Single- and Two-Phase Thermal Transport in Microchannels With Embedded Staggered Herringbone Mixers

Fanghao Yang, Mohammad Alwazzan, Wenming Li, and Chen Li
Single- and Two-Phase Thermal Transport in Microchannels With Embedded Staggered Herringbone Mixers

Fanghao Yang, Mohammad Alwazzan, Wenming Li, and Chen Li

Abstract—Improving mixing is an effective means to enhance single- and two-phase heat transfer in microchannels. However, it is challenging to induce since the flow in microchannels is laminar in the most working conditions. We report that heat transfer rate and critical heat flux (CHF) on 1-methoxyheptafluoropropane (HFE-7000) can be significantly enhanced by patterning embedded micromixers on the bottom walls in a parallel silicon microchannel array, which consists of five parallel channels (height, width, length: 250 μm × 220 μm × 10 mm). Compared with a plain-wall microchannel array at a mass flux range of 1018 to 2206 kg/m²·s and a heat flux range of 10 to 198 W/cm², single-phase heat transfer rate, two-phase heat transfer rate, and CHF are enhanced up to 221%, 160%, and 61% using microscale staggered herringbone mixers in microchannels, respectively. These mixers consist of 7 or 3.5 Hz with 12 staggered herringbone grooves (50 μm in depth and width) with 90° between two asymmetric arms in each cycle. Its asymmetry is defined in accordance with the off center position of the apex of the herringbone groove. Finally, experimental results suggest that the locations and coverage of the micromixers have significant impacts on both single and two-phase heat transfer in microchannels.[2013-0372]

Index Terms—Electronics cooling, dielectric fluid, micromixer, microchannel, flow boiling.

I. INTRODUCTION

TWO-PHASE transport in microchannels [1], [2] has drawn extensive attentions in the last two decades because of its high performance as monolithic integratable technique in high-power microelectronics and Microelectromechanical systems (MEMS) compared to the single-phase transport [3], [4]. Applications include electronics cooling, [5], compact heat exchangers [3], [6], [7], multi-phase mixing and reactions [8], etc.

Promoting mixing has been found to be an effective method to enhance both single and two-phase transport since the laminar flow, due to the viscous and capillary flows, governs the fluid flow and hinders the mass and heat transfer in microchannels. Recent studies have been carried out to enhance flow boiling by promoting mixing via either active or passive methods. Active methods such as impingement jets [5], and synthetic jets [9], [10] can suppress flow instabilities, enhance heat transfer, and CHF. However, these methods usually require additional actuators or pumping power, which results in more complicated systems and makes them challenging to be integrated.

Numerous types of passive mixing and yet cost-effective methods at microscale have been developed to enhance fluid mixing during single and two-phase flow in microchannels. For example, passive mixing was achieved by various stretching and folding techniques such as creating helical flow using serpentine [11] or curved microchannels [12], using pump to stir fluid through multiple sides’ channel [13], and embedding slanted ribs or grooves in the bottom walls of microchannels [14], [15]. However, most of existing passive mixing methods in microscale laminar flows focus on promoting mass transfer. Micro-mixers that aim to enhance heat transfer are as yet lacking.

Several recent studies suggested that convective heat transfer and flow boiling inside microchannels can be also significantly enhanced by generating strong fluid mixing using passive methods with less complicated designs, such as using wavy microchannels to generate dean vortices [16], using the self-sustained high frequency two-phase oscillations that are induced by rapid thermal bubble growth/collapse [17], and flow separation technique by using a passive microjet [18]. Beside the existing mixing techniques to enhance heat transfer, mixing in microchannels could also be generated by using passive micro-mixers to manipulate the dominant laminar flow based on either enhanced advection or diffusion transport mechanisms. As shown in Fig. 1a, the mixing design firstly introduced by A. D. Stroock, et al [15], also known as staggered herringbone mixer (SHM), produces two helical flows with two vertices that induce effective mixing in microchannels and has not yet been studied for enhancing single and two-phase transport in silicon or metallic microchannels. It is mostly because of the difficulty in microfabricating SHMs in silicon or metallic microchannels. Recent advances in silicon etching technique promote the opportunity to study single and two-phase heat transfer in microchannels by implementing SHMs on bottom walls.

