International Communications in Heat and Mass Transfer 49 (2013) 36–40

Contents lists available at ScienceDirect

International Communications in Heat and Mass Transfer

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

Simultaneous integration, control and enhancement of both fluid flow and heat transfer in small scale heat exchangers: A numerical study ☆

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ARTICLE INFO

Available online 29 October 2013

Keywords:
Heat transfer enhancement
Dynamic deformation
Wall morphing
Narrow channel

ABSTRACT

Compactness, efficiency and control of heat exchangers are of great interest in many processes. A technological breakthrough must be achieved to go further in their ability to respond to needs. A new concept of heat exchanger is proposed. It consists in dynamically deforming at least one of the walls of a low hydraulic diameter channel. Heat transfer and mass flow rate enhancements are investigated in single-phase flow. When the deformation is a progressive wave with a relative amplitude of 98.5% and frequency of 50 Hz, it generates a flow having a mass velocity of up to 510 $kg \cdot m^{-2} \cdot s^{-1}$. Although the Reynolds number is low the heat transfer coefficient is enhanced by up to 450% compared to a straight channel.

1. Introduction

The increasing demands of new and effective mixing and heat transfer technologies for various industrial fields such as engineering, chemicals, pharmaceuticals, biochemistry and chemical analysis, associated with the increasing need to control highly exothermic or explosive chemical reactions, have contributed to the development of new and effective technologies involving low diameter channels. It is nowadays common to meet chemical or thermal systems whose channels have sub-millimetric hydraulic diameters. If these systems can increase the compactness, one major drawback is the difficulty to disturb the boundary layers as flow is mostly laminar in such small devices. Consequently, the net gain in terms of mass or heat transfer is due to the increase in the exchange surface as the mass and heat transfer coefficients are usually reduced compared to classical systems.

Moreover such devices lead usually to very high-pressure drops. Indeed, to address the problem of mass and heat transfer coefficients, high velocities of the fluid must be achieved, leading to high pressure losses and consequently to large mechanical power consumed by the pump which is generally located further more or less far upstream the heat exchanger. Thus, the flow distribution in the heat exchanger may be difficult to control and additional pressure losses are generated.

Besides the performance issue, the operating conditions of heat exchangers and chemical reactors are generally constrained, since the mass and heat transfer coefficients are closely related to the fluid flow rate. Integrating the pump to these systems appears thus as a solution to the architectural optimization of chemical or thermal process. One possible way to simultaneously disturb the boundary layers and to integrate the pumping function within the heat exchanger is to generate a dynamic deformation of the channel's wall. Indeed, using a progressive wave deformation produces the motion of the fluid (peristaltic pump) and the disturbance of the boundary layers.

The literature lacks investigation of the effects of dynamic deformations on flow and heat transfer inside channels. Several authors have considered flow inside squeezed thin films like Langlois [1]. However, only a few of them have also analyzed heat transfer such as Hamza [2], Bhattacharyya et al. [3] or Debbaut [4] but, in these works, the squeezing was not of oscillatory type. Nakamura et al. [5] investigated numerically the influence of the wall oscillation on the heat transfer characteristics in a two dimensional channel. Khaled and Vafai [6] considered flow and heat transfer inside incompressible oscillatory squeezed thin films. Kumar et al. [7] recently studied heat transfer inside circular millimetric tubes with static and moving sinusoidal corrugated walls. Numerical analyses were performed to study the effect of spatial wavelengths ($\lambda = 1/2, 2/3, 1, 2$ mm), Reynolds number (1–120) and amplitude (1–20% of tube diameter, D_0) on heat transfer and pressure drops. The heat transfer coefficient for the moving wavy walls had a higher value for all frequencies compared to the static wall case (up to 35–70%). A sharp decrease in pressure drop (by a factor of 1.2 to 5) was also obtained at high amplitudes. Mainly, heat transfer and pressure drop values apparently changed erratically with wall frequency. No general trend of heat transfer and pressure drop values in respect to operating parameters was found.

Several authors performed studies about the effect of heat transfer on peristaltic flow such as Ali et al. [8], Nadeem and Safia [9], Srinivas

☆ Communicated by J. Taine and A. Soufiani.

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0735-1933/$ – see front matter
http://dx.doi.org/10.1016/j.icheatmasstransfer.2013.10.004

et al. [10,11], Hina et al. [12], Al-Amiri and Khanafer [13] and Hayat et al. [14].

According to these studies, the dynamic deformation of a wall efficiently enhances heat transfer. Additionally, if the amplitude of the deformation is high enough, consequent peristaltic pumping may be obtained. It therefore appears possible to realize an interesting multi-functional device using such a deformable system. A virtual prototype constituted of a single channel of small hydraulic diameter with one of the walls dynamically deformed in a large range of amplitude and frequency has then been studied as described in the following sections.

