Flow visualization and heat transfer characteristics of gas–liquid two-phase flow in microtube under constant heat flux at wall

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ABSTRACT

Flow visualization of gas–liquid two-phase flows in circular glass tubes with inner diameter of 500 and 300 μm have been conducted. The 300 μm tube is further utilized for the investigation of the heat transfer characteristics under constant heat flux at wall. The constant heat flux is supplied by joule heating via applied electric current through a thin uniform indium tin oxide coating. To ensure steady flow patterns, the two-phase flow is achieved by injecting nitrogen gas coaxially through a centrally positioned tube to the continuous liquid phase flow. Measurements of wall temperature along the heating zone, various flow patterns and pressure drop are recorded simultaneously. Thermal performance is found to depend on bubble size, void fraction, Reynolds number and flow patterns. The two-phase Nusselt number (NuL) for bubbly flow is found to increase by 176% compared to the single phase flow, while the corresponding pressure drop increases by less than 27%.

1. Introduction

The increasing demand for portable electronic devices will inevitably lead to miniaturization. However, one of the drawbacks of miniaturization is that it will greatly increase the number of the transistors residing on a single chip. The temperature of the chip will increase significantly due to the accumulated heat flux. Hence, to ensure the normal operation of the electronic components, efficient cooling methods are necessary.

Due to the fact that heat transfer coefficient for fully developed laminar flow is inversely proportional to the hydraulic diameter of the channel, the reduction of the channel size will lead to higher heat transfer rate. Micro-channel, which has the potential to enhance the heat transfer performance, has been given a lot of attention in the past decades. The first single-phase heat transfer experiments in micro-channel heat sink was conducted by Tuckerman and Pease [1]. Their experimental results demonstrated that the single-phase micro-channel heat sink could effectively achieve a heat removal rate of 790 W/cm², but with the penalty of over 2 bars pressure increment for a 1 cm² chip. In order to achieve even higher heat removal rate, flow boiling has also been extensively investigated. Although the phase change of the fluid can provide higher heat transfer coefficient, our limited understanding of the heat transfer mechanism in small capillary channels and micro-channels have inhibited their widespread applications. Previous heat transfer work on flow boiling had shown tremendous increase in pressure drop [2] and the occurrence of flow reversal has posed further challenges for its applications [3].

In view of the great complexities and instability of the flow boiling, non-boiling two phase segmented flows are still being considered favorably. Prior to the heat transfer investigation for the two phase non-boiling flow, two phase flows under adiabatic conditions have been investigated by many researchers. In spite of extensive research, large deviation of the reported results still exists, especially for the observed flow patterns and heat transfer coefficients. Kawahara et al. [4] reported that the bubbles generated in the channel were released periodically while Chen et al. [5] observed bubbly train where the number of the bubbles in the flow through the channel appeared in random. This can be attributed to the different inlet configurations designed for the investigation. Hayashi et al. [6] had conducted the two phase flow visualization experiments so as to analyze the flow patterns. The dispersed phase was injected into the continuous phase via a center tube inserted into the test section. This inlet configuration showed that steady flow patterns could be maintained. However, the heat transfer characteristics of the two phase flow was not discussed in details.

Regarding the non-boiling two phase flow, Betz and Attinger [7] demonstrated that segmented flow could enhance heat transfer by 140% in a micro-channel heat sink comparing to the single phase flow case. Recently, Asthana et al. [8] employed micro-PIV and Laser Induced Fluorescence to determine the velocity field and
temperature of the segmented flow of two immiscible liquids in square channel with hydraulic diameter of 100 μm. Though flow circulation was observed and their results had shown that the Nusselt number had a four-fold increase over the single phase flow, no thermal performance relating to the flow patterns had been discussed.

Heat transfer of the two phase non-boiling flow is rarely investigated for channel with internal diameter less than 1 mm, especially for the steady flow patterns where the segmented media is having a constant size and spacing. Careful study of the liquid slug length effects to the heat transfer performance with controllable flow patterns had been conducted by Walsh et al. [9] using a 1.5 mm inner diameter stainless steel tube for heating and a transparent tube connected immediately behind the stainless steel tube for flow visualization. They reported that the shorter liquid slug would have better heat transfer enhancement than the longer liquid slug due to internal circulation. However, no pressure drop effect was studied.