In this experimental study, the SHMs are adapted as micro-mixers (Fig. 1) to improve both single and two-phase transport...
II. DESIGN OF MICROCHANNEL ARCHITECTURE

The micro-device has five microchannels (H = 250 µm; W = 220 µm; L = 10 mm). The SHM design, which was used in previous study on single-phase mixing [15], is employed to investigate its effects on transport in microchannels. The embedded SHMs Fig. 1a are comprised of 3.5 or 7 cycles with 12 staggered herringbone grooves (50 µm in depth and width) with 90° between two asymmetric arms in each cycle. Given a constant heat flux input, the amount of vapor generated owing to evaporation and boiling will increase linearly along the flow direction and result in an increasing local vapor quality and Reynolds number. Thus, the location of SHMs can be a major factor in manipulating two-phase transport by inducing mixing at different locations. The SHMs are expected to generate mixing and modify local flow patterns, thus, alter thermo-hydraulic characteristics. In this study, four different configurations with various coverages and locations of SHMs as illustrated in Fig. 1b were tested to determine their impacts on transport in microchannels. On the 1st configuration, SHMs (7 cycles) embed in the full length of the microchannels. On the 2nd and 3rd configurations, SHMs (3.5 cycles) covered the front half and the rear half of the microchannels, respectively. The 4th configuration is plain-wall microchannels without integrating SHMs. The detailed dimensions of microchannels and SHMs are summarized in TABLE I.

The design of micro-device is illustrated in Fig. 1c and d. As shown in Fig. 1c, a 1-mm-diameter inlet port, 1-mm diameter-outlet port, and two 1-mm-diameter pressure ports were fabricated in the micro-device. The inlet to exit pressure drop was measured between the two pressure ports. Flow stabilizers at the entrance to the inlet manifold were established to evenly distribute flow. To minimize heat loss, two thermal isolation gaps (air gaps) were etched on both sides of a microchannel array to reduce heat loss as displayed in Fig. 1d. An aluminum thin film heater was deposited onto the back side of the silicon microchannels (Fig. 1d). The heater area (10 mm × 2 mm) is the same as that of the total base area of a microchannel array. This thin-film heater is also a thermistor to measure the average wall temperature on the bottom [17].

III. DEVICE AND FABRICATION

The micro-devices [20] were made of bonded silicon/glass wafers by a microfabrication process (Fig. 2) detailed in previous studies [17] and [21]. First, a thin film heater was deposited to provide approximately constant heat flux input on the backside of microchannels. This thin-film serves also as a thermistor to measure the average wall temperature. Pre-patterned silicon wafer was then covered by photoresist (Shipley S1827) and thermal oxide layers (500-nm-thinkness). Shallow grooves were etched in the first deep reactive ion etching (DRIE) step (Fig. 2a) and their edges define the
embedded micro-mixers. The depth of SHMs, \(d_{SHM}\), in this step is 50 \(\mu m\). Then, the mask layers were re-patterned for microchannels as shown in Fig. 2b. In the second DRIE step, deep trenches were etched to achieve the required depth, 250 \(\mu m\). When the etching depth reached to the top of reentrant profiles of sidewalls, the top of reentrant structures started to create residual structures [22] as reported in a previous study (Fig. 2c). These residual structures formed submicron fins/tips on all edges of SHMs and the depression were gaps between these fins/tips. A Pyrex glass cover (500-\(\mu m\)-thickness) was anodically bonded onto the silicon substrate to package the micro-device (Fig. 2d). A single test chip as shown in Fig. 1b (length 30 mm; width 10 mm; thickness 1 mm) was cut from the wafer by a dice-saw cutting machine.

The major features of the integrated SHM micromixers on the bottom wall in microchannels are shown in scanning electron microscope (SEM) images (Fig. 3). In each cycle, a group of staggered herringbone grooves consists of 12 SHMs. In each microchannel, 7 or 3.5 cycles were embedded on the bottom wall as indicated in Fig. 3a. The major dimensions of a herringbone mixer are specified in Fig. 3b. The depth of each herringbone groove is 50 \(\mu m\).

IV. EXPERIMENTAL PROCEDURES

The micro-device is placed in the middle of the test package module (Fig. 4a) as detailed in our previous study [17], which provides hydraulic ports and electrical connections. Major components of the experimental setup include an optical imaging system, a data acquisition unit, and an open loop for coolant supply (Fig. 4b). A pressurized coolant tank was utilized to supply HFE-7000 (its major fluid properties listed in Table II), which was degassed prior to testing and pumped by compressed nitrogen (N\(_2\)). Mass fluxes were monitored by an Omega FPR-1501 liquid flow sensor. Electrical power was provided by a high-precision digital and programmable power supply (B&K Precision XLN10014). The voltage on the micro heater was monitored by an Agilent digital multi-meter. As shown in Fig. 4a, two pairs of K-type thermocouples were used to monitor the inlet and outlet fluid temperatures, respectively. Flow rate, local pressure, inlet and outlet temperature, and the voltage and current of the heater were recorded automatically by a data acquisition system developed from NI LabVIEW\textsuperscript{®}. The visualization system comprised of a high-speed camera (Phantom V 7.3) and Olympus microscope (BX-51) fitted with 400X optical magnifications. The visualization, with approximate 40,000 frames per second (FPS) at a resolution of 256 \(\times\) 256 pixels, allows to carefully study bubble dynamics and two-phase flow patterns.

