Enhanced flow boiling in microchannels using auxiliary channels and multiple micronozzles (I): Characterizations of flow boiling heat transfer

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ABSTRACT

Flow boiling in parallel microchannels can be dramatically enhanced through inducing self-excited and self-sustained high frequency two-phase oscillations as demonstrated in our previous studies using a two-nozzle microchannel configuration. Two-phase mixing induced by the rapid bubble collapse is shown to be the major enhancement mechanism. However, in the two-nozzle configuration microchannels, the mixing effect is limited to the downstream of the microchannels, meaning that only half length of the entire microchannel is functionalized as designated. In this study, a four-nozzle microchannel configuration is developed with an aim at extending the highly desirable mixing effect to the entire channel. Flow boiling in the four-nozzle configuration microchannels is experimentally studied with deionized water and the mass flux ranging from 120 kg/m² s to 600 kg/m² s. The onset of nucleate boiling temperature is considerably reduced by ~14% because of more nucleation sites created by the multiple nozzles. Equally important, the improved microchannel configuration successfully extends the mixing to the entire channel as validated by the enhanced heat transfer rate and visualization study. Compared to the previous two-nozzle configuration, the overall heat transfer coefficient (HTC) is significantly improved primarily owing to the enhanced nucleate boiling. For example, the peak overall HTC of 262 kW/m² K is achieved at a mass flux of 150 kg/m² s, accounting for ~83.7% enhancement. Additionally, the peak effective HTC reaches 97.6 kW/m² K, accounting for ~67% enhancement.

1. Introduction

As high power electronic devices miniaturize down to nanometer size, high cooling capability is demanded. Among all cooling techniques, flow boiling in microchannels is one of the most effective and efficient approaches to cool high power electronic devices [1,2]. However, it is challenging to improve the boiling heat transfer in microchannels without compromise because of the limitations primarily imposed by the confined vapor bubbles [3], which usually lead to flow choking and premature dryout [4,5]. The persistent bubble confinement inherently retards primary and efficient heat transfer modes including nucleation, convection, and evaporation. Extensive studies have been conducted to enhance flow boiling in microchannels in last decade [4,6–10]. Numerous techniques have been explored to promote flow boiling performance through structure and surface modifications [5,8,11] or change of working fluids [12–14]. These include enhancing nucleate boiling by increasing number of nucleation sites through integrating micro/nano-structures [8,11,15] as well as promoting thin film evaporation by creating micro-structure on the bottom surface of channels [5,16]. However, these approaches have not shown strong effectiveness on removing or managing the bubble confinement. The existence of confined bubbles/vapor slugs in microchannels severely deteriorates the heat transfer performances. For example, confined bubbles in microchannels during boiling process can induce severe two-phase flow instabilities and hence, result in liquid flow crisis, eventually retarding highly efficient thin film evaporation [2,3]. Therefore, rapidly and passively removing these confined bubbles can be an effective method to substantially improve microchannel flow boiling performances.

Technically, bubble collapse/removal can be achieved by direct interface condensation [17–20]. But a slow bubble collapse/removal cannot meet the need of high working heat loads, where generation of vigorous vapor would result in chaotic and violent two-phase flows. Many techniques have been developed to remove or manage confined bubbles. For example, inlet orifices or restrictors were designed to overcome the bubble flow reversal [3,10]; taper geometry was explored to enhance bubble removal by changing cross-section area [21]. However, they either induce...
addition pressure drop or require high flow rate. Most recently, without using external controllers and forces, a microfluidic transistor was devised to timely collapsing confined bubbles and generate highly desirable two-phase mixing by inducing high frequency self-sustained two-phase oscillations (TPOs) in our previous studies [4,22]. However, with one pair of nozzles located in the middle of each main channel, persistent confined vapor slugs in the upstream led to a limited mixing range. Additionally, less nucleation sites are unfavorable to enhance flow boiling heat transfer rate.

To enhance nucleate boiling and extend the mixing range, in this study, an improved microchannel configuration is proposed. For better compactness, an auxiliary channel is shared by two neighboring main channels. The new configuration is developed based on the experimental and theoretical study of two-nozzle microchannel configuration [4,9]. On the improved configuration, nucleate boiling can be further enhanced and the range of mixing can be extended to the entire channel.

