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Parametric numerical study of the flow and heat transfer in microchannel with dimples☆

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A R T I C L E I N F O

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A B S T R A C T

The characteristics of flow and heat transfer in microchannel with dimples were numerically investigated. The geometric parameters of dimpled channel, including aspect ratio, dimple depth and dimple spacing, were independently studied under constant Reynolds number 500. A constant heat flux 1 W/mm^2 was adopted in the central area at the bottom of the microchannel heat sink to simulate a high power device. In comparison to straight channels, dimpled surface reduced the local flow resistance and also improved thermal performance of micro-channel heat sink. Compared to flat channel case, the optimal dimpled case has 3.2 K decrease of temperature, 15% gain of Nusselt number and 2% reduce of pressure drop.

1. Introduction

Miniaturizations of integrated circuits (ICs) are facing many difficult challenges including packaging, line width, and materials. One of the most serious challenges is to manage the high thermal power density and to control temperature. According to the 2012 International Technology Roadmap for Semiconductors (ITRS), the maximum power density of a single-chip with high performance and multiple purpose units (MPU) will reach 1 W/mm^2 by the end of 2020 [1].

Traditional forced air convection cooling cannot manage such high heat flux without large radiators and powerful fans; which may result in unacceptable noise level and difficulty in system integration. Liquid cooling method with microchannels, separating the heat absorption from ICs and heat dissipation to air, has higher efficiency and high degree of integration. It will be an important option for cooling the next generation electronics.

In 1981, Tuckerman and Pease [2] built the first microchannel heat sink and proved its huge potential for IC cooling. Since their pioneering work, many studies on friction factors and Nusselt numbers in flat channels have been reported [3–8]. These works showed that the Navier–Stokes equations are still valid for these micron level channels. The axial heat transfer, surface roughness, and viscous heating are also negligible. Wavy channel [9–12], branched channel [13–15], ribs, fins or rough elements [16–19] (such as triangular, rectangular, dimple elements and so on) have also been investigated, the rough elements method can strengthen the convection heat transfer without large pressure penalty and is a possible low-cost solution.

Jonghyeok Lee and Kwan-Soo Lee [20] studied plate heat exchanger with dimples and protrusions. The Reynolds numbers were varied from 500 to 15,000. A genetic algorithm was used to determine the optimal dimple and protrusion shape. The optimal design enhances the performance factor by as much as 28% and is independent of Reynolds number. Alshroof et al. [21] investigated laminar flow (Reynolds number of 1600) and heat transfer of single dimple, single protrusion and their combinations. With a single dimple, a 9% increase in the performance factor was obtained. This was achieved with a 7% heat transfer enhancement and a 1.5% decrease in the average shear stress. With a single protrusion application, the performance factor increased by 32% even with higher average shear stress. Bi et al. [22] numerically studied the heat transfer in 1 mm hydraulic diameter channel with dimples, cylindrical grooves and low height fins for Reynolds numbers from 2700 to 6100. It was concluded that, when the Reynolds number is larger than 3323, the dimpled channel has the best overall performance. They also concluded that the dimple diameters and spacings can be optimized to increase performance. Wei et al. [23] studied steady laminar flow and heat transfer inside a microchannel with one dimple using periodic boundary conditions, the results show that the dimple could reduce the pressure drop under low Reynolds number. Lan et al. [24] studied rectangular microchannel with dimples and protrusions with Reynolds number from 100 to 900, for the dimple-only case they also find the dimple could reduce the reduce the pressure drop under low Reynolds number.

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Nomenclature

α	channel aspect ratio
ρ	density, kg/m^3
λ	thermo conductivity, W/m·K
μ	dynamic viscosity, Pa·s
ϕ	viscous dissipation rate, W/mm^3
ν	kinematic viscosity, m^2/s
λ	thermo conductivity, W/m·K
Δp	pressure drop, Pa
A_c	inlet cross-sectional area of the flow channel, mm^2
A	area, mm^2
c_p	isobaric heat capacity, J/K·kg
d	dimple depth, mm
D_e	equivalent diameter of channel, mm
H	heat sink height, mm
h	convective heat transfer coefficient, W/m^2·K
H_c	channel height, mm
H_w	channel width, mm
h_w	local convective heat transfer coefficient, W/m^2·K
L	channel length, mm
L_h	length of heating source, mm
Nu	Nusselt number
p	pressure, Pa
P	perimeter of the flow channel at inlet, mm
PF	performance factor
Q	total heat power of heating surface, W
q	heat flux, W/mm^2
r	dimple radius, mm
R	thermal resistance, K/W
Re	Reynolds number, $U_{in} \cdot D_e/\nu$
s	dimple spacing, mm
T	temperature, K
U	velocity magnitude, m/s
u, v, w	dimensionless velocity components in x, y and z directions
x, y, z	Cartesian coordinates

