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Experimental study of flow boiling of FC-72 in fractal-like flow channels

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ABSTRACT

In this study, symmetrical and dichotomous fractal flow channel designs with various configurations in terms of the number of channels being split into smaller channels, were investigated experimentally for their flow boiling heat transfer performance with FC-72 as the working fluid. A multi-channel parallel channel design depicted as *Parallel* was used as a benchmark. The channels were fabricated by Selective Laser Melting (SLM). Three mass fluxes were studied, viz., 600, 900 and 1200 kg/m²·s. The peak heat transfer coefficient and pressure drop were investigated. High-speed visualization studies were also employed to examine the fluid flow characteristics in the channels. The fractal flow channels were found to perform comparably with the *Parallel* channel at low mass fluxes but achieved higher heat transfer coefficients at higher mass fluxes, with the design $n = 2$, i.e., channel splitting into smaller channels twice, being the highest. The pressure drop of the fractal flow channel designs is higher than the *Parallel* channel due to the multiple bifurcations experienced, with $n = 4$, i.e., channel splitting into smaller channels four times, experiencing the highest pressure drop for all mass fluxes. Visualization studies show flow reversals in the *Parallel* channel especially at higher mass fluxes, which suggest no enhancement in the heat transfer coefficient. The fractal flow channels, however, do not experience any flow reversal which could be attributed to the acceleration of the two-phase flow after each bifurcation, which in turn increases the inertia force to overcome the force from vapor expansion. The visualization studies also suggest that increasing the complexity does not result in better flow boiling heat transfer performance. This is due to the multiple flow separations which resulted in no two-phase heat transfer interaction and thus, under-utilization of the entire channel surface area for heat transfer.

KEYWORDS: Flow boiling; fractal flow channels; minichannels; flow reversals; Selective Laser Melting

Nomenclature*Symbols*

A	Area [m ²]
D	Diameter of channel [m]
D'	Characteristic dimension [m]
F'_M	Force per unit length due to evaporation of vapor [N/m]
F'_I	Inertial force per unit length [N/m]
G	Mass flux [kg/m ² ·s]

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h	Heat transfer coefficient [W/m ² ·K]
h_{fg}	Latent heat of vaporization [J/kg]
k	Thermal conductivity [W/m·K]
L	Length of channel [m]
$\dot{m}$	Mass flow rate [kg/s]
n	Branch level [-]
$\dot{Q}$	Heat transfer rate [W]
$\dot{q}$	Heat flux [W/m ²]
T	Temperature [K]
t	Thickness [m]

Greek symbols

β	Length ratio
γ	Diameter ratio
ρ	Density

Subscripts

Al	Aluminum
FB	Flow boiling
FC	Flow channel design
L	Heat loss
l	Liquid phase
m	Position of thermocouple
n	Branch level
sat	Saturated condition
$supp$	Heat supplied
TC	Thermal compound
v	Vapor phase
w	Channel surface wall

1. Introduction

Thermal management schemes have been growing in importance as more powerful and miniature electronic devices are being developed. Due to the increased power of electronic devices, current power densities could reach as high as 300 W/cm² at hot-spots [1, 2] and were even predicted to go as high as 1000 W/cm² in 2015 [3]. Such high power densities render single-phase cooling schemes inadequate [4]. Consequently, two-phase thermal management is one of the viable options to manage the ever-increasing heat dissipation by using both the sensible and latent heat of the coolant to dissipate the high heat fluxes, while at the same time keeping the temperatures relatively low and more uniform.

1.1. Flow boiling in mini and microchannels

Flow boiling is one of the widely investigated two-phase cooling schemes where an external source such as a pump, induces forced convection on the coolant undergoing phase change. One aspect of flow boiling is the use of minichannels and microchannels heat sinks. These heat sinks allow dissipation of high heat rates within a constrained space, providing high heat transfer coefficients and temperature uniformity during the phase change process of the coolant [4]. Extensive studies of mini and microchannels were conducted by many investigators in the recent years. Kandlikar [5] attempted to define the heat transfer mechanisms deterministically in microchannels. Lu et al. [6] investigated flow boiling in

conventional and mini channels in the vertical orientation. Wu and Sundén [7] developed correlations for stable flow boiling in minichannel and microchannels with flow patterns as the basis. These sustained efforts in the research on mini and microchannels could lead to the eventual implementation of these configurations for heat transfer applications.