2. Experimental apparatus and procedures

2.1. Design of the device architecture

As shown in Fig. 1(a–c), a four-nozzle microchannel configuration is developed with an aim at extending the mixing to the entire channel length, which can potentially double the mixing range compared to our previous two-nozzle microchannel configuration (Fig. 1(d–g)) [4]. Herein, an array of five parallel microchannels (W = 200 μm, H = 250 μm, L = 10 mm), where each main channel is connected to two auxiliary channels (H = 250 μm, W = 60 μm, L = 8 mm), is developed to experimentally characterize flow boiling heat transfer. Each 60 μm wide auxiliary channel has four 20 μm wide nozzles connected to the main channel. An inlet restrictor (H = 250 μm; W = 20 μm; L = 400 μm) is integrated in each main channel to trap elongated bubbles. A resistor, which serves as both a micro heater to generate heat flux and a thermistor to measure the wall temperature, is deposited onto the back side of the silicon chip. The heating area (10 mm × 2 mm) is identical to the total base area of microchannel arrays. A comparison of main features between the present design and our previous microchannel configuration is summarized in Table 1. More descriptions of device and experimental details can be found in our previous studies [4,22].

Fig. 1. The design and major dimensions of the present four-nozzle microchannel configuration. The two-nozzle microchannel configuration [4,22] is included for comparisons. (a) The improved four-nozzle microchannel configuration. (b) SEM image of the micronozzle distribution with a pitch of 2 mm. (c) SEM image of the micronozzles. (d) The two-nozzle configuration developed in our previous study [4]. (e) A close look of the two micronozzle locations [4]. (f–g) SEM images of a pair of micronozzles and a close-look of the micronozzle, respectively [4].
2.2. Microfabrication of the device

The microdevice was made from a silicon wafer bonded to a Pyrex wafer by a standard microfabrication process detailed in our previous study [4]. This process started with a double-side-polished n-type (1 0 0) silicon wafer. First, before fabricating a microheater, a 1 ± 0.01 μm thick thermal oxide layer was grown on both sides of the silicon wafer. The silicon oxide film provides electrical insulation for the micro heater and acts as a mask for deep reactive ion etching (DRIE) in the subsequent microfabrication steps. Next, a thin film heater was fabricated through a lift-off process on the backside of the wafer. In the lift-off process, a pattern mold of a thin film heater was prepared with a negative photosist by photolithography first. Then, a 1.5 ± 0.05 μm thick layer of aluminum was deposited followed by a 63 A thick of titanium layer by DC sputtering. Once the thin films were successfully deposited, a thin film heater was formed by a lift-off process. A 1 ± 0.05 μm thick plasma-enhanced chemical vapor deposition (PECVD) oxide layer was then deposited to protect the thin film heater in the subsequent fabrication processes.

After the heater was formed on the backside, a patterned mask of microchannels on the top side of the wafer were etched in the silicon oxide through photolithography and reactive ion etching (RIE). The area under the oxide mask was protected and the remaining areas were etched out to create 250 ± 3 μm deep trenches by DRIE.

A Pyrex glass wafer was anodically bonded to the silicon substrate to seal the device. The transparent glass cover also serves as a visualization window. RIE was used to remove oxide coatings on the backside to expose the contact pads after anodic bonding. The individual microchannel test chips (length 30 ± 0.005 mm; width 10 ± 0.005 mm; thickness 1 ± 0.005 mm) were cut from the wafer by a dice saw.

2.3. Experimental setup

The experimental setup and test procedures detailed in our previous study [4] were used and applied to the present study. Major components of the experimental setup include an optical imaging system, a data acquisition unit, and an open coolant loop. A pressurized water tank was used to supply deionized (DI) water, which was degassed prior to tests and pumped by compressed nitrogen (N2). Flow rates were measured by an Omega flow meter (FLV4604A) with a 1 ml/min resolution. Electrical power was supplied by a high precision digital programmable power supply (BK-PRECISION XLN10014). The voltage on the micro heater was measured by an Agilent digital multimeter (34972A). Two Omega K type thermocouples were used to measure the inlet and outlet fluid temperatures. Flow rates, pressures, inlet and outlet temperatures, and voltage and current were recorded by a customized data acquisition system developed from NI LabVIEW®. A visualization system comprised of a high-speed camera (Phantom V 7.3) with 256 × 256 pixels at approximate 40,000 frames per second and an Olympus microscope (BX-51) with 400× amplifications was used to study the bubble dynamics and two-phase flow structures.

