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Nucleate flow boiling enhancement on engineered three-dimensional porous metallic structures in FC-72

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ABSTRACT

Flow boiling heat transfer experiments were conducted in a closed-loop facility on engineered hollow structures using FC-72 as the working fluid. The use of Selective Laser Melting, an additive manufacturing technique, has allowed the fabrication of specialized 3D porous metallic structures with repeating unit cells of spherical voids and cylindrical connecting channels. Three porous substrates which consist of a small, large and gradient unit cells were fabricated. The porous substrates show superior flow boiling heat transfer as compared to a plain surface due to the increased number of nucleation sites and more vigorous flow mixing. For each substrate, the heat transfer coefficient shows an initial increasing trend from low to mid-range of the exit vapor quality and deteriorates thereafter when approaching the dry-out condition. Visualizations using a high-speed camera reveal that bubbly and churn flows were prevalent in the substrates and suggest the strong dominance of nucleate boiling over the convective mechanism. The Gradient-reverse substrate showed superior performance due to the channeling of the liquid-vapor mixture into smaller cross-sectional areas and the delay of dry-out for high vapor quality.

KEYWORDS: Nucleate flow boiling; porous media; 3D printing; Selective Laser Melting; FC-72

1. Introduction

Higher heat generation in electronic chips has posed challenges for effective cooling schemes. As conventional convective cooling methods by single-phase fluids are becoming increasingly difficult to meet the cooling demands, nucleate boiling has been shown to be capable of removing the high heat fluxes from electronic devices [1]. Often, when direct contact with the electronic devices is required, dielectric fluids such as FC-72, FC-87 and PF-5060 have been used due to their properties of electrical non-conductivity and chemical compatibility [2].

Nucleate boiling can be broadly classified under pool and flow boiling. Pool boiling works under the effects of buoyancy alone, whereas in flow boiling, a pump is used to drive the fluid in a closed-loop system. Due to the additional convective effect, flow boiling typically offers higher heat transfer rates than pool boiling.

For flow boiling, the main heat transfer mechanisms can be classified as nucleate boiling or convective dominated and is largely dependent on the vapor quality. At low vapor quality, nucleate boiling is likely to dominate, whereas the convective mechanism is more dominant at

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high vapor quality [3]. The flow pattern in two-phase flow is important for the understanding of heat transfer mechanisms. For plain tubes, the commonly observed flow patterns in increasing order of vapor qualities are, namely, bubbly, slug, churn, annular and misty flow [4].

Nomenclature

A	area (m ²)	<i>Greek Symbols</i>	
c_p	liquid specific heat (J/kg·K)	σ	surface tension (N/m)
G	mass flux (kg/m ² ·s)	ρ	density (kg/m ³)
h	heat transfer coefficient (W/cm ² ·K)	η	relative enhancement
h_{fg}	latent heat of vaporization (J/kg)		
I	current (A)	<i>Subscripts</i>	
$\dot{m}$	mass flow rate (kg/s)	al	aluminum
q	power (W)	ave	average
q''	heat flux (W/cm ²)	cu	copper
k	thermal conductivity (W/m·K)	in	inlet
L	length (m)	l	liquid
P	pressure (Pa)	out	outlet
R	thermal resistance (K/m ² ·W)	sat	saturated
T	temperature (°C)	TC	thermal compound
V	voltage (V)	v	vapor
$\dot{V}$	volume flow rate (m ³ /s)	w	wall
x	vapor quality		

1.1. Flow boiling enhancement techniques

Many researchers have used surface enhancement techniques to enhance the heat transfer coefficient in flow boiling. A thorough review of flow boiling in dielectric fluids using surface enhancement techniques was recently conducted by Leong et al. [2]. Finned surfaces in flow boiling were investigated by some researchers. Ma et al. [5] used a dry-etching method to produce micro-pin-fins for flow boiling of FC-72. In their studies, different fluid velocities, liquid subcooling and fin heights were investigated. All finned-surfaces showed considerable heat transfer enhancements as compared to a benchmark plain surface. Krishnamurthy and Peles [6] fabricated a single row of inline micro-pin-fins in a microchannel for subcooled and saturated flow boiling in HFE-7000. A higher heat transfer coefficient was obtained as compared to a plain microchannel due to the large surface area and fluid mixing. Visualizations showed different flow patterns of isolated bubbles, bubbles interacting and annular regions.

Porous coated surfaces were investigated by other researchers. Rainey et al. [7] investigated the effect of microporous coated surfaces for flow boiling in subcooled FC-72, whereas Ammerman and You [8] used microporous coated surfaces in FC-87. Both studies show that the coated surfaces resulted in boiling incipience at lower wall superheats and increased heat transfer coefficients. On the other hand, Sun et al. [9] sintered copper particles of 20, 50 and 120 μm sizes onto horizontal minichannels and found that heat transfer enhancements of between 7 and 10 times as compared to uncoated channels can be achieved.