The temperature of the micro heat exchanger in steady state, i.e. the temperature difference between the device and the ambient without fluid flow, was evaluated and plotted as functions of input heat flux using linear curve fitting. Then, the obtained curve was used to accurately estimate heat loss [23]. The thermistor (thin film heater) was calibrated using an isothermal oven prior to tests. The temperature as a function of electric resistance was obtained by curve fitting with an accuracy of 0.8 K (TABLE III).

The experiments were conducted at three mass fluxes of 1018 kg/m\(^2\)·s, 1527 kg/m\(^2\)·s, and 2206 kg/m\(^2\)·s. In each run, a mass flux was maintained constant. A uniform heat flux was applied with a step of approximately 2 W to 4 W until the CHF conditions are reached. In each step measurement, 120 sets of steady-state data (voltage, current, local pressure, both inlet and outlet temperatures) were recorded through a data acquisition system in a 4-minute interval. The experimental results were reduced by averaging steady-state data.

V. DATA REDUCTION

Using the collected data including voltage \(V\), current \(I\), and pressure measurements, the average wall temperature, and inlet and outlet temperatures, average heat transfer coefficient

![Fig. 2. Major microfabrication steps of a microchannel heat sink with embedded SHM micromixers [20]. (a) Defining mixers. (b) Defining channels. (c) Etching mixers and microchannels. (d) Bonding cover layer.](image)

![Fig. 3. SEM images of the integrated SHM micromixers on the bottom wall in microchannels [20]. (a) SEM image of micromixers in a microchannel. (b) A close-look of micromixers.](image)
can be derived. From the measured voltage and current, the electric power input $P$, and heater resistance $R$, respectively, were determined as follow:

$$ P = VI $$

(1)

and

$$ R = V / I $$

(2)

Then the effective heat flux was calculated after excluding the heat loss, $Q_{\text{loss}}$, (calibrated between the environment and the test micro-device) from the measured power, $P$, as follow,

$$ q_{\text{eff}}'' = \frac{P - Q_{\text{loss}}}{A} $$

(3)

where $A$ is the heating area. Then, the contribution of latent heat during phase-change heat transfer was calculated as,

$$ Q_{\text{latent}} = P - Q_{\text{loss}} - Q_{\text{sensible}} $$

(4)

where $Q_{\text{sensible}}$ is the sensible heat resulting from the subcooled liquid. It was calculated as,

$$ Q_{\text{sensible}} = GA_c C_p (T_o - T_i) $$

(5)

where $G$ is the mass flux; $A_c$ is the cross-sectional area; $T_o$ is the outlet fluid temperature; and $T_i$ is the inlet fluid temperature. When flow boiling occurs,

$$ Q_{\text{sensible}} = GA_c C_p (T_{\text{sat}} - T_i) $$

(6)

where $T_{\text{sat}}$ is the saturated temperate of working fluid. From the pre-calibrated temperature-electrical resistance slope, $K$, the average temperature of the thin film heater (i.e. The thermistor) was calculated as,

$$ \bar{T}_{\text{heater}} = K (R - R_a) + T_a $$

(7)

where $R_a$ is the resistance of the micro heater at ambient temperature $T_a$. The average wall temperature at the base of the microchannels was then calculated as,

$$ T_{\text{wall}} = \bar{T}_{\text{heater}} - \frac{q_{\text{eff}}''}{k_s} $$

(8)

where $r$ and $k_s$ are the substrate thickness, thermal conductivity of silicon, respectively.

The fin efficiency $\eta_f$, of a finite fin was estimated as

$$ \eta_f = \frac{\tanh (mH)}{mH} $$

(9)

The $\eta_f$ is used to characterize fin performance and then to calculate the total effective heat transfer areas, where the parameter $m$ was calculated as,

$$ m = \sqrt{2h} \frac{(L + W)}{k_s W L} $$

(10)

before the onset of the nucleate boiling (ONB), the fluid temperature, $T_f$, is estimated as,

$$ T_f = \frac{T_o + T_i}{2} $$

(12)

When flow boiling occurs,

$$ T_f = \frac{(T_{\text{sat}} + T_i)}{2} $$

(13)

To normalize the sensible heat transfer in both single and two-phase regimes, the Nusselt number of single-phase heat transfer was calculated as,

$$ Nu = \frac{h_{sp} D_h}{k_f} $$

(14)

where $D_h$ is the hydraulic diameter of the microchannel and $k_f$ is the thermal conductivity of the working fluid (liquid phase).

