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Optimal geometric structure for nanofluid-cooled microchannel heat sink under various constraint conditions

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ABSTRACT

A numerical model is developed to analyze the flow and heat transfer in nanofluid-cooled microchannel heat sink (MCHS). In the MCHS model, temperature-dependent thermophysical properties are taken into account due to large temperature differences in the MCHS and strong temperature-dependent characteristics of nanofluids, the model is validated by experimental data with good agreement. The simplified conjugate-gradient method is coupled with MCHS model as optimization tool. Three geometric parameters, including channel number, channel aspect ratio, and width ratio of channel to pitch, are simultaneously optimized at fixed inlet volume flow rate, fixed pumping power, and fixed pressure drop as constraint condition, respectively. The optimal designs of MCHS are obtained for various constraint conditions and the effects of inlet volume flow rate, pumping power, and pressure drop on the optimal geometric parameters are discussed.

1. Introduction

Since the pioneer work by Tuckerman and Pease [1], the MCHS has attracted extensive attention over the past two decades. The MCHS has become an important cooling device to high power light emitting diode, very-large-scale integrated circuits and Micro-Electro Mechanical System applications [1–5]. The performance of a MCHS is closely related to properties of solid material and coolant fluid, to the flow state (laminar flow or turbulent flow, inlet flow rate and temperature of coolant, etc.), and to its geometric structure. The most frequently used coolants in the MCHS study were air, water, and fluorochemicals etc. Recent studies indicated that nanofluids which have high thermal conductivities were applied to MCHS as coolants, the MCHS performance was significantly improved [6–16].

There are two approaches in modeling of nanofluid flow and heat transfer in MCHS. First approach to describe the heat transfer and flow for nanofluid is to treat the nanofluid as a real two-phase mixture in which irregular and random movement of particle increases the heat exchanging rate. The second approach is to treat the nanofluid as a single-phase fluid based on the fact that nanofluid has good uniformity with low particle volume fraction due to nanoscale particle size. In the second approach, the themophysical properties of base fluid, including density, specific heat, thermal conductivity and viscosity, must be substituted by nanofluid's ones. Based on single-phase approach, the thermal conductivity and viscosity of nanofluids have been extensively investigated from both experimental and theoretical viewpoint [17]. The density, thermal conductivity and viscosity of the nanofluids are increased while the specific heat is reduced by the addition of the nanoparticles [18]. The thermal conductivity enhancement mechanisms for nanofluids include the interfacial nanolayer ordering, Brownian motion, Brownian-motion-induced microconvection, particle clustering structures and ballistic transport of energy carriers [18–21].

Table 1 summarizes the recent numerical investigations on nanofluid-cooled MCHS. All models adopted the single-phase approach. Apart from Chen and Ding's model [13], the models were not validated by experiment, because experimental data on flow and heat transfer characteristics for nanofluid-cooled MCHS were available in open literatures until 2007 years [14]. Chein and Chuang [14] tested experimentally the MCHS performance using copper oxide/water nanofluids with 0.2–0.4% particle volume fractions as the coolants. Later, Ho et al. [15] tested experimentally forced convective cooling performance of MCSH with alumina/water nanofluid as the coolant. Both the numerical predictions and the experimental data confirmed that use of nanofluids enhances the cooling performance of MCHS and produces only small increases in pressure drop or pumping power at low particle volume fractions.

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Nomenclature

A_1	convective heat transfer area (m^2)
c_p	specific heat (J kg^{-1} K^{-1})
D	hydraulic diameter (m)
H_{ch}	channel height (m)
J	objective function
k	thermal conductivity (W m^{-1} K^{-1})
$k_{nf,eff}$	effective thermal conductivity for the nanofluid flow
L_x	channel length (m)
L_y	height of heat sink (m)
L_z	width of heat sink (m)
N	channel number
p	coolant pressure (Pa)
q_w	heat flux applied to bottom surface of heat sink (W m^{-2})
Q	total volumetric flow rate (m^3 s^{-1})
R_T	total thermal resistance (K W^{-1})
T	temperature (K)
u, v, w	velocity component in x, y, z direction (m s^{-1})
u_m	average velocity of coolant over channel cross-section (m s^{-1})
W_{ch}	channel width (m)
W_r	rib width (m)

Greek

α	aspect ratio of the channel
β	width ratio of channel to pitch
$\gamma_N^{(k)}, \gamma_\alpha^{(k)}, \gamma_\beta^{(k)}$	conjugate gradient coefficients of (N, α, β) in the kth search step
δ	thickness of bottom wall of solid (m)
μ	viscosity (kg m^{-1} s^{-1})
$\xi_N^{(k)}, \xi_\alpha^{(k)}, \xi_\beta^{(k)}$	search direction of (N, α, β) in the kth search step
ρ	density (kg m^{-3})
φ	particle volume fraction
$X_N^{(k)}, X_\alpha^{(k)}, X_\beta^{(k)}$	search step size of (N, α, β) in the kth search step
Ω	pumping power (W)

subscripts

bf	base fluid
in	inlet
l	liquid
nf	nanofluid
out	outlet
p	nanoparticle
s	solid phase

