International Journal of Heat and Mass Transfer 101 (2016) 656–666

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Enhancement thermodynamic performance of microchannel heat sink by using a novel multi-nozzle structure

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ARTICLE INFO

Article history:
Received 26 January 2016
Received in revised form 7 April 2016
Accepted 30 April 2016

Keywords:
Channel length
Multi nozzle microchannel heat sink
High heat flux

ABSTRACT

In the present study, a novel multi-nozzle microchannel heat sink (MN-MCHS) was proposed. The channel length, channel aspect ratio, rib width, pumping power, and heat flux were numerically investigated in detail. It was found that the MN-MCHS with a shorter channel length not only could significantly improve the temperature uniformity on the bottom wall and thermodynamic performance index, but it also could significantly reduce the overall thermal resistance. With the decrease in the channel length from 10 mm to 1 mm, the temperature uniformity was enhanced by approximately 10 times, the overall thermal resistance improved 62% and the pressure drop was reduced approximately 12 times. For all cases in this study, the optimal structure of the MN-MCHS could dissipate a heat flux up to 1300 W/cm^2, and kept the temperature rising above the inlet coolant temperature under 77.5 °C. In addition, at the same pumping power, it could improve the overall thermal resistance up to 62% and 47.3% compared to that of the single layer MCHS and the double layer MCHS, respectively. This structure of MN-MCHS is really a promising structure of MCHS because it can improve thermal performance and reduce the pressure drop by optimizing its geometric dimensions.

1. Introduction

With the rapid development of scientific technology, air-cooling heat sinks (ACHS) have been gradually replaced by fluid-cooling heat sinks (FCHS) because the higher heat dissipation demands of electronic devices in a smaller volume have gradually exceeded the heat dissipation capacity of the ACHSs. A typification for FCHSs is the microchannel heat sink (MCHS) that was first proposed by Tuckerman and Pease [1] in the early 1980s. Since then, many investigations on MCHS have been conducted in many aspects, and a great number of studies have focused on single-layer and parallel microchannel heat sinks (SL-P-MCHS). Yu et al. [2] and Zhang et al. [3,4] reported investigations on the fractal-like MCHS. They found that the fractal tree-like MCHS had a much higher heat transfer coefficient than that of the straight microchannels, but it consumed a much higher pumping power. Azizi et al. [5] and Nebbati et al. [6] reported investigations on convection heat transfer of the MCHSs using Cu–water and Al_2O_3–water nanofluids, respectively. Azizi et al. [5] found that nanofluids not only could increase the average and local Nusselt numbers but could also decrease the bottom surface temperature. Sakanova et al. [7] reported a study on wavy MCHS using nanofluids. Their research revealed that the wavy MCHS could improve heat transfer performance compared to that of the straight MCHS. Especially, the wavy MCHS would work more efficiently than with the nanofluids. Leng et al. [8] studied porous fins MCHS, and they reported that the pressure drop on the porous fins MCHS could decrease remarkably compared to that of the solid fins. Chu et al. [9] studied the triangle channel shape, and Wong et al. [10] studied the triangle fin shape. They reported that a high temperature gradient existed in the regions between the inlet and outlet of the triangle channel [9]. The heat transfer rate increased when the rib width or rib height increased, but it decreased when the rib length increased [10]. Square and circle channel shapes were investigated by Normah et al. [11]. They concluded that at the same hydraulic diameter and pumping power, the thermal resistance of the circle MCHS was lower than that of the square MCHS. The thermal performance of MCHS with internal Y-shaped bifurcations was much better than that of the rectangular MCHS, as reported by Xie et al. [12]. Although the high heat flux and small thermal resistance of SL-P-MCHS were proven in published literature, the nonuniform bottom-wall temperature and

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2016.04.111
0017-9310/

Nomenclature

ACHS	air-cooling heat sinks
c_p	specific heat at constant pressure (J/kg K)
B_{th}	bottom thickness (m)
D_c	channel depth (m)
FCHS	fluid-cooling heat sinks
H_t	top cover height (m)
H_s	substrate height (m)
L_c	channel length (m)
L_m	length of computational model (m)
L_n	nozzle length (m)
L_s	substrate length (m)
L_t	top cover length (m)
M	mass flow rate (kg/s)
MCHS	microchannel heat sink
N_c	channel number of a column
N_r	channel number of a row
N_t	total number of the channels on the heat sink
ΔP	pressure drop (Pa)
Q	dissipated power (W)
q	heat flux (W/cm^2)
R	thermal resistance (°C/W)
T	temperature (°C)
W_c	channel width (m)
W_m	width of computational model (m)
W_r	rib width (m)
W_s	substrate width (m)