Then, the latent heat transfer rate was evaluated as the average two-phase HTC, $h_{tp}$, in two-phase regime as below,

$$ h_{tp} = \frac{Q_{\text{latent}}}{(\sum (WL + 2HL \eta_f)(\bar{T} - T_{\text{sat}}))} $$

(15)
Major physical properties of dielectric fluid HFE-7000 are given in TABLE II. $T_{sat}$ is a function of working pressure, $p$, in the middle of a microchannel, which is estimated as,

$$p = p_i - \frac{\Delta p}{2}$$

where $p_i$ is the inlet pressure; and $\Delta p$ is the pressure drop.

The uncertainties of the measured parameters were detailed in TABLE III. The uncertainties of the derived parameters were estimated by analysis of the propagation of uncertainty [24].

VI. RESULTS AND DISCUSSION

A. Boiling Curves

Average wall temperatures were plotted a function of effective heat flux in Fig. 5 at three mass fluxes. Boiling curves illustrate that the embedded SHMs can significantly enhance single and two-phase heat transfer rates, but not always. In Fig. 5a, the reduction of superheat $\Delta T$ is 6.3 K for a mass flux of 1018 kg/m$^2$·s at an effective heat flux of 80 W/cm$^2$. Such a significant enhancement primarily results from strong mixing induced by SHMs and enhanced nucleation boiling from the artificial cavities formed by grooves and/or residual structures [25], [26]. Noticeably, the boiling curves of microchannels with front-half (2nd configuration) and rear-half (3rd configuration) coverages of SHMs show different trends from the 1st configuration with fully-covered SHMs. In Fig. 5, the wall temperature before nucleate boiling is reduced on three microchannel configurations with SHMs. However, distinct trends were observed in the two-phase regime. On the 2nd configuration, the enhancement of phase-change heat transfer is significant and achieved the lowest wall temperature at the highest mass flux of 2206 kg/m$^2$·s. On the 3rd configuration, the reduction of wall temperatures is not noteworthy as other two microchannel configurations with SHMs. On the contrary, the wall temperatures were elevated at a higher mass flux of 1527 and 2206 kg/m$^2$·s since boiling curves of two microchannel configurations (3rd and 4th) overlap (Fig. 5b and c). The Nu in the single-phase regime and HTC in two-phase regime were derived to further explain these observations.

B. Characteristics of Single and Two-Phase Heat Transfer

1) Single-Phase Heat Transfer: To gain a better understanding of the impacts of micro-mixers on single-phase and two-phase heat transfer, the percentage of sensible heat transfer during entire heat transfer processes was calculated. The percentage transferred by single-phase flow is defined as $Q_{sensible}/(Q_{latent} + Q_{sensible})$. It should be noted...
that this percentage of $Q_{\text{sensible}}$ is not significant for water-based coolants because latent heat, $h_{fg}$, of water is 14.9 times higher than that of HFE-7000 and hence the contribution of sensible heat transfer on water during flow boiling is a small fraction. On the contrary, with HFE-7000, the sensible heat contribution is significant and hence cannot be neglected as observed from this experimental study (Fig. 6). As shown in Fig. 6, the percentages of sensible heat transfer are compared between four microchannel configurations as detailed in the Section II.

Compared to that in plain-wall microchannels, the contribution of sensible heat transfer was promoted on the 1$^{\text{st}}$ configuration at mass fluxes of 1018 kg/m$^2$.s (Fig. 6a) and 1527 kg/m$^2$.s (Fig. 6b). This promotion faded away when mass flux increased to 2206 kg/m$^2$.s (Fig. 6c). In addition, such an enhancement of sensible heat transfer contribution on the 1$^{\text{st}}$ configuration becomes insignificant in two-phase regime, i.e., less than 10%, and holds the lowest ratio among all configurations. Particularly, among all configurations, the single-phase contribution degrades most noticeably on the 1$^{\text{st}}$ configuration. Since high heat flux and mass flux lead to high mean velocity $\bar{u}$ and thus high Reynolds number ($Re = \bar{u}D_h / \nu$, which cannot be determined in this study since some vapor properties of HFE-7000 are not available), the observation on the 1$^{\text{st}}$ configuration implied that the effects of the SHMs on sensible heat transfer is more significant at a condition of low Reynolds number than that at high Reynolds number, especially on the 1$^{\text{st}}$ configuration.

On the 2$^{\text{nd}}$ and 3$^{\text{rd}}$ configurations, the overall contribution of single-phase heat transfer keeps decreasing with increasing heat fluxes as well, i.e., Reynolds number. However, there is a meaningful difference from the 1$^{\text{st}}$ configuration. The contribution of single-phase heat transfer gradually increases with increasing mass fluxes on both 2$^{\text{nd}}$ and 3$^{\text{rd}}$ configurations as shown in Fig. 6. The overall enhancement of sensible heat transfer reaches the highest at the highest mass flux of 2206 kg/m$^2$.s on the 2$^{\text{nd}}$ configuration (Fig. 6c). This results indicated that the SHMs at the entrance area can generate stronger mixing in two-phase flow regime with higher Reynolds number and hence considerably enhance the sensible heat transfer. This impact is opposite to that on the 1$^{\text{st}}$ configuration. This mixing phenomena is further validated and described in Fig. 8. Additionally, the experimental results on the 3$^{\text{rd}}$ configuration as shown in Fig. 6 indicated that SHMs at the exit area (rear-half) of microchannels didn’t function well since the sensible heat transfer ratio becomes even lower than that on the plain-wall configuration at all three mass fluxes.