1.2. *Issues faced in mini and microchannels and proposed solutions*

Despite the advantages mentioned in Section 1.1, there are certain issues faced in the two-phase flow of mini and microchannels that may hinder their feasibility. One of the major issues faced is flow reversal when a bubble, which is confined within a channel expands upstream and downstream, causing a premature dryout in the channel before it is expelled. Steinke and Kandlikar [8] investigated the flow boiling of water in six horizontal parallel microchannels with a hydraulic diameter of 207 μm and observed flow reversals. Mosyak et al. [9] observed flow reversals during low mass flux subcooled flow boiling of water. Qu and Mudawar [10] studied the critical heat flux (CHF) of a heat sink of 21 parallel $215 \times 821 \mu\text{m}$ channels. They observed flow instabilities when approaching CHF, which altered the inlet coolant temperature. Kuang et al. [11] observed periodic reversed flows even in mini-channels of 1.1 mm. Several researchers have developed solutions that mitigate flow reversals. Kuan and Kandlikar [12] implemented inlet flow restrictors in their microchannels which acted as a physical pressure drop element (PDE) to prevent the expanding bubble from expanding upstream. Lee et al. [13] studied expanding microchannels and inlet orifices and compared these to a regular parallel microchannel using an instability parameter. It was found that expanding microchannels and inlet orifices can reduce flow instability and minimize temperature oscillations compared with regular microchannels. Law et al. [14] and Law and Lee [15] explored oblique-finned microchannels and found that such a configuration suppressed flow reversal and consequently improved heat transfer performance. Deng et al. [16] investigated Ω -shaped reentrant configurations and showed a reduction in flow reversal through accommodation of vapor with the large circular cavity.

1.3. *Fractal-like and constructal flow channels*

Another promising solution that mitigates flow reversal, while also improving the thermal performance of the two-phase heat sinks, is the implementation of fractal-like and constructal tree-like flow channels. Zhang et al. [17] conducted a numerical simulation of a constructal tree-like flow channel and a serpentine channel under saturated flow boiling conditions. The constructal tree-like flow channel had a higher coefficient of performance (COP) compared to the serpentine channel. Cullion et al. [18] experimentally studied circular fractal flow channel networks using deionized and degassed water as the media. Flow reversal was found to be mitigated when the vapor slug was split at the bifurcations as long there was no obstruction downstream. There are, however, some studies that reveal the disadvantages of such implementations. Daguenet-Frick et al. [19] parametrically studied a disk-shaped constructal microchannel numerically with different combinations of flow channels and pairing levels. It was found that increasing the complexity, i.e., increasing the pairing levels, or bifurcations, did not necessarily lead to increased thermal performances. Revellin et al. [20] arrived at the same conclusion based on the critical heat flux (CHF). Nevertheless, most studies were performed numerically while experimental results for fractal and constructal flow channels are scarce, with one of the possible reasons being the complexity in manufacturing such channels conventionally.

1.4. *Selective Laser Melting*

Developments in additive manufacturing become more evident in the recent years. One of the additive manufacturing techniques is Selective Laser Melting (SLM) which uses metallic powders of aluminum, titanium and stainless steel to manufacture the desired parts. The main advantage of such a technology is its capability in fabricating complex three-dimensional (3D) geometries that are almost 100% dense [21] for applications in the aerospace, marine and electronics industries. Recent developments in SLM have inspired many researchers to develop unique heat sinks for various applications. Ho and Leong [22] studied cylindrical porous inserts in cold plates subjected to water flow. Neugebauer et al. [23] showcased the high accuracy of complex geometries when fabricating heat transfer devices. Wong et al. [24] fabricated various heat sinks using SLM, such as circular pin, elliptical, rectangular and rounded rectangular fins and lattices. The heat sinks were investigated based on their heat transfer coefficient and pressure drop. Wong and Leong [25] investigated octet-truss porous lattice structures fabricated by SLM under pool boiling conditions. The above brief literature review shows that additive manufacturing, and in particular, SLM, has the potential to develop unique heat transfer enhancement devices.