2.4. Experimental procedures

The microheater served as a thermistor was calibrated in an isothermal oven prior to tests. The relationship between microheater temperature and its electric resistance was generated using a linear curve fitting. The confidence of the correlation coefficient was estimated to be higher than 0.9999. The heat loss as a function of temperature difference between micro device and ambient temperature was experimentally evaluated. Without fluid flows, the device was gradually heated up into steady state. The device temperature change is proportional to the heat input. The curve obtained by linear fitting was used to estimate heat loss with a high accuracy [4].

After assembling the microchannel device on the test package, the flow rate was kept constant at a set value ranging from 120 kg/m²s to 600 kg/m²s. Uniform heat fluxes at a constant step of around 3–20 W/cm² were applied by a digital power supply through the microheater until approaching CHF conditions. For each data point, the data acquisition system recorded 90 sets of steady state experimental data including voltages, currents, local pressures and temperatures at inlet and outlet at 4 min intervals.

3. Data reduction

3.1. Flow boiling heat transfer

In this study, the data of voltage, current and pressure drop are obtained to deduce wall temperatures and HTCs. The electrical input power (P) and resistance of the heater (R) is calculated as, respectively,

\[
P = V \times I
\]

(1)

and

\[
R = \frac{V}{I}
\]

(2)

Then heat loss (Ploss) between the environment and the testing chip is deducted from P to calculate the effective power:

\[
P_{eff} = P - P_{loss}
\]

(3)

The overall temperature of the thin film heater is calculated as,

\[
\theta_{heater} = K(R - R_0) + \theta_0
\]

(4)

where \(R_0\) is the resistance of the microheater at room temperature and \(K\) is the slope of the heater electrical resistance-temperature calibration curve. The surface temperature (\(\theta_{wall}\)) at the base area of the microchannels is then calculated using the overall surface temperature of the thin film heater as

\[
\theta_{wall} = \theta_{heater} - \frac{q''_{eff} t}{K_s}
\]

(5)

where \(q''_{eff} = P_{eff}/A_b\).

To accurately derive the two-phase heat transfer rate, the microchannels length (\(L\)) is divided into two sections: single-phase region (\(L_{sp}\)) and two-phase region (\(L_{tp}\)) [4]. The length of the two sections varies significantly with vapor quality as observed

<table>
<thead>
<tr>
<th>Geometry configuration</th>
<th>Nozzle number in each channel</th>
<th>Length of auxiliary channel</th>
<th>Alignment of nozzles</th>
<th>Number of main channel</th>
<th>Opening of nozzle</th>
<th>How nozzles connected</th>
</tr>
</thead>
<tbody>
<tr>
<td>The present configuration</td>
<td>4</td>
<td>8 mm</td>
<td>At a pitch of 2 mm</td>
<td>5</td>
<td>25 μm</td>
<td>4 nozzles share one auxiliary channel</td>
</tr>
<tr>
<td>Two-nozzle configuration</td>
<td>2</td>
<td>5 mm</td>
<td>At 5 mm</td>
<td>4</td>
<td>20 μm</td>
<td>1 nozzle in each auxiliary channel</td>
</tr>
</tbody>
</table>
by flow visualization. The two-phase HTC ($h_{tp}$) considering fin efficiency is used to evaluate flow boiling heat transfer performance in the two-phase region ($L_{tp}$) by excluding the weight of the single-phase heat transfer from the overall temperature. The single-phase effective HTC considering fin efficiency is calculated from,

$$h_{sp} = \frac{P_{eff}}{A_k - NA_f(1 - \eta_f)}\left[\frac{1}{T_{in} - T_{sat}} - \frac{1}{T_{exit} - T_{sat}}\right]$$ \hspace{1cm} (6)

where the fin efficiency ($\eta_f$) is estimated from

$$\eta_f = \frac{\eta_{Tsp}}{\eta_{Ttp}} = \frac{\tan^2(mH)}{mH}$$ \hspace{1cm} (7)

In the above equation, the inlet and the exit temperatures are estimated as,

$$T_{in,sp} = T_{in} + \frac{P_{eff}}{h_{sp}A_k}$$ \hspace{1cm} and \hspace{1cm} $$T_{exit,sp} = T_{sat} + \frac{P_{eff}}{h_{sp}A_k}$$ \hspace{1cm} (8)