The enhancements of heat transfer need to be discussed in two portions, i.e., the single-phase heat transfer portion and two-phase portion. Based on the transport efficiency of $Q_{\text{sensible}}$, the Nusselt numbers (Nu) of single-phase portion are compared to further reveal the mechanisms of the enhanced sensible heat transfer as shown in Fig. 7(a, c and e). Before the onset of nucleate boiling (ONB), the SHMs generally enhance sensible heat transfer under all working conditions no matter where the SHMs are located. Single-phase Nu numbers on all three microchannel configurations with SHMs are larger than that of the plain-wall microchannels.

When nucleate boiling occurs, the dimensionless heat transfer curves agree with the tendencies of the percentage of $Q_{\text{sensible}}$ in Fig. 6. First, the consequences on the 1$^{\text{st}}$ configuration indicate that the enhancement of sensible heat transfer disappeared at the highest mass flux of 2206 kg/m$^2$.s (Fig. 6c and Fig. 7e). Nonetheless, on the 2$^{\text{nd}}$ configuration, sensible heat transfer becomes more significant at a higher mass flux working condition. Thus, at the mass flux of 2206 kg/m$^2$.s, the 2$^{\text{nd}}$ configuration has the highest single-phase Nu among four configurations (Fig. 7e). In addition, the single-phase Nu number on the 3$^{\text{rd}}$ configuration is (Fig. 7a) even lower than that on the 4$^{\text{th}}$ configuration, i.e., the plain-wall microchannels (Fig. 7c and e), which illustrated that the 3$^{\text{rd}}$ configuration is not favored by latent heat transfer, especially in high Re conditions.

![Fig. 6. Percentage of sensible heat transfer of total heat transfer during two-phase flow boiling in microchannels as $Q_{\text{sensible}}/(Q_{\text{latent}}+Q_{\text{sensible}})$. The mass flux is (a) 1018 kg/m$^2$.s, (b) 1527 kg/m$^2$.s and (c) 2206 kg/m$^2$.s.](image_url)
Thus we hypothesize that the single-phase flow regime in SHMs may be similar with these in multi-louvered geometries, where there are two asymptotic flow regimes: “duct” flow regime with flow directed in the axial direction and “louver” flow in the louver direction [27], [28]. The former is associated with a low heat transfer rate and the latter with a high heat transfer rate [29]. Thus, the trends of sensible heat transfer as shown in Fig. 6 and Fig. 7 can be caused by the reduced “flow efficiency”, i.e., the ratio of flow allocated to the louver direction, because the SHM dimensions and locations (such as the angle between two asymmetric arms, the groove width, etc.,) were not optimized in this study [29]. As a result, the SHMs in certain configurations failed to direct sufficient fluids in the louver direction and led to non-enhancement of sensible heat transfer in high Reynolds number operating conditions.

2) Two-Phase Heat Transfer: Basing on the “V” or “N”-shaped two-phase heat transfer curves in Fig. 7b, d and f, the two-phase heat transfer curves can be separated into two regions. In the initial region, heat transfer could be dominated by convection and nucleate boiling. The reason that HTC decreases with increasing input heat flux on all configurations could result from two aspects: 1) the degraded fluid properties such as reduced thermal conductivity and density and 2) the transition from bubbly flow to slug flow [30]. Moreover, HTC in microchannels with SHMs (1st, 2nd and 3rd configurations) in this flow regime is noticeably higher than that in plain-wall microchannels (4th configuration). It demonstrated that two-phase HTC can be effectively enhanced by promoting nucleation boiling through creating artificial cavities using grooves or residual structures as well as by inducing mixing.

