Heat transfer characteristics of gas-liquid two-phase flow in microtubes

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HEAT TRANSFER CHARACTERISTICS OF GAS-LIQUID
TWO-PHASE FLOW IN MICROTUBES

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SCHOOL OF MECHANICAL AND AEROSPACE ENGINEERING

2014

**HEAT TRANSFER CHARACTERISTICS OF GAS-LIQUID
TWO-PHASE FLOW IN MICROTUBES**

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Abstract

It is well-known that microelectronic devices would generate high heat fluxes. Given the same heat flux generated, smaller size means more concentration of heat in one particular area. Hence, efficient thermal management is required for these devices to ensure the effective operations of these devices. Heat dissipation via microchannels has been considered as an effective method for removal of high heat fluxes due to its high surface to volume ratio.

Despite the extensive research in fluid flow in microchannels over the past decade, the reported results among different researchers are still at variance. The present work is to elucidate some of the unresolved issues on one and two phase flow heat transfer characteristics in micro-sized tubes with the inner diameter less than 1 *mm*.

Experiments have been carried out to examine the single and two-phase flow characteristics in micro-sized tubes with inner diameters at 100, 300, 500 and 900 μm respectively. Generally, pressure drop measurements are in good agreement with Poiseuille flow theory. Transition flow is found to start slightly earlier at Reynolds number below 2000 as compared to the onset of transition flow at Reynolds number of 2300 in macroscale tubes. 300 and 500 μm tubes are employed for the flow visualization investigations of adiabatic two-phase flow characteristics. Two-phase flow patterns, two-phase frictional pressure losses and void fraction are discussed and the results are compared with the available literature. Two-phase pressure losses for the bubbly and slug flows are found to be underestimated by the Lockhart-Martinelli correlation. 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 applying 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 co-axially 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 ($Nu_{G,L}$) for bubbly flow is found to increase by 176% comparing to the single-phase flow, while the corresponding pressure drop increases by less than 27%.

To extend our understanding of the heat transfer characteristics in microtubes, numerical simulations for the heat transfer characteristics in axisymmetric air-water two-phase flow have been carried out. Comparison between the numerical and experimental flow patterns shows that the difference is within 10%. Subsequent simulation results show that the Nusselt number enhancement can be as high as 200% while the two-phase frictional pressure loss is about 20% increment for the bubbly flow comparing to that of liquid flow alone. The numerical results also show that the heat transfer performance varies with the bubble size, frictional pressure drop and Reynolds number. Analysis of the velocity and temperature profiles near a bubble shows that the bubble obstructs the path of the liquid flow, forcing the redistribution of the axial and radial velocities around the bubble. This redistribution enhances the thermal mixing and is found to be the main reason that enhances the heat transfer performance.

In addition, a new approach to enhance the heat transfer characteristics in microtube has been proposed. The main idea of this approach is to inject two immiscible fluids alternately into a microtube/microchannel, resulting in completely separated slugs for the two phases. Characteristics of both heat and mass transfer are investigated numerically. The overall Nusselt number could be increased by sixfold over the single-phase fully developed flow when the ratio of primary phase to the tube inner diameter (L_1/D) is kept at about 7. Characteristic of the average Nusselt number is found to be almost independent to the properties of the secondary phase. The friction factor is much dependent on the viscosity of the secondary phase. Secondary phase with higher viscosity would induce higher pressure loss and large pressure fluctuation is observed at regions close to the interfaces. The flow field that is far from the interfaces could be approximated as the single-phase fully developed flow. Weber number of the primary phase and

Capillary number of the secondary phase should be kept below certain critical values in order to maintain the discrete flow patterns. Frictional loss is a more important consideration than the heat capacity when choosing a particular secondary phase. Secondary phase with lower viscosity is preferred due to the overall heat transfer enhancement will become less beneficial if the pressure loss induced by the secondary phase becomes significant. Liquid-gas discrete flow is found to have better heat transfer performance than the liquid-liquid discrete flow.

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Nomenclature

A	cross sectional area	$[m^2]$
Ca	Capillary number	$[-]$
C	Chisholm parameter	$[-]$
C_p	heat capacity	$[J/kg.K]$
d	diameter	$[m]$
d_{actual}	is the actual diameter of the object	$[m]$
d_{image}	is the average diameter measured from the images	$[m]$
D_b	equivalent bubble diameter	$[-]$
f	friction factor	$[-]$
$\vec{g}$	gravitational force	$[\text{vector}]$
$\bar{h}$	average heat transfer coefficient	$[W/m^2.K]$
j	superficial velocity	$[m/s]$
k	thermal conductivity	$[W/m.K]$
k_{liquid}	thermal conductivity of the liquid phase	$[W/m.K]$
k_{tube}	thermal conductivity of the solid wall	$[W/m.K]$
L	length	$[m]$
L_1	length of the primary phase	$[m]$
L_2	length of the secondary phase	$[m]$
$L_{b,c}$	distance between the center of two bubbles	$[m]$
M	magnification factor	$[-]$
$\dot{m}$	mass flow rate	$[kg/s]$
$\vec{n}$	unit normal vector	$[\text{vector}]$
q''	heat flux	$[W/m^2]$
q_f	volume fraction of fluid	$[-]$

r	radius	[m]
r	radius for non-zero wall thickness model	[m]
R	radius for zero wall thickness model	[m]
t	time	[s]
T	temperature	[K]
T_b	bulk fluid temperature	[K]
T'_b	bulk fluid temperature	[K]
T_i	fluid inlet temperature	[K]
T_w	local wall temperature	[K]
$\overline{T_w}$	average wall temperature	[K]
$\overline{T_b}$	average bulk fluid temperature	[K]
u	velocity	[m/s]
$\vec{u}$	velocity vector	[m/s]
U	dimensionless velocity	
U_x	dimensionless axial velocity	
U_y	dimensionless radial velocity	
v	volume	[m ³]
w_i	thickness of the inner wall	[m]
w_o	thickness of the outer wall	[m]
We	Weber number	
x	axial location	[m]
y	radial location	[m]
X	dimensionless axial location	
Y	dimensionless radial location	
$\overline{Nu}$	average Nusselt number	
Nu_X	local Nusselt number along the tube axis	
Nu'_X	local Nusselt number along the tube axis	
Nu_{TP}	two-phase Nusselt number	
Nu_{SP}	single-phase Nusselt number	
∇	gradient operator	
$\nabla \cdot$	divergence operator	

Greek symbols

α	void fraction	
δ	Delta function	
ΔP_L	pressure drop for flow of only liquid phase	$[Pa]$
ΔP_G	pressure drop for flow of only gaseous phase	$[Pa]$
ΔP_{SP}	pressure drop for single-phase flow	$[Pa]$
ΔP_{TP}	pressure drop for two-phase flow	$[Pa]$
$\bar{\tau}$	average axial wall shear stress	$[N/m^2]$
κ	local curvature	
μ	dynamic viscosity	$[kg/ms]$
ν	kinematic viscosity	$[m^2/s]$
ρ	density	$[kg/m^3]$
σ	interfacial tension coefficient	$[N/m]$
χ	Martinelli parameter	
Φ	two-phase friction multiplier	

Subscripts

1	primary phase
2	secondary phase
<i>b</i>	bubble
<i>c</i>	cross section
<i>center</i>	center
<i>channel wall</i>	channel wall
<i>cond</i>	conduction
<i>conv</i>	convection
<i>cr</i>	critical
<i>f</i>	fluid
<i>G</i>	gas phase

G, L	liquid and gas phase
$heat$	heating zone
i	inner
loc	local
L	liquid phase
o	outer
s	solid
s	slug
sc	slug cap
SP	single phase
TP	two phase
uc	unit cell
w, i	wall inner
w, o	wall outer

Abbreviation

$i.d.$	inner diameter
$o.d.$	outer diameter

Chapter 1

Introduction

1.1 Background

With the advancement of manufacturing techniques, the size of the microelectronic components has been scaling down over the years. However, one of the main obstacles of miniaturizing the microelectronic components is the high heat generation in compact chips. Hence, in order to ensure the effective operation for these microelectronic devices, microchannels which can provide better cooling effectiveness (i.e., by pumping the liquid or gas liquid co-flow inside the channels to dissipate the generated heat) have been given a lot attention over the past decade.

The experiments conducted in the channels with inner diameter of a few millimeters to channels with inner diameter less than 1 millimeter by many researchers have found that the flow characteristic of fluid flow inside the small channels deviated from the conventional internal flow characteristic. For the general flow inside a conventional size tube, the critical Reynolds number for the laminar flow develops into turbulent flow is at about 2300. However, the critical Reynolds number was reported to have deviated from this number as the channel diameter decreased [1–4]. Additionally, some experiments have also reported lower Nusselt numbers than the conventional Nusselt number. The discrepancy between experiment and

theory has been discussed by Morini et al. [5]. Experimental errors have been identified to be the main factor that causes the deviation.

On the cooling applications, single-phase heat transfer has been widely utilized in micro heat exchanger. This is because reducing the size of the channel could increase the surface to volume ratio and hence leads to high heat transfer rate. Tuckerman and Pease [6] were among the first to conduct experiments on the microchannel heat sink. They reported that the heat flux removed from the heat sink could be as high as 790 W/cm^2 accompanying by pressure loss of $2 \times 10^7 \text{ Pa/m}$. Since then, heat transfer enhancement using microchannel has been widely acknowledged and being utilized on many cooling applications [7, 8]. Various technologies for the fabrication of microchannel heat exchangers have also been explored [9].

Despite the fact that single phase liquid convective flow has successfully achieved high magnitude of heat transfer rate, in order to achieve even higher heat transfer performance, extensive research has also been carried out on multiphase flow such as the flow boiling and condensation in microchannels. Hetsroni et al. [10] found that under certain flow and heat transfer conditions the explosive boiling, where bubble grew rapidly, would occur in the micro-channels and the alternative occurrence of single phase liquid flow and two phase liquid vapor flow in the channels would cause the pressure drop to fluctuate. This pressure fluctuation could potentially damage the device and cause unstable working condition. Zhang et al. [11] experimentally examined the phase change phenomena in silicon channels with internal diameter ranging from 27 to 171 μm . They concluded that the flow boiling mechanism was largely dependent on the channel size. Nucleate boiling was expected to appear in channels larger than 100 μm and explosive boiling would appear in channels smaller than 50 μm . The pressure pulses occurred at the onset of boiling was expected to affect the local boiling and equilibrium conditions. Schilder et al. [12] performed visualization experiment of single phase and two phase flow boiling on inner diameter 0.6 mm tubes. Their experimental results showed that local Nusselt numbers for flow boiling could be increased by 3 to 8 times higher than the single phase flow in a micro-channel.

Although two-phase flow such as flow boiling was found to have higher transfer rate than the single-phase flow, the not well understood mechanism of the two-phase flow in small capillary channels and microchannels have limited the widespread applications of microchannels. Therefore, in view of the great complexities and instability of the flow boiling [13], non-boiling two phase segmented flows are still being considered favorably.

Fluid flow in microscale is usually laminar. One way to enhance the heat transfer rate of micro-channel is to improve the heat exchange within the fluid by increasing the mixing rate of the hot and cool liquid phases. There are several ways to enhance the mixing. Some researchers focused on the modification of the micro-channel geometry. Lee and Teo [14] experimentally investigated the micro-channel with slanted grooves. Their results showed that the heat transfer performance could be enhanced by 12% without incurring additional pressure loss. Placing obstacle, such as triangular prism [15], along the flow path so as to create some turbulence to the flow could also improve the heat transfer rate. Enhancement through surface modification could increase the thermal performance, but higher flow rate is usually required for significant improvement to take place. The increase of the required pumping power has to be evaluated also.

Nanofluids has also become a popular way to enhance the heat transfer in micro-channels. Experimental data with two thermal conductivity models for nanofluids compared by Li and Kleinstreuer [16] showed that nanofluids could enhance the heat transfer performance of the heat sink with small increase in pressure loss. Analysis performed by Tsai and Chein [17] showed that there were both benefits and drawbacks for the nanofluids in micro-channels. Although the heat transfer rate could be enhanced, the increase in viscosity and decrease in heat capacity of the fluid cause the nanofluids used in micro-channel became less beneficial. They concluded that optimization for the heat sink was necessary so as to achieve better heat removal performance.

The above mentioned methods demonstrate the improvement in the heat transfer performance in micro-scale, however, modification of the inner wall surface might

not be an easy task. For instance, the manufacturing of the micro-channel with complex geometry will be costly and time consuming for mass production. Hence, the need for a simple solution to improve the heat transfer performance in micro-channels is still necessary.

In order to have a better understanding in the heat transfer mechanism for the two phase flow in micro-channel. Micro particle image velocimetry (μ PIV) method which can be used to measure the velocity field of the fluid flow in micro-channel is introduced [18]. Laser induced fluorescence (LIF) method which is capable of measuring the temperature field of the fluid flow is also adopted for micro-channel experiments [19]. However, to complete the task with the μ PIV and LIF methods, the refractive index of the fluid interface has to be carefully matched so as to obtain the distortion free results.

Decreasing the inner diameter of the channel will increase the difficulty of the fluid to flow through and hence implying the increase in frictional loss in the microchannels. Hence, though microchannels can effectively enhance the heat transfer performance, an essential parameter which must also be taken into account is the undesired pressure drop. This is because the increment of heat transfer rates could be easily offset by the pumping power required to overcome the pressure loss.

In recent years, researchers have been dedicated to study the heat transfer of microchannel with computational fluid dynamics (CFD). The powerful visualization capabilities enable the evaluation of the flow and heat transfer around the gas bubble. Gupta et al. [20] and Talimi et al. [21] had performed detailed analysis on the heat transfer increase around the gas slug region. In addition, Horvath et al had conducted the numerical investigation for the two-phase flow at different Reynolds numbers. Their results were limited to $\alpha_G = \alpha_L = 0.5$, where α_G and α_L are the void fraction of the gas and liquid phase, respectively. However, the numerical simulation results were not compared with the experimental results especially on the effect of flow patterns to the heat transfer characteristic.

In short, not only the reported deviated results that could be affected by the experimental skills and the equipments used, the experimental data especially

the two-phase flow characteristic with visualization in the microchannels with the inner diameters less than 1 *mm* are still found to be scarce. For that reason, tubes having the inner diameter within the range of 100 to 900 μm with length of 300 *mm* will be employed in the present study. The frictional loss, critical Reynolds number of single-phase flow as well as the two-phase frictional pressure loss and void fraction of two-phase flow with the aid of visualization has been investigated. Heat transfer mechanism of two-phase flow has been examined experimentally and numerically.

1.2 Objectives and scopes

In order to investigate the flow characteristics in microchannels, experimental and numerical work have been conducted.

The purposes of this study are:

Single-phase flow

- (i) Clarify the deviation of results obtained by various researchers on frictional loss and onset of transitional flow in small diameter channels.
- (ii) Experimentally examine the heat transfer characteristic in micro-sized tubes.

Two-phase flow

- (i) Investigate the behavior of two-phase flow patterns in microtubes with flow visualization.
- (ii) Utilizing flow visualization to study the optimal flow configuration that results in better heat transfer enhancement.
- (iii) Investigate the important parameters (e.g. bubble size and distance between two bubbles) that affect the heat transfer performance.

(iv) Improve heat transfer in micro-sized tubes with multiphase flow.

The scopes of the present study are:

Single-phase flow

Microtubes with the inner diameter of 100, 300, 500 and 900 μm are employed to conduct the experiments.

To achieve objective (i), pressure drop measurement will be conducted at different length of microtubes. The second experimental setup for single-phase flow which can provide high water flow rate is employed for the investigations to observe the onset of transitional flow. Furthermore, the experimental pressure loss results will be compared with theoretical Poiseuille flow equation and the results available in literature.

Objective (ii) can be achieved by coating a transparent conducting ITO (Indium Tin Oxide) film on the outer wall of the glass tube. ITO coating will act as a uniform heating source to the microchannel. Temperature on the outer wall surface will be measured by thermocouples and Nusselt numbers will be calculated.

Two-phase flow

Glass tube with inner diameter 300 and 500 μm will be used to conduct the two-phase flow experiments.

To achieve objective (i) a high speed camera (up to 16000 frame rate) is employed in the experiments to capture the images of gas-liquid two-phase flow patterns.

Objective (ii) can be achieved by coating a transparent conducting ITO (Indium Tin Oxide) film on the outer wall of the glass tube. ITO coating will act as a uniform heating source to the microchannel. Temperature on the outer wall surface will be measured by thermocouples and Nusselt numbers will be calculated.

The objectives (iii) and (iv) can be investigated by adopting the numerical methods to systematically analyze the flow field of the two-phase flows. Discrete flow which is potentially having higher heat transfer performance is proposed.

1.3 Overview of the thesis

The manuscript is organized as follows:

Chapter 1 introduces the background, objectives and scopes of the present study.

Chapter 2 provides a comprehensive review by previous researchers on single and two-phase flow in small diameter channels. It is presented by the adiabatic flows followed by the flows with heat transfer.

Chapter 3 presents the definition of main variables and basic equations used in two-phase flow and two-phase pressure drop analysis. The terminology of flow patterns used in present study is also provided.

Chapter 4 describes the experimental setup for single and two-phase flow with and without heat transfer. Some of the preliminary experimental results for the validation of experimental setup are presented.

Chapter 5 presents the flow visualization and heat transfer measurement results for bubbly and slug flow patterns. The bubbly flow patterns is found to have better overall heat transfer performance than the slug flow patterns due to its relatively lower pressure loss.

Chapter 6 provides a more in depth understanding of the liquid gas bubbly flow patterns via numerical simulations. The flow field around the gas bubbles are analyzed. Optimal bubble size and length between two bubbles are suggested.

Chapter 7 discusses the completely separated discrete flow patterns numerically. Its overall heat transfer performance is found to be better than the bubbly flow patterns. Though liquid-liquid discrete flow has higher heat capacity than the liquid-gas discrete flow, the liquid-gas discrete flow could outperform the liquid-liquid discrete flow due to its relatively lower pressure loss. Optimal primary phase length for the discrete flow pattern is proposed.

Chapter 8 concludes the current work. Future work is proposed.

Chapter 2

Review of Previous Work

This chapter gives an overview of the history of the previous work done by different researchers. Literature review of single-phase liquid flow will be presented first and followed by the two-phase flow.

2.1 Previous work on single-phase liquid flow

2.1.1 Deviation of transition regime

Pehlivan et al. [22] conducted the single-phase pressure drop experiments for water and air. They confirmed that experimental results for both working fluids agreed with the Poiseuille conventional theory for laminar flow and Blasius theory for turbulent flow in the circular section with inner diameter of 3, 1 and 0.8 *mm*, respectively. No early transition phenomenon was observed in the investigations of Lee and Lee [23] on rectangular channels of height ranging from 0.4 to 4 *mm* and width of 20 *mm*.

Along with the reduction of the tube diameter, the critical Reynolds number tends to deviate from the conventional internal flow critical Reynolds number ($Re = 2300$). According to Mala and Li [1], as the Reynolds number increases, the measured pressure gradient is significantly higher than the theoretical predicted

gradient. Early transition from laminar to turbulent flow was found to occur at $Re > 300 - 900$. Conversely, Hwang and Kim [2] reported that the onset of flow transition occurred only at Reynolds number of slightly less than 2000. However, Peng and Peterson [24] claimed that the transition of laminar flow to turbulent flow had occurred at Reynolds number greater than the convention critical Reynolds number of 2300. Addressing to the debate of early transition, Vijayalakshmi et al. [25] suggested that the deviation of Poiseuille number from a constant value should be an indicator of the onset of compressible flow when the exit Mach number exceeded 0.3 at $Re = 850$ (Fig. 2.1). This is because the corresponding slope of the frictional curve shown in Fig. 2.2 did not show any changes at $Re = 850$. Hence, they confirmed that the earlier deviation of Poiseuille number was caused by the compressible effect but not the occurrence of turbulent flow.

FIG. 2.1: fRe vs Re for $D_h = 124 \mu m$ (reprinted from Vijayalakshmi et al. [25]).