The potential of using porous foams in nucleate boiling has also been investigated. Porous foams typically possess high surface-area-to-volume ratio and can be made of high thermal conductivity materials such as aluminum, copper and graphite. Various researchers have used commercial porous foams for pool boiling applications. Xu et al. [10] studied the pool boiling of acetone using copper foams of different porosities and pore densities. Coalescence of bubbles within the foam structure was observed to form ‘cage-shaped’ bubbles which were larger than the pore diameter. In another study, Xu et al. [11] postulated that while the heat transfer in pool boiling is enhanced by the larger surface area in a porous foam, the higher bubble escape resistance countered the gain. In order to reduce the bubble escape resistance, Xu and Zhao [12] used copper foams with V-shaped grooves and showed that the critical heat flux could be delayed.

Kim et al. [13] used copper foams of 10 and 20 pores per inch (PPI) for flow boiling in water and FC-72. It was shown that the increase in mass flux from 20 to 48 kg/m²·s yielded an enhanced heat transfer coefficient of 15% although a further increase to 72 kg/m²·s did not yield further enhancement. A larger pore size distribution seemed to favor the release of vapor at high heat flux. On the other hand, Zhao et al. [14] performed flow boiling experiments in metal foam tubes using R-134a as the refrigerant. It was concluded that the heat transfer coefficients can be doubled when the metal foam cell size decreases from 20 PPI to 40 PPI. The enhancement was attributed to increased surface area and better flow mixing within the foamed channel. Madani et al. [15] used metallic copper foams of 36 PPI in a flow boiling study using n-pentane as the fluid. It was found that the use of metallic foams could increase the heat transfer coefficients by 2 to 4 times for low exit vapor quality although the enhancement decreases as the exit quality increases. Pranoto and Leong [16] used graphite foams for flow boiling studies. Flow boiling enhancements of up to 2.5 times were achieved and this was attributed to the high thermal conductivity of graphite and increased nucleation site density.

From the above literature review, the use of commercial foams for flow boiling is favored due to their large internal surface areas and their ability to dissipate heat in a three-dimensional manner. However, there are associated issues of poor inter-connectivity in the pores and high pressure drop across the foams. Due to manufacturing constraints, specialized pore structures are difficult to fabricate. However, this could be overcome with the process of additive manufacturing.

1.2. Selective Laser Melting technique

Additive Manufacturing (AM), commonly known as 3D printing, is an emerging field of manufacturing due to its design freedom and ability to produce parts of complex internal features. Selective Laser Melting (SLM) is a powder-bed fusion technology which is a branch of AM that allows the fabrication of dense metallic components. SLM uses a high-power fiber laser to melt metallic powder in a layer-by-layer fashion to generate a 3D substrate from a Computer-Aided Design (CAD) model [17]. The ability to fabricate near-net shape parts eliminates the need for a mold which is required in conventional casting techniques.

In recent years, SLM has been employed to fabricate heat transfer devices due to its lower cost-to-complexity ratio and lower production time as reported by Jafari and Wits [18]. Commonly used materials are aluminum alloys such as AlSi10Mg and Al6061 because of their high thermal conductivities. For single-phase studies, Wong et al. [19] and Ho et al. [20] fabricated pin fin arrays of airfoil geometry using SLM for air convective heat transfer studies. They have shown that the airfoil fins possess superior heat transfer performance with lower pressure drop than cylindrical fins due to their streamline shape. Neugebauer et al. [21] fabricated a water cooler unit and demonstrated the ability of using SLM to build delicate and complex internal geometries. On the other hand, Ho and Leong [22] fabricated porous inserts for water-cooled cold plate studies and showed that heat transfer enhancement of up to 383% can be obtained as compared to an empty channel.

Numerous two-phase studies have been conducted using substrates fabricated using SLM. The ability to manufacture complex and customized wick structures for heat pipes is an advantage of SLM. Esarte et al. [23] demonstrated the ability to manufacture optimized wick design for a two-phase cooling system, whereas Ameli et al. [24] managed to customize the wick structures with various thicknesses, porosities and permeabilities at different regions of a heat pipe. For nucleate boiling studies, Ho et al. [25] investigated pool boiling enhancement in saturated FC-72 using micro-fin and micro-cavities fabricated using SLM, whereas Wong and Leong [26] produced lattice substrates of different unit cell sizes and substrate heights for pool boiling studies. The above literature review highlights the potential of using SLM to produce specialized substrates for heat transfer applications and especially for two-phase studies.