The experiments performed by Muzychka et al. [10] showed that the increase in the local velocity at constant mass flow had no effect on enhancing the overall heat transfer rate due to the reduction in wetted surface area. However, Kreutzer et al. [11] reported that the increase in mass transfer in gas–liquid segmented flow could enhance the heat transfer rate while keeping the pressure drop at an acceptable level are therefore required. While the previous reported results only shows that the heat transfer performance can be increased with the two phase flow [8], the pressure drop and flow visualization (flow patterns and void fraction) have not been analyzed simultaneously.

The focus of the present study is to experimentally investigate the optimum conditions by which the thermal performance of a microtube can be enhanced but the pressure drop can be kept to a minimum. The mechanisms responsible for heat transfer enhancement for the gas–liquid segmented flow in microtube can be identified via temperature measurements and flow visualization simultaneously. In addition, the importance of the liquid slug length to the thermal performance will also be discussed.

2. Experimental setup

Fig. 1 shows the schematic of the experimental setup for the present investigation. The working fluid is DI (de-ionized) water. Nitrogen gas (N₂) is introduced into the channel for the two phase flow visualization experiments. Test channels are 300 mm long borosilicate circular glass tubes with outer/inner diameter of 1200/500 and 900/300 μm so as to provide visualization for the flow patterns when the nitrogen gas is injected into the DI water flow coaxially. Both 500 and 300 μm tubes are used to carry out the single phase pressure drop measurements and the two phase flow visualization under different water and nitrogen gas flow rates. Subsequently, the 300 μm tube is chosen to study the heat transfer.
characteristic under constant heat flux at wall. The tube length is 300 mm and only 80 mm has been used as the heating zone. The flow will firstly pass through a 200 mm unheated zone; to ensure that the flow is hydrodynamically fully developed before entering the heating zone.

Halogen lamp (cold light) is placed behind the test section for illumination. High speed camera (Phantom Miro 4) is employed and positioned at the entrance of the centre tube which is about 3 cm (100 diameter) away from the inlet to provide visualization of about 10 mm (33 diameter) of the channel length. Images are captured at a rate of 16,000 frames per second with the resolution of 512 x 64 pixels for each frame.

DI water is pumped into the test channel by the twin plunger pump (HPLC Pump PU714i) with the flow rate in the range of 0.01–20.0 ml/min. The flow rate of the HPLC pump is further confirmed by collecting the water in a beaker over a fixed period of time. The mass of the water is weighted using an electronic balance and the difference is found to be less than 1%. The inlet and outlet pressure of the test section are measured by the digital pressure gauges (DPG1000B) with an accuracy of 0.25%. Injection of nitrogen gas is controlled by low mass flow controller (Sierra Smart-Trak model 101).

Uniform ITO (Indium Tin Oxide) coating with thickness of 2 μm is applied at the outer wall of the glass tube to act as the heating source. It is a conductive transparent film which could generate heat when applying electricity. In order to prevent the degeneration of the ITO coating, 1.1 μm oxidation protection film (Parylene HT) is coated on top of the ITO coating. This is to make sure the resistivity of the ITO coating remains consistent throughout the experiments which would be conducted over a period of time. An AC power supply is used to provide joule heating on the ITO coating. The applied AC power is up to 8 Watts over the 80 mm long heating zone. Tube outer wall temperatures are measured by five thermocouples (k-type) and the data are digitized and recorded by a computer. The thermocouples are arranged with the following sequence where T1 is placed 10 mm in front of the heating zone and T2 is positioned 10 mm away from the starting point of the heating zone. The other thermocouples are having a distance of 20 mm between each other, as shown in Fig. 1. The experiments are performed by varying the gas flow rate while keeping the water flow rate constant. All the measurements are recorded when gas and water flow rates as well as the flow patterns become steady. i.e. when the bubble size and the distance between two adjacent bubbles would remain unchanged. This condition can be achieved after switching on the pump for about 10–15 mins. The experimental conditions are listed in Tables 1 and 2.