Other than the divergence of the critical Reynolds number, Steinke and Kandlikar [26] had performed an analysis on the entrance and exit losses. They concluded that researchers who reported the discrepancies of pressure loss from the classical theory were due to the entrance and exit losses or developing length were not properly accounted for. On the contrary, researchers corrected the entrance and exit losses have reported good agreement with the classical theory.

FIG. 2.2: f vs Re for $D_h = 124 \mu m$ (reprinted from Vijayalakshmi et al. [25]).

Recent experimental designs that have carefully taken the channel inlet and exit losses effects into consideration have shown good agreement with the theoretical macroscale channels data. Dutkowski [27] had performed experiments on the test section made from stainless steel pipes with internal diameters of 1.10, 0.64 and 0.55 mm . The pressure drop was measured at the fully developed flow region. Hence, the inlet pressure losses were accounted for. He concluded that the friction factor of laminar flow could be correctly predicted by convection correlation $f = 64/Re$. Furthermore, studies on pressure drop conducted by Hetsroni et al. [3] in circular, rectangle, triangular and trapezoidal channels with hydraulic diameter ranging between 1.01 to 4010 μm had drawn the conclusion that the flow behavior in microchannels had no differences from the macroscale flow.

To investigate the discrepancies of previous experimental data on the onset of flow transition zone, Kohl et al. [28] examined the air and water flow in rectangular channels with hydraulic diameters ranging between 25 to 100 μm . Their results agreed with the correlations for macroscale channels and suggested that the inconsistencies in the previous work were probably due to the instrumentation errors. More recently, pressure losses of low Reynolds number flow measured by Petropoulos et al. [29] in rectangular channel with hydraulic diameter of 146 μm

had shown good agreement with the conventional frictional pipe flow theory as well.

2.1.2 Surface roughness

Microchannels made from different materials may have influence on the surface roughness. Channel surface roughness could be another issue that caused the frictional pressure losses to deviate from the classical theory. Friction factors measured by Peiyi and Little [30] were generally greater than the predicted values for smooth surface in conventional size channels. Mala and Li [1] have performed the single-phase water flow experiments with channels of inner diameter ranging from 50 to 254 μm and concluded that the experimental results were in good agreement with the theoretical value when the Reynolds number was less than 500. The deviation increased when the Reynolds number of the flow increased. And as the channel diameter decreased, the deviation would increase correspondingly. Early transition from laminar to turbulent was found. Furthermore, Li et al. [4] studied the frictional resistivity in the stainless steel channels where the relative roughness (ε/D) were 3 to 4%. The measured Poiseuille numbers were about 15 to 37% higher than 64.

However, Hetsroni et al. [3] confirmed that the transition from laminar to turbulent flow occurred between Reynolds numbers of 1800 to 2200 for channel with relative roughness of 0.32 to 7%. Similarly, the study conducted by Celata et al. [31] reported that small inner diameter was the main reason that caused the frictional pressure losses to increase. Effect of surface roughness could be neglected as long as the relative roughness was less than 1%. In addition, no significant surface roughness effect was found by Faghri and Turner [32] in the microchannels made of silicon wafer.

A summary of the above mentioned literature is shown in Table. 2.1. The existing contradictory results of the previous research suggest that there are still rooms for future investigations.

	Hydraulic diameter	Poiseuille flow theory	Onset of transition	Relative surface roughness
Pehlivan et al. [22]	3, 1 and 0.8 mm	Agree	Not discussed	Not discussed
Lee and Lee [23]	0.4 to 4 mm and width of 20 mm	Agree	No early transition	Not discussed
Mala and Li [1]	50 to 254 μm	Higher	$Re \approx 300-900$	0.68 to 3.5 %
Hwang and Kim [2]	244, 430 and 792 μm	Agree	$Re \approx 1700-1900$	Not discussed
Peng and Peterson [24]	133 to 367 μm	Higher	$Re > 2300$	Not discussed
Judy et al. [33]	15 to 150 μm	Agree	Not discussed	Not discussed
Dutkowski [27]	0.55, 0.64 and 110 mm	Agree	Not discussed	Not discussed
Hetsroni et al. [3]	1.01 to 4010 μm	Agree	$Re \approx 1800-2200$	0.32 to 7 %
Koh et al. [28]	25 to 100 μm	Agree	No transition up to $Re=2067$	Not discussed
Petropoulos et al. [29]	146 μm	Agree	Not discussed	Not discussed
Li et al. [4]	79.9 to 205.3 μm	Higher	$Re \approx 1700-2000$	3 to 4 %
Celeta et al. [31]	30 to 500 μm	Agree	$Re \approx 2160-4430$	< 1 %

TABLE 2.1: Summary of literature on single-phase flow in small diameter channels.

2.2 Previous work on two-phase gas liquid flow

2.2.1 Two-phase frictional pressure drop

Two-phase pressure drop measurement performed by Hwang and Kim [2] had shown that the previous correlations used to predict their experimental data failed due to the dominance of surface tension effect.

Revellin and Thome [34] measured the two-phase pressure drop over the two-phase Reynolds number from laminar region ($Re_{TP} < 2000$) up to turbulent region ($Re_{TP} < 8000$). They concluded that the gradient of the two-phase frictional pressure drop is much steeper than the gradient of single-phase pressure drop and the transition regime was not well-predicted by any of the theoretical methods. New correlations for homogeneous two-phase frictional pressure drop were proposed as $f_{TP} = 0.08Re_{TP}^{-1/5}$ for $i.d. = 0.509\text{ mm}$ and $f_{TP} = 6Re_{TP}^{-3/5}$ for $i.d. = 0.790\text{ mm}$.

Sun and Mishima [35] evaluated various frictional pressure drop correlations proposed by various researchers. They concluded that Lockhart-Martinelli Method [36], Mishima and Hibiki correlation [37], Zhang and Mishima correlation and Lee and Mudawar [38] correlation were in fact very close to each other and that they could predict the pressure drop well in the laminar region to within $\pm 30\%$ for the mean relative errors.

2.2.2 Effects of channel diameters and surface wettability

Fukano and Kariyasaki [39] conducted the experiment of gas liquid flow in the capillary tubes with the inner diameter of 100, 240, 490 and 900 μm , respectively. According to the observed results, they concluded that the critical tube size for the surface tension to dominate over the gravitational force was between 500 to 900 μm . Hence, the stratified flow pattern would not occur when the channel inner diameter was less than 900 μm .

Triplett et al. [40] has studied the gas liquid flow in the circular channels with 1.45 and 1.1 *mm* inner diameters. The flow patterns observed in the experiments were bubbly, churn, slug, slug annular and annular flow. Furthermore, the observed flow patterns in the experiment performed on circular channels with 1.5 and 1 *mm* inner diameter by Chen et al. [41] were bubbly flow, slug flow, bubble-train slug flow, churn flow and annular flow. No stratified flow was observed in both experiments. In addition, Suo and Griffith [42] and Damianides et al. [43] confirmed that stratified flow pattern did not occur in microchannels less than 1 *mm*.

Besides, Kawahara et al. [44] confirmed that bubbly flow pattern with bubbles much smaller than the channel diameter of 100 μm was not observed. Furthermore, the liquid ring flow observed by Serizawa et al. [45] in channels with i.d. 25 μm was a flow pattern which cannot be discerned in macroscale channels.

Addressing to the surface wettability, Barajas and Panton [46] examined the effect of the contact angle to the two-phase flow patterns in an inner diameter 1.6 *mm* tube by using four different tube materials. The materials and contact angles were Pyrex ($\theta = 34^\circ$), polyethylene ($\theta = 61^\circ$), polyurethane ($\theta = 74^\circ$) and FEP ($\theta = 109^\circ$). The first three were partially wetting systems and the last one was a non-wetting system. As shown in Fig. 2.3, the contact angle had little effects on the transition lines when it was less than 74° and it showed that the contact angles moved the transition boundaries between bubble and plug flow, bubble and slug flow, annular and dispersed flow, slug and multiple rivulet flow as well as the slug and annular flow. In contrast, contact angles had little effect on the boundaries of plug and slug flow together with the bubble and dispersed flow. The observed flow patterns are shown in Fig. 2.4.

Furthermore, Iguchi and Terauchi [47] experimentally studied the wettability of two-phase flow in vertical channels with inner diameter of 5 to 15 *mm*. They concluded that the boundary between the bubbly flow and slug flow regimes was hardly dependent on the wettability of the pipe. However, they proposed that the wettability effect of the pipe would disappear at Weber number exceeding 100.

FIG. 2.3: Comparison of contact angle effects (reprinted from Barajas and Panton [46]).

FIG. 2.4: Definition of flow patterns by Barajas and Panton [46].

2.2.3 Terminology of flow patterns

Different researchers may have different definitions for the flow patterns they observed. Saisorn and Wongwises [48] studied the flow patterns in circular channel with inner diameter of 0.53 mm experimentally and compared the flow pattern definitions with other researchers. Figs. 2.5 and 2.6 are the flow region maps constructed by Saisorn and Wongwises [48]. The slug annular flow region categorized by Triplett et al. [49] and Chung and Kawaji [50] in the flow pattern maps were identical to the region defined as the throat annular flow by Saisorn and Wongwises [48].

FIG. 2.5: Comparison of the observed flow patterns with the transition lines by Triplett et al. [40] (reprinted from Saisorn and Wongwises [48]).

In addition, Akbar et al. [51] made a summary of the inconsistent terminology used among different researchers when describing a particular flow pattern. They pointed out that the churn flow defined by Zhao and Rezkallah [52] might fall in the regime defined by other researchers where they called it slug annular regime or wavy annular regime. Flow regimes were insensitive to the channel orientation when the hydraulic diameter of the channel was less than 1 mm . One of the obvious inconsistent terminology descriptions is that Triplett et al. [40] defined slug flow shown in Fig. 2.7 and they are different from the definition of Barajas and Panton [46] in Fig. 2.4 (c).

FIG. 2.6: Comparison of the observed flow patterns with the transition lines by Chung and Kawaji [50] (reprinted from Saisorn and Wongwises [48]).

FIG. 2.7: Slug flow pattern defined by Triplett et al. [40].

To avoid the above mentioned inconsistencies, terminology of the flow patterns observed by the present study will be redefined in Section 3.2.1.

2.2.4 Channel inlet condition

In order to investigate the two-phase flow characteristic in the microchannel, stable flow pattern that will not change over time is highly essential. Hence, stable liquid and gas pump which is capable of providing constant flow rate over a long period of the time has become a vital requirement in the design of the test system. For the liquid flow, water was pumped by pressurizing the water vessel with constant pressure provided by the high pressure nitrogen cylinder was commonly employed in the experiment due to the pulsation free mechanism. For the gas flow, the

nitrogen gas was often introduced to the test system by employing a mass flow controller to discharge an accurate amount of nitrogen gas from the high pressure nitrogen cylinder.

Although the pumping sources employed were similar in a number of the experiments conducted by researchers, the flow characteristic observed could be very different. Some researchers found that the two-phase flow pattern tends to change from one to another over a period of time.

Undeniably, the different flow patterns observed could be caused by the channel material wettability, channel diameter, surface roughness and fluid properties. If these properties are constants, and the input conditions remain unchanged, the flow pattern should not vary over a period of time. Hence, the most possible reason that causes the flow pattern to be different over the time could be the design of the gas liquid injection methods.

According to the literature, the inlet design with a mixing chamber before the two-phase flow was introduced into the microchannel will cause the bubbles to be released periodically [53], or bubble train which contained a random number of bubbles in each train will be observed [41]. The inlet mixing designed by Chen et al. [41] and Kawahara et al. [53] are shown in Figs. 2.8 and 2.9, respectively.

In addition, Kawahara et al. [44] studied the two-phase flow characteristic using the channel inlet shown in Fig. 2.10. They found that the appearance of liquid alone, thin liquid film and ring liquid films flow patterns were unpredictable in the channel though the water and gas flow rate were remain constant. Hence, the flow patterns observed by them were rather random. However, no periodically flow pattern was observed by Hayashi et al. [54] with the channel inlet design shown in Fig. 2.11.

Kawaji et al. [55] designed the inlet geometric shown in Fig. 2.12 so as to examine the effects of inlet conditions of two-phase flow. Their experimental results showed that the inlet geometric and method of gas liquid mixing could significantly change

FIG. 2.8: Channel inlet design of Chen et al. [41].

FIG. 2.9: Channel inlet design of Kawahara et al. [53].

FIG. 2.10: Channel inlet design of Kawahara et al. [44].

FIG. 2.11: Channel inlet design of Hayashi et al. [54].

the void fraction for the same microchannel at given gas and liquid superficial velocities.

FIG. 2.12: Channel inlet design of Kawaji et al. [55].

In summary, the mixing of two fluids or gas-liquid could be classified into two categories. First, injected gas-liquid with a T-junction or having a mixing chamber before the gas-liquid flow into the interested channels may be the reason that bubbles will periodically be released into the test channel. Second, injecting the gas into another fluid via a center tube could generate a more consistent flow pattern.

2.2.5 Optical distortion

Circular glass tube will cause optical distortion which would magnify the inner diameter of the glass tube. Kawahara et al. [44] investigated the optical distortion of the circular tube. Their experiment showed that the microchannel could be enlarged by about 50% without any optical correction box. The optical distortion only had the magnifying effect in the radial direction. They confirmed that, with the magnification effect, the details at the near wall region could be better identified.

FIG. 2.13: Images of gas core surrounded by liquid film: (a) with and (b) without an optical correction box Kawahara et al. [44].

Hence, the magnification effect of the circular glass tube in the radial direction is not corrected during the present experiment in order to have a clearer boundary view of the bubble and wall tube.

2.2.6 Void fraction

Measurements of void fraction in microchannel have been difficult due to the small diameter. A variety of methods to estimate the void fraction had been proposed by many researchers. Most of the void fractions were obtained by image processing. The main difficulty of void fraction estimation using the optical method is the optical distortion which has been discussed in Section 2.2.5.

Kawahara et al. [44] conducted the experiments to investigate the two-phase flow characteristics in a $100\ \mu\text{m}$ diameter circular tube by using de-ionized water and

nitrogen gas as the working fluids. The void fraction was determined using the following equation.

Time averaged void fraction:

$$\alpha = \frac{\text{Number of gas core images}}{\text{Total number of images counted}} \quad (2.1)$$

And they proposed the following empirical correlation:

$$\alpha = \frac{C_1 \beta^{0.5}}{1 - C_2 \beta^{0.5}} \quad (2.2)$$

where the constants were sensitive to the channel size: $C_1=0.02$ and $C_2=0.98$ for the 50 μm diameter channel and $C_1=0.03$ and $C_2=0.97$ for the 100 μm diameter channel.

Chung and Kawaji [50] conducted the experiments of nitrogen gas and water two phases flow in the circular channels of 530, 250, 100 and 50 μm inner diameters. Void fraction was determined by fitting the region occupied by gas to shapes of symmetrical volume and estimating the fraction of gas volume. Meanwhile for the flow with thick smooth film, the volume fraction of gas was estimated by squaring the ratio of the gas core radius to the channel radius [44]. The experimental results shown in Fig. 2.14 reported that 530 μm channel was well-predicted by the homogeneous flow model, the 250 μm channel agreed favorably with the Armand type correlation, whereas the 100 and 50 μm followed the proposed empirical correlation of Eq. 2.2.

The cross-sectional void fraction results, concerning only with bubbly and slug flow, of Serizawa et al. [45] shown in Fig. 2.15 shows that the void fractions are very close to Armand correlation for the i.d. 20 μm channel. This is a rather different trend observed by Kawahara et al. [44]. Hence, the large deviation of the void fraction from the homogenous and Armand correlation for channels below 100 μm remains an open question.

FIG. 2.14: Relationship between void fraction and volumetric quality (reprinted from Chung and Kawaji [50]).

FIG. 2.15: Void fraction of 20 μm i.d. silica tube (reprinted from Serizawa et al. [45]).

Saisorn and Wongwises [48] conducted a study of the air water flow through a 0.53 mm i.d. circular channel. The volumetric void fraction was estimated from the images taken. Air phases were assumed to be symmetrical volumes, covering spherical and ellipsoidal segments and cylinders for the analysis of void fraction. Their results for void fraction could be well correlated by homogeneous flow model.

Assuming one dimensional flow and small slip velocity, drift flux model (discussed in Section 3.2.5) is another popular method to estimate the void fraction. Hayashi et al. [54] estimated the void fraction $\alpha = j_G/u_G$ and concluded that the void fractions for 0.6 mm channel was in good agreement with Armand correlation but the void fraction for 0.3 mm channel disagreed with the Armand correlation and varied with the change of liquid velocity.

Kawahara et al. [53] determined the void fraction by $\alpha = j_G/u_G$, where u_G is the bubble velocity estimated from the images taken by calculating the distance that the bubble moves over a number of frames. The image width is 2.6 mm and recorded at 8000 frames per second. Using the drift flux model $u_G = C_0 j + V_{Gj}$. Setting the V_{Gj} as zero because the studies were on horizontal channels, the correlation derived became $C_0 = aBo^{0.19}Re_L^{-0.01}We_G^{0.01}$. Furthermore, Chen et al. [41] investigated the inner diameter of 1 mm microchannel. No other bubbles other than the large Taylor bubble were normally observed in their experiments. Hence the average void fraction was indirectly evaluated with $\alpha = j_G/u_G = j_G/u_B$. Drift flux model was modified as $\alpha = j_G/(0.932j^{1.11})$.

Mishima and Hibiki [37] conducted the two-phase flow experiment for channel i.d. from 1 to 4 mm to investigate the void fraction. They concluded that the distribution parameter for bubbly and slug flows was a decreasing function of inner diameter and could be correlated by the equation: $C_0 = 1.2 + 0.510e^{-0.691d}$, where the distribution parameter C_0 will be further discussed in Section 3.2.5.

2.3 Previous work on heat transfer

Enhancing the efficiency of the heat transfer in small diameter channels without increasing the pressure drop has been a key motivation that attracts researchers in this field of research. The two-phase flow in particular would induce higher pressure loss when the gas is introduced into the channel. The proper understanding of the flow characteristic and heat transfer mechanisms in microchannels are crucial in designing micro heat exchanger or electronic cooling devices.

2.3.1 Single-phase flow

Peng and Peterson [24] performed single-phase forced convective heat transfer in rectangular channels with hydraulic diameters of 0.113 to 0.367 *mm*. Geometric configuration was pointed to have significant effect to the heat enhancement and flow characteristics.

Wu and Little [56] performed the experiment on nitrogen gas flowing through micro-heat exchanger so as to investigate the heat transfer characteristic of flow in both laminar and turbulent regime. It was found that the Nusselt numbers were higher than the prediction of conventional theory and large relative roughness was concluded to be capable of increasing the heat transfer coefficient and the friction factor. Besides, Kandlikar et al. [57] experimentally examined the effect of surface roughness on the single-phase heat transfer in channels with i.d. of 0.62 and 1.067 *mm*. They concluded that surface roughness could increase the heat transfer. At the same roughness level ($\varepsilon/d = 0.003$), the i.d. 0.62 *mm* tube had shown a better heat transfer rate than the i.d. 1.067 *mm* tube.

On the contrary, Qu et al. [58] compared the single water phase heat transfer experimental results with numerical predicted results in trapezoidal silicon channels of hydraulic diameter ranging from 62 to 169 μm . Nusselt numbers of experimental results were found to be much lower than the numerical analysis and surface roughness was suspected to be the reason that caused the deviation.

Furthermore, Guo and Li [59] also suggested that surface roughness could be the main reason causing early transition from laminar to turbulent flow. This may explain the corresponding increase in the friction factor and Nusselt numbers.

Experiments conducted by Acosta et al. [60] on smooth rectangular channels agreed with the conventional heat transfer correlations. Owhaib and Palm [61] conducted experiments on 1.7, 1.2 and 0.8 *mm* inner diameters circular channels and compared their experimental results with the macroscale heat transfer correlations and the proposed microscale correlations. They found that the experimental results agreed with the macroscale correlations but not the proposed microscale correlations. In addition, Lin and Yang [62] experimentally proved that the conventional single-phase heat transfer correlations could be applied on the channel with inner diameter of 123 μm . Studies of Rosa et al. [63] and Grohmann [64] also shown that conventional heat transfer correlations could be used to predict the microchannel heat transfer well. Careful measurements conducted by Lelea et al. [65] and Celata et al. [66] where the experiments were conducted in vacuum chamber so as to minimize the heat loss had shown that the conventional theory could correlate the experimental results well.