1.3. Aims of study

From the thorough literature review conducted, there is a research gap in the use of specialized three-dimensional structures for flow boiling enhancement. In this study, the authors aim to fabricate engineered porous 3D substrates using the SLM technique for flow boiling in FC-72. The substrates possess repeating cells of hollow spherical features to promote nucleation and reduce coalescence of bubbles. The flow boiling heat transfer process will be studied in a two-phase closed-loop rig and high-speed visualization will be used to understand the flow pattern evolution within the substrates.

2. Substrate preparation and characterization

2.1. Description and design concept of porous hollow substrates

Porous substrates with engineered hollow structures were designed and fabricated in this study. Each substrate consists of repeating unit cells with an internal spherical void and six connecting cylindrical through-holes. Each unit cell is fixed as a cube of 5-mm length on all sides.

A total of three substrates were designed in SolidWorks. Two of the substrates were populated with uniform unit cells but with different porous parameters and were labeled as Sphere-small and Sphere-large substrates. The Sphere-small substrate has a connecting hole of 2-mm diameter and an inner spherical void of 2-mm radius, whereas the Sphere-large substrate has a connecting hole of 2.8-mm diameter and an inner spherical void of 2.8-mm radius, as shown

in Fig. 1(a-b). A third substrate, labeled as Gradient, consists of unit cells of increasing spherical void radii varying from 2-mm to 2.8-mm in a length-wise manner. A length-wise cutout of the Gradient substrate is shown in Fig. 1(c) for illustration purpose. For each substrate, the unit cells were arranged to form geometries of 80 mm (length) by 60 mm (width) by 10 mm (height), which correspond to a total of 16 by 12 by 2 unit cells. The fabricated substrates are shown in the next section. The parameters of the substrates are summarized in Table 1. The porosity is defined as the ratio of void to actual bulk volume occupied by the substrates.

Fig. 1. Illustration of unit cells of different porous substrates

Table 1. Summary of substrate characteristics

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

Fig. 2. shows a schematic of the cross-section viewed from the front of the substrate. The flow direction is from left to right and the heat source is located at the bottom. There are two major considerations to the concept of this design, which are to increase the surface area to promote nucleation within the spherical void, and to reduce the coalescence of bubbles as the fluid flows downstream. The solid matrix helps to conduct the heat upwards. The black arrows illustrate

the various possible flow paths for the liquid and vapor within the hollow channels. Due to the interconnecting channels within the substrate, fluid mixing is likely to enhance the heat transfer too. This design overcomes the problem of flow in conventional metallic foams where random pore distribution and poor inter-connectivity could pose difficulties.

Fig. 2. Schematic of front cross-section of substrate

2.2. Selective Laser Melting facility and procedure

The SLM 250 HL (SLM Solutions GmbH) facility was employed for the fabrication of substrates in the present study. It is housed in the Future of Manufacturing Laboratory 1 under the Singapore Centre for 3D Printing (SC3DP) at Nanyang Technological University (NTU), Singapore. In this study, AlSi10Mg powder was used because of its high thermal conductivity of up to 170 W/m·K for as-built products [27]. The powder has a distribution size of 20 - 63 μm and possesses a spherical geometry.

During the manufacturing process, the powder was first distributed evenly on a substrate base plate. The machine uses a Gaussian distributed Yb:YAG laser of maximum power of 400 W and laser beam spot size of 80 μm to melt the metal powder. During operation, the build chamber was flushed with inert argon gas to attain an oxygen level of less than 0.2 % to minimize oxidation and combustion of powder. For each layer, the laser scanning melted the powder according to a pre-programmed pathway that was defined in a CAD file. The powder layer was rapidly melted and then solidified to form a thin solid layer. The powder bed was then lowered by one layer and redeposited. Laser scanning subsequently melted and bonded the new layer to the previous layer. This process was repeated until the entire three-dimensional part was formed.

2.3. Substrate characterization

The fabricated substrates are shown in Fig. 3. After fabrication, residual powder was removed from the substrates which were then separated from the metal baseplate using the wire-cutting technique. The internal hollow features were observed to be formed in good order and accuracy. The bulk external dimensions were measured using a Vernier caliper and were found to be accurate within ± 0.1 mm. Further measurements were obtained using a light optical microscope (Olympus SZX7). The diameters of 20 random external cylindrical holes were measured for each Sphere-small and Sphere-large substrate. The average values when

compared to the design showed a small deviation of ± 0.1 mm. Thus, it has been demonstrated that SLM is able to produce three-dimensional substrates with complex and intricate features that is otherwise difficult to achieve using conventional subtractive methods.