### 3. Data reduction

#### 3.1. Void fraction

Void fractions of the two phase flow patterns are determined by the images captured by the high speed camera. For the bubbly

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Experimental conditions for i.d. 500 μm.</th>
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<tr>
<td>jL [m/s]</td>
<td>jG [m/s]</td>
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<tr>
<td>0.08</td>
<td>0.04</td>
</tr>
<tr>
<td>0.17</td>
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</tr>
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<tr>
<th>Table 2</th>
<th>Experimental conditions for i.d. 300 μm.</th>
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<tr>
<td>jL [m/s]</td>
<td>jG [m/s]</td>
</tr>
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<td>0.12</td>
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<td>0.35</td>
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<tr>
<td>0.42</td>
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<td>0.47</td>
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<td>0.71</td>
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<tr>
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<td>0.71</td>
</tr>
</tbody>
</table>

Fig. 1. Experimental setup.
flow, the volume of the bubble can be approximated as a sphere (Fig. 2). For the slug flow, the elongated bubble is divided into a cylinder with two ellipsoidal endcaps (Fig. 3). The volumetric void fractions for both bubbly and slug flows are calculated as the ratio of the gas volume to the channel volume within a unit cell and expressed by the following equations. For bubbly flow pattern:

\[
\alpha = \frac{4}{3} \pi \left( \frac{d_b}{2} \right)^3 = \frac{4}{3} \pi d_b^3 \left( \frac{C_0}{M_{\text{center}}} \right)^3
\]

For slug flow pattern:

\[
\alpha = \frac{L_s A_c + \frac{4}{3} \pi \left( \frac{d_s}{2} \right)^3 L_c}{L_{\text{uc}} A_c} = \frac{L_s + \frac{4}{3} \pi \left( \frac{d_s}{2} \right)^2}{L_{\text{uc}}} \left( \frac{d_c}{C_0} \right)^2
\]

3.1.1. Optical Distortion

Optical distortion will magnify the inner diameter of the glass tube. The experiments of Kawahara et al. [4] of the two phase flow had shown that the inner diameter of the micro-channel could be enlarged by almost 50%.

The radial magnification factor is determined by inserting a known diameter center tube into the glass test tube shown in Fig. 4. It can be expressed by the following equation,

\[
M = \frac{D_i}{D_o}
\]

Therefore, in order to account for the magnification effect of the glass tube, Eqs. (1) and (2) will be modified into Eqs. (4) and (5), respectively.

\[
\alpha = \frac{\frac{4}{3} \pi \left( \frac{d_b}{2} \left( 1 - M_{\text{center}} \right) \right)^3}{L_{\text{uc}} \left( \frac{d_c}{C_0} \right)^3}
\]

\[
\alpha = \frac{L_s + \frac{4}{3} \pi \left( \frac{d_s}{2} \left( 1 - M_{\text{center}} \right) \right)^3}{L_{\text{uc}}} \left( \frac{d_c}{C_0} \right)^2
\]

3.2. Heat Transfer

The present experiments have been conducted so as to determine the value of the Nusselt number for the single and two phase flow heat transfer in the micro-channel. In order to determine the Nusselt number, temperature at the channel inlet, outlet and channel wall are measured. Channel wall outside temperature measurements are obtained from the thermocouples attached on the surface of the channel outer wall. The inner wall temperature is not possible to measure directly such that the inner wall temperature is solved by the one dimensional heat conduction equation in cylindrical coordinates:

\[
T_{w,i} = T_{w,o} - \frac{q_{00} R_i}{k_{\text{tube}}} \ln \frac{R_o}{R_i}
\]

Furthermore, the liquid bulk temperature along the heating zone can be estimated as:

\[
T_{\text{bulk},i} = T_{\text{bulk},\text{inlet}} - \frac{\left( q_{00} \right) \left( 2 \pi R_i x \right)}{mc_p}
\]

where \( q_{00} \) is the amount of energy absorbed by the water flow:

\[
q_{00} = \frac{m c_p \left( T_{\text{bulk},\text{inlet}} - T_{\text{bulk},\text{inlet}} \right)}{2 \pi R_i t_{\text{heat}}}
\]

The wall heat flux over the heating zone is constant and the local heat transfer coefficient can be obtained by:

\[
\frac{h_{\text{loc},x}}{k_{\text{liquid}}} = \frac{\frac{q_{00}}{2 \pi R_i}}{T_{w,i} - T_{\text{bulk},x}}
\]

The local Nusselt number is:

\[
Nu_{\text{loc},x} = h_{\text{loc},x} D_i = \frac{h_{\text{loc},x} 2 R_i}{k_{\text{liquid}}}
\]

Since the gas phase has a comparatively low heat capacity, the heat transfer will mainly occur in the liquid phase. Therefore, only the water properties are used for Eqs. (7) and (8).

