Experimental Study of Flow Boiling Heat Transfer in Spider Netted Microchannel for Chip Cooling

Hui Tan, Jiajing Chen, Mingyang Wang, Pingan Du*

Abstract—**This study investigates the flow boiling heat transfer performance of the spider netted microchannel which is applied in the chip cooling at high heat flux. The boiling curves, heat transfer coefficients are obtained in the heat flux range of 10 to 100 W/cm², volumetric flow rate of 0.2 and 0.3L/min. Compared to the straight microchannel, spider netted microchannel can achieve significant augmentation in heat transfer coefficients and lower superheat temperatures in the boiling process under the same condition. Moreover, the wall superheat of the spider netted microchannel at a volumetric flow rate of 0.2L/min is lower than that of the straight microchannels at 0.3L/min. The results indicate that spider netted microchannel presents better heat transfer performance than straight microchannel.**

Keywords: **Spider netted microchannel, Straight microchannel, Flow boiling heat transfer, Boiling curve**

I. Introduction

Microchannel cooling has attracted great attentions in high flux devices for its highly efficient heat transfer performance since it was proposed firstly by Tuckerman and Pease [1]. Numerous studies have been conducted to improve the single-phase heat transfer performance of the straight microchannels by optimizing geometry parameters [2], or shapes [3-11], including fractal-like branching channels [3], tree-like channels [4-6], and wavy channel [7-11].

Two-phase flow boiling heat transfer in microchannel is more efficient than the single-phase, as the flow boiling offers a high heat transfer rate via the latent heat of coolant with small rate of coolant flow. Several experimental studies have been conducted to explore the flow boiling performance of microchannels with various coolants as water, FC-72, R-134a, multi-component mixture, nanofluids [12-19] and different shapes as straight, diverging cross section, pin fins [20-27].

G. Hetsroni [12] investigated the instability and heat transfer phenomenon of the flow boiling in parallel micro channels with deionized water. J.B. Copetti [13] gave the conclusion that the effect of heat flux on the heat transfer coefficient is much more obvious in the low vapor quality region than that of in the high quality region with R-134a. B.H. Lin and B.R.Fua [14-15] concluded that the small additions of ethanol into water could increase the CHF. Krishnamurthy and Peles [16] thought that there were obvious heat transfer enhancements in micro circular silicon pin fins with coolant HFE7000 compared to the straight micro channels. Saeid Vafaei and D. Wen[17] investigated the CHF of subcooled flow boiling of aqueous based alumina nano-fluids in a 510 μm single microchannel under low mass flow rate conditions. It was concluded that nanoparticle deposition and a subsequent modification of the boiling surface were common features associated with nanofluids L. Xu [18] concluded that nanofluids significantly mitigated the flow instability and enhanced heat transfer. Zhou [19] proposed a theoretical saturated flow boiling heat transfer coefficient correlation for nanofluid in minichannel and compared the experimental and theoretical data.

Kandlikar [20] improved the stabilization of flow boiling in rectangular microchannels with inlet pressure restrictors and artificial nucleation sites. Lu and Pan [21] explored flow boiling in a single microchannel with a converging/diverging cross section. The angle of converging/diverging was 0.183°. Chun [22] investigated the diverging microchannels with different distributions of artificial nucleation sites to reduce the wall superheat and enhance flow boiling heat transfer performance. K. Balasubramanian [23] found that the expanding microchannel had a better heat transfer performance than the straight microchannel with lower pressure drop and wall temperature fluctuations. D. Deng [24] developed a copper microchannel with unique Omega-shaped reentrant configurations, which present significant augmentation in two-phase heat transfer and a reduction of two-phase pressure drop and mitigation of two-phase flow instabilities. W. Wan [25] compared flow boiling performance of four types of micro pin fin heat sinks, i.e., square, circular, diamond and streamline, and found that the square micro pin fin presented the best boiling heat transfer, followed by circular, streamline and diamond ones. Zhang [26] introduced the interconnecting channel with higher heat transfer and suppressed instability at small to medium mass flux but not satisfactory at higher mass flux. Hong [27] investigated the two configurations of ultra-shallow microchannels with rectangular and parallelogram con-figurations respectively.