The investigations also found that the geometric structure has remarkable effect on the thermal resistance of nanofluid-cooled MCHS [6,7,10,12]. With different constraint conditions, including fixed pumping power [6,9,10,16], fixed pressure drop [8,10], fixed inlet volume flow rate [13], fixed inlet velocity [7,11], and fixed inlet Reynolds number [12], some conclusions made by these investigations are different. For example, with the fixed pumping power by Tsai and Chein [10] and with the fixed inlet velocity by Li and Kleinstreuer [11] as constraint conditions, the thermal performance of MCHS increases with particle volume fraction, but Ghazvini and Shokouhmand [12] demonstrated that there is an optimal particle volume fraction to reach the maximum heat transfer with fixed inlet Reynolds number. Lelea [16] revealed that contrary to the analysis based on a Re = constant basis, in the fixed pumping power case the heat transfer enhancement rises along the microchannels. Also the heat transfer augmentation increases as the particle's concentration increases. Similarly, the effects on the geometric parameters on the cooling performance are also different in these investigations due to different constraint conditions being used. Tsai and Chein [10] proposed that there are the optimal porosity (ratio of channel width to total width of MCHS) and aspect ratio under a given pressure drop across the MCHS, nanofluids can enhance the MCHS performance when the porosity and aspect ratio are less than the optimum porosity and aspect ratio, oppositely, nanofluids did not produce a significant change in MCHS thermal resistance. Ghazvini and Shokouhmand [12] found that the increase in the porosity and channel aspect ratio always improved MCHS performance under fixed inlet Reynolds number.

The fixed pumping power condition for evaluating cooling performance of the MCHS is physically practical constraint condition because which means the power required to drive the fluid through the MCHS is the same. However, from the viewpoint of practical operation of MCHS the fixed inlet volume flow rate or pressure drop is more easily controlled. In addition, for a MCHS, the geometric parameters include the channel number, the channel aspect ratio, and the width ratio of channel to pitch, and all parameters have coupled effect on the MCHS cooling performance. An individual parameter study is useful but it cannot answer how one can obtain the optimal design. Therefore, a multi-parameter coupled/combined effect is needed to account for to obtain optimal nanofluid-cooled MCHS performance. Based on the above reasons, this work develops an inverse problem optimization method, which combines a complete three-dimensional solid–fluid conjugated MCHS model and simplified conjugate-gradient method, to optimize geometric parameters of nanofluid-cooled MCHS for various constraint conditions, including fixed inlet volume flow rate, the fixed pumping power, and fixed pressure drop. In the MCHS model, temperature-dependent thermophysical properties are taken into account due to large temperature differences in the MCHS and strong temperature-dependent characteristics of nanofluids, the model is validated by experimental data by Ho et al. [15] with good agreement. The optimal designs of MCHS are obtained for various constraint conditions and the effects of pumping power, inlet volume flow rate, and pressure drop on the optimal geometric parameters are discussed.

2. Parameters of nanofluid-cooled MCHS for optimization

The schematic of MCHS with dimensions of $L_x = 10$ mm, $L_y \leqslant 1$ mm, and $L_z = 10$ mm is shown in Fig. 1, which consists of N parallel microchannels and N ribs with rectangular cross-section. Usually, the bottom of the MCHS is mounted on electronic equipment or other heat dissipating component. Heat is removed primarily by conduction through the solid and then dissipated away by convection of the cooling fluid in the microchannels. The channel has a height H_{ch}, width W_{ch}, and the rib has a width W_r with the same height as the channel, thus, we have $W_{ch} + W_r = L_z/N$. The thickness of the top and bottom plates are fixed to $\delta = 0.1$ mm. The channel aspect ratio and the width ratio of channel to pitch are defined as $\alpha = H_{ch}/W_{ch}$, $\beta = W_{ch}/(W_{ch} + W_r)$, respectively. Once N, α, and β are given, the geometric structure of the MCHS is determined uniquely. Therefore, N, α, and β are chosen as optimized parameters and are optimized simultaneously at fixed inlet volume flow rate, fixed pumping power, and fixed pressure drop in the present work, respectively.

3. Optimization method

3.1. The nanofluid-cooled MCHS model

The solid–fluid conjugated model is refined from that adopted in our previous work for water-cooled MCHS [5] with modified