3.3. Temperature Measurement

Empty tube (where there is no fluid flowing through the tube) is heated so as to check the uniformity of the heat flux generated by the coating. The temperature is measured along the heated zone while the heat flux is gradually increased. The results are shown...
in Fig. 5. The temperature measured by the thermocouple (T1) located at 10 mm before the heating zone shows no significant temperature increment like the other thermocouples located inside the heating zone (Fig. 5). Hence the axial conduction along the heating zone can be neglected in the present experiments.

In addition, axial conduction along the flow direction is checked with the axial conduction number proposed by Maranzana et al. [12]. The axial conduction number, \( M \), is defined as the ratio of axial heat conduction in the channel wall to the heat convection in the fluid flow. It can be expressed by the following equation.

\[
M = \frac{q_{\text{cond}}}{q_{\text{conv}}} = \frac{k_w A_w}{k_f A_f} \frac{1}{RePr} \tag{11}
\]

where \( k_w \) and \( k_f \) are the thermal conductivities of the wall and the fluid, respectively. \( A_w \) is the cross sectional area of the channel wall and \( A_f \) is the surface area wetted by fluid. Maranzana et al. [12] has proposed the criteria that in order for the axial conduction in the solid wall to be neglected, the value of \( M \) has to be less than 0.01. The calculated values of \( M \) in the present experiments are in the range of \( 1.51 \times 10^{-5} \) to \( 3.03 \times 10^{-5} \) for Reynolds number in the range of 71 to 354. Therefore the axial conduction can be neglected in the present experiments.

Fig. 5 also shows that the instantaneous measured temperatures can fluctuate within 3 °C about the average temperature. On account of the ITO coating is uniformly coated along the heating zone, this temperature deviation measured by each thermocouple may likely to be caused by the contact problem of the thermocouples with the tube surface or the heat loss to the surrounding environment. The measured wall temperatures are therefore time-averaged for the Nusselt number calculation. Typical temperature profiles after calibration are shown in Fig. 6.

4. Results and discussion

4.1. Single phase

4.1.1. Frictional pressure loss

Single phase pressure drop for incompressible flow through horizontal channels of constant cross sectional area is governed by Darcy Weisbach equation:

\[
f = 2 \frac{\Delta P}{d \rho U^2} \tag{12}
\]

The pressure loss across the test channel is measured between the inlet and outlet chamber of the test section. In order to account for the pressure loss at the inlet contraction region and hydrodynamic development region, measurements are performed on a shorter test channel. Subtracting the pressure loss across the shorter test channel from that across a longer test channel, the pressure difference is subsequently divided by the length difference of the two channels. This could provide the pressure loss per unit length for a particular flow rate. The pressure loss of laminar fully developed flow can be expressed by the following equation:

\[
\left( \frac{\Delta P}{T} \right)_{\text{fully developed}} = \frac{\Delta P_{\text{long}} - \Delta P_{\text{short}}}{L_{\text{long}} - L_{\text{short}}} \tag{13}
\]

Fig. 7 shows the pressure loss measurement results of the single phase fully developed flow. Pressure loss measurement results presented in Fig. 7 shows that the single phase pressure drop in channels with inner diameter of 300 and 500 μm can be well-predicted by Poiseuille flow theory. Comparing to the macrochannel, where the transitional flow usually occurs at Reynolds number at about 2300, earlier transition from laminar flow to transitional flow is noted and it starts at Reynolds number about 1600.

Fig. 5. Empty tube heating (i.d. 300 μm).

Fig. 6. Typical single phase heat transfer temperature profiles along the microtube (i.d. 300 μm) under the experimental condition of \( Re_L = 141 \) and \( q^* = 80.800 \) W/m²

Fig. 7. Single phase pressure drop for fully developed flow.
4.1.2. Heat transfer

Single phase heat transfer experiments are conducted. As shown in Fig. 6, the temperature increases linearly with the axial location, as may have been repeated. The Nusselt numbers for the single phase heat transfer in channel with i.d. 300 \( \mu \text{m} \) are presented in Fig. 8. The local Nusselt number is rather uniform along the axial location implying that thermal boundary layer has been fully developed at axial location \( x = 0.01 \text{ m} \). Fig. 8 also shows that single phase Nusselt number is independent/insensitive to the wall heat flux and the increase of the Reynolds number would cause the Nusselt number to increase accordingly. The Nusselt numbers are further compared with the results obtained by previous researchers [13,14] and are shown in Fig. 9. The experimental data show that as the Reynolds number increases from 71 to 354, the Nusselt number increases linearly from 1.35 to 8.83. Though the present experimental results show different gradient from the previous reported results, it has similar trend where the Nusselt number increases with the Reynolds number.

