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**NANYANG
TECHNOLOGICAL
UNIVERSITY**

**STUDY OF ENHANCED BOILING HEAT TRANSFER
FROM POROUS GRAPHITE FOAM STRUCTURES**

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

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FROM POROUS GRAPHITE FOAM STRUCTURES**

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SCHOOL OF MECHANICAL AND AEROSPACE ENGINEERING

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ABSTRACT

Rapid developments of new-generation electronic devices in the last few decades have provided the impetus to develop compact and effective cooling systems that are able to dissipate high heat fluxes and maintain the devices at desirably low temperatures. To address this need, much effort had been put in by engineers and researchers to develop densely-packed electronic cooling systems. Two-phase cooling is widely recognised as a promising technique to remove high heat fluxes. In addition, porous materials such as porous graphite foams of low density, high effective thermal conductivity and large total exposed surface area have been recognised as potential heat sinks or evaporators. However, the boiling process is a complex phenomenon. Combining it with the complexity of the internal structure of the graphite foams has rendered the boiling mechanism from such structures to be not well understood.

This study represents a comprehensive experimental and analytical study of boiling heat transfer from porous graphite foams. Pool and flow boiling experimental facilities were developed to study the boiling heat transfer performance, bubble characteristics, and boiling parameters with graphite foam evaporator and dielectric liquid coolant. Pool boiling experiments with different graphite foam types, foam geometry and working fluid types were conducted in the study. “Pocofoam” of 61% porosity, “Pocofoam” of 75% porosity, “Kfoam” of 78% porosity, and “Kfoam” of 72% porosity were tested as evaporator inserts. Three dielectric liquids *viz.* FC-72, HFE-7000, and HFE-7100 were selected as the working fluids. Important boiling parameters such as bubble departure diameter,

bubble departure frequency and nucleation site density were determined. Bubble growth and boiling phenomena from the porous graphite foam were analysed by a visualisation study. Boiling heat transfer enhancements of up to 3 times were obtained with porous graphite foam evaporators. It was also found that the critical heat flux (CHF) was enhanced by about 2 times to 225 W/cm^2 .

The role of internal pore structure of the graphite foam was investigated in this study by testing block and fins structures of different fin-to-block-surface-area ratios. The study shows that the block structure attained higher boiling heat transfer performance of up to 1.6 times as compared to fin structures which indicates the important contribution of the internal pore structure of the porous graphite foams to the enhancement of boiling performance. The boiling phenomena from the block and fin structures were captured and analysed through a visualisation study. It was concluded that the entire pore structure of the foam provides possible nucleation sites for generation of bubbles which contribute to the boiling heat transfer coefficient. From the experimental data, pool boiling heat transfer correlations from the porous graphite foams were proposed. The Rohsenow correlations were modified by accounting for the equivalent boiling surface area of the foam.

Beside pool boiling experiments, flow boiling heat transfer on the porous graphite form structure was also investigated. The effects of foam properties, coolant mass flux and evaporator gaps on the cooling performance were studied. Bubble flow characteristics from the porous graphite foam structure were captured and analysed. The experimental results show that the foam properties, coolant mass flux, and evaporator gaps have affected significantly the cooling performance. The

results suggested that higher coolant mass fluxes are able to sweep away the generated and flowing bubbles more effectively and increase the liquid replacement rate to the pore structures. It was also found that a smaller evaporator gap produces larger coalesced bubbles and causes bubble confinement and the formation of vapour layers on the top foam surface resulting in significant increase of the wall temperatures. Working fluid replenishment rate was reduced accordingly by reducing the evaporator gap and resulted in the decrease of the bubble departure frequency.

Semi-analytical models of bubble departure diameter and active nucleation site density for porous graphite foams were developed in the study. It was found that the proposed bubble departure diameter model is a function of pore radius, coolant properties, contact angle and bubble growth time. The bubble departure diameter from the graphite foams were measured from the captured boiling images and compared with the predicted values. Two models which predict the active nucleation site density from the foam structures were developed. The first model was constructed based on a heat and mass balance analysis, while the second model was developed based on the probability density function of cavity radius in porous graphite foams. The deviations from both models are about 27%.

It is the author's hope that the results of this present study can be used as guidelines in the design and use of porous structures in current and future high heat flux electronic device cooling systems.

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NOMENCLATURE

A	area (m ²)
Bo	Bond number
Ca	capillary number
C_p	specific heat (J/kg·K)
C_{sf}	surface-fluid constant in boiling heat transfer
d	diameter (m)
Gr	Grashof number
g	gravitational acceleration (m/s ²)
h	heat transfer coefficient (W/m ² ·K)
h_{lv}	latent heat of vaporization (J/kg)
k	thermal conductivity (W/m·K)
L	length (mm)
M	molecular weight (kg/mol)
N	active nucleation site density (sites/mm ²)
$\bar{N}$	average measured cavity density (sites/mm ²)
P	pressure (Pa)
Pr	Prandtl number
q	heat power (W)
q''	heat flux (W/cm ²)
r	radius (mm)
$\bar{r}$	mean radius (mm)
s	standard deviation
T	temperature (°C)
u	velocity (m/s)
V	volume (m ³)

Greek

α	thermal diffusivity (m^2/s)
α_{sf}	specific contact surface area (m^2/m^3)
β	thermal expansion coefficient (K^{-1})
	total internal surface-area-to-volume ratio (m^2/m^3)
δ	thickness (mm)
ε	porosity
θ	contact angle (rad)
μ	dynamic viscosity ($\text{kg}/\text{m}\cdot\text{s}$)
ρ	density (kg/m^3)
$\Delta\rho$	density difference ($\rho_v - \rho_l$) (kg/m^3)
σ	surface tension (N/m)
ν	kinematic viscosity (m^2/s)

Subscripts

b	boiling parameter
eff	effective
f	fluid
lv	vaporization
g	gas
h	heater
L	losses
l	liquid
min	minimum
max	maximum
nuc	nucleation

<i>p</i>	foam pore
<i>s</i>	solid
<i>sat</i>	saturation
<i>t</i>	total
<i>v</i>	vapour
<i>W</i>	wall
<i>w</i>	width

PUBLICATIONS

A. Publications Arising from this Thesis

Journal Papers

1. Leong, K.C., Jin, L.W., **Pranoto, I.**, Li, H.Y., and Chai, J.C., "Experimental study of enhanced pool boiling heat transfer using graphite foam inserts", *Defect and Diffusion Forum*, Vols. 312-315, pp. 352-357, 2011.
2. Jin, L.W., Leong, K.C., **Pranoto, I.**, Li, H.Y., and Chai, J.C., "Experimental study of a two-phase thermosyphon with porous graphite foam insert", *ASME Journal of Thermal Science and Engineering Applications*, Vol. 3, No. 2 pp. 024502-1 – 024502-6, 2011.
3. Jin, L.W., Leong, K.C., and **Pranoto, I.**, "Saturated pool boiling heat transfer from highly conductive graphite foams", *Applied Thermal Engineering*, Vol. 31, Nos. 14-15, pp. 2685-2693, 2011.
4. **Pranoto, I.**, Leong, K.C., and Jin, L.W., "The role of graphite foam pore structure on saturated pool boiling enhancement", *Applied Thermal Engineering*, Vol. 42, pp. 163-172, 2012.
5. **Pranoto, I.** and Leong, K.C., "An experimental study of flow boiling heat transfer from porous foam structure in a channel", *Applied Thermal Engineering*, Vol. 70, No. 1, pp. 100-114, 2014.

Conference Papers

1. Leong, K.C., Jin, L.W., **Pranoto, I.**, Li, H.Y., and Chai, J.C., "Experimental study of enhanced pool boiling heat transfer using graphite foam inserts", *Proceedings of the 6th International Conference on Diffusion in Solids and Liquids, DSL 2010*, July 5-7, 2010, Paris, France.

2. Leong, K.C., Jin, L.W., **Pranoto, I.**, Li, H.Y., and Chai, J.C., "Experimental study of a two-phase thermosyphon with porous graphite foam insert", *Proceedings of the 14th International Heat Transfer Conference, IHTC 14*, August 8-13, 2010, Washington, D.C., USA.
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5. **Pranoto, I.** and Leong K.C., "Experimental study of enhanced flow boiling heat transfer and bubble dynamics from porous graphite foam structures", *Proceedings of the 8th International Symposium on Heat Transfer ISHT-8*, October 21-24, 2012, Beijing, China.

B. Publications Related to this Thesis

1. **Pranoto, I.**, Yang, C., Zheng, L.X., Leong, K.C., and Chan, P.K., "Flow boiling heat transfer enhancement from carbon nanotube-enhanced surfaces", *Defect and Diffusion Forum*, Vol. 348, pp. 20-26, 2014.
2. **Pranoto, I.**, Yang, C., Zheng, L.X., Leong, K.C., and Chan, P.K., "Flow boiling heat transfer enhancement from carbon nanotube-enhanced surfaces", *Proceedings of the 9th International Conference on Diffusion in Solids and Liquids, DSL 2013*, 24 - 28 June 2013, Madrid, Spain.
3. Ho, J.Y., Leong, K.C., Yang, C. and **Pranoto, I.**, "An experimental study of carbon nanotube coatings for pool boiling heat transfer enhancement", *Proceedings of the 15th International Heat Transfer Conference, IHTC-15*, August 10-15, 2014, Kyoto, Japan.

CHAPTER 1 INTRODUCTION

1.1 BACKGROUND

1.1.1 Challenges in Recent Developments in Electronic Cooling

Rapid developments in the design of new-generation electronics devices have led to significant increase in device performance and very high power densities. The decrease of size versus increase of processor speed has caused the decrease of heat transfer surface area which leads to very high heat fluxes. It is known that today's high performance computer chips generate high power densities of up to 100 W/cm^2 as reported by IBM in their website (IBM, 2006). The power density of high performance electronic devices is projected to reach 200 W/cm^2 in the years 2014 – 2020 (ITRS, 2007). Furthermore, it is shown that the current research on interlayer cooling system in IBM is targeted to dissipate heat fluxes of up to 250 W/cm^2 (IBM, 2014). With regard to advanced high heat flux cooling systems, Mudawar (2001) categorised future high-heat-flux thermal management systems into the *high-flux* range of $10^2 - 10^3 \text{ W/cm}^2$ and the *ultra-high-flux* range of $10^3 - 10^5 \text{ W/cm}^2$.

The high levels of current and projected power densities have led to the significant increase of surface temperature of the devices and will affect device reliability. Without a good cooling system, future computer chips will reach very high surface temperatures that are close to 6000°C (IBM, 2006). Bar-Cohen *et al.* (1983) stated that for every 2°C temperature rise, the reliability of a

silicon chip decreases by about 10%. Hence, thermal management becomes extremely important to address the problem of dissipating the high heat fluxes from the devices. This high demand on high heat flux electronic devices has challenged researchers to develop novel cooling systems that will provide the required cooling performance, reliability and cost. An effective cooling system must be able to remove the high heat flux and at the same time maintain the devices at desirably low temperatures which are typically below 85°C. Current and future developments of high power density electronics devices for supercomputers, electric vehicles, advanced military vehicles and avionics have constantly face the challenge to develop advanced high heat flux cooling systems.

1.1.2 Development of Advanced Cooling Systems

In order to develop an advanced cooling system to dissipate high heat flux from the devices, many cooling methods were proposed by various researchers. Different cooling techniques based on the different categories such as passive-active cooling, direct-indirect cooling and single phase-two phase cooling methods have been studied to engineer advanced cooling systems. Techniques such as air cooling, liquid cooling, mini and micro-channels, jet impingement, spray cooling, thermosyphons, heat pipes, and other exotic designs have been explored to dissipate the high heat fluxes from these devices. Cooling technologies for electronic devices have shifted from natural convection to single-phase forced convection and then to phase-change cooling systems. The heat transfer performance limitations of natural and single phase forced

convection have driven the development of two-phase cooling or boiling heat transfer. The advantage of high heat transfer coefficients of two phase systems over natural and single-phase forced convection is shown clearly in Fig. 1-1. Based on this fact, two-phase cooling is generally considered to be the more promising technique for high heat flux electronic devices in the future.

Fig. 1-1 Heat transfer coefficients for natural, single-phase forced convection and boiling for different coolants (Mudawar, 2001).

As mentioned in the previous section, future electronic devices would generate very high power densities with smaller package sizes. Therefore, effective heat dissipation materials are those which possess high thermal conductivity, large surface area and low weight. Porous materials that provide large fluid-solid contact surface and possess high thermal conductivity have been applied in high heat flux thermal management systems. Many researchers have conducted extensive studies on heat transfer in porous materials (Koh and Stevens, 1975; Kim *et al.*, 1994; Rainey *et al.*, 2001, 2003; Jiang *et al.*, 2004; Li and Peterson, 2007, 2010; El-Genk and Ali, 2010; Forest *et al.*, 2010; Furberg and Palm,

2011; Narasimhan *et al.*, 2012; Xu *et al.*, 2015; Joshi and Dede, 2015). However, most of these studies have focused on the packed bed as the porous medium. As compared to the porous channel packed with metal particles, spheres or woven-screens and microcellular foams, materials such as metals or graphite foams belong to a class of novel porous media which has an open structure. Figure 1-2 shows the typical internal structures of aluminium and graphite foams observed under a Scanning Electron Microscope (SEM). It can be seen from Fig. 1-2(a) that the aluminium foam has a reticulated structure of open, shaped cells connected by continuous solid metal ligaments. Figure 1-2(b) shows the internal structure of high-conductivity graphite foam developed at Oak Ridge National Laboratory (ORNL), USA. The foam has an open cell structure with highly aligned graphitic ligaments. It possesses high thermal conductivity with predominantly spherical pores with smaller openings between the cells.

Fig. 1-2 Typical internal structures of (a) Aluminium and (b) Graphite foams.

Open cell foams with fully inter-connected structure, high surface area to volume ratio and high thermal conductivity lend itself to many applications, especially in heat transfer enhancement. Metallic porous foams have been successfully used as heat exchangers for airborne equipment and as compact heat sinks for electronics cooling (Fu *et al.*, 2001; Kim *et al.*, 2001; Bhattacharya and Mahajan, 2002; Boomsma *et al.*, 2003; Leong and Jin, 2005, 2006; Xu *et al.*, 2008; Zhao *et al.*, 2009; Yang *et al.*, 2010; Li and Leong, 2011; Souza and Barbosa, 2012; and Qu *et al.*, 2012; Chen *et al.*, 2013; Ventola *et al.*, 2014; Chumpia and Hooman, 2015; Xu and Zhao, 2015; Xu *et al.*, 2015). Attracted by the advantages of the thermal properties of graphite foam, several studies have been performed to investigate its use in thermal management applications (Klett, 1999; Ott *et al.*, 2002; Gallego and Klett, 2003; Parker and El-Genk, 2005; Williams and Roux, 2006; Leong *et al.*, 2010; Jin *et al.*, 2011; Pranoto *et al.*, 2012; Lin *et al.*, 2013; Pranoto and Leong, 2014; Solmus, 2015). The results from these studies show that better performance can be achieved by using open cell porous foams as heat sinks. It is envisaged that the use of highly conductive graphite foams in two phase cooling systems will add to more options for cooling electronic components in the near future.

1.2 OBJECTIVES AND SCOPE

The objectives of this study are to achieve fundamental understanding of the boiling heat transfer mechanism from porous graphite foam structures and to investigate the boiling heat transfer performance from graphite foams. Pool and flow boiling heat transfer characteristics and phenomena on these foams are

investigated experimentally and theoretically. In order to achieve the objectives, the following scope of work has been performed:

- (1) Comprehensive experimental studies were conducted to investigate the boiling heat transfer from porous graphite foams in a thermosyphon in the case of pool boiling and pumped two-phase cooling facility for flow boiling.
- (2) Parametric studies on the effects of graphite foam properties, graphite foam geometry and structures, coolant properties and coolant volumes were conducted for the pool boiling system. For the flow boiling system, the effects of graphite foam properties, foam height, evaporator channel gap, coolant mass flux, and coolant type were investigated. The role of the graphite foam pore structure was investigated extensively to understand the boiling heat transfer mechanism.
- (3) Important boiling parameters such as *Critical Heat Flux* (CHF), boiling heat transfer coefficient, bubble departure diameter, frequency, and active nucleation site density were determined. The boiling phenomena and bubble growth-departure mechanism were extensively studied by using a high-speed camera and image analysis for visualisation purposes.
- (4) Theoretical analyses were performed on the bubble dynamics and boiling mechanisms. The bubble growth mechanism from the porous graphite foam were explored and modelled. Semi-analytical models of

the bubble departure diameter and active nucleation sites density are proposed in this thesis.

1.3 OUTLINE OF THESIS

The preceding section of this chapter introduces the background of this study. The objective and scope of the research are also presented. A comprehensive literature review of high heat flux cooling systems, porous structures, porous foam heat sinks and evaporators and boiling heat transfer performance and parameters are presented in Chapter 2. The experimental setup and procedures for this study are described in Chapter 3 complete with the details of the pool and flow boiling experimental facilities and the measurement systems. The experimental procedures, sensor calibration and uncertainty analyses are also discussed. Finally, graphite foam and working fluids properties are presented in this chapter.

Pool boiling characteristics from porous graphite foam structures are discussed in Chapter 4. Parametric studies on the effects of graphite foam properties, working fluid types, and working fluid levels are elaborated in detail. Important boiling characteristics such as the active nucleation site density, bubble departure frequency, and critical heat flux are presented. Pool boiling heat transfer and characteristics from the porous graphite foam structures are further discussed in Chapter 5. In this chapter, the roles of the internal pore structure of the foam to the boiling enhancement are studied experimentally. Block and fin graphite foam structures with different exposed surface area ratios were tested and analysed in this chapter. The heat transfer rate prediction

models for different graphite foam types are presented in this chapter. Chapter 6 discusses the flow boiling characteristics from porous graphite foam structures. Similar to the parametric study presented in Chapter 4, the effects of coolant mass fluxes, graphite foam properties, coolant properties, and evaporator gap are discussed here. In this chapter, the enhanced boiling mechanism and characteristics are discussed in terms of the forced convection and flow boiling heat transfer, bubble departure diameter, bubble departure frequency and critical heat flux from porous graphite foam structures.

Boiling heat transfer models from porous graphite foam structures are discussed in Chapter 7. The bubble growth and mechanism theory and the semi-analytical models of bubble departure diameter and active nucleation site density are presented and compared with experimental data. Chapter 8 ends the report with conclusions derived from this research and recommendations for future research work.

CHAPTER 2 LITERATURE REVIEW

2.1 HIGH HEAT FLUX COOLING SYSTEMS IN ELECTRONIC PACKAGES

As mentioned in the previous chapter, many methods have been developed to remove the high heat fluxes from the electronic devices. These can be categorised into active and passive techniques, direct and indirect methods, and single-phase and two-phase systems. In the active technique, an external activator or power supply is used to enhance the cooling performance. Passive cooling, on the other hand, does not require any external activator or power supply. Direct and indirect methods are categorised based on whether the coolant agents are in direct contact with the hot spots or not. The direct method has the advantage of minimising the contact resistance between the hot spots and the cooling medium such that the cooling performance will be enhanced. As described in the previous chapter, two-phase cooling systems have become an effective method to remove high heat fluxes in current and future advanced electronic cooling systems. As elaborated by Kandlikar *et al.* (2009), heat transfer enhancement in two-phase cooling systems is achieved through the following methods:

- Generation of repeatable boiling cavities for increasing the nucleation site density and the associated latent heat flux while decreasing the wall superheat.

- Increase in the total area of the thin liquid film under the boiling bubbles and inside the cavities for enhancing the latent heat flux.
- Increase in the forced convection heat transfer between the boiling surface and heat transfer liquid for increasing the accumulated sensible superheat in liquid.

Several reviews of high heat flux cooling systems for electronic devices are presented in this section. Such devices have been used in many applications such as computers, telecommunications equipment, portable electronic devices, automobiles, military and aerospace. Yeh (1995) reviewed the thermal control and management of electronic equipment. He showed that thermal control had become a critical factor in the design of electronic equipment because of the increased miniaturisation of components and device heat dissipation. It was mentioned that the great demands placed on system performance and reliability intensify the need for better thermal management. It was also mentioned that based on the survey by the U.S. Air Force, more than fifty percent of all electronic failures are caused by undesirable temperature control. A review of air-cooled electronic packages was carried out by Sathe and Sammakia (1998). They reviewed developments in heat sink designs and applications intended for high-end high-power dissipation systems. A literature survey on the effects of card in the thermal enhancement of electronic packages was also presented. They had stated that in certain applications, the card heat-sinking effect can play a major role in the thermal management of a package, accounting for more than 50 percent of the total power dissipation of the package. They had also

mentioned that better manufacturing tolerances, lower defects, and an improved understanding of card and enclosure effects have been attained and shown to be critical to achieving the desired thermal performance.

An updated review on high heat flux thermal management was presented by Mudawar (2000). His assessment was focused mainly on the application of thermal management techniques using phase change cooling such as pool boiling/thermosyphon, channel flow boiling, micro and mini channel cooling, jet impingement cooling and also thermal management in high heat flux avionic systems. Another review on thermal management for high power electronics devices was presented by McGlen *et al.* (2004). They proposed to combine two-phase cooling with liquid cooling for high flux electronic devices to achieve higher heat transfer performance.

Two-phase cooling systems in electronic devices have been investigated and developed by many researchers. Most of the techniques were based on heat pipes and thermosyphons. These techniques rely on evaporation and condensation heat transfer. Palm and Khodabandeh (2003) stated that the principal difference between the two types is that, in heat pipes, the liquid is returned from the condenser to the evaporator by means of surface tension forces acting in a wick whereas in a thermosyphon, the liquid is returned by gravity force. Chu *et al.* (1999) investigated a cooling technique which used a thermosyphon heat loop and water as the working fluid. They observed the effects of working fluid level, heat load and condenser air flow rate on heat transfer performance. Their results showed that the thermosyphon provided a

viable alternative to either direct air cooling or forced water-air hybrid cooling. They also stated that the level of coolant had a significant effect on heat transfer performance. A fluid volume of 400 cm^3 gave the best performance for the configuration tested. Below this level, the flow was unstable with pulsating behaviour and periodically uncovered the evaporator surface. They recommended surface treatment to enhance the boiling heat transfer coefficient.

The use of a compact thermosyphon for cooling Pentium 4 microprocessor in a HP Vectra PC with 80 W power dissipation was investigated by Pal *et al.* (2001). The effects of working fluid and system inclination on the thermal performance were studied. Numerical simulation of condenser efficiency in natural and forced convection was also performed. Copper fins were used in the evaporator and aluminium fins for the condenser. Methanol (PF5060) and water (DIUF) were used as the working fluids. They concluded that water showed better heat transfer performance than methanol and a large inclination angle had no significant effect on heat transfer performance. Jiang *et al.* (2001) designed and characterised a two-phase microchannel heat sink for an electro-kinetic VLSI chip. Experiments were conducted in open loop operation with water as the working fluid. Their results showed that a thermal resistance of 1°C/W was achieved for a $1.2 \times 1.2 \text{ cm}$ silicon chip.

Palm and Khodabandeh (2003) investigated the effects of working fluid in the performance of a two-phase thermosyphon. They concluded that high pressure fluids generally showed better heat transfer performance. They stated that there is no ideal working fluid for two phase thermosyphon systems. It depends on

the equipment under consideration for the local and global environments. An experimental and theoretical study of two-phase closed thermosyphon (TPCT) behaviour in transient regimes was conducted by Farsi *et al.* (2003). Their experimental results showed the existence of two kinds of TPCT responses. The first category was concerned with the regular and monotonous operating system variables, and the second category was concerned with the response presenting temperature excursions. Their mathematical model showed good agreement between experiments and numerical results.

Gima *et al.* (2005) conducted an experimental study on high-power CPU of notebook computers using a close-loop two-phase thermosyphon with FC-72 as the working fluid. The effects of heat input, working fluid volume and types of evaporator surface were investigated. Four types of copper evaporator surfaces were used in the experiments. Their results showed that working fluid volume had no significant effect on the CPU temperature, but the evaporator surface had significant effect on the CPU temperature. Khodabandeh (2005) investigated the effect of heat flux, system pressure, mass flow rate, vapour fraction, diameter of evaporator channel and tubing distance between evaporator and condenser on the performance of a two-phase thermosyphon loop. A block copper evaporator with 7, 5, 4, 3 and 2 vertical channels of diameters 1.1, 1.5, 1.9, 2.5 and 3.5 mm, respectively and a length of 14.6 mm were used. Isobutane was used as the working fluid and tests were conducted at heat fluxes between 28.3 and 311.5 kW/m². Khodabandeh concluded that the vapour fraction had no significant effect on heat transfer performance which depended significantly on heat flux and system pressure. He also indicated that

nucleate boiling was the dominant mechanism and reported that the evaporator diameter had no effect on heat transfer performance. A further investigation of two phase cooling systems was undertaken by Campbell *et al.* (2007). R-134a was used as the working fluid for cooling the Power Electronics (PE) for hybrid-electric vehicles. The PE was submerged in a refrigerant bath. Their results showed no damaging effect on normal operation for 300 days on a submerged IGBT and gate controller card. Two phase cooling system with copper foam heat sink was studied by Ji *et al.* (2012). The experimental results have shown that the water and ethanol coolants have performed the best and worst thermal performance, respectively, among the tested three working fluids of water, acetone and ethanol. It was also mentioned that there were a combined effect of charge ratio and inclined angle to the cooling performance. They have also concluded that the maximum heating power of 170 W and minimum thermal resistance of 0.09 K/W were achieved without the capillary or boiling limits reached.

It is noted from the above review that the porous media enhance the cooling performance and increase the CHF. However, there are still many aspects and parameters pertaining to the mechanism and characteristics of boiling on porous media evaporator that need to be explored, especially in highly conductive graphite foams. To the best of the author's knowledge, the literature in this area is still rather scarce. Research on the two phase cooling system for high heat flux electronic devices by using the porous graphite foam evaporator are further discussed in Section 2.2 of this chapter.

2.2 POROUS STRUCTURE AND POROUS FOAM HEAT SINK AND EVAPORATOR

Research on two-phase cooling systems for high heat fluxes were presented by Mudawar (2000). He conducted an assessment on the development of two-phase cooling which included pool boiling or thermosyphon, channel boiling, micro and mini channel cooling, and phase change cooling for avionic devices. He concluded that all two-phase cooling systems were capable of dissipating heat fluxes above 100 W/cm^2 by using dielectric coolants. He also stated that two-phase cooling systems are generally more complicated to implement than single-phase liquid cooling systems. The primary concern is the difficulty of predicting CHF in order to ascertain a comfortable and safety factor. However, most research efforts on two-phase cooling systems were based mainly on applications of enhanced evaporator and condenser made of copper or aluminium. On the other hand, capillary pumped loop cooling systems mostly used silicon microchannels as the evaporator and condenser. The novelty of graphite foams with high thermal conductivity, low density, and very large surface area to volume ratio has opened up the possibility of developing two-phase cooling systems with higher heat transfer performance. The open cells of the graphite foam with fully inter-connected structures improve heat transfer by the increased surface area in contact with the working fluid.

The first development of carbon foams by researchers in the late 1960's is the reticulated vitreous (glassy) carbon foam (Ford, 1964; Googin *et al.*, 1967). This was followed by research on novel graphite foams to exploit the better

thermophysical properties for thermal management. The first graphitic foams with bulk thermal conductivities greater than 58 W/m·K was reported by Klett *et al.* (1998). In the following year, Klett (1999) reported a study on graphite foams with thermal conductivities up to 180 W/m·K. The graphite foam heat exchanger (radiator) for a passenger automobile which had a significantly smaller size was developed and investigated by Klett (1998). His results showed that the graphite foam radiator performed two times better than the traditional radiator. The graphite foam has a bulk thermal conductivity of 150 W/m·K and specific conductivity of up to 6 times greater than copper. He stated that the unique thermal properties combined with the continuous graphitic open-celled network of such foams (unlike carbon fiber reinforced composites), should lead to novel and interesting methods of thermal management. Klett *et al.* (2000) reported on the manufacturing process and properties of high thermal conductivity graphite foam. The effect of pitch precursor characteristics on the foam structure and properties were also investigated. Their results showed that thermal conductivity was relatively unaffected by precursor characteristics.