The overall temperature of the two-phase heat transfer region ($T_{sp}$) is obtained by a weighted overall method [10] in terms of the single-phase and overall wall temperatures:

$$\bar{T}_{sp} = \frac{T_{wall}L_{sp} + T_{exit}L_{tp}}{L_{tp}}$$ \hspace{1cm} (9)

These length values ($L_{sp}$ and $L_{tp}$) are measured through visualization. Considering fin effects on a single microchannel, the effective two-phase HTC considering fin efficiency is calculated as,

$$\bar{h}_{tp} = \frac{P_{eff}}{\sum(WL + 2HL) \eta_f\left(T_{sp} - T_{sat}\right)}$$ \hspace{1cm} (10)

Because the thermal conductivity of Pyrex glass is approximately 1% of silicon, the interface between the microchannel walls and the cover glass is assumed to be thermally insulated in the fin approximation. Then, $h_{tp}$ was iteratively obtained from Eq. (1) through Eq. (10). Additionally, the exit vapor quality was calculated with mass flow rate and input power according to:

$$\chi = \frac{P_{eff} - Q_{sensible}}{mH_g}$$ \hspace{1cm} (11)

Then, the contribution of latent heat during phase-change heat transfer was derived as,

$$Q_{latent} = P_{eff} - Q_{sensible}$$ \hspace{1cm} (12)

where $P_{eff} = P - Q_{latent}$. $Q_{sensible}$ is the sensible heat resulting from the subcooled liquid. It was derived as,

$$Q_{sensible} = Gc\left(T_{sat} - T_i\right)$$ \hspace{1cm} (13)

In above equations, $T$ and $k_0$ are the substrate thickness, the thermal conductivity of silicon, respectively. Major physical properties of DI-water are given in atmosphere. $T_{sat}$ is a function of working pressure ($p$) in the middle of a microchannel, which is estimated as,

$$p = pi - \frac{\Delta p}{2}$$ \hspace{1cm} (14)

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

In addition, to evaluate the overall heat transfer performance of the device, the overall HTC based on the heating area is estimated as

$$\bar{h} = \frac{P_{eff}}{A_k(T_{wall} - T_{sat})}$$ \hspace{1cm} (15)

3.2. Uncertainty analysis

The uncertainty of mass flux, electrical voltage, electrical current, electrical power, ambient temperature and average wall temperature are ±1%, ±0.5%, ±0.5%, ±0.7%, ±0.5°C and ±0.86%, respectively. Uncertainty propagations are calculated using methods developed by Kline and McClintock [23]. For results of overall HTC, the uncertainty is ±6%.

4. Results and discussion

4.1. Flow boiling curves

4.1.1. Flow boiling curves based on heating area

Fig. 2 shows the overall HTCs based on the heating area as a function of heat flux at mass flux ranging from 120 kg/m²·s to 600 kg/m²·s. The overall HTCs increase with increasing mass flux. Fig. 2 indicates that the slope of each HTC curve initially declines sharply with the increase in effective heat flux and exit vapor quality, and then becomes flat. During the early boiling stage, nucleate boiling is dominant associated with a low vapor quality. The dispersed bubbles in the channel may promote the convection heat transfer by increasing the liquid superficial velocity. Hence, the overall HTC shows the highest value after onset of nucleate boiling (ONB). With further increase in heat flux and thus, the vapor quality, the role of nucleate boiling in determining heat transfer gradually decreases while convective boiling and thin film

![Fig. 2](image-url)
evaporation prevail. Fig. 2(b) shows two distinct bands of HTC curves as indicated by exit vapor quality among different flow rates. At low mass fluxes between 120 kg/m² s and 180 kg/m² s, the HTCs dramatically decrease among exit vapor quality from 0.25 to 0.6. At a higher mass flux ranging from 325 kg/m² s to 600 kg/m² s, the HTCs gradually decrease among exit vapor quality from 0.1 to 0.35. Previous studies demonstrate that the HTC descends with the increase in exit vapor quality due to the increasing heat flux [4,24,25].

