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Heat transfer enhancement of microchannel heat sink using transcritical carbon dioxide as the coolant

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ABSTRACT

Microchannel heat sink (MCHS) is regarded as a promising cooling scheme for high power microelectronic devices. To further enhance its cooling capacity and improve the temperature uniformity, the investigation of the MCHS employing transcritical CO_2 as coolant was conducted in this work for the first time. A three-dimensional solid–fluid conjugated model is developed to investigate the performance of CO_2-cooled MCHS. Sufficiently changed thermophysical properties of the transcritical CO_2 is taken into account in the model. The results demonstrate that, with the same pumping power of 0.03 W, the thermal resistance (R) of the CO_2-cooled MCHS is reduced by 23.34–34.62% in the inlet temperature range of 285–305 K, as compared with the water-cooled MCHS. Likewise, the maximum temperature drop ($\Delta T_{b,max}$) on the bottom surface is decreased by 24.18–48.75%. The improved performance is attributed to the lower viscosity and higher specific heat of the transcritical CO_2. Moreover, as compared with the water-cooled MCHS, R and $\Delta T_{b,max}$ for the CO_2-cooled MCHS exhibit a more significant reduction when the MCHS has a larger number of channels, a larger channel aspect ratio, or a smaller channel width to pitch ratio. For a given inlet pressure p_{in} of CO_2, the optimal inlet temperature should be appropriately lower than the pseudocritical temperature at the given p_{in} to ensure the larger specific heat for CO_2 flowing in microchannels.

1. Introduction

With the advancement of technologies, the electronic chip has been widely used in the fields of energy, power, aerospace, metallurgy, chemical engineering, and so on. Meanwhile, electronic chips are moving toward miniaturization as well as larger and larger heat fluxes. When heat fluxes reach up to as high as 100 W cm^{-2}, conventional single-phase cooling technologies such as forced-air heat exchangers are failed. To more effectively dissipate the high heat fluxes, Tuckerman and Pease [1] proposed a concept of microchannel heat sink (MCHS) made of the solid material with a high thermal conductivity. In their experiments, the heat sink was made of silicon and the microchannels had rectangular cross-section. The results showed that using water as coolant, the MCHS can remove an ultra-high heat flux of 790 W cm^{-2} with a temperature rise of 71 °C on the bottom surface. Besides, the MCHS also exhibited the other advantages such as compact size, low coolant requirement, and low operation cost, as compared with the conventional single-phase cooling technologies.

After the landmark work by Tuckerman and Pease [1], many efforts have been devoted to improving the cooling capability of MCHS. Some studies focused on the enhancement of convective heat transfer by changing the channel shape and flow field arrangement. For example, trapezoidal [2], circular [3], triangle [4], triangular grooved [5], wavy [6], tapered [7], and converging [8] channels were investigated, and some new flow field configurations such as the heat sink consisting of four compartments with separate coolant inlet and outlet plenum for each compartment [9], as well as serpentine, coiled, and hybrid flow fields [10] were proposed. Besides the above designs, Hung et al. [11] inserted porous materials into channels to improve the performance of MCHS. Recently, Leng et al. [12] proposed a new MCHS design concept, in which solid ribs were replaced by porous ribs. Their results demonstrated that the pressure drop across the channel was reduced by

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Nomenclature

A	bottom surface area of MCHS (m^2)
A_c	cross-sectional area of channel (m^2)
c_p	specific heat (J kg^{-1} K^{-1})
H_c	channel height (m)
k	thermal conductivity (W m^{-1} K^{-1})
L_x	length of MCHS (m)
L_y	height of MCHS (m)
L_z	width of MCHS (m)
N	the number of channels
p	pressure (Pa)
q_w	heat flux on the bottom surface (W m^{-2})
Q_v	total volumetric flow rate (m^3 s^{-1})
R	overall thermal resistance (K W^{-1})
T	temperature (K)
T_{max}	maximum temperature observed in MCHS (K)
T_{min}	minimum temperature observed in MCHS (K)
$T_{b,max}$	maximum temperature observed on the bottom surface (K)
$T_{b,min}$	minimum temperature observed on the bottom surface (K)
u, v, w	velocity component in the x-, y-, and z-directions (m s^{-1})
W_c	channel width (m)
W_r	vertical rib width (m)
x, y, z	coordinates (m)

Greek symbols

β	channel width to pitch ratio
γ	channel aspect ratio
δ	thickness of the horizontal rib (m)
μ	coolant viscosity (kg m^{-1} s^{-1})
ρ	coolant density (kg m^{-3})
Ω	total pumping power (W)

Subscripts

c	channel
f	fluid phase
in	inlet
out	outlet
pc	pseudocritical point
s	solid phase

about 40%, as compared with the conventional design. Some optimization algorithms such as simplified conjugate-gradient method [13], evolutionary algorithm [14], simulated annealing method [15], and genetic algorithm [16] were combined with the MCHS model to search for the optimal MCHS geometry. These studies can also be classified into single-objective optimization [13,15] and multi-objective optimization [14,16].