Subscripts

0	flat channel
f	fluid
heat	heating region
in	inlet
max	maximum
out	outlet
P	point P on fluid solid interface
s	solid
wall	fluid solid interface
x	slice at x location

Ge et al. [25] studied the cosine-shaped dimpled heat sink with Reynolds number from 50 to 3000. They concluded that transition from laminar to turbulent flow begins at Reynolds number of 1000 and the flow becomes fully turbulent around Reynolds number of 3000. Lienhart et al. [26] reported experimental measurements and numerical simulations for turbulent flow over shallow (dimple depth to dimple print diameter ratio of 0.05) dimpled surfaces. Both in internal and external flow, the drag force consists of shear stress and pressure force didn't decrease. Though the shear stress was marginally decreased, the pressure force caused by dimples overcame these reduction, dimples couldn't decrease the flow resistance. Turnow et al. [27] carried out experimental measurements and large eddy simulations on staggered dimpled surfaces for Reynolds numbers of 6521 and 13,042. 93% heat transfer enhancement and 14% decrease in pressure drop was the best overall performance obtained with dimple depth to dimple print diameter (diameter of dimple edge) ratio of 0.26. Isaev et al. [28] presented numerical solutions for single dimple in narrow channel. The Reynolds number was varied from 20,000 to 60,000. The average Nusselt number enhancement was independent of Reynolds number. When both heat transfer enhancements and pressure losses were considered, the overall performance was better at lower Reynolds numbers. Tay et al. [29] conducted experiments on dimpled microchannels with dimple depth to dimple diameter ratios from 0.05 to 0.5. Flows were visualized by injecting dye into the flow passages. The Reynolds number was varied from 1000 to 28,000. Six different flow stages were observed when Reynolds number was growing. The flow stages development achieved under lower Reynolds number as the dimple's depths increase.

Silva. et al. [30] reviewed published literature on heat transfer enhancement with dimples for cooling of microelectronics. They concluded that heat transfer enhancements (1) are independent of the Reynolds number in laminar and transition flows, (2) are better at low Reynolds number due to the small pressure drop, (3) are better with shallow dimple (dimple depth to dimple diameter ratio of 0.2) and (4) can be optimized by properly spaced dimples placements.

Xie et al. [31] presented numerical solutions of turbulent flows through microchannels with tear drop dimple and protrusion. It was concluded that tear drop dimple and protrusion enhanced heat transfer better than hemispherical dimple or protrusion, albeit with higher flow resistance. Yoon et al. [32] reported results from direct numerical simulations of flow through microchannels with tear drop dimple for a Reynolds number of 2800. It was concluded that a 4% heat transfer enhancement was obtained with the use of tear drop dimples.

From the above, majority of the available studies focused on turbulent flows, transition flows or higher spectrum of laminar flows. Microchannel heat sink used for electronic cooling must be highly reliable. As the leakage of fluid or failure of structure will damage the electronic or even the whole system. Thus properly designed cooling arrangements with laminar flow which has low pressure drop and higher efficiency may be more suitable for electronic cooling. The objective of this article is to study the effectiveness of cooling facilities utilizing dimples to enhance performance. The flow is kept laminar with a Reynolds number of 500. The effects of channel aspect ratios, dimple sizes and dimple arrangements are numerically studied to find the effect of dimple and a proper way to control the maximum temperature of electronics.

2. Geometry model

Fig. 1 shows the schematic of the microchannel heat sink considered in this article. Cooling is provided by flowing water through the fluid zone in the 1 mm × 1 mm × 20 mm (outer dimensions) heat sink with dimples along the bottom of the fluid zone. There is a 10 mm heat source (Fig. 1b) in the middle of the bottom surface with a constant heat flux 1 W/mm^2. As a result, the 5 mm upstream and 5 mm downstream sections of the bottom wall are unheated. The first dimple is placed 2 mm from the entrance and there are a total of 9–21 dimples in the channel depending on dimple spacing s.

The effects of the flow area aspect ratios (Fig. 1c), the dimple spacings (Fig. 1d) and the dimple depths (Fig. 1e) are examined in this article. Four flow area aspect ratios (width/height) namely, 1, 2, 3 and 4 are considered as shown in Fig. 1c. Dimple spacings s of 0.7 mm, 1.0 mm, 1.2 mm, 1.6 mm and 2.8 mm are studied. Dimple depths d of 0.05 mm, 0.1 mm, 0.15 mm and 0.2 mm are evaluated. The dimple diameter is kept at 0.2 mm.