Fig. 3. Substrates fabricated using SLM for (a) Sphere-small, (b) Sphere-large, (c) Gradient and (d) Top views of all substrates

3. Flow boiling facility and data reduction

3.1. Experimental flow boiling facility

A two-phase closed-loop experimental flow boiling facility was developed to study the nucleate flow boiling performance of the substrates as shown in Fig. 4. The facility consists of five main components, namely, the fluid reservoir, gear micro-pump, preheater section, evaporator test section and condenser. A funnel was located at the reservoir to introduce the fluid before the start of experiment. Two cartridge heaters were installed in the reservoir to warm up the fluid. A gear micro-pump (Cole-Parmer R-73011-18) was used to circulate the fluid in the system. It can be varied accurately between 0.2 and 5.0 L/min. A flow meter (McMillan S-112) was installed after the micro-pump to measure the volumetric flow rate. A preheater section was installed before the evaporator to allow fine control of temperature of the fluid before it enters the test section. The test substrates were embedded inside the evaporator for heat transfer testing and a heat simulator was located under the evaporator to control the heat input. The details of evaporator are elaborated in Section 3.2. The condenser was a plate-type heat exchanger through which water at 20°C was circulated by a chiller to condense the

liquid-vapor mixture before it was returned to the reservoir to complete the cycle. “Swagelok” stainless steel tubings with outer diameter of 3/8 inch (9.5 mm) were used to connect the different components and insulation was provided around the tubing to minimize heat losses.

The temperatures of the fluid were monitored using four type-K thermocouples at various locations. The first thermocouple was installed in the reservoir, and subsequent ones in the flow channel before the pre-heater section, and before and after the evaporator section, as shown in Fig. 4. The pressures were monitored at various locations. A pressure gage was installed at the reservoir. In the flow channel, a pair of absolute pressure transducers (Cole-Parmer EW-68074-08) and a differential pressure sensor (Omega PX409-005DWUV) were installed before and after the evaporator test section.

Fig. 4. Schematic of experimental flow boiling facility

3.2. Evaporator test section

The evaporator section consists of a rectangular flow channel with dimensions of 160 mm (length) by 60 mm (width) by 12 mm (height), as shown in Fig. 5. An expanding and a contracting vane before and after the rectangular flow channel allow smooth convergence and divergence of the flow. The flow channel is made of an aluminum base and a polycarbonate top cover to allow observation of the boiling process during the experiments. The polycarbonate top cover is secured to the base and sides with 24 screws and gaskets and no leakage was observed during the experiments.

The heat simulator is attached below the rectangular flow channel. It contains a copper block to provide a heating surface of 80 mm by 60 mm with a thickness of 10 mm. Eight cartridge

rod heaters were embedded in the copper block and a power transformer allowed the power to the rod heaters to be varied accurately up to 3 kW. The copper block is then encompassed by a 10-mm thick Teflon layer to provide insulation. A thin layer of thermal grease “Omega-therm-201” was applied between the copper block and the aluminum base of rectangular channel to ensure good thermal contact.

Fig. 5. Illustration of evaporator section

As mentioned previously, the test substrates were fabricated to the dimensions of 80 mm by 60 mm by 10 mm. During each experiment, a substrate was fitted securely in the evaporator section. A silicone rubber was embedded at the top of the substrate to minimize fluid bypass. A thin layer of thermal compound (Artic MX-4) was applied between the test substrate and aluminum base to reduce contact resistance. Surface temperatures of the substrate were measured using six Type-K thermocouples which were inserted through drilled holes in the aluminum base. They were located 3 mm beneath the substrate surface and distributed evenly throughout the surface. The locations of the thermocouples are shown in Fig. 6 which illustrates the detailed schematic of the evaporator with the test substrate embedded.

Fig. 6. Detailed schematic diagrams of evaporator section and location of thermocouples

3.3. Properties of fluid – FC-72

FC-72 from 3M Ltd was selected as the phase-change fluid due to its high dielectric strength and electrical non-conductivity. The fluid has a boiling point of 56°C under atmospheric condition. It is a dielectric fluid which has been widely used for electronic cooling applications. FC-72 is also thermally and chemically stable, non-flammable, and practically non-toxic and have low Global Warming Potential. The thermophysical properties of FC-72 are summarized in Table 2.

Table 2. Thermophysical properties of FC-72 at 1 atm [28]

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

3.4 Experimental procedure and data reduction

Before the start of each experiment, heaters in the reservoir were used to boil the liquid vigorously for 30 min and the degassing valve was opened to purge any non-condensable gas. After the degassing procedure, the micro-pump was switched on to allow the fluid to circulate through the entire facility. The pre-heater section was then tuned to heat the fluid to 56°C and careful management was taken to avoid boiling before it enters the evaporator section. Only after the temperature of the fluid has stabilized was the power supplied to the heat simulator at the test section at increasing intervals of between 50 W and 150 W. For each interval, it takes about 15 to 20 min for the system to reach quasi-steady conditions, whereby the temperatures recorded do not fluctuate more than 0.2°C within a 5-min interval. The temperatures and pressures were recorded using a MX100 data acquisition system. For each set of readings at steady-state, time-averaging values were recorded within 1 min with sampling rate of 2 Hz and total of 120 readings to minimize data uncertainty.