1.5. *Aims of study*

The aim of this paper is to study the flow boiling performance of fractal flow channels fabricated by SLM. The fractal flow channels were compared against a multi-channel parallel flow design fabricated by SLM as well. A dielectric fluid, FC-72, was used as the working fluid. The flow boiling performance, pressure drop, and the two-phase flow mechanics of fractal flow channels are studied in this paper.

2. **Design and characterization**

2.1. *Selective Laser Melting process*

To fabricate the designs in this present study, the SLM 250 HL (SLM Solutions GmbH) facility located in the Future of Manufacturing Laboratory 1 of the Singapore Centre of 3D Printing (SC3DP) at Nanyang Technological University, Singapore was employed. The machine uses a Gaussian distributed Yb:YAG laser of maximum power of 400 W, with a spot size of 80 μm . Inert argon gas is pumped into the build chamber to achieve 0.2% oxygen level to minimize possible oxidation and combustion of the powder.

During manufacturing, a thin layer of powder is deposited over the base plate. Based on the sliced CAD drawing of the part, the laser selectively melts the powder. The melted powder quickly solidifies, forming the first 2D geometry of the part. The base plate is then lowered, and another thin layer of powder is deposited over it. The laser then selectively melts the next 2D geometry layer of the part, bonding the first layer to the second. The process is then repeated until the desired 3D geometry is completed. AlSi10Mg powder is used for the present study as it can produce a solid part of thermal conductivity as high as 170 W/m²·K. The powder used has the same composition and distribution size as that used by Wong and Leong [24], where further details could be found.

2.2. Description of designs

In this study, the fractal flow channel designs consisting of a single large channel upstream, and multiple, smaller channels downstream were employed. The level of channels, or branch level, is defined by a parameter n , starting from 0 which represents a single large channel. The fractal flow channel designs are governed by the following equations:

$$\gamma = \frac{D_{n+1}}{D_n} \quad (1)$$

$$\beta = \frac{L_{n+1}}{L_n} \quad (2)$$

where γ and β are the diameter and length ratios, respectively.

In the present study, the designs are symmetrical and dichotomous, i.e., the channels are split into two equal parts and are symmetrical about the axis of the large channel ($n = 0$). A schematic of a fractal flow channel design is shown in Fig. 1.

Fig. 1. Schematic of a symmetrical, dichotomous fractal flow channel design, $n = 2$.

The length and diameter ratios were set to 0.5 and $2^{-\frac{1}{3}}$, respectively, and four designs were fabricated, ranging from $n = 1$ to $n = 4$. A multi-channel parallel flow channel design (depicted as *Parallel* for ease of presentation) was also fabricated as a benchmark to compare with the four fractal designs. All of the cross-sections of the channels in each design are semi-circular. The total mass of the designs are kept constant with a difference of $\pm 1.0\%$. The basic dimensions of the designs are listed in Table 1. Figs. 2 and 3 show the fabricated designs.

Table 1

Dimensions of all designs.

Fractal flow channel	Initial length [mm]	Initial diameter [mm]	Final length [mm]	Final diameter [mm]
$n = 1$	54.80	4.65	27.40	3.45
$n = 2$	47.40	4.60	11.85	2.54
$n = 3$	44.40	4.55	5.55	1.87
$n = 4$	43.00	4.50	2.69	1.37
<i>Parallel</i>	79.70	2.80	-	-

Fig. 2. Fabricated fractal flow channel designs. (a) $n = 1$, (b) $n = 2$, (c) $n = 3$, (d) $n = 4$.

Fig. 3. Parallel flow channel design (depicted as *Parallel*).

2.3. Light optical imaging of fabricated designs

An Olympus SZX7 light optical microscope coupled with an Olympus SC30 digital camera was employed to conduct optical imaging on the fabricated flow channel designs. An in-built measurement software was used to measure the diameter of each channel. It was found that the average error of the diameters is $\pm 0.41\%$, with the maximum error of $\pm 1.55\%$.

(a)

(b)

Fig. 4. Optical imaging of (a) $n = 1$ channel at branch level 1, (b) $n = 4$ channel at branch level 4.

The channels are also observed to be rough due to the SLM fabrication process. Since the flow channel designs were fabricated using the same powder and the same printing parameters, the surface roughness was not considered in the study, i.e., the flow channels are assumed to have a similar surface roughness.