Development of high thermal conductivity graphite foam was continued at Oak Ridge National Laboratory (ORNL), USA by Gallego and Klett (2003). They proposed a new fabrication process without blowing and stabilisation processes. The process had a better effect on manufacturing time and cost. They showed that the overall heat transfer coefficients of carbon foam heat sinks were up to two orders of magnitude greater than those of conventional heat sinks. Klett and Trammel (2004) described a unique evaporative cooling technique where

the chip or actual electronic device could be bonded directly to the graphite foam and then immersed in the cooling fluid. Their results showed that the thermal performance of this system was 30% better than the standard design. A study on the application of high thermal conductivity carbon foam was conducted by White *et al.* (2004). They investigated the use of graphite foams in automotive power electronics under forced air convection, water cooled heat exchangers, and evaporative cooling.

Klett *et al.* (2004) studied the graphitic structure of the graphite foam and attempted to correlate the morphology of the ligaments with bulk thermal properties of up to 182 W/m·K. In the same year, Klett and Trammell (2004) reported a parametric investigation of a high thermal conductivity graphite foam evaporator in a thermosyphon cooling system for a CMOS chip. The effects of evaporator density, evaporator geometry, types of working fluid and volume level of working fluid were investigated. FC-87 and FC-72 were used as the working fluids. Their results showed that the thermal performance of the system was affected by foam density. Foams with a lower density performed significantly better than higher density foam, although they stated that the mechanism of improvement was still not fully understood. Their results also showed that the evaporator structure had significant effect on the heat transfer performance. They also found that working fluid type and volume had no significant effect on heat transfer performance and stated that the critical heat flux (CHF) was not reached in their investigation.

2.2.1 Pool Boiling Cooling System from Porous Foam Structures

Research on the use of two-phase thermosyphons for electronics cooling using graphite foam as evaporator was conducted by Coursey *et al.* (2005). Fluorinerts FC-72 and FC-87 were used as working fluids in their experiments. Their results showed that working fluid type had no effect on heat transfer performance. They stated that liquid level, condenser temperature, geometry and density of graphite foam had significant effect on heat transfer performance. An investigation of the pool boiling heat transfer enhancement of graphite foams with FC-72 coolant was conducted by Parker and El-Genk (2005). They had also compared the cooling performance from the graphite foam evaporators with a smooth copper surface. They found that the nucleate boiling heat transfer coefficients obtained from graphite foams were significantly higher than those of copper. In addition, they found that the overall critical heat flux (CHF) values were higher than those from a copper evaporator.

Lu *et al.* (2006) developed a vapour chamber using graphite foam for chip heat spreading. They investigated the effect of water and ethanol on heat transfer performance. Their results showed that water performed twice as well as ethanol. They also stated that the performance of a vapour chamber using graphite foam was about two times better than the copper wick structure.

Xu *et al.* (2008) investigated pool boiling heat transfer and bubble dynamics from the copper foams of 30 – 90 pores per inch evaporators with heating surface area of 12×12 mm by using acetone as the working fluid. They

obtained boiling curves of copper foams which show three distinct regions. Region I and II are those of natural convection heat transfer, and nucleate boiling heat transfer for all the foam covers. Region III is that of either a resistance to vapour release for the 30 and 60 PPI foam covers, or a capillary-assist liquid flow towards foam cells for the 90 PPI foam covers. It was found that the value of PPI has an important effect on the thermal performance. Boiling curves are crossed between the high and low PPI foam covers. Low PPI foams had better thermal performance at low surface superheats, but high PPI foams performed better at moderate or large surface superheats and extended the operation range of surface superheats. The effects of other factors such as pool liquid temperature, foam cover thickness on the thermal performance are also discussed.

Gandikota and Fleischer (2009) investigated the thermal performance of graphite foam for evaporator enhancement in pool boiling of a thermosyphon system. The influences of heat load and fluid level on the overall system performance were examined. Their results showed that the thermal resistance of the foam heat sink was about 0.024 K/W. They also reported that the thermal resistance increased with increasing heat flux, but still remained advantageously low and exhibited excellent potential for high heat flux dissipation for thermal management of advanced electronics.

Jin *et al.* (2011) investigated the pool boiling enhancement of porous graphite foam. Their experimental results showed that the boiling thermal resistances of the system with a graphite foam evaporator were about 2 and 3 times lower

than those of tested copper block with FC-72 and HFE-7000, respectively. A pool boiling study from the porous graphite foam structures with FC-72 and HFE-7100 dielectric liquids was performed by El-Genk (2012). Their experimental results showed that the natural convection data obtained with the graphite foam performed higher heat removal rates up to 12.4% than on plane surfaces. Their results also showed that the values of nucleate boiling heat transfer coefficient and CHF performed by the porous graphite foam were much higher than those reported by others on plane, micro-porous and structured surfaces and on surfaces with micro-porous coatings. Furthermore, the role of the internal pores in the graphite foam on the boiling heat transfer enhancement was studied by Pranoto *et al.* (2012). Boiling heat transfer from different graphite foam inserts structures were investigated in saturated pool boiling condition. They concluded that the internal pores of a graphite foam contributed significantly to a larger active nucleation site density and enhanced boiling heat transfer performance.

2.2.2 Flow Boiling Heat Transfer from Porous Foam Structures

Many researchers have studied flow boiling for various cooling applications. Rainey *et al.* (2001) studied flow boiling from microporous coated surfaces in sub-cooled FC-72. Their experimental results showed that microporous surfaces outperformed plain surfaces and enhanced the Critical Heat Flux (CHF). Muwanga and Hassan (2007) studied flow boiling heat transfer of FC-72 in a microtube by liquid crystal thermography. They observed the effect of mass and heat fluxes on the heat transfer coefficient. Their results showed that

the heat transfer coefficient was influenced slightly by heat flux at a lower mass flux ($770 \text{ kg/m}^2\cdot\text{s}$) and was less dependent on heat flux at a higher mass flux ($1040 \text{ kg/m}^2\cdot\text{s}$). They also observed that the wall temperature oscillated over a wide range under two-phase conditions with amplitude of 1 to 7°C for low heat flux and frequency of about 10 Hz.

Wang *et al.* (2008) experimentally investigated micro capillary pumped loop (CPL) cooling for high power devices with water as the working fluid. They found that the groove-enhanced surface design improved the thermal performance of the cooling system. They suggested that the micro CPL system must be used at low pressures and that automatic circulation had to be maintained for preheating to ensure the equilibrium of transmission without an additional power supply. In addition, the system started up successfully and depriming did not occur until the heat flux had reached 185.2 W/cm^2 . A MEMS-based integrated capillary pumped loop (CPL) cooling module was developed by Jung *et al.* (2008). The module consisted of an evaporator and condenser with cone-shaped capillary structure insertions. Their experimental results showed that the module possessed good transient characteristics. They stated that the CPL can handle a heat flux of 6.22 W/cm^2 and an allowable evaporator surface temperature of up to 110°C with FC-72 as the coolant. Further investigations on pumped two-phase cooling were carried out by Agostini *et al.* (2008). They developed a pumped two-phase cooling system with a silicon multi-microchannel heat sink using refrigerant R236fa as the working fluid. They stated that the system had successfully removed heat

fluxes up to 255 W/cm^2 and maintained the base chip temperature below 52°C . They also reported that the flow boiling heat transfer coefficient increased with heat flux, decreased with mass velocity, and decreased slightly with vapour quality. Kim *et al.* (2008) studied the flow boiling characteristics of three porous copper foams *viz.* 95% porosity and 10 PPI, 95% porosity and 20 PPI, and 92% porosity and 20 PPI, which were soldered to a heated wall of a 10 mm wide $\times$ 37 mm long $\times$ 7 mm high channel. Their experimental results showed that the high porosity and large pore size foam, *i.e.* the 95%, 10 PPI copper foam, gave the best result, achieving a heat transfer coefficient of $10 \text{ kW/m}^2\cdot\text{K}$. They stated that the results for water were in good agreement with the available sparse porous matrix correlations, using the recommended dispersion conductivity coefficient of 0.06. Finally, the results of their study had also indicated that the porous foam had more influence in enhancing the convective heat transfer coefficient for FC-72 than water despite its relatively low dispersion coefficient.

Many research investigations on the application of porous material in two-phase flow cooling systems have been conducted over the years. Zhao *et al.* (2009) investigated flow boiling heat transfer in horizontal metal-foam filled tubes with R134a as the working fluid. Their experimental results showed that the heat transfer coefficient was almost doubled by reducing the cell size from 20 to 40 pores per inch (PPI) for a given porosity. They also stated that the boiling heat transfer coefficient kept increasing steadily, albeit slowly, with increasing vapour quality for high mass fluxes, while the same trend was not observed for low mass fluxes. They obtained heat transfer coefficients of

copper foam tubes which were approximately three times higher than that of plain tubes. An experimental study on the flow boiling heat transfer performance of FC-72 on silicon chips was conducted by Ma *et al.* (2009). They fabricated micro-pin-fins on the chip surface by using a dry etching technique to enhance the boiling heat transfer. Different fluid velocities, liquid subcooling and chip configuration were investigated in their study. They found that all the micro-pin-finned surfaces showed considerable heat transfer enhancement and increased CHF compared to the smooth surface. They also stated that the CHF values for all surfaces increased with fluid velocity and subcooling. In their study, the maximum CHF value of nearly 150 W/cm² was reached for chip PF30-120 at the fluid velocity of 2 m/s and liquid subcooling of 35 K.

An experimental study of flow boiling heat transfer inside a channel filled with metallic foam was performed by Madani *et al.* (2013). The metallic copper foam of 36 PPI and 97% porosity and the n-pentane working fluids were used in this study. The coolant mass flux and the heating power were varied from 10 ~ 100 kg/m²s and 0 ~ 25 W/cm², respectively. Their experimental results showed that the heat transfer coefficient was enhanced by a factor from 2 to 4 as compared to the plain tube heat transfer coefficient calculated with *Gungor-Winterton* correlation.

In summary, flow boiling systems provide the advantages of enhanced cooling performance and orientation flexibility compared to pool boiling systems. However, there are still many aspects pertaining to the characteristics of flow

boiling on porous media evaporator that need to be explored, especially in highly conductive graphite foams. To the best of the authors' knowledge, the literature in this area is still rather scarce. In this study, a pumped flow boiling cooling systems was developed by using porous graphite foams as the evaporator insert in the channels. The effects of the coolant mass flux, evaporator gap, and thermophysical properties of the graphite foam on the bubble characteristics and performance were investigated.

2.3 BOILING HEAT TRANSFER PERFORMANCE AND PARAMETERS

2.3.1 Nucleation Site Density

In pool boiling systems, as the heat flux at the surface increases, more and more nucleation sites will be activated. It is important to determine how many sites are active per unit area, which is usually expressed as the nucleation site density. Many studies have been performed to predict the nucleation site density in boiling heat transfer. Kocamustafaogullari and Ishi (1983) have assumed that the active nucleation site density in pool boiling was influenced by both thermophysical properties of the fluid and the surface condition. They developed the following correlation based on the water data:

$$N_{np}^* = f(\rho^*) R_c^{*-4.4} \quad (2-1)$$

$$\text{where } N_{np}^* = N_{np} D_B^2$$

$$f(\rho^*) = 2.157 \times 10^{-7} \rho^{*-3.2} (1 + 0.00409)^{4.13} \quad (2-2)$$

$$\rho^* \equiv (\Delta\rho / \rho_g) \text{ and } R_c^* \equiv \left(\frac{R_c}{(D_B / 2)} \right)$$

They determined the bubble departure diameter (D_B) by adapting and modifying the bubble departure diameter (D_{BF}) correlation proposed by Fritz (1935) as

$$D_B = 0.0012 \rho^{*0.9} D_{BF} \quad (2-3)$$

$$\text{where } D_{BF} = 0.0148 \theta \sqrt{\frac{2\sigma}{g\Delta\rho}} \quad (2-4)$$

Yang and Kim (1988) proposed the first attempt to predict quantitatively the active nucleation site density from the knowledge of the radius and cone angle distribution of cavities that were actually measured on the surface. They measured the probability density function of the cavity radius by using a scanning electron microscope (SEM) and a differential interference contrast microscope. They proposed that the cavity radius and cone angle distributions can be fitted by the Poisson and normal distributions, respectively and expressed the active nucleation site density in pool boiling as

$$N = \bar{N} \int_0^{\theta/2} \frac{1}{\sqrt{2\pi s^2}} \exp \frac{(\beta - \bar{\beta})^2}{2s^2} d\beta \times \int_{r_{\min}}^{r_{\max}} \lambda e^{-\lambda r} dr \quad (2-5)$$

The nucleation site density model proposed by Yang and Kim (1988) will be discussed in detail in Chapter 7.

An empirical correlation of active nucleation site density including the contact angle effect during pool boiling of water on vertical surface was proposed by Wang and Dhir (1993). They conducted pool boiling experiments in saturated water at atmospheric pressure using mirror-finished copper surfaces in the experiments by well-defined procedures. They changed the wettability of the surface by controlling the surface oxidation degree and expressed the active nucleation site density model (SI units) as

$$N_{np} = 7.81 \times 10^{-29} (1 - \cos \theta) D_c^{-6.0} \quad (2-6)$$

They stated that the model could reproduce almost all of their data on the sites of the surfaces which have contact angles between 90° and 18° to within $\pm 60\%$. Benjamin and Balakrishnan (1997) conducted an experimental investigation on the nucleation site density during nucleate pool boiling of saturated pure liquid at low to moderate heat fluxes. They investigated the surface-liquid interaction during the boiling phenomena and its effect on the nucleation site density. Stainless steel and aluminium boiling surfaces of different surface finishes were investigated. The liquids used in their experiments were distilled water, carbon tetrachloride, n-hexane, and acetone. They stated that the nucleation site density depended on the surface roughness, liquid surface tension, thermophysical properties of the heating surface-liquid and the wall superheat and expressed the nucleate site density as

$$N_{np} = 218.8 \text{Pr}^{1.63} \left(\frac{1}{\gamma} \right) \Theta^{-0.4} \Delta T_w^3 \quad (2-7)$$

$$\text{where } \gamma \equiv \left(\frac{\kappa_w \rho_w C_{pw}}{\kappa_f \rho_f C_{pf}} \right) \text{ and } \Theta = 14.5 - 4.5 \left(\frac{R_a P}{\sigma} \right) + \left(\frac{R_a P}{\sigma} \right)^{0.4}$$

Hibiki and Ishi (2003) proposed a model of active nucleation site density based on Yang and Kim (1988)'s approach. They stated that although the basic concept of Yang and Kim's model might be sound, the approximated functions of cavity radius and half cone angle might not be physically appropriate. They approximated the distribution of the cavity number as a function of the cavity radius as

$$f'(r) = \frac{\lambda'}{r^2} \exp\left(\frac{\lambda'}{r}\right) \quad (2-8)$$

where the cavity number distribution is expressed as

$$f'(\beta) = \frac{\beta}{\mu^2} \exp\{-\beta^2/(2\mu^2)\} \quad (2-9)$$

and the active nucleation site density is given by

$$N = \bar{N} \int_0^{\theta/2} f'(\beta) d\beta \times \int_{R_c}^{r_{\max}} f'(r) dr \quad (2-10)$$

They also stated that the active nucleation site density is a function of the wall superheat with a different exponent in the wall superheat from the existing models. Their model also supported Yang and Kim's proposal that the active nucleation site density can be developed by knowing the size and cone angle distributions of cavities that are actually present on the surface.

2.3.2 Bubble Departure Diameter and Frequency

Bubble departure diameter (D_B) and frequency (f_B) are very important parameters in bubble dynamics that directly affect boiling heat transfer performance. In bubble departure models, the departure bubble shape is assumed to be spherical to simplify the analysis. Surface heating, liquid heating, nucleation, bubble growth and departure occur continuously and repeatedly during the boiling process such that normally the cycle is referred to as an ebullition cycle. During the boiling process, a bubble is generated from an activated cavity on the boiling surface. The generated bubble grows in a spherical shape during the bubble growth time and then departs from the nucleation cavity. The bubble diameter at the time of departure from the nucleation site is called the bubble departure diameter. Based on the literature survey, the bubble departure diameter is affected mainly by the force balance on the nucleation cavity of the boiling surface.

The bubble departure frequency indicates how fast the bubble grows and departs from the cavity. It is affected directly by the bubble departure diameter. At the same heat flux, a smaller bubble departure diameter will result in a higher bubble departure frequency. During the previous decade, numerous studies on the bubble departure diameter and frequency predictions were reported. Carey (2008) elaborated and summarised the significant D_B and f_B models. Table 2-1 shows some models of the boiling departure diameter and frequency proposed by many researchers that can represent the D_B and f_B model development.

Table 2-1 Summary of the bubble departure diameter and frequency models

List of Authors	Models	Eq. number
Fritz (1935)	$Bo^{1/2} = 0.0208\theta$	(2-11)
	$Bo = \frac{g(\rho_l - \rho_v)D_B^2}{\sigma}$	(2-12)
Zuber (1959)	$Bo^{1/2} = \left[\frac{\sigma}{g(\rho_l - \rho_v)} \right]^{-1/6} \left[\frac{6k_l(T_W - T_{sat})}{q''} \right]^{1/3}$	(2-13)
Zuber (1964)	$Bo^{1/2} = \left[\frac{3\pi^2 \rho_l \alpha_l^2 g^{1/2} (\rho_l - \rho_v)^{1/2}}{\sigma^{3/2}} \right]^{1/3} Ja^{4/3}$	(2-14)
	$Ja = \frac{\rho_l c_{pl} [T_W - T_{sat}(P_o)]}{\rho_v h_{lv}}$	(2-15)
Cole (1967)	$Bo^{1/2} = 0.048Ja$	(2-16)
Cole and Rohsenow (1968)	$Bo^{1/2} = C(Ja^*)^{5/4}$	(2-17)
	$Ja^* = \frac{T_c c_{pl} \rho_l}{\rho_v h_{lv}}$	(2-18)
	$C = 1.5 \times 10^{-4}$ for water $C = 4.65 \times 10^{-4}$ for fluids other than water	
Kutateladze and Gogonin (1979)	$Bo^{1/2} = 0.25(1 + 10^5 K_1)^{1/2}$	(2-19)
	$K_1 = \left(\frac{Ja}{Pr_l} \right) \left\{ \left[\frac{g \rho_l (\rho_l - \rho_v)}{\mu_l^2} \right] \left[\frac{\sigma}{g(\rho_l - \rho_v)} \right]^{3/2} \right\}^{-1}$	(2-20)
	$K_1 < 0.06$	
Jensen and Memmel (1986)	$Bo^{1/2} = 0.19(1.8 + 10^5 K_1)^{2/3}$	(2-21)
Stephan (1992)	$D_B = 0.25 \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \left[1 + \left(\frac{Ja}{Pr_l} \right)^2 \frac{1}{Ar} \right]^{1/2}$	(2-22)
	for $5 \times 10^{-7} \leq \left(\frac{Ja}{Pr_l} \right)^2 \frac{1}{Ar} \leq 1$	

	$Ar = \frac{\rho_l^2 g}{\mu_l^2} \left(\frac{\sigma}{\rho_l g} \right)^{3/2}$	(2-23)
Jakob and Fritz (1931)	$f_B D_B = 0.078$	(2-24)
Peebles and Garber (1953)	$f_B D_B = 1.18 \left(\frac{t_g}{t_g + t_w} \right) \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_l^2} \right]^{1/4}$	(2-25)
Zuber (1963)	$f_B D_B = 0.59 \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_l^2} \right]^{1/4}$	(2-26)
Cole (1967)	$f_B D_B = C_1 \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_l^2} \right]^{1/4}$ $0.15 < C_1 < 1.4$	(2-27)
Ivey (1967)	$f_B^2 D_B = \text{constant}$ for inertia controlled growth $f_B^{1/2} D_B = \text{constant}$ for thermally controlled growth	
Mikic and Rohsenow (1969)	$f_B^{1/2} D_B = 0.83 Ja \sqrt{\pi \alpha_l}$	(2-28)

The bubble departure diameter and frequency models listed in Table 2-1 are the models that are based on boiling heat transfer from smooth surfaces. There are limited bubble dynamics models from structured surfaces especially from porous foam structures in the published literature. Bubble dynamics from artificial structured surfaces were experimentally and analytically studied by Nakayama *et al.* (1980). Active nucleation site density and bubble frequency predictions were proposed from their work. In their model, bubble dynamics from the structured surface are categorised into three phases: pressure build-up, pressure reduction and liquid intake. Their D_B model is expressed as

$$D_B = C_B \left[\frac{2\sigma}{g(\rho_l - \rho_v)} \right]^{1/2} \quad (2-29)$$

where $C_B = 0.42$ for R-11 and 0.22 for water.

Haider (1994) proposed the following equation for constant C_B in Eq. (2-29):

$$C_B = (3s_g \sin \theta) \left[\frac{2\sigma}{g(\rho_l - \rho_v)} \right]^{-1/6} \quad (2-30)$$

where s_g is the surface gap width of the boiling system.

Chien and Webb (1998) conducted a visualisation and analytical study on the bubble dynamics from a structured surface with a circular fin base that was produced by a machining process. Their structured surface possessed surface pores and sub-surface tunnels to enhance the boiling surface area and the bubble escaping process. Semi-analytical models of the bubble departure diameter and frequency were proposed from their study. They stated that their models can predict the bubble departure diameter and bubble frequency with accuracies of $\pm 20\%$ and $\pm 30\%$, respectively compared to their experimental data. Their models D_B and f_B models are expressed as

$$D_B = \left[\frac{\text{Bo} + \sqrt{\text{Bo}^2 + 2304(96/\text{Bo} - 3)}}{192 - 6\text{Bo}} \right]^{1/2} d_p \quad (2-31)$$

$$f_B = \frac{1}{0.0296 \left[\frac{7}{\pi} \frac{\rho_l T_w}{h_{lv} \rho_v \Delta T_{sat}} \frac{(D_B + d_p)}{(D_B - d_p)} \right]^{1/2} \left(\frac{D_B - d_p}{2} \right)} \quad (2-32)$$

Furberg (2011) proposed a model for f_B and D_B for a dendritic and micro-porous structure. The bubble frequency was determined based on the heat and mass balance during the nucleation process while the bubble diameter was determined by balancing the forces acting on the micro-structure surface. The bubble frequency is expressed as

$$\frac{f_B}{P} = \frac{q_{lat}'' / P}{V_B \Delta h_{lv} \rho_v} \quad (2-33)$$

Recently, further studies on the bubble growth and mechanism from the porous structures were performed by some researchers. Witharana *et al.* (2012) have performed an experimental study to measure the nucleation temperature of water on custom-fabricated nano- to micro-scale cavities by infrared thermometry. The results showed that the measured cavity data were in agreement with the predictions of the *Young-Laplace* equation, thus re-affirming the correctness of the classic view of heterogeneous bubble nucleation. They have also stated that the individual posts of any size have an insignificant effect on bubble nucleation.

Nucleate boiling heat transfer of saturated FC-72 on micro-pin-finned surface was experimentally investigated by Zhang *et al.* (2014) in microgravity environment. Their experimental results showed that the bubble detachment radius increases with increasing heat flux. A bubble departure radius model on micro-pin-finned surface in microgravity was developed. In their model, both bubble force balance and bubble coalescence are considered as two main factors influencing the size of bubble departure radius.

An experimental study on the effects of orientation angle, subcooling, heat flux, mass flux, and pressure on bubble departure diameter subcooled flow boiling were performed by Sugrue *et al.* (2014). The results indicated that the bubble departure diameter increases with increasing heat flux, decreasing mass flux, decreasing subcooling, and decreasing pressure. They also stated that the bubble departure diameter increases with decreasing orientation angle. They concluded that the mechanistic bubble departure diameter model of Klausner *et al.* (1993) and its modification by Yun *et al.* (2012) were found to correctly predict all the observed parametric trends, but with large average errors and standard deviation of $65.5 \pm 75.8\%$ for Klausner's and $37.9 \pm 51.2\%$ for Yun's.

Recently, Thiagarajan *et al.* (2015) have performed nucleate pool boiling experiments were on microporous copper and plain surfaces with HFE-7100. The bubble dynamics parameters such as the nucleation site density, bubble diameter at departure, and bubble departure frequency were obtained by using high-speed visualisation. Their results showed that the bubble site density and frequency of departure increases with temperature, while the diameter at departure stays relatively insensitive to temperature. They concluded that on the plain surfaces, both evaporative and quenching components contribute nearly equally to the total heat flux, while, on the microporous surfaces, the evaporative component contributes about 70% of the total heat. Hence, confirming that the microporous coating enhances nucleate boiling heat transfer as a direct result of its significantly higher active nucleation site density.

2.4 SUMMARY

A review of two-phase cooling systems for high heat flux electronic devices, pool and flow boiling heat transfer from the porous structures and the boiling parameters is carried out. The key findings from this review are as follows:

- 1) In general, two-phase cooling systems show better heat transfer performance compared with their single phase counterparts (natural or forced convection).
- 2) Two-phase cooling systems based on pool boiling or thermosyphon have been demonstrated to successfully dissipate high heat fluxes above 100 W/cm² from electronic devices. However, such systems will not be applicable to devices which are oriented away from the vertical direction during operation. This limitation has driven the development of pumped two-phase cooling systems for electronic packages.
- 3) The application of graphite foams in two-phase cooling systems was investigated by many researchers. However, the boiling phenomenon from the graphite foam is still not clearly understood. Inconsistent results of the effects of parameters such as type of coolant, working fluid level, and inclination angle on heat transfer performance were obtained. Therefore, research in this direction becomes very challenging. It is also observed that there are very limited published studies on pumped two-phase cooling systems using graphite foam inserts in the evaporator. With this motivation, the study of phase change cooling systems for densely-packaged electronics packages based on the pumped two-phase cooling system with graphite foam

inserts in the evaporator will be performed by the author as part of this research.

- 4) Nucleation site density, bubble departure diameter and frequency models have been proposed by many researchers for smooth and structured surfaces. However, to the best of the author's knowledge, there are no bubble dynamics models for porous foam structures especially for porous graphite foams. It is shown that the main factors which affect the boiling parameters (*i.e.* N_a , D_B and f_B) in a pool boiling system are thermophysical properties of fluid, surface conditions, contact angle and wall superheat. In order to understand the boiling mechanism and bubble dynamics from the porous graphite foam, bubble departure diameter, bubble departure frequency and nucleation site density are studied and presented in Chapter 7.

CHAPTER 3 EXPERIMENTAL SETUP AND PROCEDURES

To conduct a comprehensive experimental study on pool and flow boiling heat transfer from porous graphite foams, a thermosyphon and a pumped two-phase cooling experimental facility were designed and fabricated. The designed thermosyphon consists of three main parts: heating base, evaporator chamber and condenser while, the pumped two-phase facility consists of six main parts: flow boiling channel, heat simulator, micro-pump, air-cooled condenser, coolant reservoir, and tubing system. In the pumped two-phase system, the generated vapour in the evaporator rises to the air-cooled condenser and the condensed liquid go to the coolant reservoir. The coolant was circulated by using a micro-pump placed before the evaporator.

The experimental facilities were housed at the Thermal and Fluids Laboratory of the School of Mechanical and Aerospace Engineering, NTU. Details of the thermosyphon and pumped two-phase cooling facilities are described in the following sections.

3.1 EXPERIMENTAL FACILITY – POOL BOILING SYSTEM (THERMOSYPHON)

A thermosyphon consists of heating base, an evaporator and a condenser that allows a working fluid to travel between the other two components. Vapour generated at the evaporator rises due to buoyancy. It then condenses at the top of the chamber at the condenser, releasing its latent heat. Finally, gravity

returns the condensate back to the evaporator and the process is repeated. At the evaporator, heat will vaporise the working fluid and ultimately the heat will be dissipated at the condenser. Two different thermosyphons with the air-cooled and water-cooled condensers were fabricated and used in this study. A schematic of the experimental facility is shown in Fig. 3-1.