Besides, Gao et al. [67] designed an experiment setup that was flexible to change the channel thickness from 1 to 0.1 *mm* while maintaining the same top and bottom channel surface so as to keep the surface roughness constant and investigate the effect of channel dimensions. Their experiment results showed that friction factor and Nusselt number were in agreement with channel thickness that was greater than 0.4 *mm*, but the Nusselt number was lower than the conventional theoretical results when the channel thickness became smaller. In addition, greater friction factor and lower Nusselt number than the macroscale correlation for the quartz microtubes with i.d. 19.6 μm was reported in the experiments conducted Zhigang et al. [68]. Effect of the variation of the thermophysical properties with temperature was concluded to be the reason for the deviation.

Peng and Wang [69] investigated single-phase flow through the rectangular channels with cross-section of 0.6×0.7 *mm*. They compared their experimental results

with the conventional size channel heat transfer correlations and showed that the Nusselt number was lower than the predictions by conventional correlations used when the Reynolds number was low. In addition, Wang and Peng [70] obtained lower Nusselt numbers for their experiments conducted on rectangular stainless steel channel. The experimental results were in good agreement with the modified Dittus-Boelter equation.

Peng and Peterson [71] conducted experiment of water flow through rectangular cross-sectional channel so as to investigate the effect of the thermophysical fluid properties towards the heat transfer in microchannel and demonstrated that an optimum channel size existed for the forced convective flow heat transfer of single-phase flow in rectangular channel. Furthermore, Peng et al. [72] concluded that a transition region existed in microchannel would change the thermophysical properties of fluid and hence change the Reynolds number along the channel as well as heat transfer performance.

Maranzana et al. [73] studied analytical models of channels between parallel plates and the results showed that neglecting the axial conduction in walls for low flow rates heat transfer might lead to an underestimation of heat transfer coefficient. More recently, Lelea [74] numerically analyzed the heat transfer in channel having inner diameter of 0.1 *mm*. Temperature dependent thermal conductivity was considered and found that it significantly influenced the Nusselt numbers when the Reynolds number was lower than 417.

A comparative review on heat transfer correlations proposed for microchannels in open literature with the conventional macroscale heat transfer correlations as shown in Fig. 2.16 was presented by Sobhan and Garimella [75]. Overall, the predictions showed that the Nusselt numbers increased with increasing of Reynolds number and the dependence of Nusselt number on Reynolds number in microchannel predictions are stronger than the conventional correlations. Lastly, little agreement between the predictions of different investigators could be concluded. In addition, Morini [76] has done a comprehensive review for the single-phase convective

heat transfer in microchannels and tabulated the summary of Nusselt numbers obtained by various researchers in comparison with the conventional theory shown in Fig. 2.17. Accordingly, it is clear to see that the Nusselt numbers obtained by previous researchers did not follow the same trend.

FIG. 2.16: Heat transfer predictions from the literature for microchannels and conventional channels, in the laminar and turbulent regimes (reprinted from Sobhan and Garimella [75]).

Experimental results on the Nusselt number for single-phase internal flows through microchannels

	Nu	D_h [μm]	Cross-section	Test fluids
Wu and Little [78]	$\uparrow\uparrow$	55.8–72.4	Trap.	N_2
Choi et al. [22]	$\uparrow\uparrow$	3–81	Circ.	N_2
Yu et al. [33]	$\uparrow\uparrow$	19–102	Circ.	Water, N_2
Rahman and Gui [24]	$\uparrow\uparrow$	176–325	Trap.	Water
Peng and Wang [79]	$\downarrow\downarrow$	646	Rect.	Water
Wang and Peng [73]	$\downarrow\downarrow$	311–747	Rect.	Water Methanol
Peng et al. [80]	$\downarrow\downarrow$	311–646	Rect.	Methanol
Peng et al. [31]	$\downarrow\downarrow$	133–367	Rect.	Water
Peng and Peterson [74]	$\downarrow\downarrow$	311–747	Rect.	Water
				Methanol
Peng and Peterson [32]	$\downarrow\downarrow$	133–367	Rect.	Water
Acosta et al. [17]	$\approx$	369–990	Rect.	He
Harms et al. [40]	$\approx$	404	Rect.	Water
Cuta et al. [81]	$\uparrow\uparrow$	425	Rect.	R124
Ravigururajan et al. [82]	$\uparrow\uparrow$	425	Rect.	R124
Nguyen et al. [76]	$\uparrow\uparrow$	690	Trap.	Water
Adams et al. [83]	$\uparrow\uparrow$	102–1090	Circ.	Water
Adams et al. [84]	$\approx$	> 1200	quasi Tri.	Water
Tso and Mahulikar [85]	$\downarrow\downarrow$	717–741	Circ.	Water
Celata et al. [53]	$\uparrow\uparrow$	130	Circ.	R114
Qu et al. [86]	$\downarrow\downarrow$	62–169	Trap.	Water
Rahman [87]	$\uparrow\uparrow$	299–491	Rect.	Water
Debray et al. [59]	$\downarrow\downarrow$	590–2218	Rect.	Water
Jiang et al. [62]	$\uparrow\uparrow$	300	Rect.	Water
Kandlikar et al. [64]	$\uparrow\uparrow$	620–1067	Circ.	Water
Gao et al. [65]	$\downarrow\downarrow$	199.2–1923	Rect.	Water
Qu and Mudawar [69]	$\approx$	349	Rect.	Water
Warrier et al. [66]	$\approx$	750	Rect.	FC-84
Bucci et al. [70]	$\approx$	290	Circ.	Water
Wu and Cheng [88]	$\approx$	69.2–160	Trap.	Water

 $\uparrow\uparrow$ — Nu higher than the conventional theory. $\downarrow\downarrow$ — Nu lower than the conventional theory. $\approx$ — Nu agrees with the conventional theory.

FIG. 2.17: Summary of single-phase Nusselt numbers (reprinted from Morini [76])

2.3.2 Two-phase flow

2.3.2.1 Flow boiling (Phase change)

The increasing requirement for dissipation of high heat fluxes while maintaining specified surface temperatures has made the single-phase liquid cooling and two-phase cooling as another alternative solution. Two-phase flow is found have a higher heat transfer rate than the single-phase flow. The flow boiling is even more attractive than the single-phase liquid cooling due to the high heat transfer coefficient during flow boiling and high heat removal capacity of the coolant at a given mass flow rate.

Experiments conducted by Tran et al. [77] in circular and rectangular channels with both hydraulic diameters of 2.46 mm using R-12 as the working fluid. Two regimes, where lower and higher wall superheated temperature dominated by convective boiling and nucleative boiling respectively, were distinguished. Kandlikar [78] observed that flow reversal caused by expanding bubbles pushing the liquid-vapor interface in both upstream and downstream direction. Flow instability due to large pressure fluctuation generated by the bubble nucleation was noted as well. Furthermore, Huh and Kim [79] conducted flow boiling experiment in a single microchannel with i.d. of $100\text{ }\mu\text{m}$ and length of 40 mm . The boiling flow was dominated by nucleate boiling mechanism and the main flow was characterized by the fast and elongated slug bubbles grown from single bubbles. Experimental heat transfer coefficient was concluded to be independent of mass flux and vapor quality.

Yen et al. [80] conducted convective boiling on the channels with i.d. of 190 , 300 and $510\text{ }\mu\text{m}$. The heat transfer coefficient was found to be independent of the heat and mass fluxes. Inner diameter, surface roughness, system pressure and latent heat of the working fluid were concluded to be the factors affecting the heat transfer coefficient. In addition, Yen et al. [81] carried out convective flow boiling heat transfer with visualization in circular and square Pyrex glass channel with inner diameter of $210\text{ }\mu\text{m}$ and hydraulic inner diameter of $214\text{ }\mu\text{m}$, respectively. Active

nucleation cavities at the corners of the square channel were found to be the reason that the heat transfer coefficient was higher than the circular microchannels.

Hetsroni et al. [82] found that under certain flow and heat transfer conditions the explosive boiling, where bubble grew rapidly, would occur in the microchannels and the alternative occurrence of single-phase liquid flow and two-phase liquid vapor flow in the channels would cause the pressure drop to fluctuate. Zhang et al. [83] experimentally examined the phase change phenomena in silicon channels with i.d. ranging from 27 to 171 μm . They concluded that the flow boiling mechanisms were dependent on the channel size. Nucleate boiling was expected in channels larger than 100 μm while explosive boiling was expected in channels smaller than 50 μm . Additionally, the pressure pulses occur at the onset of boiling was expected to affect the local boiling and equilibrium conditions.

Furthermore, experiments conducted by Zhang et al. [84] to measure the pressure drop of flow boiling for the microchannel cool plate where the number of microchannels was 100 with hydraulic diameters of 99.6 μm each. The obtained pressure drop results is shown in Fig. 2.18 where the DI water volume flow rate and power input were 7.9 ml/min and 60 W , respectively. The instability of the flow boiling could be shown from this pressure drop results. The pressure drop across the microchannel increased while the pressure drop across the flow meter fell to almost zero with a period of 20 to 30 seconds indicating that there was no incoming water into the microchannel cold plate. Hence the wall temperature of the microchannels should be oscillating as well.

In addition, Schilder et al. [12] performed visualization experiment of single-phase and two-phase flow boiling on inner diameter 0.6 mm tubes. single-phase friction factors and Nusselt numbers were reported to agree with the conventional values from Poiseuille flow. Local Nusselt numbers for flow boiling were increased by 3 to 8 times higher than the single-phase flow in microchannel. However, the pressure drop for flow boiling had been fluctuating at the value that was about 2.5 to 3 times higher than the pressure drop for single-phase flow (Fig. 2.19).

FIG. 2.18: Experimental flow oscillations in boiling microchannel system (reprinted from Zhang et al. [84]).

FIG. 2.19: Pressure drop, transition from single-phase flow to boiling, $\dot{m} = 237 \text{ kg/m}^2 \text{ s}$ and $q'' = 79000 \text{ W/m}^2$ (reprinted from Schilder [12]).

2.3.2.2 Non-boiling flow (No phase change)

Early investigation for the non-boiling two-phase flows at different Reynolds numbers was conducted by Horvath et al. [85]. Their results were limited to $\alpha_G = \alpha_L = 0.5$, where α_G and α_L are the void fraction of the gas and liquid phase, respectively. However, they demonstrated that the ratio of tube length to diameter (L/D) has only minor effect to the overall heat transfer rate.

Thulasidas et al. [86] investigated the bubble train flow in capillaries of circular and square cross-sectional channels. The enhanced heat transfer rates were concluded to be attributed to the bubble diameter and the thickness of the thin film between the bubble and the channel wall.

Gas liquid concurrent flow without phase change experiments were conducted by Hayashi et al. [54] to examine the effects of inlet flow conditions on glass tube of 300 and 600 μm channels. They found that the bubble formation process was strongly affected by the inlet conditions and pressure loss was affected by the bubble size. Though the heat transfer rate was concluded to be increased by the mixing effect, no further details for the heat transfer mechanism such as flow patterns, void fraction and bubble velocity had been elaborated

Furthermore, Asthana et al. [87] conducted an experimental study of convective heat transfer in serpentine microchannels by using water as the continuous phase and light mineral oil as the segmented phase of the flow. The local velocity and temperature were measured by micro-PIV method and the LIF (Laser-Induced Fluorescence) method. Their experimental results for local velocity and temperature are shown in Figs. 2.20 and 2.21. The velocity field shown in Fig. 2.20 illustrates that strong circulation occurred in the water slug. This phenomenon explains that why the temperature at the center of the water slug (Fig. 2.21) is higher than the wall temperature. Their experimental results show that the circulation in the water slug has increased the Nusselt number by 2 to 3 times than the single-phase flow. However, no relationship of flow patterns and the heat transfer rate were investigated.

FIG. 2.20:

The relative velocity field between two consecutive slugs (where the leading slug is adjacent to the left side of the image and is not shown) at $100 \mu\text{l}/\text{min}$ water flow rate: (a) $30 \mu\text{l}/\text{min}$ oil, (b) $20 \mu\text{l}/\text{min}$ oil and (c) $10 \mu\text{l}/\text{min}$ oil. (reprinted from Asthana et al. [87])

FIG. 2.21:

A snapshot of the local temperature distribution between two slugs (water flow rate: $100 \mu\text{l}/\text{min}$, oil flow rate: $30 \mu\text{l}/\text{min}$). (reprinted from Asthana et al. [87])

2.4 Previous numerical work

Computational fluid dynamics (CFD) is a powerful tool for the investigation of multiphase flow. The visualization capabilities could provide valuable insights into the complex flow field of the multiphase flow. Nevertheless, difficulties still exists. This is because solving the full Navier-stokes equation becomes challenging due to the constantly deforming interface between the phases. Moreover, the difficulty increases as the computation domain turn into microscale where the surface tension effects become dominant.

Some of the numerical methods employed for the multiphase flow interface tracking are listed below.

- Front tracking method

Glimm et al. [88] were the first to implement the front tracking method. This method was basically developed by Glimm's group and Tryggvason's group. Front tracking method utilizes the marker particles to localized the interface. It is combination of fixed and moving mesh method. The front grid moves through the fixed grid to track the location and geometry of the interface. The computation of the interfacial tension is performed at the front grid and transferred to the global grid.

The advantage of the front tracking method is that no numerical diffusion of mass across the fluid interface is allowed. At the meantime, the drawback of the front tracking method would be the difficulties in applying for the interfaces with topological changes.

- Level set method

Level set method was firstly developed by Osher and Sethian in 1980s. Review of the level set method can be found in Osher and Fedkiw [89]. The level set function is as follows, $\frac{\partial \phi}{\partial t} + u \cdot \nabla \phi = 0$. where ϕ is the signed distance function from the point to the interface. Generally, the interface of the shape is the zero level set of signed distance function. The interior and exterior of

the shape are positive and negative values, respectively. Whenever there is a topological change, the level set method will handle it automatically.

However, due to the level set function will tend to deviate from the signed distance function, a reinitialization process is needed to maintain the accuracy of the level set function to the signed distance function. Hence, the mass of the shape may not be gain or loss. The mass conservation becomes one of the major concern when level set function is used.

- Volume of fluid method (VOF)

The first publication of the volume of fluid method was by Hirt and Nichols [90] in 1981. Review of the VOF method can be found in Scardovelli and Zaleski [91]. VOF method utilized the fraction function $\frac{\partial q_f}{\partial t} + \vec{u} \cdot \nabla q_f = 0$, to trace the interface between phases. When the value of q_f is zero, the function will see no traced fluid exists in the cell. When the cell is full of the traced fluid, q_f becomes one. When the value of q_f is within zero and one, the interface is contained within the cell.

The advantage of the VOF method over the other methods is its ability to conserve the mass of each fluid. However, due to the nature of VOF method, q_f is discontinuous. Thus the interface will tend to be smeared. Therefore, Piecewise-Linear Interface Calculation (PLIC) scheme (which is also known as geometric reconstruction in Fluent) is often served as a treatment to this imperfection.

Among the previous discussed methods, level set method and volume of fluid method are the most used interface tracking methods. Therefore, other methods such as Lattice Boltzmann Method [92] and phase-field method [93] are not discussed in the present study due to their lesser significant applications to the two-phase flows in microchannel.

CFD for the multiphase flow is not a new topic. However, the numerical simulations for the multiphase flow are not as extensive as the experimental investigations. The possible reason could be due to the existence of the interface that

increases the complexity of the flow field. Another potential reason that causes the CFD study to lag behind the experimental investigations could be the not so well-understood flow conditions in microscale that has slowed down the progress of validation for the computation results.

Only until recently, the flow visualization experimental results has proved that the numerical computation method could well-predicting the shape and obtains the flow field in microscale. Some of the numerical investigations for non-boiling two-phase flows are gradually appearing in literatures.

Fukagata et al. [94] numerically investigated the heat transfer effect of a gas bubble immersed in water in the channel of i.d. $20\ \mu m$. They found that bubble length could be one of the factors that affecting the heat transfer characteristics and concluded that the enhancement of thermal performance of the flow was due to the strong circulations in the water phase induced by the gas bubbles. However, the assumption of an initial shape of the bubble and fixed pressure drop across the bubble might be difficult to reproduce in laboratory.

The heat transfer simulation performed by Lakehal et al. [95] showed that the heat removal rate varies with different flow patterns (e.g. bubbly, slug and slug-train flows). Slug and slug train flows was found to have higher heat removal rate than the bubbly flow due to the larger blockage to the flow. Heat transfer performance could be two to three times higher than the single water phase flow. Correlation for Nusselt number was proposed as $Nu_D = 3.67 + 0.022Re_D^{4/5}Pr^{0.4}$, where Re_D is based on the velocity of the gas phase.

In addition, Narayanan and Lakehal [96] has also numerically examined the heat transfer enhancement for the small diameter channels. Though the influence of the slug length on the heat transfer rate was investigated, it was only limited to the cases they studied. More details, such as optimal slug length for the greatest gain in heat transfer performance was not considered.

To clarify the existence of the thin liquid film between the Taylor bubble and the wall, Gupta et al. [97] has conducted a CFD modelling for Taylor flow in

microchannels using the ANSYS Fluent. Their results showed that the thin liquid film could be adequately captured if sufficiently fine mesh was applied.

He et al. [98] proposed a model which decomposing the whole flow field into an adherent liquid film near the wall and the gas liquid slug regions, where both liquid and gas slugs were assumed to be cylinders. This model has reasonably improved the heat transfer prediction by accounting the thermal coupling between the slug and the film regions, but an error of about 25% was confirmed.

Numerical simulation performed with Fluent and Transat by utilizing both level set and volume of fraction method to capture the interface was compared by Gupta et al. [20]. The comparison showed that the results given by both methods were closely matched. Except that the bubble size generated by Fluent was slightly larger due to the late necking process. The thin film thickness between the bubble and wall was found to depend on the Capillary number and Reynolds number. The average Nusselt number was approximately 2.5 times higher than that of the single-phase laminar flow.

Similarly, Carlson et al. [99] also compared the bubbles generated by Fluent and Transat. Their simulation results showed that the bubble frequency was similar using both softwares, but the obtained flow fields were rather different. Fluent would fail to capture the gas liquid interface when the flow rates were high.

A simplified model for the heat transfer of plug flows in microchannels with different cross-sectional shapes was proposed by Muzychka et al. [100]. Their approach was based on the Graetz problem where the wall temperature and fluid inlet temperature were assuming to be uniform. This simplified model was analytically solved by considering uniform velocity profile for the entire liquid phase. Therefore, the influence of the fluid circulations within the liquid flow might be neglected.

Simulation of the multiphase flow is computation costly. In order to reduce the computation time, moving frame reference method has been utilized. Moving frame reference simulation which could accelerate the computation process has

been examined by Talimi et al. [21]. Their simulation results showed that the moving frame of reference method would over-predict the pressure drop and heat transfer. Additionally, they also pointed out that the large value of Nusselt number around that region where the bubble exists might be caused by the small temperature difference between the thin liquid film around the bubbles and the wall temperature. However, no experimental flow patterns and comparison for the bubble size and Nusselt numbers were concluded.

More recently, Che et al. [101] analyzed the heat transfer rate of a single plug flows through the microchannel numerically. Their analysis of different plug length ranging from 1 inner diameter to 4 inner diameters showed that the shortest plug length (1 inner diameter) has the highest heat transfer performance. However, only the heat transfer rate of a single plug, but not series of plug, had been investigated.

Generally, almost all of the above reviewed articles were more focusing on the local flow field around the bubble phase. Fluid circulations within the slug itself have been identified to be the main reason that enhances the heat transfer performance. However, the optimal slug length which could give maximum heat transfer enhancement while the pressure penalty could be kept to the minimum was not discussed.