The purpose of the study is to explore the flow boiling heat transfer performance with various heat fluxes and inlet flow rates in the spider netted microchannel (SNMC). For

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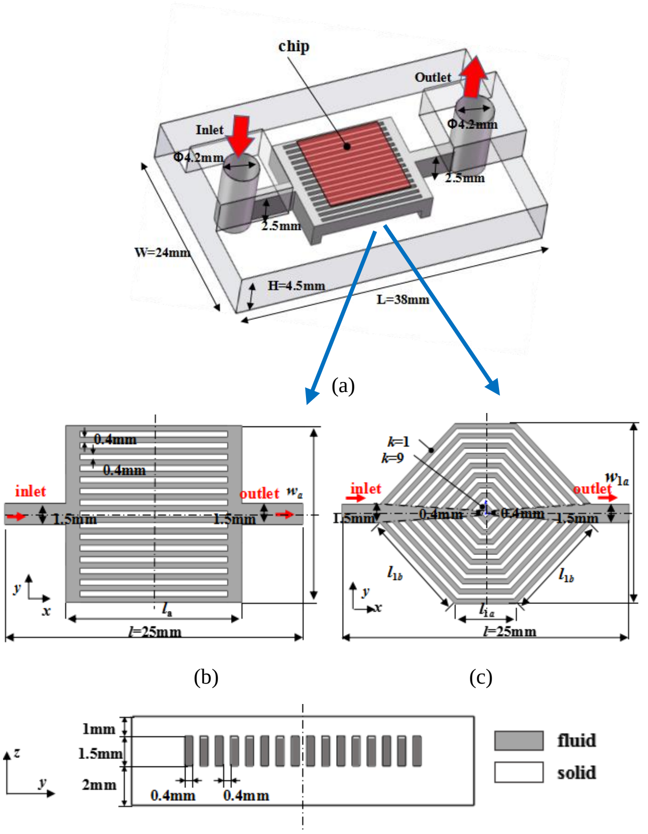
comparison, experiments are performed in two test pieces, i.e., spider netted microchannel proposed in our previous work and straight microchannel (SMC) respectively under the condition of volumetric flow rate of 0.2 to 0.3L/min and heat flux 10 to 100 W/cm^2.

II. EXPERIMENT DESCRIPTION

A. Fabrication of microchannels

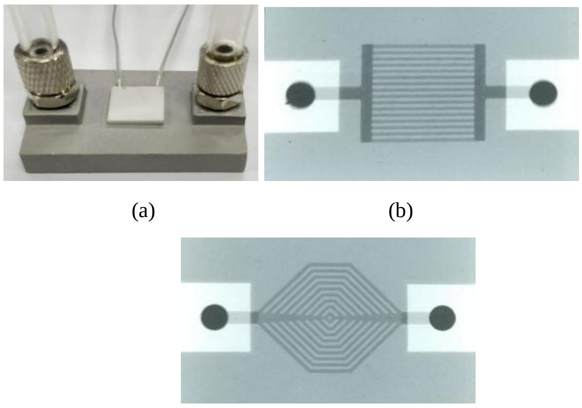
The traditional fabrication method of the heat sink is to make the cover plate and the microchannel structure respectively, which are installed together by diffusion welding or bolts. However, the way of the bolted connection may cause the coolant leakage, while the interface thermal resistance is generated during the welding process, which may lead to the decrease of the heat dissipation efficiency. In this paper, the metal additive manufacturing technique is adopted to make the test pieces on 6063 aluminum alloy base.

Two types of microchannels are designed in Fig.1. For comparison, the coverage area of microchannels in SMC (183.5mm^2) is nearly equal to SNMC(183.4mm^2). The dimensions are as follows: wall thickness of each microchannel 0.4mm, channel width 0.4mm, channel height 1.5mm. The cross section is rectangular for inlet and outlet with 1.5mm width and 2.5mm height. Subscript k represents the branching level at a bifurcation. Seen from Fig. 1, the first branch emanating from the inlet flow and the last branch in the center region are respectively the first-order branch and ninth-order branch, i.e., k=1 and k=9. Fig. 2 shows the X optical scanning images of two fabricated microchannels.



(d)

Fig. 1. Geometry dimensions of microchannels (a) 3D view (b)front view of SMC (c)front view of SNMC (d) side view of the microchannel



(a) (b)

(c)

Fig. 2. Test pieces. (a)outside view (b)X optical scanning images of SMC (c)X optical scanning images of SNMC by industrial CT

B. Flow boiling experiment procedure

Fig.3 shows the experiment facility in a closed loop consisted of a heating system, coolant driving system, and test section. The coolant driving system includes liquid storage tank, constant temperature bath, pump, filers, condenser, and flowmeter. The working liquid of ethanol, adjusted to the certain temperature and volumetric flow rate, enter the test section and heat exchange occurs inside the microchannels. The inlet subcooling is 2℃. The heat source is a square High-temperature co-fired ceramics (HTCC) bonded to the upper surface of the heat sink by thermal conductive adhesive. One type-K shielded thermocouple with a diameter of 1 mm is bounded at the center of the top surface of HTCC. Once reaching steady state, all temperatures are collected by an Agilent 34972A data acquisition system. The range of the flow meter is from 0.2L/min to 0.3L/min, and the heat power is increased in an increment of 5W with the maximum total power up to 100W.

