International Journal of Heat and Mass Transfer 74 (2014) 333–341

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Heat transfer characteristics of alternating discrete flow in micro-tubes

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ARTICLE INFO

Article history:
Received 5 July 2013
Received in revised form 7 February 2014
Accepted 14 March 2014
Available online 12 April 2014

Keywords:
Numerical simulation
Two phase flow
Heat transfer
Micro-channel

ABSTRACT

In this paper, the heat transfer performance of the alternating discrete two phase flow in micro-tubes has been numerically investigated. Two immiscible fluids are injected into a micro-tube alternately, generating a plug flow pattern with two completely separated phases. 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 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.

1. Introduction

In recent years, more densely packed miniaturized electronic devices have been integrated into our daily lives. However, one of the major factors that limits the performance of these devices is the heat generated and removal by the electric circuits. To prevent failure, heat has to be dissipated as quickly and efficient as possible. Hence the high surface to volume micro-channels has been receiving a lot of attention. Single phase flow with tube diameter of about 100 μm could easily reach the heat transfer rate in the order of MW/m^2 [1].

Heat removal method, such as flow boiling which utilized the latent heat of the fluid to remove large amount of heat has also been extensively investigated. Despite the high heat transfer rate, the instabilities in terms of the pressure fluctuation [2], reverse flow [3] due to the nucleation of gas phase may cause the implementation of flow boiling into the micro-channel cooling extremely challenging.

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 [4] 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 [5], 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 [6] 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 [7] 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

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2014.03.041
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Nomenclature

Ca	Capillary number
C_p	heat capacity [J/kg K]
D	diameter [m]
$\vec{g}$	gravitational force [vector]
$\bar{h}$	average heat transfer coefficient [W/m^2 K]
j	superficial velocity [m/s]
k	thermal conductivity [W/m K]
L_1	length of the primary phase [m]
L_2	length of the secondary phase [m]
$\bar{n}$	unit normal vector [vector]
q''	heat flux [W/m^2]
q_f	volume fraction of fluid
r_i	inner radius for non-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]
u	velocity [m/s]
$\bar{u}$	velocity vector [m/s]
U_x	dimensionless axial velocity
w_o	thickness of the outer wall
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_{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]
v	kinematic viscosity [m^2/s]
ρ	density [kg/m^3]
σ	interfacial tension coefficient [N/m]

Subscripts

1	primary phase
2	secondary phase
cr	critical
f	fluid
i	inner
s	solid
SP	single phase
TP	two phase

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 (μPIV) method which can be used to measure the velocity field of the fluid flow in micro-channel is introduced [8]. Laser induced fluorescence (LIF) method which is capable of measuring the temperature field of the fluid flow is also adopted for micro-channel experiments [9]. However, to complete the task with the μPIV and LIF methods, the refractive index of the fluid interface has to be carefully matched so as to obtain the distortion free results.

In microscale, due to the strong surface tension, the dispersed phase tends to form bubbly or slug flow. As a result, the reported two phase flow patterns [2] are mostly either bubbly or slug flow, or a combination of both flow patterns. Other flow patterns, such as wavy and annular flows [10] could also be observed when the gas velocity is much higher than the liquid velocity. Apart from the flow patterns mentioned above, very few investigations were reported for the discrete plug flow especially the heat transfer performance of the discrete plug flow. Some available literatures related to the discrete plug flow are the dry-out of the gas slugs observed in the experiments performed by Serizawa et al. [11] and the experiments of local and global dry-out of the channel walls by moving bubbles in square capillary channels conducted by Cubaud and Ho [12].

Non-boiling two phase flow has been experimentally examined by Lim et al. [13], their measurements showed that the introduction of the gas phase into the continuous liquid phase could successfully increase the heat transfer performance in micro-channel. However, the local flow field have not been discussed. More recently, numerical computation method has shown its capability in predicting the shape and obtain the flow field in microscale. Che et al. [14] analyzed the heat transfer rate of a single plug flows through the micro-channel numerically. Their analysis of different plug length ranging from one inner diameter to four inner diameter 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 plug-train, has been investigated.

The objective of this paper is to investigate the overall heat transfer performance of the alternating discrete flow in relation to the Nusselt number and pressure loss.

2. Simulation approach

2.1. Numerical modeling

Fig. 1(a) and (b) show the computational domain for the present investigation. Since circular micro-tubes 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 micro-tube. The primary and the secondary phase are injected alternately from the same inlet of the micro-tube. In Fig. 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 3.3).