3. Experimental setup

3.1. Flow boiling facility

The flow boiling facility used in the experiments is a closed-loop, pumped system that recirculates the fluid. The facility was used by Pranoto and Leong [26], with various modifications made. Figs. 5 and 6 show the photograph and schematic of the facility, respectively.

Fig. 5. Flow boiling experimental facility.

Fig. 6. Schematic of the flow boiling experimental facility.

The facility consists of a reservoir, a micro-pump, a pre-heater, a heater assembly and a heat exchanger that acts as a condenser. The test section is made of two main parts: 1) an aluminum base, and 2) a clear polycarbonate assembly to facilitate visualization of the two-phase flow. The two parts form a rectangular channel of 60 mm width, 160 mm length and 14 mm in height in the test section. The test section also has an expansion duct and a contraction duct with lengths of 150 mm and 110 mm, respectively which are implemented to allow a smooth transition for the coolant to the rectangular channel and away from the channel.

The heater assembly consists of seven cartridge heaters rated up to 4750 W of power in total through a variable transformer. The cartridge heaters were embedded into a copper block to generate uniformity of heating. To minimize heat losses to the surroundings, Teflon insulation was attached on all sides of the copper block except the exposed top surface in contact with the aluminum base mentioned previously. The heater assembly was clamped below the test section with OMEGATHERM 201 thermal grease applied in between to reduce the thermal contact resistance.

Two heaters rated at 400 W each were embedded in the reservoir to heat the coolant fluid to near-saturated temperature. A Cole-Parmer micro-pump with a high precision gear pump was installed to provide flow throughout the system. The pre-heater will heat the coolant fluid further to the 1 atm saturated temperature of 56°C just before entering the test section where the boiling phenomenon takes place. The vapor-liquid mixture then leaves the test section to the second heat exchanger, where it will be condensed to liquid and returned to the reservoir. Six K-type thermocouples were inserted into the aluminum base of the test section to estimate the average temperature of the surface of the test section. Fig. 7 shows the positions of the thermocouples as seen from the plan view. Two OMEGA PX409 absolute pressure transducers were installed before and after the test section to measure the absolute pressures. An additional OMEGA PX409 Wet/Wet Differential Pressure Transducer was attached

between the inlet and outlet of the test section to measure the pressure drop across the test section. Three Digi-Sense RTD thermocouples were inserted before the pre-heater, before the test section and after the test section to measure the temperature of the coolant. A McMillan Co. Model 112 flow sensor was installed before the first heat exchanger to measure the flow rate of the coolant. A YOKOGAWA MX100 Data Acquisition Unit collects all signals from the respective sensors and the data is collected and processed by a personal computer. A Photron FASTCAM 1024 PCI High-Speed Camera was used for visualization of the flow boiling conditions. The camera was set to 6000 frames per second with resolutions of 512×256 and 384×352 pixels.

Fig. 7. Positions of the thermocouples. All dimensions are in mm.

To ensure that the coolant flows within the channels only, a piece of translucent silicone was sandwiched between the design and a piece of acrylic, and the silicone was pressed down by the top cover of the test section. Since the silicone rubber is thin, at 1 mm, there is sufficient transmission of light for the high-speed camera to capture the two-phase flow phenomenon. Fig. 8 shows the arrangement of the flow channel design, silicone rubber sheet and acrylic.

Fig. 8. The arrangement of the flow channel, silicone rubber sheet and acrylic.

For each flow channel design installed, a commercial thermal grease, ARTIC MX-4 thermal compound with thermal conductivity, k_{TC} , of 8.5 W/m·K was used.

The thickness of the thermal compound is determined by calculating the difference in thickness of the base and the flow channel installed with the thermal compound, and the thickness of the same setup without the thermal compound. This measurement was done with a pair of Vernier callipers. It was found that the thickness of the thermal compound, t_{TC} , is 0.1 mm. The measured thicknesses are consistent for all experiments conducted in this study.

3.2. Dielectric fluid FC-72

Fluorinert Electronic Liquid FC-72 is a dielectric fluid developed by 3M Company. The fluid was selected due to its high dielectric strength of 38 kV and low boiling point of 56°C at 1 atm. FC-72 is also thermally and chemically inert, non-flammable and non-toxic, thus allowing its safe use for the study. The thermophysical properties of FC-72 at 25°C and of atmospheric pressure obtained from 3M [27] are listed in Table 2. Thermodynamic properties of FC-72 such as enthalpy at various temperatures were obtained directly from 3M.