A visualization study would help explain different trends of flow boiling curves between the low and high mass fluxes. Fig. 3 shows that TPOs present in upstream at a low mass flux of 150 kg/m² s. The confined bubble shrinks due to interfacial condensation. The effect of two-phase mixing induced by jetting flows is limited in enhancing HTC at low heat flux and mass flux as indicated in Fig. 3 and Fig. 4. In another word, nucleate boiling is dominant. As mass flux increases, for example, at 325 kg/m² s as illustrated in Fig. 5, strong mixing induced by jetting flow and TPOs are well established, indicating the dominance of convective boiling. Compared to upstream, however, Fig. 4 and Fig. 6 show that the mixing is not as obvious and frequent as observed in the downstream. Partial dryout or liquid detaching from walls starts to develop there. The force analysis of flow boiling in microchannels by Alam et al. [26] has demonstrated that evaporation momentum force increases with increasing heat flux and then dominates over shear force at high heat flux ranges, which makes it difficult to spread liquid on walls and hence, leading to liquid detaching as illustrated in Figs. 4 and 6. This could explain that a reduced HTC with the increasing heat flux is a result of a reduced effective heat transfer areas.

4.1.2. Flow boiling curves considering the fin effect

Compared to overall HTCs, effective HTCs considering the fin effect could provide more insights into the heat transfer mechanisms. Figs. 7 and 8 shows the effective HTC’s versus effective heat flux and vapor quality at low and high mass fluxes, respectively. Effective HTCs can better evaluate the heat transfer mechanisms since it considers all effective heat transfer areas. Fig. 7 shows that a peak value of effective HTCs exists at low mass fluxes of 120 kg/m² s, 150 kg/m² s and 180 kg/m² s. The effective HTCs initially increase with the increase in effective heat flux until the establishment of the fully developed boiling; and then gradually decrease with increasing heat flux. With an increase in effective heat flux, dryout was observed in the downstream as illustrated in Fig. 4 at a mass flux of 150 kg/m² s and a heat flux of 117 W/cm², resulting in descending effective HTC curves. In the effective HTC curves versus exit vapor quality, the effective HTCs reach their maximum values at exit vapor qualities between 0.3 and 0.45 when boiling is fully established in the entire channel, as shown in Fig. 7. For example, at the low mass flux of 150 kg/m² s, nucleate boiling prevails; while the confined bubble shrinks in upstream owing to interfacial condensation. Although TPOs were observed, the intensity in terms of frequency and magnitude is low. The effective HTCs sharply drop as the exit vapor quality higher than 0.4 because the increase of exit vapor quality leads to even weaker mixing (as indicated in Figs. 3 and 4) and hence, the local dryout starts to develop from outlet region. For example, Fig. 6 shows that mixing is not obvious and intensive in the downstream at a mass flux of 325 kg/m² s and a heat flux of 298 W/cm².
Fig. 8 demonstrates that the effective HTCs initially increase when nucleate boiling is dominant and then gradually decline after the full establishment of mixing and TPOs for higher mass fluxes. Specifically, Fig. 8 shows that the effective HTC keeps a relative high value as the boiling process at a mass flux of 600 kg/m² s associated with exit vapor quality lower than 0.4. The maximum value is ~115 kW/m² K during fully developed flow boiling owing to strong mixing.

4.2. Enhanced flow boiling

Fig. 9 shows that the overall HTCs based on the heating area are significantly enhanced compared to our previous two-nozzle configuration [4]. An enhancement up to 83.7% is achieved at a low mass flux of 150 kg/m² s (Fig. 9(a)). With an increase in mass fluxes, the enhancements of overall HTCs decrease to ~14.5% at a moderate mass flux of 325 kg/m² s (Fig. 9(b)) and ~30.5% at a moderate mass flux of 430 kg/m² s (Fig. 9(c)). The larger enhancement of overall HTC in low mass fluxes may be attributed to the enhanced nucleate boiling primarily owing to increased nucleation site density compared to that in the two-nozzle configuration [15].

