International Communications in Heat and Mass Transfer xxx (2016) xxx–xxx

Contents lists available at ScienceDirect

International Communications in Heat and Mass Transfer

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

Experimental investigation for sequential triangular double-layered microchannel heat sink with nanofluids☆

Q2 Hamdi E. Ahmed [a,b,*], M.I. Ahmed [a], Islam M.F. Seder [a], B.H. Salman [c]

[a] Department of Mechanical Engineering, International Islamic University Malaysia, 50728, Gombak, Malaysia
[b] Department of Mechanical Engineering, University of Anbar, Ramadi 31001, Iraq
[c] Department of Mechanical Engineering, University of Nevada, Las Vegas, NV 89154, USA

A R T I C L E I N F O

Available online xxxx

Keywords:
Double-layer
Microchannel heat sink
Sequential triangular
Nanofluids

A B S T R A C T

This study examined new innovative design of aluminum rectangular and triangular double-layered microchannel heat sink (RDLMCHS) and (TDLMCHS), respectively, using Al_2O_3–H_2O and SiO_2–H_2O nanofluids. A series of experimental runs for different channel dimensions, different nanoparticles concentrations and types and several pumping powers showed excellent hydrothermal performance for DLMCHS over traditional single-layer (SLMCHS). The results showed that the sequential TDLMCHS provided a 27.4% reduction in the wall temperature comparing with RDLMCHS and has better temperature uniformity across the channel length with less than 2 °C. Sequential TDLMCHS provided 16.6% total thermal resistance lesser than the RDLMCHS at low pumping power and the given geometry parameters. Pressure drop observation showed no significant differences between the two designs. In addition, larger number of channels and smaller fin thickness referred less thermal resistance rather than only increasing the pumping power. Higher nanoparticle concentration showed better thermal stability for both nanofluids than pure water. The Al_2O_3–H_2O nanofluid (0.9 vol.%) showed best performance with the temperature difference of 1.6 °C and lowest thermal resistance of 0.13 °C/W·m^2.

1. Introduction

Thermal management is a vital part to maintain the functionality and durability of any successful engineering device. As the world goes toward smaller in size, cheaper devises, lighter in weight, and higher in efficiency, microelectronic mechanical systems enable an innovative cooling design to be used in miniature designs, especially electronic applications. Enhancing the performance of electronic chips requires utilizing more resistors which causes massive heat generated in these devices. This pushed researchers to adapt more efficient cooling system. In 1981, Tuckerman and Pease [1] utilized the concept of micro heat exchanger in electronic cooling, opening the door to a new mechanism in dissipating heat rather than the traditional low-thermal capacity of air heat sink. Another important factor in increasing the thermal performance of the cooling system is adapting a proper coolant. It has been proved that different fluids have different flow properties that strongly affect the overall performances. Recently, nanofluids have been used in some engineering applications in order to enhance the thermal conductivity of the based-fluids [2]. Heat transfer coefficient can be improved by increasing the ratio of channel surface area to channel volume and also the properties of fluids decide the abilities of fluid to remove heat by influencing the heat transfer coefficient. Thus, improvement in overall heat transfer performance must be concentrated on dimensions of the channel and type of fluids used in heat exchanger.

Two main challenging obstacles are still faced in this filed. First, the high pressure drop between the inlet and outlet of the channel which imposes using high pumping power. Second, the large difference in temperature distribution along the channel which increase thermal stresses in the cooled-electronic elements thus reduces the electrical performance due to the electrical–thermal instability and thermal breakdown. This pushed engineers to move from traditional single-layer microchannel to double or multi-layer microchannel. The non-circular geometry of microchannels are widely used such as rectangle, trapezoid, triangle, and diamond. Moreover, width of channel and width of fin are among critical parameters that are usually investigated since they strongly affect the thermals conductivity and convection coefficient of heat transfer.

Double-layered microchannel heat sink (DLMCHS) was suggested by Vafai and Zhu [3] to represent an innovative design that adds more features for cooling system in miniature devices where the pressure drop was deceased dramatically and the heat was better distributed along the MCHS length. Gunnasegaran et al., [4] showed that the smallest hydraulic diameter (D_h) of the MCHS provided better

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☆ Communicated by W.J. Minkowycz.

* Corresponding author at: Department of Mechanical Engineering, International Islamic University Malaysia, 50728, Gombak, Malaysia.
E-mail address: hamdi.ahmed@uoa.edu.iq (H.E. Ahmed).

http://dx.doi.org/10.1016/j.icheatmasstransfer.2016.06.010
0735-1933/© 2016 Published by Elsevier Ltd.