3.1.1 Heating Base

The heating base made of Teflon with a customised cartridge heater with maximum power rating of 300 W inserted at the centre of the base was used to generate a certain amount of heat, which can be adjusted by a power controller. The external dimensions of the heating base are $80(L) \times 80(W) \times 80(H)$ (mm). The customised cartridge heater of “Solidheat” has a non-uniform heat distribution along the heater such that the density of the heating element increases from the middle to the top of the heater. The heater also has a conductive top surface and an unheated section at the bottom part of the heater. To minimise the heat loss, the heater was insulated with ceramic insulation layers and a Teflon base. In the experiments, the rate of generated heat was adjusted by a “Carlo Gavazzi” WM14-96 power meter that is attached to the power controller to monitor the generated heat level. The structure and dimensions of the heating base and insulation layer are shown in Figs. 3-2 and 3-3, respectively.

Fig. 3-1 Schematic diagram of the thermosyphon cooling system.

Fig. 3-2 Structure and dimensions of heating base.

(All dimensions are in mm)

Fig. 3-3 Structure and dimensions of base insulation layer.

The customised cartridge heater was installed in an insulation sheath and three thermocouples were attached on the sheath to estimate heat loss from heater. At the bottom, a screw was used to fasten the heater to provide good contact at the interface between heater tip and copper plate. A graphite foam evaporator was bonded on a copper plate which was clamped tightly on the heater's surface. In addition, the customised heater was integrated to a copper plate which can be used as a platform to attach the graphite foam. The graphite foam sample was bonded on the copper plate by highly thermal conductive epoxy

adhesive “Omega Bond 101”. The properties of “Omega Bond 101” are provided in Table A-1 of Appendix A. With this arrangement, the contact resistance between heater and graphite foam was minimised. At the bottom, the heating element was fastened by screws. A rubber O-ring was employed at the interface between the copper plate and Teflon surface to ensure air- and liquid-tightness.

3.1.2 Evaporator Chamber

The evaporator chamber is the container where boiling occurs. The chamber was made of aluminium with a transparent polycarbonate window for allowing the operator to observe the boiling process. A graphite foam was placed in the evaporator chamber which was filled with the working fluid. The chamber was made of transparent polycarbonate so that the phase change phenomenon can be observed. It was assembled to the heating base and cooling top by two pairs of flanges and rubber O-rings were applied at each interface to ensure that the set-up is leak-proof. Pressure tests were carried out before the start of the experiments. The detailed structure and dimensions of the evaporator chamber are given in Fig. 3-4.

Fig. 3-4 Structure and dimensions of evaporating chamber with polycarbonate window. (All dimensions are in mm)

3.1.3 Air-Cooled and Water-Cooled Condensers

The heat carried by the evaporated vapour are dissipated and condensed by condenser. Two types of condenser viz. air-cooled and water-cooled condensers were used in the thermosyphon system. Originally, the experiments were carried out by using the air-cooled condenser. To improve the cooling performance, a water-cooled condenser was used. The air-cooled condenser consists of two copper pin fin arrays attached on two sides of the top cover by using a highly thermal conductive adhesive “Omegabond 101”. The air-cooled

condenser which can be fixed on the top of the evaporator chamber was designed to remove the heat from the thermosyphon system. For the air-cooled condenser, the main factors which affect convection heat transfer are the exposed surface area of the condenser, thermal conductivity of material and convection coefficient. To obtain high heat transfer rates, it is necessary to choose a highly thermal conductive material and maximise the contact surface area in an available space. The fin configuration is a popular way to extend the surface area of condenser and enhance the convection heat transfer. Pin fins were configured at both sides of the heat sink to maximise the fluid contact surface area. The heat carried by the vapour is conducted by the fins from the bottom to the top and then dissipated by forced convection. To enhance the heat removal of the condenser, a 300-mm diameter AC fan “ORIX” 18W was used to enhance heat removal from the condenser.

The water-cooled condenser is made up of circular copper coils which provide the heat transfer surface. It has one inlet and one outlet which allow cooling water from the circulation bath to provide indirect cooling of vapour formed during experiments. The copper coiling coil has a total heat transfer surface area of 18,000 mm². The main reason for the choice of its shape is the ease of fabrication. In this water-cooled condenser, the water was circulated by using a “PolyScience” bath circulator. The bath circulator can supply and circulate the water up to 30 L/min with the liquid temperature of 20°C. The heating base, evaporating chamber and condenser were clamped tightly onto each other through eight screws distributed evenly on the fringes. A rubber O-ring was employed at the interfaces to improve the air-tightness of the system. The

photographs of the assembled test section and the experimental facilities with the air-cooled and water-cooled thermosyphon described above are shown in Figs. 3-5 and 3-6, respectively.

(a)

(b)

Fig. 3-5 Photographs of the pool boiling test section with the (a) Air-cooled and (b) Water-cooled condenser.

(a)

(b)

Fig. 3-6 Photographs of the pool boiling experimental facility with the (a) Air-cooled and (b) Water-cooled condenser.

In this study, the boiling phenomena were observed and captured by using a high speed camera. Photographs of the high speed camera arrangements in the thermosyphon are shown in Fig. 3-7.

Fig. 3-7 Photographs of visualisation study layout with a high speed camera of the thermosyphon with the (a) Air-cooled and (b) Water-cooled condenser.

3.2 EXPERIMENTAL FACILITY – FLOW BOILING SYSTEM (PUMPED TWO-PHASE COOLING)

As presented in Chapter 1, the development of cooling technologies for electronic devices had shifted from natural convection and single-phase forced convection to phase change cooling techniques. In a phase change system, the evaporation, condensation and fluid flow rates are determined by the system design and properties of the wick. A capillary two-phase loop system has high heat flux capability due to its sintered wick structure. However, there are two important mechanisms that limit its thermal performance. The first is the capillary limit, which occurs when the sum of the pressure drops in the liquid

and vapour sections exceeds the driving force generated by the capillary pressure created by the fluid surface tension at the evaporator section. The second is the boiling limit, which occurs when the heat flux causes bubbles to nucleate and grow in the wick structure of the evaporator section. This restricts the local flow of liquid and leads to evaporator dryout.

For high heat flux cooling systems with densely packed components, the pumped flow boiling system is a promising technique due to the flexible positioning of the evaporator. An external power source provided by a pump overcomes the limited driving force of the capillary loop. In addition, there is no orientation problem for a pumped cooling system and the evaporator and condenser can be positioned easily. To enhance the evaporator performance, porous graphite foam is selected as the material to extend the surface area and to promote boiling. To investigate the flow boiling heat transfer and phenomena from the porous graphite foams, a single-channel flow boiling facility was fabricated.

A single-channel pumped two-phase cooling experimental facility was designed to remove the heat from a heat simulator of dimensions 80 mm $\times$ 60 mm in this study. The designed facility consists of six main parts: flow boiling channel (test section), heat simulator, micro-pump, air-cooled condenser, coolant reservoir, and tubing system. The schematic diagram and photograph of the facility are shown in Fig. 3-8. The heater assembly consists of a copper block and eight cartridge heaters to generate up to 5000 W of heat. The heat simulator was insulated with a Teflon base to reduce the heat loss. The

evaporator channel was located and clamped tightly on top of the heater. The test section was designed to ensure that the coolant can completely fill and flow in the evaporating channel. Before the coolant is directed into the evaporator, it flows through an expansion channel with guide vanes to achieve uniform flow. The inlet of the evaporator is connected to the expansion channel, whereas its outlet is connected to a 6.35 mm (1/4") pipe via a NPT connector. The evaporator, together with a top cover and two side covers, form a channel of 12 mm $\times$ 60 mm $\times$ 220 mm. The wide cross sectional area design minimises the effect of variation in the transverse direction. By changing the thickness of the top cover, the inner height of the evaporator channel are adjusted, and thus the gap between the porous foam and the top cover can be varied.

The evaporator channel was located and clamped tightly on top of the heater. Thermal conductive thermal grease, “Omegatherm 201”, was applied to reduce the thermal contact resistance between the heater and evaporator channel surface. Graphite foam inserts were attached on the bottom of channels by thermal conductive adhesive, “Omegabond 101”. The evaporator cover was also made of aluminium and clamped tightly to the evaporator through 18 M4 screws. A square rubber ring was attached to the interface between the evaporator channel and cover to ensure air- and liquid-tightness of the system. To facilitate visualisation, the side covers were fabricated with transparent polycarbonate windows. The flow boiling channel and its photograph are shown in Fig. 3-9.

(a)

(b)

Fig. 3-8 (a) Schematic diagram and (b) Photograph of the flow boiling experimental facility.

Fig. 3-9 (a) The design and (b) Photograph of the flow boiling channel.

A Cole-Parmer “Micropump R-73011-18” pump was used to convey the coolant from the coolant reservoir to the evaporator channel and continuously circulate it in the closed-loop tubing system. The pump was equipped with a high precision Cole-Parmer gear pump drive which can be varied from 0 ~ 5000 rpm and can provide accurate and pulse-less fluid delivery. The chemical resistance of the pump makes it compatible with a wide range of liquids. The phase change coolant inside the evaporator channel was heated such that the

resulting liquid-vapour mixture flows to the air-cooled condenser section. The condensed liquid returns to the reservoir and will be circulated continuously through the tubing system. The air-cooled condenser was designed to ensure that the vapour can be completely condensed before it is circulated to the evaporator.

The condenser tubes were made of aluminium which is compatible with most coolants including FC-72. They were fitted with aluminium fins to enhance heat transfer with aluminium-brazed joints which ensured that the system was leak-free. The condenser of dimensions $200 (L) \times 50 (W) \times 230 (H)$ mm was mounted on a metal base. Two fans were used to remove heat from the condenser. The speed of the fans can be controlled to vary the heat removal rate. “Swagelok” stainless steel tubes and fittings were used in the entire closed-loop system to satisfy the requirements of the system, especially the micro pump and filter compatibility, and to ensure leak-tightness. The tubing system was designed for both vacuum and high pressure operational conditions. Seven “Swagelok” needle valves were installed in the system to facilitate pressure and vacuum tests and to charge/discharge the coolant.

3.3 SENSOR AND MEASUREMENT SYSTEMS

In the experiments, all the signals were acquired by a PC-based data acquisition (DAQ) Yokogawa MW-100 and MX-100 for pool and flow boiling systems, respectively. All the sensors used in the experimental setup were calibrated before the start of the experiments. Quasi-steady-state measurements,

uncertainty measurements and heat loss calculations from the setup are discussed.

3.3.1 Measurement System in Pool Boiling Facility

In the pool boiling cooling system, thermocouples and pressure transducer were used to measure the temperatures and pressure, respectively. Ten type K thermocouples were installed to measure the temperatures of the liquid, vapour, wall, and at the location along the cartridge heater. To measure the wall temperatures, two thermocouples (T_{w1} and T_{w2}) were placed on the top surface of the heater to obtain the average surface temperature. The liquid and vapour temperatures of the coolant were measured by the thermocouples in the chamber inserted through the top section of the chamber. To obtain the temperature distribution along the heater, three thermocouples (T_{h1} , T_{h2} and T_{h3}) were spaced equally along the heater. Another three thermocouples (T_{i1} , T_{i2} and T_{i3}) were attached to the insulation layer to estimate the heat loss. The thermocouple positions in the heater are shown in Fig. 3-10. All thermocouples were calibrated before the experiments. Prior to the experiments, 1-D thermal contact resistance from the positions of Thermocouple 1 and 2 of Fig. 3-10 to the top boiling surface was analysed with thermal conductivity values of the thin copper block and very thin thermal adhesive layer taken as 400 W/m·K and 1.1 W/m·K, respectively. The calculated total thermal contact resistance is $1.7 \times 10^{-4} \text{ m}^2 \cdot \text{K/W}$.

The saturation temperature of the working fluid depends on the operation pressure. In order to measure the pressure inside the chamber, a “OMEGA PX-

409” pressure transducer was installed on the cooling top section of the evaporator chamber.

Fig. 3-10 Thermocouple positions on the heater and insulation.

All the data measured by thermocouples were monitored and acquired by a “Yokogawa MW-100” DAQ system. In addition, the fully separated channels with differential connections provide better accuracy for millivolt-level measurements. The unit consists of a main module equipped with an Ethernet port, I/O modules for input and output of signals, and a base plate to which the first two items are mounted. The “Yokogawa MW-100” system consists of a main control unit and two attached modules with 20 channels. The recorded data were processed in the Yokogawa Data Viewer Software. The software is easy to program for user-end application and the data can be exported to common spreadsheet applications such as Microsoft Excel. The I/O modules of MW 100 Data acquisition Unit were calibrated by a “Yokogawa MW 100”

Calibrator. Calibration between 20 mV – 100 V was performed with the standard procedure described in the manual (Yokogawa Manual Sheet, 2008).

To obtain accurate temperature measurements, all the thermocouples have to be calibrated prior to the commencement of the experiments. Ten (10) K-type thermocouples were calibrated using a thermocouple calibrator device (Fig. 3-11) which has an adjustable temperature console. The thermal calibrator serves as a constant heat source to ensure accurate calibration of the thermocouples. The ten thermocouples wires were inserted into the cylindrical chamber of the calibrator and the temperatures were set to known points. The other ends of the thermocouple wires were connected to the data acquisition card and to the data logger software where the temperatures were measured and displayed directly. The setup was left to stand for about 15 minutes while the thermocouples were stabilised to the pre-set temperature. The corresponding measured temperature was then recorded and a calibration plot of measured temperature versus reference temperature was drawn. A separate plot was done for each thermocouple junction. The complete calibration plots are provided in Fig. B-1 in Appendix B.

Fig. 3-11 Thermocouple calibrator.

3.3.1.1 Steady State Tests

Pressure and steady state tests were performed before the start of experiments. The pressure test was carried out to ensure chamber pressure capability and air tightness. In the pressure test, the chamber was subjected to 150 kPa pressure and maintained for 8 hours. If no pressure drop was observed, the steady state test can be performed. Prior to the experiments, degassing procedures were conducted to remove any dissolved and non-condensable gas in the coolant and porous graphite foam. The degassing procedures were performed by applying the heating power of 20 W to the system for 60 minutes and then releasing it to the atmosphere by opening the charging ports of the pool boiling and flow boiling experimental setups as shown in Figs. 3-1 and 3-8, respectively

To perform steady state tests, a heating power of 35 W was imposed on the system and the average values of temperature variations according to the readings detected by the two thermocouples on the heater surface are shown in

Fig. 3-12, where $T_n - T_{n-1}$ is the temperature difference between two consecutive measurements. It was found that after 15 minutes, the average temperature variation was only 0.5°C , which can be considered to be steady. After the system had reached steady state, all data were recorded for 2 minutes at a sample interval of 200 ms.

Fig. 3-12 Temperature variations between two consecutive measurements for wall temperature steady state test.

3.3.1.2 Uncertainty Analysis

Measurement uncertainties generally occur during data acquisition and reduction. Improper performance of instruments, limitation in probe sensitivity and frequency response, and inaccuracy of instruments contribute to experimental uncertainties. The uncertainty can be classified into systematic uncertainty and random uncertainty. Systematic uncertainty may be due to poor measurement technique, errors in instrumental calibration, software errors or failure to correct for external conditions (*e.g.* temperature). In simple terms,

systematic uncertainty is mainly due to the wrong operation of instruments and other human errors. With careful operations and regular calibrations, systematic uncertainty can be minimised. The uncertainties of the interested parameters calculated in the following test refer to random uncertainties. The uncertainty analysis performed in this study is based on the method described by Taylor (1997). $v, \dots y, \dots z$ are measured quantities with uncertainties $\delta v, \dots \delta y, \dots \delta z$, and the measured values of $v, \dots y, \dots z$ are the functions of the parameter q as

$$q = f(v, \dots y, \dots z) \quad (3-1)$$

Assuming that the uncertainties of the measured quantities are independent and fall within random normal distributions, the uncertainty of parameter q can be calculated from

$$\frac{\delta q}{|q|} = \sqrt{\left(\frac{\delta q}{v}\right)^2 + \dots + \left(\frac{\delta q}{y}\right)^2 + \dots + \left(\frac{\delta q}{z}\right)^2} \quad (3-2)$$

From the above equation, it can be seen that the uncertainty of a calculated parameter depends on the uncertainties of the independent quantities $v, \dots y, \dots z$. For N measurements, the standard deviation of the mean is used to describe the uncertainty. For example, the standard deviation of the quantities v in N measurements can be obtained by

$$\sigma_v = \sqrt{\frac{1}{N} \sum_{i=1}^N (v_i - \bar{v})^2} \quad (3-3)$$

where v_i and $\bar{v}$ are the measured value and average value for quantity v , respectively. Therefore, the uncertainty of v can be given by

$$\delta_v = \frac{\sigma_v}{\sqrt{N}} \quad (3-4)$$

Equation (3-4) indicates that the uncertainty of quantity v due to data reduction results mainly from the use of finite data in calculating a statistical mean.

The rate of heat loss along the y -direction $Q_{L(y)}$ of the cartridge heater was calculated (Fig. 3-13). The temperature distribution along the y -direction of the heater measured by four thermocouples T_w , T_{h1} , T_{h2} , and T_{h3} (Fig. 3-10) is shown in Fig. 3-14. $Q_{L(y)}$ was calculated by assuming one-dimensional heat conduction along the heater with copper material from the high temperature side (T_w) to the low temperature side (T_{h3}) as

$$T_{sat} = f(P_v) \quad (3-5)$$

$$Q_{L(y)} = k_{Cu} A_h \frac{(T_w - T_{h3})}{L_h} \quad (3-6)$$

$$\text{where } A_h = \frac{\pi}{4} d_h^2$$

The heater was well insulated using two layers of insulation of ceramic and Teflon material. Therefore, the effect of natural convection from the bottom and outer side of the Teflon base can be considered to be negligible. Using Eq. (3-6), the uncertainty in the rate of heat loss of the heater is about $\pm 8 \%$. The

data and calculations of the heat loss are provided in Tables B-1 to B-4 in Appendix B.

Fig. 3-13 Schematic diagram of heat loss along the y-direction of the heater.

Fig. 3-14 Temperature distribution along the y-direction of the heater.

3.3.2 Measurement System in Flow Boiling Facility

In the flow boiling experiments, temperatures, pressures, coolant flow rates, and heating powers were measured and acquired by using sensors and DAQ system. The electrical power can be adjusted by a power controller and displayed by a power meter. Eight K-type “Cement-On” thermocouples were inserted at the interface between the evaporator and heater surface to measure the average evaporator surface temperature. Two K-type thermocouples were inserted in the copper block to measure the temperature of the heater and another two thermocouples were screwed to the evaporator cover to measure the liquid temperature. Three K-type thermocouples were attached in the tubing system to measure the coolant temperatures at the reservoir, at the location before the evaporator inlet, and at the evaporator outlet. Due to the difficulty of securing thermocouples within the graphite foam, temperatures within the foam were not measured. Two “Omega PX409” gauge pressure transducers were installed on the evaporator cover to measure the pressure inside the evaporator channel. The system volumetric flow rate was obtained by a “Digimesa” flow meter installed between the micro pump outlet and the evaporator inlet. All the sensors were carefully calibrated before the commencement of the experiments. The digital signals from the sensors were channelled to the “Yokogawa” (MX100) data acquisition system attached to a personal computer.

Pressure tests were performed before the start of the experiments to ensure that the system was leak-proof. The steady-state condition was considered to be

reached when the difference between two consecutive average heater wall temperatures was smaller than 0.2°C . The tests showed that 15 minutes were required to reach quasi steady state from the initial condition, and 5 minutes for the system to reach the next steady state after increasing the power input. The heat flux was increased gradually by using the power controller to achieve respective power level in the tests. Systematic uncertainty can be minimised by careful experimentation. To minimise data reduction uncertainty, the time-averaging method was employed. The time-averaged value is defined as the average value of a parameter measured for a prescribed duration (1 minute with a sampling rate of 5 Hz) under quasi-steady-state condition. The uncertainty of the thermocouple measurements is $\pm 0.5^{\circ}\text{C}$ while the uncertainties of the pressure transducer, power meter, and flow meter measurements are within $\pm 0.25\%$, $\pm 0.3\%$, and $\pm 0.5\%$ of their full-scales, respectively. It is noted that all the thermocouples, pressure transducers, flow meter and the DAQ system used in the multi-channel flow boiling system are the same with the sensors and the DAQ system used in the single-channel flow boiling system. In this study, the uncertainties of the boiling parameters were determined by the method of Taylor (1997) as described previously in the pool boiling system.

The rate of heat loss ($Q_{L(y)}$) of the test section was evaluated before the commencement of the boiling experiments. $Q_{L(y)}$ was calculated from the difference between the total power input (q_t) to the cartridges heaters and the heat transfer rate to the single-phase liquid of the coolant through the channel

(q_c). With $q_t = VI$ and $q_c = GA_{ff}c_{p,f}(T_{f,e} - T_{f,i})$, the relative rate of heat loss (ε) was calculated by

$$\varepsilon = \frac{q_t - GA_{ff}c_{p,f}(T_{f,e} - T_{f,i})}{q_t} \quad (3-7)$$

By using Eq. (3-7), the average value of ε was estimated to be about 7.5%. $T_{f,e}$ and $T_{f,i}$ in Eq. (3-7) are the liquid temperatures at the exit and inlet of the channel, respectively. The vapour quality at the channel exit (x_e) can be determined from the energy balance calculation as follows:

$$x_e = \frac{q''A_h - GA_c c_{p,f}(T_{sat} - T_{f,i})}{GA_c h_{fg}} \quad (3-8)$$

3.4 POROUS GRAPHITE FOAMS AND WORKING FLUIDS

3.4.1 Thermophysical Properties

In this study, four types of graphite foam with different thermophysical properties *viz.* “Pocofoam” of 61% porosity, “Pocofoam” of 75% porosity, “Kfoam” of 72% porosity, and “Kfoam” of 78% porosity were used as evaporator inserts in the pool and flow boiling cooling systems. The pore diameter at up to 30 ~ 50 locations of the graphite foams was measured by Image-Pro software based on photographs captured by Scanning Electron Microscopy (SEM). The average pore diameter (d_p) was determined and compared to its properties provided by the manufacturer. The internal structures of the tested foams obtained by SEM are shown in Fig. 3-15.

It can be seen that the internal structures of the graphite foams consist of near-spherical pores with small openings on the pore walls. The method of determination of the pore diameter in the software environment is presented in Fig. 3-16. All the measured d_p values from the graphite foam samples were averaged. The d_p values of all tested foams are presented in Table 3-1.

Fig. 3-15 SEM images of graphite foams (a) “Pocofoam” 61% porosity, (b) “Pocofoam” 75%, (c) “Kfoam” 78% porosity and (d) “Kfoam” 72% porosity.

Fig. 3-16 Determination of pore diameter of graphite foams.

Porosity is defined as the total void volume divided by the total volume occupied by the solid matrix and void volume, which can be expressed as

$$\varepsilon = \frac{V_t - V_s}{V_t} \quad (3-9)$$

where V_t and V_s are the total volume and the volume of solid fraction of porous material, respectively. The pertinent properties of the graphite foam provided by the manufacturer are presented in Table 3-1.

Table 3-1 Pertinent properties of the graphite foams (Poco Graphite, 2008; Kopper Inc., 2008)

Properties	“Pocofoam” 61%	“Pocofoam” 75%	“Kfoam” 72%	“Kfoam” 78%
d_p (μm)	350	350	650	500
	310 (measured)	310 (measured)	645 (measured)	495 (measured)
ε (%)	61	75	72	78
ρ (g/cm ³)	0.90	0.50	0.48	0.34
k_{eff} (W/m·K)	245	135	110	55

3.4.2 Working Fluids

In two-phase cooling systems, the working fluid plays an important role in heat transfer performance. Thermophysical properties of the working fluids directly influence the heat transfer coefficient of the system. In the pool boiling system, the working fluid has to be chemically compatible with the components and must have a large dielectric strength to prevent electrical connections. As reported by Palm and Khodabandeh (2005), there are several considerations for choosing the working fluid for two-phase cooling systems in electronic products. Component reliability and risk level to the equipment and environment have to be considered. Thermal and physical properties of the coolants have to be examined to achieve high heat transfer performance of the systems.

In the current system, FC-72, HFE-7000 and HFE-7100 dielectric liquids obtained from 3M Ltd., USA were chosen as coolants. These phase change liquids are commonly used for cooling electronic devices or in other thermal management schemes. Both these fluids are thermally and chemically stable, non-flammable, and practically non-toxic and have low Global Warming Potential (GWP). The selected thermophysical properties of the coolants at 25°C and 1 atm are shown in Table 3-2.

Table 3-2 Thermophysical properties of the coolants at 25°C (3M Product Information, 2009)

Properties	FC-72	HFE-7000	HFE-7100
Boiling point at 1 atm [°C]	56	34	61
Vapour pressure [kPa]	30.9	64.6	28.0
Liquid density [kg/m ³]	1680	1400	1520

Specific heat [J/kg·K]	1100	1300	1170
Latent heat of vaporisation [kJ/kg]	88	142	112
Thermal conductivity [W/m·K]	0.057	0.075	0.068
Surface tension [mN/m]	10	12.4	13.6
Dielectric strength [kV]	38	40	28

The saturation temperatures T_{sat} of the coolants corresponding to the saturated pressures P_{sat} were calculated from the following equations given by the manufacturer:

$$\text{FC-72: } T_{sat}(K) = \frac{1562}{\left[9.729 - \log_{10}(P_{sat})\right]} \quad (3-10)$$

$$\text{HFE-7000: } T_{sat}(K) = \frac{3549}{\left[22.978 - \ln(P_{sat})\right]} \quad (3-11)$$

$$\text{HFE-7100: } T_{sat}(K) = \frac{3641.9}{\left[22.415 - \ln(P_{sat})\right]} \quad (3-12)$$

3.5 SUMMARY

Details of the pool and flow boiling experimental setups are presented in this chapter. All sensor and measurement systems, uncertainty analyses and the procedures of experiment are presented in this chapter. The methods to calculate the rate of heat losses from the pool and flow boiling setups are also presented. As discussed, the uncertainties of the parameters are determined from the individual measured uncertainties which are calculated by using the method described by Taylor (1997). However, the results of the uncertainties

of the boiling parameters such as boiling heat transfer coefficients, dimensionless numbers and heat loss will be presented in the respective chapter that discusses the parameters. The thermophysical properties of the graphite foams and dielectric coolants used in this study are also presented.

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In this chapter, pool boiling heat transfer and phenomena from the porous graphite foam insert with different parameters are discussed. The effect of graphite foam type, working fluid type and working fluid volume were investigated. The boiling process from the porous graphite foams were also observed and recorded to study the pool boiling phenomena such as the bubble dynamics, Critical Heat Flux (CHF), and bubble parameters. The experimental parameters tested in this study are given in Table 4-1.

Table 4-1 Experimental parameters

No	Parameter	Type/ level
1.	Graphite foam type	1) Copper block (basis test)
		2) “Pocofoam” 61%
		3) “Pocofoam” 75%
		4) “Kfoam” 72%
		5) “Kfoam” 78%
2.	Working fluid type	1) FC-72
		2) HFE-7000
		3) HFE-7100
3.	Working fluid volume (V_l)	1) $V_{l1} = 1.0$ cm
		2) $V_{l2} = 1.5$ cm
		3) $V_{l3} = 2.0$ cm

Boiling heat transfer experiments from a smooth copper block were also performed as a basis for comparison. The boiling heat transfer phenomenon was analysed from the surface temperature readings, boiling curves and non-dimensional analysis of the tested parameters.