Fig. 7. Sensible and latent heat transfer during flow boiling in microchannels with and without SHMs. (a), (c) and (e) Are single-phase Nu as a function of heat flux. (b), (d) and (f) Are two-phase heat transfer coefficients representing two-phase portion. The onsets of nucleate boiling (ONB) are highlighted.
The other region prevails in churn flow and annular flow [30] as validated in the visualization study (Fig. 8). In this flow regime, the convection and thin-film evaporation primarily determines the heat transfer, which is expected to be enhanced by inducing mixing. HTC curves on the 1st, 3rd, and 4th are of the same shape. However, HTC on the 2nd configuration are quickly dropping in the high heat flux conditions, which induces an “N”-shaped curve. It indicated that the low heat transfer rate should result from the local dry-out owing to liquid supplying crisis [30] because of the absence of capillarity in the rear section of microchannels as discussed in Section VI-C. Moreover, HTC on the 2nd configuration become more prominent with increasing mass fluxes, thus, it has the highest HTC at the highest mass flux of 2206 kg/m²·s (Fig. 7f). This observation is consistent with the highest sensible heat transfer on the 2nd configuration (Fig. 7e), which strongly suggested that the induced advection and enhanced nucleate boiling by SHMs located in the front half section of channels plays a more critical role that those in the other locations. On the other hand, the 3rd configuration with SHMs shows different trends. At relatively low mass fluxes of 1018 and 1527 kg/m²·s, the enhancement of HTC is noteworthy on the 3rd configuration and nearly same on the 1st configurations (Fig. 7b and d). However, such an enhancement was quickly degraded to that of the 4th configuration (plain-wall microchannels) when mass flux increased to 2206 kg/m²·s (Fig. 7f). Mixing generated by the SHMs in rear half section appears not to induce favorable advection and have no opportunities to enhance nucleate boiling since annular flow and thin film evaporation prevails there as discussed in the Section IV.

Furthermore, it is interesting to compare overlapped sections of two-phase heat transfer curves on three configurations with SHMs. Specifically, at mass flux of 1018 kg/m²·s, these three configurations perform nearly unchanged at a heat flux range of 20 to approximately 60 W/cm². The overlapped sections were also noted at a mass flux of 1527 kg/m²·s from approximately 40 to 90 W/cm². The overlapped sections indicated that two-phase heat transfer rate is independent on location of SHMs.

These observations suggest that phase-change heat transfer in microchannels with SHMs is strongly related to the locations of SHMs. However, at this stage of study, the relationship between HTC and flow patterns was not quantified due to the complexity of transitional two-phase flow patterns with intense mixing.

C. Flow Patterns

Thermo-hydraulic characteristics of two-phase flow in microchannels strongly depend on flow patterns. Local flow patterns in microchannels with and without SHMs were visualized in this study. Two cases, i.e., at low (Fig. 8a) and high (Fig. 8b) vapor qualities, are compared. Here, the vapor quality is calculated as the vapor quality at the exit,

$$\chi = \frac{Q_{\text{latent}}}{G_A L_v}$$  

where $L_v$ is the specific latent heat of vaporization.

In this experimental study, SHMs can induce intense mixing in two-phase flow and hence reduce bubble size (Fig. 8a) compared to plain-wall microchannels under similar working conditions. Therefore, large vapor slugs were not observed in the 1st configuration. This indicates that bubbly flow can prevail in microchannels with SHMs when vapor quality is low. However, at low vapor qualities, the vapor slugs exist in the sections with plain-wall on the 2nd and 3rd configurations. Fig. 8a is a clear indication that a vapor slug was rebuilt after a two-phase flow went through the SHMs (the 2nd configuration). Furthermore, as illustrated in Fig. 8a, a train of vapor slugs was observed in the plain-wall microchannels (the 4th configuration), which is in agreement with results reported in [19]. Interestingly, a wriggly two-phase flow pattern was noted on the 3rd configuration when the vapor slug enters the area covered by SHMs. The wave of two-phase flow has a wavelength, $\lambda$, which is just exactly equal to the axial length of a cycle of SHMs on the bottom of microchannels (Section II). This result is strongly recommended that SHMs can manipulate the flow direction and even flow structure of two-phase flows.

With vapor quality increasing, under the effects of vortices induced by SHMs, the large liquid slugs or droplets are reduced into small droplets as two-phase flow becomes opaque under an optical microscopy (Fig. 8b). Generally, the local mixing effect of SHMs produces finer droplets and smaller bubbles as the whole channel of the 1st configuration, however, the situations on the 2nd and 3rd configurations are more complicated. On the 2nd configuration, wriggly flow formed in the first-half section with SHMs transformed to annular flow in the rest of section with plain-wall [19]. On the 3rd configuration, churn flow formed in the plain-wall section was regulated into a “wriggly” flow by SHMs in the downstream. However, the “wriggly” flow was not observed on the 1st configuration (Fig. 8b), where the mixing effect may be
too strong to form regular flow patterns. This also implied that the interaction between the section with and without SHMs is intense and complex. On the 4th configuration, the flow patterns transit to churn flow and/or annular flow [31] as indicated by the visible liquid films and droplets (Fig. 8b).

In summary, SHMs induce vortices and strong mixing, which can significantly change two-phase flow patterns in microchannels. The intrinsic link between the flow patterns and two-phase transport needs to be further studied.