Please cite this article as: H.E. Ahmed, et al., Experimental investigation for sequential triangular double-layered microchannel heat sink with nanofluids, Int. Commun. Heat Mass Transf. (2016), http://dx.doi.org/10.1016/j.icheatmasstransfer.2016.06.010

Symbols

A	area, m^2
C_p	Specific heat capacity, J/kgK
D_h	hydraulic diameter, m
DW	distilled water
f	Fanning friction factor
g	gravitational acceleration, m/s^2
h	convection heat transfer coefficient, W/m^2·°C
H	channel height, m
I	current, A
k	thermal conductivity, W/m·°C
L, *l*	channel Length, mm
M	molecular weight, g
N	number of channel
Nu	Nusselt number
PP	pumping power, W
Pr	Prandtl number
ΔP	pressure drop, Pa
q	heat flux, W/m^2
Q	heat transfer rate, W
R	thermal resistance, °C/W·m^2
R	variable
Re	Reynolds number
RWP	rectangular winglet pair
T	temperature, K
t	thickness, m
u	velocity, m/s
V	voltage, v
vol	volume fraction
W	weight, g
x, *y*, *z*	3D Cartesian coordinate

Greek symbols

ρ	density, kg/m^3
μ	dynamic viscosity, kg/m·s

Subscripts

avg	average
bf	base fluid
c	channel
eff	effective
f	fin
f	fluid
in	inlet
m	mean
nf	nanofluid
out	outlet
p	nanoparticle
s	surface
tot	total
w	water

uniformities in heat transfer coefficient and temperature distribution and better performance of pressure drop and friction factor. It was also shown that the higher heat transfer coefficients were for rectangular channels, followed by trapezoidal and then triangular channels. Alfaryjat et al. [5] concluded that hexagonal MCHS was the best channel shape for the heat transfer coefficient and pressure drop compared to circular and rhombus channels. However, rhombus cross-section MCHS was the best channel in terms of temperature, friction factor, and thermal resistance.

Hung et al. [6] and Hung and Yan [7] illustrated that the number of channel, channel upper and lower width ratio, channel lower and lower AR were crucial parameters for heat transfer augmentation. Thus, optimal values of the geometric parameters could be obtained to reach the lowest thermal resistance.

Hung et al. [8] showed that DLMCHS provided higher heat transfer by about 6.3% than SLMCHS with significant decrease in pressure drops. One critical drawback of SLMCHS was the dramatic rise in temperature along the microchannels due to relatively small amount of fluids that take heat generated from the electronic chips, thus the coolant experiences a massive temperature increment. Lin et al. [9] found that channels number, coolant velocity in the lower channel, bottom channel height, and vertical rib affected the values of pumping powers of DLMCHS. Hung et al. [10] emphasized that the optimal design depends on the pumping power of MCHS. Sakanova et al. [11] reported that DLMCHS caused a reduction of 15% in thermal resistance compared to SLMCHS. Moreover, better uniformity for temperature distribution was obtained for DLMCHS leading to enhance the performance of the electronic chips and semiconductor devices. Wei et al. [12] showed that DLMCHS provided larger flow passages than SLMCHS, thus the penalty of pressure drop was dramatically reduced. The thermal resistance was also reduced as the flow rate increased inside the lower-layer microchannels than through the upper layer. Wang et al. [13] emphasized that as the pumping power increased, the heat transfer was remarkably improved.

The surface roughness affects the values of the friction factors even under laminar flow condition as Weilin et al. [14] and Wu and Cheng [15] stated. They displayed the dependency of Nusselt number and friction factor on geometric shape of microchannel.

Comprehensive review, done by Liu and Garimella [16] ascertained that conventional correlations provide consistent prediction for low Reynolds number regime while Lee and Garimella [17] found that the measured Nusselt number agrees with prediction ones over the whole length of the micro-channel. Thus, the correlations of heat transfer coefficient, which can be found by Nusselt number using Eq. (6). Nusselt number derived by Shah and London [18] and Kays and Crawford [19] is

$$Nu = \left[\left(2.22\left(\frac{l}{Re\, D_h\, \mathrm{Pr}}\right)^{-0.33}\right)^3 + (-0.02 + 8.31\, G)^3\right]^{1/3} \qquad (1)$$

Sakanova et al. [20] tested the application of nanofluids in wavy channel structure having different wavy amplitudes and wavelengths. They found that the wavy microchannels outperformed the straight microchannels thermally using water as coolant. Moreover, this thermal improvement was enhanced when water was replaced by nanofluid. Hung et al. [21] reported that by using base fluids (having lower dynamic viscosity such as water) and substrate materials (having high thermal conductivity) enhanced the thermal performance of the MCHS. Nanofluids enhanced the thermal performance of MCHS better than water particularly when the volume fraction increased. Ho et al. [22] and Ho and Chen [23] recorded a slight increase in the friction factor of alumina oxide nanofluids flow in copper MCHS compared to water. Besides, a significant enhancement in heat transfer rate (lower thermal resistance and wall temperature) was registered with nanofluids. Mohammed et al. [24] recorded an increase in both the heat transfer coefficient and wall shear stress when the volume fraction increased whereas a slight increase in the pressure drop across the MCHS was monitored compared to the base fluid.

Yang et al. [25] found that the thermal resistance of trapezoidal MCHS was smaller compared when water was used, which decreased as the particle concentration and *Re* number increased. While a small increase in the pressure drop was recorded with increasing the concentration. Chen and Ding [26] reported that the effect of fluid inertia caused a reduction in the total thermal resistance and the temperature difference between the channel wall and nanofluid phase. Halelfadl et al. [27] observed that nanofluids reduced the total thermal resistance and enhanced significantly the thermal performances of a rectangular MCHS

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