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Fluid flow and heat transfer in microchannel heat sink based on porous fin design concept ☆

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ABSTRACT

In this work, we propose a new design concept of microchannel heat sink, in which solid fins are replaced by porous fins, to reduce the pressure drop across the heat sink. The Forchheimer–Brinkman–Darcy model is used to investigate the effectiveness of this design. The results show that the pressure drop of the new design is reduced by 43.0% to 47.9% at various coolant flow rates as compared with that of the conventional heat sink, with only about 5% increase in the thermal resistance. The pressure drop reduction is attributed to "slip" of coolant on the channel wall due to the presence of porous fins. The drag reduction efficiency for the new design is also calculated by the slip theory extensively used in ultrahydrophobic surfaces, and the calculated value is very close to our simulation value. The results also show that the drag reduction efficiency strongly depends on the porous fin parameters. A larger "slip" velocity of coolant occurs at a higher permeability, a smaller quadratic drag factor, or a larger width ratio of fin-to-pitch, which corresponds to a larger "slip" length. As a result, the pressure drop becomes lower at these conditions.

1. Introduction

Microchannel heat sink has gained extensive attention over the past two decades due to the wide range of applications in cooling of microelectronic devices [1,2]. The heat sink performance is generally characterized by its thermal resistance defined as $R_T = (T_{max} - T_{min}) / Q_w$, where T_{max} and T_{min} are the maximum and minimum temperatures observed in the heat sink, and Q_w is the heat amount dissipated by the heat sink [3]. A lower R_T means a smaller temperature difference required by coolant to dissipate the same heat amount. Many studies demonstrated that the low R_T can be achieved by using the solid material with a high thermal conductivity [4,5] and nanofluid coolant [6,7], as well as optimizing the heat sink geometry [3,8,9].

Recently, great efforts have been devoted to enhancing convective heat transfer and hence to decreasing the thermal resistance through fully inserting a porous medium or partially inserting porous plates, blocks, or baffles into the channel [10–20]. These studies showed a significant heat transfer enhancement in the channel due to the porous inserts; however, this was also accompanied by a drastic increase in pressure drop or pumping power. Several studies also attempted to improve the performance of microchannel heat sink by insertion of porous media [21–23] and found a maximum 76.6% improvement in the thermal resistance when the porous insert with a high porosity was adopted. They also reported that a significant pressure drop was induced across the microchannel. As well known, high pressure drop has become a serious obstacle to the actual applications of microchannel heat sink, thus, this design is restricted.

In this work, a new design of porous microchannel heat sink is proposed to reduce the pressure drop across the heat sink. Different from the designs in Refs. [21–23], the new design adopts porous fins instead of porous media inserted into the channel. The Forchheimer–Brinkman–Darcy model is adopted to examine the effectiveness of the new design. The mechanism of the pressure drop reduction is then explained. Finally, a parametric analysis is performed to optimize the structure of porous fins for further performance improvement of the new design.

2. Analysis

2.1. Microchannel heat sink with porous fins

The schematic of the microchannel heat sink with porous fins is shown in Fig. 1(a). The heat sink has a dimension of $L_x \times L_y \times L_z$

