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Research Paper

Numerical study of laminar flow and heat transfer in microchannel heat sink with offset ribs on sidewalls

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HIGHLIGHTS

- Three-dimensional model for laminar flow and heat transfer characteristics.
- Heat transfer performance for microchannel heat sink with offset ribs on sidewalls.
- Nusselt number, friction factor and performance evaluation criteria have been shown.

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ABSTRACT

A numerical investigation has been carried out to examine the characteristics of laminar flow and heat transfer in microchannel heat sink with offset ribs on sidewalls. The three-dimensional equations considering entrance effect, conjugate heat transfer, viscous heating and temperature-dependent properties are solved for the fluid flow and heat transfer in the microchannel heat sink. Five different shapes of offset ribs are designed, including rectangular, backward triangular, isosceles triangular, forward triangular and semicircular. Results show that the offset ribs result in significant heat transfer enhancement and higher pressure drop. Depending on the different offset ribs and Reynolds number ($190 \le Re \le 838$) studied in the present work, Nusselt number and friction factor for the microchannel heat sink with offset ribs are 1.42–1.95 and 1.93–4.57 times higher than those for the smooth one, leading to performance evaluation criteria of 1.02–1.48. Further, as a consequence of significant pressure drop, the microchannel heat sink with offset ribs gradually loses its advantage as an effective heat transfer enhancement method at higher Reynolds number.

1. Introduction

The pioneering work by Tuckerman and Pease [1] opened the door for further investigations in the use of microchannels for high heat flux dissipation devices. Since then, much effort has been dedicated to the capabilities of microchannel heat sink to remove heat generated by electronic chips. Steinke and Kandlikar [2] studied the friction factor for single-phase liquids in microchannels and performed an in-depth comparison of experimental data to identify the discrepancies. Rosa and Karayiannis [3] reviewed the experimental and numerical results for microscale single-phase heat transfer and concluded that heat transfer in microchannels could be satisfactorily described by theory and correlations for macroscale, but indicated that the scaling effects should be accounted for, including entrance effect, conjugate heat transfer, viscous heating, temperature-dependent properties, etc. Adham et al. [4] investigated the heat transfer and hydrodynamic performance of microchannel heat sinks and reviewed the methodologies used to analyze and optimize the overall performance of microchannel systems with regard to channel geometry, flow conditions and coolant used.

More recently, with increasing and ongoing emphasis on size reduction and strict temperature limitation in microscale thermal systems, microchannel heat sink with passive microstructures is considered to be efficient means to meet the demand. Xu et al. [5,6] experimentally and numerically studied heat transfer with laminar flow in a microchannel heat sink comprising parallel longitudinal microchannels with several transverse microchambers. They found that the heat sink could significantly reduce the pressure drop and enhance heat transfer, due to the shortened effective flow length. Chai et al. [7] studied the pressure drop and heat transfer characteristics of interrupted microchannel heat sinks with rectangular ribs in transverse chambers. Based on performance evaluation criteria, they obtained the optimal dimensions and locations of the

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rectangular ribs. Cheng [8] numerically simulated heat transfer in a stacked two-layer microchannel heat sink with passive microstructures. They found that the stacked microchannel heat sink had better performance than the smooth one. Chen et al. [9] numerically analyzed laminar flow and heat transfer in microchannel heat sink with rough surfaces and found that the heat transfer coefficient increased almost linearly with Reynolds number and was larger than the smooth one. Sui et al. [10,11] and Mohammed et al. [12] experimentally and numerically studied laminar flow and heat transfer in wavy microchannel heat sink and found that it had better heat transfer performance than the smooth one. Ghaedamini et al. [13] numerically studied the developing forced convection in converging–diverging microchannel heat sink. Based on the performance factor introduced, the superiority of converging–diverging design showed itself at higher Reynolds number for which higher performance of up to 20% was observed. Chai et al. [14] experimentally and numerically studied the heat transfer performance of microchannel heat sink with periodic expansion-constriction cross-sections. They found that the maximum of performance evaluation criteria was about 1.8 at Reynolds number ranged from 147 to 752. Danish et al. [15] and Xia et al. [16,17] analyzed the effect of geometric parameters on pressure drop and heat transfer in the microchannel heat sink with grooved structures or reentrant cavities, and obtained the optimal geometric parameters based upon performance evaluation criteria. Foong et al. [18] and Liu et al. [19] experimentally and numerically investigated the fluid flow and heat transfer characteristics in microchannel heat sink with longitudinal vortex generators (LVGs) and found that the microchannel heat sink with LVGs could enhance heat transfer but have larger pressure drop, compared with the smooth one. Promvonge et al. [20] carried out a numerical investigation to examine laminar flow and heat transfer characteristics in a three-dimensional isothermal wall square channel with 45°–angled baffles. They found that the channel with 45°–angled baffles showed the best performance evaluation criteria of 2.6 at Reynolds number ranged from 100 to 1000.