Table 2

Properties of FC-72 at 25°C and 1 atm.

Properties (1 atm)	Value
Boiling point [°C]	56
Liquid density [kg/m ³]	1680
Liquid dynamic viscosity [kg/m·s]	6.4×10^{-4}
Liquid specific heat [J/kg·K]	1100
Liquid thermal conductivity [W/m·K]	0.057
Latent heat of vaporization [kJ/kg]	88
Liquid surface tension [N/m]	0.01
Dielectric strength at 0.1 inch (2.54 mm) gap (kV)	38

4. Experiment and data reduction

The mass flux, G , was set for each experiment by adjusting the flow rate of the coolant. The mass flux is also dependent on the inlet cross-sectional area of each flow channel, A_{in} . Therefore, the mass flux is determined as

$$G = \frac{\dot{m}}{A_{in}} \quad (3)$$

Three mass fluxes were selected for each flow channel design: 600, 900 and 1200 kg/m²·s. Each experiment was conducted until the steady-state condition was achieved when the temperature fluctuated less than 0.2°C for a period of 5 minutes. Time-averaged methods were employed to acquire each experimental data. This was done by recording the data for one minute at 500 ms intervals and averaging the readings. Heat was supplied to the flow channel through the heater assembly from 30 W with increments of 60 W, and the experiment

was ceased when either dryout was observed, or when the temperature of the flow channel increased continuously in an unsteady fashion. The latter action was taken to prevent possible equipment damage and injury to the investigator.

The heat loss, $\dot{Q}_L$, from the test section was determined by conducting an experiment with an empty channel, and taking the difference between the heat input from the heater assembly and the heat transferred to the coolant across the channel. The heat losses for various mass flow rates for the study were determined and the total heat transfer rate was compensated based on the heat loss for the particular mass flow rate. The total heat transfer rate is therefore expressed as

$$\dot{Q} = \dot{Q}_{supp} - \dot{Q}_L \quad (4)$$

where $\dot{Q}_{supp}$ is the heat transfer rate supplied from the variable transformer and is calculated by the product of the voltage and current displayed on the transformer. The total rate of heat loss was found to be between 5% and 32% of the heat supplied throughout the study depending on the mass flow rate used.

The average temperature of the flow channel design, T_{FC} , was calculated by taking the average of six temperatures measured at the base of the test section. One-dimensional heat conduction modeling was employed to determine the temperatures of the flow channel. The thickness between the thermocouples and the surface of the channel, t , is 3 mm. Accounting for the thickness of the thermal compound as mentioned in Section 3.1, the temperatures are determined from

$$T_{FC,m} = -\frac{tk_{TC} + t_{TC}k_{Al}}{k_{Al}k_{TC}A} \dot{Q} + T_m \quad (5)$$

where T_m is the temperature measured within the base of the test section, and $T_{FC,m}$ is the temperature at the flow channel design for the particular thermocouple measured. k_{Al} is the thermal conductivity of the aluminum base of the test section which is taken as 203 W/m·K. The average temperature of the flow channel design is therefore calculated as

$$T_{FC} = \sum_{m=1}^6 T_{FC,m} \quad (6)$$

The heat transfer performance of each flow channel design is defined based on the heat transfer coefficient. The experiments were conducted at the saturated temperature of the coolant, T_{sat} at atmospheric pressure. The flow boiling heat transfer coefficient, h_{FB} , is expressed as

$$h_{FB} = \frac{\dot{Q}}{A_w(T_{FC} - T_{sat})} \quad (7)$$

where A_w is the total wall surface area of the channels.

The measurement uncertainties of the K-type thermocouples and RTD sensors are $\pm 0.5^\circ\text{C}$ and $\pm 0.1^\circ\text{C}$, respectively. The uncertainty of the PX409 pressure transducers is $\pm 0.02\%$ and that of the differential pressure transducer is $\pm 0.01\%$ of their full scales. The uncertainty of the flow meter is $\pm 0.5\%$ of its full scale. The variable transformer powering the heater

assembly has an uncertainty of $\pm 0.38\%$ for the voltage and $\pm 0.07\%$ for the current in their full scales. The uncertainty of h_{FB} is determined by a method as described by Moffat [28]. It is found that the average uncertainty of h_{FB} is $\pm 3.0\%$, with a maximum uncertainty of $\pm 13.8\%$. The higher uncertainty generally occurs at lower heat supplied.