4.1 THE EFFECT OF GRAPHITE FOAM TYPE

As shown in Table 4-1, four types of graphite foam evaporator inserts *viz.* “Pocofoam” of 61% porosity, “Pocofoam” of 75% porosity, “Kfoam” of 78% porosity, and “Kfoam” of 72% porosity and smooth copper block evaporator inserts were investigated. Experiments were conducted to explore the possible boiling heat transfer enhancement from the porous graphite foam (*i.e.* as compared to copper block) and study the boiling phenomena from the porous graphite foam structures. The pertinent thermophysical properties of the graphite foams are shown in Table 3-1 of Chapter 3. All samples were fabricated into cubes of dimensions 30(L) × 30(W) × 20(H) (mm) by Wire Cut Electron Discharge Machining (EDM). The dimensions and photograph of the sample are shown in Fig. 4-1. The properties and SEM images of the graphite foams are given in Table 3-1 and Fig. 3-15 of Chapter 3, respectively.

Fig. 4-1 Dimensions (in mm) and a photograph of the graphite foam insert sample.

Table 3-1 in Chapter 3 shows that “Pocofoam” graphite foams of 61% and 75% porosities possess the same average pore diameter (*e.g.* 350 μm) with significant difference in the bulk thermal conductivity. On the other hand, “Kfoam” graphite foams of 78% and 72% porosities have a larger pore diameter and lower bulk thermal conductivity when compared to the “Pocofoam” foams. In this study, the effects of graphite foam types were evaluated by using FC-72 and HFE-7000 dielectric liquids. To evaluate the boiling performance from different graphite foam types, wall temperature readings and boiling curves were derived from the experimental data. The wall temperature readings and boiling curves from different graphite foams and smooth copper block are shown in Figs. 4-2 and 4-3.

As shown in these figures, the graphite foams inserts were compared with a smooth copper block immersed in phase change coolants FC-72 and HFE-7000, respectively. It is obvious that the wall temperatures of the copper block are much higher than those of the graphite foams. It can be seen that at heat flux of $q'' = 112.68 \text{ W/cm}^2$, the wall temperature T_w is maintained at about $87.9 \sim 99.1^\circ\text{C}$ and $79.8 \sim 93.1^\circ\text{C}$ by the tested graphite foams with FC-72 and HFE-7000, respectively whereas it approaches 158°C and 149°C in the copper block with FC-72 and HFE-7000, respectively.

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Fig. 4-2 (a) Wall temperature readings and (b) Boiling curves for different graphite foam types with saturated FC-72.

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Fig. 4-3 (a) Wall temperature readings and (b) Boiling curves for different graphite foam types with saturated HFE-7000.

The results show that the average wall temperatures of the graphite foams of “Pocofoam” 61%, “Pocofoam” 75%, “Kfoam” 78%, and “Kfoam” 72% with FC-72 coolant were lowered by up to 28.4, 21.7, 23.3, and 20.1°C compared to those of the copper block, respectively. For HFE-7000, the wall temperature

differences between the cooper block and graphite foams of “Pocofoam” 61%, “Pocofoam” 75%, “Kfoam” 78%, and “Kfoam” 72% are 30.5, 21.8, 23.5, and 19.1°C, respectively. It can also be seen that CHF with the smooth copper block occurred at 112.68 W/cm² for FC-72 and HFE-7000 coolants. With the porous graphite foam evaporator inserts, the CHF values were delayed to 218.31 ~ 225.35 W/cm² (*i.e.* enhanced by up to 1.9 ~ 2 times). As shown in Figs. 4-2 and 4-3, the graphite foam evaporators have reduced significantly the superheat temperatures at the respective heat fluxes. Based on the experimental data, the boiling heat transfer coefficients h_b were calculated from Eq. (4-1).

$$h_b = \frac{q''}{(T_w - T_{sat})} \quad (4-1)$$

$$q'' = \frac{q}{A_h} \quad (4-2)$$

Saturation temperatures T_{sat} of FC-72 and HFE-7000 in Eq. (4-1) were calculated at their respective pressures as expressed in Eqs. (3-10) and (3-11), respectively. The wall temperature differences and boiling heat transfer ratios between different graphite foams and copper block evaporator are presented in Tables 4-2 and 4-3. It can be clearly seen that the graphite foam evaporator inserts have enhanced significantly the boiling heat transfer coefficient by up to 2 ~ 3 times as compared to the smooth copper block evaporator.

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Table 4-2 Wall temperature differences and boiling heat transfer coefficient ratios between different graphite foams and Cu block for FC-72

$\overline{\Delta T_{W(Copper-Graphite\ Foam)}} \text{ (}^{\circ}\text{C)}$		$h_{b(Graphite\ Foam)} / h_{b(Copper)}$	
$\overline{\Delta T_{W(Copper-Pocofoam61\%)}}$	28.4	$h_{b(Pocofoam61\%)} / h_{b(Copper)}$	2.84
$\overline{\Delta T_{W(Copper-Pocofoam75\%)}}$	21.7	$h_{b(Pocofoam75\%)} / h_{b(Copper)}$	2.04
$\overline{\Delta T_{W(Copper-Kfoam78\%)}}$	23.3	$h_{b(Kfoam78\%)} / h_{b(Copper)}$	2.19
$\overline{\Delta T_{W(Copper-Kfoam72\%)}}$	20.1	$h_{b(Kfoam72\%)} / h_{b(Copper)}$	1.89

Table 4-3 Wall temperature differences and boiling heat transfer coefficient ratios between different graphite foams and Cu block for HFE-7000

$\overline{\Delta T_{W(Copper-Graphite\ Foam)}} \text{ (}^{\circ}\text{C)}$		$h_{b(Graphite\ Foam)} / h_{b(Copper)}$	
$\overline{\Delta T_{W(Copper-Pocofoam61\%)}}$	30.2	$h_{b(Pocofoam61\%)} / h_{b(Copper)}$	2.97
$\overline{\Delta T_{W(Copper-Pocofoam75\%)}}$	21.8	$h_{b(Pocofoam75\%)} / h_{b(Copper)}$	2.10
$\overline{\Delta T_{W(Copper-Kfoam78\%)}}$	23.5	$h_{b(Kfoam78\%)} / h_{b(Copper)}$	2.17
$\overline{\Delta T_{W(Copper-Kfoam72\%)}}$	19.1	$h_{b(Kfoam72\%)} / h_{b(Copper)}$	1.94

The results also show that there are significant differences in the boiling heat transfer coefficients for different types of the graphite foam. The use of “Pocofoam” of 61% porosity resulted in the highest boiling performance as compared to the other graphite foams. As can be extracted from Tables 4-1 and 4-2, the boiling heat transfer coefficient ratios of “Pocofoam” of 61% porosity to “Pocofoam” of 75% porosity, “Kfoam” of 78% porosity, and “Kfoam” of

72% porosity are about 1.4, 1.3, and 1.5, respectively with FC-72 and HFE-7000 coolants.

To evaluate different boiling performances from the tested graphite foams, the generated boiling bubble parameters (*i.e.* nucleation site density and bubble departure frequency) are analysed. The superior boiling performance of the graphite foams to the copper block can be explained from the porous structures and properties of the foams. As mentioned previously in Chapter 3, the porous graphite foams structures consist of spherical pores with small openings on the pore walls. This structure would give larger solid-fluid contact area (*e.g.* larger total exposed surface area) and more nucleation sites as compared to the copper block structure. The resulting larger nucleation site density in the nucleate boiling mode means higher boiling heat transfer coefficient can be achieved with the use of porous graphite foams. To support this argument, boiling images recorded by a high-speed camera were used to compare the nucleation sites or generated bubbles from the graphite foams and copper block evaporators. The boiling images from the copper and graphite foams with FC-72 at different heat fluxes are shown in Fig. 4-4.

Figure 4-4 shows that at the same heat flux, the nucleation site densities from the graphite foams are much larger qualitatively than the copper block for both tested coolants. It can also be seen that the number of nucleation sites in the “Pocofoam” 61% are higher than those of other graphite foams. This also shows that “Kfoam” of 78% porosity had generated larger bubble density than

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“Pocofoam” of 75% porosity. It can be seen that the lowest nucleation site density from the graphite foam inserts is from “Kfoam” of 72% porosity.

Fig. 4-4 Captured boiling images from the copper and graphite foams with FC-72 coolant

Based on the captured boiling images, it is difficult to quantify the active nucleation site density from the graphite foam from the exposed surface, and it will be more difficult to quantify it from the internal structure of the graphite foam. To evaluate the potential nucleation site density from the porous

structure of the graphite foams, the total internal surface area to volume ratio (β) was determined from the graphite foam samples. The values of β for different graphite foam types were calculated based on the models of Yu *et al.* (2006) and Leong and Li (2011).

In Yu *et al.*'s model (2006), β is expressed as

$$\beta = \frac{\pi d_p}{H^3} (3H - 2d_p) \quad (4-3)$$

whereas in the model proposed by Leong and Li (2011), β can be calculated by

$$\beta = \frac{1}{a^3} (6a^2 - \frac{3}{2} \pi d_c^2) \quad (4-4)$$

The differences in the β values from the two models shown in Table 4-4 can be attributed to the different geometric and selected parameters of the graphite foam unit cell used which nevertheless are of the same order of magnitude. The values of β can be used to quantify the potential active nucleation site density. The total exposed area of the porous graphite foams and the role of its internal structure in boiling heat transfer will be discussed in more detail in Chapter 5.

As mentioned in Chapter 2, besides the nucleation site density, the boiling heat transfer performance will be affected significantly by the bubble departure frequency.

Table 4-4 Total internal surface area to volume ratio (β) calculated from two models

β based on Yu <i>et al.</i> (2006)	β [m^2/m^3]
“Pocofoam” 61%	10,400
“Pocofoam” 75%	9,650
“Kfoam” 78%	5,600
“Kfoam” 72%	5,900
β based on Leong and Li (2011)	β [m^2/m^3]
“Pocofoam” 61%	19,500
“Pocofoam” 75%	16,000
“Kfoam” 78%	15,300
“Kfoam” 72%	12,500

The bubble departure frequency was determined and analysed to further explore the different boiling performance from the graphite foams. To determine the bubble departure frequency, the bubble growth and departure processes were recorded by using a high-speed camera at 2005 frames per second (fps). The captured images were analysed frame by frame to determine the bubble growth and departure phenomena. The bubble departure frequency calculation method is portrayed in Fig. 4-5 which shows that a period of bubble growth and departure can be divided into waiting and bubble departure times. Once these times are determined, the bubble departure frequency can be calculated from

$$f_d = \frac{1}{t_w + t_d} \quad (4-5)$$

where t_w and t_d are the waiting and departure times, respectively. At the frame rate of 2005 fps, the number of frames in the waiting and departure periods are M and N , respectively. Therefore, t_w and t_d can be calculated as

$$t_w = (M \text{ frames}/2005 \text{ fps}) \quad (4-6)$$

$$t_d = (N \text{ frames}/2005 \text{ fps}) \quad (4-7)$$

The bubble departure frequencies from different graphite foams were obtained by averaging the measured values from 30 different foam cavities. The comparison of the bubble departure frequency from the different graphite foams are shown in Fig. 4-6. As shown in Fig. 4-5, at $q'' = 56.34 \text{ W/cm}^2$, the measured f_d from the “Pocofoam” 61%, “Pocofoam” 75%, “Kfoam” 78%, and “Kfoam” 72% are 171, 156, 163, and 149 Hz with FC-72 coolant, respectively. For HFE-7000 coolant, the f_d values are 166, 152, 159 and 149 Hz, respectively.

Fig. 4-5 Bubble departure frequency determination from the captured images of bubble growth and departure.

(a)

(b)

Fig. 4-6 Measured bubble departure frequency (f_d) from different graphite foam inserts with (a) FC-72 and (b) HFE-7000 coolant.

From the total internal surface area to volume ratio (β) and bubble departure frequency analysis, “Pocofoam” of 61% porosity had produced larger β and highest bubble departure frequency as compared to the other tested foams. Therefore, “Pocofoam” of 61% porosity exhibits the highest boiling heat transfer performance. The same analyses apply to the boiling heat transfer

performance of “Kfoam” of 72% porosity. This foam has the smallest β value and lowest bubble departure frequency among the other graphite foams that resulted in the lowest boiling heat transfer coefficient.

An interesting result is shown from the comparison of “Pocofoam” of 75% and “Kfoam” of 78% porosity. The results show that the boiling heat transfer performance of “Kfoam” of 78% porosity is about 8 to 15% higher than “Pocofoam” of 75% porosity. From the calculation of β , “Pocofoam” of 75% porosity possesses larger values compared to those of “Kfoam” of 78% porosity (See Table 4-3). Conversely, from the f_d measurements, “Kfoam” of 78% porosity generated higher bubble departure frequency compared to “Pocofoam” of 75% porosity. It is noted from the thermophysical properties of the graphite foams shown in Table 3-1 that “Pocofoam” of 75% porosity has an effective thermal conductivity which is about 2.5 times larger than that of “Kfoam” of 78% porosity. Thus, “Pocofoam” 75% should be more efficient in dissipating heat from the heater section to the fluid-solid contact surface where bubble nucleation had occurred. Hence, a higher bubble frequency should be generated from the “Pocofoam” of 75% porosity. However, the experimental results show that “Kfoam” of 78% porosity showed higher bubble departure frequency and higher boiling performance compared to “Pocofoam” of 75% porosity.

With similar pore structures, the main morphological difference between “Pocofoam” 75% and “Kfoam” 78% is the pore diameter d_p . The pore diameter would affect the bubble departure mechanism and boiling heat transfer

significantly. The effect of d_P and other foam parameters on the bubble departure frequency and nucleation site density are studied and presented in detail in Chapter 7. In this case, the larger potential nucleation site density of “Pocofoam” of 75% porosity competes with the superior bubble departure frequency from the “Kfoam” of 78% porosity and the results show that the advantage of higher bubble departure frequency from the “Kfoam” of 78% porosity resulted in the latter’s superior boiling performance.

4.2 EFFECT OF WORKING FLUID TYPE

To investigate the effects of working fluid properties on the boiling heat transfer performance, three different dielectric liquids *viz.* FC-72, HFE-7000, and HFE-7100 were tested in the experiments. The information and thermophysical properties of the working fluids are presented in Section 3.4.2 of Chapter 3. Boiling heat transfer performances from different working fluids were evaluated for “Pocofoam” of 61% porosity and “Kfoam” of 78% porosity graphite foams under saturated pool boiling condition. The dimensions of the graphite foam inserts are shown in Fig. 4-1. To evaluate the boiling heat transfer of different working fluid types, wall temperature readings and boiling curves were constructed. The wall temperature readings and boiling curves for different coolant types with “Pocofoam” of 61% porosity and “Kfoam” of 78% porosity are shown in Figs. 4-7(a) and 4-7(b), respectively.

(a)

(b)

Fig. 4-7 (a) Wall temperatures readings and (b) Boiling curves for different working fluid types with “Pocof foam” of 61% porosity.

Fig. 4-8 (a) Wall temperatures readings and (b) Boiling curves for different working fluid types with “Pocofoam” of 78% porosity.

Figures 4-7 and 4-8 show that the wall temperatures obtained by using HFE-7000 are much lower than those of FC-72 and HFE-7100 for all tested graphite foams and heat fluxes. The results show that the wall temperature differences between FC-72 and HFE-7000 with “Pocofoam” of 61% porosity and “Kfoam” of 78% porosity are 8.7°C and 7.1°C, respectively while those between FC-72

and HFE-7000 are about 33.2°C and 33.4°C for “Pocofoam” 61% and “Kfoam” 78%, respectively. The lower wall temperature obtained with HFE-7000 coolant can be explained by its lower saturation temperature as compared to FC-72 and HFE-7100 at corresponding pressure. The saturation pressure and temperature relationship provided by the coolant manufacturer are given as Eqs. (3-10) to (3-12).

The wall superheat readings for different coolant types with “Pocofoam” of 61% and 78% porosity are shown in Figs. 4-7(b) and 4-8(b), respectively. As shown in Figs. 4-7(b) and 4-8(b), the wall superheats of FC-72 are lower than those of HFE-7000 and HFE-7100 for all the tested foams and heat fluxes. As defined in Eq. (4-1), a lower wall superheat will lead to a higher boiling heat transfer coefficient at a given heat flux. The results show that the boiling heat transfer ratio by using FC-72 to HFE-7000 coolants for “Pocofoam” 61% and “Pocofoam” 78% are about 1.19 and 1.25, respectively while, the boiling heat transfer ratio performed by FC-72 to HFE-7100 coolants are found to be 1.37 and 1.29 with “Pocofoam” 61% and “Kfoam” 78%, respectively. It is shown clearly that FC-72 coolant have performed superior boiling heat transfer performance as compared to HFE-7000 and HFE-7100 coolants for both tested graphite foams.

To analyse and explain the difference in the thermal performance of the different coolants, a discussion of pertinent dimensionless parameters is presented here. The dimensionless parameters considered are the Bond number (Bo), ratio of the buoyancy force to surface tension force; Grashof number (Gr),

ratio of the buoyancy force to viscous force; and capillary number (Ca), ratio of the viscous force to surface tension force. Non-dimensional number Bo, Gr, and Ca are defined, respectively, as

$$Bo = \frac{g(\rho_l - \rho_v)d_p^2}{\sigma} \quad (4-8)$$

$$Gr = \frac{g\beta(T_w - T_{sat})d_p^3}{\nu^2} \quad (4-9)$$

$$Ca = \frac{\mu u_l}{\sigma} \quad (4-10)$$

As shown in Eqs. (4-8) ~ (4-10), the Bond number can be calculated based on working fluid properties and the pore diameter of foam. To calculate the Grashof number, superheat $T_w - T_{sat}$ can be determined from the experimental data. For the calculation of Ca, the liquid velocity u_l can be obtained from

$$u_l = \frac{q}{\rho_l A_h h_{fg}} \quad (4-11)$$

where A_h is the heated area and h_{fg} is the enthalpy of vaporization. Bo, Ca and Gr for the cases of “Pocofoam” 61% porosity and “Kfoam” 78% porosity in FC-72 and HFE-7000 at heat flux of 84.5 W/cm² are presented in Table 4-5.

The results show that a large Gr and small Ca indicate that the dynamic viscosity of the fluid is less significant as compared to buoyancy and surface tension forces. It is found that the Bond number of FC-72 is higher than HFE-7000 and HFE-7100 for both types of foams, which implies that the bubbles

need to overcome a larger surface tension force before departing the foam wall in HFE-7000 and HFE-7100. A lower surface tension corresponds to higher bubble frequency, which leads to better boiling heat transfer performance.

**Table 4-5 Dimensionless numbers for FC-72, HFE-7000 and HFE-7100 at
 $q'' = 84.5 \text{ W/cm}^2$**

Parameters	“Pocofoam” 61%			“Kfoam” 78%		
	FC-72	HFE-7000	HFE-7100	FC-72	HFE-7000	HFE-7100
Bo	0.17	0.12	0.10	0.66	0.45	0.32
Gr	79.5	170.3	215.6	822.5	998.6	1250.8
Ca	2.86×10^{-5}	1.25×10^{-5}	1.12×10^{-5}	2.95×10^{-5}	1.15×10^{-5}	0.96×10^{-5}

As discussed previously, the use of HFE-7000 results in lower wall temperatures than FC-72. The compromise between the lower wall temperatures obtained using HFE-7000 and better boiling heat transfer with FC-72 indicates that the selection of phase change coolant can be considered to have met the specific requirements for actual applications.

4.3 THE EFFECT OF WORKING FLUID LEVEL

Boiling heat transfer experiments with different working fluid levels were conducted in the study. Three working fluid levels of 1.0, 1.5 and 2.0 cm were tested by using “Pocofoam” of 61% porosity graphite foam with FC-72 and HFE-7000 coolants. The fluid level is defined as the distance from the top of

the graphite foam insert to the top fluid surface, where the graphite foam insert is fully covered by working fluid for all cases in the experiments. The wall temperature readings and boiling curves from different working fluid levels with “Pocofoam” 61% with FC-72 and HFE-7000 coolants are shown in Figs. 4.9 and 4.10, respectively.

Fig. 4-9 (a) Wall temperatures readings and (b) Boiling curves for different working fluid levels with “Pocofoam” of 61% porosity and FC-72 coolant

(a)

(b)

Fig. 4-10 (a) Wall temperatures readings and (b) Boiling curves for different working fluid levels with “Pocofoam” of 61% porosity and HFE-7000 coolant

As shown in Figs. 4-9 and 4-10, the liquid level has no significant effect on the heater wall temperature and boiling performance for both coolants. Figures 4-9(a) and 4-10(a) show that the working fluid level of 2.0 cm resulted in slightly lower wall temperatures about 0.2 ~ 0.3°C and 0.3 ~ 0.4°C as compared to

those of fluid level of 1.5 cm and 1.0 cm, respectively for both FC-72 and HFE-7000 coolants.

The readings of wall superheat for different working fluid levels for FC-72 and HFE-7000 with “Pocofoam” 61% are shown in Figs. 4-9(b) and 4-10(b), respectively. The results show that the wall superheats of $V_l = 2.0$ cm are lower slightly than those of $V_l = 1.5$ cm and $V_l = 1.0$ cm. As defined in Eq. (4-1), a lower wall superheat will lead to a higher boiling heat transfer coefficient at a given heat flux. It was found that the boiling heat transfer coefficients obtained for a working fluid level of 2.0 cm are slightly higher than those of $V_l = 1.5$ cm and $V_l = 1.0$ cm by about 1% and 2%, respectively for both coolants.

It can be concluded that the working fluid level (e.g. filling volume of coolant) has negligible effect on the boiling heat transfer performance. It was observed that the evaporative and condensate rates from all tested working fluid levels were found to be approximately equal during the boiling process. In the application, as long as the porous graphite foam inserts are immersed fully in the working fluid, the addition of coolant would not enhance the cooling performance significantly. Therefore, an optimum working fluid volume should be considered.

4.4 SUMMARY

Boiling heat transfer performance and characteristics from the porous graphite foam with different properties, working fluid properties and working fluid levels are presented and discussed in this chapter. The wall temperature

readings and boiling curves were used to evaluate the boiling heat transfer coefficient. Nucleation site density and bubble departure frequency from the porous foam structure were evaluated to analyse the boiling performance. Bubble departure frequencies were calculated from the boiling images captured by a high speed camera. The potential active nucleation site density was predicted by calculating the internal surface area to volume ratio (β) from the porous graphite foams.

Three working fluids with different thermophysical properties *viz.* FC-72, HFE-7000 and HFE-7100 were investigated for their boiling heat transfer performance. It was found that the use of FC-72 resulted in higher boiling heat transfer coefficient than HFE-7000 and HFE-7100. The dimensionless parameters of Bond number (Bo), Grashof number (Gr), and capillary number (Ca) were analysed to evaluate the boiling heat transfer performance from different working fluids. The effects of working fluid level on the boiling heat transfer are discussed in this chapter. Three working fluids levels of 1.0 cm, 1.5 cm and 2.0 cm were investigated. The results show that working fluid level has insignificant effect on boiling heat transfer performance as long as the graphite foam inserts are fully immersed in the working fluids. Finally, this chapter provides important guidelines on how to select the graphite foam type, working fluid type and working fluid level in cooling system applications.

CHAPTER 5 POOL BOILING CHARACTERISTICS FROM POROUS GRAPHITE FOAM STRUCTURES – PART 2: THE ROLE OF PORE STRUCTURE ON BOILING HEAT TRANSFER ENHANCEMENT

5.1 POOL BOILING HEAT TRANSFER FROM BLOCK AND FINS STRUCTURES

As mentioned in Chapter 2, the use of porous graphite foams has enhanced significantly the boiling heat transfer performance. The larger exposed surface area and high thermal conductivity of the foam were identified as the main reasons for the enhancement. It is also shown that the graphite foam possesses interconnected pores that allow the generated bubbles to escape from the pore structures. It is therefore important to investigate the role played by the internal pore structure on boiling heat transfer performance. In this study, block and fin evaporators of different fin-to-block-surface-area ratios (AR) were tested in the air-cooled thermosyphon described in Section 3.1 of Chapter 3 under saturated pool boiling condition with FC-72 and HFE-7000 as dielectric working fluids. The schematic diagram of the experimental setup is shown in Fig. 3-1.

To analyse the boiling performance and enhancement mechanism, the wall temperatures were measured and the boiling heat transfer coefficients were calculated. A visualisation study of the bubble phenomena from the block and

fin structures was conducted to support the analysis. The results of this work have been published in Pranoto *et al.* (2012).

5.1.1 Design of the Graphite Foam Evaporator Inserts

In this study, two types of graphite foams *viz.* “Pocofoam” of 61% porosity and “Kfoam” of 78% porosity were used as the evaporator inserts. The typical internal structures of the graphite foams observed under a Scanning Electron Microscope (SEM) are shown in Fig. 3-19 of Chapter 3. Selected properties of the graphite foams are listed in Table 3-1 of Chapter 3.

Two different evaporator structures were tested in the experiments. Block and fin structures were fabricated using a wire cut electron discharge machine (EDM). The structures, dimensions and photographs of the tested porous evaporator of the block and fin structures with $AR = 3.70$ and 2.73 are shown in Figs. 5-1 and 5-2, respectively. The samples were designed with different fin-to-block-surface area ratios (AR) which is defined by Eq. (5-1). Based on these structures and dimensions, the internal and external surface areas were calculated. The total internal surface area to volume ratio (β), total external exposed area to flat surface ratio (S_F), and total exposed area (A_T) of graphite foam were determined from the foam samples.

$$AR = \frac{A_F}{A_B} \quad (5-1)$$

The values of β and S_F for different graphite foam types were calculated based on the model of Yu *et al.* (2006). In order to evaluate the potential nucleation

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site density of a porous graphite foam, the total internal surface area (A_{Int}) and external surface area (A_{Ext}) were determined. The total exposed surface area of graphite foam can be calculated as

$$A_T = A_{Int} + A_{Ext} \quad (5-2)$$

and the ratio of total exposed areas of the block and fin structures (ER) is expressed as

$$ER = \frac{A_T(B)}{A_T(F)} \quad (5-3)$$

In Yu *et al.*'s model (2006), β and S_F are expressed as

$$\beta = \frac{\pi D}{H^3} (3H - 2D) \quad (5-4)$$

$$S_F = \frac{2 \left(A_{avg1} c + A_{avg2} \left(\frac{H}{2} - c \right) \right)}{H} + \frac{\pi D^2 - 6\pi D h}{2} \quad (5-5)$$

The total surface area of the foams can be determined from the knowledge of the physical properties of the foam samples *viz.* pore diameter (D), porosity (ϵ), density (ρ), and interconnected pore diameter (d). A summary of the calculated results for the tested graphite foams of $AR = 3.70$ and 2.73 are given in Tables 5-1 and 5-2, respectively.

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Fig. 5-1 Dimensions of the block and fin structures of the graphite foam for $AR = 3.70$.

Table 5-1 Calculated parameters of the “Pocofoam” of 61% porosity and “Kfoam” of 78% porosity graphite foams for $AR = 3.70$.

Parameters	“Pocofoam” 61%		“Kfoam” 78%	
	Block	Fins	Block	Fins
$\beta \text{ [m}^2\text{/m}^3\text{]}$	8885		5569	
S_F	1.72		1.42	
$A_{Int} \text{ [m}^2\text{]}$	239.9	142.9	150.4	89.5
$A_{Ext} \text{ [m}^2\text{]}$	0.0077	0.029	0.0064	0.024
$A_T \text{ [m}^2\text{]}$	239.9	142.9	150.4	89.6
ER	1.68		1.68	

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Fig. 5-2 Dimensions of graphite foam of block and fin structures for $AR = 2.73$.

Table 5-2 Calculated parameters of “Pocofoam” of 61% porosity and “Kfoam” of 78% porosity graphite foams for $AR = 2.73$

Parameters	“Pocofoam” 61%		“Kfoam” 78%	
	Block	Fins	Block	Fins
$\beta \text{ [m}^2\text{/m}^3\text{]}$	8885		5569	
S_F	1.72		1.42	
$A_{Int} \text{ [m}^2\text{]}$	205.4	142.2	128.7	89.0
$A_{Ext} \text{ [m}^2\text{]}$	0.0066	0.018	0.0055	0.015
$A_T \text{ [m}^2\text{]}$	205.4	142.12	128.8	89.1
ER	1.44		1.44	

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Pertinent thermophysical properties of FC-72 and HFE-7000 at 25°C and 1 atm are shown in Table 3-2 of Chapter 3. The vapour pressure readings of FC-72 and HFE-7000 with “Pocofoam” 61% porosity evaporator for different evaporator structures are shown in Fig. 5-3.