4.2. Two phase flow

4.2.1. Flow visualization

Fig. 10 shows typical flow visualization images taken in the present experiments for i.d. 500 \( \mu \text{m} \) channel. No coalescence of bubbles or breakdown is observed within the range of gas flow rate investigated. The distance between two bubbles remains constant throughout. No small bubble tailing the slug flow or in the thin liquid film are observed. The flow patterns are steady when the gas and liquid flow rates are fixed at particular values.

Stratified flow, which normally exists in the macro-scale pipe flow, is not observed in the present study. This observation shows that the surface tension effect dominates over the gravitational effect in the micro-channel when the inner diameter is less than 1 mm. Keeping the water flow rate constant while gradually increasing the gas flow rate, the first flow pattern that can be observed in Fig. 10(A, B) is the bubbly flow where the diameters of the bubbles are smaller than the channel diameter. These bubbles appear to be slightly elliptic with its longest radius in the vertical direction. This is due to the magnification effect in the radial direction by the curvature of the wall.

Second flow pattern which could be observed is the growth in the bubble size. Bubbles with diameter smaller than the channel diameter will slowly grow into the bubbles having the diameter larger than the channel diameter. Due to the physical restriction by the tube wall, the bubbles will eventually elongate to become slugs (Fig. 10(C–G)). Longer gas slugs will be formed when the gas flow rate is further increased (Fig. 10(H)).

Fig. 11 shows the flow visualization images for i.d. 300 \( \mu \text{m} \). Similar to the flow patterns observed in i.d. 500 \( \mu \text{m} \) tube, no stratified flow occurs and the flow patterns is gradually transforming from bubbly flow (Fig. 11(A–E)) into slug flow (Fig. 11(F–G)) with increasing gas flow rate.

4.2.2. Frictional pressure loss

Two phase pressure gradient can be obtained from the well-known single pressure gradient using the two phase multiplier proposed by Lockhart and Martinelli [15] for macro-sized channels.

\[
\frac{dp}{dz}_{TP} = \left( \frac{dp}{dz} \right)_L \times \Phi_L^2
\]

\((14)\)
The Martinelli parameter is defined as,
\[ X = \frac{\sqrt{dp/dz}}{C_{16}/C_{17}} G \] (15)

Chisholm [16] introduced a conventional expression to correlate two phase multiplier and Martinelli parameter. The equation is as follow,
\[ \Phi_0^2 = 1 + \frac{C}{X} \frac{1}{\alpha^2} \] (16)

Fig. 12 shows the comparison of the present experimental two phase frictional pressure drop results with the Lockhart-Martinelli correlation which is known to be applicable for the conventional channel. This figure shows that the pressure drop of the two phase flow for both with and without heat transfer is well within the \( C = 5 \) and \( C = 13 \). To better interpret Fig. 12, the experimental conditions are listed in Table 3. The flow pattern in region (Z) is the bubbly flow pattern, which is observed when the volumetric void fraction (\( \alpha \)) is less than 0.34. Increasing the gas flow rate implying an increase of the volumetric void fraction. Hence, when the volumetric void fraction is between 0.39 to 0.56, the corresponding Martinelli parameter is between 1.83 to 3 and the governing flow pattern becomes slug flow which falls in region (X) of Fig. 12. Region (Y) is the region where the bubbly flow transforms into slug flow. Hence the flow patterns observed in this region could be either bubbly or slug flow patterns. Further increase of the gas flow rate, churn and annular flow patterns which generally observed when the gas flow rate is comparably larger than water flow rate is expected to be observed. Nonetheless, on account of the limitation of the gas flow meter, experiment for \( j_G/j_L \) greater than 5.25 was not performed.

Fig. 12 also shows that the temperature has significantly reduced the pressure drop across the channel by 5–18% comparing to the adiabatic two phase pressure drop. This discrepancy may be attributed to the viscosity changes of the liquid phase [8]. For the bubbly flow, the frictional pressure loss is increased by around 27% comparing to the slug flow which has the pressure loss increase by 209% for the same water superficial velocity of 0.71 m/s. This may be due to the fact that when the water flow rate is fixed, the required gas flow rate to generate the slug flow pattern is larger than the bubbly flow pattern (also see Section 4.2.1). Thus, the larger pressure loss of the slug flow pattern over the bubbly flow pattern can be attributed to the comparative higher total flow rate.