2.5 Summary

Microchannels were first proposed for electronics cooling applications by Tuckerman and Pease [6]. The silicon chips embedded with microchannels was demonstrated to be able to dissipate $7.9 \text{ MW}/\text{m}^2$ with a maximum substrate temperature to inlet water temperature difference of $71 \text{ }^\circ\text{C}$ by employing direct circulation of water to the microchannels. However, the pressure losses for the respective one square centimeter of chip were over 2 bars.

For most of the electronic components, constant temperature of $85 \text{ }^\circ\text{C}$ or below is essential to maintain smooth operation. Heat fluxes on electronic chips could easily be as high as $300 \text{ W}/\text{cm}^2$, which is the heat flux generated in the state of the art chips.

In the case where the size of the electronic device is small, the pumps utilized will be small in size as well. Thus heat transfer efficiency would be indirectly limited by the pumping power of the micro pump. Fig. 2.22 shows the maximum back pressure of the micro pump at zero flow rate. It is clear that the micro pump would fail to deliver required flow rate for the microchannel cooling due to the high pressure drop in the microchannel. So that, flow boiling which requires high pumping power might not be suitable for the micro-electronic cooling and an alternative two-phase cooling method is necessary.

In short, many experiments with flow visualization carried out by previous researchers to investigate the flow characteristic of two-phase flow as discussed in Section 2.2 were performed under adiabatic condition. The literature investigations on two-phase flow with heat transfer in microchannels were mostly related to the flow boiling, where the nucleative boiling and explosive boiling may cause the flow to be unstable for investigations of the heat transfer phenomenon as discussed previously.

In addition, some factors that cause the high heat flux cooling systems with flow boiling to lag behind of the single-phase liquid cooled systems was addressed by Kandlikar [103] as follows:

FIG. 2.22:

Maximum back pressure of the micro pump at zero flow rate. (reprinted from Singhal et al. [102].

1. The need for low pressure water or a suitable refrigerant to match the saturation temperature requirement for electronics cooling.
2. Unstable operation due to rapid bubble expansion and occasional flow reversal.
3. Low critical heat flux levels due to instability, unavailability of critical heat flux data over a wide operating range, and a lack of fundamental understanding.

However, the inlet condition design proposed by Hayashi et al. [54] had proved that the bubble formations could be controlled and suggesting that the further investigations for the stable flow patterns become possible. Although the heat transfer of the two-phase flow without phase change were simulated [20, 21, 85, 94, 95, 98, 100, 101], no experimental data were yet available. Furthermore, Asthana et al. [87] discerned the strong circulation for flow in water slug was the reason that

the Nusselt numbers were increased. But no relationship of flow patterns and the heat transfer rate were discussed. Therefore, the experimental investigations for the two-phase flow without phase change are essential to further our understanding on the actual heat transfer mechanism in microchannels.

The present work is to conduct experiments to investigate the heat transfer characteristics of non-boiling two-phase flow in the microtubes. To extend our understanding for the two-phase flow heat transfer mechanism in the microtubes, numerical computational method has been adopted to analyze the liquid velocity profile and bulk fluid mixing mechanism. In addition, the optimal heat transfer condition for the non-boiling co-axial flow has been identified.

Additionally, a controllable discrete flow where two immiscible fluids would flow alternately in microtubes to obtain fully separated phases are analyzed. Unlike the bubbly flow and slug flow, the two phases flow in completely separated mode and no dispersed phases is encapsulated by the continuous phase. Our goal is to assess the potential and make optimizations for such multiphase flow in microchannels, based on the analysis of the characteristics of heat transfer and pressure loss.

Chapter 3

Terminology and Equations

3.1 Single-phase flow

3.1.1 Pressure drop measurement

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

$$\frac{\Delta P}{L} = f \frac{\rho u^2}{2d} \quad (3.1)$$

Where d is the inner diameter of the channel, ρ is the density of the fluid, U is the mean fluid velocity which can be obtained by the flow meter and $\Delta P/L$ is the measured pressure loss per unit length across the microchannel that can be obtained by a digital pressure gauge.

3.1.2 Volumetric flow rate measurement

In order to determine the volumetric flow rate for the experiment, the discharged water can be collected in a beaker and the collection time is recorded. The mass

of the discharged water can be measured by the electronic balance. The following equation can be used to obtain the volume of the water.

$$v = \frac{m}{\rho} \quad (3.2)$$

The volumetric flow rate can thus be obtained by dividing the value of the volume with the recorded collection time.

$$\dot{Q} = \frac{v}{t} \quad (3.3)$$

3.1.3 Friction factor measurement

The friction factor measurement for the circular channel of inner flow is governed by the equation below.

The experimental friction factor is:

$$f = \frac{64}{Re} \quad (3.4)$$

The friction factor of the present experiment can be calculated from the eq. 3.5 which is obtained by rearranging the Darcy Weisbach equation as shown in eq. 3.1.

$$f = 2 \frac{\Delta P}{L} \frac{d}{\rho u^2} \quad (3.5)$$

3.2 Two-phase flow

3.2.1 Definition of flow patterns

Flow patterns defined in this study are:

a) Dispersed bubbly flow

Many small bubbles with the diameter much smaller than the channel

b) Bubbly flow (Fig. 3.1 (a))

Bubbles having diameter smaller than the channel inner diameter

c) Slug flow (Fig. 3.1 (b))

Bubbles having diameter larger than the channel inner diameter and hence being elongated in the streamwise direction

d) Slug-annular flow (Fig. 3.1 (c))

An intermediate flow between slug and annular flow

e) Churn flow (Fig. 3.1 (d))

Large elongated bubbles which exist when the pressure fluctuates. The missing interface between water and gas slugs represents a high level of mixing is taking place.

FIG. 3.1: Two-phase flow patterns (taken from the present experimental setup).

3.2.2 Superficial velocity

For multiphase flow in a pipe, the superficial velocity of each phase is the mass flow rate $\dot{m}_G$ of that phase divided by the channel cross-sectional area A_C and material density. The superficial velocity for the dispersed phase can be written as:

$$j_G = \frac{\dot{m}_G}{\rho_G A_C} = \frac{\rho_G A_G u_G}{\rho_G A_C} \quad (3.6)$$

Alternatively,

$$j_G = \frac{\dot{Q}_G}{A_C} \quad (3.7)$$

Where u_G is the phase velocity, $\dot{Q}_G$ is the volumetric flow rate of the phase. The subscript G is referring to gas phase. In short, superficial velocity is the velocity of gas or liquid in the pipe when each is occupying the channel alone.

3.2.3 Phase velocity

The phase velocity u_G is the actual velocity of the phase. The superficial velocity and the phase velocity are related by the volume fraction.

$$j_G = \frac{\alpha}{u_G} \quad (3.8)$$

Bubble velocity is measured directly from the images taken from the high speed camera using the following equation:

$$u_G = \frac{N_{bubblepixel}}{\Delta t} \times \frac{L}{N_{pixel}} \quad (3.9)$$

Where $N_{bubblepixel}$ is the distance of the bubble traveled in the continuous images measured in the unit of pixel. Δt is the time interval for the bubble to travel across the measured distance. Furthermore, L is the reference length and N_{pixel} is the number of pixels over a reference length (usually 1 mm).

3.2.4 Volumetric gas flow ratio

Volumetric gas flow ratio is defined as:

$$\beta = \frac{\dot{Q}_G}{\dot{Q}_G + \dot{Q}_L} = \frac{j_G}{j_G + j_L} = \frac{j_G}{u_{TP}} \quad (3.10)$$

Where,

$\dot{Q}_G$ is the volumetric flow rates of the gas phase [m^3/s]

$\dot{Q}_L$ is the volumetric flow rates of the liquid phase [m^3/s]

j_G is the superficial velocity of the gas phase [m/s]

j_L is the superficial velocity of the liquid phase [m/s]

u_{TP} is the two-phase mixture velocity [m/s]

3.2.5 Void fraction

Void fraction is important in the calculation of pressure gradients and is relevant to the calculation of the amount of liquid and gas present in a system. In a control volume made up of liquid and gas mixture, void fraction is the volume fraction of the gas phase.

The void fraction is given by,

$$\alpha = \frac{\text{volume of gas}}{\text{volume of the control volume}} \quad (3.11)$$

The void fraction correlation given by Wallis [104] is:

$$\alpha = \frac{j_G}{u_G} \quad (3.12)$$

The followings are the homogenous flow model and drift flux model that are widely used to predict the void fraction in the channels.

Homogenous flow model

Velocity ratio S is defined as:

$$S = \frac{u_G}{u_L} = \frac{\dot{Q}_G/A_c\alpha}{\dot{Q}_L/A_c(1-\alpha)} = \frac{j_G/\alpha}{j_L/(1-\alpha)} = \frac{j_G(1-\alpha)}{j_L\alpha} \quad (3.13)$$

Rearrange eq. 3.13, α becomes:

$$\alpha = \frac{j_G}{Sj_L + j_G} \quad (3.14)$$

For homogeneous model, gas and liquid velocity are assumed to be flowing at the same velocity.

Thus, $u_G = u_L$, $S = 1$, eq. 3.14 becomes:

$$\alpha = \frac{j_G}{j_L + j_G} = \beta \quad (3.15)$$

Drift flux model (void quality correlation)

$$u_G = j + (u_G - j) \quad (3.16)$$

Where $(u_G - j)$ is the gas velocity with respect to the mixture center volumetric velocity.

Multiply both sides of eq. 3.16 by α , and yield

$$j_G = \alpha j + \alpha(u_G - j) \quad (3.17)$$

The local velocity and local void fraction usually vary across the channel dimension and perpendicular to the direction of flow, so applying the flow area averaging to all the terms in the equation and obtain,

$$\langle j_G \rangle = \langle \alpha j \rangle + \langle \alpha(u_G - j) \rangle \quad (3.18)$$

The second term on the right hand side is the gas drift flux. Both terms on the right hand side involve the average of products of two quantities and are difficult to handle. Therefore, two additional parameters are defined:

The distribution parameter is:

$$C_0 = \frac{\langle \alpha j \rangle}{\langle \alpha \rangle \langle j \rangle} \quad (3.19)$$

The gas drift velocity is:

$$V_{Gj} = \frac{\langle \alpha(u_G - j) \rangle}{\langle \alpha \rangle} \quad (3.20)$$

Substituting eqs. 3.19 and 3.20 into eq. 3.18, the equation thus becomes:

$$\langle j_G \rangle = C_0 \langle \alpha \rangle \langle j \rangle + \langle \alpha \rangle V_{Gj} \quad (3.21)$$

Alternatively,

$$\frac{j_G}{\alpha} = C_0 j + V_{Gj} \quad (3.22)$$

One way to check the suitability of the drift flux model for the experimental data is through plotting the $\frac{j_G}{\alpha}$ versus j . A linear asymptotic curve fit to the data points

can be performed. The slope of the curve will provide C_0 and ordinate intercept of the curve will be the V_{gj} .

Correlates eqs. 3.12 and 3.22, one dimensional drift flux model can be used to express the relationship between gas velocity and the mixture volumetric flux as:

$$\frac{j_G}{\alpha} = u_G = C_0 j + V_{Gj} \quad (3.23)$$

Eq. 3.23 was also used by Mishima and Hibiki [37] and Mishima et al. [105] to measure the void fraction. Besides, there are quite a number of researchers modifying the distribution parameter C_0 to better correlate their experimental results.

Furthermore, Eq. 3.23 can be expressed as:

$$\alpha \approx \frac{j_G}{C_0 j + V_{Gj}} \quad (3.24)$$

When the drift velocity is low, ($V_{Gj} \ll j$), Eq. 3.24 can be approximated to:

$$\alpha \approx \frac{j_G}{C_0 j} = K\beta \quad (3.25)$$

where $\beta = \frac{j_G}{j}$ is the volumetric quality and $K = \frac{1}{C_0}$

K is the Armands flow parameter when it is equal to 0.8

3.2.6 Two-phase frictional pressure drop

Two-phase pressure gradient could be obtained from the well-known single pressure gradient using the two-phase multiplier proposed by Lockhart and Martinelli R. C. [36]

$$\left(\frac{dp}{dz}\right)_{TP} = \left(\frac{dp}{dz}\right)_L \times \Phi_L^2 \quad (3.26)$$

The two-phase multiplier is defined as:

$$\Phi_L^2 = \frac{\left(\frac{dp}{dz}\right)_{TP}}{\left(\frac{dp}{dz}\right)_L} \quad (3.27)$$

The Martinelli Parameter is defined as:

$$\chi = \sqrt{\frac{\left(\frac{dp}{dz}\right)_L}{\left(\frac{dp}{dz}\right)_G}} \quad (3.28)$$

Where the subscript of TP is two phase, L is liquid phase and G is gas phase. Chisholm [106] introduced a conventional expression to correlate two-phase multiplier and Martinelli parameter. The equation is as follows:

$$\Phi_L^2 = 1 + \frac{C}{\chi} + \frac{1}{\chi^2} \quad (3.29)$$

Alternatively,

$$\Phi_G^2 = 1 + C\chi + \chi^2 \quad (3.30)$$

The values of coefficient C are different for different flow conditions. The coefficient is shown in Table 3.1

Liquid	Gas	C
Turbulent	Turbulent	20
Laminar	Turbulent	12
Turbulent	Laminar	10
Laminar	Laminar	5

TABLE 3.1: Chisholm [106] Coefficient C .

3.3 Heat Transfer

FIG. 3.2: Schematic drawing of glass tube with heat flow

Nusselt number is the ratio of convective heat transfer to the conductive heat transfer under the same condition. The present experiments will be conducted to measure the value of Nusselt number for the laminar single phase heat transfer in the micro-channel.

In order to measure the Nusselt number, temperatures 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 be measured directly so that the inner wall temperature is solved by one dimensional heat conduction equation in cylindrical coordinates:

$$T_{w,i} = T_{w,o} - \frac{q_{loc}'' r_i}{k_{tube}} \ln \frac{r_o}{r_i} \quad (3.31)$$

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

$$T_{bulk,x} = T_{bulk,inlet} + \frac{(q_{loc}'')(2\pi r_i x)}{\dot{m} C_p} \quad (3.32)$$

where q_{loc}'' is the amount of energy absorbed by the water flow:

$$q_{loc}'' = \frac{\dot{m} C_p (T_{bulk,outlet} - T_{bulk,inlet})}{2\pi r_i L_{heat}} \quad (3.33)$$

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

$$h_{loc,x} = \frac{q_{loc}''}{T_{w,i} - T_{bulk,x}} \quad (3.34)$$

The local Nusselt number is:

$$Nu_{loc,x} = \frac{h_{loc,x} d_i}{k_{liquid}} = \frac{h_{loc,x} 2r_i}{k_{liquid}} \quad (3.35)$$

Note that, the equations derived above (Eqn. 3.31 to Eqn. 3.35) are for thermally and hydrodynamically fully developed situation. Since the gas phase has a comparative low heat capacity, the heat transfer will mainly occur in the liquid phase. Therefore, only the water properties is used for Eqn. 3.32 and Eqn. 3.33.

The definitions for the above parameters are:

C_p is heat capacity of the fluid

$h_{loc,x}$ is the local convective heat transfer coefficient at the distance of x away from the heating zone

k_{tube} is the thermal conductivity of the channel

k_{liquid} is the thermal conductivity of the fluid

r_i is the radius of the inner wall

r_o is the radius of the outer wall

$T_{w,i}$ is the channel inner wall temperature

$T_{w,o}$ is the channel outer wall temperature

$T_{bulk,x}$ is the mean temperature of the fluid at the distance of x away from the heating zone

$T_{bulk,inlet}$ is the mean temperature of the fluid before entering the heated channel

$T_{bulk,outlet}$ is the mean temperature of the fluid at the exit of the heated channel

x is the distance away from starting of the heating zone

3.4 CFD Governing Equations and Boundary Conditions

In microscale, surface tension is dominant over the gravitational force. Hence the gravitational force is neglected. Reynolds number is small (typically $Re < 1000$). Therefore the assumptions and simplification for the simulations are listed as following:

- Both phases are immiscible
- The fluid does not undergo phase change or no phase change will occur
- The flow is unsteady, laminar, incompressible and newtonian fluid is assumed
- Non slip boundary condition is applied at the wall
- Constant physical properties
- Heat transfer at the outer wall is uniform
- Constant flow rate for either phase

Ansys Fluent 14.0 has been used for the two-phase flow simulation. The interface is captured by using the volume of fluid (VOF) method along with the Navier-Stokes and energy equation sets for the fluid phases. These equations are as below:

Continuity equation,

$$\nabla \cdot \vec{u} = 0 \quad (3.36)$$

Navier Stokes's equation,

$$\rho_f \left(\frac{\partial \vec{u}}{\partial t} + \vec{u} \cdot \nabla \vec{u} \right) = -\nabla p + \nabla \cdot [\mu(\nabla \vec{u} + \nabla \vec{u}^T)] + \sigma \kappa \vec{n} \delta \quad (3.37)$$

The effect of surface tension is included in the Navier Stoke's equation due to its significant physical important in the microscale.

VOF equation,

$$\frac{\partial q_f}{\partial t} + \vec{u} \cdot \nabla q_f = 0 \quad (3.38)$$

The energy conservation equations for the fluid and solid are taken as following,

$$\rho_f \left[\frac{\partial C_{pf} T}{\partial t} + \vec{u} \cdot \nabla C_{pf} T \right] = \nabla \cdot (k_f \nabla T) \quad (3.39)$$

where subscript f is replaced by G and L for the gas phase and liquid phase, respectively. σ is the surface tension coefficient, κ is local curvature and δ is delta function which is zero everywhere except on the interface. The last term appearing on the right hand side of Eqs. 3.37 is the interfacial force and only takes effect on the liquid/gas interface. The variable q_f in Eqs. 3.38 is the volume fraction of fluid, f can be either gas or water phase. Eqs. 3.37 to Eqs. 3.39 are discretized by finite volume method using second order upwind scheme interpolation. PISO scheme is used to solve coupling of flow and pressure fields.

Uniform velocity profile and constant temperature for both gas and liquid phases are assumed at the inlet. Pressure outlet boundary condition is applied at the exit. No-slip boundary condition and a constant wall heat flux boundary condition are applied at the wall. The same velocity profile as applied at the inlet is used to initialize the flow in the whole domain. The gauge pressure and temperature in the domain are set to 0 Pa and 300 K, respectively.

Generally, VOF method can better conserve mass than the level set method. However the drawback of the VOF method is that, it is not able to provide sharp interface between two phases. The smeared interface may bring extra uncertainties in evaluating the heat transfer performance. Therefore, geometric reconstruction of interface (PLIC VOF) is employed to achieve this sharp interface. This numerical method has proven to be appropriate to the study of mass and heat transfer characteristic related to air-water two-phase flow in microscale [107].

In microscale, buoyant force is expected to be small comparing to the surface tension across the interface. Hence the 3D simulations for the horizontal flows with gravitational force was not performed.

Chapter 4

Experimental Setup and preliminary measurements

DI (De-ionized) water is used as the working fluid for the single liquid phase flow experiment. Nitrogen gas which is insoluble in water is chosen as the gas phase in the two-phase flow experiment. Experiments are carried out at room temperature and standard atmospheric pressure. Circular borosilicate glass tubes with 300 *mm* long and inner diameters of 100, 300, 500 and 900 μm are utilized in the single liquid phase flow experiment.

4.1 Single-phase flow experimental setup

Two different experimental setups are employed under the room temperature and standard atmospheric pressure.

A schematic drawing of first experimental setup is shown in Fig. 4.1. It is a setup designated for low flow rate experiment. DI water is pumped to the test section by the twin plunge pump (HPLC Pump PU714i) with the flow rate in the range of 0.01 to 20.0 *ml/min*. Meanwhile the inlet and outlet pressure of the test section are measured by the digital pressure gauges (DPG1000B) with an accuracy of 0.25%. The volumetric flow rate of the distilled water is further

confirmed by weighting the collected discharged distilled water over a period of time with electronic balance. The details for the measurement of volumetric flow rates are discussed at Section 3.1.2

The pressure drop across the channel is measured by reading the two pressure gauges when steady state is reached. The stability of the flow could be monitored by the digital pressure gauges. Usually after a few minutes, the reading will stop fluctuating and the flow is considered to be steady.