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

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

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

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

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

Combining Eqs. (1) and (2), the rate of heat loss can be determined using Eq. (3),

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

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

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

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

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

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

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

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

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

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

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

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

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

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

Finally, the local vapor quality can be calculated from an energy balance analysis as shown in Eq. (10).

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

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

The measurement uncertainty for the flow meter is 1% FS (0.02 L/min). The uncertainties of the absolute pressure transducer and differential pressure sensor are 0.13 % FS (22 Pa) and 0.08% FS (28 Pa), respectively. The uncertainties of Type-K thermocouples and power supply yields are ± 0.3 °C and 1% FS (10 W), respectively. The main uncertainties are contributed by the heat flux and thermocouple readings, while the uncertainties from the pressure sensors are negligible. Using the method described by Taylor [29], the uncertainty of the heat transfer coefficient (h_{ave}) was determined from Eq. (11). The uncertainty in h_{ave} was determined to be higher in the low heat flux region with a maximum of $\pm 15\%$ and decreased rapidly to $\pm 5\%$ at heat fluxes above 8 W/cm².

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

3.5 Design of experiments

The empty channel, which was labeled as Plain surface, was used as a benchmark in the experiments. For the Gradient substrate, it can be mounted in the evaporator with the gradient structure varying from small-to-big (forward) or big-to-small (reverse) with respect to the flow direction. Thus, it has been labeled as Gradient-forward and Gradient-reverse substrates for simplicity. Together with the Sphere-small, Sphere-large and plain surface, a total five types of substrates have been tested. For each substrate, experiments for three mass fluxes of 20, 30 and 40 kg/m²·s were conducted to study the effect of mass flux. The experiment for each condition was conducted twice to ensure repeatability. The summary of the design of experiments is shown in Table 3. The experiments were conducted for heat fluxes between 2.0 and 20.0 W/cm².

Table 3. Summary of design of experiments

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

4. Results and discussion

4.1 Effect of mass flux

The effect of mass flux on the heat transfer coefficients for Sphere-small and Sphere-large substrates are shown in Fig. 7. Both figures show a common trend of the heat transfer coefficient increasing with heat flux up to a peak value before decreasing. For both figures, it can be observed that at low heat fluxes, the curves for all the mass fluxes collapse to a single curve. This indicates that the heat transfer coefficient is independent of mass flux within this range and suggests that nucleate boiling is the dominant heat transfer mechanism. At higher heat fluxes, the heat transfer performance starts to diverge for different mass fluxes, indicating the higher contribution of convective heat transfer mechanism. However, the transitions occur at different heat fluxes for the two substrates, which correspond to about 12 and 16 W/cm² for the Sphere-small and Sphere-large substrates, respectively. The earlier transition for Sphere-small substrate is likely due to the smaller cross-sectional areas which cause the flow to accelerate more within the flow channels, and hence, increase the contribution of the convective heat transfer mechanism.

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

4.2 Effect of outlet vapor quality

In Fig. 8, the effect of outlet vapor quality on heat transfer coefficient is shown for the Sphere-small and Sphere-large substrates. For both substrates, the heat transfer coefficient curves show a peak value before declining at high outlet vapor quality. At low mass flux of $20 \text{ kg/m}^2\cdot\text{s}$, the outlet vapor quality approaches the dry-out vapor quality of $x = 1.0$. Typically, the heat transfer performance rapidly deteriorates near dry-out due to transition to misty flow. Comparing both substrates, the deterioration for the Sphere-large substrate at high vapor quality is more pronounced as observed from the gradient of the curves. From both figures, the peak values of the heat transfer coefficient occur in the mid-range of exit vapor qualities such that $0.3 < x < 0.6$. Therefore, for practical consideration, the mass flux should match the heat flux level where the corresponding vapor quality would be in that range for optimal heat transfer performance. However, it is noteworthy that the optimal range varies for different substrates.

Fig. 8. Effect of outlet vapor quality on heat transfer coefficients for (a) Sphere-small and (b) Sphere-large substrates.

