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Optimization of geometry and flow rate distribution for double-layer microchannel heat sink

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ABSTRACT

A three-dimensional solid–fluid conjugated model is coupled with a simplified conjugate-gradient method to optimize the flow and heat transfer in a water-cooled, silicon-based double-layer microchannel heat sink (MCHS). Six design variables: channel number, bottom channel height, vertical rib width, thicknesses of two horizontal ribs, and coolant velocity in the bottom channel are optimized simultaneously to search for a minimum of global thermal resistance. The optimal design variables are obtained at fixed pumping powers, coolant volumetric flow rates, and pressure drops through the MCHS, respectively. The dependences of design variables on the increased pumping power, volumetric flow rate, and pressure drop are discussed. Although the combined optimization is proven effective only for the double-layer MCHS with a specific dimension, it is expected that the proposed design strategy provide a valuable guide for the practical double-layer MCHS design.

1. Introduction

Over the last two decades, with the advances in MEMS technology, micro-systems become more and more compact and small, which results in an ever-increasing heat generation rate from microelectronic devices. The microchannel heat sink (MCHS) originally proposed by Tuckerman and Pease [1] is found to have many advantages such as higher heat dissipation, smaller size and volume per heat load, lower coolant requirement and lower operational cost over the convectional cooling techniques, and has received extensive studies [2–17]. Especially, Ng's group for the first time introduced the concept of electric double layer to explain the microscale deviation between flows in microscale channels and large-scale channels [12–14].

In original MCHS configuration, there is only one layer of parallel microchannels separated by solid ribs. Due to the limitation of pumping power, only a small coolant flow rate can be adopted. When the coolant passes through the channels and takes heat away from the heat dissipating microelectronic device attached below, its temperature gradually increases and the cooling capacity gradually deteriorates. Thus, a relatively higher temperature rise along the microchannel is inevitably formed. This undesirable temperature gradient may produce a large thermal stress in microelectronic device due to mismatch of thermal expansion coefficients between different materials. In addition, the large temperature gradient may also produce instability and thermal breakdown of microelectronic device.

In order to reduce the undesirable temperature gradient, Vafai and Zhu [18] originally proposed a new design concept based on stacking two layers of microchannel heat sink structures, one atop the other, with coolant flow in the opposite direction in each of the microchannel layers. Followed by Vafai and Zhu's work, many investigations [19–28] studied the cooling performance of double-layer MCHSs. These investigations confirmed that the double-layer structure significantly improved the temperature uniformity at the bottom wall compared with the single-layer one, and therefore the pressure drop required for the double-layer design can be much smaller than that of the single-layer design when small temperature rise is required for microelectronic devices. Optimal geometric structure for the double-layer MCHS was also investigated by optimization algorithms [29,30]. Chong et al. [29] adopted a thermal resistance model linked with a multivariable constrained direct search optimization algorithm to optimize the performance of single- and double-layer MCHSs at fixed pressure drops. Their results showed that both single- and double-layer MCHSs operating

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in the laminar out performed the heat sinks under turbulent flow conditions, both in heat transfer and hydrodynamic considerations. Jeevan et al. [30] optimized the channel height, channel width, and rib width to search for a minimum thermal resistance using genetic algorithms and box optimization method at fixed pumping powers. Their results indicated that the double-layer MCHS produced less thermal resistance compared to the single-layer one.

Fixed pumping power for evaluating the cooling performance of the MCHS is a physically practical constraint condition because which means the power required to drive the coolant through the MCHS is the same. However, practical MCHS generally operates at a fixed volumetric flow rate or a fixed pressure drop. As a result, the optimal double-layer MCHS design at various operation conditions should be investigated. In addition, when the geometric structure of the double-layer MCHS is determined, its cooling performance still strongly depends on the flow rate distribution between bottom and upper channels. To our best knowledge, simultaneous optimizations of the geometric structure and flow rate distribution for the double-layer MCHS at fixed pumping powers, coolant volumetric flow rates, or pressure drops have not been conducted.

The objective of this work is to develop an optimization approach to look for the optimal double-layer MCHS design. The optimization approach integrates a three-dimensional solid–fluid conjugated MCHS model and a simplified conjugated-gradient method. The channel number, bottom channel height, vertical rib width, thicknesses of two horizontal ribs, and coolant velocity in the bottom channel as search variables are optimized simultaneously to reach a minimum thermal resistance at fixed pumping powers, coolant volumetric flow rates, and pressure drops, respectively. The effects of these three constraint conditions on the design direction of search variables are discussed. The present optimization approach and the corresponding design strategy are expected to provide a guide for practical double-layer MCHS design.