On account of the limitation of the maximum volumetric flow rate of the HPLC pump, a second experimental setup (Fig. 4.2) is implemented to provide higher flow rate which is corresponding to higher Reynolds number in the test channels to observe the transition phenomenon.

The test section in second experimental setup is similar to the setup in first experimental setup. The only difference is that the constant flow rate DI water is supplied by pressurizing the DI water inside the water vessel with the high pressure gas nitrogen cylinder instead of using the twin plunge pump.

FIG. 4.1: Schematic drawing of low flow rate experimental setup.

FIG. 4.2: Schematic drawing of high flow rate experimental setup.

4.1.1 Details of single-phase flow channel inlet design

The reservoir chambers at the inlet and outlet of the test microchannel have $d_{chamber}/d_{channel}$ greater than 16 to ensure the measurement of the static pressure only. The gradual reduction of the cross-section inside the chamber area to the inlet of the test tube shown in Figs. 4.3 and 4.4 is designed to reduce the pressure head loss due to the sudden contraction.

FIG. 4.3: Single-phase flow channel inlet design.

FIG. 4.4: Actual view of single-phase flow channel inlet design.

4.2 Two-phase flow experimental setup

4.2.1 Non-heating experimental setup

The experimental setup presented in Section 4.1 is modified for the two-phase flow experiment by inserting a stainless tube co-axially into the test channels. The tube is to introduce gas nitrogen into the continuous DI water flow (Fig. 4.5).

The transparent borosilicate glasses employed as the test channels are to provide visual of the inner flow conditions when the nitrogen gas is injected into the DI water flow co-axially. Halogen lamp gooseneck light is placed behind the test section for illumination and high speed camera is employed and placed at the entrance of the centre tube which is about 3 *cm* away from the water inlet to collect the images at the resolution of 512 x 64 pixels and 16000 frames per second to over a 10 *mm* of the channel length. The reason to have such a high frame rate is to make sure that the bubble movement is being monitored closely. This is because bubbles are looked alike along the channel and it is impossible to identify the same bubble if the frame rate is low.

Accurate amount of nitrogen gas introduced into the channels through the centre stainless tube is controlled by low mass flow controller (Sierra Smart-Trak model 101). 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 are steady. The flow is confirmed to be steady when the reading on the pressure gauges stops fluctuating.

The water flow rates are further confirmed by weighing the water collected over a period of time, which is normally more than 10 minutes. The high speed camera is placed at the entrance of the centre gas tube which is about 3 *cm* away from the water inlet to minimize the entrance effect.

FIG. 4.5: Schematic drawing of two-phase flow experimental setup.

4.2.1.1 Details of two-phase flow channel inlet design

The schematic drawing and actual view of the two-phase flow channel inlet design are shown in Figs. 4.6 and 4.7, respectively. The reason for choosing this design is because other inlet designs discussed at Section 2.2.4 are hard to control the amount of bubbles and bubble injection frequency to the interested channel and made the investigation to be difficult. The present design is capable of generating repeatable results. As shown in Fig. 5.11, the bubble length and distance between two adjacent bubbles are stable after the flow becomes steady. The disadvantage of this design is that it will increase the pressure drop and difficult to make sure the centre tube was concentric with the glass tube.

FIG. 4.6: The two-phase flow channel inlet design.

FIG. 4.7: Actual view of two-phase flow channel inlet design.

4.2.2 Heat transfer experimental setup

Figs. 4.9 and 4.10 show the schematic drawing of the experiment setup for single and two-phase flow heat transfer respectively. The working fluid is DI (de-ionized) water. Nitrogen gas (N_2) 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 nitrogen gas is injected into the DI water flow co-axially.

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* in which 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 16000 frames per second with a 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 $2\ \mu m$ is applied at the outer wall of the glass tube to act as the heating source (Fig. 4.8). It is a conductive transparent film which could generate heat when applying electricity. In order to prevent the degradation of the ITO coating, $1.1\ \mu m$ 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 [12]. An AC power supply is used to provide joule heating on the ITO coating via two wires connected to the test section. 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 attached on the tube outer wall with Kapton tape which can withstand the temperature of up to $204\ ^\circ C$. The arrangement of the thermocouples are as following sequence where T_1 is placed $10\ mm$ in front of the heating zone and T_2 is positioned $10\ mm$ away from the starting point of the the heating zone. The other thermocouples are having a distance of $20\ mm$ between each other.

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 minutes. The experimental conditions are listed in Tables 4.1 and 4.2.

FIG. 4.8: Schematic drawing of the layer of coating.

FIG. 4.9: Schematic drawing of single-phase heat transfer experimental setup.

FIG. 4.10: Schematic drawing of two-phase heat transfer experimental setup.

j_L [m/s]	j_G [m/s]												
0.08	0.04	0.08	0.13	0.17	0.25	0.30	0.34	0.42					
0.17	0.04	0.08	0.13	0.17	0.25	0.30	0.34	0.42	0.47	0.51	0.55	0.59	0.64
0.25	0.08		0.13	0.17	0.25	0.30	0.34	0.42	0.47	0.51	0.55	0.59	0.64
0.42				0.17	0.25	0.34	0.42		0.51				

TABLE 4.1: Experimental conditions for i.d. 500 μm .

j_L [m/s]	j_G [m/s]							
0.12	0.12	0.18	0.24	0.35	0.59	0.94		
0.24	0.12	0.18	0.24	0.35	0.59	0.94	1.18	
0.35	0.12	0.24		0.35	0.59	0.71	0.94	
0.47	0.12	0.24		0.35	0.59	0.71	0.83	1.41
0.71	0.12						0.71	1.41

TABLE 4.2: Experimental conditions for i.d. 300 μm .

4.3 Preliminary measurements

4.3.1 Validation of the experimental setup

Many experiments were conducted by using pressurizing liquid via high pressure cylinder by previous researchers because the liquid supplied from the water vessel was constant and pulsation generated by mechanical pump could be avoided. Although the high pressure nitrogen cylinder and water vessel system is good enough to be a water supply source, HPLC pump is introduced in the present study due to its easy control of accurate flow rate especially when the required flow rate is low.

Even though the HPLC pump is designed to have low pulsation, the pulsation which may be caused by the mechanical mechanism of HPLC pump is necessary to be checked before the experiments. Consequently, i.d. $500\ \mu\text{m}$ glass tube is utilized to examine the pulsation of the HPLC pump. Single-phase DI water pressure drop measurements are conducted using the experimental setups discussed in Section 4.1. Experimental results using both experimental setups are shown in Fig. 4.11.

The results show that using both experimental setups gives the same trend of pressure drop per unit length at different flow rate and hence verified that the HPLC pump is capable of producing stable flow rate. As shown, either HPLC pump or high pressure nitrogen cylinder with water vessel can be used to supply the DI water in the present study to obtain a larger range of pressure drop measurement. The results as shown in Fig. 4.11 are mainly to show the trend of the pressure drop measurement obtained by two different pumps. The head losses across the channel are not yet accounted for. Therefore, it may explain the departure of the experimental results from the Poiseuille flow theory.

FIG. 4.11: Comparison of the pressure drop measurements using the HPLC pump and N_2 Cylinder.

4.3.2 Measurement of pressure head loss

The pressure losses due to the contraction and expansion at the inlet and outlet of the microchannels are checked.

4.3.2.1 Channel outlet pressure loss

i.d. $300\ \mu\text{m}$ glass tube had been used to examine the outlet condition of the test section. It is found that the results with and without the outlet chamber (exposed directly to atmosphere) give the same results as shown in Fig. 4.12. Therefore the outlet head loss is confirmed to be very small and is negligible in the present study. Again, the departure of the experimental results from the Poiseuille flow theory might be caused by the reason that the pressure head losses are not accounted for.

4.3.2.2 Channel inlet pressure loss

The pressure loss across the test channels are measured at the inlet and outlet chamber of the test section. In order to account for the pressure loss at the inlet

FIG. 4.12: Comparison of outlet loss.

contraction region and hydrodynamic development region, pressure loss measurements are performed on a shorter test channel which was cleaved from the same glass tube. Subtracting the pressure loss across the shorter test channel from the pressure loss across the longer test channel divided by the length difference of these two test channels could give us the pressure loss per unit length of the laminar flow. The pressure losses of laminar fully developed flow can be expressed by the following equation:

$$\left(\frac{\Delta P}{L}\right)_{fully\ developed} = \frac{\Delta P_{long} - \Delta P_{short}}{L_{long} - L_{short}} \quad (4.1)$$

The determination of head losses had performed on i.d. 500, 300 and 100 μm glass tubes. The length of the shorter tube is also long enough to ensure the flow is fully developed and the experimental conditions are listed at the Table 4.3.

According to the experimental conditions in Table 4.3, the pressure drop measurements across four different inner diameter channels are plotted in Fig. 4.13. The open circular symbols are the pressure drop measurements obtained before any losses are accounted for. The closed circular symbols are the pressure drop for the developed flow after accounted for the losses by utilizing Eq. 4.1.

i.d. (μm)	Length (mm)	L/d	$Q(ml/min)$	Re	$u_{mean}(m/s)$
900	300	333	34-237	800-5588	0.9-6.2
500	300	600	4.21-52.7	178-2238	0.36-4.5
500	100	200	8.83-86.8	356-3504	0.68-6.66
300	300	1000	2.315-22.6	175-1712	0.62-6.1
300	200	667	1-20	75.5-1510	0.27-5.37
100	300	3000	0.1-0.556	20-112	0.2-1.07
100	150	1500	0.2-0.75	40-150	0.38-1.44

TABLE 4.3: Experimental conditions for head loss estimation.

The error bars in Fig. 4.13 shows that the experimental error could be up to 15% if the head losses are not carefully accounted for. Furthermore, it shows that the pressure drop is increasing and deviates earlier as the channel diameter becoming smaller. Especially for the case of i.d. 100 μm tube, it deviates earlier at Reynolds number below 100 (Fig. 4.13 (d)).

The sudden contraction at the channel inlet could be the reason for the observed early deviation. This is because after the head losses have been accounted for, the pressure drop per unit length becomes closer to the conventional theory again. The smaller the test tube becomes, the greater the contraction loss. This could be illustrated by the pressure loss results of i.d. 900 μm and i.d. 100 μm channels shown in Fig. 4.13 (a) and 4.13 (d), respectively. i.d. 900 μm channel agrees with the Poiseuille flow theory without accounting for any head losses. However i.d. 100 μm channel agrees well with the conventional theory after accounted for the head losses. This trend also shows that head losses increases with the decreasing of channel size. Researchers who did not consider the pressure head losses are expected to have a great deviation from the conventional theory from the beginning of the pressure loss measurements. Hence, the head losses for the channels i.d. below 900 μm should be carefully measured beforehand.

FIG. 4.13: Pressure losses for channels with inner diameter of (a) $900 \mu m$ (b) $500 \mu m$ (c) $300 \mu m$ (d) $100 \mu m$, respectively.

4.3.3 Investigation on the effect of L/D

The experimental results conducted with different length of the same tubes are shown in Fig. 4.14. The glass tube of i.d. $300\ \mu\text{m}$ with length of $300\ \text{mm}$ and length of $50\ \text{mm}$ are tested to observe the L/D effect to the pressure drop. Though, in general, the difference does not appear to be very significance, it is worthwhile to mention that the pressure drop results show that shorter tube tend to have a higher pressure drop than the longer tube which they are cleaved from. In addition, the gradient of pressure drop for the shorter tube changes at about 1500 Reynolds number (Fig. 4.14). It is an indication that transition flow is occurring which could be the reason that some previously literature had reported much earlier transition when the channels they used were short (28.5 to $88\ \text{mm}$ long tubes were used in the experiments performed by Mala and Li [1]). In this case, the effect of L/D is negligible in the laminar region of the present study.

FIG. 4.14: Investigation on the effect of L/D .

4.4 Uncertainty analysis

4.4.1 Uncertainty of temperature measurement

The uncertainty of temperature measurement could be verified by putting thermocouples into a constant temperature bath to measure the variation of the temperature read by each thermocouple. The measured temperature, averaged measured temperature and experimental error are listed in the Table. 4.4.

Bath temperature $T_{bath} (^{\circ}C)$	Thermocouples ($^{\circ}C$)							
	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8
32	32.11	32.08	31.93	31.69	31.50	31.69	31.85	31.02
40	40.10	40.10	40.00	39.80	39.57	39.74	39.60	40.08
50	50.16	50.22	50.07	49.82	49.57	49.48	49.53	50.22
60	60.18	60.19	60.13	59.88	59.40	59.38	59.00	60.24
70	70.25	70.23	70.10	69.77	69.51	69.49	69.70	70.19

Bath Temperature $T_{bath} (^{\circ}C)$	Average temperature $T_{ave} (^{\circ}C)$	Experimental error $\frac{T_{bath}-T_{ave}}{T_{bath}} \times 100 (\%)$
32	31.73	0.83
40	39.87	0.32
50	49.88	0.23
60	59.80	0.33
70	69.91	0.14

TABLE 4.4: Uncertainty of temperature measurement.

where $T_{ave} = (T_1 + T_2 + T_3 + T_4 + T_5 + T_6 + T_7 + T_8)/8$

4.4.2 Uncertainty of velocity measurement

The uncertainty of the fluid flow velocity could be determined by comparing the flow rate given by the flow controller and volume of fluid collected over time. The equations for the volumetric flow rate measurement have been discussed previously in section 3.1.2.

Velocity is directly proportional to both the cross section of the channel and the volumetric flow rate. Since the uncertainty measurements are conducted with the same channel, the obtained experimental error for the volumetric flow rate could be treated as the experimental error for the velocity measurement.

The collected liquid volumetric flow rate and experimental error are listed in Table. 4.5

Liquid flow rate	HPLC pump $\dot{Q}$ (ml/min)	Collected volumetric flow rate $\dot{Q}_{ave}$ (ml/min)	Experimental error $\frac{\dot{Q}-\dot{Q}_{ave}}{\dot{Q}} \times 100$ (%)
Case 1	1	0.92	8.00
Case 2	4	3.67	8.25
Case 3	8	7.31	8.63

TABLE 4.5: Uncertainty of the liquid volumetric flow rate measurement.

The collected gas volumetric flow rate and experimental error are listed in Table. 4.6

Gas flow rate	Mass flow controller $\dot{Q}$ (ml/min)	Collected volumetric flow rate $\dot{Q}_{ave}$ (ml/min)	Experimental error $\frac{\dot{Q}-\dot{Q}_{ave}}{\dot{Q}} \times 100$ (%)
Case 1	1.0	1.05	5.00
Case 2	1.5	1.52	1.33
Case 3	2.0	2.11	5.50

TABLE 4.6: Uncertainty of the gas volumetric flow rate measurement.

The average experimental errors for the liquid and gas flow rate are about 8.29% and 3.94%, respectively.

4.4.3 Uncertainty of bubble measurement

The uncertainty of the bubble size could be determined by measuring a few bubbles under the same experimental conditions. The bubble diameters could be obtained from the images captured by the high speed camera. Hence, the measured diameter is in the unit of pixel. The measured diameter, average diameter and standard deviation are shown in Table. 4.7

<i>i.d.</i> (μm)	Flow condition	Bubble diameter (pixels)				Standard deviation
		d_{b_1}	d_{b_2}	d_{b_3}	$d_{b_{ave}}$	
300	$j_L = 0.354 \text{ m/s}, j_G = 0.118 \text{ m/s}$	19	18	18	18.33	0.58
300	$j_L = 0.472 \text{ m/s}, j_G = 0.118 \text{ m/s}$	17	18	17	17.33	0.58
500	$j_L = 0.085 \text{ m/s}, j_G = 0.042 \text{ m/s}$	29	30	29	29.33	0.58

TABLE 4.7: Uncertainty of the bubble measurement.

where $d_{b_{ave}} = (d_{b_1} + d_{b_2} + d_{b_3})/3$

Standard deviation is defined as $s = \sqrt{\frac{(b_1 - b_{ave})^2 + (b_2 - b_{ave})^2 + (b_3 - b_{ave})^2}{3-1}}$

4.4.4 Uncertainty of pressure measurement

The uncertainty of the digital pressure gauge with the model DPG1000B used for the pressure measurement is given by the manufacturer to be about $\pm 0.25\%$

Chapter 5

Flow Visualization and Heat Transfer Measurements of two-phase Flow in Microtubes

The focus of the this chapter is to experimentally investigate the optimum conditions by which the thermal performance of a microtube can be enhanced while 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 monitored by flow visualization simultaneously. In addition, the importance of the liquid slug length to the performance will also be discussed. The detail description of the experimental setup has been described previously in Section 4.2.2

5.1 Data reduction

5.1.1 Void Fraction

Void fractions of the two-phase flow patterns are determined by the images captured by the high speed camera. For the bubbly flow, the volume of the bubble can

be approximated as a sphere (Fig. 5.1). For the slug flow, the elongated bubble is divided into a cylinder with two ellipsoidal endcaps (Fig. 5.2). 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:

$$\alpha = \frac{\frac{4}{3}\pi(d_b/2)^3}{L_{uc}A_c} = \frac{\frac{2}{3}d_b^3}{L_{uc}d_c^2} \quad (5.1)$$

For slug flow pattern:

$$\alpha = \frac{L_s A_s + \frac{4}{3}\pi(d_s/2)^2 L_{sc}}{L_{uc} A_c} = \frac{L_s + \frac{4}{3}L_{sc}(\frac{d_s}{d_c})^2}{L_{uc}} \quad (5.2)$$

FIG. 5.1: Image processing method to measure the void fraction for bubbly flow.

5.1.2 Optical Distortion

Optical distortion will magnify the inner diameter of the glass tube. The experiments of Kawahara et al. [44] of the two-phase flow had shown that the inner diameter of the microchannel could be enlarged by almost 50%.

FIG. 5.2: Image processing method to measure the void fraction for slug flow.

The radial magnification factor is determined by inserting a known diameter center tube into the glass test tube shown in Fig. 5.3. It can be expressed by the following equation,

$$M = \frac{d_{image}}{d_{actual} \times N_{pixel}} \quad (5.3)$$

where,

d_{image} is the diameter measured from the image [*pixels*]

d_{actual} is the actual diameter of the object [*mm*]

N_{pixel} is the number of pixels over the length of 1 *mm* in the image. [*pixels/mm*]

FIG. 5.3: Optical magnification in radial direction of the circular channel.

Fig. 5.4 is the image taken from one of the flow visualization experiments with the i.d 500 μm glass tube and center tube of o.d. 262 μm .

FIG. 5.4: Optical magnification in radial direction of the circular channel with i.d. of 500 μm .

Magnification for the center tube is given by,

$$M_{center} = \frac{d_{image}}{d_{actual} \times N_{pixel}} = \frac{11}{0.262 \times 34} = 1.23 \quad (5.4)$$

Magnification for the inner channel wall is given by,

$$M_{channel \text{ wall}} = \frac{d_{image}}{d_{actual} \times N_{pixel}} = \frac{23}{0.5 \times 34} = 1.35 \quad (5.5)$$

Similarly, the magnification factor for the i.d. 300 μm glass tube with the center tube of o.d. 152.4 μm could be estimated with the image shown in Fig. 5.5.

FIG. 5.5: Optical magnification in radial direction of the circular channel with i.d. of 300 μm .

