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Optimization of thermal performance of multi-nozzle trapezoidal microchannel heat sinks by using nanofluids of Al_2O_3 and TiO_2

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ABSTRACT

In this study, a new multi-nozzle trapezoidal microchannel heat sink (MNT-MCHS) was proposed. Five substrate materials, two nanofluids with nanoparticle volume fractions, $0.1\% \leq \varphi \leq 1\%$, and channel hydraulic diameters, $157.7\ \mu m \leq D_h \leq 248.2\ \mu m$, were numerically examined in detail. In addition, heat fluxes in the range of 100–1450 W/cm^2 subject to inlet coolant temperature from 15 °C to 75 °C were examined in detail. A locally optimal MNT-MCHS was defined, and a novel equation was proposed for predicting the maximum temperature on the locally optimal MNT-MCHS depending on the heat flux, coolant inlet temperature, and the Reynolds number. It was found that at a Reynolds number of 900, the overall thermal resistance of a MNT-MCHS using copper as a substrate material is improved up to 76% as compared to that using stainless steel 304. The locally optimal MNT-MCHS, using TiO_2-water nanofluid with $\varphi = 1\%$, could dissipate a heat flux up to 1450 W/cm^2 at a *Re* of 900. A minimum thermal resistance in the present study is improved up to 11.6% and 36.6% in association with those of a multi-nozzle MCHS and a double-layer MCHS, respectively.

1. Introduction

In parallel with the development of scientific technology, electronic products not only have continued to shrink their sizes, but also have added more functions and operated at even higher processing speeds. As a consequence, a huge amount of heat is generated in a rather small volume that makes traditional air-cooling heat sinks in a dilemma. Hence, liquid-cooling heat sinks had been implemented to resolve the tough thermal management in many high-power applications. A great breakthrough in the implementation of liquid-cooled heat sinks was created by Tuckerman and Pease [1] in 1981. They proposed microchannel heat sinks (MCHS) using water as the working fluid. One of the MCHSs could dissipate a heat flux up to 790 W/cm^2 with an overall thermal resistance of 0.09 °C/W and kept a maximum junction temperature under 94 °C. Thereafter, many researchers were involved in this field due to its promisingly heat-dissipation capability.

Investigations in MCHSs can be classified into three main groups. Most research efforts devoted to the first group by examining the effects of geometric parameters on the heat transfer and fluid flow characteristics of the MCHSs, including influences of channel shapes, channel dimensions, manifold shapes, manifold dimensions, channel layers, channel distributions, substrate dimensions and the like. The second group investigated the effects of the thermophysical parameters of the coolants on the heat transfer and fluid flow characteristics of the MCHSs, and the third group, with the fewest reports in this field, studied the effects of the thermophysical properties of the substrate's materials on the heat transfer of the MCHSs. In the first group, a great number of the studies had concentrated on the performance of single-layer parallel microchannel heat sinks (SL-P-MCHS). However, the SL-P-MCHSs had not been widely applied due to concerns of high-pressure drop and non-uniform temperature distribution. To tailor the non-uniform temperature distribution, Vafai and Zhu [2] first proposed a double-layered microchannel heat sink (DL-MCHS) in 1999. The inlets of the upper layer of a DL-MCHS were located on top of the outlets of the lower layer, thereby lowering the temperature near the outlet region, and a more uniform temperature prevails on the bottom surface. However, the DL-MCHS design showed no appreciable improvements in overall thermal resistance and pressure drop as compared to those of the SL-MCHSs. To ease pressure drop penalty of the MCHSs, Boteler et al. [3] were the first to propose a manifold microchannel heat sink (M-MCHS). The coolant paths of the M-MCHS were shortened

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Nomenclature

a	length of the top edge of the trapezoidal channel (m)
$A_{bt,s}$	heat transfer area of the bottom wall of the heat sink (m^2)
b	length of the bottom edge of the trapezoidal channel (m)
B_{th}	bottom thickness (m)
C_p	specific heat at constant pressure (J/kg K)
D_h	channel hydraulic diameter (m)
D_c	channel depth (m)
H_s	substrate height (m)
L_c	channel length (m)
L_m	model length (m)
L_n	inlet-nozzle length (m)
L_t	top cover length (m)
MCHS	microchannel heat sink
Q	heat transfer rate (W)
q_w	heat flux applied on the bottom wall (W/cm^2)
Re	Reynolds number
R_T	thermal resistance (°C/W)
$T_{bt,max}$	maximum temperature on the bottom wall (°C)
T_i	inlet-coolant temperature (°C)
ΔT	temperature difference between inlet and outlet coolant (°C)
W_m	model width (m)
W_s	substrate width (m)

Greek symbols

μ	dynamic viscosity of the coolant (kg/ms)
ρ	density of the coolant (kg/m^3)
φ	nanoparticle volume fraction (%)