5. Results and discussion

5.1. Heat transfer coefficient

In this study, the heat transfer coefficient is used to define the heat transfer performance of the flow channel designs. Both the effects of mass flux and the branch levels are investigated.

5.1.1. Effect of mass flux

Fig. 9 shows the heat transfer performance for the fractal flow channel designs $n = 1$ and $n = 2$, and *Parallel*. The *Parallel* design does not show a significant increase in the heat transfer coefficient for all mass fluxes, with only a modest 5.7% increase when the mass flux is increased from 600 kg/m²·s to 900 kg/m²·s and only 1.0% from 900 kg/m²·s to 1200 kg/m²·s. The mass flux plays a significant role in the performance for all fractal flow channel designs as the peak heat transfer coefficient increases as the mass flux is increased. The largest increase is from 600 kg/m²·s to 900 kg/m²·s, where it is observed that the improvement of the peak heat transfer coefficient is approximately 22.7% on average. Modest increase is observed from 900 kg/m²·s to 1200 kg/m²·s, with an average increase of 6.6%. This increase is therefore not linear. This lack of increase of heat transfer coefficient at high mass flux also suggests that forced convective boiling is dominant in all fractal flow channel designs [6]. It is also possible that at lower mass fluxes, the nucleate boiling mechanism is dominant, and thus, the peak heat transfer coefficient increases significantly due to the transition from nucleate dominant to convective dominant heat transfer. Contrary to the fractal flow channel designs, *Parallel* has minimal improvement in the peak heat transfer coefficient as the mass flux is increased. This design is likely to exhibit the same phenomenon as the fractal flow channel designs at higher mass fluxes, which is convective dominant.

(a)

(b)

Fig. 9. Heat transfer coefficient versus wall superheat ($T_{FC}-T_{sat}$) of (a) *Parallel*, (b) $n = 1$, (c) $n = 2$.

5.1.2. Effects of branch level

The fractal flow channels designs and *Parallel* are compared with the respective mass flux, expressed in Fig. 10.

(b)

(c)

Fig. 10. Heat transfer coefficient of all flow channel designs for (a) 600 kg/m²·s, (b) 900 kg/m²·s, (c) 1200 kg/m²·s.

At low mass flux, the fractal flow channel design does not show significant enhancement in the heat transfer coefficient when compared with *Parallel* which has a peak heat transfer coefficient of up to 7.0% higher. By increasing the mass flux, the peak heat transfer coefficient of the fractal flow channel surpasses *Parallel*, with $n = 2$ achieving the highest value compared to all flow channel designs. Table 3 shows the enhancements for various mass fluxes.

Table 3Enhancements in peak heat transfer coefficient over *Parallel* for various mass fluxes.

Mass flux [kg/m ² ·s]	Enhancement over <i>Parallel</i> [%]			
	$n = 1$	$n = 2$	$n = 3$	$n = 4$
600	-3.9	-2.1	-0.1	-7.0
900	11.2	15.1	14.1	8.8
1200	14.1	28.5	13.5	22.4

For all mass fluxes, it is also observed that $n = 4$ does not achieve similar peak heat transfer coefficient compared to $n = 2$. The results lead to a possible conclusion that increasing the complexity (i.e., increasing branch level) may not necessarily be beneficial in two-phase flow. Such a conclusion was also reached by various researchers [19, 20].

5.2. Pressure drop

The pressure drop generally increases linearly for all flow channel designs for all mass fluxes as shown in Fig. 11.

(b)

(c)

Fig. 11. Pressure drop of flow channel designs for (a) 600 kg/m²·s, (b) 900 kg/m²·s, (c) 1200 kg/m²·s.

Pressure drop in the two-phase flow is vastly different compared to single-phase flow, where research showed a reduction in pressure drop for the fractal flow channel designs compared to their parallel counterparts [29, 30]. In this study, it is observed that the pressure drop of *Parallel* is much lower than the fractal flow channel designs. The rate of increase in the pressure drop with the increase of heat supplied in *parallel* is also lower than all fractal flow channel designs for all mass fluxes. The average rate of increase for all designs at all mass fluxes are shown in Table 4.