Greek symbols

α	channel aspect ratio
δ	ratio between rib width and channel width
η	thermodynamic performance index
μ	dynamic viscosity of the coolant (kg/m s)
ρ	density of the coolant (kg/m^3)
Ω	pumping power (W)

Subscripts

bt,max	bottom maximum
i	inlet
o	outlet
T	total
w	wall

high pressure dropped on the channels of the SL-P-MCHSs were still barriers for wide application. To overcome the challenge of nonuniform bottom-wall temperatures of the heat sink, Vafai and Zhu [13] first proposed a concept of double-layer MCHS (DL-MCHS) in 1999. Eight years later, it was reintroduced by Wei et al. [14] with the thermal resistance improved up to 0.09 °C/Wcm^2. Recently, DL-MCHSs were continuously investigated by other groups of authors such as Hung et al. [15–17], Leng et al. [18–20], Lin et al. [21], Rajabifar et al. [22], Wu et al. [23], and Zhai et al. [24]. The channel inlets of the upper layer of a DL-MCHS were designed on the top of the channel outlets of the lower layer; therefore, the temperature in the lower-layer-outlet region would be reduced compared to that of the single layer leading to the temperature on the bottom wall being more uniform. Sakanova et al. [25] reported an optimization and comparison of double-layer and double-side micro-channel heat sinks with nanofluid as the coolant. They revealed that the thermal resistance of the heat sink with sandwich structure and counter flows was much smaller than those of the single-layer and the double-layer with unidirectional or counter flow. A truncated structure of the upper layer channel of the DL-MCHS was proposed by Leng et al. [19]. This truncation could improve the temperature uniformity on the bottom wall. Although the channel length of the upper layer was shorter than the original, the channel length of the lower layer was not changed compared to the original. This indicates that with the same hydraulic diameter and channel length, the pressure drop of the lower-layer channels of the DL-MCHS was approximately the same as that of the single layer MCHS. This means that the DL-MCHS could not reduce the pressure drop compared to that of the single layer. Furthermore, Zhai et al. [24] suggested that DL-MCHSs should not be used for cooling microelectronic equipment under a small volumetric flow rate due to a larger irreversibility. In 2012, Boteler et al. [26] were first to propose the concept of a manifold microchannel heat sink (M-MCHS). Following that proposition, the investigations on the M-MCHS were continued by other groups of authors [27–29]. They reported that the M-MCHS could significantly reduce the pressure drop and improve the temperature uniformity on the bottom wall due to the truncation of the path of the coolant. Yin et al. [30] proposed a structure of heat sink with AlN-base micro-channels in direct bond copper. They reported that the thermal resistance of heat sink with the proposed structure decreased by 15% and 80% compared with that of the conventional packaged structures with Cu-based MCHS being bonded to the direct bond copper by solder or thermal interface material, respectively. Based on the foundations of the reviewed literature, a novel multi-nozzle microchannel heat sink (MN-MCHS) is presented in the present work to improve the thermal performance of the heat sink, reducing the pressure drop along the channel and enhancing the uniformity of the bottom wall's temperature.

2. Methodology

2.1. Geometric design of a novel multi-nozzle microchannel heat sink

The schematic of a novel multi-nozzle MCHS is presented in Fig. 1a. The blue and red parts in Fig. 1a are the inlet and outlet channel nozzles, respectively. The gray part on the upper layer is the top cover, and the lower gray part is the substrate. With this structure of a heat sink, the channels can be arrayed on the substrate following both row and column directions. The number of channels can be calculated for a row, N_r, a column, N_c, or a total number of the channels on the heat sink, N_t, respectively, as follows:

$$N_r = (W_s - W_r)/(W_c + W_r) \tag{1}$$

$$N_c = (L_s - W_r)/(L_c + W_r) \tag{2}$$

$$N_t = N_r * N_c \tag{3}$$

where W_s, L_s, W_r, W_c and L_c are the substrate width, substrate length, rib width, channel width, and channel length, respectively. Fig. 1b presents a unit cell of the MN-MCHS which includes parts of the MN-MCHS such as the substrate, channel, top cover, inlet and outlet. This unit cell would be used as a computational model for analysis. Fig. 1c presents the dimensions of the MN-MCHS. Fig. 1d presents the dimensions of the unit cell. With the original parallel MCHS, supposedly, the flow is laminar; the coolant will flow along the channel from the channel inlet to the channel outlet without changing direction. However, in the MN-MCHS, the inlet and outlet nozzles are located on the top of the channel, as shown in Fig. 1b; therefore, the flow direction of the coolant will be