Based on the passive heat transfer enhancement methods mentioned above, the offset ribs are mounted on the two opposite channel sidewalls for microchannel heat sink with the purposes as follows. One is to develop the converging–diverging or wavy microchannels, which have played an important role in fluid mixing and heat transfer as indicated by Ghaedamini et al. [13] and Sui et al. [10,11]. Two is to work as micromixers or baffles, which can enhance the mixing of cold and hot fluids and the heat transfer as indicated by Cheng [8] and Promvonge et al. [20]. The last is to have the advantage of simple structure, easy manufacturing by micro machining method, low cost, and higher reliability than the active measures. In the present investigation, numerical simulation is conducted for laminar flow and heat transfer in the microchannel heat sink with offset ribs on sidewalls. Five different shapes of offset ribs are designed, including rectangular, backward triangular, isosceles triangular, forward triangular and semicircular. Since the offset ribs may play a fundamental role in establishing flow structure, and therefore heat transfer and pressure drop performances, the present work focuses on these characteristics and the effects of the different offset ribs.

2. Computational method

2.1. Conservation equations

A three-dimensional, incompressible, steady state laminar flow model is designed. As indicated by Zhang et al. [21], the conjugate effect of wall conduction and fluid axial conduction can be important for simultaneously developing laminar flow and heat transfer in microchannels, thus the thermal conduction along flow direction and viscous dissipation are considered in this model. The governing equations in the Cartesian tensor form are:

Continuity equation

$$\frac{\partial}{\partial x_i}(\rho_f u_i) = 0 \tag{1}$$

Momentum equation

$$\frac{\partial}{\partial x_i}(\rho_f u_i u_j) = -\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_i}\left[\mu_f\left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j}\right)\right] \tag{2}$$

Energy equation

$$\frac{\partial}{\partial x_i}(\rho_f u_i c_{pf} T) = \frac{\partial}{\partial x_i}\left(k_f \frac{\partial T}{\partial x_i}\right) + \mu_f\left[2\left(\frac{\partial u_i}{\partial x_i}\right)^2 + \left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j}\right)^2\right] \tag{3}$$

For the solid $\frac{\partial}{\partial x_i}\left(k_s \frac{\partial T}{\partial x_i}\right) = 0$ (4)

where ρ is density, μ is dynamic viscosity, c_p is specific heat capacity, k is thermal conductivity, x_1, x_2 and x_3 are x, y and z coordinates, respectively, as shown in Fig. 1. Subscripts f and s refer to fluid and solid, respectively.

2.2. Computational domain and boundary conditions

Figure 1 illustrates the computational domain, corresponding coordinate system and key notations used. The fluid velocity at the microchannel inlet is assumed uniform

$$x = 0: u_f = u_{in} \quad \text{and} \quad T_f = T_{in} \tag{5}$$

where u_{in} and T_{in} are the given fluid velocity and temperature at the channel inlet. At the channel outlet, a pressure-outlet boundary condition is applied

$$x = L: p_f = p_{out} \tag{6}$$

where p_{out} is the given pressure. For the inner wall/ fluid contact surface,

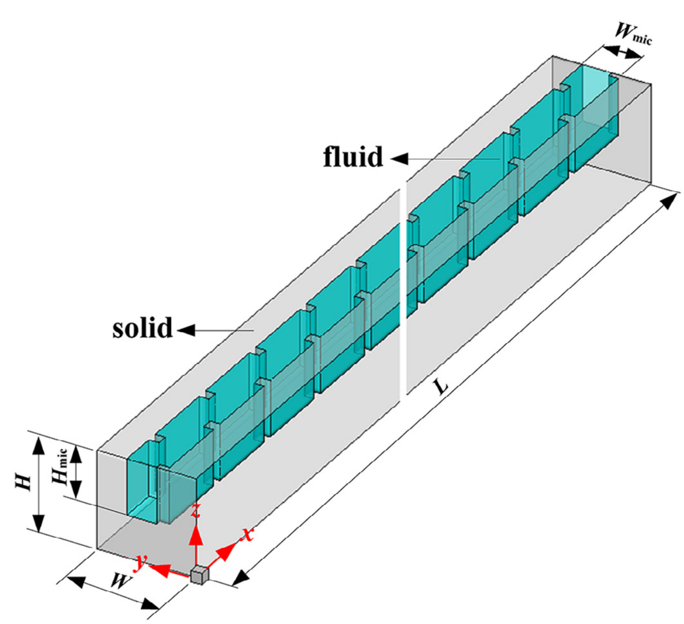

Fig. 1. Structure of microchannel heat sink with offset ribs on sidewalls.