Table 4

The rate of increase of pressure drop.

Mass flux [kg/m ² ·s]	Rate of increase of pressure drop [kPa/W]				
	$n = 1$	$n = 2$	$n = 3$	$n = 4$	<i>Parallel</i>
600	132.9	186.9	173.8	213.8	87.2
900	122.0	166.7	160.9	198.9	81.6
1200	109.3	153.2	140.5	186.5	76.5

The higher rate of pressure drop is likely due to the number of bifurcations that the flow experienced in the fractal flow channel designs. The higher the number of bifurcations, the higher the pressure drop, as seen in $n = 4$, which has a total of 15 bifurcations, compared to $n = 1$, where there is only one bifurcation. In addition, the flow impacts the inlet of the next set of channels at an angle, further increasing the pressure drop. Further studies would be necessary to investigate the effects of the bifurcation angle on the pressure drop. It is noteworthy that the rate of pressure drop decreases with increasing mass flux, with a larger change in the fractal flow channel designs compared to *Parallel*. The significant change suggests the possibility of employing fractal flow channel designs at high flow rate applications. However, as far as this study is concerned, the *Parallel* configuration still possesses a lower pressure drop compared to any fractal flow channel design.

Since increasing the branch level causes higher pressure drop, the increase in complexity of design configuration may not be beneficial as mentioned in Section 5.1.2.

5.3. High-speed video images of the flow channels and mechanisms

High-speed video images of the two-phase flow in the designs were recorded for various mass fluxes. During boiling, flow reversals were observed in *Parallel* at certain heating powers. Fig. 12 shows the schematic of the flow reversal observed in this study, while Fig. 13 and 14 show high speed images of the expulsion of vapor on one of the channels depicted in the regions circled in red, and vapor flowing downstream respectively. During heating, vapor begins to build up as the coolant is boiled. Eventually the vapor confines the channel shown in Fig. 12(a). The process continues as the vapor expands upstream and downstream as shown in Fig. 12(b). The vapor continues to expand until it reaches the inlet of the channel, where it is expelled from the channel. Eventually, the inertial force from the coolant flow pushes the expelled vapor back into the channel as depicted in Fig. 12(d) (Refer to Appendix A - supplementary video 1). At the mass flux of 600 kg/m²·s, bubbles can be seen to be expelled upstream and downstream before accelerating downstream for as low as 150 W of heat supplied in two out of the four channels. Increasing the heat supplied, however, reduces the effects of the reversal. At 900 kg/m²·s, flow reversal was observed at 210 W heating power, even though it was not as pronounced as that at 600 kg/m²·s. When increasing the heat supplied, unlike the phenomenon at the mass flux of 600 kg/m²·s, the effects of the flow reversal remain unchanged. Increasing the mass flux to 1200 kg/m²·s seems to prevent the flow reversal up to a higher heating power of 330 W instead of the lower heat supplied for the other two lower mass fluxes. Flow reversal occurs at higher heat supplied at increasing mass fluxes, likely due to the higher inertial force that prevents vapor from being expelled upstream. The vapor is expelled upstream when the force due to its expansion overcomes the inertial force. Referencing from Kandlikar [5], the force per unit length caused by the evaporation of vapor and the inertial force are shown in Eqs. (8) and (9) respectively

$$F'_M = \left(\frac{\dot{q}}{h_{fg}} \right)^2 \frac{D'}{\rho_v} \quad (8)$$

$$F_I' = \frac{G^2 D'}{\rho_l} \quad (9)$$

where D' is the characteristic dimension suggested by Kandlikar [5].

Vapor expelling upstream overcoming the inertial force can be seen as $F_I' < F_M'$. Forming the inequality and rearranging gives

$$\frac{G}{\dot{q}} < \frac{1}{h_{fg}} \sqrt{\frac{\rho_l}{\rho_v}}. \quad (10)$$

Eq. (10) shows the increase in heat flux will lead to the possibility of flow reversal, while the increase in mass flux can suppress flow reversal, which is aligned with the observations in the study.

It is noteworthy that the flow reversal was observed to be not as prominent at low mass fluxes compared to higher mass fluxes. This might have caused the insignificant enhancement in the heat transfer coefficient as the mass flux increases. That is due to the flow reversal disrupting the proper replenishment of the liquid coolant to the channels for the latent heat transfer process to occur.

