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Effect of pumping power on the thermal design of converging microchannels with superhydrophobic walls

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ABSTRACT

This paper investigates the thermal efficiency of converging microchannels with superhydrophobic walls under various coolant flow rates. To enhance the thermal performance of microchannels, geometric optimization is performed for different values of pumping power, Ω, ranging from 0.01 up to 10 Watts. For this purpose, Navier-Stokes and energy equations with slip boundary conditions are solved for various design points by the Finite Volume Method. Response functions for pumping power and thermal resistance are then constructed. Finally, desirability function approach is employed to minimize the total thermal resistance for a fixed pumping power to give the optimum design set. It is found that for each value of Ω, by the use of superhydrophobic walls, optimal number of channels (N), width-tapered ratio (Λ_z) and height-tapered ratio (Λ_y) increase, whereas the optimal width ratio (β) decreases. It is also found that with an increase in pumping power, the optimal N increases, whereas the optimal values of β, Λ_z, and Λ_y decrease. Finally, it is shown that the thermal performance enhancement by the use of superhydrophobic walls is only achieved for a low pumping power (Ω < 0.5 Watts), and conversely, for high values of Ω (> 5 Watts), thermal performance is deteriorated by their use.

1. Introduction

Water-cooled Microchannel Heat Sink (MCHS) has been regarded as one of the most prominent means of cooling microelectronic devices, ever since its superb heat dissipation capability was pointed out by Tuckerman and Pease [1]. Using the liquid as coolant, however, imposes a great flow pressure drop inside the microchannels. Therefore, novel approaches are needed to facilitate fluid flow in microchannels and increase the efficiency of MCHSs by reducing the pressure drop and accompanying required pumping power. Compared to straight microchannels, converging microchannels are shown to considerably reduce the pumping power requirement when a prescribed cooling capacity is to be achieved [2]. Superhydrophobic walls are also suggested to reduce the excessive pressure drop when water flows through microchannels [3]. Hence, the combined effects of the convergence of microchannels and the use of superhydrophobic walls on the overall performance are investigated in our study.

Many previous works have proposed methods to enhance the thermal performance in a heat sink, for instance by mounting ribs (in forms of fans, rectangular, triangular and ellipsoidal) on the sidewalls of the microchannels [4], inserting micro pin-fins (in forms of circles, rectangles, and diamonds) in the microchannels [5,6], or by using channels filled with phase change materials [7], wavy microchannels [8,9], porous-filled microchannels [10], transverse microchannels [4,11], impinging jets [12], and coolants different from pure water [9,13]. Although numerous studies have attempted to enhance the overall performance by improving the heat transfer resistance, limited studies have targeted overall performance enhancement in microchannels by decreasing frictional and thermal resistances simultaneously. Xu et al. [14] proposed a novel MCHS, in which, in addition to parallel longitudinal microchannels, several transversal microchannels were also included. They succeeded in reducing both the pressure drop and thermal resistance, and were able to enhance the overall performance of the heat sink considerably. They attributed this enhancement to the repeated occurrence of thermal boundary layer development. As an improvement to the conventional plate type condenser which induced very large pressure drops, Yu et al. [15] put forward a novel micro condenser with varied cross section of gas and liquid phases. Micro structures were also included to ensure a sustainable phase separation. They found that this method is able to enhance both hydraulic and thermal performances of the condenser. Pressure drop reduction was attributed to the decrement in the interfacial area between gas-liquid phases. Xu et al. [16] proposed a novel "gradient-porous-wall" microchannel heat sink in which porous walls with varied populated pin-fins were used. This concept allows the vapor to flow into the bare channels by means of surface tension force. Also, by creating the so-

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Nomenclature

A	heat transfer area (m^2)
C_1,C_2	constants used in Eq. (18)
C_p	specific heat (J kg^{-1} K^{-1})
d_R	thermal resistance desirability function
d_Ω	pumping power desirability function
D	overall desirability function
h	convective heat transfer coefficient (W m^{-2} K^{-1})
H	channel height (m)
k	thermal conductivity (W m^{-1} K^{-1})
l_S	slip length (m)
L_x	length of the heat sink (m)
L_y	height of the heat sink (m)
L_z	width of the heat sink (m)
n	wall normal vector
N	number of channels
P	pressure (Pa)
q	heat flux (W m^{-2})
Q	volumetric flow rate (m^3 s^{-1})
R	thermal resistance (K W^{-1})
R_S	apparent slip thermal resistance in Eq. (15)
T	temperature
u	velocity component in x direction (m s^{-1})
$\vec{V}$	velocity vector (m s^{-1})
W	channel width (m)