Subscripts

bf	base fluid
bt,max	bottom maximum
f	fluid
i,o	inlet_outlet
nf	nanofluid
p	nanoparticle

considerably when compared to those of the SL-MCHSs and DL-MCHSs. In this regard, the M-MCHS remarkably reduced the pressure drop and considerably increased the temperature uniformity as compared to the SL-MCHSs and the DL-MCHSs. Although the pressure drop and temperature uniformity of the M-MCHS were conspicuously improved, it still suffers from comparatively high overall thermal resistance as compared to that of the SL-MCHSs or DL-MCHSs. For further improvements to balance the pressure drop penalty, temperature uniformity, and thermal resistance into a microchannel heat sink, Tran et al. [4] recently proposed a multi-nozzle microchannel heat sink (MN-MCHS). The key features of the MN-MCHS are those channel lengths are much shorter as compared to those of the SL-MCHS and DL-MCHS, and channel heat transfer areas are larger than that of the M-MCHS subject to the same channel shape and hydraulic diameter. They showed that the overall thermal resistance, temperature uniformity and the pressure drop of the MN-MCHS were significantly improved.

Normally, water was utilized as a working fluid in MCHSs for its superior thermal-physical properties. For further augmentation of the heat-transfer capability of the coolants, Choi and Eastman [5] first proposed a nanofluid concept. Eastman et al. [6] reported that by adding 5% CuO nanoparticles, it could improve thermal conductivity by approximate 60% in comparison with that of pure water. In recent years, the number of investigations on nanofluids in the MCHSs increased drastically. For example, Hung et al. [7,8] used Al_2O_3-water nanofluids as coolants in a SL-MCHS and a DL-MCHS, respectively. They concluded that the thermal resistance of SL-MCHS and DL-MCHS could be minimized by properly adjusting the nanoparticle volume fraction. Kuppusamy et al. [9] created various combinations amid nanoparticles (Al_2O_3, CuO, SiO_2, and ZnO) and base fluids (water, ethylene glycol, and engine oil), and conducted tests in a triangular grooved microchannel heat sink (TG-MCHS). They reported that the thermal performance of the TG-MCHS with Al_2O_3-water nanofluid having a volume fraction of 0.04% (nanoparticle diameter of 25 nm) outperformed the straight MCHS using pure water. A MCHS using CuO-water nanofluids as working fluids was experimentally investigated by Rimbault et al. [10]. Their results showed that the thermal performance of the MCHS using CuO-water nanofluids with $\varphi \leqslant 1.03\%$ was slightly increased when compared to that using water. Sakanova et al. [11,12] reported results from a DL-MCHS and a double-side MCHS (DS-MCHS) using Al_2O_3-water nanofluid [11], with a wavy MCHS using three different nanofluids [12]. They showed that the thermal performances of the heat sinks using the Al_2O_3-water nanofluid with $\varphi = 1\%$ and $\varphi = 5\%$ were substantially better than those using water. The convective heat transfer and friction factor of Cu-water nanofluids in a cylindrical microchannel heat sink were experimentally studied by Azizi et al. [13,14]. For the same Reynolds number, the corresponding Nusselt numbers of the heat sink with φ being 0.05%, 0.1% and 0.3% were enhanced approximately 17%, 19%, and 23%, compared to that using pure water, respectively. A ribbed MCHS using Al_2O_3-water nanofluids was numerically investigated by Ghale et al. [15]. They revealed that the Nusselt number and friction factor of nanofluids in the ribbed microchannel were higher than those of a straight microchannel without rids inside. Nebbati and Kadja [16] presented a numerical study on a microchannel heat sink using Al_2O_3-water nanofluid. They reported that the heat transfer coefficient of a heat sink using nanofluid was significantly increased in comparison with pure water. Parametthanuwat et al. [17] presented an experimental study on thermal properties of silver nanofluids. They revealed that a silver nanofluid with $\varphi = 1\%$ at a temperature of 80 °C yielded the highest heat transfer enhancement. PCM slurry and nanofluid coolants were numerically investigated in a DL-MCHS by Rajabifar [18]. The author revealed that the pumping power of both PCM slurry and nanofluid was significantly raised for their higher viscosity. A M-MCHS using nanofluids was numerically investigated by Yue et al. [19]. They concluded that the increase in the nanoparticle diameter led to the decrease in the Nusselt number, pumping power, and performance index. Recent investigations using nanofluids in microchannel heat sinks were reported by Kim et al. [20], Radwan et al. [21], Xia et al. [22], Yang et al. [23], and Zhao et al. [24]. The results in [20] showed that the thermal resistance of the flat-plate heat pipes with Al_2O_3-Acetone nanofluids containing sphere-, brick- and cylinder-shaped nanoparticles were decreased by 33%, 29%, and 16%, respectively, as compared to that using pure acetone. Radwan et al. [21] concluded that the use of nanofluids in MCHSs was quite effective. Xia et al. [22] revealed that the thermal conductivity and dynamic viscosity of Al_2O_3 and TiO_2 nanofluids were both increased with the increase in φ. Yang et al. [23] reported that the nanoparticle volume fraction and the fins aspect ratio played a very important role in the enhancement in the heat transfer of the MCHSs. Zhao et al. [24] reported that the pressure drop could