Fig. 12. Schematic of flow reversal observed in *Parallel*. (a) Vapor is confined in the channel. (b) Vapor expands upstream and downstream. (c) Vapor is expelled out at the inlet of channel, while part of it continues downstream. (d) Vapor that is initially expelled flows back into channel and downstream.

Fig. 13. Flow on a channel in *Parallel*, $600 \text{ kg/m}^2\text{s}$, 210 W heat supplied. Frames are shown at $1.67 \times 10^{-4} \text{ s}$ intervals. Bubbles were expelled upstream, depicted by the regions marked in red.

Fig. 14. Flow on a channel in *Parallel*, $600 \text{ kg/m}^2\text{s}$, 210 W heat supplied. Confined bubbles shown moving downstream depicted by the red circles. Frames are shown at $1.67 \times 10^{-3} \text{ s}$ intervals.

It has been observed, however, that the fractal flow channels do not result in any flow reversal. Fig. 15 shows a snapshot of $n = 1$ at three different mass fluxes. The flow is consistent for all mass fluxes and heat supplied. As the two phase flow approaches the bifurcation, it splits relatively equally, while also increasing in velocity. This phenomenon is consistent for all mass fluxes and at any heat supplied as well (Refer to supplementary video 2). The lack of flow reversal is likely due to the large dimensions of the channel at the inlet, preventing the flow to be confined due to bubble expansion. Downstream, at the bifurcation point (i.e., the point where the channel is split into two smaller channels), the flow accelerates, possibly increasing the inertial forces and therefore preventing the force due to vapor expansion from causing flow reversal. In addition, upon the flow splitting at the bifurcation, the two-phase flow separated from the wall, indicated by the maroon boundaries in Fig. 15. This contributed to the lack of flow reversal since there is ample space for the vapor to expand. However, this also leads to empty spaces where no boiling occurs, and therefore underutilization of the whole channel surface for heat transfer. This may explain why the fractal flow channels with higher bifurcations or branch levels (i.e., $n = 4$) could not perform as well as those with lower branch levels, as $n = 4$ will experience more regions where the flow separates from the wall. Generally, it is observed that the increase in peak heat transfer coefficient in fractal flow channel designs is mainly due to the increase in mass flux without flow reversal offsetting its performance.

(a)

(b)

(c)

Fig. 15. Two-phase flow of $n = 1$, 330 W heat supplied, and (a) 600 kg/m²·s, (b) 900 kg/m²·s and (c) 1200 kg/m²·s. Maroon boundaries with dark blue shading represent regions where two-phase flow was not observed.

6. Conclusions

The conclusions from this study are summarized as follows:

1. The peak heat transfer coefficients for all flow channel designs are comparable at 600 kg/m²·s. At higher mass fluxes, the *Parallel* channel does not show an increase in the peak heat transfer coefficient whereas the fractal flow channel designs have heat transfer coefficients which are between 8.8% to 15.1% higher than *Parallel*. The fractal design with $n = 2$ has the highest peak transfer coefficient among all flow channels.
2. The pressure drop in *Parallel* is lower than all of the fractal flow channel designs for all mass fluxes. The design for $n = 4$ has the highest pressure drop among all the fractal flow channel designs, with a lower pressure drop with lower branch level. The higher pressure drop in the higher branch level is likely due to the multiple bifurcations that the coolant has to flow through.
3. Increasing complexity, i.e., increasing branch level, does not necessarily improve heat transfer performance and pressure drop in flow boiling applications. It is shown that $n = 2$ possesses the highest peak heat transfer coefficient, whereas $n = 4$ shows a lower peak heat transfer coefficient.
4. Visualization studies suggest that *Parallel* does not show any increase in its heat transfer coefficient with increasing mass flux due to the presence of flow reversals at higher mass fluxes.
5. Visualization studies of the fractal flow channels show no flow reversal due to the large dimensions of the first channel, and the high inertial force downstream upon reaching the bifurcation. The flow separation, however, may have hindered higher branch level fractal flow channels from performing better than the lower branch level fractal flow channels.

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Appendix A

Supplementary videos 1 and 2 associated with this article can be found in the online version.

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