4.3 Effect of substrate type

Fig. 9 shows the effect of substrate type on the heat transfer performance at mass fluxes of 20 and $40 \text{ kg/m}^2\cdot\text{s}$. Compared to the benchmark Plain surface, all hollow porous substrates showed significant heat transfer enhancements. The enhancements could be attributed to the internal structure of the spherical hollow cells, which provides larger surface areas and more favorable nucleation sites as compared to the Plain surface. Also, as the liquid-vapor mixture flows within the spherical hollow cells, the alternate expanding and contracting cross-sections is likely to intensify the flow mixing. The phenomenon of flow mixing will be discussed in the subsequent section on the visualization study.

For both figures, the Gradient-reverse substrate demonstrates superior heat transfer coefficients as compared to other substrates across the entire heat flux range. Also, while the other substrates experience decreasing heat transfer coefficients at higher heat fluxes, the Gradient-reverse substrate shows a delay in the deterioration to a higher range of heat flux. For example, in Fig. 8(a) for mass flux of $20 \text{ kg/m}^2\cdot\text{s}$, the Gradient-reverse substrate only show deterioration

in the heat transfer coefficient above 18 W/cm^2 , whereas the other substrates tend to show deterioration from 12 W/cm^2 onwards.

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

Fig. 9. Effect of substrate type at mass fluxes of (a) $20 \text{ kg/m}^2\cdot\text{s}$ and (b) $40 \text{ kg/m}^2\cdot\text{s}$.

4.4 Pressure drop analysis

Fig. 10(a) shows the pressure drop curves of the different substrates at adiabatic condition. The data were collected for each substrate prior to heat input at the evaporator section. The pressure drops were obtained at various mass fluxes for comparison. In general, the pressure drop increases at higher mass flux due to increasing single-phase frictional losses. Logically, the Plain surface shows the lowest pressure drop due to the lack of obstruction in the flow. For the other substrates, the pressure drops correspond well to the porosities of the substrates. The Sphere-small substrate has the highest pressure drop due to the smallest cross-sectional flow area. Both gradient substrates presented very similar pressure drops despite the different directions of variation in the unit cell sizes.

Fig. 10(b) shows the pressure drops at various heat fluxes during the actual experimental runs. As compared to Fig. 10(a) at adiabatic condition, the magnitude of the two-phase pressure drop for each substrate is noted to be several times higher. Under conditions of flow boiling, the formation of vapor causes the two-phase fluid to accelerate and contribute to the accelerative pressure drop. In Fig. 10(b), the pressure drop generally increases at higher heat fluxes due to the corresponding higher outlet vapor quality. Among the structured substrates, although the Sphere-large and Sphere-small substrates retained the lowest and highest pressure drop, respectively, it is interesting to note that the pressure drops of the gradient substrates deviate from each other. The Gradient-reverse substrate consistently shows a higher pressure drop than

that of the Gradient-forward substrate. Due to the gradually expanding channel size in Gradient-forward, it is likely to relieve the accelerative pressure drop contribution.

Fig. 10. Pressure drop characteristics at (a) adiabatic condition and (b) flow boiling condition

The higher heat transfer performance achieved by the porous substrates comes at the penalty of higher pressure drop, which translates to the need for higher pumping power. To relate the heat transfer enhancement to the increased pressure drop experienced by the various substrates as compared to the plain surface at the same mass flux, the relative enhancement ratio (η) is proposed in Eq. (12).

$$\eta = \frac{(h/h_{ref})}{(\Delta P/\Delta P_{ref})} \quad (12)$$

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

Fig. 11 shows the relative enhancement ratio of each substrate for the various mass fluxes. In general, for individual substrate, the higher mass flux corresponds to a higher relative enhancement ratio, except for the Sphere-large substrate where the mid mass flux performed the best. If pumping power is an important concern, the Sphere-large substrate would be favored. Otherwise, the Gradient-reverse substrate shows the highest absolute heat transfer coefficient as discussed in the previous section and has reasonably high relative enhancement ratio as shown in Fig. 10.

Fig. 11. Relative enhancement ratios of porous substrates at different mass fluxes

4.5 Comparison with literature

Due to the diverse types of fluid, mass fluxes and configurations of setup used by other researchers in flow boiling experiments, it is challenging to draw out meaningful comparisons. The authors have selected the works by Madani et al. [15] and Kim et al. [13] for comparisons due to the similarity in range of heat and mass fluxes tested.