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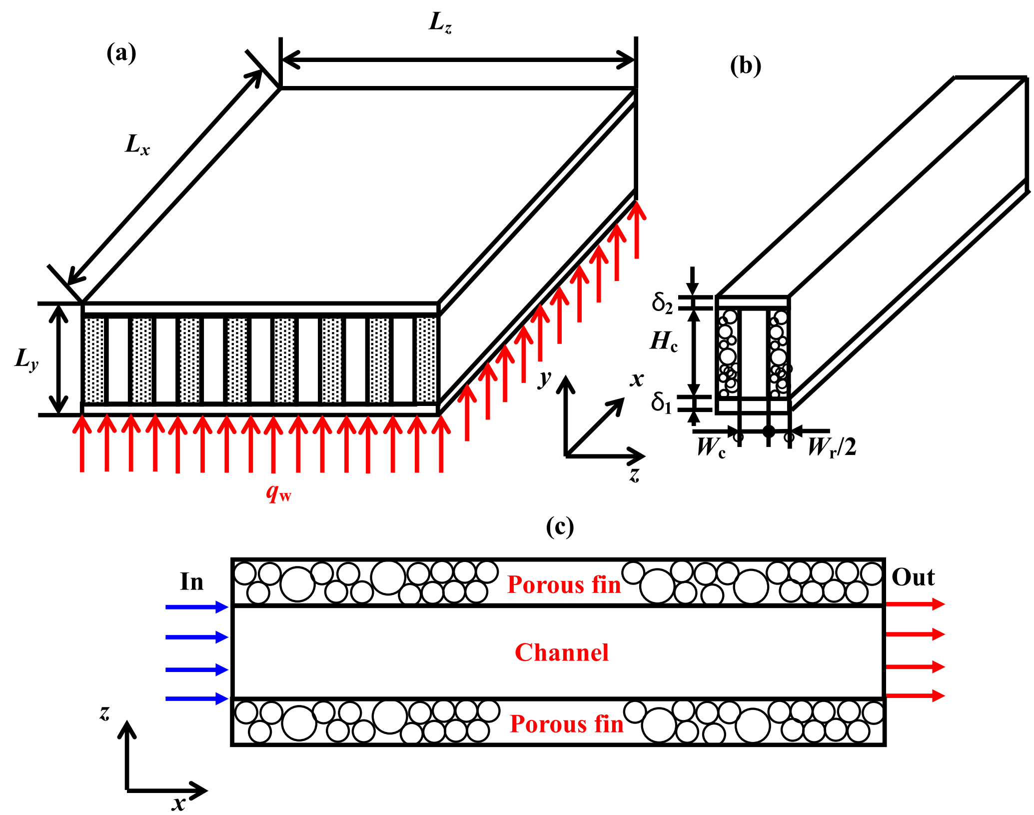

Fig. 1. (a) Microchannel heat sink with porous fins; (b) computational domain; and (c) x–z cross section of the computational domain.

composed of N parallel rectangular microchannels separated by $N+1$ vertical fins. The channel height and width are denoted by H_c and W_c, respectively, while the vertical fin width is denoted by W_r. Here, a width ratio of fin-to-pitch is defined as $\beta = W_r/(W_c + W_r)$. The thickness is δ_1 for the bottom horizontal fin and δ_2 for the top horizontal fin. A uniform heat flux, q_w, is applied to the bottom wall of the heat sink. The heat is firstly transferred from the bottom wall to the heat sink by the horizontal and vertical fins and then dissipated by water coolant.

2.2. Mathematical formulation

Three-dimensional solid–fluid conjugate model is used to solve the convective flow and heat conduction occurred in channels and porous fins of the heat sink based on the Forchheimer–Brinkman–Darcy equation. Considering the symmetry, only a symmetric unit is selected as the computational domain, which is composed of one microchannel and two half-fins, as shown in Fig. 1(b). The model adopts the following assumptions:

1. The flow is single phase, steady state, incompressible, and laminar. The properties of solid fins and coolant are temperature-independent.
2. The viscous dissipation, gravitational force, radiation heat transfer between the heat sink and ambient, and contact thermal resistance between the heat sink and cooled objective are neglected.
3. The porous vertical fins are considered to be homogeneous, isotropic, consolidated, saturated with coolant, and in local thermal equilibrium with the coolant.

The governing equations for the porous fins and channel can be combined into one set by introducing a binary parameter λ [24,25], the values of which are zero in the channel and 1 in the porous fins.

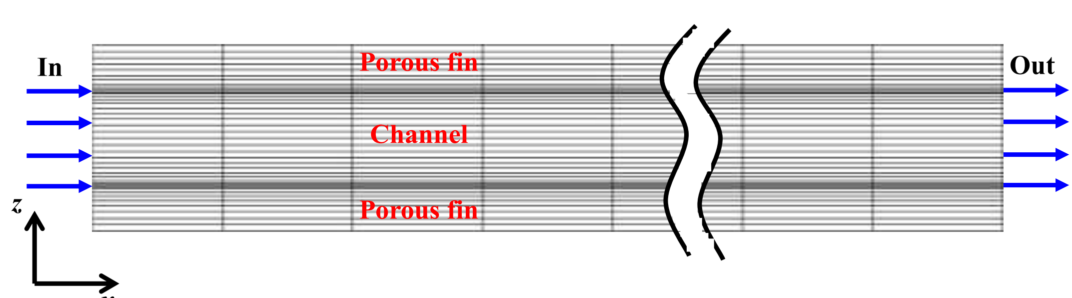

Fig. 2. Typical grid distribution in x–z cross section.