Magnification for the center tube is given by,

$$M_{center} = \frac{d_{image}}{d_{actual} \times N_{pixel}} = \frac{13}{0.1524 \times 67} = 1.27 \quad (5.6)$$

Magnification for the inner channel wall is given by,

$$M_{channel \ wall} = \frac{d_{image}}{d_{actual} \times N_{pixel}} = \frac{26}{0.1524 \times 67} = 1.29 \quad (5.7)$$

Therefore, in order to account for the magnification effect of the glass tube, Eqn. 5.1 and Eqn. 5.2 will be modified into Eqn. 5.8 and Eqn. 5.9, respectively.

$$\alpha = \frac{\frac{2}{3}[d_b(1 - M_{center})]^3}{L_{uc}[d_c(1 - M_{channel \ wall})]^2} \quad (5.8)$$

$$\alpha = \frac{L_s + \frac{4}{3}L_{sc}}{L_{uc}} \left[\frac{d_s(1 - M_{center})}{d_c(1 - M_{channel \ wall})} \right]^2 \quad (5.9)$$

5.1.3 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. 5.6. The temperature measured by the thermocouple (T_1) located at 10 mm before the heating zone shows no significant temperature increment like the other thermocouples located inside the heating zone (Fig. 5.6). Hence the axial conduction along the heating zone can be neglected in the present experiments.

FIG. 5.6: Empty tube heating (i.d. 300 μm).

In addition, axial conduction along the flow direction is checked with the axial conduction number proposed by Maranzana et al. [108]. The axial conduction number, M , reduced 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.

$$M = \frac{q''_{cond}}{q''_{conv}} = 4 \frac{k_w}{k_f} \frac{A_w}{A_f} \frac{1}{RePr} \quad (5.10)$$

where k_w and k_f are the thermal conductivity 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. [108] had proposed the criteria that in order for the axial conduction in the solid wall to be neglected, the value of M should be less than 0.01. The calculated value of M in the present experiments are in the range of 1.51×10^{-5} to 3.03×10^{-6} for Reynolds number in the range of 71 to 354. Therefore the axial conduction can be neglected in the present experiments.

Fig. 5.6 also shows that the instantaneous measured temperatures can fluctuate within $3^\circ C$ 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. 5.8.

5.1.4 Estimation of the heat loss

The heat loss of the test section could be estimated by heating the empty tube [12]. Due to no working fluid is supplied through the tube, the joule heat is transferred to the ambient and could be determined by the following equation.

$$q_{loss}'' = \frac{VI}{\pi d_{out} L_{heat}} \quad (5.11)$$

where V and I are the voltage and the current measured at the copper wires connected to the tube. d_{out} is the tube outer diameter and L_{heat} is the heated length.

A second order polynomial could be used to provide a continuous relationship for q_{loss}'' depending on $T_{w,o} - T_{ambient}$

$$q''_{loss} = (T_{w,o} - T_{ambient})^2 c_1 + (T_{w,o} - T_{ambient}) c_2 + c_3 \quad (5.12)$$

where c_1 , c_2 , and c_3 are the coefficients that fit the experimental data best in a least squares sense.

The measured wall temperature and the corresponding joule heat in Fig 5.6 are shown in Table 5.1.

Measure wall temperature ($^{\circ}C$)	Joule heat VI (W/m^2)
29.04	398.23
46.61	1684.42
71.47	3849.56

TABLE 5.1: Measured wall temperature and the corresponding joule heat in Fig 5.6.

where $T_{ambient}$ is taken as $22^{\circ}C$ and solving the equations simultaneously. The heat loss correlation could be estimated to be,

$$q''_{loss} = 0.334(T_{w,o} - T_{ambient})^2 + 62.45(T_{w,o} - T_{ambient}) - 58.02 \quad (5.13)$$

5.1.5 Investigation on the thermal resistance of the oxidation protective film and the heat source effect of ITO coating

On account of both ITO ($2 \mu m$) and Parylene HT ($1.1 \mu m$) coatings are very thin, $T_{w,o}$ is actually measured on top of the Parylene HT coating and the measured temperature is assumed to be the temperature of the glass tube outer surface temperature.

In order to verify the effect of the Parylene HT coating, small modification of the experiment setup has been made. The heating zone is adjusted to become 100 mm. Hence the thermocouple T_1 is included in the heating zone. Location of

T_2 , T_3 , T_4 and T_5 , are in the zone where ITO and Parylene HT are both applied. However, thermocouple T_1 is at the location where no Parylene HT coating is applied. The comparisons of temperature measurements for thermocouples with and without Parylene HT coating are shown in the Fig. 5.7.

This figure shows that the T_1 has not much difference (to within 1%) from the thermocouples (T_4 and T_5). Hence the effect of the Parylene HT coating can be considered negligible. On the other hand, ITO is a conducting film and hence it will have a higher thermal conductivity than the Parylene HT coating. Therefore, the temperature difference within the ITO coating could be neglected as well.

FIG. 5.7: Thermal resistance of the oxidation protective film and the heat source effect of ITO coating.

5.2 Single-phase flow

5.2.1 Heat Transfer

Single-phase heat transfer experiments are conducted. As shown in Fig. 5.8, the temperature increases linearly with the axial location, as may have been expected. The Nusselt numbers for the single-phase heat transfer in channel with i.d. $300\ \mu m$ are presented in Fig. 5.9. The local Nusselt number is rather uniform along the axial location implying that thermal boundary layer has been fully developed at axial location $x=0.01\ m$. Fig. 5.9 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 [109, 110] and are shown in Fig. 5.10. 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 gradients from the previous reported results, it has similar trend where the Nusselt number increases with the Reynolds number.

FIG. 5.8: Typical single-phase heat transfer temperature profiles along the microtube (i.d. $300\ \mu m$) under the experimental condition of $Re_L = 141$ and $q'' = 80\ 800\ W/m^2$.

FIG. 5.9: Nusselt number for single-phase heat transfer in i.d. 300 μm with water flow rate of 1 ml/min ($Re = 71$), 2 ml/min ($Re = 141$) and 3 ml/min ($Re = 212$).

FIG. 5.10: Comparison of the single-phase Nusselt number with the results reported by other researchers.

5.3 Two-phase flow

5.3.1 Flow visualization

Fig. 5.11 shows typical flow visualization images taken in the present experiments for i.d. $500\ \mu m$ channel. No coalescence of bubbles or breakdown are observed within the range of gas flow rate investigated. The distance between two bubbles remain 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 macroscale pipe flow is not observed in the present study. This observation shows that the surface tension effect dominates over the gravitational effect in the microchannel 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. 5.11 (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. 5.11 (C-G)). Longer gas slugs will be formed when the gas flow rate is further increased (Fig. 5.11 (H)).

Fig. 5.12 shows the flow visualization images for i.d. $300\ \mu m$. Similar to the flow patterns observed in i.d. $500\ \mu m$ tube, no stratified flow occurs and the flow patterns is gradually transforming from bubbly flow (Fig. 5.12 (A-E)) into slug flow (Fig. 5.12 (F-G)) with increasing gas flow rate.

FIG. 5.11: Two-phase flow visualization (i.d. $500\ \mu\text{m}$). Water velocity is $0.08\ \text{m/s}$, gas velocity is (A) $0.04\ \text{m/s}$, (B) $0.08\ \text{m/s}$, (C) $0.13\ \text{m/s}$ (D) $0.17\ \text{m/s}$, (E) $0.25\ \text{m/s}$, (F) $0.30\ \text{m/s}$, (G) $0.34\ \text{m/s}$, (H) $0.42\ \text{m/s}$.

FIG. 5.12: Two-phase flow visualization (i.d. $300\ \mu\text{m}$). Water velocity is $0.47\ \text{m/s}$, gas velocity is (A) $0.12\ \text{m/s}$, (B) $0.24\ \text{m/s}$, (C) $0.35\ \text{m/s}$ (D) $0.47\ \text{m/s}$, (E) $0.59\ \text{m/s}$, (F) $0.71\ \text{m/s}$, (G) $1.41\ \text{m/s}$.

5.3.2 Frictional pressure loss

Fig. 5.13 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.

To better interpret Fig. 5.13, the experimental conditions are listed in Table 5.2. The flow pattern in region (Z) is the bubbly flow pattern, which is observed when the volumetric void fraction (α) 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 5.6 and the governing flow pattern becomes slug flow which falls in region (X) of Fig. 5.13. 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. However, comparing the present bubbly and slug flows data with the churn and annular flows results reported previously by Hayashi et al. [54], the present pressure loss measurements at low gas flow rate is found to be matching well with the pressure measurements conducted by Hayashi et al. [54] where their measurements were performed at a higher gas flow rate using a similar set up.

Region	Flow patterns	χ	α	j_G/j_L
(X)	slug	1.83 to 5.6	0.39 to 0.56	1.76 to 5.25
(Y)	bubbly or slug	5.6 to 6.8	0.34 to 0.39	1.33 to 1.76
(Z)	bubbly	>6.8	<0.34	<1.33

TABLE 5.2: Experimental conditions for the measurements shown in Fig. 5.13.

Fig. 5.13 illustrates that the conventional correlation for macro-sized channel (which is represented by $C = 5$ when both phases are laminar) will underestimate

the pressure drop for the bubbly and slug flows. Conversely, it will overestimate the pressure drop for the churn and annular flows according to the results reported by Hayashi et al. [54]. Furthermore, Fig. 5.13 also shows that the flow patterns may be predicted well by measuring the inlet and outlet pressure drop. This figure would be useful when the channel is not transparent.

Fig. 5.13 also shows that the temperature has significantly reduce 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 [87]. 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 5.3.1). 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.

FIG. 5.13: Comparison of two-phase friction loss with and without heat transfer in i.d. $300 \mu\text{m}$ with water flow rate of 2 ml/min ($j_L = 0.47 \text{ m/s}$) and 3 ml/min ($j_L = 0.71 \text{ m/s}$).

5.3.3 Void fraction

The relation between the volumetric void fraction (α) and the volumetric gas flow ratio (β) of the bubbly and slug flows are presented in Fig. 5.14. The void fraction results are compared with the homogeneous correlation ($\alpha = \beta$), Armand correlation ($\alpha = 0.833\beta$) and the Kawahara correlation [44] ($\alpha = \frac{C_1\beta^{0.5}}{1-C_2\beta^{0.5}}$). Void fraction 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. 5.14 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 microchannel is rather significant.

FIG. 5.14: Void fraction α against volumetric gas flow ratio β .

Figs. 5.15 and 5.16 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, Fig. 5.15 and Fig. 5.16 shows 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. 5.12). 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.

FIG. 5.15: Bubble velocity (i.d. 500 μm).

5.3.4 Heat transfer

5.3.4.1 Pressure loss and wall temperature at different flow patterns

Fig. 5.17 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

FIG. 5.16: Bubble velocity (i.d. 300 μm).

figures show that the wall temperature has become lower when the second phase is introduced into the channel.

Fig. 5.17 (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. 5.17 (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. 5.17 (a) and Fig. 5.17 (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. 5.13). Bubbly flow is found to be more efficient in this case.

FIG. 5.17: Measured outer wall temperature at different flow patterns under the heat flux of 80707 W/m^2 .

5.3.4.2 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. 5.18 and 5.19 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. 5.18 and 5.19 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 (sections A, B and C). It is obvious that at the same water flow rate, the 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 3.3. 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. 5.18 and 5.19. 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. [111] where they concluded that shorter liquid slug would have better heat transfer enhancement than the longer liquid slug.

FIG. 5.18: Temperature measurement for different flow patterns in i.d. $300\ \mu\text{m}$ with water flow rate of $2\ \text{ml/min}$ ($j_L=0.47\ \text{m/s}$).

FIG. 5.19: Temperature measurement for different flow patterns in i.d. $300\ \mu\text{m}$ with water flow rate of $3\ \text{ml/min}$ ($j_L=0.71\ \text{m/s}$).

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 ($Nu_{G,L}$) and the single-phase Nusselt number (Nu_L) which has the same liquid superficial velocity are shown in Fig. 5.20. 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. 5.21 shows the heat transfer enhancement ($\frac{Nu_{G,L}}{Nu_L \Phi_L^2}$), where the pressure drop is included, over the void fraction (α). Two cases, shown previously in Fig. 5.18 and Fig. 5.19, where the water flow rates are fixed at 2 and 3 *ml/min* are presented. Both cases show that the heat transfer performance are better for the bubbly flow and starts to become poorer when the bubbly flow pattern transforms into slug flow pattern as the gas superficial velocity increases. This is because the flow resistance is related to the bubble size and thus Φ_L^2 increases rapidly with the increase of the void fraction. Especially for the case of $j_L = 0.47$ *m/s*, the increase of pressure drop is higher than the increase of heat transfer and therefore the heat transfer is not enhanced. Conversely, for the case of $j_L = 0.71$ *m/s*, the heat transfer is about 2.17 times enhanced over the single-phase flow. Consequently, it is estimated that when j_L

is greater than 0.6 m/s (or $Re_L \geq 180$) and the bubble size which has the diameter close to the channel diameter will give the optimal heat transfer performance where the heat transfer rate is enhanced while the increase of the pressure drop is kept to the minimum.

FIG. 5.20: Comparison of single and two-phase Nusselt number in i.d. 3 ml/min ($j_L = 0.71 \text{ m/s}$) and heat flux of 84321 W/m^2 .

FIG. 5.21: Two-phase heat transfer enhancement at different void fractions.

5.4 Conclusions

Single-phase and two-phase pressure drop under the adiabatic and heat transfer condition are examined. It is found that the pressure drop is significantly decreased by 5-18% under the heat transfer conditions due to the change in fluid viscosity. Single-phase pressure drop results show that the classical friction loss theory is applicable on the channel up to Reynolds number about 1600. The results of two-phase frictional pressure drop (Fig. 5.13) show that Lockhart and Martinelli correlation would underestimate the pressure loss for the bubbly and slug flows.

The present experiment reveals some insight two-phase flow characteristic with the flow visualization. The gas-water two-phase flow does enhance the heat transfer in microchannel with i.d. $300\ \mu m$. The overall heat transfer performance of the bubbly flow exceeding 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 ($Nu_{G,L}$ increases 176%) and the increase of the pressure loss can be kept to the lowest level (less than 27%).

Chapter 6

Bubbly Flow in Microtubes

6.1 Introduction

The experimental results in Chapter 5 shows that bubbly flow has a better overall heat transfer enhancement over the slug flow. Nevertheless, the optimal bubble size for better heat transfer performance has not been fully identified. Furthermore, the liquid velocity profile and bulk fluid mixing mechanism were yet to be discussed. Hence the objectives of this numerical method are to identify the mechanisms of heat transfer enhancement for air water co-axial flow in microtubes and to provide more detail information for its behavior over various operating parameters.

6.2 Simulation Approach

6.2.1 Numerical Modeling

Fig. 6.1 shows the schematic configuration of the two co-axial circular microtubes for the present investigation. The details of the experimental setup have already been described in Section 4.2.2. r_i and w_i are the inner radius and the wall thickness of the center microtube, respectively. Similarly, the radius and the wall thickness of the outer tube are denoted as r_o and w_o . The thickness of

the coating film is indicated as w' . Dispersed gas phase is introduced into the continuous liquid phase through the center microtube. Since both microtubes are axisymmetric, the computation domain can be reduced to half of the tube in the flow direction, as shown in Fig. 6.2.

The effect of the wall thickness of the inner and outer tube have been considered while simplifying the computation model. The influence of the inner tube wall is investigated by comparing the cases of zero and non-zero wall thickness. Ratio of bubble diameter to the distance between the center of two bubbles are compared at different velocity ratios. Fig. 6.3 shows that the error between the zero and non-zero inner wall thickness is within 10% as long as the the inner diameter of the zero and the non-zero wall model are the same. Hence, in Fig. 6.2, R_i is equivalent to r_i and the inner tube wall thickness w_i is reduced to zero.

FIG. 6.1: Schematic of co-axial flow setup.

FIG. 6.2: Computational domain.

In addition, our numerical results of some selected cases show that the wall temperature difference between the non-zero and zero wall model is less than 7%. The overall wall temperature for the non-zero wall model would be slightly higher than the zero wall model due to the axial conduction along the channel wall. Using the axial conduction number, $M = \frac{q_{cond}}{q_{conv}}$ defined by Maranzana et al. [108], to determine the axial conduction effect. For all cases presented in this paper, M is much less than the criteria of 0.01 if the tube is made of glass. Hence the axial conduction effect is neglected. Since our main objective is the overall of the heat transfer

enhancement instead of details in some local area, the zero wall thickness model can evaluate the overall Nusselt number as accurate as the model included wall thickness. Therefore, the geometry of computational domain used in the present simulation consists only of two microtubes with zero wall thickness.

The outer microtube length is constructed sufficiently long (60 times of R_o) to ensure that the flow and bubbles are fully developed. Inlet velocity for the two phases are kept to constant and denoted as u_i and u_o . The physical properties employed for the gas and liquid phase are assumed to be constant, i.e., $\rho_L = 1000 \text{ kg/m}^3$, $\rho_G = 1.225 \text{ kg/m}^3$, $C_{P,L} = 4182 \text{ J/kgK}$, $C_{P,G} = 1006.43 \text{ J/kgK}$, $k_L = 0.6 \text{ W/mK}$, $k_G = 0.024 \text{ W/mK}$ and $\sigma = 0.072 \text{ N/m}$, where the subscript G denotes the gas phase and L denotes the liquid phase.

FIG. 6.3: Ratio of bubble diameter to distance between two bubbles at different velocity ratios with zero and non-zero center tube wall thickness.

6.2.2 Verification of Computed Results

Generally, grid independent results can be achieved by increasing the number of cells. However, the larger the mesh size, the higher the computational cost. To achieve the grid independent results, different sizes of meshes, such as 44×2500 , 58×3333 and 85×5000 , have been tested. It was found that more meshes are required to obtain grid independent results when the weber number increases in the regime of slug flow. The present investigation is focused on in the regime of bubble flow, hence lower mesh size 44×2500 is sufficient for all cases studied in this paper. There are 2500 uniformly spaced meshes in the x-direction and 44 meshes in the y-direction. Among which 5 fine meshes with the ratio of 1.08 constitute boundary layers adjacent to the tube wall and resulting in a total mesh size of about 110000. The criterion of selecting mesh size is that the difference of the computed results (e.g., bubble size and distance between two adjacent bubbles) should be within 3% with significant increase in mesh size. The mesh comparison results are presented in Fig. 6.4. It can be seen that the interface of the bubble becomes sharper when the number of the mesh increases. The difference of average bubble diameter is less than 2%. Additionally, the velocity profiles at the middle of two bubbles located at $x = 3D$ are shown in Fig. 6.5. The velocity is found to be almost identical and the difference of average velocity among different grid size is about 3%.

To save computing cost, the simulation for the comparison of the flow patterns is done for $30R_o$ and only cases for heat transfer analysis will be done for $60R_o$. In the present simulation results, as shown in Fig. 6.6, the similarity of the flow patterns for bubbly flow and the experimental result captured by the high speed camera is within 90%. This matching quality is considered good enough for our analysis, since our interest for this simulation is the overall heat transfer performance of the bubbly flow patterns.

FIG. 6.4: Grid independent results of (a) 44x2500, (b) 58x3333 and (c) 85x5000

FIG. 6.5: Velocity profiles at different grid size

FIG. 6.6: Flow visualization comparison of simulation and experimental result. For i.d. $300\ \mu\text{m}$ (a) $j_L = 0.354\ \text{m/s}$, $j_G = 0.118\ \text{m/s}$ (b) $j_L = 0.472\ \text{m/s}$, $j_G = 0.118\ \text{m/s}$ For i.d. $500\ \mu\text{m}$ (c) $j_L = 0.085\ \text{m/s}$, $j_G = 0.042\ \text{m/s}$ (d) $j_L = 0.255\ \text{m/s}$, $j_G = 0.168\ \text{m/s}$

In addition, Fig. 6.7 shows the relationship of Lockhart-Martinelli parameter χ and the two-phase frictional multiplier Φ_L^2 . Where the χ and Φ_L^2 are defined as $\chi = \sqrt{\frac{\Delta P_L}{\Delta P_G}}$ and $\Phi_L^2 = \frac{\Delta P_{TP}}{\Delta P_L}$, respectively. Two-phase frictional pressure drop of the bubbly flow is found to be well-agreed with the experimental results and following the Lockhart-Martinelli correlation expressed as $\Phi_L^2 = 1 + \frac{C}{\chi} + \frac{1}{\chi^2}$, where $C = 5$ for laminar flow. This good agreement validates that the numerical method is capable of simulating the air water co-axial flow in microtubes.

FIG. 6.7: Two-phase frictional pressure loss.