2. Optimization method

2.1. Geometric structure of double-layer MCHS and optimized parameters

The schematic of double-layer MCHS with dimensions of $L_x = 10$ mm, $L_y = 1$ mm, and $L_z = 10$ mm is shown in Fig. 1. The MCHS has two layers of rectangular channels and ribs. The bottom and upper layers have the same channel number, N, rib number, N, channel width, W_c, and rib width, W_r. The heights of bottom and upper channels are H_{c1} and H_{c2}, respectively. The thicknesses of two horizontal ribs are δ_1 and δ_2, respectively. For easy fabrication, $\delta_1 \geq 50$ μm and $\delta_2 \geq 50$ μm. Flow in bottom channels is along the x-direction and flow in upper channels is opposite to the x-direction. Generally, the bottom wall of the MCHS is attached to an electric equipment or other heat dissipating component, thus, a uniform heat flux, q_w, is assumed to be applied to the bottom wall.

With the fixed dimensions of $L_x = 10$ mm, $L_y = 1$ mm, $L_z = 10$ mm, the parameters N, W_r, H_{c1}, δ_1, and δ_2 can determine the MCHS geometric structure uniquely. In addition, when the pumping power, inlet volumetric flow rate, or pressure drop is specified, the cooling performance of the MCHS is also closely depends on the flow rate distribution between bottom and upper channels. Consequently, the five geometric parameters N, W_r, H_{c1}, δ_1, and δ_2, as well as the inlet flow velocity in bottom channel, u_{in1}, are chosen as the optimized parameters or search variables. They are optimized to reach an optimal MCHS performance at fixed pumping powers, coolant volumetric flow rates, and pressure drops, respectively. The present double-layer MCHS can be fabricated using the selective reactive ion etching (RIE) technique. Each one of the five optimized geometric parameters can be realized by the RIE technique. For the double-layer MCHS, the reason that the pressure drop is chosen as the constraint condition is based on the following consideration. To control the pressure drops across bottom and upper channels, two micro-pumps scheme may be adopted, where one pump powers the coolant flow in bottom channels and the other in upper channels. As a result, when performance of such two double-layer MCHS designs are compared, the sum of pressure drops of the two pumps should be constrained to a constant value.

2.2. Three-dimensional solid–fluid conjugate model for double-layer MCHS

Three-dimensional solid–fluid conjugate model is developed to solve the convective flow and heat conduction occurred in channels and ribs of the MCHS. The model adopts the following assumptions: (1) water and silicon are used as the coolant and solid material; (2) the flow is single phase, laminar, and steady state; (3) constant fluid and solid properties are adopted; (4) gravitational force is ignored; (5) contact thermal resistance between the MCHS and electric equipment is ignored; (6) except for the bottom wall, heat losses of the MCHS to the ambient are ignored. The governing equations are as follows.

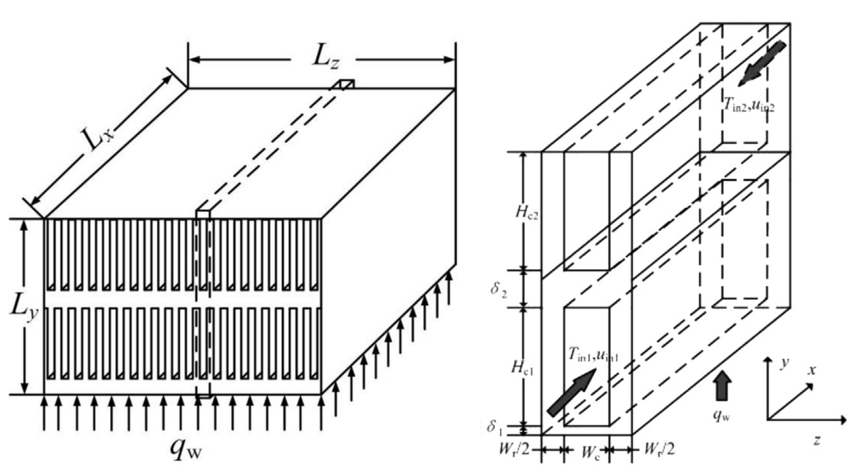

Fig. 1. Schematics of double-layer MCHS and its computational domain.