2. Virtual prototype

A virtual prototype is developed in order to study fluid flow and heat transfer behavior in such a device and gain a better understanding of operating parameters' influence on heat transfer and pumping performances. A simple system is studied. It is constituted of two parallel plates placed 1 mm apart. The bottom one is fixed and is heated at a constant heat flux. The top one is adiabatic and dynamically deformed by a progressive sinusoidal wave (Fig. 1). The influence of the movement parameters i.e. frequency (f), amplitude (A_0) and wavelength (λ) is analyzed. The calculations are performed using the commercial software StarCCM+ (CD-Adapco). Constant fluid properties (water at 20 °C) are assumed and the flow is considered three-dimensional and laminar. The transient conjugate flow and heat transfer problems are solved simultaneously using the following set of equations. The continuity equation is written as follow, considering the fluid is non-compressible (constant density):

$$div\left(\vec{v}\right) = 0.$$

The momentum equation is:

$$\rho\frac{\partial\vec{v}}{\partial t} + \rho\left(\vec{v}.\overrightarrow{grad}\right)\vec{v} = \rho\vec{g} - \overrightarrow{grad}(p) + \mu\triangle\vec{v}.$$

The energy equation is written as follow, considering the dissipation source term is negligible and considering that the physical properties of the fluid are constant and uniform:

$$\rho c_p\frac{\partial T}{\partial t} + \rho v\frac{\partial v}{\partial t} + \rho c_p\overrightarrow{grad}(T).\vec{v} + \frac{\rho}{2}\overrightarrow{grad}\left(v^2\right).\vec{v} = k\triangle T + \frac{\partial p}{\partial t}.$$

In these equations, μ is the dynamic viscosity and k is the fluid thermal conductivity.

Fig. 1 presents the studied geometry as well as the imposed boundary conditions. At the entrance, uniform arbitrary values of the pressure and the temperature are imposed ($p(x=0, y, z) = p_{in}$ and $T(x = 0\,\text{cm}, y, z) = T_{in}$, respectively).

The imposed exit boundary condition is a uniform pressure ($p(x = 10\,\text{cm}, y, z) = p_{out} = p_{in} - 20\,\text{Pa}$).

The shape of the upper (non-elastic) wall is imposed by the following equation:

$$Z(x, y, t) = \delta + A_0\sin(2\pi(ft - Kx)).$$

A no-slip condition is imposed to this boundary:

$$\vec{v}(x, y, Z(x, y, t)) = \vec{v}_{wall}(x, y, Z(x, y, t)).$$

Remark: the amplitude of the deformation is weak compared to the wavelength, $\vec{v}_{wall}(x, y, Z(x, y, t)) \approx \frac{\partial Z}{\partial t}\vec{z}$. The upper wall is considered adiabatic. Thus the following equation is imposed:

$$\overrightarrow{grad}(T).\vec{n} = 0.$$

The bottom wall is the heated wall. Thus a heat flux is imposed:

$$\lambda\overrightarrow{grad}(T).\vec{n} = 30,000\ \text{W}\cdot\text{m}^{-2}.$$

Furthermore, the no-slip condition is imposed:

$$\vec{v}(x, y, 0) = \vec{v}_{wall}(x, y, 0) = \vec{0}.$$

The numerical resolution used was a segregated approach with implicit second order temporal discretization. The time step chosen is small enough to allow an accurate description of the wall movement: typically 20 up to 30 time steps per period. We used a structured anisotropic mesh that is dynamically deformed in order to obtain the desired shape and movement of the top wall (Fig. 2). Mesh convergence was studied. A structured mesh with 10 up to 20 cells along the height of the channel, 50 up to 200 cells along the main flow axis and 5 up to 20 cells along the transverse direction has been tested. A mesh of $20 \times 10 \times 200$ cells is found to be sufficient to capture both flow and temperature patterns.

The following procedure is chosen to manage the mesh deformations: we first mesh a parallelepiped channel whose height is equal to the smallest height in the deformed configuration. Then this mesh is dilated to the channel height and, finally, we superimpose the displacement of the surface corresponding to the chosen wave. Using this procedure allows not only stretching the initial mesh but also obtaining a fair control of cell size at all times. The wall motion redistributes mesh vertices in response to the movement of control points. The latter and their associated displacements are used by the mesh morpher procedure included in StarCCM+ to generate an interpolation field throughout the region which is then used to displace the actual vertices of the mesh. Each control point has an associated distance vector which specifies the displacement of the point within a single time-step. More details on the description and validation of such a numerical scheme can be found in [15].

In order to observe the fully developed regime, the number of wavelengths must be sufficiently high: ten wavelengths are imposed along the length of the channel (10 cm), corresponding to $K = 100\,\text{m}^{-1}$. The

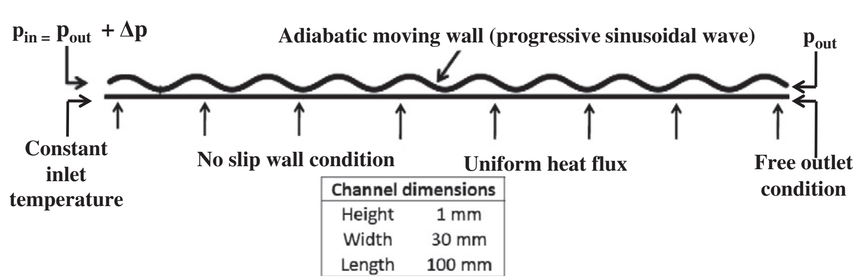

Fig. 1. Geometry of the model: the bottom plate is uniformly heated, the upper plate is adiabatic and dynamically deformed. A constant pressure difference is imposed between the inlet and outlet sections.