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Asymmetric heat transfer in liquid–liquid segmented flow in microchannels

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ABSTRACT

Heat transfer in segmented flow in microchannels can be significantly enhanced by recirculating vortices, due to the presence of interfaces. The processes of heat transfer in segmented flow subjecting to asymmetric boundary conditions are studied. Two types of boundary conditions are considered, asymmetric constant surface temperature and asymmetric constant surface heat flux. The paths of heat flow and the effects of the thermal conductivity, the plug length, and the Peclet number are studied. The results show different features from those at symmetric boundary conditions. The heat transfer process at asymmetric boundary conditions is controlled by both thermal advection and diffusion at the mid-plane of the channel. The coupling effects between the adjacent plugs complicate the process by the heat transfer across plug–plug interfaces.

1. Introduction

Heat generated by microprocessors with ultra large scale integration causes a problem to many modern electronic devices and thus hinder their development. Efficient cooling is crucial for maintaining their desired performance. If heat cannot be removed immediately, microprocessors will become unreliable or be permanently destroyed. High heat flux cooling is also required in applications such as high heat-load optical components, laser diode arrays, and X-ray medical devices. Different methods have been proposed to cope with the high heat flux in heat exchangers, such as spray cooling [1–3], jet impingement [4], heat pipes [5,6], and microchannels [7–9]. Heat transfer in microchannels is enhanced by large heat transfer areas per unit volume. Another advantage of microchannels is the possibility to directly integrate the cooling components into microchips to achieve efficient cooling.

In comparison with single phase microchannel heat exchangers, multiphase microchannel heat exchangers have a much higher efficiency, thanks to the recirculating vortices in multiphase flow. Due to the presence of interfaces, the recirculating vortices can enhance the heat transfer by advecting heated fluid from the wall to the central region of the microchannel, and by supplying fresh fluid from the central region of the microchannel to the wall [10,11]. The flow dynamics [12–15], mass transfer [16,17] in segmented flow has been widely studied experimentally [12,18], numerically, and analytically.

The heat transfer enhancement of multiphase flow in microchannels has been proven experimentally by several research groups. Betz and Attinger [19] performed experiments on a polycarbonate heat sink consisting of an array of seven parallel microchannels, using water and air to form segmented flow. Sharon et al. [20] utilized a copper tube with 2 mm inner diameter as the flow channel, while using water and nitrogen as the two immiscible phases. The temperature was measured using several thermal couples along the tube. Lim et al. [21] used borosilicate circular glass tubes with inner diameters of 300 and 500 μm, and a thin layer of Indium Tin Oxide (ITO) coating on the tubes as the heat source.

To analyze the heat transfer process in plug flow, Muzychka et al. [22,23] proposed a simplified model to consider the whole process as a Graetz problem with steady state heat transfer. Numerical simulations were performed to study multiphase heat and mass transport in microchannels. Some simulations were based on fixed frames of references [24–26], while others were based on frames of references following the droplets/plugs [27,28], which could significantly reduce the simulation time.

Besides the traditional pressure-driven method, another strategy to actuate droplets is to use electrowetting on dielectric (EWOD), which manipulates individual droplets by applying a series of electrical potentials to an array of patterned electrodes [29,30]. This subfield of microfluidics is often termed digital microfluidics. Digitized heat transfer on the basis of EWOD was proposed by Mohseni and Baird [31,32]. In comparison with pressure-driven

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flow in microchannels, the digital approach could avoid high pressure problems, which is encountered in pressure-driven flow in microchannels at high flow rates.

Although many investigations of heat transfer in segmented flow have been reported, many of the observed phenomena remain unclear. The investigation is challenging because heat transfer in segmented flow in microchannels is a complex process and involves many parameters. In the previous studies, we studied the heat transfer process of gas–liquid plug flow subjecting to constant surface temperature in two dimensional (2D) microchannels [10], and the heat transfer process of gas–liquid plug flow subjecting to constant surface heat flux in cylindrical microchannels [11]. The results showed that heat transfer could be significantly enhanced by the recirculating vortices. Here we extended this method to study the heat transfer process of liquid–liquid segmented plug trains moving in microchannel subjecting to asymmetric boundary conditions, where a *plug train* refers to a series of N consecutive different plugs which repeat periodically in microchannels. With asymmetric boundary conditions, one wall of the microchannel is adiabatic, and the other wall is maintained at a constant surface temperature or a constant surface heat flux, as shown in Fig. 1. When the flow in microchannel heat sinks is in the flow regimes of plug trains, the heat transfer benefits from the recirculating vortices, and is further enhanced by the interaction of both phases. In addition, asymmetric heat transfer problems are often encountered when only one side of the heat sink is in contact with heat sources.

For most microchannels, the typical widths are about several tens to hundreds of micrometers. At this scale, the continuity of mass, momentum, and energy is still valid for the liquid in the microchannels [33,34], and is used in this investigation. This paper is organized as follows. The numerical methods are described in Section 2. The results are presented and discussed in Section 3. The process of heat transfer is analyzed, the asymmetric features of heat transfer are discussed, and the effect of thermal conductivity is studied.

