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A study on five different channel shapes using a novel scheme for meshing and a structure of a multi-nozzle microchannel heat sink

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In this study, a copper plate measuring 9.8 mm × 9.8 mm × 0.5 mm was used as a fixed substrate for designs with single-layer-and-parallel or multi-nozzle microchannel heat sinks. Water was applied as the coolant. Channel lengths from 0.2 to 5.6 mm and five different channel shapes, including a circle, square, trapezium, two concave surfaces, and two convex surfaces, were numerically investigated in detail at a constant hydraulic diameter of 200 μm with a Reynolds number in the range of 700–2200. A novel scheme for meshing was proposed. A structure for a multi-nozzle microchannel heat sink was presented. For all cases in this study, it was found that the best thermal performance was achieved with a circular channel shape which could dissipate a heat flux up to 1500 W/cm^2, and the maximum temperature was kept at less than 75 °C at a Reynolds number of 2200. Furthermore, novel equations were proposed to predict the temperature differences between inlet and outlet coolant temperatures depending on the channel length and Reynolds number, as well as to predict the maximum temperatures on the bottom walls of the circular channel shape depending on the Reynolds number and heat flux.

1. Introduction

Requirements for the heat dissipation capacity and maximum working temperature for each applied field were different. For example, the maximum power flux for chips, reported by Phillips [1], was around 100 W/cm^2, and the maximum working temperature for this field was about 100 °C. The VLSI industry requires a heat flux up to 10^3 W/cm^2 and a maximum working temperature of 125 °C, as reported by Mudawar [2], and Lee and Mudawar [3], respectively. These demands have grown every single day and have gradually exceeded the heat dissipation capabilities of air-cooling heat sinks; therefore, fluid-cooling heat sinks have been employed. In 1981, Tuckerman and Pease [4] first proposed a microchannel heat sink (MCHS) for VLSI with a heat flux of 790 W/cm^2. Since then, the MCHS has been investigated in many aspects. Single-layer and parallel microchannel heat sinks (SL-P-MCHS) have been found in a great number of investigations in the MCHS field. This may divide ideas in published literature for enhancing thermal performance of a microchannel heat sink into two main solution groups as follows: (1) using substrate materials and coolants with higher thermal conductivity and (2) creating optimal structures for heat sinks in general and for channels, manifolds or inlet–outlets in particular. The influences of substrates' thicknesses and the type of materials on heat transfer in MCHS were investigated by Kosar [5]. Kosar found that substrate-thermal conductivity affected the change in thermal resistance much more than the effect of the substrate's thickness. For increasing thermal conductivities of the coolant, Choi and Eastman [6] first used the term, nanofluids, in 1995. Two year later, Eastman et al. [7] reported that a nanofluid, including water and 5% CuO nano-particles, could improve thermal conductivity by approximately 60% compared to water alone. For the geometric structure of MCHS, Zhang et al. [8,9] and Yu et al. [10] reported investigations on fractal-like MCHS. They found that fractal tree-like MCHS has a much higher heat transfer coefficient than that of straight microchannels, but it requires a much higher pumping power. Leng et al. [11] studied porous fin MCHS, and their research revealed that the pressure drop on porous fin MCHS could decrease remarkably compared to solid fins. Circular and square channel shapes were investigated by Normah et al. [12]. They concluded that at the same hydraulic diameter and pumping power, the thermal resistance of a circular MCHS was lower than that of a square MCHS. Wong et al. [13] studied the triangular fin shape, and Chu et al. [14] studied the triangular channel shape. They reported that the heat transfer rate increased with the increase in the rib width or height, but it decreased with the increase in the rib length [13]. A high temperature gradient was found in the

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Nomenclature

a	length of the top channel edge, (m)
A_c	channel cross-sectional area, (m)
A_{ch}	heat transfer area of channel, (m)
A_{bt}	bottom wall area of the computational domain, (m)
b	length of the bottom edge of trapezoidal channel, (m)
c_p	specific heat at constant pressure, (J/kg K)
D_h	channel hydraulic diameter, (m)
h	height of the channel, (m)
L_{cmax}	the maximum channel length in this study, (m)
L_x	channel length at x location
MCHS	microchannel heat sink
ΔP	pressure drop on the channel, (Pa)
Q	heat transfer rate, (W)
q	heat flux, (W/cm^2)
r	radius of arcs in two concave and two convex channels, (m)
R_T	thermal resistance, (°C/W cm^2)
Re	Reynolds number
T_{btmax}	maximum temperature on the bottom wall, (°C)
T_{w_inlet}	inlet water temperature, (°C)
ΔT_r	temperature rise of the bottom wall above the inlet water temperature, (°C)
$\Delta T_{i,o}$	temperature difference between inlet and outlet fluid, (°C)