4.2.3. Void fraction

The relation between the volumetric void fraction (\( \alpha \)) and the volumetric gas flow ratio (\( \beta \)) of the bubbly and slug flows are presented in Fig. 13. The void fraction results are compared with the homogeneous correlation (\( \alpha = \beta \)), Armand correlation (\( \alpha = 0.833/\beta \)) and the Kawahara correlation [4] (\( \alpha = C_{16}/(1+C_{17}) \)). Void fraction

Table 3

<table>
<thead>
<tr>
<th>Region</th>
<th>Flow patterns</th>
<th>( \alpha )</th>
<th>( \beta )</th>
</tr>
</thead>
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<tr>
<td>(X)</td>
<td>Slug</td>
<td>1.83 to 3</td>
<td>0.39 to 0.56</td>
</tr>
<tr>
<td>(Y)</td>
<td>Bubbly or slug</td>
<td>3 to 3.6</td>
<td>0.34 to 0.39</td>
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<tr>
<td>(Z)</td>
<td>Bubbly</td>
<td>&gt;3.6</td>
<td>&lt;0.34</td>
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</table>

Fig. 11. Two-phase flow visualization (i.d. 300 \( \mu \)m). Water velocity is 0.47 m/s, gas velocity is (A) 0.12 m/s, (B) 0.24 m/s, (C) 0.35 m/s, (D) 0.47 m/s, (E) 0.59 m/s, (F) 0.71 m/s, (G) 1.41 m/s.

Fig. 12. Comparison of two-phase friction loss with and without heat transfer in i.d. 300 \( \mu \)m with water flow rate of 2 ml/min (\( j_L = 0.47 \) m/s) and 3 ml/min (\( j_L = 0.71 \) m/s).
estimated by the image processing method gives the results which are close to the Kawahara correlation. Taking a closer look on the plotted results, it is noted that when the volumetric gas flow ratio becomes larger (when $\beta > 0.5$), the slope of the void fraction will become steeper. The void fraction will increase at a faster rate when the gas superficial velocity becomes larger. This is due to the fact that larger gas superficial velocity will change the flow pattern to a longer gas core where higher slip ratio may take place.  

Fig. 13 also shows that the void fraction results are clustering between slip ratio $S = 3$ and $S = 7$, where slip ratio is defined as the ratio of the velocity of the gas phase to the velocity of the liquid phase ($S = U_G/U_L$). Although the bubble velocity is slowing down, the gas phase is still traveling faster than the liquid phase. This indicates that the slip effect of two phase flow in the micro-channel is rather significant.

Figs. 14 and 15 show the relationship of the bubble velocity ($U_b$) and the mixture velocity ($j_G + j_L$). Initially, at a fixed liquid superficial velocity and low gas superficial velocity, the liquid phase will push the gas phase through the channel causing the bubble velocity to be much larger than the gas superficial velocity. Keeping the same water superficial velocity and increasing the gas superficial velocity, Figs. 14 and 15 show that the increase in bubble velocity is rather flat beyond $j = 0.4$ m/s, especially in the case of i.d. 300 µm channel. This may be explained by the reason that the increase of the gas flow rate only causes the bubbles to grow in size and shorten the distance between two adjacent bubbles due to the confined space in the small diameter channel (refer to Fig. 11). Furthermore, the relatively large bubble size with the channel size also makes the bubble to experience more resistance and difficult to rupture through the liquid slug.

4.2.4. Heat transfer

4.2.4.1. Pressure loss and wall temperature at different flow patterns. Fig. 16 shows the relationship of the wall temperature for the single phase and two phase flow at different flow patterns. Having the same liquid velocity, both figures show that the wall temperature has become lower when the second phase is introduced into the channel. Fig. 16(a) demonstrates that bubbly flow is able to lower down the wall temperature even though the single phase water flow is having the same superficial velocity as the two phase bubbly flow mixture velocity. Further increase of the gas superficial velocity will cause the bubble size to increase and to become the slug flow patterns (Fig. 16(b)). The wall temperature for the two phase slug flow is about 13.7% higher than that of the single phase flow which has the same velocity as the mixture velocity. Therefore, Fig. 16(a) and Fig. 16(b) suggest that the increase of gas flow rate that transforms bubbly flow pattern into slug flow pattern may have disadvantages over the single phase flow which has the same velocity as the mixture velocity.