(a)

(b)

Fig. 5-3 Vapour pressure readings of FC-72 and HFE-7000 with “Pocofoam” of 61% porosity graphite foam for (a) AR = 2.73 and (b) AR = 3.70.

5.1.2 Wall Temperature Reading and Boiling Curve from Block and Fins Structure

As mentioned in the previous section, the block and fin evaporators with AR of 3.70 and 2.73 were tested with FC-72 and HFE-7000. Figure 5-4 shows the wall temperatures of the “Pocofoam” 61% porosity foam tested with FC-72 and HFE-7000, respectively.

(a)

(b)

Fig. 5-4 Wall temperature variation with heat flux for “Pocofoam” of 61% porosity graphite foam in FC-72 and HFE-7000 for different block and fin structures.

As shown in Fig. 5-4, all tested block evaporators exhibit lower wall temperatures compared with the fin evaporator for all heat fluxes. The results show that the wall temperature difference between the block and fin structures becomes more significant for heat fluxes above 30 W/cm^2 which is in the nucleate boiling regime. It was found that the average wall temperature differences between the block and fin structures for FC-72 and HFE-7000 are 8 and 10°C for $AR = 3.70$ and 2.73 , respectively.

Figure 5-5 shows the wall temperatures of the block and fin structures of “Kfoam” 78% porosity graphite foam tested with FC-72 and HFE-7000, respectively. It can be seen that the block structures resulted in lower wall temperatures for both coolants. The average wall temperature differences between the block and fin structures are found to be about 8 and 10°C for $AR = 3.70$ and 2.73 , respectively. All these give a clear indication that the fin structures with lower total exposed surface areas resulted in higher wall temperatures in all the tests. A more detailed analysis on the boiling mechanism will be presented in Section 5.1.3.

From the same tests, the boiling curves for the tested samples with FC-72 and HFE-7000 are presented in Figs. 5-6 and 5-7. The saturation temperatures of FC-72 and HFE-7000 were calculated from Eqs. (3-10) and (3-11) which are based on the corresponding vapour pressure measured by the pressure transducer. Figures 5-6 and 5-7 clearly show that at the same heat flux, all block structures exhibit lower superheats than the fin structures in FC-72 and HFE-7000.

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Fig. 5-5 Wall temperature variation with heat flux for “Kfoam” of 78% porosity graphite foam in FC-72 and HFE-7000 for different block and fin structures.

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(a)

(b)

Fig. 5-6 Boiling curves of “Pocofoam” 61% porosity graphite foam in FC-72 and HFE-7000 for different block and fin structures.

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(a)

(b)

Fig. 5-7 Boiling curves of “Kfoam” 78% porosity graphite foam in FC-72 and HFE-7000 for different block and fin structures.

The boiling heat transfer coefficients (h_b) and heat flux (q'') were calculated using Eqs. (5-6) and (5-7), respectively. Based on this equation and for the same heat flux, a lower superheat will correspond to higher h_b . Hence, the

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boiling curves obviously show that the block structures performed better than the fin structures.

$$h_b = \frac{q''}{(T_w - T_{sat})} \quad (5-6)$$

$$q'' = \frac{q}{A_h} \quad (5-7)$$

The average boiling heat transfer coefficients ($\bar{h}_b$) were calculated and presented in Tables 5-3 and 5-4. By combining the results of wall temperatures and superheats, it can be deduced that the designed fin structures resulted in poorer boiling performance and higher wall temperatures which are undesirable for a cooling system.

Table 5-3 Average boiling heat transfer coefficient and its ratio for $AR = 3.70$

Parameters		“Pocofoam” 61%		“Kfoam” 78%	
		Block	Fins	Block	Fins
$\bar{h}_b$ [kW/m ² ·K]	FC-72	41.9	34.8	36.8	23.5
	HFE-7000	36.0	28.3	29.9	22.9
BR	FC-72	1.2		1.6	
	HFE-7000	1.3		1.3	

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Table 5-4 Average boiling heat transfer coefficient and its ratio for $AR = 2.73$

Parameters		“Pocofoam” 61%		“Kfoam” 78%	
		Block	Fins	Block	Fins
$\bar{h}_b$ [kW/m ² ·K]	FC-72	41.3	33.8	35.7	24.6
	HFE-7000	35.8	27.9	29.5	21.6
BR	FC-72	1.2		1.5	
	HFE-7000	1.3		1.4	

As presented in Table 5-3, $\bar{h}_b$ of the “Pocofoam” 61% porosity foam with FC-72 for the block and fin evaporators are 41.9 and 34.8 kW/m²·K, respectively while for HFE-7000, $\bar{h}_b$ for the block and fin evaporators are 36.0 and 28.3 kW/m²·K, respectively. Lower boiling heat transfer coefficients were obtained for the “Kfoam” 78% porosity evaporator. $\bar{h}_b$ of the block and fin evaporators with FC-72 as the coolant are 36.8 and 23.5 kW/m²·K, respectively while by using HFE-7000, $\bar{h}_b$ of the block and fin structures are 29.9 and 22.9 kW/m²·K, respectively. In order to quantify the difference in the boiling heat transfer performance between the tested block and fins structures, the ratio of average boiling heat transfer coefficient of block-to-fin structures (BR) as expressed by Eq. (5-8) are presented in Tables 5-3 and 5-4.

$$BR = \frac{\bar{h}_{b(B)}}{\bar{h}_{b(F)}} \quad (5-8)$$

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It can be seen that the block structures result in boiling heat transfer coefficients which are about 1.2 ~ 1.6 times higher than the fin structures for all tested foams and coolants.

Another important point to highlight is the effect of thermophysical properties of the coolant on the wall temperatures and boiling heat transfer coefficients. As shown in Figs. 5-4 and 5-5, the use of HFE-7000 always resulted in lower wall temperatures than FC-72. The difference of wall temperatures between the two coolants is due to the difference in the boiling points of the liquids. From Eqs. (3-10) and (3-11) in Chapter 3, it can be seen that the saturation temperatures of HFE-7000 are lower than those of FC-72 at the same pressure. The experimental results have also shown that the use of FC-72 resulted in higher boiling heat transfer coefficients compared to HFE-7000.

The saturated boiling phenomena of FC-72 and HFE-7000 were analysed using the dimensionless parameters of Bond, Grashof, and Capillary numbers as defined in Eqs. (4-5) to (4-7) and discussed in Chapter 4. The experimental data of the block and fin structures of “Pocofoam” 61% porosity and “Kfoam” 78% porosity foams in FC-72 and HFE-7000 at the heat flux of 70.42 W/cm^2 were used to calculate Bo, Ca and Gr. The results are presented in Tables 5-5 and 5-6. The calculations of Bo, Gr and Ca are discussed in greater detail in Chapter 4. Based on the calculated dimensionless numbers, it was found that the capillary force was less significant compared to the buoyancy and surface tension forces. The results show that HFE-7000 has lower Bond numbers compared with FC-72 for all graphite foam types and structures.

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Table 5-5 Calculated dimensionless numbers for block structure ($AR = 3.70$) at $q'' = 70.4 \text{ W/cm}^2$

Parameters	“Pocofoam” 61%		“Kfoam” 78%	
	FC-72	HFE-7000	FC-72	HFE-7000
Bo	0.17	0.14	0.65	0.46
Gr	82.6	174.6	822.5	998.6
Ca	2.94×10^{-5}	1.16×10^{-5}	2.95×10^{-5}	1.15×10^{-5}

Table 5-6 Calculated dimensionless numbers for fin structure ($AR = 3.70$) at $q'' = 70.4 \text{ W/cm}^2$

Parameters	“Pocofoam” 61%		“Kfoam” 78%	
	FC-72	HFE-7000	FC-72	HFE-7000
Bo	0.20	0.15	0.75	0.52
Gr	86.5	176.4	825.4	998.8
Ca	2.98×10^{-5}	1.18×10^{-5}	2.97×10^{-5}	1.17×10^{-5}

This indicates that during the bubble growth and departure process with HFE-7000, the bubble has to overcome a larger surface tension than with FC-72 which will result in both lower boiling departure frequency and poorer boiling performance.

5.1.3 Boiling Performance on the Structured Porous Graphite Foam

The wall temperatures and boiling curves obtained for the designed block and fin structures are presented and discussed in Section 5.1.2. This section will discuss how the porous graphite foam evaporator structures significantly affect the boiling heat transfer performance. The superior boiling performance of the block over the fin structures show that the internal pore structure of the graphite

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foam plays an important role in the boiling process and performance. In a porous graphite foam evaporator, both the internal and external surfaces can generate nucleation sites and affect the boiling performance. As presented in Tables 5-1 and 5-2, the fin structures have much larger external exposed surface areas than the block structures ($AR = 3.70$ and 2.73). However, the calculation results show that the total exposed areas of the block structures are about 1.68 and 1.44 times larger than the fin structures for the “Pocofoam” 61% and “Kfoam” 78% porosity foams, respectively.

During the boiling process, the interconnected pores in the graphite foams allow the coolant to permeate through the internal pore structure and will also allow the bubbles to escape from the porous evaporator. Thus, the entire pore structure contains active nucleation sites for generation of bubbles compared to only the voids of the surface for conventional heater surfaces. The fin structures with a smaller value of A_T will therefore have fewer active nucleation sites than the block structure with a larger A_T .

The results of this study also explained how the internal structure of graphite foam significantly affects the boiling performance which is caused by the larger boiling total exposed area. The calculated results show that the “Pocofoam” 61% porosity foam possesses a much larger β than the “Kfoam” 78% porosity foam as shown in Table 4-4 of Chapter 4. The larger β values of the “Pocofoam” 61% porosity foam coupled with its excellent thermophysical properties resulted in its better boiling performance.

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In this study, the boiling processes were captured by a high-speed camera and the images were used to analyse the boiling phenomena from the structured graphite foam evaporators. Figure 5-8 shows the captured images of the nucleation processes from the block and fin evaporators under saturated pool boiling condition with “Pocofoam” 61% porosity foam and FC-72. Based on the observed boiling process, nucleation bubbles were seen to be generated after the heat fluxes have reached 42.3 and 26.2 W/cm^2 , respectively for FC-72 and HFE-7000. From Fig. 5-8, bubble generation from the structured graphite foam evaporators subjected to different heat fluxes can be observed. Isolated bubbles from the block and fin structures can only be identified for heat fluxes below 70.4 W/cm^2 [Figs. 5-8(a) – 5-8(f)].

The comparison of the bubble densities for the block and fin structures indicates that significantly more bubbles were generated from the block structure for all heat flux ranges. This supports the evidence that the block structure had more nucleation sites compared to the fin structure as discussed previously in this section.

Fig. 5-8 Captured boiling images of block and fin structure with “Pocofoam” 61% porosity graphite foam evaporator with $AR = 2.73$ and FC-72 for different heat fluxes.

5.2 PREDICTION OF POOL BOILING HEAT TRANSFER FROM POROUS GRAPHITE FOAM STRUCTURES

In general, the nucleate pool boiling heat transfer coefficient is proportional to the heat flux or a non-dimensional number containing the heat flux term raised to the power of about $2/3$ (Pioro *et al.*, 2004). Based on a detailed comparison of the prediction accuracies of various boiling correlations, the Rohsenow correlation with the constants and powers for the Prandtl number depending on a surface-fluid combination parameter (C_{sf}) obtained from experimental data, is found to be the most accurate (Pioro *et al.*, 2004). Considering the present case, the boiling of dielectric liquid is not only from the surface of the foam, but

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also from the internal pore walls due the interconnected open pores. To obtain an accurate prediction of the boiling heat transfer coefficient from the porous graphite foam, the original Rohsenow correlation

$$\frac{c_{pl}}{h_{fg}}(T_w - T_{sat}) = C_{sf} \left[\frac{q''}{u_l h_{fg}} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right]^{0.33} \left(\frac{c_{pl} u_l}{k_l} \right)^n \quad (5-9)$$

is modified to

$$\frac{c_{pl}}{h_{fg}}(T_w - T_{sat}) = C_{sf} \gamma^m \left[\frac{q''}{u_l h_{fg}} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right]^{0.33} \left(\frac{c_{pl} u_l}{k_l} \right)^n \quad (5-10)$$

by adding a correction term to account for the equivalent boiling surface area γ (Ng *et al.*, 2006), which is defined as the ratio of the total wetted surface to the heated area. The definitions and calculations of pressure and temperature dependent parameters in Eq. (5-10) are given in Table 5-7. The constants C_{sf} , m and n are obtained by fitting the correlation to experimental data. For porous graphite foams, the specific internal surface area ϕ (surface area to volume ratio) is given by Yu *et al.* (2006) as

$$\phi = \frac{\pi d_p}{h^3} (3h - 2d_p) \quad (5-11)$$

where d_p is the pore diameter and h is the characteristic length of the void unit. Therefore, for a graphite foam of volume V , the total wetted surface area is ϕV and the equivalent boiling surface area γ can then be calculated as

$$\gamma = \frac{\phi V}{A_h} \quad (5-12)$$

where A_h is the heated surface of graphite foam. All the measured liquid and vapour temperatures and saturation pressure along with the coolant properties at various test conditions are used to calculate the parameters in Eq. (5-10). The data for graphite foams boiling in FC-72 and HFE-7000 are plotted in Figs. 5-9 and 5-10, respectively. The x axis represents the surface-fluid and Prandtl number combination, while the y axis represents the combination of the boiling heat transfer coefficient and the interfacial velocity based Reynolds number. Using the Quasi-Newton nonlinear optimisation algorithm (Schoenberg, 2001), the constants C_{sf} , m , and n were determined and the two correlations are expressed as follows:-

for FC-72,

$$\frac{c_{pl}}{h_{fg}}(T_w - T_{sat}) = 0.00013\gamma^{-0.54} \left[\frac{q''}{u_l h_{fg}} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right]^{0.33} \left(\frac{c_{pl} u_l}{k_l} \right)^{4.1} \quad (5-13)$$

and for HFE-7000,

$$\frac{c_{pl}}{h_{fg}}(T_w - T_{sat}) = 0.00013\gamma^{-0.32} \left[\frac{q''}{u_l h_{fg}} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right]^{0.33} \left(\frac{c_{pl} u_l}{k_l} \right)^{4.26} \quad (5-14)$$

Comparing the two equations, it can be seen that the effective boiling surface area affects the thermal performance significantly. The prediction errors $\left(\frac{Pred - Exp}{Exp} \right)$ of wall superheats based on the experimental data obtained for

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FC-72 and HFE-7000 are presented in Figs. 5-11 and 5-12.

Table 5-7 Determination of variables in Eqs. (5-13) and (5.14)

Variables	FC-72	HFE-7000
ρ_l (kg/m ³)	$1740 - 2.61T_l$	$1472 - 2.88T_l$
ρ_v (kg/m ³)	13.6	8.1
T_{sat} (°C)	$\log_{10} P_{sat} = 9.729 - \frac{1562}{T_{sat}}$	$\log_{10} P_{sat} = 22.978 - \frac{3549}{T_{sat}}$
σ (N/m)	0.01	0.012
μ (kg/m·s)	$\rho_l(-0.04T_l + 5.2)10^{-7}$	$\rho_l(-0.03T_l + 3.8)10^{-7}$
u_l (m/s)	$u_l = \frac{Q}{\rho_l A_f h_{fg}}$	
h_{fg} (J/kg)	8.8×10^4	1.42×10^5

Fig. 5-9 Correlation of FC-72 boiling from porous graphite foams.

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Fig. 5-10 Correlation of HFE-7000 boiling from porous graphite foam.

Fig. 5-11 Prediction errors of superheat of using FC-72 with graphite foams.

Fig. 5-12 Prediction errors of superheat of using HFE-7000 with graphite foams.

It is found that the maximum errors for predicting the wall superheat $T_w - T_{sat}$ in FC-72 and HFE-7000 are $\pm 35\%$ and $\pm 25\%$, respectively. According to a comprehensive review of various correlations of pool boiling heat transfer by Piro *et al.* (2004), an error within $\pm 40\%$ can be considered as acceptable due to the complexity of the boiling process. Therefore, the current correlations can be used to predict the performance of the porous thermosyphons in the development phase with reasonable accuracy, thus reducing the duration of the product design cycle.

5.3 SUMMARY

The experimental results show that the graphite foam structures significantly affect the boiling performance. It was found that the wall temperatures of the

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block structure are about 8 and 10 °C lower than the fin structure for $AR = 3.70$ and 2.73, respectively. For all the tested foams, coolants, and structures of different aspect ratios, the average boiling heat transfer coefficients of the block structure are about 1.2 ~ 1.6 times higher than those of the fin structure for the tested heat flux level of up to 112 W/cm². A maximum average boiling heat transfer coefficient of 41.9 kW/m²·K can be achieved in this study using “Pocofoam” 61% porosity graphite foam evaporator in FC-72 coolant. Furthermore, the poorer boiling performance from the tested fin structures indicates the important role of the internal pore structure of the porous graphite foams on the enhancement of boiling performance. Larger values of A_T and β from the block structures imply more nucleation sites and hence, enhanced boiling performance of these structures. The calculated values of Bo , Ca , and Gr indicate that the large ratio of buoyancy to surface tension force of FC-72 promotes bubble growth and departure resulting in higher bubble departure frequency compared to those of HFE-7000. From this study, it can be concluded that the internal pores of a graphite foam contributed significantly to a larger active nucleation site density and enhanced boiling heat transfer performance. Hence, careful selection of the evaporator structure and the type of foam and coolant in terms of their thermophysical properties is extremely important to achieve enhanced boiling heat transfer performance.

Pool boiling heat transfer prediction models from the porous graphite foams “Pocofoam” 61%, “Pocofoam” 75%, “Kfoam” 72%, and “Kfoam” 78% with FC-72 and HFE-7000 were proposed and compared to the experimental data.

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The proposed pool boiling heat transfer models were developed from the modified Rohsenow's model by accounting for the equivalent boiling surface area which is the ratio of the total wetted surface to the heated area of the porous graphite foams. The proposed models are expressed as Eqs. (5-13) and (5-14) for FC-72 and HFE-7000 coolants, respectively. It was found that the maximum errors for the wall superheat ($T_w - T_{sat}$) prediction are $\pm 35\%$ and $\pm 25\%$ for FC-72 and HFE-7000, respectively.

CHAPTER 6 FLOW BOILING CHARACTERISTICS FROM POROUS GRAPHITE FOAM STRUCTURES

Flow boiling heat transfer performance and bubble flow characteristics from experiments conducted on porous graphite foam structures are discussed in this chapter. The effect of graphite foam types, graphite foam height, and coolant mass fluxes were investigated in a single flow boiling cooling facilities. The boiling phenomena and bubble flow phenomena were observed and analysed by using a high-speed camera and images analysis. The results of this work have been published in Pranoto and Leong (2014).

6.1. GRAPHITE FOAM INSERTS AND WORKING FLUIDS

To enhance the boiling heat transfer performance of the system, two types of graphite foams *viz.* “Pocofoam” of 61% porosity and “Kfoam” of 72% porosity were used as evaporator inserts. These foams of different heights were fabricated by Electrical Discharge Machining (EDM) to study the effect of foam height on the system performance. The samples were fabricated into block structures with dimensions of 80 mm (L) $\times$ 60 mm (W) $\times$ 6 mm (H). Scanning Electron Microscope (SEM) images of internal structure of the graphite foams are given in Fig. 3-19 of Chapter 3. The thermophysical properties of the graphite foams are provided in Table 3-1 in Chapter 3. In all the experiments, there will always be a gap in the channel. It is important to note that with “Pocofoam” of 61% porosity and “Kfoam” of 72% porosity, the coolant flows predominantly through the evaporator gaps and any flow through

the foams is considered to be negligible. In this study, FC-72 was selected as the phase change coolant used in the experiments. The properties of FC-72 working fluid are shown in Table 3-2 of Chapter 3.

6.2 EXPERIMENTAL PARAMETERS

To explore the flow boiling performance and phenomena from the graphite foam, three parameters, *viz.* coolant mass flux (G), evaporator gap (d), and foam properties (*i.e.* porosity and pore diameter) were considered. In the experiments, different mass fluxes were obtained by adjusting the speed of the micro pump according to the respective free flow area of the channel (A_{ff}). The evaporator gap is defined as the distance from the top of the porous insert to the evaporator top surface as illustrated in Fig. 6-1. Evaporator gaps of 2 mm, 4 mm, and 6 mm were investigated in this study. These were adjusted by changing the thicknesses of the interface material attached on the top cover of the channel. In the study, experiments on the aluminium smooth surface (*i.e.* evaporator channel without any graphite foam insert) were performed as a base case to evaluate the heat transfer enhancement of the porous channel (*i.e.* evaporator channel with the graphite foam inserts).

The parameters tested in the experiments are summarised in Table 6-1. The wall temperatures T_w were evaluated as the average of eight temperature readings obtained by thermocouples placed at different locations on the heater surface and is given by

$$T_w = \frac{T_{w1} + T_{w2} + T_{w3} + T_{w4} + T_{w5} + T_{w6} + T_{w7} + T_{w8}}{8} \quad (6-1)$$

The schematic of wall temperature thermocouple arrangement on the evaporator is presented in Fig. 6-2.

Table 6-1 Experimental parameters

Parameters	Value
Evaporator gap (d)	d_1 6 mm
	d_2 4 mm
	d_3 2 mm
Coolant mass flux (G)	G_1 50 kg/m ² ·s
	G_2 100 kg/m ² ·s
	G_3 150 kg/m ² ·s
Graphite foam type	“Pocofoam” 61% porosity
	“Kfoam” 72% porosity

Fig. 6-1 Schematic of the test section and the evaporator gap.

Fig. 6-2 Schematic of the heat simulator and wall temperature thermocouple arrangements.

6.3 WALL TEMPERATURE READINGS AND BOILING CURVES

The cooling performances of the pumped two-phase cooling system were analysed based on the flow boiling heat transfer coefficient. Many researchers have proposed various correlations to determine the flow boiling heat transfer coefficient. Chen (1966) proposed that the local flow boiling heat transfer coefficient can be expressed as the superposition of forced convection and nucleate boiling heat transfer coefficients given by

$$h_{fb} = Fh_{fc} + Sh_{nb} \quad (6-2)$$

Kandlikar (1990) and Kandlikar and Balasubramanian (2004) proposed generalised correlations of h_{fb} for both vertical and horizontal tubes in the form of

$$h_{fb} = \max[h_{fc}, h_{nb}] \quad (6-3)$$

In the current experiments, the heating powers were varied from 4.2 to 83.3 W/cm² at intervals of 4.2 W/cm². It was found that boiling incipience occurred for a heat flux of 20.8 W/cm². Therefore, it can be concluded that nucleate boiling heat transfer had dominated the heat transfer process and thus, the flow boiling heat transfer coefficient can be determined from

$$h_{fb} = \frac{q''}{(T_w - T_{sat})} \quad (6-4)$$

$$q'' = \frac{q}{A_h} \quad (6-5)$$

In this study, the saturation temperatures T_{sat} of FC-72 were calculated at their corresponding saturated pressures P_{sat} as shown in Eq. 3-10 of Chapter 3. The average values of the flow boiling heat transfer coefficients $h_{fb(avg)}$ were calculated to evaluate the boiling performance for different coolant mass fluxes, evaporator gaps, and graphite foam properties.

6.3.1 Effects of Coolant Mass Flux

Wall temperature readings and boiling curves derived from the experiments on “Pocofoam” of 61 % porosity and “Kfoam” of 72% porosity for various mass fluxes of 50, 100, and 150 kg/m²·s with evaporator gaps of 6 and 4 mm are presented in Figs. 6-3 and 6-4. As defined in Eq. (6-4), the flow boiling heat transfer coefficient h_{fb} is inversely proportional to the superheat $T_W - T_{sat}$. For constant heat flux q'' , a lower superheat corresponds to a higher h_{fb} . The experimental results show that the coolant mass flux affects the wall temperatures and flow boiling heat transfer coefficient for a given gap size. The results show that regardless of the evaporator gap and the graphite foam used, the highest coolant mass flux resulted in the lowest wall temperature and the smallest wall superheat at all tested heat fluxes. The higher the mass flux, the better is the flow boiling heat transfer performance. The average wall temperature differences $\overline{\Delta T_{W(G_1-G_x)}}$ between G_1 and various coolant mass fluxes; and the ratio of $h_{fb(avg)}$ values of the porous channels with various mass fluxes to that at G_1 defined as $h_{fbG_x(avg)} / h_{fbG_1(avg)}$ are presented in Table 6-2 and 6-3. G_x refers to the coolant mass flux which was varied from G_2 to G_3 .

(a)

(b)

(c)

(d)

Fig. 6-3 Wall temperature readings of different coolant mass fluxes at (a) $d = 6$ mm and (b) $d = 4$ mm; and (c) Boiling curves of different mass fluxes at (c) $d = 6$ mm and (b) $d = 4$ mm with “Pocofoam” of 61% porosity.

(a)

(b)

(c)

(d)

Fig. 6-4 Wall temperature readings of different coolant mass fluxes at (a) $d = 6$ mm and (b) $d = 4$ mm; and (c) Boiling curves of different mass fluxes at (c) $d = 6$ mm and (b) $d = 4$ mm with “Kfoam” of 72% porosity.

Table 6-2 Average wall temperature differences and average flow boiling heat transfer coefficient ratio between different coolant mass fluxes of with “Pocofoam” of 61% porosity at (a) $d = 6$ mm and (b) $d = 4$ mm

Wall temperature difference $\left(\overline{\Delta T_{W(G_1-G_x)}}\right)$	(°C)	Boiling heat transfer coefficient ratio $\left(h_{fb(G_x)} / h_{fb(G_1)}\right)$	
$\overline{\Delta T_{W(G_1-G_2)}}$	4.3	$h_{fb(G_2)} / h_{fb(G_1)}$	1.13
$\overline{\Delta T_{W(G_1-G_3)}}$	9.2	$h_{fb(G_3)} / h_{fb(G_1)}$	1.33
(a)			

Wall temperature difference $\left(\overline{\Delta T_{W(G_1-G_x)}}\right)$	(°C)	Boiling heat transfer coefficient ratio $\left(h_{fb(G_x)} / h_{fb(G_1)}\right)$	
$\overline{\Delta T_{W(G_1-G_2)}}$	4.2	$h_{fb(G_2)} / h_{fb(G_1)}$	1.12
$\overline{\Delta T_{W(G_1-G_3)}}$	9.6	$h_{fb(G_3)} / h_{fb(G_1)}$	1.34
(b)			

Table 6-3 Average wall temperature differences and average flow boiling heat transfer coefficient ratio between different coolant mass fluxes of with “Kfoam” of 72% porosity at (a) $d = 6$ and (b) $d = 4$ mm

Wall temperature difference $\left(\overline{\Delta T_{W(G_1-G_x)}}\right)$	(°C)	Boiling heat transfer coefficient ratio $\left(h_{fb(G_x)} / h_{fb(G_1)}\right)$	
$\overline{\Delta T_{W(G_1-G_2)}}$	4.1	$h_{fb(G_2)} / h_{fb(G_1)}$	1.13
$\overline{\Delta T_{W(G_1-G_3)}}$	8.6	$h_{fb(G_3)} / h_{fb(G_1)}$	1.22
(a)			

Wall temperature difference $\left(\overline{\Delta T_{W(G_1-G_x)}}\right)$	(°C)	Boiling heat transfer coefficient ratio $\left(h_{fb(G_x)} / h_{fb(G_1)}\right)$	
$\overline{\Delta T_{W(G_1-G_2)}}$	3.6	$h_{fb(G_2)} / h_{fb(G_1)}$	1.12
$\overline{\Delta T_{W(G_1-G_3)}}$	8.7	$h_{fb(G_3)} / h_{fb(G_1)}$	1.23

(b)

It is shown that by increasing the mass flux from 50 kg/m²·s to 100 and 150 kg/m²·s the average wall temperatures were reduced by up to 4.3°C and 9.6°C, respectively. The results also show that enhancements of $h_{fb(avg)}$ by up to 13% and 34% were obtained by increasing the coolant mass fluxes from 50 kg/m²·s to 100 and 150 kg/m²·s, respectively.