4.2.1. Enhanced nucleate boiling

To fairly compare the boiling performance in these two microchannel configurations, Fig. 10 compares effective HTCs for mass flux ranging from 150 kg/m² s to 430 kg/m² s. Though enhancement varies with mass fluxes, the effective HTC has been substantially enhanced under all three compared mass fluxes when the superheat is less than ~10 K, at which nucleate boiling is believed to prevail. Specifically, at low mass fluxes, such as 150 kg/m² s, an enhancement of ~67% is achieved (Fig. 10(a)) and then the enhancement drops to ~31% at the peak value. As superheat increasing, the enhancement is further reduced. The trend of the two boiling curves is similar. The effective HTC firstly increases and then decreases with an increase in effective heat flux. The peak effective HTC is achieved at the onset of fully developed boiling (OFB). However, the enhancement decreases to ~54% as the mass flux increases to 325 kg/m² s in nucleate boiling region, as shown in Fig. 10(b). The trend of boiling curves between the present study and the two-nozzle configuration is quite different as mass flux reaches 325 kg/m² s or higher. In this study, the peak effective HTCs occurred after ONB and then effective HTCs gradually decline with an increase in heat flux for mass fluxes over 325 kg/m² s (as shown in Fig. 10(b, c)). In contrast, the peak effective HTC for the two-nozzle configuration happens when bubbles reach the inlet for all the mass fluxes [4]. The superheat is ~15 K (in Fig. 10(b)) at a mass flux of 325 kg/m² s when boiling is fully established. Our previous study [4] has demonstrated that the gradually decline of effective HTCs was linked to the formation of persistent vapor slugs in the section between the cross-junction and the inlet. Finally, the results in Fig. 10 show that the effective HTC is significantly enhanced compared to the two-nozzle configuration, including significant enhancement in the whole boiling process at a mass flux of 150 kg/m² s as well as in the nucleate.
boiling region at mass flux of 325 kg/m²·s and 430 kg/m²·s. At a higher mass flux of 430 kg/m²·s, the effective HTC values of the present study in the fully developed boiling region are lower than that of two-nozzle configuration, as indicated in Fig. 10(c).

As aforementioned, effective HTCs are significantly enhanced during nucleate boiling region compared to the two-nozzle configuration. Fig. 11 indicates that significant reduction of ONB is observed owing to the enhanced nucleation because of an increase of nucleation site density. The overall wall superheat at ONB is approximately 15 °C lower compared to that of the two-nozzle configuration. Fig. 11 also shows that the ONB gradually increases with the increase of mass flux, requiring higher average wall temperature to trigger nucleate boiling.

4.2.2. Enhanced mixing
Due to the small hydraulic diameter, laminar flow is dominant during flow boiling in conventional microchannels in most working conditions [3,5,15,16]. Improving mixing can be an effective way to enhance flow boiling heat transfer in microchannels. Mixing is highly efficient, but hard to be passively achieved during flow boiling in microchannels. Active methods such as impingement jets [27–29] can enhance flow boiling by promoting fluid mixing.
Our previous study has demonstrated that embedded staggered herringbone mixers can significantly improve flow boiling performance through enhancing mixing [30]. Recently, mixing was also achieved using the self-sustained high frequency TPOs induced by rapid thermal bubble growth-collapse [4]. However, the mixing is only limited to the central section (Fig. 12(c)) and there are no TPOs in the upstream (Fig. 13). In this study, the extended intense-mixing (as indicated in Fig. 12(a, b)) due to high frequency jetting flows could be another responsible factor for the enhanced effective HTCs of the present configuration, as shown in Figs. 7 and 8.

As depicted in Fig. 14, the frequency of TPOs is at a magnitude of 100 Hz in both configurations. Due to four nozzles shared one auxiliary channel, the average frequency of TPOs in the present study is about 2 times lower than that in the two-nozzle configuration at the same working conditions, in which one auxiliary channel is connected to the main channel by one nozzle. In the present study, the order of magnitude of TPOs is approximately 100 Hz and the highest frequency as measured is around 571 Hz as illustrated in Fig. 14.

Although the frequency of TPOs is much lower in this study compared to that of the two-nozzle configuration in similar working conditions (i.e., effective heat fluxes) as shown in Fig. 14, intense mixing induced by high frequency jetting flows is successfully extended to the entire channel by integrating evenly-distributed four nozzles in each main channel as indicated in Fig. 12(a and b). Fig. 15 shows the high frequency of jetting flows from one of the four micronozzles. Thereby, the heat transfer rate of the present study is comparable to that of two-nozzle configuration after OFB for mass flux of 150 kg/m²s and 325 kg/m²s; while the effective HTCs are lower at a mass flux of 430 kg/m²s in the region of OFB. In summary, our studies show that at low mass fluxes such as 150 kg/m²s, the primary enhancement mechanism is the augmented nucleate site density, particularly with the superheat less than 10 K. However, in high mass flux conditions, although the mixing has been successfully extended to the entire channel, the HTC enhancement is shown to be compromised due to the reduced overall two-phase oscillation frequency as well as the mild mixing (or even liquid detachment) as observed in the downstream in the present configuration.