2. Methods

2.1. Flow field in plug trains moving in microchannels

To study the flow field in plug trains, a moving frame of reference was built on each plug following the plug train, as shown in Fig. 2. The moving frames of reference were used for the flow field and the heat transfer. To reduce the computation time, a theoretical flow field for plug trains [35] was employed and

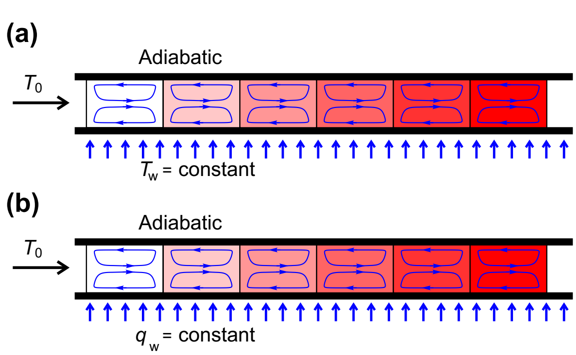

Fig. 1. Schematic diagrams of asymmetric heat transfer in symmetric segmented flow in microchannels: (a) Asymmetric constant surface temperature boundary condition (abbreviated to TA): the bottom wall is maintained at a constant surface temperature while the top wall is adiabatic; (b) Asymmetric constant surface heat flux boundary condition (abbreviated to QA): the bottom wall is maintained at a constant surface heat flux while the top wall is adiabatic.

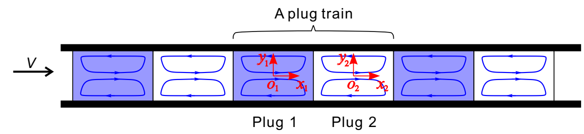

Fig. 2. Schematic diagram of a typical plug train which consists of two immiscible liquid plugs of different fluid properties.

substituted into the simulation of heat transfer process, which has the form of

$$\begin{aligned}\hat{u}_{x,i} &= \sum_{m=1}^{\infty}\left[p_m^{o'}(\hat{y}_i)X_m^o + p_m^{e'}(\hat{y}_i)X_m^e\right]\frac{(-1)^m}{\alpha_m}\cos(\alpha_m\hat{x}_i)\\ &\quad + \widehat{L}_i\sum_{l=1}^{\infty}(-1)^l\left[q_l^e(\hat{x}_i)Y_l^e + q_l^o(\hat{x}_i)Y_l^o\right]\cos(\beta_l\hat{y}_i) \qquad (1)\end{aligned}$$

$$\begin{aligned}\hat{u}_{y,i} &= \sum_{m=1}^{\infty}\left[p_m^o(\hat{y}_i)X_m^o + p_m^e(\hat{y}_i)X_m^e\right](-1)^m\sin(\alpha_m\hat{x}_i)\\ &\quad - \widehat{L}_i\sum_{l=1}^{\infty}\frac{(-1)^l}{\beta_l}\left[q_l^{e'}(\hat{x}_i)Y_l^e + q_l^{o'}(\hat{x}_i)Y_l^o\right]\sin(\beta_l\hat{y}_i) \qquad (2)\end{aligned}$$

where $\hat{u}_x \equiv u_x/V$, $\hat{u}_y \equiv u_y/V$, $\hat{x} \equiv x/h$, $\hat{y} \equiv y/h$, V is the speed of the plug train, $h \equiv w/2$ is the half width of the microchannel, and i is the index of the plug unit in the plug train. Simplification about the film has been made to theoretically study the plug train flow. It is valid when the liquid film is thin and its effect on the flow field and heat transfer is negligible. The thickness of the liquid film is mainly determined by the force balance between the viscous force and the surface tension force, which can be represented by the capillary number. At a low capillary number, the surface tension force is strong. The strong surface tension force can push the liquid away from the liquid film and results in a thin liquid film. As the capillary number increases, the effect of surface tension decreases while the effect of the viscous force increases. Then the thickness of the liquid film increases. A typical flow speed 0.1 m/s of water corresponds to a capillary number of 1.3×10^{-3}, which leads to a film thickness of 1.5% of the radius of capillary tubes, based on the correlation in [36]. The details of the coefficients in Eqs. (1) and (2) and the method to obtain the theoretical solution are explained in [35]. In this paper, we only consider plug trains consisting of two plugs, as shown in Fig. 2. Plug trains with more plugs could be studied similarly.

2.2. Simulation of heat transfer

The governing equation for heat transfer in plug trains is expressed as

$$\frac{\partial(\rho c_p T)}{\partial t} + \frac{\partial(\rho c_p u_x T)}{\partial x} + \frac{\partial(\rho c_p u_y T)}{\partial y} = \frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) \qquad (3)$$

where ρ, c_p, k are, respectively, the density, the specific heat capacity, and the thermal conductivity of the fluid. In Plug 1, $\rho = \rho_1$, $c_p = c_{p1}$, and $k = k_1$, while in Plug 2, $\rho = \rho_2$, $c_p = c_{p2}$, and $k = k_2$.

Using the properties of Plug 1 as the characteristic values, Eq. (3) could be non-dimensionalized to

$$\begin{aligned}&\frac{\partial\left(\widehat{\rho c_p}\widehat{T}\right)}{\partial\hat{t}} + \frac{\partial\left(\widehat{\rho c_p}\hat{u}_x\widehat{T}\right)}{\partial\hat{x}} + \frac{\partial\left(\widehat{\rho c_p}\hat{u}_y\widehat{T}\right)}{\partial\hat{y}}\\ &\quad = \frac{1}{Pe}\left[\frac{\partial}{\partial\hat{x}}\left(\hat{k}\frac{\partial\widehat{T}}{\partial\hat{x}}\right) + \frac{\partial}{\partial\hat{y}}\left(\hat{k}\frac{\partial\widehat{T}}{\partial\hat{y}}\right)\right] \qquad (4)\end{aligned}$$

where $\hat{t} \equiv t/\tau$, $\tau \equiv h/V$, $\widehat{T} \equiv (T - T_0)/T_c$, $\widehat{\rho c_p} \equiv \rho c_p/(\rho_1 c_{p1})$, $\hat{k} \equiv k/k_1$, $Pe \equiv hV/\alpha_1$, $\alpha_1 \equiv k_1/(\rho_1 c_{p1})$. τ is the characteristic time, ρc_p is the volumetric heat capacity, and Pe is the Peclet number. With this