Greek symbols

γ	ratio between the top and bottom trapezoidal channel
μ	dynamic viscosity of the coolant (kg/ms)
ρ	density of the coolant (kg/m^3)

Subscripts

i,o	inlet_outlet
btmax	bottom maximum
cmax	channel maximum

region between the inlet and outlet of the triangular channel [14]. Xie et al. [15] reported that the thermal performance of MCHS with internal Y-shaped bifurcations was much better than that of the rectangular MCHS. The high heat flux and low thermal resistance of SL-P-MCHS were proven in published literatures; however, barriers for wide application of the SL-P-MCHS were non-uniform bottom-wall temperatures and high pressure drops in the channels. To surmount the problem of non-uniform bottom-wall temperatures of the heat sink, in 1999 Vafai and Zhu [16] first proposed a concept of a double-layer MCHS (DL-MCHS). Eight years later, this was reintroduced by Wei et al. [17] improving thermal resistance up to 0.09 °C/W cm^2. Recently, DL-MCHSs have been investigated by Leng et al. [18,19], Hung et al. [20–22] and other groups of authors, such as Sakanova et al. [23], Rajabifar et al. [24] and Wu et al. [25]. The channel inlets of the upper layer of a DL-MCHS were designed on top of the channel outlets of the lower layer; therefore, reducing the temperature in the lower-layer outlet region compared to that of a single layer. This could lead to the temperature on the bottom wall being more uniform. A truncated structure of the upper layer channel of the DL-MCHS was proposed by Leng et al. [18]. This truncation could significantly improve temperature uniformity on the bottom wall. The channel length of the upper layer was shorter than the original; however, the channel length of the lower layer was unchanged. This indicated 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 a single layer MCHS. This means that the DL-MCHS could not reduce the pressure drop compared to a single layer. Furthermore, Zhai et al. [26] 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. [27] were the first to propose a concept of a manifold microchannel heat sink (M-MCHS). Following that proposal, investigations of M-MCHS were continued by other groups of authors [28–30]. They reported that the M-MCHS could significantly reduce the pressure drop and improve temperature uniformity on the bottom wall due to truncation of the path of the coolant. Recently, Tran et al. [31] proposed a multi-nozzle microchannel heat sink. They revealed that the optimal structure of a multi-nozzle microchannel heat sink could dissipate a heat flux up to 1300 W/cm^2 and keep the temperature rising above inlet coolant temperature to under 77.5 °C. Based on the foundations of the reviewed literature, it was found that studies on channel shapes are still needed to improve thermal performance and pressure drop of MCHS. In addition, it was also found that applications of simulated results, to predict the heat transfer and fluid flow in a heat sink, are economic and flexible; however, simulated results are more accurate if the meshing scheme is managed better. The mentioned demands are motivation for us to investigate influences of various channel shapes on heat transfer and fluid flow of a multi-nozzle microchannel heat sink with the aim to improve the uniformity of the bottom wall's temperature, increasing heat flux and decreasing the pressure drop of the heat sink. The presented multi-nozzle microchannel heat sink is expected to be widely applicable in the VLSI industry in the near future.

2. Methodology

2.1. Work description

To examine the above-mentioned ideas, a copper plate with bottom dimensions of 9.8 mm × 9.8 mm and a thickness of 0.5 mm was considered as a fixed substrate for the design. The first design for this substrate was an MCHS including 22 parallel microchannels and two manifolds, as shown in Fig. 1a and c. The channels were designed with a square cross-section area of 200 × 200 μm and a length of 5.6 mm; two manifolds were designed with a width of 1.5 mm, a depth of 200 μm and a length of 8.6 mm. To analyze the thermodynamic processes in the heat sink, a CFD-ACE+ software package was employed, and a unit cell with a width of 0.4 mm, a length of 5.6 mm and a thickness of 0.7 mm was used as a computational domain. The domain included parts of the substrate, channel and top cover, as shown in Fig. 1b. To examine the influences of the channel length on heat transfer and fluid flow in MCHSs, channel lengths from 0.2 mm to 5.6 mm were numerically investigated in detail. Their results are presented in Section 3.1. To examine the effects of channel shapes on the thermodynamic properties of the heat sink, five different channel shapes, including a circle, square, trapezium, two concave surfaces and two convex surfaces, were carefully investigated. Their results are presented in Section 3.2, and their shapes are shown in Table 1. It is noted that for all cases in this study, the channel's hydraulic diameter, D_h, and channel height, h, were kept at a constant value of 200 μm. However, the top edges, a, and the two surface radiuses, r, were altered with different values based on the constraints, as