The two phase flow can generally reduce the wall temperature but it may not necessary increase the heat transfer efficiency due to the corresponding increase of pressure drop (see Fig. 12). Bubbly flow is found to be more efficient in this case.

4.2.4.2. Local Nusselt number at different flow patterns. Two phase flow heat transfer experiments are conducted by varying gas flow rate into the continuous water flow. Figs. 17 and 18 show the temperature measurements along the channel where the water flow rates are fixed at 2 ml/min and 3 ml/min, respectively. The gas flow rates are then increased from 0.5 ml/min (section A), 3.0 ml/min (section B) until 6.0 ml/min (section C) and the gas injection stops at section D. The observed flow patterns are bubbly flow in section A and slug flow in section C. Section B in Figs. 17 and 18 are slug and bubbly flows, respectively. While section D is only water flow inside. The temperature for each flow rate is measured continuously for two hours. The increase of the gas flow rate causes the bubble size to grow inside the channel and transforms the bubbly flow pattern into slug flow. Although the distance between two adjacent bubbles are shortened and the liquid wetted surface area decreases correspondingly, the wall temperature do not have significant
difference among different flow patterns (Section A, B and C). It is obvious that at the same water flow rate, the single phase wall temperature (when there is only water phase flow inside (Section D)) is higher than the wall temperature of the two phase flow. Hence two phase flow can effectively reduce the wall temperature when the water flow rate is fixed.

In addition, the bulk water inlet and outlet temperature remain the same throughout different tested flow patterns. It shows that the increase of the gas flow rate does not have any changes to the bulk temperature of the mixture due to poor heat capacity of the gas phase. Hence it supports the fact that only the properties of the water phase should be used for the calculation at Section 3.2. Besides, though the bubble size is increased from section A to C and the length of the liquid slug between two bubbles has significantly change among these three cases, the wall temperature remains the same, compare both Figs. 17 and 18. This suggests that the bubble size and the length of the liquid column do not have significant effects on the wall temperatures. The present measurement contradicts the results reported previously by Walsh et al. [9] where they concluded that shorter liquid slug would have better heat transfer enhancement than the longer liquid slug.

One explanation for this reason may be that the heat transfer enhanced by the liquid circulation is no longer important due to the heat from the channel inner surface will easily reach the liquid core in such a small channel. The increase of liquid velocity due to the injection of gas phase may actually be the reason that reduces the wall temperature. However, the wall temperature cannot be decreased further with the increase of the gas superficial velocity. This may be attributed to the reason that the water phase has reached its limit on the amount of heat that can absorb. Hence, the experimental results also imply that there is a limit for decreasing the channel size for two phase flow without phase change.

The success of lowering down the wall temperature with the two phase flow actually implying that the Nusselt number for the two phase flow would become higher than the Nusselt number for the single phase flow. Since the gas flow rate does not have significant influence on the wall temperature, the comparison of the two phase Nusselt number \( \left( \text{Nu}_{G} \right) \) and the single phase Nusselt number \( \left( \text{Nu}_{L} \right) \) which has which has the same liquid superficial velocity are shown in Fig. 19. The two phase flow has about 100% increase comparing to the single phase flow for the water flow rate at 3 ml/min. However, the growth of the bubble size increases the pressure drop across the test channel due to the increase of flow resistance. Therefore the increase of the two phase pressure drop has to be accounted for the total heat transfer enhancement of the two phase flow over single phase flow. Fig. 20 shows the heat transfer enhancement \( \left( \text{Nu}_{G} \right) / \left( \text{Nu}_{L} \right) \) over the void fraction \( (\alpha) \). Two cases, shown previously in Figs. 17 and 18, where the water flow rates are fixed at 2 and 3 ml/min are presented. Both cases show that the heat transfer performance are
The present experiment reveals some insight into phase flow characteristic with the flow visualization. The gas–water two phase flow does enhance the heat transfer in micro-channel with i.d. 300 μm. The overall heat transfer performance of the bubbly flow exceeds slug flow is due to the pressure loss generated by the bubbly flow is generally smaller than the slug flow pattern. Higher Reynolds number which corresponds to higher liquid superficial velocity will result in better thermal performance. The optimal heat transfer condition for the two phase flow is to keep the bubble diameter close to the channel diameter and the void fraction to be around 10%. Under this condition, the thermal performance is enhanced to the maximum (NuG increases 176%) and the increase of the pressure loss can be kept to the lowest level (less than 27%).

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References