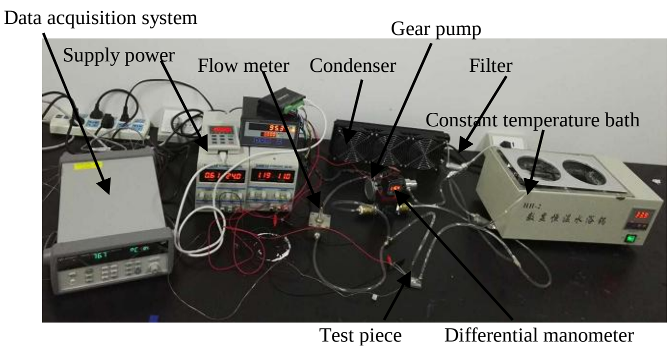


Fig. 3. Experimental platform

C. Data reduction

The effective heat flux, q, is computed from

$$q = \varphi VI / A_w \tag{1}$$

where φ is the heat transfer ratio denoting the absorbed heat by the coolant against the total power, V and I are the input voltage and current, and A_w is the heat transfer area of HTCC. Uniform heat flux is expected to be supplied for the heating surfaces in this paper. The range of φ was from 0.85 to 0.9 depending on the inlet temperature, flow rate and heat flux and the mean heat transfer ratio in the flow boiling experiment.

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This method has been used in many works [12,25].

The local heat transfer coefficient, h, is defined as

$$h = q / T_w - T_{sat} \quad (2)$$

where T_{sat} is saturation temperature, T_w is the wall temperature computed from

$$T_w = T_c - q\left(l_{HTCC} / k_{HTCC} + l_{glue} / k_{glue} + l_c / k_c\right) \quad (3)$$

where T_c is the local thermocouple reading, l_{HTCC}, l_{glue}, l_{c} are the thickness of HTCC and glue, distance of from heat sink base to the top of the microchannel surface, respectively. k_{HTCC}, k_{glue}, k_{c} are the thermal conductivities of HTCC, glue and Aluminum base, respectively.

Uncertainties in individual temperature measurements are ±0.3℃ for thermocouple. The wall temperature uncertainty comes from the thermocouple errors and the correction from the temperature drop of the glue and aluminum base. Using the standard error analysis method, the maximum experimental uncertainties in the two-phase heat transfer coefficient can be estimated to be within 10%.

III. RESULTS AND DISCUSSION

A. Boiling curve

Fig. 4 shows the boiling curves for both the SNMC and SMC with effective heat flux versus wall superheat temperature at the inlet subcooling of 2℃. It can be noted that the slope of the boiling curve in the SNMC is larger, demonstrating that the wall superheat is relatively small for a given heat flux and volumetric flow rate in SNMC, and conversely, the wall superheat of SMC is higher. This is partially due to the more stable two-phase flow through the SNMC and the vapor generated can pass through the channel smoothly. Otherwise, the heat transfer area is larger in SNMC and more alternative pathways for bubbles to exit the microchannels. Therefore, the flow boiling heat transfer performance in SNMC is better than the straight one under the same volumetric flow rates and heat fluxes. Moreover, the wall superheat of the SNMC at a volumetric flow rate of 0.2L/min is lower than that of SMC at 0.3L/min. It may be expected that the SNMC will have wider application range.