Madani et al. [15] soldered metallic copper foam of 36 PPI into a tube for their flow boiling investigation. The fluid used was n-pentane with a boiling point of 36°C which is lower than that of FC-72. The latent heat of vaporization of n-pentane is 357 kJ/kg, which is about 4 times the value of 88 kJ/kg for FC-72. It is shown in Fig. 12 that an increase of mass flux from 36 to 57 kg/m²·s causes a corresponding increase of h . However, at high heat fluxes, both the heat transfer coefficients of n-pentane and FC-72 decrease. According to Madani et al., the decrease could be attributed to the coalescence of bubbles and the resistance of the metallic foam to bubble movement

Kim et al. [13] investigated flow boiling of metallic foams using FC-72. In their case, subcooled boiling was performed. For the mass flux of 20 kg/m²·s, nucleate boiling was initiated at lower heat flux range. However, for the case of mass flux of 40 kg/m²·s, the heat transfer coefficients increase at increasing heat flux due to transition from subcooled to nucleate boiling. The peak heat transfer coefficient increased by about 10% by increasing mass flux from 20 to 48 kg/m²·s.

Fig. 12. Comparison of results with literature

5. Flow pattern analysis

For this section, visualizations were conducted using a high-speed camera (Photron FASTCAM 1024 PCI) for the Sphere-large and Sphere-small substrates. To observe the flow pattern within the substrates clearly, milling was applied at the edge of each substrate for about 1-mm depth to reveal more of the internal features without major disruption of the flow structure. High-speed videos were recorded for different heat fluxes and at two locations of each substrate, which corresponds to $L = 30$ mm and $L = 60$ mm downstream of the leading edge. For each video, capturing frequencies of between 2000 Hz and 6000 Hz and 600 images were obtained for analysis.

5.1 General flow patterns

The presence of spherical hollow unit cells used in this study caused the flow patterns to deviate from that of plain tubes. The general flow patterns of the Sphere-large substrate at various heat fluxes are shown in Fig. 13. Each figure focuses on the flow pattern within two adjacent unit cells. At low heat flux and correspondingly low vapor quality, bubbly flow was observed as shown in Fig. 13(a). As the heat flux and vapor quality increased, the bubbles coalesced and increased in size. The flow also became more turbulent and is classified as churn flow in Fig. 13(b). Flow mixing intensified at increasing vapor quality and the flow is classified as wavy-churn in Fig. 13(c). Finally, when the vapor quality was sufficiently high, partial dry-out was observed and was characterized by lower bubble count in a dominant vapor medium, as shown in Fig. 13(d).

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

The general flow patterns of the Sphere-small substrate are shown in Fig. 14. At low vapor quality as in Fig. 14(a), the flow pattern was visibly more vigorous than that of the Sphere-large substrate at the same vapor quality and thus is classified as bubbly-churn flow. It is likely that the smaller cross-sectional area in the Sphere-small substrate caused the bubbles to be more confined and resulted in more fluid mixing. Subsequently, the flow intensified at higher vapor qualities and evolved to churn and wavy-churn as shown in Fig. 14(b) and (c), respectively. It is interesting to note that the relative bubble sizes were smaller than that of Sphere-large substrates, although the flow patterns were similar. At high vapor quality, however, minimal dry-out was observed and the wavy-churn flow pattern was sustained as shown in Fig. 14(d). However, the bubble sizes were observed to increase generally at higher vapor quality due to coalescence.

Fig. 14. Bubble regimes for different heat fluxes at mid-section for the Sphere-small substrate ($G = 40 \text{ kg/m}^2\cdot\text{s}$, $L = 60 \text{ mm}$)

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

5.2 Transient behavior of flow patterns at different vapor qualities

Further analyses were conducted on the various flow patterns which occurred in the Sphere-small and Sphere-large substrates. Fig. 15 shows the transient behavior of the flow pattern in the Sphere-small substrate at low heat flux and vapor quality, which corresponds to bubbly-

churn flow. The images were taken at 5-ms intervals. The figure on the left shows the actual flow pattern, while the figure on the right shows the schematic for better clarity. For all images, the flow direction is from the left to the right.

At the beginning of the flow in Fig. 15(a), a plug bubble was observed to emerge from the left and to move downstream to the right. As the plug bubble approached the constriction point connecting two adjacent spherical unit cells, it underwent elongation and broke up into two smaller bubbles in Fig. 15(b). Upon emerging from the constriction point, the frontal bubble began to re-form into a more spherical geometry in Fig. 15(c). The next wave of bubbles underwent a similar process of elongation, breaking up and squeezing through the constriction points in Fig. 15(d). The presence of cylindrical connecting channels as constriction points between the spherical hollow cells helped to reduce the coalescence of bubbles in the Sphere-small substrate.