The boiling curves in Figs. 6-3(c) and 6-3(d) for “Pocofoam” foam of 61% porosity and Figs. 6-4(c) and 6-4(d) for “Kfoam” foam of 72% show that the coolant mass flux affects significantly the flow boiling heat transfer performance. The results show that at a given heat flux, the higher coolant mass fluxes had resulted in lower wall superheats for all evaporator gaps and heat fluxes. The values of $h_{fb(avg)}$ for coolant mass fluxes of 50, 100, and 150 kg/m²·s with “Pocofoam” foam of 61% porosity at $d = 6$ mm and $d = 4$ mm were found to be 12.5, 14.5, and 16.5 kW/m²·K; and 11.7, 13.2, and 15.7 kW/m²·K, respectively. $h_{fb(avg)}$ for different coolant mass fluxes of 50, 100, and 150 kg/m²·s with “Kfoam” of 72% porosity at $d = 6$ mm were found to be 10.8, 12.1, and 13.1 kW/m²·K while $h_{fb(avg)}$ for different mass fluxes of 50, 100, and 150 kg/m²·s with “Kfoam” of 72% porosity at $d = 4$ mm are 10.2, 11.4, and

12.4 kW/m²·K, respectively. The results also show that the enhancements of $h_{fb(avg)}$ due to the increase of the mass flux with $d = 6$ mm and $d = 4$ mm are not significantly different which suggests that the flow boiling enhancement mechanisms for both tested gaps might be the same.

In flow boiling systems, the mechanism of heat transfer is a combination of forced convection and nucleate boiling. In this study, the graphite foam is fully immersed in the coolant with the possibility of the liquid filling all the voids in the pore structure. The heat from the heater will be conducted through the heater wall and porous insert resulting in boiling of the liquid inside the pores. Subsequently, the supplied fresh coolant refills the pores and evaporating bubbles will be continuously generated. The coolant flows through the evaporator gaps enhancing the heat removal by forced convection and sweeping away the generated bubbles from the top surface of graphite foams. Therefore, higher coolant mass fluxes will increase the liquid replacement rate to the pores and increase the forced convection capacity resulting in an increase in the bubble departure frequency. The flow boiling mechanism for different coolant mass fluxes will be discussed in greater detail in Section 6.3.4.

6.3.2 Effect of Evaporator Gap

The effects of evaporator gap were investigated for “Pocofoam” of 61% porosity and “Kfoam” of 72% porosity with mass fluxes of 100 and 150 kg/m²·s. Wall temperature readings and boiling curves for different gaps tested with different graphite foam types and coolant mass fluxes are shown in Figs. 6-5 and 6-6. As expected, the experimental results show that the evaporator

gap affects the wall temperatures and flow boiling heat transfer coefficient. Subsequent decreases of the evaporator gap from 6 mm to 4 and 2 mm result in lower flow boiling performance. As shown in Figs. 6-5(a), 6-5(b), 6-6(a), and 6-6(b), the evaporator gaps affect the wall temperatures at a given mass flux. The results show that the wall temperatures had increased gradually when the evaporator gap was decreased from $d = 6$ mm to 4 and 2 mm. The average wall temperature differences for various evaporator gaps $\overline{\Delta T_{W(d_x-d_1)}}$ at $G = 100$ kg/m²·s and $G = 150$ kg/m²·s for “Pocofoam” of 61% porosity and “Kfoam” of 72% porosity are presented in Tables 6-6 and 6-7, respectively where d_x refers to the evaporator gaps of d_2 and d_3 . It is shown that by decreasing the gap from 6 mm to 4 and 2 mm, the average wall temperatures were increased by about 3.4 ~ 3.9°C and 5.3 ~ 7.5°C, respectively. To evaluate the boiling performance difference, the $h_{fb(avg)}$ ratio of the evaporator gap of 6 mm to those at 4 and 2 mm, h_{fbd_1} / h_{fbd_x} are determined, where x refers to the evaporator gap of d_2 and d_3 . The values of h_{fbd_1} / h_{fbd_x} for various evaporator gaps are presented in Tables 6-4 and 6-5. The results show that the flow boiling heat transfer coefficients were reduced by about 5 ~ 6% and 20 ~ 23% by decreasing the evaporator gap from 6 mm to 4 and 2 mm, respectively. The boiling phenomena on the effects of the evaporator gap will be further discussed in Section 6.3.4.

The superheats readings for various evaporator gaps on “Pocofoam” graphite foam of 61% porosity and “Kfoam” graphite foam of 72% porosity are shown in Figs. 6-5(c), 6-5(d) and 6-6(c), 6-6(d), respectively.

(a)

(b)

(c)

(d)

Fig. 6-5 Wall temperature readings and boiling curves for different evaporator gaps tested with “Pocofoam” graphite foam of 61% porosity for (a) and (b) for $G = 100 \text{ kg/m}^2\cdot\text{s}$; (c) and (d) for $G = 150 \text{ kg/m}^2\cdot\text{s}$.

(a)

(b)

(c)

Fig. 6-6 Wall temperature readings and boiling curves for different evaporator gaps tested with “Kfoam” of 72% porosity and FC-72 coolant for (a) and (c) for $G = 100 \text{ kg/m}^2\cdot\text{s}$; (b) and (d) for $G = 150 \text{ kg/m}^2\cdot\text{s}$.

Table 6-4 Average wall temperature differences and average flow boiling heat transfer coefficient ratio between different evaporator gaps with “Pocofoam” of 61% porosity at (a) $G = 100 \text{ kg/m}^2\cdot\text{s}$, and (b) $G = 150 \text{ kg/m}^2\cdot\text{s}$.

Wall temperature difference $\left(\overline{\Delta T_{W(d_x-d_1)}}\right)$	(°C)	Boiling heat transfer coefficient ratio $\left(h_{fb(d_1)} / h_{fb(d_x)}\right)$	
$\overline{\Delta T_{W(d_2-d_1)}}$	3.4	$h_{fb(d_1)} / h_{fb(d_2)}$	1.06
$\overline{\Delta T_{W(d_3-d_1)}}$	5.3	$h_{fb(d_1)} / h_{fb(d_3)}$	1.20

(a)

Wall temperature difference $\left(\overline{\Delta T_{W(d_x-d_1)}}\right)$	(°C)	Boiling heat transfer coefficient ratio $\left(h_{fb(d_1)} / h_{fb(d_x)}\right)$	
$\overline{\Delta T_{W(d_2-d_1)}}$	3.9	$h_{fb(d_1)} / h_{fb(d_2)}$	1.05
$\overline{\Delta T_{W(d_3-d_1)}}$	6.7	$h_{fb(d_1)} / h_{fb(d_3)}$	1.21

(b)

Table 6-5 Average wall temperature differences and average flow boiling heat transfer coefficient ratio between different evaporator gaps with “Kfoam” of 72% porosity at (a) $G = 100 \text{ kg/m}^2\cdot\text{s}$, and (b) $G = 150 \text{ kg/m}^2\cdot\text{s}$.

Wall temperature difference $\left(\overline{\Delta T_{W(d_x-d_1)}}\right)$	(°C)	Boiling heat transfer coefficient ratio $\left(h_{fb(d_1)} / h_{fb(d_x)}\right)$	
$\overline{\Delta T_{W(d_2-d_1)}}$	3.8	$h_{fb(d_1)} / h_{fb(d_2)}$	1.06
$\overline{\Delta T_{W(d_3-d_1)}}$	7.5	$h_{fb(d_1)} / h_{fb(d_3)}$	1.23
(a)			
Wall temperature difference $\left(\overline{\Delta T_{W(d_x-d_1)}}\right)$	(°C)	Boiling heat transfer coefficient ratio $\left(h_{fb(d_1)} / h_{fb(d_x)}\right)$	
$\overline{\Delta T_{W(d_2-d_1)}}$	4.1	$h_{fb(d_1)} / h_{fb(d_2)}$	1.05
$\overline{\Delta T_{W(d_3-d_1)}}$	7.3	$h_{fb(d_1)} / h_{fb(d_3)}$	1.20
(b)			

It can be seen that subsequent decreases of the evaporator gap from 6 mm to 4 and 2 mm result in higher superheats. By using Eq. (6-4), h_{fb} for various evaporator gaps are further quantified. The $h_{fb(avg)}$ values of the porous channel with “Pocofoam” of 61% porosity and $G = 100$ and $150 \text{ kg/m}^2\cdot\text{s}$ for the evaporator gaps of 6, 4, and 2 mm are found to be 14.5, 13.2, and 12.1 $\text{kW/m}^2\cdot\text{K}$ and 16.5, 15.7, and 13.6 $\text{kW/m}^2\cdot\text{K}$, respectively. For “Kfoam” of 72% porosity, the values of $h_{fb(avg)}$ for the evaporator gaps of 6, 4, and 2 mm are 12.1, 11.4, and 9.8 $\text{kW/m}^2\cdot\text{K}$ and 13.1, 12.5, and 11.2 $\text{kW/m}^2\cdot\text{K}$, respectively. It is noted that the highest flow boiling heat transfer coefficient of 16.5 $\text{kW/m}^2\cdot\text{K}$ was achieved by using “Pocofoam” of 61% porosity with an evaporator gap of 6 mm and a coolant mass flux of $150 \text{ kg/m}^2\cdot\text{s}$. It can be seen

that the effects of decreasing the evaporator gap from 6 to 4 mm on the flow boiling heat transfer coefficients are less significant as compared to the decrease of the evaporator gap from 6 to 2 mm.

Based on the results, it is found that the evaporator gap has an important role in the flow boiling mechanism from the graphite foam structures. As the heat flux increases in the boiling process, the generated bubbles will coalesce and form larger bubbles on the graphite foam surface and in the channel gap. The sizes of the merged bubbles were found to be much larger than the individual isolated bubbles generated from the foam structure. At a constant mass flux, a smaller evaporator gap will tend to cause more bubble confinement as compared to larger evaporator gaps. The bubble confinement leads to a decrease in the bubble departure frequency, vapour layer formation, and thus a decrease in the boiling performance. The flow boiling mechanism from different evaporator gaps will be discussed in Section 6.3.4.

6.3.3 Effect of Graphite Foam Types

As mentioned in the previous section, two graphite foams of different porosities and thermal conductivities were tested in the experiments. The wall temperatures and boiling curves for “Pocofoam” of 61% porosity and “Kfoam” of 72% porosity with evaporator gaps of 6 and 4 mm at $G = 150 \text{ kg/m}^2\cdot\text{s}$ are presented in Fig. 6-7. The experimental results show that the thermophysical properties of the foams had significantly affected the flow boiling heat transfer. It can be seen that the wall temperatures obtained by “Pocofoam” foam of 61% porosity are lower than those of “Kfoam” foam of 72% porosity for all tested

cases. The average difference of wall temperatures obtained by “Kfoam” foam of 72% porosity and “Pocofoam” foam of 61% porosity are found to be 8.0 and 8.4°C for the evaporator gaps of 6 and 4 mm, respectively. The boiling curves show that the use of “Pocofoam” foam of 61% porosity result in much lower superheats compared to “Kfoam” foam of 72% porosity. By using Eq. (6-4), the values of $h_{fb(avg)}$ obtained using “Pocofoam” foam of 61% porosity $G = 150$ kg/m²·s are 16.5 and 15.7 kW/m²·K for evaporator gaps of 6 and 4 mm, respectively while for “Kfoam” foam of 72% porosity, the values of $h_{fb(avg)}$ are found to be 13.1 and 12.5 kW/m²·K, respectively. It is also found that the $h_{fb(avg)}$ ratio of “Pocofoam” 61% to “Kfoam” 72% porosity foams are 1.26 for both evaporator gaps of 6 and 4 mm.

(b)

(c)

Fig. 6-7 Wall temperature readings and boiling curves of different graphite foam types with FC-72 coolant at $G = 150 \text{ kg/m}^2\cdot\text{s}$ for (a) and (b): $d = 6 \text{ mm}$; (c) and (d): $d = 4 \text{ mm}$.

As shown in Fig. 6-7, the wall temperature readings and boiling curves of the smooth surface are also presented as a basis of comparison. It can be seen that the use of porous graphite foams have enhanced significantly the boiling performance of the smooth surface. The average differences of wall temperatures between the use of “Pocofoam” graphite foam of 61% porosity and the smooth surface are found to be 21.1 and 19.6°C with the evaporator gaps of 6 mm and 4 mm, respectively. On the other hand, the average difference of wall temperatures obtained by “Kfoam” of 72% porosity and the smooth surface are about 15.1 and 13.2°C, respectively. “Pocofoam” of 61% porosity had better flow boiling heat transfer performance compared to “Kfoam” of 72% porosity. It is also found that the $h_{fb(avg)}$ ratios of “Pocofoam” graphite foam of 61% porosity to the smooth surface are 2.49 and 2.44 for the

evaporator gaps of 6 and 4 mm, respectively while the $h_{fb(avg)}$ ratios of “Kfoam” graphite foam of 72% porosity to the smooth surface are 1.98 and 1.93, respectively.

As shown in Table 3-1 of Chapter 3, “Pocofoam” foam of 61% porosity possesses higher effective thermal conductivity and lower porosity compared to “Kfoam” foam of 72% porosity which has effects on the flow boiling mechanism. The combined effect of these two properties will determine the bubble departure frequency and the active nucleation site density of the porous graphite foam. The larger bubble departure frequency and larger nucleation sites density of the “Pocofoam” foam of 61% porosity account for its superior flow boiling heat transfer performance as compared to “Kfoam” of 72% porosity. This will be further discussed in Section 6.3.4.

6.3.4 Flow Boiling Mechanism and Discussions

The flow boiling heat transfer from porous graphite foams with different coolant mass fluxes and evaporator gaps are discussed in Sections 6.3.1 ~ 6.3.3. As presented in Section 6.3.1, increases in the mass flux from $50 \text{ kg/m}^2 \cdot \text{s}$ to 100 and $150 \text{ kg/m}^2 \cdot \text{s}$ had increased the flow boiling performance by 13% and 34%, respectively. The flow boiling images for different mass fluxes on “Pocofoam” of 61% porosity and “Kfoam” of 72% porosity at $q'' = 83.3 \text{ W/cm}^2$ for evaporator gaps of 6 and 4 mm are shown in Figs. 6-8 and 6-9, respectively. They show that the increase of the coolant mass flux has reduced the number of the coalesced flowing bubbles on the graphite foam surfaces and evaporator

gaps. It can be seen that there was a significant reduction in the number of coalesced bubbles when the mass flux was increased from 50 to 100 kg/m²·s.

This flow boiling visualisation supports the experimental results of enhancement of flow boiling heat transfer with increased coolant mass flux. The higher coolant mass flux will sweep away the generated and coalesced bubbles more effectively from the graphite foam surfaces and evaporator gap and increase the liquid replacement rate to the graphite foam pores. Combining these mechanisms with the heat removal enhancement by forced convection from the higher coolant mass flux resulted in the increase of the bubble departure frequency. It is also shown that there are insignificant differences in the bubble phenomena on the graphite foam for mass fluxes of 100 and 150 kg/m²·s. Therefore, further increase of the coolant mass flux beyond 150 kg/m²·s, might not enhance significantly the flow boiling heat transfer in this system. The selection of the coolant mass flux for a flow boiling cooling system should consider the cooling performance, cost efficiency and the system requirements.

Fig. 6-8 Flow boiling images with various coolant mass fluxes at $q'' = 83.3 \text{ W/cm}^2$ and $d = 6 \text{ mm}$ for graphite foams (a) “Pocof foam” of 61% porosity and (b) “Kfoam” of 72% porosity.

As discussed in Section 6.3.2, the evaporator gaps have affected the flow boiling heat transfer coefficient on the graphite foam structures. By decreasing the evaporator gap from 6 to 4 mm, the flow boiling performance was reduced slightly by 5 ~ 6% for all tested graphite foams and coolant mass fluxes. Further decrease of the evaporator gap from 6 to 2 mm resulted in a significant decrease in the boiling performance by 20 ~ 30 %. The flow boiling images from the graphite foams for various evaporator gaps with the coolant mass fluxes of 100 and 150 $\text{kg/m}^2\cdot\text{s}$ are shown in Figs. 6-10 and 6-11, respectively. At the same heat flux and coolant mass flux, much larger coalesced bubbles were formed in the smaller evaporator gaps, especially at $d = 2 \text{ mm}$ and $G = 100 \text{ kg/m}^2\cdot\text{s}$.

Fig. 6-9 Flow boiling images with various coolant mass fluxes at $q'' = 83.3 \text{ W/cm}^2$ and $d = 4 \text{ mm}$ for graphite foams (a) “Pocof foam” of 61% porosity and (b) “Kfoam” of 72% porosity.

The larger coalesced bubbles on the graphite foam surfaces and evaporator gaps will result in more bubble confinement and vapour layers covering the top foam surface leading to an increase in the graphite foam local temperatures. Simultaneously, it will also diminish fresh liquid replacement rate to the graphite foam pore structures and decrease the bubble departure frequency. Further decrease of the evaporator gap below 2 mm may cause dry-out on the porous foam which will significantly increase the surface temperatures, decrease the cooling performance and result in the appearance of CHF. Therefore, the selection of the evaporator gap in the flow boiling system becomes important to achieve maximum cooling performance and system reliability. In this study, the evaporator gap of 6 mm gave better flow boiling performance for all tested cases.

Fig. 6-10 Flow boiling images with various evaporator gaps at $q'' = 83.3$ W/cm² and $G = 100$ kg/m²·s graphite foams (a) “Pocofoam” of 61% porosity and (b) “Kfoam” of 72% porosity.

Fig. 6-11 Flow boiling images with various evaporator gaps at $q'' = 83.3$ W/cm² and $G = 150$ kg/m²·s graphite foams (a) “Pocofoam” of 61% porosity and (b) “Kfoam” of 72% porosity.

Based on the captured images of boiling process, several bubble regimes can be observed. The first regime is termed as “laminar” in which isolated bubbles were generated intermittently from the active cavities. This bubble regime was

generally observed from the boiling process at low heat flux for all mass flux levels. By increasing the heat flux, the number of active nucleation sites was also increased. The bubble regime at high heat flux is termed as the “turbulent” regime. In this regime, some bubbles had coalesced with their predecessors and formed larger bubbles. When the heat flux was further increased, more coalesced bubbles in the channel were formed. This phenomenon is more obvious in the captured images of high heat flux with low mass flux. Similar flow boiling phenomena and regimes were also found in the visualisation study performed by Ramaswamy *et al.* (2002). The continuous bubble coalescence process had led to the formation of larger “mushroom cloud” bubbles and a thin vapour layer on the porous foam surface as shown in Figs. 6-12 and 6-13, respectively.

Fig. 6-12 Typical “mushroom cloud” of large coalesced bubbles at high heat flux and low mass flux.

Fig. 6-13 Typical thin vapour film formed in channel at low mass flux and high heat flux.

As discussed in Section 6.3.3, the porous graphite foams have enhanced significantly the flow boiling performance of the smooth aluminium surface. In flow boiling from the smooth surface, the nucleation bubbles were generated from the smooth aluminium surface. It was observed from the experiments that relatively fewer bubbles departed from the smooth surface. This is due to the relatively fewer active nucleation sites from the smooth aluminium surface. By attaching the graphite foam inserts, the number of active nucleation sites was increased significantly due to the increased total exposed area to the fluid. The small openings between the pores allow the liquid to fill the entire pore network which in turn leads to a larger number of nucleation bubbles. The significant role of the internal pore structure of the graphite foam on the enhancement of nucleation sites was discussed in Section 5.1 of Chapter 5. It was also mentioned in Section 3.3 of Chapter 3 that the use of “Pocofoam” graphite foam of 61% porosity resulted in significantly higher boiling heat transfer coefficients as compared to “Kfoam” graphite foam of 72% porosity.

The superior boiling heat transfer performance of “Pocofoam” graphite foam of 61% porosity compared to “Kfoam” graphite foam of 72% porosity can be deduced from the properties of the foams. As shown in Table 6-1, “Pocofoam” foam of 61% porosity possesses higher effective thermal conductivity compared to “Kfoam” foam of 72% porosity. In the boiling process, the porous channels with “Pocofoam” graphite foam of 61% porosity inserts will conduct the heat more effectively and the saturation temperature of the foam will be achieved in a shorter time which results in higher bubble departure frequency.

Based on the physical properties of the foams, the total internal surface-area-to-volume ratio (β) of the porous graphite foam was determined as discussed previously in Section 4.1 of Chapter 4. Table 4-4 of Chapter 4 shows that the calculated values of β for “Pocofoam” graphite foam of 61% porosity and “Kfoam” graphite foam of 72% porosity are about 10,400 and 5,900 m^2/m^3 , respectively by using Yu *et al.*’s (2006) model. On the other hand, the unit cell model of the graphite foam proposed by Leong and Li (2011) yields values of β for “Pocofoam” of 61% porosity and “Kfoam” of 72% porosity graphite foams that are about 19,500 and 12,500 m^2/m^3 , respectively.

It is shown that the “Pocofoam” graphite foam of 61% porosity has a much higher internal surface-area-to-volume ratio compared to the “Kfoam” graphite foam of 72% porosity. Therefore, more active nucleation sites will be generated within the “Pocofoam” graphite foam of 61% porosity resulting in higher flow boiling heat transfer coefficients. Flow boiling images from “Pocofoam” of 61% and “Kfoam” of 72% with different evaporator gaps and heat fluxes are shown in Figs. 6-14 and 6-15.

It can be seen that the number of generated bubbles from “Pocofoam” foam of 61% porosity is much larger than “Kfoam” foam of 72% porosity for all tested cases. This is in line with the results of the effect of the foam types and provides the evidence that “Pocofoam” foam of 61% porosity gives a higher nucleation site density compared to “Kfoam” foam of 72% porosity.

Fig. 6-14 Flow boiling images with different graphite foam types with $G = 150 \text{ kg/m}^2 \cdot \text{s}$ at $q'' = 58.3 \text{ W/cm}^2$ for (a) $d = 6 \text{ mm}$ and (b) $d = 4 \text{ mm}$.

Fig. 6-15 Flow boiling images with different graphite foam types with $G = 150 \text{ kg/m}^2 \cdot \text{s}$ at $q'' = 83.3 \text{ W/cm}^2$ for (a) $d = 6 \text{ mm}$ and (b) $d = 4 \text{ mm}$.

The bubble departure frequency was determined and analysed to further explore the different boiling performance from the graphite foams. The methods to determine the bubble departure frequencies are explained in Section 4.1 of Chapter 4. By using the method, the average bubble departure frequency of “Pocofoam” of 61% and “Kfoam” of 72% porosity at 45.8 W/cm^2 are found to be 163 and 145 Hz, respectively. At 58.3 W/cm^2 , the average values of f_d are 169 and 150 Hz for “Pocofoam” of 61% and “Kfoam” of 72% porosity, respectively. Therefore, the combination of larger total internal surface area and higher thermal conductivity of “Pocofoam” of 61% porosity gives rise to a higher active nucleation site density and higher bubble departure frequency and thus, better boiling heat transfer performance.

6.4 SUMMARY

In this study, the effects of coolant mass flux, evaporator gap, and graphite foam properties were investigated. The flow boiling heat transfer enhancements of two different graphite foams were evaluated and compared with the boiling performance of the smooth surface. A visualisation study was performed to understand the flow boiling phenomena and bubble characteristics from different experimental parameters. The following conclusions can be drawn:-

- 1) The coolant mass flux has affected significantly the flow boiling cooling performance. The experimental results show that the increase of mass fluxes from $50 \text{ kg/m}^2 \cdot \text{s}$ to 100 and $150 \text{ kg/m}^2 \cdot \text{s}$ have increased the boiling heat transfer coefficients up to 13% and 34%, respectively. Higher

coolant mass fluxes are able to sweep away the generated and flowing bubbles effectively and increase the liquid replacement rate to the pore structures. In addition, the larger forced convection heat transfer from the higher coolant mass flux resulted in the increase of the bubble departure frequency. However, there are insignificant differences of the bubble phenomena and cooling performance by increasing the mass fluxes from 100 to 150 kg/m²·s. Further increase of the coolant mass flux beyond 150 kg/m²·s might not result in significant enhancement of the flow boiling performance. Therefore, an optimal coolant mass flux should be selected to achieve maximum cooling performance, cost efficiency and the system requirements.

- 2) The experimental results show that the evaporator gap had affected the flow boiling performance. It was found that the flow boiling heat transfer coefficients were reduced by about 5 ~ 6% and 20 ~ 23% by decreasing the evaporator gap from 6 mm to 4 and 2 mm, respectively. From the study, a smaller evaporator gap produces larger coalesced bubbles on the graphite foam surfaces and evaporator gaps and cause bubble confinement and vapour layers which cover the top foam surface resulting in significantly increase of the wall temperatures. In addition, the fresh liquid replenishment rate to the graphite foam pore structures will also be diminished and result in the decrease of the bubble departure frequency. It was found that the highest flow boiling heat transfer coefficient of 16.5 kW/m²·K was achieved by using “Pocofoam” graphite foam of 61% porosity with the evaporator gap of 6 mm.

- 3) With the use of “Pocofoam” graphite foam of 61% porosity and “Kfoam” graphite foam of 72% porosity, the flow boiling heat transfer coefficients were increased significantly by up to 2.5 and 1.9 times, respectively as compared to those of the smooth surface. In addition, the wall temperatures were reduced by 19.6 ~ 21.1°C with the use of the porous graphite foams. The graphite foam’s thermophysical properties and its interconnected pores structures play an important role in the flow boiling enhancements. It was also found that the use of “Pocofoam” of 61% porosity resulted in a heat transfer coefficient which was 1.3 times higher than that of “Kfoam” 72% porosity. The difference of flow boiling heat transfer performance from the tested graphite foam types was evaluated by determining the surface-area-to-volume ratio and the bubble departure frequency. It was found that the surface-area-to-volume ratio of the “Pocofoam” 61% porosity is about 1.2 ~ 1.3 times larger than those of “Kfoam” 72% porosity while the measured bubble departure frequency from “Pocofoam” of 61% porosity is 1.6 ~ 1.7 times higher than those of “Kfoam” of 72% porosity.
- 4) The visualisation study shows that the larger coalesced bubbles and bubble confinements were observed with the decreases of the coolant mass flux and evaporator gap. The flow boiling images also show that larger bubble density was generated by the use of “Pocofoam” 61% porosity as compared to “Kfoam” 72% porosity. “Laminar” and “turbulent” flow boiling regimes were observed in the flow boiling process. Typical “mushroom cloud” bubbles and vapour film layers were

found from the boiling process at high heat flux and low coolant mass flux which indicated that the boiling process had approached the onset of CHF.

CHAPTER 7 BOILING MECHANISM AND SEMI-ANALYTICAL MODELS FOR THE BUBBLE DEPARTURE DIAMETER AND ACTIVE NUCLEATION SITE DENSITY FROM POROUS FOAM STRUCTURES

7.1 BOILING MECHANISM FROM POROUS FOAM STRUCTURES

The experimental results of the boiling heat transfer from the porous graphite foam structures with different experimental parameters are discussed in detail in the previous chapters. It was presented clearly that the pore structure of the graphite foam has significant effect on the enhancement of boiling heat transfer due to its significantly higher nucleation site density and bubble departure frequency. However, the bubble dynamics and boiling mechanism from the foam structures are currently still not well understood. From the experimental results and visualisation study that are presented in Chapters 4 and 5, the bubble dynamics and boiling mechanism in a porous graphite foam structure are hypothesised as follows:

Every pore in a porous foam structure may become an active nucleation site during the boiling process. The nucleation bubbles from the internal pore structure can escape through the open cells between the pores and depart from the exposed-surface porous foam. During the bubble growth and escape process in the porous structure, the bubbles may merge or coalesce with other growing or departed bubbles.

