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Three-dimensional numerical optimization of a manifold microchannel heat sink

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Abstract

A three-dimensional analysis procedure for the thermal performance of a manifold microchannel heat sink has been developed and applied to optimize the heat-sink design. The system of fully elliptic equations, that govern the flow and thermal fields, are solved by a SIMPLE-type finite volume method, while the optimal geometric shape is traced by a steepest descent technique. For a given pumping power, the optimal design variables that minimize the thermal resistance are obtained iteratively. The procedure is robust and the optimal state is reached within six global iterations. Comparing with the comparable traditional microchannel heat sink, the thermal resistance is reduced by more than a half while the temperature uniformity on the heated wall is improved by tenfold. The sensitivity of the thermal performance on each design variable is also examined and presented in the paper. Among various design variables, the channel width and depth are more crucial than others to the heat-sink performance. The optimal dimensions and corresponding thermal resistance have a power-law dependence on the pumping power.

1. Introduction

The recent trend in the electronic equipment industry toward denser and more powerful products requires higher thermal performance from a cooling technique. Many ideas for innovative cooling methods have been proposed including a microchannel heat sink. Two representative types of the microchannel heat sink are (1) the traditional microchannel (TMC) type and (2) the manifold microchannel (MMC) type. A TMC heat sink, which was proposed first by Tuckerman and Pease [1], is characterized by the long microchannels that run in one direction parallel to the heat-sink base. It has been successfully investigated by using a simple one-dimensional model [2] or more sophisticated three-dimensional numerical methods [3,4]. Even though the TMC heat sink has brought substantial improvements in the cooling performance, it has two disadvantages: the relatively high pressure loss and the significant temperature variation within the heat source. An MMC heat sink, on the other hand, differs from a TMC type in that the coolant flows through the alternating inlet and outlet manifolds in the direction normal to the heat-sink base to and from the segmented microchannels as shown in Fig. 1(a). The flow path is greatly reduced to a small fraction of the total length of a heat sink; the shortened flow path is expected to reduce the pressure drop and restrain the growth of the thermal boundary layer along the streamwise direction.

Harpole and Eninger [5] proposed an MMC system having between 10 and 30 manifold channels and reported that, for constant flow rate or pumping power, the maximum temperature and the temperature variation within the heat source were substantially reduced from that of a TMC heat sink. Copeland et al. [6] tested a variety of MMCs experimentally and reported that the thermal resistance was inversely proportional to the volume flow rate in a log–log scale. Copeland et al. [7] also found in a comparative study of an MMC heat sink

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Nomenclature

c_p	heat capacity of fluid
D_h	hydraulic diameter of the channel, $2w_cH/(w_c+H)$
h	heat transfer coefficient defined in Eq. (9)
H	channel depth
k	thermal conductivity
$L \times W$	dimension of a heat sink
$M_{dv}, M_{in}, M_{out}, M_{tot}$	manifold dimensions (see Fig. 1)
N	number of channels
p	pressure
P	pumping power, $Q\Delta p$
Pr	Prandtl number, $\mu c_p/k_f$
$\dot{q}$	total heat-flow rate loaded on the bottom wall
q_w	heat flux loaded on the bottom wall
Q	total volume flow rate
Re	Reynolds number, $\rho U D_h/\mu$
R_t	thermal resistance
S	search direction vector
T	temperature
ΔT_{max}	maximum temperature difference in a heat sink
$\vec{u}$	velocity vector in the fluid region
U	average velocity in the channel
w_c	channel width
W_w	fin thickness
w	$w_c + w_w$
x, y, z	cartesian coordinates
X	design variable vector

Greek symbols

δ	substrate thickness
θ	nondimensionalized temperature
μ	fluid viscosity
ρ	fluid density
γ	inlet/outlet width ratio, M_{in}/M_{out}

Superscript

$*$	nondimensionalized variable

Subscripts

cf	channel floor
f	fluid
in	inlet
max	maximum
out	outlet
s	solid

that the simple analytical model based on correlations for a straight channel is not satisfactory in predicting the performance.

The purpose of the present study is to develop a three-dimensional analysis procedure for the thermal performance of an MMC heat sink and apply it to optimize the geometric shape and the operating condition. The SIMPLE-type finite volume method is coupled with an optimization scheme based on the steepest descent method [8]. The geometric parameters that minimize the thermal resistance are obtained. The effects of the channel number and the pumping power on the performance of a heat sink are examined.

2. Numerical analysis

The problem under consideration concerns the forced convection through the MMC heat sink depicted in Fig. 1(a). A coolant, guided by the inlet, divider, and outlet manifolds, passes through a number of microchannels and takes heat away from an electronic component attached below. In analyzing the problem, it is assumed that the flow is laminar, incompressible, and all thermophysical properties are constant.

Due to the periodicity, it suffices to consider only a single periodic module of the geometry, that includes the flow passage, the substrate, the fin, and the manifold divider, shown in Fig. 1(b). The nondimensionalized continuity, Navier–Stokes, and energy equations in three dimensions that govern the flow may be written as

Continuity equation:

$$\nabla^* \cdot \vec{u}^* = 0 \tag{1}$$

Momentum equation:

$$\vec{u}^* \cdot \nabla^* \vec{u}^* = -\nabla^* p^* + \frac{1}{Re} \nabla^{*2} \vec{u}^* \tag{2}$$

Energy equation:

$$\vec{u}^* \cdot \nabla^* \theta = \frac{1}{Re\,Pr} \nabla^{*2} \theta \quad \text{(fluid region)} \tag{3}$$

$$0 = \nabla^{*2} \theta \quad \text{(solid region)} \tag{4}$$

where

$$Re = \frac{\rho U D_h}{\mu}, \quad Pr = \frac{\mu c_p}{k_f},$$
$$\nabla^* = D_h \nabla, \quad \vec{u}^* = \vec{u}/U, \quad p^* = p/\rho U^2,$$
$$\theta = \frac{T - T_{f,in}}{q_w D_h / k_s} \tag{5}$$

Here $\vec{u}$, U, p, and T are the velocity vector, the mean velocity in the channel, the pressure, and the temperature, respectively. The fluid properties, ρ, μ, k, and c_p