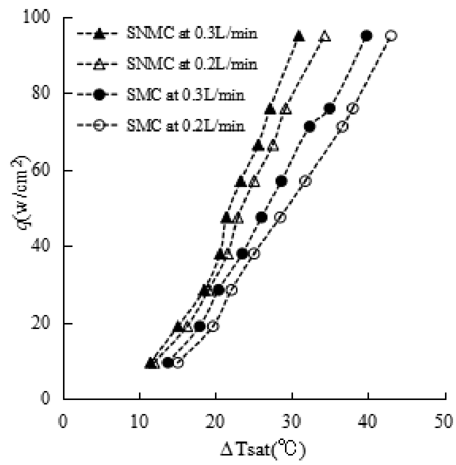

Fig. 4. Flow boiling curve for SNMC and SMC

B. Heat transfer performance

Fig. 5 illustrates that the heat transfer coefficient versus effective heat flux in two test pieces with the volumetric flow rate of 0.2L/min. It is clear that trend of the two boiling curves is similar, while the SNMC has higher values of the heat transfer coefficient under the same condition, especially at higher heat fluxes, an enhancement of 33% is achieved compared to the straight channel at the heat flux of 75W/cm^2. Moreover, the heat transfer coefficient increases sharply with increasing heat flux at lower heat flux ranges. The heat transfer coefficients in this region are sensitive to hear flux. This may suggest that the heat transfer mechanism is dominated by nucleate boiling in the range of low heat fluxes, where the increase of heat flux could generate more bubbles. It is publicly recognized that the bubbles motion plays a positive role on the heat transfer enhancement. In the nucleate boiling region, the larger heat transfer area of SNMC increases the nucleation site density, thus enhancing the density of the bubbles. The higher density of nucleating bubbles increases the rate of heat transfer which results in lower wall temperature.

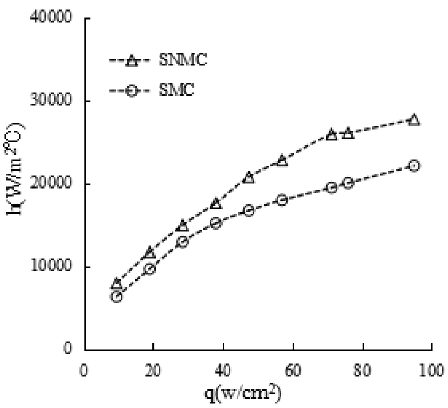

Fig. 5. Variation of flow boiling heat transfer coefficient with heat flux for SNMC and SMC at V=0.2L/min.

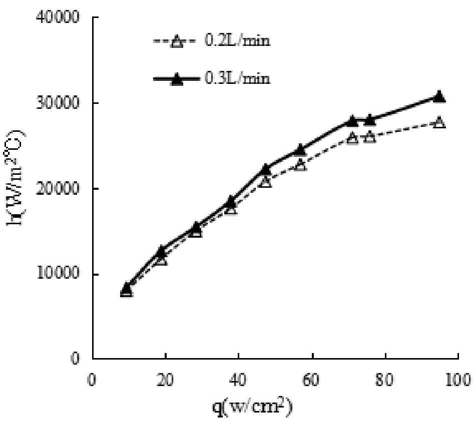

Fig. 6. Variation of flow boiling heat transfer coefficient with heat flux for SNMC as a function of flow rate.

Fig. 6 shows the flow boiling heat transfer coefficient in SNMC at a volumetric flow rate of 0.2L/min and 0.3L/min respectively. We can see that the heat transfer coefficient increases with increasing of volumetric flow rates, yet this is not very significant, thus in the flow boiling process, the nucleate-boiling is still active. The reason for the enhanced flow boiling heat transfer is the improving fluid mixing and improving bubble movement in microchannels. The fluid

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mixing plays positive role in the continuous interruption of the boundary layer, which is recognized to be beneficial for the heat transfer performance. Besides, the geometric characteristic of bifurcation in spider netted channel provides more alternative pathways for bubbles to exit the microchannels.

From the discussion above, the advantages of the spider netted microchannel for flow boiling in parallel diverging microchannels are a reduction in the wall superheat for boiling and an enhancement in the boiling heat transfer performance.

IV. Conclusion

The flow boiling heat transfer performance of SNMC is studied experimentally under various volumetric flow rates and heat fluxes. The results reveal that：

a) The flow boiling heat transfer performance in SNMC is better than that in the straight one under the same conditions of volumetric flow rates and heat fluxes.

b) The wall superheat of SNMC at a volumetric flow rate of 0.2L/min is lower than SMC at 0.3L/min.

c) The SNMC has higher values of the heat transfer coefficient under the same condition, especially at higher heat fluxes, an enhancement of 33% of the heat transfer coefficient is achieved at the heat flux of 75W/cm^2. Besides, the heat transfer coefficient increases with increasing of volumetric flow rates.

d) The main reasons for the better heat transfer performance of SNMC are the following: firstly, the larger heat transfer area of the SNMC which increases the amount of nucleation sites, thus trigger more bubble generation, secondly, geometric characteristics of bifurcation provides more alternative pathways for bubbles to flow and exit the microchannels, and lastly, the improving fluid mixing plays positive role in the enhancement of the flow boiling heat transfer.

Therefore, the above research will be probably attributed to the engineering application of the new microchannel at high heat flux.

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