The performance of MCHS is also significantly influenced by the choice of coolant. Thus, the increase in the rate of heat generated by increasingly powerful electronics forces the designers to search for alternative coolants with better heat transfer characteristics. The liquid coolant owns a better heat transfer ability than the gas coolant. Therefore, besides water methyl alcohol [17], R134a [18], R410a [19], FC-77 [20], FC-72 [21], HEF-7100 [22], and other liquid coolants were also adopted to cool the MCHS. Compared with the above liquid coolants, nanofluids were found to have many advantages such as larger thermal conductivity and higher thermal capacity. Hence, many studies focused their attentions on the nanofluid-cooled MCHS. Chein and Huang [23] investigated the heat transfer performance of nanofluid-cooled MCHS. Their results showed that nanofluid-cooled MCHS yielded a better heat transfer performance than water-cooled MCHS, and the enhancement was explained by the increase in thermal conductivity of nanofluid and the nanoparticle thermal dispersion effect. Subsequently, various nanofluids prepared by diverse base fluids and nanoparticles were employed as coolant of MCHS, for example, Cu/water [24], Al_2O_3/water [25,26], CuO/water [26], diamond/water [27], diamond/ethylene glycol [27], diamond/oil, and diamond/glycerin [27] nanofluids. These studies [24–27] demonstrated that the heat transfer capability of MCHS can be improved when nanofluids are used. Recently, Gunnasegaran et al. [28] conducted a comprehensive study on nanofluid-cooled MCHS. Three different channel shapes and various volume fractions and types of nanoparticles were examined. An inverse geometric optimization for the nanofluid-cooled MCHS was performed by Wang et al. [29] at a constant pumping power. Their results, however, indicated that the use of nanofluid does not always lead to a better MCHS performance than the use of pure water. Similar results were also reported by Escher et al. [30] and Gunnasegaran et al. [28]. The performance deterioration was mainly attributed to the strong increase in dynamic viscosity with nanoparticle loading [28–30]. In Wang et al.'s another work [31], the same optimizations were carried out at three different constraint conditions: constant pumping power, constant inlet volumetric flow rate, and constant pressure drop across the heat sink. Their results indicated that the optimal geometric structure of MCHS strongly depended on the constraint condition.

Although the nanofluid-cooled MCHS has been found to have the superior cooling performance at lower nanoparticle loadings as compared with the conventional liquid-cooled MCHS, the microchannel may be clogged by nanoparticles when the MCHS operates for a long time. Therefore, it is still needed to find new coolants for MCHS. Carbon dioxide is considered as a major alternative refrigerant of this century for automotive air-conditioners and heat pump systems due to its prominent thermodynamic, transport, and environmentally-benign properties [32]. The viscosity of carbon dioxide becomes very low while its specific heat becomes very large near the critical point. The thermophysical properties of carbon dioxide are so outstanding that it can be used in many technical applications. Though the pressure of carbon dioxide is really high, but the carbon dioxide may be a good choice if the designers want to achieve a better MCHS performance without considering the cost.

In this work, transcritical carbon dioxide is employed as the coolant of MCHS. The use of terminology of transcritical CO_2 is based on the fact that CO_2 may go from the subcritical to the supercritical region or vice versa due to its temperature and pressure changes during its passage through the heat sink. It is expected to further reduce the overall thermal resistance and improve the temperature uniformity on the bottom surface of MCHS. A three-dimensional solid–fluid conjugate heat transfer model with the temperature-dependent coolant thermophysical properties is built to investigate the cooling performance of CO_2-cooled MCHS. The simulations are firstly carried out at various inlet temperatures ranging from 285 to 305 K to examine the effectiveness of the CO_2-cooled MCHS, and then the mechanism of the performance improvement is revealed. Subsequently, the performances of the MCHSs cooled by CO_2 and water are compared at various numbers