6.3 Results and Discussions

Several non-dimensional parameters are introduced to describe the flow and heat transfer characteristic for the present investigation. These parameters are defined as following, $Re_L = \frac{j_L D_o}{\nu_L}$, $X = \frac{x}{D_o}$, $Y = \frac{y}{D_o}$, $U = \frac{u}{u_{TP}}$, $u_{TP} = j_L + j_G$. Subscripts L and G represent the liquid and gas phase, respectively. X and Y are non-dimensional length scale. X denotes the ratio of axial distance to the channel diameter and Y denotes the ratio of radial distance to the channel diameter.

6.3.1 Local wall friction and temperature at different flow patterns

Figure 6.8 shows the comparison of the wall temperature and wall friction for the single and two-phase flow along the x-direction. The wall temperature is found to be increasing from the front and peaks appeared at the rear of the gas bubbles for these three cases. This can be attributed to the fact that the gas phase will decrease the mixture heat capacity at the region where the gas phase passing by.

In addition, Fig. 6.8 (a1) and (b1) show that the wall temperatures of the two-phase flow have become much lower than the single-phase flow when the gas phase is introduced into the channel. Conversely, when more and large gas bubbles are occupying the channel (Fig. 6.8 (c1)), the two-phase wall temperature is only slightly lower than the single-phase flow. Hence this suggests that the bubble size and number of the bubbles might affect the heat transfer performance. At a given wall heat flux, lower wall temperature implying that the Nusselt number for the two-phase flow has already become higher than that of the single-phase flow and thus the heat transfer is enhanced.

Figure 6.8 (a2) and (b2) show the relationship of the wall friction at the corresponding axial location. It is observed that the wall friction fluctuates at high values on top of the gas bubbles. One explanation for this wall friction fluctuation could be the differences in the gas and liquid velocities. Gas bubble which moves as a solid object will obstruct the liquid flow. Hence, the friction peaks at the moment the water phase flows through the sides of the bubble. In addition, the more and bigger the bubbles are, the more flow resistance fluctuations are observed (Fig. 6.8 (c2)). However, Poiseuille number, $fRe = 64$ at the liquid phase shows that Poiseuille flow is redeveloping immediately after each bubble.

In summary, flow resistance and wall temperature are found to be related to the bubble size and void fraction. Comparing Fig. 6.8 (a2) and (c2), small and less bubbles are favored due to the relatively low wall friction. It can be estimated that the average highest Poiseuille number is about 200 comparing to the large bubble where the average highest Poiseuille number is above 1500.

FIG. 6.8: Wall temperature (K) and friction (fRe) along the x-direction for $R_o/R_i = 5/1$ (a1) and (a2), $j_G/j_L = 1/30$, $Re_L = 127.3$, $q'' = 20 \text{ W/cm}^2$ (b1) and (b2), $j_G/j_L = 1/12$, $Re_L = 127.3$, $q'' = 20 \text{ W/cm}^2$ (c1) and (c2), $j_G/j_L = 1/2$, $Re_L = 42.4$, $q'' = 10 \text{ W/cm}^2$.

6.3.2 Local Nusselt number at different flow patterns

Fig. 6.9 shows the local Nusselt number of the cases discussed in Fig. 6.8. The corresponding Nusselt number of the single-phase flow, where its flow rate equals to the total flow rate of the two-phase flow, are also plotted for comparison. The parameters represented in Fig. 6.9 are defined as following,

$$Nu_X = \frac{q'' D_o}{k_L(T_w - T_b)} \quad (6.1)$$

and

$$T_b = \frac{\int \rho C_p u T dY}{\int \rho C_p u dY} \quad (6.2)$$

where X is the dimensionless axial location along the flow direction defined earlier. T_b is the classic definition based on single-phase flow and both integrals are evaluated over the entire cross section.

In Fig. 6.9, Nu_X is significantly increased from the rear of the gas phase and peaks at its front for all three cases. These results demonstrate that high heat transfer has been taking place within the thin liquid film which locates between the wall and gas bubbles. However, this finding contradicts the results of wall temperature in Fig. 6.8 where the wall temperature peaks at the location where the gas bubbles appear. This is because the wall temperature should be lower down if high heat transfer occurs. One explanation for this contradiction should be that due to the low heat capacity of the gas phase, the fluid temperature calculated at the thin film is very close to the wall temperature. This causes small temperature difference resulting in a high Nu_X at the region of gas phase.

In this case, the Nusselt number correlation is needed to be modified so as to correctly describe the heat transfer characteristics of multiphase flow. The modified Nusselt number Nu'_X is proposed as,

$$Nu'_X = \frac{q'' D_o}{k_L(T_w - T'_b)} \quad (6.3)$$

and

$$T'_b = T_o + \frac{\int \rho C_p u (T - T_o) dY}{\int \rho_L C_{p,L} u dY} \quad (6.4)$$

The difference between these two Nusselt numbers is the definition of the bulk temperature. These two Nusselt numbers give the same value at the region where only liquid phase exists. However, when the cross section consists of both phases, the bulk temperature T'_b is determined as if the flow was single-phase flow. This modification could accurately assess the capacity of heat transfer for single and multiphase flow having the same total volumetric flow rate.

In order to compare Nu_X and Nu'_X , the relationship of Nu'_X against X are also presented in Fig. 6.9. The trend of Nu'_X is similar to the Nu_X except at locations where bubbles exist. Comparing to Nu_X , these results demonstrate that the modified Nusselt number Nu'_X would be more suitable to define the heat transfer characteristic for the co-axial gas-liquid flow and could be able to give better estimation for multiphase flow. Furthermore, it is noticed that Nu'_X peaks in front of each bubble and redevelops within the liquid phase. Hence, it is clear that substantial changes within the liquid phase had taken place and this could be the main reason that the heat transfer is enhanced. Therefore, further study on the velocity profiles before the bubble and the temperature profiles after the bubble would provide clearer understanding for this mechanism.

6.3.3 Velocity and temperature profiles near a bubble

Axial and radial velocities could provide some ideas on how the liquid phase flow field between two adjacent bubbles develops before it comes into contact with the bubble. Five curves denoted as Y_0 to Y_4 across the tube cross section are plotted from $X_1 = 2$ to $X_1 = 0$ in Fig. 6.10 to describe the evolution of the axial and radial velocities. X and Y are non-dimensional parameters defined previously at Section 6.3. X_1 is introduced to be the distance from a point (e.g. point P) upstream of the bubble up to the bubble surface and Y is the vertical distance along radial direction. The two figures show that both velocities remain

FIG. 6.9: Nusselt number along the x-direction for $R_o/R_i = 5/1$ (a1) and (a2), $j_G/j_L = 1/30$, $Re_L = 127.3$, $q'' = 20 \text{ W/cm}^2$ (b1) and (b2), $j_G/j_L = 1/12$, $Re_L = 127.3$, $q'' = 20 \text{ W/cm}^2$ (c1) and (c2), $j_G/j_L = 1/2$, $Re_L = 42.4$, $q'' = 10 \text{ W/cm}^2$.

undisturbed at about $X > 0.75$. This information suggests that the liquid phase is fully developed before it is in contact with the bubble. However, significant changes in both velocities take place when X approaches zero. The axial velocities converge to the same value and the radial velocities start to build up concurrently. This changes reveal that the presence of the bubble might have obstructed the way of liquid flow and forced the fully developed velocity profile to redevelop by reducing the velocity gradient in the y-direction. For example, the axial velocity along the axis (Y_0) reduced dramatically from about 2.1 to slightly less than 1 while the axial velocity near the wall (Y_4) increases marginally from 0.75 to about 1. Reduction in axial velocity had led to the increase of the radial velocity due to continuity. Hence, this redistribution of axial velocity to radial velocity just in front of the bubble had enhanced the fluid mixing and makes the fluid temperature becomes more uniform.

The higher Nusselt number right after the bubble in Fig. 6.8 could also be understood by analyzing three cross sectional planes after the bubble to capture the temperature profile. Fig. 6.11 (a) shows the locations of the cross sectional planes. X is the dimensionless distance between each cross sectional plane. Fig. 6.11 (b) shows the temperature profiles and the single-phase thermally fully developed temperature profile is also plotted for comparison.

The temperature profile is defined as,

$$\theta = \frac{T - T_w}{T_w - T_b} \quad (6.5)$$

The gradient represents the heat transfer rate. These three temperature profiles are plotted at the same instantaneous time. Comparing with the thermally developed flow, the gradient for the two-phase flow is much steeper (i.e. a higher heat transfer rate) at the near wall region, especially when $Y > 0.4$. Furthermore, the temperature gradient at cross sectional plane 3 is generally lower than the gradient at cross sectional plane 1. It indicates that the heat transfer rate is gradually decreases as X_2 increases. This results is in accordance with the Nu'_x trend

FIG. 6.10: Velocity profile near a bubble ($R_o/R_i = 5/1$, $Re_o = 127.3$, $j_G/j_L = 1/12$).

presented in Fig. 6.9 where the Nusselt number starts to decrease immediately after every gas bubble.

FIG. 6.11: Temperature profile near a bubble ($R_o/R_i = 5/1, Re_o = 127.3, j_G/j_L = 1/12$).

6.3.4 Bubble size and distance between two bubbles

In the present investigation, bubble size and the distance between two adjacent bubbles could be determined by Re_L and Re_G . First, the bubble size increases with the increasing gas flow rate. Consequently, when the bubble grows to the size which is about channel diameter, the distance between two bubbles will decrease.

To further understand the effects of the bubble distance to the heat transfer performance, bulk fluid temperature contours and its wall temperature profile of the two cases discussed previously at Section 6.3.1 are presented in Fig. 6.12. Blue color represents lower temperature while the green and yellow colors represents higher temperature. It can be observed that the temperature of the liquid slugs between bubbles are becoming more uniform as they approaching downstream. This shows that the existence of the bubbles could improve the mixing of the bulk fluid by refreshing the hot liquid layer from the wall to the center of the channel. In addition, the proportion of the green color in case (b1) is comparably higher than the case (a1), which contains only three bubbles, also shows that more bubbles would result in a quicker mixing over a short distance. This explains the reason that the wall temperature of the case (b1) could be lower than the case (a1). However, at the downstream of these two flow patterns, the bulk fluid temperatures turn out to be closer to each other. This shows that the fluid temperature has become rather uniform over a short distance after entering the heating zone. Therefore, this “entrance effect” is estimated to become negligible after 20 inner diameter of the heating length.

Fig. 6.13 shows that Φ_L^2 increases with the increase in bubble size. This trend is especially obvious when $D_b > 0.9$. This is because when $D_b > 0.9$, the growth of the bubble size is comparably slower than the decrease of bubble distance between bubbles and thus the pressure loss increases rapidly. Hence the pressure loss should be taken into account when evaluating the heat transfer performance.

The relationship of bubble size (D_b) to the heat transfer enhancement ($\frac{Nu_{G,L}}{Nu_L \Phi_L^2}$) is plotted in Fig. 6.14. $Nu_{G,L}$ and Nu_L are the Nusselt numbers for the two-phase

FIG. 6.12: Comparison of temperature contour and wall temperature of different flow patterns at $Re_L = 127.3$ and $q'' = 20 \text{ W/cm}^2$.

FIG. 6.13: Two-phase multiplier at different bubble size.

and single-phase flow respectively at the same liquid flow rate. Both two-phase and single-phase Nusselt numbers are defined as following,

$$Nu_{G,L} = \frac{q'' D_o}{k_L(\overline{T_w} - \overline{T_b})} \quad (6.6)$$

and

$$Nu_L = \frac{q'' D_o}{k_L(\overline{T_w} - \overline{T_b})} \quad (6.7)$$

The overbars in Eqs. 6.6 and 6.7 denote the average wall and bulk fluid temperature. Φ_L^2 is the two-phase multiplier which is defined previously at Section 3.2.6. D_b is the ratio of the equivalent bubble diameter to the channel diameter.

FIG. 6.14: Overall heat transfer enhancement at different bubble diameter.

The heat transfer performance for the low Reynolds number $Re_L = 42.4$ is found to be not enhanced due to the comparably high pressure loss caused by the big bubble size. Increase of the Re_L helps the heat transfer performance. For example at $Re_L = 212.2$, the heat transfer performance is about two times higher than the heat transfer performance at $Re_L = 42.4$. Additionally, it can be observed that the heat transfer enhancement becomes rather flat and seem to be reaching the maximum when $Re_L > 127.3$. This could be attributed to the fact that small

bubbles has less impact to the hot liquid flow near the wall and the reduction of bubble quantity in the channel would decrease the refreshing rate of the hot liquid as well. On the other hand, increases the bubble size to increase the impact of the bubble to the fluid near the wall by increasing the Re_G , it can be observed that the $Re_G = 2.18$ does not give better heat transfer performance. The reason is also due to the high pressure loss caused by the big bubble size. Hence comparing both Fig. 6.13 and 6.14, the optimal bubble diameter ratio is estimated to be about 0.8 where $\frac{Nu_{G,L}}{Nu_L \Phi_L^2}$ has reached the maximum.

In addition, the effect of the distance between two bubbles to the heat transfer enhancement is plotted in Fig 6.15. L_b is the ratio of the distance between the center of two bubbles to the channel diameter. This figure shows that the heat transfer is greatly enhanced when L_b is less than 6 and barely increase when L_b is greater than 6. At a constant Re_L , large value of L_b could be obtained by decreasing the Re_G . However, further decrease of Re_G to obtain large value of L_b could not increase the heat transfer performance unlimitedly due to the fact that the size and number of bubble inside the channel will be reduced accordingly and resulted in lower refreshing rate for the liquid phase.

FIG. 6.15: Overall heat transfer enhancement at different bubble distance.

6.4 Conclusions

The success in lowering down the wall temperature shows that the two-phase non-boiling system does enhance the heat transfer in microtubes. Non-boiling two-phase flow utilized the intrusion of the bubbles to enhance the mixing between hot liquid layer adjacent to the wall and the cool liquid core. The temperature profile of the liquid phase tends to become more uniform through flow recirculation. The thermal boundary layer is redeveloping after each bubble passing through. However, the existence of the bubble also results in the increase of the pressure loss by obstructing the water flow. In most cases, the pressure loss is comparably smaller than the gain of thermal performance. The gain in Nusselt number of the two-phase bubbly flow is about 200% higher than single-phase flow while the increase of Φ_L^2 is about 20%. Hence, this is why the heat transfer could be enhanced by the bubbly flow.

The optimal bubble diameter ratio is found to be slightly less than 0.9. Beyond this value, the number of bubbles in the channel will increase and the bubble size will start to grow resulting in rapid increase of the pressure loss. The distance between two bubbles should be kept to a minimum of 6 channel diameters to obtain the optimal heat transfer performance. The overall heat transfer performance is found to be better for the small bubble is due to the lower pressure loss comparing to the big bubbles.

Chapter 7

Alternating Discrete Flow in Microtubes

7.1 Introduction

This chapter presents a new method based on controllable multi-phase flow in a microtube. Two immiscible fluids would flow through a microtube to form the completely separated discrete two-phase flow. This flow pattern is different from the bubbly and slug flow patterns as no dispersed phase is encapsulated by the continuous phase. Our objectives are to assess the potentials of such multi-phase flow in the microchannels and determine the optimal working conditions by analysing the characteristics of heat transfer and pressure loss.

7.2 Simulation Approach

7.2.1 Numerical Modeling

Figs. 7.1 (a) and (b) show the computational domain for the present investigation. Since circular microtubes are axisymmetric, the computation domain can be reduced to contain only half of the tube in the flow direction. r_i is the inner radius

of the microtube. The primary and the secondary phase are injected alternately from the same inlet of the microtube. In Fig. 7.1 (b), the tube wall thickness w_o is introduced in cases when the secondary phase is gas. This is because zero wall thickness model would result in large deviation from reality when the heat capacity of the two phases has a large gap (This issue will be further discussed in Section 7.3.3).

The microtube length is set to be at 30 inner diameter to ensure that most part of the flow train would be fully developed. For cases with long primary phase where length of primary phase, L_1 , is larger than 10 inner diameter, the microtube length is set to be at 60 inner diameter. The wall material is assumed to be aluminum and the wall thickness is set to be 1/4 of the tube radius. Flow rates for the two phases are kept at constant and the inlet velocity is denoted as u_{TP} .

The physical properties employed for the gas and liquid phases are assumed to be constant and listed in Table 7.1.

FIG. 7.1: Schematic of (a) zero wall model (b) thin wall model

Physical properties	water	air	heptane (liquid)
ρ [kg/m^3]	1000	1.225	684
C_p [J/kgK]	4128	1006.4	2219
k [W/mK]	0.6	0.024	0.14
σ [N/m]	0.072		

TABLE 7.1: Physical properties of the gas and liquid phases.

7.2.2 Verification of Computed Results

In Fig. 6.6, the numerical simulation of the two-phase bubbly flow patterns has been compared with the flow visualization images obtained from the experiments. This high matching quality has given us the confidence to carry out the simulation for the alternative discrete flow.

Additionally, the simulated and experimentally measured outer wall temperature of the bubbly flow at $j_1 = 0.472 \text{ m/s}$ and $j_2 = 0.118 \text{ m/s}$ are compared and shown in Fig. 7.2. It can be seen that the computed wall temperature and the experimentally measured wall temperature, under the same flow condition and same wall heat flux, are almost identical (to within $\pm 2^\circ\text{C}$). This shows that the numerical method employed can provide reasonable simulation results for the two phase flow in micro-channel.

The selection of mesh size has been carefully considered based on the best compromise between the numerical accuracy and the computational cost. Different sizes of meshes such as 28×1500 , 53×3000 and 78×4500 have been tested. The differences of average temperature and pressure over a time span are within 3% despite changes in the mesh size. The mesh sizes used for all the cases studied in this paper is 28 meshes in y-direction and 1500 mesh in x-direction for the tube length of 30 inner diameter. For the cases that longer tubes length is required, such as 60 inner diameter long tube, the mesh configuration is 28 meshes in y-direction and 3000 meshes in x-direction. The aspect ratio of the meshes are set to be 1:1 in both x and y-directions. This is to ensure the computed pressure variation induced by the interfaces could be as accurate as possible.

FIG. 7.2: Comparison of the computed and measured outer wall temperature at $j_1 = 0.472 \text{ m/s}$ and $j_2 = 0.118 \text{ m/s}$ under the heat flux of 80707 W/m^2

7.3 Results and Discussion

7.3.1 Flow characteristics

Figs. 7.3 (a) and (b) shows the flow patterns of the bubbly and alternating discrete flow patterns, respectively. For bubbly flow, due to strong surface tension, the dispersed phase tends to form a spherical shape. The distance and shape of the dispersed phase ejected from the co-axial microtubes configuration remains unchanged after the steady state has reached. However, for the alternating discrete flow patterns, the interfaces actually fluctuate as they advance forward. He et al. [112] explained that the motion of the slug was due to the change in the chemical potential. Movement of the interface is shown schematically in Fig. 7.3 (c). Point A at the wall moves to point A' and the border point B moves towards the wall and become point B' as the flow advances forward. In order to show the flow field more clearly, the streamlines of the primary and secondary phase are plotted in Fig. 7.4. Circulations are observed within the two phases and this may be the reason why the liquid layer near the wall could be constantly refreshed by the cooler liquid from the center core.