![Fig. 11. Significant reduction of ONB in the present configuration compared to the two-nozzle one.](image)

![Fig. 12. (a, b) Extended mixing in the entire channel is achieved in the four-nozzle microchannel configuration at a mass flux of 600 kg/m²s and a heat flux of 439 W/cm². (c) Mixing was observed at the central section of channel in two-nozzle configuration microchannel at 430 kg/m²s and a heat flux of 106 W/cm² (Number 1–4 represents the four locations of micronozzles from inlet to outlet. Scale bars are 200 μm).](image)

![Fig. 13. Persistent vapor slugs near the inlet section of each main channel after OFB were observed at a mass flux of 430 kg/m²s and a heat flux of 399 W/cm² in the two-nozzle configuration microchannels.](image)

![Fig. 14. A comparison of two-phase oscillation (TPO) frequency between the two configurations. The frequency is calculated based on the time interval of the bubble-growth-collapse cycle.](image)
4.2.3. Enhanced flow boiling stability

The enhanced removal of confined bubbles of flow boiling in microchannels is critical in reducing two-phase flow instabilities in terms of both wall temperature and pressure drop fluctuations. Numerous techniques have been explored to remove or at least manage confined bubbles, for example, using taper geometry [21]. In our previous study, the two auxiliary microchannels has been designed to connect with a main microchannel through two-microwave nozzles. This configuration has been demonstrated in effectively stabilizing flow boiling in terms of both wall temperature and pressure drop fluctuations by inducing high frequency TPOs compared to microchannels with IRs [3]. The removal of confined bubbles can also lead to significantly enhanced heat transfer rate and CHF. In the present study, the extended range of intense mixing to the entire channel can further manage two-phase flow instabilities.

Specifically, the wall temperature fluctuation in the present design is substantially reduced compared to that of the two-nozzle configuration. Standard deviation of wall temperature between the present study and the two-nozzle microchannel configuration at a mass flux of 380 kg/m² s was compared in Fig. 16. As shown in Fig. 16, the difference of standard deviation of wall temperature varies with increasing effective heat flux. At low heat flux, the standard deviation of wall temperature of the two-nozzle configuration is ~150% larger than that of the present study since confined bubbles still persist in the downstream channel. In contrast, the increased number of micronozzles could generate jetting flow near outlet section to facilitate the removal of confined bubbles. Consequently, the wall temperature fluctuation is reduced owing to enhanced mixing range. As the increase in effective heat flux, the standard deviations of wall temperature are dramatically reduced on both configurations owing to intensified mixing (as shown in Fig. 12), which is induced by high frequency TPOs. The frequency of TPOs of the two-nozzle configuration is much higher, as shown in Fig. 14. But the increased number of jetting flow can still increase the mixing effect in two aspects. First, jetting flows from four evenly-distributed micronozzles can enhance bubble removal, leading to enhanced oscillations in main channels. Second, jetting flows themselves would induce mixing around the micro-nozzle area. As the further increase in effective heat flux, confined slugs have been better managed in the present configuration and hence, leading to smaller deviation in Fig. 16.

5. Conclusions

In our previous study, the two-nozzle microchannel configuration [4] has demonstrated the effectiveness in enhancing flow boiling owing to generation of intense mixing induced by high frequency TPOs and jetting flows. However, mixing cannot be generated in the entire channel length. In this study, we show that through an improved microchannel configuration, such an enhancer can be extended to the entire channel. In the part (1) of this study, heat transfer performance is further enhanced due to extended intense mixing induced by four evenly-distributed nozzles. Meanwhile, nucleation is substantially enhanced because of increased nucleation sites. The main conclusions are summarized below:

(a) Mixing is extended to the entire microchannel length.
(b) An enhancement of overall HTCs up to ~83.7% is achieved at a mass flux of 150 kg/m² s. The enhancement of overall HTCs decreases to ~14.5% at a moderate mass flux of 325 kg/m² s. To fairly compare the enhanced mechanism, effective HTCs considering fin efficiency are examined. As shown in Fig. 10, an enhancement of ~67% is achieved at a mass flux of 430 kg/m² s due to the enhanced nucleate boiling, which is consistent with the 14% lower ONB over the two-nozzle configuration.
(c) Effective HTCs curves are converging during fully developed boiling. Meanwhile, the effectiveness of TPOs decreases with the increase in exit vapor quality.
(d) Flow boiling stability in terms of wall temperature is noticeably enhanced in this present configuration.

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Conflict of interest

The authors declare that there are no conflicts of interest.

References