D. Thermo-Hydraulic Properties

First, the total pressure drop in microchannels with and without SHMs during flow boiling was measured and compared in Fig. 9. The experimental results indicate that the integration of SHMs can cause a sharp increase of pressure drop, for example, up to 300% in both single-phase and two-phase regimes. The growth of boundary layers on the inner walls of microchannels with SHMs is supposed to be interrupted because of the induced mixing [18]. Accordingly, flow resistance would increase as a result of discontinuous boundary layers. Additionally, the SHMs and residual fins/tips on the edges of SHMs significantly increase the roughness of the bottom wall in the microchannels.

Interestingly, according to the experimental results on the 2nd and 3rd configurations (Fig. 9), the flow resistances are not reduced with decreasing the quantity and coverage of SHMs in microchannels. On the contrary, the flow resistances are significantly elevated, especially, on the 3rd configuration. These results reveal that the flow patterns alternated by SHMs (Fig. 8) significantly affected pressure drop of two-phase flow [32]. For example, if the SHMs are only located at the exit end of the microchannels, it would increase the flow resistance at the exit, which prevents the vapor from flowing out of channels. As a result, bubble confinement (Fig. 8a) and then severe reverse flows in the entrance area were induced [3].

To further analyze the thermo-hydraulic performance, the effective heat fluxes are plotted as a function of pumping power for a given wall temperature in Fig. 10. The idea pumping power is calculated as,

\[ P_p = \frac{\Delta p G A_c}{\rho} \]  

(18)

where \( \rho \) is the density of working fluid. This figure combined the capability of heat dissipation and required \( P_p \), which could hopefully provide some guidance in practical applications. For example, in electronics cooling, the thin-film heater is the mimic of the semiconductors under the heat sink. For given wall temperatures at 40 °C, 60 °C and 80 °C, the pumping power increases with the thermal power of the electronics, i.e., effective working heat fluxes. The experimental study shows that although microchannels with SHMs would consume more pumping power as shown in Fig. 9 and 10, they could deliver much higher working heat fluxes than plain-wall microchannels do, especially at a relatively low temperature (Fig. 10a). It also indicated that plain-wall microchannels has a narrow range of working temperature. As illustrated by the dash line in Fig. 10c, plan-wall microchannels reached CHF conditions before wall temperature reached 80 °C. In another word, a plan-wall configuration cannot maintain the required working temperature at 80 °C.

Generally, the 1st and 2nd configurations work in a wider range of heat fluxes for all given working temperatures. The 1st configuration needs lowest pumping power at high heat flux conditions. The 3rd configuration can only work well at a lower temperature of 40 °C since working heat flux fails to increase with increasing pumping power at working temperatures of 60 °C and 80 °C [purple lines in (Fig. 10b and c) because of the low heat transfer rate resulting from vapor slug confinement as shown in Fig. 8a.

In order to manage the pumping power, the structure of SHMs should be optimized to better manage flow structures. Additionally, even though the contribution of the residual fins or tips (Fig. 3b) on the total pressure drops has not been
quantified, the pressure drop of microchannels with SHMs could be reduced by removing residual structures as shown in Fig. 3b. The removal can be done by improving DRIE process.

E. Flow Instabilities

Previous studies demonstrated that the use of dielectric coolant (e.g. HFE-7100) can alleviate static flow instabilities in plain-wall microchannels as a result of small surface tension [33]. Bubble confinements and reverse flows, which are pronounced in flow boiling of water [34], were not significant in plain-wall microchannels with HFE-7100. The large-amplitude oscillations of temperatures and pressure drops during flow boiling of water [33] were not observed on HFE-7100.

In this study, at a moderate mass flux of 1527 kg/m²·s and heat flux of 104 W/cm², the transient wall temperature and pressure drop in the parallel microchannels were measured (Fig. 11). The standard deviations (SDs) of temperatures and pressure drops as functions of time in 4 minutes were plotted. The experimental results demonstrated that the two-phase flow boiling of HFE-7000 in this parallel microchannels is stable on the 1st and 4th configurations in terms of fluctuations of wall temperature and pressure drop. Specifically, on the 1st and 4th microchannel configurations, the SDs of wall temperature are only 0.007 °C and 0.004 °C; the SDs of pressure drops are only 0.169 kPa and 0.100 kPa, respectively. The results indicated that the fully-covered mixing does not lead to significant flow instabilities.