Based on this hypothesis, physical models of the boiling mechanism from the pore structure are developed. The bubble departure diameter, nucleation site density and bubble departure frequency are then modelled by using semi-analytical methods.

7.1.1 Physical Models of Boiling in Pore Structure of the Foam

To develop the boiling parameter models of the bubble departure diameter, active nucleation site density and bubble departure frequency, physical models of the pore structure of the foam are constructed. As shown in Fig. 7-1, a typical porous foam is constructed with nearly spherical pores that are interconnected. The number of channels between the pores varies for different interconnected pores. The shape of the pores and opening cells are closed to spherical with variation of their sizes.

Fig. 7-1 Typical structure of the porous foam with interconnected pores with the opening cells between the pores.

In this study, the foam structure is modelled as a unit-cube foam with a sub-surface pore, opening cell and surface pore. The physical models of the unit-cube foam are constructed based on the following assumptions:

- 1) A unit pore structure of the foam is constructed with a spherical subsurface and a hemispherical surface located at the centre of a unit-cube connected by a cylindrical opening cell between the pores.
- 2) The diameter of the pore (void) is uniform for the entire porous foam structure.
- 3) The pores are regularly arranged in unit-cube volume of pore structure and each pore is connected to other pores via an opening cell.

The proposed unit-cube geometry with a spherical pore and an opening cell between the pores is shown in Fig. 7-2. It can be seen that the structure consists of an array of unit-cubes in a rectangular configuration. The detailed parameters of a unit graphite foam model are presented in Fig. 7-3.

Fig. 7-2 Proposed geometry model for (a) A unit-cube porous foam and (b) Configuration of the units of the foam.

Fig. 7-3 Porous foam unit model and its parameters.

The proposed geometry of the porous foam structure the boiling heat transfer environment is shown in Fig. 7-4.

Fig. 7-4 Schematic of boiling process on porous foam structure.

During the nucleation process, the liquid phase of the coolant inside the sub-surface and surface pores is changed to the vapour phase and departs from the surface pore cavity mouth as a spherical bubble at a bubble radius rate $R_b(t)$ and

contact angle (ϕ) as illustrated in Fig. 7-4. For the proposed boiling mechanism, the following assumptions are made:

- 1) The liquid temperature at the pool is at saturation temperature (T_{sat}) corresponding to its system pressure (P_{syst}).
- 2) The temperature of the porous graphite foam (T_{GF}) is uniform during the process.
- 3) The vapour bubble in the pore structure is in the saturated condition.
- 4) The departing bubble shape from the sub-pore and pore of porous foam is spherical.

The mechanisms of boiling nucleation process in the porous foam structure are described in Fig. 7-5.

Fig. 7-5 Schematic of boiling mechanism (a) Liquid intake and waiting process, (b) ~ (d) Bubble growth process with different instantaneous bubble radius $R_b(t_1)$ - $R_b(t_3)$ and (e) Bubble departure and liquid re-intake process.

As shown in Fig. 7-5, the boiling mechanism in the pore structure can be divided into three phases as described below.

Phase I: *Liquid-intake and waiting process*

After the sub-surface and surface pores are completely filled by fresh liquid, the temperature of the porous foam structure rises and reaches the superheat temperature necessary to initiate the nucleation process (boiling incipience).

Phase II: *Nucleation and bubble growth*

Evaporation of liquid in the pores causes the internal vapour pressure to build up until the vapour located at the mouth of the surface-pore becomes a spherical-shaped bubble, protruding outward to the external pool liquid. Before the growing bubble departs from the surface pore, a necking process will occur in the bubble.

Phase III: *Bubble departure process*

The bubble nucleus will grow into a spherical shape. When the resultant of the forces acting on the bubble is zero, the bubble will start to depart from the surface pore mouth to the surrounding pool-liquid. The liquid intake process will be started and the process will be repeated.

Based on the proposed pore structure model and defined boiling mechanism, the bubble departure diameter, active nucleation site density and bubble departure frequency models are developed and presented in the following sections.

7.1.2 Bubble Departure Diameter (D_B) Model

During nucleation and bubble growth process, a vapour bubble attached to a boiling surface (cavity) will be subjected to different forces. Negative forces will hold the bubble to the surface while positive forces will pull the bubble to detach from the surface. When the positive forces increases faster than the negative forces until the resultant force becomes positive, the bubble will be pushed out from the cavity. Therefore, bubble departure diameter can be determined by a force balance on the foam surface pore.

Four (4) different main forces acting on the growth bubble are considered in the model. These are

1. buoyancy force (F_B),
2. pressure force (F_P),
3. drag force (F_D), and
4. surface tension force (F_σ)

The schematic of forces acting on the cavity is shown in Fig. 7-6.

Fig. 7-6 Schematic of forces acting on a bubble growing on a boiling pore structure.

The proposed bubble departure diameter model assumes the departing bubble to be spherical in shape and attached to the surface and sub-surface pores. The bubble will depart from the cavity if the resultant force on the cavity is positive. Therefore, the bubble departure diameter can be determined by the force balance on the cavity as

$$\sum F_C = 0 \quad (7-1)$$

$$F_B + F_P = F_\sigma + F_D \quad (7-2)$$

7.1.2.1 Buoyancy Force (F_B)

The buoyancy force is an upward acting force exerted by a fluid that opposes an object's weight. For a spherical bubble, it is equal to the weight of the fluid displaced in a liquid-vapour system. Therefore the buoyancy force will be calculated as

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$$F_B = V_B(\rho_L - \rho_V)g \quad (7-3)$$

For a spherical bubble,

$$V_B = \frac{4\pi R_B^3}{3} \quad (7-4)$$

$$F_B = \frac{4\pi R_B^3}{3}(\rho_L - \rho_V)g \quad (7-5)$$

$$R_B = \frac{D_B}{2} \quad (7-6)$$

where

V_B : volume of a spherical bubble (m^3)

ρ_l and ρ_v : liquid and vapour densities (kg/m^3)

R_B : bubble growth radius (m)

7.1.2.2 Excess Pressure Force (F_P)

The dynamic excess vapour on the dry area where the bubble is attached will contribute to lift the bubble and is identified as pressure force which can be calculated as

$$\Delta P = \frac{2\sigma \sin \varphi}{D_p} + \frac{\sigma}{R_p} \quad (7-7)$$

The first term comes from the Laplace equation applied to the bubble base diameter D_p

$$P_v - P_L = \frac{2\sigma}{R_{nuc}} \quad (7-8)$$

and the second term accounts for the effect of curvature at the base of the bubble. Based on the analysis of Keshock and Siegel (1964), this term is too small compared to the first term and can be neglected. The excess pressure force acting on the base area of diameter D_p can be calculated as

$$F_p = \left(\frac{2\sigma \sin \phi}{D_p} \right) \left(\frac{\pi}{4} D_p^2 \right) \quad (7-9)$$

$$F_p = \frac{\pi D_p \sigma \sin \phi}{2} \quad (7-10)$$

where

P_v : vapour pressure (N/m^2)

P_L : liquid pressure (N/m^2)

7.1.2.3 Surface Tension Force (F_σ)

The attraction of the liquid to the surface which acts around the perimeter of the bubble will cause a surface tension. The surface tension forces acting in the system are described in Fig. 7-11. It is proportional to the surface tension of the fluid and the sine of the contact angle and is expressed as

$$F_{\sigma} = 2\pi R_p \sigma \sin \varphi \quad (7-11)$$

where

σ : fluid surface tension (N/m)

φ : contact angle (degree)

R_p : pore radius (m)

Fig. 7-7 Schematic of surface tension acting on the bubble.

7.1.2.4 Drag Force (F_D)

When the bubble grows during a nucleation process, it will be subjected to a drag force. By assuming a spherical bubble rising freely in a liquid and growing away from the wall at velocity equal to the change of its radius with time (dR_B/dT), the drag force can be calculated as

$$F_D = C_D \frac{1}{2} \rho_L \pi R_B^2 \left[\frac{dR_B}{dt} \right]^2 \quad (7-12)$$

where C_D is the drag coefficient that can be determined from the experimental correlation given by Keshock and Siegel (1964) as

$$C_D = \frac{45}{\text{Re}}$$

$$\text{and } \text{Re} = \frac{2\rho_L}{\mu_L} R_B \frac{dR_B}{dt}$$

The bubble departure diameter can be solved by substituting Eqs. (7-5), (7-10), (7-11) and (7-12) into Eq.(7-2). In this analysis, the bubble growth radii for different boiling surfaces and liquids are assumed to be proportional to $t^{1/2}$ and can be expressed as

$$R_B(t) = at^{1/2} \tag{7-13}$$

If $\rho_L \gg \rho_V$, $(\rho_L - \rho_V) \approx \rho_L$, and hence, all the acting forces can be re-expressed as

$$F_B = \frac{4}{3} \pi \rho_L g t^{3/2} a^3 \tag{7-14}$$

$$F_p = \pi R_p \sigma \sin \phi \tag{7-15}$$

$$F_D = \frac{45}{4} \pi \nu_L \rho_L a^2 \tag{7-16}$$

By substituting Eqs. (7-5), (7-10), (7-11) and (7-12) into Eq. (7-2), D_B can be determined as

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$$D_B = \left[\frac{5.62v_l}{gt^{3/2}} + \frac{79.72\pi v_l^2 \rho_l}{gt^{3/2} \left[2847.66\pi^3 v_l^3 \rho_l^3 - 48\pi^3 \rho_l^2 g^2 t^3 R_p \sigma \sin \varphi + 5.20 \left(10125\pi^6 \rho_l^5 g^2 t^3 v_l^3 R_p \sigma \sin \varphi + 85.33\pi^6 \rho_l^4 g^4 t^6 R_p^2 \sigma^2 \sin^2 \varphi \right)^{1/2} \right]} \right]^{1/3} + \left[\frac{\left(2847.66\pi^3 v_l^3 \rho_l^3 - 48\pi^3 \rho_l^2 g^2 t^3 R_p \sigma \sin \varphi + 5.20 \left(10125\pi^6 (\rho_l^5 g^2 t^3 v_l^3 R_p \sigma \sin \varphi + 85.33\pi^6 \rho_l^4 g^4 t^6 R_p^2 \sigma^2 \sin^2 \varphi) \right)^{1/2} \right)}{0.4\pi \rho_l g t^{3/2}} \right]^{1/2} t^{1/2} \quad (7-17)$$

As shown in Eq. (7-17), D_B is a function of pore radius (R_p), coolant properties (ρ , v , σ), contact angle (φ) and bubble growth time (t). In the next section, the values of D_B from the model and experimental measurements will be presented and discussed.

7.1.3 Measured Bubble Departure Diameter

The bubble departure diameter from the porous structure was measured from the captured boiling images using a high speed camera and “Image Pro Software”. From the captured boiling process, bubble growth diameters from the different graphite foam types can be determined with the method shown in Fig. 7-8. It was shown that the bubble growth diameters from the start of the nucleation process to the departure process can be measured from the series of the captured boiling images. The measured bubble growth diameters from the different graphite foams are shown in Fig. 7-9. From the measured bubble diameter, the average bubble departure diameter ($\overline{D_B}$) and the standard deviation (σ_{D_B}) are calculated by

$$\overline{D_B} = \frac{1}{N} \sum_{i=1}^N D_{B_i} \quad (7-18)$$

$$\sigma_{D_B} = \sqrt{\frac{\sum (D_{B_i} - \overline{D_B})^2}{N-1}} \quad (7-19)$$

with the number of measurements $N = 30$ for each graphite foam evaporator. The average bubble departure diameters and their standard deviations for FC-72 and HFE-7000 are shown in Tables 7-1 and 7-2, respectively.

Fig. 7-8 Bubble departure growth diameter measurement method

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(a)

(b)

Fig. 7-9 Measured bubble growth from different porous graphite foam structures with (a) FC-72 and (b) HFE-7000 coolant.

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**Table 7-1 Average and standard deviation of bubble departure diameter
for FC-72 at $q'' = 56.34 \text{ W/cm}^2$**

Graphite foam type	$\overline{D_B}$ (μm)	σ_{D_B} (μm)
“Pocofoam” 61%	480.8	9.5
“Pocofoam” 75%	502.4	10.2
“Kfoam” 78%	635.8	6.5
“Kfoam” 72%	750.1	8.3

**Table 7-2 Average and standard deviation of bubble departure diameter
for HFE-7000 at $q'' = 56.34 \text{ W/cm}^2$**

Graphite foam type	$\overline{D_B}$ (μm)	σ_{D_B} (μm)
“Pocofoam” 61%	520.6	8.6
“Pocofoam” 75%	580.5	9.8
“Kfoam” 78%	675.6	7.5
“Kfoam” 72%	784.5	7.6

7.2 ACTIVE NUCLEATION SITE DENSITY (Na) – MODEL 1

During the boiling process, bubbles will be generated from the activated pores and departed from the foam surfaces. The active nucleation site density is the number of active sites per unit area on the foam exposed area. If the heat given to the surface is increased, more bubbles will be generated from the active sites and the active nucleation site density will be increased accordingly. This is one of the key parameters of the pool boiling process. The boiling heat transfer coefficient will be directly affected by this parameter. Hence, many research

efforts have been undertaken by previous investigators to increase the number of the active sites by surface modification techniques such as machining, polishing, sintering, etching, and coatings on various surface materials. A schematic of the boiling process and nucleation sites is shown in Fig. 7-10. From a heat and mass balance analysis, the number of bubbles activated in the porous foam can be determined as from the following equations:-

Fig. 7-10 Schematic of the boiling and nucleation sites in a porous graphite foam.

$$q = m_B h_{lv} f_B n_B \quad (7-20)$$

$$m_B = \rho_v V_B \quad (7-21)$$

Therefore,

$$q = \rho_v V_B h_{lv} f_B n_B \quad (7-22)$$

For a spherical bubble,

$$V_B = \frac{4}{3}\pi R_B^3 \quad (7-23)$$

$$R_B = \frac{D_B}{2}$$

$$q = \rho_v \frac{4}{3}\pi \left(\frac{D_B}{2}\right)^3 h_{lv} f_B n_B \quad (7-24)$$

which can be re-written as

$$q = \frac{1}{6}\pi D_B^3 \rho_v h_{lv} f_B n_B \quad (7-25)$$

Therefore, the number of generated bubbles n_B and the active nucleation site density N_a can be expressed as

$$n_B = \frac{6q}{\pi D_B^3 \rho_v h_{lv} f_B} \quad (7-26)$$

$$N_a = \frac{n_B}{A_T} \quad (7-27)$$

giving

$$N_a = \frac{6q}{\pi D_B^3 \rho_v h_{lv} f_B A_T} \quad (7-28)$$

As shown in Eq. (7-28), the bubble departure frequency must be determined first in order to calculate the bubble departure diameter. In this study, the bubble departure frequency (f_B) was calculated from the captured bubble

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growth and departure during the boiling process. The methods to determine the bubble departure frequency are presented in Section 4-1 of Chapter 4. The numbers of sites per unit area N_a calculated with the proposed model of Eq. (7-28) are presented in Table 7-3.

Table 7-3 Calculated active nucleation sites based on Model 1 at $q'' = 56.34$ W/cm²

Coolants	Porous foam types	ρ_v (kg/m ³)	D_B (μ m)	f_B (Hz)	h_{lv} (kJ/kg)	N_{a1} (sites/cm ²)
FC-72	“Pocofoam” 61%	13.24	480.8	171	88	34
	“Pocofoam” 75%		502.4	156		44
	“Kfoam” 78%		635.8	163		23
	“Kfoam” 72%		750.1	149		15
HFE-7000	“Pocofoam” 61%	8.17	520.6	166	142	28
	“Pocofoam” 75%		580.4	152		31
	“Kfoam” 78%		675.6	159		21
	“Kfoam” 72%		784.5	149		14

7.3 ACTIVE NUCLEATION SITE DENSITY(N_a) – MODEL 2

As described in Chapter 2, many models have been proposed to predict the active nucleation site density. Yang and Kim’s (1988) model was the first attempt to predict quantitatively the active nucleation site density from the knowledge of measured radius and cone angle distribution of real cavities on the surface. This approach had been applied by Wang and Dhira (1993) and

Hibiki and Ishi (2003) to predict the active nucleation site density. In this section, a model of the active nucleation site density on a typical porous graphite foam will be developed with the following steps:

1. Sample design and surface characterisation
2. Determination of the *Probability Density Function (PDF)* of the cavity radius
3. Development of the active nucleation site density model
4. Determination of the minimum and maximum radii (R_{min} and R_{max}) of the cavity that will be categorised as an active nucleation site.
5. Solution of the model.

The model is developed based on the following considerations:

1. The graphite foam surface is predominantly constructed by pores with smaller openings between the cells. The distributed pores in the foam surface are considered as possible cavities for activation of nucleation. Physically, the graphite foam pores are nearly spherical as shown in Fig. 1-1(b). Hence, cavities on the machined foam surfaces are treated as hemispherical in shape. A schematic hemispherical cavity on a pool boiling surface is shown in Fig. 7-11.
2. The only surface characteristic parameter which has an effect on the nucleation site is the cavity radius. Since the cavity is assumed to be hemispherical, the depth of cavity is the same as its radius.

3. The sample graphite foams used in the study are assumed to be isotropic and homogeneous.

Fig. 7-11 Schematic of the cavity and the vapour entrapment in the pool boiling.

7.3.1 Sample Design and Surface Characterisation

In order to perform surface characterisation, two graphite foams of different porosities were used as samples. They are “Pocofoam” of 75% and 61% porosities. The samples were cut by Wire Cut Electron Discharge Machining (EDM) to ensure high accuracy of the surface flatness. The structure and dimensions of the sample are shown in Fig. 7-12. After the machining process, many small foam chips were found to have blocked the pores of the foam. These were removed by a pressurised air gun and strong sticky tapes.

Fig. 7-12 Structure and dimensions of the sample.

In order to determine the probability density function of the cavity radius on the foam surface, the samples were observed under a Scanning Electron Microscope (SEM) and captured at $30\times$ magnification level. As the SEM limit operation range at this level is $4205\text{ }\mu\text{m} \times 3145\text{ }\mu\text{m}$, three measurements at different locations were made for every sample. This will provide a more representative cavity distribution in the samples. The captured SEM images of “Pocofoam” of 75% and 61% porosities are shown in Fig. 3-19 of Chapter 3. In order to investigate the dependence of nucleation site density on the actual boiling surface condition, the probability density functions of the cavity radius were determined. In this model, the dependence of active nucleation site density on boiling surface conditions was developed in the form:

$$N = \bar{N} \cdot \phi(R) \quad (7-29)$$

where N is the active nucleation site density, $\bar{N}$ is the average measured cavity density, and $\phi(R)$ is the distribution function of the cavity radius expressed as

$$\phi(R) = \int_{R_{min}}^{R_{max}} f(R) dR \quad (7-30)$$

where $f(R)$ is the probability density function of the cavity in terms of cavity radius and R_{min} and R_{max} are the minimum and maximum radii of the active cavity, respectively. Finally, the active nucleation site density can be expressed as

$$N = \bar{N} \int_{R_{min}}^{R_{max}} f(R) dR \quad (7-31)$$

7.3.2 Determination of Cavity Radius

The cavities of graphite foam surface are assumed to be hemispherical. However, based on the surface measurements using SEM, the cavities were found to be not exactly hemispherical in shape. The “IMAGE PRO” Software was used to calculate the effective radius, R_{eff} , of the cavity. This software has a “trace” function to construct a high order smooth line following the cavity contour and can measure the length (perimeter of the circular shape) and area of the constructed features. The calculations of the area and perimeter of the cavities using this software are shown in Fig. 7-13. The values of R_{eff} are calculated by using Eq. (7-32) as

$$R_{eff} = \frac{2A}{L_c} = \frac{2\pi R^2}{2\pi R} = R \quad (7-32)$$

where A and L_c are the measured area and perimeter of cavity, respectively.

Fig. 7-13 Determination of area and perimeter of the cavities

7.3.3 Probability Density Function (PDF) of Cavity Radius

In order to determine the probability density function of the cavity radius, the measured radius values were grouped into several ranges. The number of cavities for each range was calculated from the SEM images manually. The cavity distribution of “Pocofoam” of 61% and 75% porosities are plotted in Fig. 7-14.

As shown Fig. 7-13, all the cavity radius distributions are assumed to fit the normal distribution. Therefore, for this model the probability density function of cavity radius of graphite foam can be expressed as

$$f(R) = \left[\frac{1}{\sqrt{2\pi}s^2} \exp\left(-\frac{(R-\bar{R})^2}{2s^2}\right) \right] \quad (7-33)$$

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(a)

(b)

(c)

(d)

Fig. 7-14 Measured cavity radius for different graphite foam types of (a) “Pocofoam” 61%, (b) “Pocofoam” 75%, (c) “Kfoam” 78% and (d) “Kfoam” 72%.

Table 7-4 Mean and standard deviation of the cavity radius (R_P)

Graphite foam types	R_P (μm)	s (μm)
“Pocofoam” 61%	195.5	34.6
“Pocofoam” 75%	201.0	36.5
“Kfoam” 78%	295.9	38.0
“Kfoam” 72%	334.9	25.8

where $\bar{R}$ and s are the mean and standard deviation of the cavity radius. By using the statistical method, the $\bar{R}$ values of “Pocofoam” foams of 75% and 61% porosity are 181.6 μm ($s = 33.4703$ μm) and 200.2 μm ($s = 33.5$ μm), respectively. The number of active nucleation sites in Eq. (7-34) can then be determined as

$$N = \bar{N} \int_{R_{min}}^{R_{max}} \left[\frac{1}{\sqrt{2\pi}s^2} \exp\left(-\frac{(R-\bar{R})^2}{2s^2}\right) \right] dR \quad (7-34)$$

7.3.4 Determination of Minimum (R_{min}) and Maximum (R_{max}) Radii

As mentioned previously, not all cavities on the boiling surface can activate the nucleation. Therefore, the requirements for nucleation activation have to be specified. Equation (7-34) shows that the requirements of active sites depend only on the cavity radius. The boiling process occurs when the superheat is large enough to initiate the growth of bubbles. Nukiyama (1966) stated that boiling, as evidenced, by the presence of bubbles did not begin until the wall superheat, $\Delta T_w \approx 5^\circ\text{C}$. Therefore even though a cavity can entrap the gas-vapour, it will not become an active nucleation site if ΔT_{sat} is below the required wall superheat. With this physical consideration, the minimum cavity radius that will initiate bubble generation should be a function of wall superheat. Before bubble growth begins, the bubble is in condition of thermostatic equilibrium. The static equilibrium of the bubble is given by the *Laplace* equation

$$\Delta P = \frac{2\sigma}{R_{nuc}} \quad (7-35)$$

where $\Delta P = (P_v - P_l)$ is the difference between vapour pressure inside the bubble (P_v) and liquid pressure (P_l), σ is surface tension of the liquid, and R_{nuc} is the nucleation radius. The condition of static equilibrium of the bubble is

illustrated in Fig. 7-7 in Section 7.1. As shown by Lord Kelvin (1871), there is an effect of the curvature on the vapour-liquid interface that will lower the pressure inside the bubble (P_v) relative to the interface pressure between bubble-liquid (P_o) at the same temperature and expressed as

$$P_v = P_o \exp\left(\frac{-2\sigma v_l M}{R_{nuc} \bar{R} T}\right) \approx P_o \left(1 - \frac{2\sigma v_l}{P_o R_{nuc} v_v}\right) \quad (7-36)$$

where M is the molecular weight in kg/mol, $\bar{R}$ is the ideal gas constant ($\bar{R} = 8.3144$ J/mol·K), and v_v and v_l are the specific volumes of the vapour and liquid, respectively.

Substituting the definition of P_v into Eq. (7-36), the equation can be rearranged as

$$P_o - P_l = \frac{2\sigma}{R_{nuc}} \left(1 + \frac{v_l}{v_v}\right) \quad (7-37)$$

To solve the slope of vapour pressure in a planar vapour-liquid interface, the *Clausius-Clapeyron* equation is applied as

$$\left(\frac{dP}{dT}\right)_{sat} = \frac{h_{lv}}{T_{sat}(v_v - v_l)} \quad (7-38)$$

where T_{sat} is in K. By assuming the behaviour of the bubble as a perfect gas, then

$$MP_v v_v = \bar{R} T \quad (7-39)$$

Since $v_v \gg v_l$, $(v_v - v_l) \approx v_v$ and introducing Eq. (7-39) into Eq. (7-38), we obtain

Eq. (7-40). With re-arrangement, this equation becomes Eq. (7-41).

$$\left(\frac{dP}{dT} \right)_{sat} = \frac{h_{lv} \bar{RT}}{T_{sat} M P_v} \quad (7-40)$$

$$\frac{1}{P} dP = \frac{h_{lv} M}{RT^2} dT \quad (7-41)$$

Integrating from P_l to P_o and from T_{sat} to T_v , Eq. (7-41) becomes

$$\ln \frac{P_o}{P_l} = -\frac{h_{lv} M}{R} \left(\frac{1}{T_v} - \frac{1}{T_{sat}} \right) = \frac{h_{lv} M}{RT_v T_{sat}} (T_v - T_{sat}) \quad (7-42)$$

Substituting Eq. (7-9) and rearranging, Eq. (7-42) becomes

$$T_v - T_{sat} = \frac{\bar{RT}_{sat} T_v}{h_{lv} M} \ln \left[1 + \frac{2\sigma}{P_l R_{nuc}} \left(1 + \frac{v_l}{v_v} \right) \right] \quad (7-43)$$

Since $v_v \gg v_l$ then $(v_v - v_l) \approx v_v$ and at nucleation condition $2\sigma / P_l R_{nuc} < 1$, Eq.

(7-15) can be simplified as

$$T_v - T_{sat} = \frac{\bar{RT}_{sat} T_v}{h_{lv} M} \frac{2\sigma}{P_l R_{nuc}} \quad (7-44)$$

The difference between T_v and T_{sat} is referred as the nucleation superheat or the wall superheat ($T_w - T_{sat}$) in previous published works (Han and Griffith, 1965;

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Mikic *et al.*, 1970; Shoukri and Judd, 1975; Singh *et al.*, 1976; Yang and Kim, 1988; and Wang and Dhir, 1993).

Finally, the wall superheat can be simplified to become

$$(T_w - T_{sat}) = \frac{2\sigma}{R_{nuc} (dP/dT)_{sat}} \quad (7-45)$$

With the definition of $(dP/dT)_{sat}$ in Eq. (7-10), R_{nuc} is expressed as

$$R_{nuc} = \frac{2\sigma T_{sat} v_{lv}}{h_{lv} (T_w - T_{sat})} \quad (7-46)$$

where $v_{lv} = v_v - v_l$

As mentioned previously, R_{nuc} is the nucleation radius, which means the radius of bubble at the initiation of the nucleation process. It is shown from Eq. (7-18) that R_{nuc} depends only on the properties of the fluid and the wall superheat. Therefore, the minimum radius of the cavity that can initiate nucleation can be determined from Eq. (7-18). The integral limit, R_{min} in Eq. (7-6) is then expressed as

$$R_{min} = \frac{2\sigma T_{sat} v_{lv}}{h_{lv} (T_w - T_{sat})} \quad (7-47)$$

R_{min} expressed in Eq. (7-47) is used to substitute R_{min} in Eq. (7-34). The value of R_{max} in Eq. (7-34) will be the same as the maximum pore diameter (R_{Pmax}) of the graphite foam and is expressed as

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$$R_{\max} = R_{p_{\max}} \quad (7-48)$$

Therefore, the active nucleation site density equation in Eq. (7-34) is expressed as

$$N_{a2} = \bar{N} \int_{\left(\frac{2\sigma T_{sat} \nu_{lv}}{h_{lv}(T_W - T_{sat})}\right)}^{R_{p_{\max}}} \left[\frac{1}{\sqrt{2\pi s^2}} \exp\left(-\frac{(R_p - \bar{R}_p)^2}{2s^2}\right) \right] dR \quad (7-49)$$

The deviation in values of the active nucleation site density obtained from the two models [Eqs. (7-28) and (7-49)] can be calculated by using

$$RD = \left[\frac{(N_{a2} - N_{a1})}{N_{a1}} \times 100 \right] \% \quad (7-50)$$

The values of N_a calculated by using both models of Eqs. (7-28) and (7-34) at the heat flux of 56.36 W/cm² and its deviation values are presented in Table 7.5.