FIG. 7.3: Flow patterns. (a) Bubbly flow (b) Discrete flow (c) Movement of the interface

FIG. 7.4: Streamline circulations inside the two phases

Fig. 7.5 (a) shows that local fRe_X along the X location for a specific case ($Re_1 = 149.6$, $u_{TP} = 0.5m/s$). f is the Darcy friction factor. It can be observed that large fluctuations occur at around each interface. Fluid impingement on the interface should be the source that attributes to these pressure fluctuations. The mechanism behind these pressure fluctuations could be due to the fluid velocity at the tube center is relatively higher than the fluid velocity near the wall. Whenever one of the phases impinges the interface, the center high velocity flow will be forced to change from its original axial flow direction to the radial flow direction due to the obstruction. Therefore, the velocity gradient imposed by this radial flow around the interfaces is the main reason that causes pressure loss peaks around each interface. The large pressure fluctuation around the interface could contribute a

large percentage to the overall pressure loss. Hence, it can be inferred that more secondary phase would decrease the overall heat enhancement due to the increase of the pressure loss. In addition, Fig. 7.5 (a) also shows that fRe_x is approximately 64 within the primary phase (where $fRe_x = 64$ is the fully developed single-phase flow in a circular channel). Owing to the small volume fraction of the secondary phase, only a small portion at the middle of the secondary phase could be approximately to be 64. Furthermore, the velocity profile at the middle of the primary phase located at $X = 15$ in Fig. 7.5 (a) is plotted in Fig. 7.5 (b). The velocity profile is found to be nearly identical to the velocity profile of the single-phase fully developed flow. This suggests that the flow characteristics in the primary flow could be approximated as the single-phase fully developed flow except the flow near the interfaces.

Primary phase of the discrete flow would be the main carrier that removes the heat. Hence, in order to achieve better thermal performance, the thermal properties (e.g. heat capacity and thermal conductivity) of the primary phase should be better than the secondary phase. Furthermore, to ensure high thermal performance for the discrete flow, the volume fraction of the secondary phase should be kept as low as possible. In present study, only void fraction below 40% is considered. This is because high void fraction would weaken the overall heat transfer enhancement.

The average friction factor $\bar{f}$ against Re_1 is plotted in Fig. 7.6. $\bar{f}$ is defined as $\bar{f} = \frac{4\bar{\tau}}{\frac{1}{2}\rho u_{TP}^2}$, where $\bar{\tau}$ is the average axial wall shear stress of the entire channel length. L_1/D is the ratio of the primary phase length to the channel diameter. Both Figs. 7.6 (a) and (b) show similar trend as the L_1/D changes. For instance, friction factor $\bar{f}$ decreases significantly towards $64/Re$ when L_1/D increases from 3.5 to 14. This is because impingement would occur on each interface. Longer primary phase simply means lesser interface in the channel. The percentage of pressure loss due to the impingement at the interface could be lower down. Hence, relatively longer primary phase is preferred. However, long primary phase might weaken the internal circulation within the primary phase as well. Therefore, the balance between the heat transfer enhancement and the pressure loss should be carefully considered. Referring to Fig. 7.6, in order not to result in large pressure

loss, the L_1/D is suggested to be higher than 7 and 14 for the liquid-gas and liquid-liquid discrete flow, respectively. Consequently, the increase of pressure loss could be controlled within 30%.

FIG. 7.5: (a) fRe_x along the axial location with $Re_1 = 149.6$, $u_{TP} = 0.5 \text{ m/s}$
 (b) Velocity profile at $X = 15$ in Fig. 7.5 (a)

FIG. 7.6: Friction factor against Reynolds number (a) liquid-liquid discrete flow
(b) liquid-gas discrete flow

7.3.2 Minimum requirement to sustain the discrete flow

Dominance of surface tension over the inertial and viscous force is the main contributing factor to maintain the separated discrete flow. Weber number (We) and Capillary number (Ca) may serve as the indicators for the existence of the discrete flow. When We_1 and Ca_2 exceed certain critical values, the discrete flow cannot be sustained. The subscript 1 and 2 represent primary and secondary phase, respectively. Figs. 7.7 (a) and (b) show the schematic process of which the discrete flow breaks down if one of these two critical numbers has been exceeded. It can be observed that when the Weber number (We_1) exceeds the critical number (We_{cr}), the center portion of the front interface of the secondary phase moves forward due to the increase of inertial force. On account of the viscous force, the secondary phase near the wall remains attached. However, as the flow advances forward, secondary phase gradually detaches from the wall and breaks down into bubbly or slug flow patterns. For the case where We_1 is less than the critical value but Ca_2 exceeds the critical Capillary number (Ca_{cr}). The increase in viscous force due to the high Capillary number would resist the motion of the secondary phase and resulted in the crescent shape shown in Fig. 7.7 (b). The crescent shape would eventually break down into bubbly or slug flow and parts of the residue will remain attached on the channel wall.

The reason of choosing We_1 instead of We_2 is due to the consideration that primary phase is the main heat carrier. Critical Weber number ($\rho_1 u_{TP}^2 D / \mu_1$) varies with the properties of the selected fluid. In the present study, only water is chosen to be the primary phase for analysis due to its high heat capacity and low cost. Furthermore, Ca_2 is chosen over Ca_1 as the criteria is attributed to the fact that the rear interface of the secondary phase can be influenced more by the near wall viscous force within the secondary phase. In addition, Fig. 7.7 also shows that inertial force dominates the front interface, while viscous force dominates over the movement of the rear interface.

The critical numbers against Re_1 are plotted in Fig. 7.8 and Fig. 7.9. These critical numbers are the largest value that the discrete flow could achieve before

FIG. 7.7: The break down of the discrete flow beyond the critical number. (a) The discrete flow detaches from the wall due to exceeding critical Weber number. (b) The discrete flow breaks down into several parts due to exceeding critical Capillary number.

it breaks down into other flow patterns. Critical Weber number is determined in the following manner. For a specific Re_1 and L_2/D , the inertia term of We_{cr} is fixed (where $We = \text{inertia force} / \text{interfacial tension force}$), interfacial tension force is tested by gradually decreasing its value until the point that the discrete flow cannot be held. Referring to Fig. 7.8, the rate of increase for the critical Weber number is not as high as the Re_1 . This figure implies that the high inertia force which tends to break the discrete flow from the center could be stabilized with the increase of surface tension.

Similarly, at specific Re_1 and L_2/D , Ca_{cr} is obtained by setting the viscous term to a specific value and gradually decreasing the interfacial tension force until the discrete flow breaks down. The critical Capillary number is found to be not so sensitive to Re_1 and tends to have the decreasing trend with the increase of Re_1 . This result shows that at high Reynolds number, the viscous force has to be lower and surface tension has to be higher to maintain the discrete flow.

In addition, it can be observed that for both critical numbers, We_{cr} and Ca_{cr} increase slowly with the increase of L_2/D . This could be simply due to the interfaces of longer secondary phase could have slightly more spaces to deform before break down.

FIG. 7.8: Critical Weber number

FIG. 7.9: Critical Capillary number

7.3.3 Local wall temperature and Nusselt number at different flow patterns

Heat transfer enhancement could be firstly evaluated by assessing the wall temperature and the calculation of Nusselt number. Several cases, such as bubbly flow and discrete flow with and without the thin wall assumption are presented in Fig. 7.10. Total flow rates of the two-phase flows are kept to be the same as the single-phase fully developed flow.

FIG. 7.10: Local wall temperature of different flow patterns (i.d. 0.3 mm , $Re_1 = 149.6$, $u_{TP} = 0.5\text{ m/s}$, $q'' = 20\text{ W/m}^2$)

Similar to the wall friction, the inner wall temperature also fluctuates around the vicinity of the secondary phase. The inner wall temperature peaks in front of each secondary phase (especially obvious in case 2). It is found that the axial conduction of the tube wall may significantly affect the temperature profiles along the tube. Comparing to the thin wall model (case 4), large temperature fluctuation is observed in the zero-wall model (case 2). This is because the axial conduction

leads to more uniform wall temperature and hence the inner wall temperature of the thin wall model (case 4) is rather uniform.

However, the difference of the average inner wall temperature is about 1.5 K for these two cases. Furthermore, the calculated axial conduction number $M = q''_{cond}/q''_{conv}$ [108] is much higher than 0.01 due to the high thermal conductivity of the aluminium tube wall. Therefore, the inner wall temperature computed from the thin wall model is insensitive to the thermal properties of the secondary phase. For instance, the inner wall temperature has only little difference when the secondary phase is changed from air to heptane (case 3 and case 4). The slightly lower wall temperature of the water-heptane over the water-air discrete flow could be attributed to the relatively higher heat capacity of the heptane over the air phase. Additionally, the minor temperature difference could be due to the lower void fraction for the secondary phase ($\alpha < 15\%$).

For the case, such as water-air discrete flow in the zero wall thickness model. The wall temperature could exceed 1000 K when the heat capacity of the primary phase is multiple times higher than the secondary phase. Therefore such results are not presented and discussed for liquid-gas flow by the zero wall thickness model.

It is noted that, though the inner wall temperature of the bubbly flow is also lower than the single-phase fully developed flow, it is still higher than the discrete flow. However, the inner wall temperature of both bubbly and discrete flows are lower than the single-phase flow. Hence, the Nusselt number for the bubbly and discrete flow may be better than the liquid alone flow.

Fig. 7.11 shows the local Nusselt number of the cases discussed in Fig. 7.10. The corresponding Nusselt number of the single-phase flow, where its flow rate equals to the total flow rate of the two-phase flow, are also plotted for comparison. The parameters represented in Fig. 7.11 are defined as following,

$$Nu'_X = \frac{q'' D_o}{k_1(T_w - T'_b)} \quad (7.1)$$

and

$$T'_b = T_i + \frac{\int \rho C_p u (T - T_i) dY}{\int \rho C_{p,1} u dY} \quad (7.2)$$

instead of the following traditional definition,

$$Nu_X = \frac{q'' D_o}{k(T_w - T_b)} \quad (7.3)$$

and

$$T_b = \frac{\int \rho C_p u T dY}{\int \rho C_p u dY} \quad (7.4)$$

where the subscript X is the dimensionless axial location along the flow direction defined earlier at Section 6.3. T_b is the classic definition based on single-phase flow and both integrals are evaluated over the entire cross section.

FIG. 7.11: Local Nusselt number of case 1 and case 3 in Fig. 7.10

Modification is needed due to the traditional definition is insufficient in evaluating the multi-phase flow when the cross section consist of two phases. For example,

Nu_X is very high at the place where the bubble is located (refer to Fig. 7.11). This contradicts with the local temperature shown in Fig. 7.10, where the wall temperature peaks right above the secondary phase. It is especially obvious for the zero wall model of the water-heptane discrete flow (case 2 in Fig. 7.10). One of the explanations for this conflicting result should be the low heat capacity of the secondary phase that resulting in little temperature difference between the wall and bulk fluid at the place where secondary phase is located.

Among the flows shown in Fig. 7.11, Nusselt numbers of both bubbly and discrete flows are higher than the single-phase flow. This may be attributed to the flow circulation inside the primary fluid of both bubbly and discrete flows. The local Nusselt number of the discrete flow is always higher than the bubbly flow. The Nusselt number of the discrete flow can be maintained at above 15 for the whole heating length. However, the Nusselt number of the bubbly flow is within 8 to 15.

In order to better understand the heat transfer mechanism, the temperature contours of the single-phase and discrete flows are plotted in Fig. 7.12 (a) and (b). Comparison of both figures shows that the temperature distribution of the discrete flow is more uniform than the single-phase flow. More uniform fluid temperature implies lower fluid temperature near the wall and thus the wall temperature could be lowered down. For example, the highest near wall fluid temperature for the single-phase and discrete flows are $328K$ and $320K$, respectively. This could be the main reason that enhance the thermal performance.

FIG. 7.12: Temperature contours of the primary phase. (a) Discrete flow (b) Single-phase flow

7.3.4 Average and overall Nusselt number

Fig. 7.13 shows the average Nusselt number ($\overline{Nu}$) against the Reynolds number (Re_1). $\overline{Nu}$ is defined as $\overline{Nu} = \frac{\bar{h}D}{k}$, where the average heat transfer rate $\bar{h}$ is obtained by averaging the wall temperature and the heat flux over the entire tube length. The bulk temperature is obtained by assuming linear variation of the inlet and outlet fluid temperature. The cases plotted in this figure comprises of both water-gas and water-liquid discrete flows. The ranges for the tested cases are $3 \sim 14$ for L_1/D , $0.5 \sim 2$ for L_2/D and $0.16 \sim 30$ for Pr_2 .

The results show that primary phase with different length would affect the average Nusselt number. The primary phase with shortest length ($L_1/D = 3.5$) gives higher average Nusselt numbers than the flow with longer primary phase length (e.g. $L_1/D = 7.0$ and $L_1/D = 14$). This could be explained with the reason that shorter primary phase length would have stronger circulation than the longer primary phase length [101]. Additionally, Fig. 7.13 also shows that though the primary phase lengths are different, they give similar trend that the Nusselt numbers increase with the increase of Reynolds numbers.

The tested flows comprise of both gas and liquid as the secondary phases. Hence, the thermal properties of the secondary phases would varies significantly. However, the average Nusselt numbers seem to be only affected by the primary phase length. Therefore, it can be deduced that the properties of the secondary phase should have minimal affects to the average Nusselt number.

In order to analyze the effect of the frictional loss to the overall heat transfer enhancement, parameter η is introduced and defined as $\frac{Nu_{TP}}{\Delta P_{TP}} / \frac{Nu_{SP}}{\Delta P_{SP}}$. The subscripts TP and SP denote two phase and single phase, respectively. Nu_{TP} and Nu_{SP} are defined the same way as the $\overline{Nu}$. ΔP_{TP} and ΔP_{SP} are the pressure loss across the tube for two-phase and single-phase flows, respectively.

The computed results plotted in Fig. 7.14 show that the overall heat transfer rate is increasing with the increase of Reynolds number. Generally, η of the water-gas discrete flows are higher than the water-liquid flows. This could be explained by

FIG. 7.13: Average Nusselt number ($\overline{Nu}$) of water-gas and water-liquid discrete flows at various primary phase length

the reason that the water-liquid flow has higher pressure loss than the water-gas flow, as shown in Fig. 7.6.

FIG. 7.14: Overall heat transfer enhancement (η) of water-gas and water-liquid discrete flows at various primary phase length

Regardless of water-gas or water-liquid discrete flows, discrete flows with the primary phase length of $L_1/D = 7$ could achieve highest η among the other flow configurations (e.g. $L_1/D = 3.5$ and $L_1/D = 14$). This is interesting as the $L_1/D = 3.5$ has highest average Nusselt numbers shown in Fig. 7.13. Hence, it suggests that shorter primary phase length which gives better average Nusselt

number may not be the best configuration when the pressure loss is considered together.

Furthermore, it could be observed that the increase of η is comparatively slow than the increase of Re_1 . The two-phase discrete flows could enhance the heat transfer performance in microtubes up to about 6 times higher than the single-phase flow. Following the trend where the Re_1 is reduced to 60. It could be extrapolated that the discrete flows would become ineffective if Re_1 is less than 60 as η would be less than unity.

When the volume fraction of the secondary phase is low, the effect of the secondary heat capacity is not significant to the overall heat transfer enhancement. Conversely for the case where the volume fraction of the secondary phase is increased to 40%, the liquid secondary phase which has higher heat capacity may result in a better heat transfer performance. Two cases, where the volume fraction of the secondary phase are high ($\alpha = 40\%$), are highlighted in Fig. 7.14 for comparison. The results show that the water-liquid flow which has higher overall heat capacity than the water-gas flow still cannot provide better heat transfer performance. Hence, it can be deduced that frictional loss may be a more important factor than the heat capacity when selecting a particular secondary phase.

7.4 Conclusions

The analysis of the alternating discrete flow shows that it is more efficient in heat transfer performance than the co-axial bubbly flow though it requires more complex implementation. Temperature profiles of the discrete flow are greatly influenced by the global circulation within the primary phase. The fluid temperature becomes more uniform and thus lowering the wall temperature as a result of the reduction of the fluid temperature adjacent to the wall.

It is found that the wall temperature of the thin-wall model is not sensitive to the properties of the secondary phase. However, the properties of the secondary phase do affect the pressure loss. For example, secondary phase with higher viscosity would cause higher pressure loss. In addition, to ensure the increase of the pressure loss to be within 30%, the ratio of the primary phase length to the channel diameter is suggested to be higher than 7 and 14 for the liquid-gas and liquid-liquid discrete flow, respectively.

Thermal performance is found to depend on the Reynolds number and phase length of the primary phase as well as the viscosity of the secondary phase. The overall Nusselt number can be up to 6 times higher than the single-phase flow while the increase of pressure loss is less than 30%. In order to achieve better heat transfer performance, the optimal primary phase length is suggested to be about 7 and the Reynolds number should be above 60.

Chapter 8

Conclusions and Future Work

8.1 Conclusions

To investigate heat transfer performance of two-phase flow in microtubes with the inner diameter less than 1 *mm*, experiments with flow visualization and heat transfer measurements were first carried out. Due to the limitation of present experimental setup, the local flow field of the two-phase flow was not able to be investigated at the current state. Therefore, numerical simulations based on the flow visualization results obtained from the experiments were carried out to extend our understanding of the heat transfer mechanism via analyzing the local flow field. Finally, a new discrete flow pattern where two phases were completely separated from each other was numerically investigated. This flow pattern was found to be potentially having higher heat transfer performance than the bubbly flow patterns where the dispersed flow was encapsulated by the continuous main fluid flow. The major findings of the present study are summarized as follows.

The experimental results showed that the pressure loss of the single-phase flow could be well-predicted by the pressure loss correlation used by conventional channel (i.d. > 1 *mm*). To avoid early deviation from the conventional pressure loss correlation, the pressure head loss of channel with i.d. less than 900 μm must be carefully account for.

Bubbly flow pattern, which has relatively lower pressure loss, was found to have better heat transfer performance than the slug flow pattern. Bubbles with diameter which were close to the channel diameter and void fraction which was less than 10% will give optimal heat transfer condition. The heat transfer enhancement of the bubbly flow was about twofold of the single-phase liquid alone flow.

The computational fluid dynamics of the bubbly flow patterns showed that the flow circulations within the liquid slug in between two bubbles was the key mechanism that enhanced the heat transfer performance. The optimal heat transfer conditions could be met if the following two conditions were fulfilled. (1) The ratio of the bubble diameter upon the channel diameter should be slightly less than 0.9. (2) The minimum distance between two bubbles should be no less than 6 channel diameters.

The design of the inlet configuration for the discrete flow patterns may be more complex than co-axial bubbly flow patterns. However, simulation for the discrete flows showed that it was potentially having a better heat transfer performance than the co-axial two-phase bubbly flow. Secondary phase with better heat capacity may not be better than the secondary phase with lower frictional loss. The optimal ratio of the primary phase length to the channel diameter ratio was suggested to be about 7 and the minimum Reynolds number should be at least 60. The heat transfer of the discrete flow could be up to sixfold higher than the single-phase liquid alone flow.

In short, non-boiling two-phase flows have been experimentally and numerically examined to be capable of improving the heat transfer performance in microchannels. The operating conditions for the optimal heat transfer performance have also been discussed and suggested in the present study. The collected results and analysis have provided a good understanding for the heat transfer mechanism in microchannels. On top of that, the present study could also serve as the guideline for the design of the two-phase microchannel heat sink or the two-phase microchannel heat exchanger.

8.2 Future work and suggestions

1) The present experiments is focused on a single microchannel. Experiments of micro heat sink with the similar inlet configuration is recommended to be carried out so as to give a more comprehensive data for the non-boiling two-phase flows in microchannel cooling applications.

2) The present continuous injection method for the co-axial bubbly flow could be modified into controllable injection method so as the distance between to bubbles and the size of the bubbles could be precisely adjusted to the desired conditions to study the optimal heat transfer performance.

3) Since the computation results for discrete flows has shown its potential over the bubbly flow, the experiments for the discrete flow pattern is recommended to be carried out to investigate its feasibility in application.

Publications

- Y.S. Lim, S. C. M. Yu and J. Schluter (2014) Heat transfer characteristics of alternating discrete flow in micro-tubes, International Journal of Heat and Mass Transfer Vol. 74, pp. 333-341.
- Y.S. Lim, S. C. M. Yu and N.T. Nguyen (2013) Flow visualization and heat transfer characteristics of gas-liquid two-phase flow in micro-tube under constant heat flux at wall, International Journal of Heat and Mass Transfer Vol. 56, pp. 350-359.
- Y.S. Lim and S. C. M. Yu (2012) Experimental investigation of two phase flow in micro-sized circular tubes, Proceedings of the ASME 2012 3rd Micro/-Nanoscale Heat & Mass Transfer International Conference MNHMT2012-75054, pp. 835-843.

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