However, the partially-covered SHMs lead to relatively more pronounced flow instabilities in terms of fluctuations of wall temperature and pressure drop as shown in Fig. 11. The SDs of wall temperatures increased to 0.308 °C and 0.413 °C; while the SDs of pressure drops reached 4.511 kPa and 0.402 kPa on the 2nd and 3rd configurations, respectively. The wall temperature fluctuations are more than 40 times higher than that in the microchannels with fully-covered SHMs. In summary, the 2nd configuration has a higher amplitude of fluctuations in terms of both temperature and pressure drops. In addition, the 3rd configuration has the most intense fluctuations in terms of wall temperatures (Fig. 11a). These instabilities could be explained as a result of spatial non-uniform (Fig. 8) and temporal transitional flow patterns in
be destroyed by discontinuous boundary layers and strong mixing. It appears that the channel is filled with vapor since the SHMs on the bottom walls can be clearly visualized by an optical microscope. Most importantly, menisci were recorded in SHMs by comparing the dry microchannels (Fig. 3a) and the microchannels near CHF conditions (the bottom picture in Fig. 13a). This strongly suggests the existence of fluid on heating surfaces (at least on the SHM-structured bottom walls) near the CHF conditions because of capillarity induced by SHMs. In this case, vapor films during flow boiling in plain-wall microchannels cannot be formed or sustained. The integration of SHMs can delay CHF conditions because of the induced capillarity on the bottom walls.

Generally, CHF is higher on these configurations with SHMs in the rear channels. Moreover, results suggest a strong CHF dependence on the location of SHMs. The 1st configuration has the highest CHF. Additionally, slopes of CHF-G curves on microchannels with partially covered SHMs, i.e., 2nd and 3rd configuration, are smaller than that on the 1st configuration. The reason could be that the CHF is also affected by two flow instabilities due to the unstable flow patterns in non-uniform structured microchannels as previously discussed in Section VI-E.

VII. Conclusion

SHMs were embedded on the bottom walls of microchannels to create strong mixing without using external actuators. In this study, heat transfer rate, pressure drop, flow instabilities, CHF, and flow patterns were experimentally characterized on four microchannel configurations with various location and coverage of SHMs.

These results indicate that the location and coverage of SHMs are critical in governing heat and mass transfer since they essentially determine the flow structures in microchannels. For example, due to the induced mixing, the size of large vapor slugs and liquid droplets was significantly reduced by SHMs. Also, a “wriggly” flow pattern was only observed on the area covered by SHMs, which illustrated the effects of mixing on flow patterns. As a result, flow instabilities are more pronounced in microchannels with partially-covered SHMs.

Equally important, the significant enhancement of CHF was demonstrated using SHMs on all working conditions in this study. Two primary enhancement mechanisms of CHF are proposed: (1) breakage of continuous vapor film on the walls by strong mixing and (2) additional liquid supply enabled by capillarity induced by SHMs.

Compared to water, sensible heat plays a more important role during flow boiling of HFE-7000. Both single and two-phase heat transfer rates are significantly enhanced by SHMs. However, heat transfer rates in microchannels with partially and fully-covered SHMs show different trends.

Moreover, the thermo-hydraulic analysis indicates that the working range is extended by using SHMs in microchannels so that the microchannel heat sink can maintain a required working temperature at a higher heat load. The increased pressure drop could be caused by the change of the flow structure as well as additional roughness resulting from SHMs and residual nano-fins. It was further observed that the pressure...
drop does not increase with the coverage of SHMs, which indicates that the change of flow structure appears to play a major role in determining the pressure drop.

REFERENCES


Fanghao Yang received his B.Eng. degree from the Beijing University of Aeronautics and Astronautics, Beijing, China, in 2008, and Ph.D. in mechanical engineering from the University of South Carolina in 2013. From 2009 to 2013, he was a Research Assistant with the department of mechanical engineering at the University of South Carolina. Now, he is a postdoctoral researcher at the University of South Carolina. His research focus is on microfabrication, microfluidics, phase-change heat transfer in MEMS, and electronics cooling.

Mohammad Alwazzan received his B.S. and M.E. degrees in mechanical engineering from the University of South Carolina, Columbia, SC, USA, in 2011 and 2012, respectively. Recently, he is pursuing the Ph.D. degree in mechanical engineering at the University of South Carolina, and is supervised by Prof. Chen Li. His research interests focus mainly on condensation heat transfer and developing novel micro/nano-engineered surfaces to enhance dropwise condensation.
Wenming Li received his M.S. degree in Fluid Mechanics from Huazhong University of Science and Technology, Wuhan, P.R. China, in 2011. He is currently pursuing the Ph.D. degree in mechanical engineering at the University of South Carolina. From fall of 2013, he has been a Research Assistant in the Department of Mechanical Engineering, University of South Carolina at Columbia. His research interest includes research in micro/nano-scale two-phase heat transfer and microfabrications.

Chen Li received his M.S. in mechanical engineering from the University of Nevada, Reno, NV, and Ph.D. degree in mechanical engineering from Rensselaer Polytechnic Institute, Troy, NY, USA, in 2006. He has been an assistant professor in the department of mechanical engineering at the University of South Carolina since 2009. He conducts researches in micro/nano-scale two-phase heat and flow physics, prediction, control, and modeling. He aims at basic research in developing and verifying theories for two-phase transport behaviors, and the application of these theories towards controlling two-phase heat and flows at micro/nano-scale.