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

Fig. 15. Transient behavior of bubbly-churn flow pattern in the Sphere-small substrate ($q'' = 8 \text{ W/cm}^2$, $G = 30 \text{ kg/m}^2\cdot\text{s}$, $L = 30 \text{ mm}$)

Fig. 16 shows the transient behavior of flow pattern in the Sphere-large substrate at low heat flux and vapor quality. In Fig. 16(a), a large bubble was observed to approach the constriction point. In Fig. 16(b), the frontal portion of the bubble managed to squeeze through the constriction point and necking occurred which resulted in the break-out into two bubbles. However, as the cross-sectional area of the constriction is much larger than that of the Sphere-small substrate, the bubble did not elongate as much to squeeze through. In fact, it appears that there was mass transfer from the preceding bubble to the frontal bubble in Fig. 16(c). Finally, in Fig. 16(d), the bubbles coalesced and re-form into a larger bubble after passing through the

constriction. Therefore, the larger spherical hollow void and wider constriction point allowed the bubbles to coalesce more easily in the Sphere-large substrate.

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

Fig. 16. Transient behavior of bubbly flow pattern in the Sphere-large substrate ($q'' = 8$ W/cm², $G = 30$ kg/m²·s, $L = 30$ mm)

Fig. 17 shows the transient behavior of flow pattern in the Sphere-small and Sphere-large substrates in the mid heat flux and vapor quality level. Churn flow was present for both substrates, which was characterized by the formation of rapidly coalescing bubbles and chaotic movements of bubbles. In this flow pattern, the heat transfer process is highly nucleate boiling dominated and corresponds to the high heat transfer coefficients as discussed in Section 4.2. It is apparent that the relative sizes of the bubbles for the Sphere-large substrate are larger than that of the Sphere-small substrate. Due to the constriction of the internal geometries, the bubble sizes tend to be of the same order of magnitude as the connecting cylindrical channels.

Fig. 17. Churn flow inside Sphere-small and Sphere-large substrates ($q'' = 8 \text{ W/cm}^2$, $G = 30 \text{ kg/m}^2\cdot\text{s}$, $L = 30 \text{ mm}$)

Fig. 18 shows the transient flow patterns of the Sphere-large and Sphere-small substrates at high vapor quality. For both substrates, some dry-out was observed but the dry-out was more prevalent for the Sphere-large substrate. Although the flow was likely to transit to convective heat transfer mechanism at high vapor quality, it appears that the contribution of nucleate boiling mechanism was crucial throughout the entire range of vapor quality due to the lack of annular flow formation. Therefore, the dry-out flow condition is not favorable for the current study, which corresponds to the rapidly declining heat transfer coefficients of the Sphere-large substrates as discussed in Section 4.2.

Fig. 18. Different flow patterns inside the Sphere-small and Sphere-large substrates ($q'' = 16$ W/cm², $G = 40$ kg/m²·s, $L = 60$ mm)

From the analyses of the different flow patterns, it is apparent that a smaller cross-sectional area is more favorable for maintaining high nucleate boiling heat transfer due to less bubble coalescence and delay of the dry-out condition.

The superior performance of the Gradient-reverse substrate can be deduced from the evolution of flow patterns for the Sphere-small and the Sphere-large substrates. Fig. 19 shows an illustration of the likely flow pattern in the Gradient-reverse substrate. Due to the decreasing unit cell size in the flow direction, the liquid volume fraction could be maintained despite vaporization along the flow direction. The changes in the local vapor quality also cause the flow to be highly turbulent, which could cause the flow pattern to be maintained as churn-wavy and delay the occurrence of dry-out.

Fig. 19. Illustration of flow pattern within Gradient-reverse substrate

5.3 Effect of mass fluxes

The flow patterns of both the Sphere-small and Sphere-large substrates at the mid heat flux of 8 W/cm^2 and various mass fluxes are shown in Fig. 20. For both the substrates, it can be observed that bubbly flow pattern with some stratification was dominant at low mass flux of $20 \text{ kg/m}^2\cdot\text{s}$. Stratification also tends to occur in the plain tube at low mass flux when the flow is not turbulent enough to cause much mixing. Generally, stratification should be avoided due to poorer flow mixing and the inability to utilize the entire substrate for heat transfer. At higher mass fluxes, churn and churn-wavy flows become dominant for both the substrates.

Fig. 20. Flow patterns of Sphere-large and Sphere-small substrates at various mass fluxes ($q'' = 8 \text{ W/cm}^2$, $L = 30 \text{ mm}$)

5.4 Summary of bubble dynamics in porous structure

Due to the unique geometry of the spherical hollow structures used in this study, the general flow patterns observed were bubbly, churn, wavy-churn and partial dry-out for low to high range of vapor quality. The alternating spherical chamber and cylindrical through-holes allow good fluid mixing and generate a highly turbulent flow, which helps to enhance heat transfer. Also, compared to plain tubes, annular flow was not observed even at high vapor quality due to the intense flow mixing which prevented the formation of orderly liquid film at the surface. This indicates that the nucleate boiling contribution is significant for the entire range of vapor quality.

6. Conclusions

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

The following conclusions are derived from this study:

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

Conflict of interest

None.

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