Table 7-5 N_a values based on Model 2 at $q'' = 56.36$ W/cm²

Coolant type	Porous foam type	N_{a1}	N_{a2}	RD (%)
		(sites/cm ²)	(sites/cm ²)	
FC-72	“Pocofoam” 61%	45	48	5.6
	“Pocofoam” 75%	44	50	12.5
	“Kfoam” 78%	23	27	17.4
	“Kfoam” 72%	15	21	34.6
HFE-7000	“Pocofoam” 61%	34	45	32.0
	“Pocofoam” 75%	31	46	48.8
	“Kfoam” 78%	21	26	24.5
	“Kfoam” 72%	14	20	39.6

As shown in Table 7.5, the deviations in values between the two proposed models varied from 5.6 ~ 48.8%. In view of the complexity of the boiling process, the magnitudes of these deviations would be considered acceptable although it should be mentioned that the proposed N_a models could not be validated due to the difficulty of obtaining the values experimentally.

7.4 SUMMARY

The bubble dynamics and boiling mechanism from a porous graphite foam structure are discussed in this chapter. The hypothesis of the bubble growth in a porous foam structure is stated. Based on this hypothesis, the physical model and bubble growth mechanism from the foam structure are proposed. A semi-analytical model of bubble departure diameter from a porous foam structure is developed and presented. The measured bubble departure diameter from different porous graphite foams are presented and discussed. Active nucleation site density models from a porous foam structure are also developed and discussed in this chapter. There are two models of the nucleation sites density are proposed. The first model (Na_1) is developed based on the heat and mass balance analysis during the nucleation process. The values of bubble departure diameter and frequency are required to calculate the active nucleation sites density in the model. The Na_1 model is expressed in Eq. (7-28) in this chapter.

The second model of the active nucleation site density (Na_2) is developed based on the probability density function of cavity radius of porous foam that have been determined from the SEM images of the porous foam. It is found that the

cavity radius distributions from the graphite foams can be fitted as a normal distribution. The Na_2 model is expressed in Eq. (7-49) in this chapter. The deviation values of the active nucleation sites density calculated from both models are found to be 6 ~ 49% for all graphite foams and coolants.

CHAPTER 8 CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

The major findings of this thesis are summarised in this chapter. Significant contributions and potential applications of this research are highlighted. In addition, recommendations for future work are proposed.

8.1 CONCLUSIONS

In this thesis, pool and flow boiling experimental facilities were designed and fabricated. Comprehensive experimental studies of pool and flow boiling heat transfer from the different porous graphite foams, dielectric working fluids, and graphite foam geometry were conducted to determine the boiling performance and characteristics. Important boiling parameters such as bubble departure diameter, bubble departure frequency, active nucleation site density and critical heat flux were also examined. A mechanism of bubble growth and departure from the pore structure of the foams is proposed. Semi-analytical models of bubble departure diameter and active nucleation sites density from the porous foam structure were developed and compared with the experimental results. The key conclusions drawn from this study are outlined below.

8.1.1 Saturated Pool Boiling Heat Transfer from Porous Graphite Foam Structures

The experimental results show that the use of porous graphite foams have enhanced significantly the boiling heat transfer coefficient by up to 3 times as compared to the smooth copper block evaporator. It was also shown that, the CHF values were enhanced by about 2 times up to 218.31 ~ 225.35 W/cm². The boiling heat transfer coefficient ratios of “Pocofoam” of 61% porosity to “Pocofoam” of 78% porosity, “Kfoam” of 75% porosity, and “Kfoam” of 72% porosity are about 1.3, 1.4, and 1.5, respectively. The bubble departure frequency from different graphite foam types were determined from the boiling images captured by a high speed camera while the internal surface area to volume ratio of the foams were calculated to predict their potential active nucleation site densities. The graphite foam evaporator that produces higher bubble departure frequency and larger potential active nucleation site density would perform better in boiling heat transfer. However, in comparing “Pocofoam” of 75% porosity and “Kfoam” of 78% porosity foams competing advantages between bubble departure frequency and active nucleation site density was found. It is noted that the largest boiling heat transfer coefficient of 42 kW/m²·K was achieved by using “Pocofoam” of 61% porosity and FC-72 coolant.

The effects of working fluid type on the boiling heat transfer were studied. The results show that boiling in FC-72 resulted in higher boiling heat transfer coefficients of up to 1.3 and 1.4 times compared to HFE-7000 and HFE-7100,

respectively. The dimensionless parameters of Bond number (Bo), Grashof number (Gr), and capillary number (Ca) were analysed to evaluate the boiling heat transfer performance from different working fluids. The results show that a large Gr and small Ca indicate that the dynamic viscosity of the fluid is less significant as compared to buoyancy and surface tension forces. The higher Bond number for FC-72 as compared to HFE-7000 and HFE-7100 indicates that the bubbles need to overcome a smaller surface tension force before departing the foam wall in FC-72. A lower surface tension corresponds to higher bubble frequency, which leads to better boiling heat transfer performance.

The experiments on the working fluid level suggest that it has no significant effect on the cooling performance as long as the graphite foam inserts are fully immersed in the working fluids.

8.1.2 The Role of the Internal Pore Structure of Graphite Foams on Boiling Heat Transfer

The role of the internal pore structure of the foams on the boiling heat transfer enhancement was studied by testing the block and fins structure of different fin-to-block-surface-area ratios $AR = 3.70$ and 2.73 . It was found that the block structure resulted in a boiling heat transfer coefficient which is up to 1.6 times higher than the fin structures. The better boiling performance from the block structures indicates the important role the internal pore structure of the porous graphite foams play on the enhancement of boiling performance. Larger values

of the total exposed surface area and surface area to volume ratio from the block structures imply more nucleation sites and enhanced boiling performance. The interconnected pores in the foams allow the coolant to permeate through the internal pore structure and will also allow the bubbles to escape from the porous evaporator during the boiling process. Therefore, the entire pore structure of the foam provides potential nucleation sites for generation of bubbles compared to only the surface voids of conventional heater surfaces. Hence, careful selection of the evaporator structure is important in the cooling system applications.

8.1.3 Flow Boiling Heat Transfer from Porous Graphite Foam Structure

1. Flow boiling heat transfer from graphite foams of different properties, coolant mass fluxes, and evaporator gaps were studied. The experimental results show that the coolant mass flux has affected significantly the flow boiling cooling performance. For instance, the increase of mass fluxes from $50 \text{ kg/m}^2\cdot\text{s}$ to 100 and $150 \text{ kg/m}^2\cdot\text{s}$ have increased the boiling heat transfer coefficients by up to 13% and 34%, respectively. Supported by a bubble flow visualisation study, it can be understood that higher coolant mass fluxes are able to sweep away the generated and flowing bubbles effectively and increase the liquid replacement rate to the pore structures. In addition, higher bubble departure frequency was produced by larger forced convection heat transfer from the higher coolant mass flux. However, after certain magnitude of coolant mass flux, boiling heat transfer might not be enhanced significantly. Therefore, an optimal coolant mass flux should be

selected to achieve maximum cooling performance, cost efficiency and the system requirements.

2. It was found that the evaporator gap has significant effect on the flow boiling performance. The results show that the flow boiling heat transfer coefficients were reduced by about 6% and 23% by decreasing the evaporator gap from 6 mm to 4 and 2 mm, respectively. The captured boiling images suggested that a smaller evaporator gap produces larger coalesced bubbles and causes bubble confinement and the formation of vapour layers on the top foam surface resulting in significant increase of the wall temperatures. It can also be concluded that by reducing the evaporator gap, the fresh liquid replenishment rate to the graphite foam pore structures will also be diminished and thus, result in the decrease of the bubble departure frequency.
3. As in pool boiling, the use of graphite foam evaporator has enhanced significantly the flow boiling heat transfer coefficient as compared to those of smooth surface. The results show that use of “Pocofoam” of 61% porosity resulted in flow boiling heat transfer coefficients of up to 1.3 times higher than those of “Kfoam” 72% porosity. This suggests that the graphite foam thermophysical properties and its interconnected pore structure play an important role in the flow boiling enhancement. Similar to the results of pool boiling heat transfer on the porous graphite foams, it was also found that the surface-area-to-volume ratio of the “Pocofoam” 61% porosity is about 1.3 times larger than those of “Kfoam” 72% porosity while the

measured bubble departure frequency from “Pocofoam” of 61% porosity is 1.7 times higher than those of “Kfoam” of 72% porosity.

4. From the visualisation study, “laminar” and “turbulent” flow boiling regimes were observed in the flow boiling process. Typical “mushroom cloud” bubbles and vapour film layers were found from the boiling process at high heat flux and low coolant mass flux which indicated that the boiling process had approached the onset of CHF.

8.1.4 Wall Superheat Correlations from Porous Graphite Foams

Pool boiling heat transfer correlations to predict the wall superheat from the porous graphite foams were proposed and compared with the experimental data. It was found that the maximum errors for the wall superheat ($T_W - T_{sat}$) prediction are about $\pm 35\%$ and $\pm 25\%$ for FC-72 and HFE-7000, respectively. These errors are considered to be acceptable for predicting the boiling heat transfer performance which is an inherently complex phenomenon.

8.1.5 Semi-analytical Models of Bubble Departure Diameter and Active Nucleation Site Density.

A semi-analytical model of bubble departure diameter from a porous foam structure was developed. On the other hand, two semi-analytical models of active nucleation site density from a porous foam structure were developed. The first model (Na_1) was developed based on the heat and mass balance analysis during the nucleation process. The values of bubble departure diameter and frequency are required to calculate the active nucleation site density in the model. The second model of the active nucleation site density

(Na_2) was developed based on the probability density function of the cavity radius of porous foam which has been determined from SEM images of the porous foam. The model was developed by modifying Yang and Kim's (1988) model to account for the real cavity distribution of the porous graphite foam. The model shows that the cavity radius distributions from the graphite foams can be fitted as a normal distribution. It was found that the deviation in values of the active nucleation sites density calculated from both models are between 6 ~ 49% for all graphite foams and coolants which are deemed to be acceptable due to the complexity of the boiling process.

8.2 HIGHLIGHTS OF THESIS CONTRIBUTIONS

The main contributions of this thesis to the field of phase change cooling systems are stated as follows:-

1. The experimental results have clearly confirmed the significant enhancement of pool boiling heat transfer obtained by using porous graphite foam as the evaporator as reported by Leong *et al.* (2011), Jin *et al.* (2011a), Jin *et al.* (2011b, Klett *et al.* (2004), Coursey *et al.* (2005) and Parker and El-Genk (2005). However, in the author's works, more details of the bubble dynamics and the enhancement mechanism were studied and explored through the visualisation study. In this thesis, important boiling parameters such as bubble departure diameter, bubble departure frequency and nucleation site density of the graphite foam were determined. This knowledge will be useful for the development an advanced cooling system using graphite foam as the evaporator insert.

2. The comprehensive experimental study of boiling heat transfer from graphite foams presented in this thesis has demonstrated that the entire internal pore structure of the graphite foams and not just the surface contributes to the generation of nucleation bubbles and enhanced boiling performance. It was found that the block structure of graphite foam is associated with a higher boiling heat transfer coefficient compared to those of the fin structure although it is well known that for solid materials, fin structures have been widely used to enhance cooling performance. This study suggests that for the porous graphite foam structure, a reduction in the area of the foam would reduce the potential nucleation site density and decrease the boiling performance. The visualisation study has also shown that more bubbles were generated from the block structures compared to the fin structures due to the larger number of nucleation sites from the block structures. This study therefore provides important guidelines on how to design the graphite foam geometry in two-phase cooling application.
3. Studies of flow boiling heat transfer from porous structures including metal foams have been conducted by previous researchers such as Kim *et al.* (2008), Zhao *et al.* (2009), and Li and Leong (2011). To the best of the author's knowledge, the first study on flow boiling heat transfer and bubble characteristics from porous graphite foams conducted as part of this thesis research was reported by Pranoto and Leong (2014). From this study, the bubble flow characteristics over the graphite foam in a single flow channel were elucidated. Important parameters such as coolant mass flux, graphite foam properties, evaporator gaps, and bubble departure frequency were

investigated. The bubble flow dynamics from the graphite foam were also clearly explored. The results will be useful to engineers in the design of flow boiling cooling systems with porous foam evaporators.

4. Pool boiling heat transfer correlations to predict the superheat temperature from the porous graphite foam structures with dielectric coolants of FC-72 and HFE-7000 are proposed from this study. The correlations were modified from the Rohsenow pool boiling correlation by accounting for the total equivalent boiling surface area of the graphite foams. The correlations are very useful to predict heat transfer performance from the porous foam evaporator in a cooling system application.
5. The bubble growth mechanism from porous graphite foam materials presented in this study is important to the design and analysis of two-phase cooling systems using graphite foam evaporators or other porous foams with similar structures. The developed semi-analytical models will be useful to predict the bubble departure diameter and the active nucleation sites density from the porous graphite foam structure.

8.3 RECOMMENDATIONS FOR FUTURE WORK

Although there are significant findings and contributions from this study, there are some limitations and simplifications that can be re-examined. Possible extensions of this present study for future research are outlined as follows:

1. This study has shown that the use of graphite foams can enhance significantly the boiling heat transfer in a laboratory scale experimental facility. Further experimental work to extend the investigations to an actual

electronic device in pool and flow boiling condition should be carried out. Mechanical stability tests need to be performed in order to ensure that the graphite foam can replace the current metal heat sinks and evaporators used. It would also be necessary to look into an improved method to bond the graphite foam to the evaporator surface to reduce the thermal contact resistance.

2. In the current work, the visualisation system can only observe the bubbling phenomena on the foam surfaces. It will be challenging and useful to develop an advanced visualisation system to explore the bubble dynamics and bubble migration in the internal pore structures of the foam using a combination system of micro and macro high-speed cameras.
3. In the current bubble departure diameter and active nucleation site models, some assumptions on the pore geometry and structure were made. Future research efforts may focus on extending the model to one with interconnected pores structures between the cells, real bubble growth and departure shape and thicker pore structure layer. With this extension, the model should be able to predict the bubble departure diameter and active nucleation site density with higher accuracy.

In addition to what was presented in this thesis, the author has also carried out studies on boiling heat transfer from carbon-nanotube (CNT) structures with high thermal conductivity as reported in Pranoto *et al.* (2014). Developing a graphite foam-CNT composite material may potentially improve the current cooling performance. The combined advantages of high thermal conductivity of CNT structures and the large total exposed surface areas of graphite foam

CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

structures with interconnecting pores may result in the development of a better cooling system for high heat flux applications.

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APPENDIX A

Table A-1 Properties of “Omega Bond 101”

Properties	Value
Material	epoxy adhesive
Thermal conductivity	7.2 (BTU) (in)/(hr)(ft ²)(°F) 1.04 W/m·K
Tensile shear	2200 psi (Min) 15.56 MPa
Flexure strength	1200 psi (Min) 8.27 MPa
Coefficient of thermal expansion	20×10^{-6} in/in/°F 36×10^{-6} m/m/°C

Table A-2 Specifications of “Fastec Hispec” high speed camera

Parameters	Specifications
Sensor	CMOS sensor, 1280 x 1024 pixels, 10-bit monochrome
Pixel size	14 × 14 μ m
Resolution	up to 500 frames per second
Light sensitivity	2500 ASA monochrome
Recording rate	up to 500 fps at full resolution and up to 112,000 fps at reduced resolution
Image amplification	digital gain from 1 to 4 steps
Power consumption	7.5 W
Dimension	63 mm <i>H</i> × 63 mm <i>W</i> × 65 mm <i>D</i>
Weight	0.28 kg
Video light power	500 W

Table A-3 Specifications of Micropump MZR-7255

Technical data	
Flow rate	0.048 – 288 ml/min
Smallest dosage volume	30 µl
Displacement volume	48 µl
Max. system pressure	80 bar (1160 psi) (inlet pressure+differential pressure)
Differential pressure range	0 – 20 bar (290 psi)
Operating temperature range	-5 ... +60 °C (-20 ... +150 °C *)
Viscosity range	0.3 – 1000 mPas
Dosage precision	< 1 % Coefficient of variation CV
Pulsation	< 6 %
Speed	1 – 6000 rpm
Fluid connection	1/8" NPT internal thread, lateral
Wetted parts	Pump case alloy C22 (2.4602), optional: stainless steel 316L; seals FFKM (Kalrez® Spectrum™ 6375), optional: FPM, EPDM; shaft/bearing sintered silicon carbide (SSiC); bearing and wetted functional parts Al ₂ O ₃ ceramics; rotors partially stabilized ZrO ₂ , optional: tungsten carbide Ni-based
Electrical interface	8-pole connector NdFeB magnetic coupling
Drive and control	DC-servomotor, 24 V DC, 44 W, with microcontroller
Interface	0–10 V, 0 (4) –20 mA, RS-232, 1 digital input/output, optional: CAN-Bus
Measurements (L x B x H)	146 x 70 x 72 mm
Weight	approx. 1650 g

Table A-4 Specifications of PolyScience Water Bath Circulator

Specifications	
Temperature Stability ¹	±0.01°C
Controller / RS232	Yes
External Temperature Probe	Functional on Programmable models / optional external probe required Not functional on Digital models
Readout Accuracy	Graphic LCD, °C or °F, ±0.25°C
Heater	1100W – 115V, 2200W – 240V
Pressure Flow Rate	15.7 to 30 LPM (60Hz); 15 to 22 LPM (50Hz)
Suction Flow Rate	11 to 22 (60Hz); 10 to 15 LPM (50Hz)
Over-Temperature Protection	Yes, user-adjustable
Low-Liquid Protection	Yes
Pump Speeds	Five-speed adjustable
Pump Inlet and Outlet	¼ inch FPT rear discharge.

APPENDIX B

Table B-1 Heat loss of heater for block “Pocofoam” of 75% porosity with FC-72

Block "Pocofoam" 75% with FC-72								
q (W)	q'' (W/cm ²)	T_W (°C)	T_{h3} (°C)	$T_W - T_{h3}$ (°C)	T_h (K)	k_{Cu} (W/m·K)	$q_{L(y)}$ (W)	$q_{L(y)}$ (%)
10	14.08	37.95	35.97	1.98	309.96	372.11	1.32	13.21
20	28.17	56.55	53.74	2.81	328.15	371.20	1.87	9.35
30	42.25	68.50	64.47	4.03	339.49	370.65	2.68	8.93
40	56.34	71.90	67.40	4.50	342.65	370.50	2.99	7.47
50	70.42	74.80	71.03	3.77	345.92	370.35	2.50	5.01
60	84.51	76.70	73.01	3.69	347.86	370.26	2.45	4.08
Average $q_{L(y)}$ in %								8.01

Table B-2 Heat loss of heater for block “Pocofoam” of 75% porosity with HFE-7000

Block "Pocofoam" 75% with HFE-7000								
q (W)	q'' (W/cm ²)	T_W (°C)	T_{h3} (°C)	$T_W - T_{h3}$ (°C)	T_h (K)	k_{Cu} (W/m·K)	$q_{L(y)}$ (W)	$q_{L(y)}$ (%)
10	14.08	36.30	33.54	2.76	307.92	372.21	1.84	18.42
20	28.17	42.65	39.45	3.20	314.05	371.90	2.13	10.67
30	42.25	46.00	43.53	2.47	317.77	371.71	1.65	5.49
40	56.34	50.50	46.98	3.52	321.74	371.51	2.34	5.86
50	70.42	53.40	50.04	3.36	324.72	371.37	2.24	4.47
60	84.51	56.60	52.95	3.65	327.78	371.22	2.43	4.05
Average $q_{L(y)}$ in %								8.16

Table B-3 Heat loss of heater for block “Kfoam” of 72% with FC-72

Block "Kfoam" 72% with FC-72								
q (W)	q'' (W/cm ²)	T_W (°C)	T_{h3} (°C)	$T_W - T_{h3}$ (°C)	T_h (K)	k_{Cu} (W/m·K)	$q_{L(y)}$ (W)	$q_{L(y)}$ (%)
10	14.08	33.55	31.83	1.72	305.69	372.32	1.15	11.48
20	28.17	43.65	40.65	3.00	315.15	371.84	2.00	10.00
30	42.25	63.85	59.43	4.42	334.64	370.88	2.94	9.80
40	56.34	71.20	67.53	3.67	342.37	370.52	2.44	6.10
50	70.42	73.35	69.58	3.77	344.47	370.42	2.50	5.01
60	84.51	77.50	73.09	4.41	348.30	370.24	2.93	4.88
Average $q_{L(y)}$ in %								7.88

Table B-4 Heat loss of heater for block “Kfoam” of 72% with HFE-7000

Block "Kfoam" 72% with HFE-7000								
q (W)	q'' (W/cm ²)	T_w (°C)	T_{h3} (°C)	$T_w - T_{h3}$ (°C)	T_h (K)	k_{Cu} (W/m·K)	$q_{L(y)}$ (W)	$q_{L(y)}$ (%)
10	14.08	35.30	33.56	1.74	307.43	372.23	1.16	11.61
20	28.17	43.30	40.68	2.62	314.99	371.85	1.75	8.73
30	42.25	50.10	46.67	3.43	321.39	371.53	2.28	7.62
40	56.34	52.80	49.32	3.48	324.06	371.40	2.32	5.79
50	70.42	54.95	48.20	6.75	324.58	371.37	4.49	8.99
60	84.51	58.85	53.68	5.17	329.27	371.14	3.44	5.73
Average $q_{L(y)}$ in %								8.08

Table B-5 Uncertainties data and calculations of T_w for block “Pocofoam” of 75% porosity with HFE-7000

q (W)	q'' (W/cm ²)	T_w (°C)					σ_{T_w} (°C)	δ_{T_w} (°C)	T_w (°C)	δ_{T_w}/T_w (%)
		N_1	N_2	N_3	N_4	N_5				
30	42.25	47.85	49.06	49.50	48.78	50.46	0.96	0.428	49.13	0.87
40	56.34	53.20	54.80	55.04	52.96	55.08	1.05	0.468	54.22	0.86
50	70.42	55.40	56.07	58.24	58.18	58.25	1.38	0.619	57.23	1.08
60	84.51	56.40	57.98	59.08	60.24	60.95	1.81	0.810	58.93	1.37
(δ _{T_w} /T _w) average (%)								1.05		

Table B-6 Uncertainties data and calculations of T_{sat} for block “Pocofoam” of 75% porosity with HFE-7000

q W	q'' W/cm ²	T_{sat} (°C)					$\sigma_{T_{sat}}$ (°C)	$\delta_{T_{sat}}$ (°C)	T_{sat} °C	$\delta_{T_{sat}}/T_{sat}$ (%)
		N_1	N_2	N_3	N_4	N_5				
30	42.25	43.11	42.86	43.14	43.86	44.34	0.62	0.276	43.46	0.63
40	56.34	47.79	48.57	49.45	47.03	48.94	0.96	0.428	48.36	0.88
50	70.42	48.66	50.90	51.54	50.76	52.36	1.38	0.615	50.84	1.21
60	84.51	48.66	51.04	52.96	51.85	53.44	1.89	0.844	51.59	1.64
(δ _{T_{sat}} /T _{sat}) average (%)								1.09		

Table B-7 Uncertainties data and calculations of q'' for block “Pocofoam” of 75% porosity with HFE-7000

q W	q'' W/cm ²	q'' (W/cm ²)					$\sigma_{q''}$ W/cm ²	$\delta_{q''}$ W/cm ²	q'' W/cm ²	$\delta_{q''}/q''$ (%)
		N_1	N_2	N_3	N_4	N_5				
30	42.25	43.66	40.85	43.66	40.85	43.66	1.54	0.690	42.54	1.62
40	56.34	57.75	54.93	57.75	54.93	57.75	1.54	0.690	56.62	1.22
50	70.42	71.83	69.01	71.83	69.01	71.83	1.54	0.690	70.70	0.98
60	84.51	85.92	83.10	85.92	83.10	85.92	1.54	0.690	84.79	0.81
(δ _{q''} /q'') average (%)								1.16		

Table B-8 Uncertainties data and calculations of T_w for block “Pocofoam” of 75% porosity with FC-72

q (W)	q" (W/cm ²)	T _w (°C)					σ_{T_w} (°C)	δ_{T_w} (°C)	T _w (°C)	δ_{T_w}/T_w (%)
		N ₁	N ₂	N ₃	N ₄	N ₅				
30	42.25	56.45	57.33	56.84	55.46	58.50	1.12	0.501	56.92	0.88
40	56.34	67.85	69.69	68.94	69.04	65.94	1.47	0.658	68.29	0.96
50	70.42	73.40	75.05	73.83	74.16	71.59	1.28	0.572	73.61	0.78
60	84.51	76.95	78.28	76.59	76.88	74.05	1.54	0.690	76.55	0.90
(δ_{T_w}/T_w) average (%)									0.88	

Table B-9 Uncertainties data and calculations of T_{sat} for block “Pocofoam” of 75% porosity with FC-72

q W	q" W/cm ²	T _{sat} (°C)					$\sigma_{T_{sat}}$ (°C)	$\delta_{T_{sat}}$ (°C)	T _{sat} °C	$\delta_{T_{sat}}/T_{sat}$ (%)
		N ₁	N ₂	N ₃	N ₄	N ₅				
30	42.25	58.84	57.90	56.58	56.02	56.45	1.17	0.524	57.16	0.92
40	56.34	63.94	65.58	67.65	65.55	61.59	2.25	1.008	64.86	1.55
50	70.42	68.46	70.54	69.77	68.83	70.84	1.04	0.464	69.69	0.67
60	84.51	70.54	72.43	72.52	71.02	72.58	0.97	0.432	71.82	0.60
$(\delta_{T_{sat}}/T_{sat})$ average (%)									0.93	

Table B-10 Uncertainties data and calculations of q'' for block “Pocofoam” of 75% porosity with FC-72

q W	q" W/cm ²	q'' (W/cm ²)					$\sigma_{q''}$ W/cm ²	$\delta q''$ W/cm ²	q'' W/cm ²	$\delta q''/q''$ (%)
		N ₁	N ₂	N ₃	N ₄	N ₅				
30	42.25	43.66	40.85	43.66	40.85	43.66	1.54	0.690	42.54	1.62
40	56.34	57.75	54.93	57.75	54.93	57.75	1.54	0.690	56.62	1.22
50	70.42	71.83	69.01	71.83	69.01	71.83	1.54	0.690	70.70	0.98
60	84.51	85.92	83.10	85.92	83.10	85.92	1.54	0.690	84.79	0.81
$(\delta_{q''}/q'')$ average (%)									1.16	

Table B-11 Uncertainties data and calculations of h_b for block “Pocofoam” of 75% porosity with HFE-7000

q (W)	$\delta_{h_b} / h_b $ (%)
30	1.95
40	1.74
50	1.89
60	2.29
(δ_{T_w}/T_w) average (%)	1.97

Table B-12 Uncertainties data and calculations of h_b for block “Pocofoam” of 75% porosity with FC-72

q (W)	$\delta_{hb} / h_b $ (%)
30	2.06
40	2.20
50	1.41
60	1.35
(δ_{T_w}/T_w) average (%)	1.73

Thermocouple 1

(a)

Thermocouple 2

(b)

Thermocouple 3

(c)

Thermocouple 4

(d)

Thermocouple 5

(e)

Thermocouple 6

(f)

Thermocouple 7

(g)

Thermocouple 8

(h)

(i)

(j)

Fig. B-1 (a) ~ (j) Plots of calibration for thermocouples 1 ~10