Greek symbols

α	channel aspect ratio
β	channel width ratio
δ	substrate thickness (m)
Δp	pressure drop (Pa)
λ	axial conduction parameter
Λ_y	height-tapered ratio (−)
Λ_z	width-tapered ratio (−)
μ	dynamic viscosity (kg m^{-1} s^{-1})
ξ_S	temperature jump coefficient (m)
ρ	density (kg m^{-3})
Ω	pumping power (W)

Subscripts

cond	conductive
conv	convective
f	fluid
fin	fin region of the heat sink
in	inlet
m	mean
max	maximum
out	outlet
s	solid
T	total
w	wall

called “eye-blinking” motion, they were able to further decrease the pressure drop through microchannels. They concluded that this new design is also able to significantly minimize the thermal oscillations and the resulting thermal stresses, bringing about a reliable and efficient cooling method.

Another class of heat sinks which has proven to increase the overall performance of cooling, is by employing converging microchannels instead of straight microchannels. Hung et al. [17] compared the thermal performances of optimized single-layer, double-layer, and converging MCHSs, and concluded that the latest case exhibited the best thermal performance. Dehghan et al. [2] showed that converging microchannels can improve the thermal performance of a heat sink by forcing the coolant to obtain a velocity component toward the center of the channel which increases the heat transfer coefficient. It is also shown that by the use of microchannels with a width-tapered ratio (channel outlet-to-inlet width ratio) of 0.5, instead of straight microchannels, the required pumping power reduces by 25% for the same cooling capacity. Hung and Yan [18] proposed an optimized design of a tapered microchannel heat sink in which the overall thermal resistance of the heat sink reduces nearly 40% in comparison with an optimized straight microchannel. They also showed that pumping power can greatly affect the optimal geometry of a micro heat sink. Osanloo et al. [19] suggested using two layers of converging microchannels, one on top of the other, with water entering each layer in the opposite direction. It was shown that with an optimum convergence angle of 4°, a significant reduction in thermal resistance and temperature gradient along the bottom wall occurs with small pressure drop penalty. Duryodhan et al. [20] experimentally and numerically studied the liquid flow through straight, converging, and diverging microchannels for Reynolds in the range of 30–274 and heat flux in the range of 0.3–9.5 W cm^{-2}. They found that the heat transfer coefficient in a converging microchannel is the highest among all tested microchannels. In fact, the value of heat transfer coefficient (h) for a converging microchannel was found 35% larger than a diverging one. They concluded that converging microchannels have the lowest ratio of non-dimensional pumping power to the thermal resistance among all considered channels.

To overcome the great pressure drop accompanying the fluid flowing in a microchannel, superhydrophobic walls (hydrophobic surface with hierarchical structures) are suggested to replace conventional (uncoated and smooth) walls [3]. Superhydrophobic walls reduce the liquid-solid interfacial drag by trapping air in the hierarchical structures. Trapped air in the textures reduces the liquid-solid interfacial areas, as water is unable to enter the cavity zones and very high contact angles may be achieved reaching up to 177° [21]. The presence of this so-called “air cavities” was experimentally observed by Ruckenstein and Rajora [22] and Churaev et al. [23], when a liquid flows over a superhydrophobic surface. Slip length, l_S, analogous to the slip coefficient proposed by Navier [24], is commonly used to address the liquid-solid interfacial slip. Watanabe et al. [25] observed liquid slip at the hydrophobic walls of a rectangular minichannel and concluded that Navier's proposed boundary slip could adequately predict the velocity profile obtained by their experiment. Tretheway and Meinhart [26] obtained the same results for a hydrophobic microchannel and pointed out the significant deviation from the no-slip boundary condition, with the fluid flow adjacent to the surface reaching a velocity as high as one tenth of the centerline velocity corresponding to the slip length of 1 μm. Slip length of about 20 μm was found by Ou et al. [3] resulting in a pressure drop reduction of as high as 40%. It is noteworthy that large slip length of 400 μm is also reported in the literature [27].

Various parameters can affect the effective slip length for flows over a superhydrophobic surface. Voronov et al. [28] studied the effect of different flow and geometrical parameters, fluid-solid and fluid-fluid interactions on the resultant contact angle and slip length. They performed dimensional analysis and obtained a relation between the slip length and contact angle based on the mentioned parameters. Recent works on quantifying the relation between the contact angle and the associated mechanism of the liquid flow can be found in Refs. [29–31]. Slip length is usually reported in terms of texture pitch, shear-free (cavity) fraction, and relative module width (i.e. the ratio between the texture pitch and the hydraulic diameter) for different micro-texture arrangements and flow Re number [3,32]. For fluid flows through