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Numerical Simulation of Nanofluids for Cooling Efficiency in Microchannel Heat Sink

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Abstract – *Numerical simulation of a 3-D microchannel heat sink with rectangular cross section was conducted to investigate the effect of various types of coolants such as water and different nanofluids on the cooling performance of the microchannel heat sink (MCHS). The rectangular MCHS adopted has a diameter of 86 μm and length of 10 mm under the boundary conditions of constant heat flux and laminar flow with uniform inlet velocity. The present study illustrates that the use of Diamond-H_2O as the coolant leads to higher efficiency of heat transfer by 0.3% compared to other nanofluid agents and base fluid. It is also shown that the trend of Nusselt number obeys the increment of Reynolds number.*

Keywords: Microchannel Heat Sink, Nanofluids, Numerical Simulation, Heat Transfer Enhancement

1.0 INTRODUCTION

Increasing process speeds, decreasing product sizes and styling requirements cause higher heat loads associated with the products. Bar Cohen in 1999 acknowledged that due to these rates, the chip level heat fluxes have gone up tremendously [1]. High heat fluxes of the order of 102-103 W/cm^2 are found by Mudawar in 2001 in opto-electronic equipment, high-performance supercomputers, power devices, electric vehicles and advanced military avionics. An increase in the heating density of these components has been a serious problem affecting the performances and reliability of the electronic devices [2].

The advanced cooling technology using microchannels were first proposed by Tuckerman and Pease in 1981 [3], featuring a higher heat transfer performance, smaller geometric size and volume per heat load, lower coolant requirement, and lower operational cost than conventional heat sinks. The concept of microchannel heat sink is considered a potential cooling device for electronic devices since high-density electronics packaging requires new advancement in thermal management.

Failure of conventional methods of cooling such as forced convection air cooling to dissipate away the excessive volumetric heats from the very small surfaces of electronic chips and circuits has triggered an increasing interest in high-performance liquid cooling systems.

Microchannel heat sink (MCHS) consists of multiple parallel microchannels containing a coolant fluid. The heat generated by a chip is carried away from the channel walls by the coolant. In most cases, base fluid such as water, ethylene glycol (EG), and engine oil (EO) are commonly used as coolants in MCHSs. However, base fluids are well known for their low

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thermal conductivity, which limits the heat transfer performance of these coolants. Hence, to improve heat transfer performance, it is necessary to increase the thermal conductivity of the coolants.

Higher thermal conductivity can be achieved by adding an appropriate amount of solid nanoparticles to the base fluid, which results in nanofluid. Nanofluids were first proposed by Choi at the Argonne National Laboratory, USA, who found that solid nanoparticles raise the thermal conductivity of the coolant [4]. Nanofluids can be considered as the next generation of heat transfer fluids.

Kawano *et al*. [5] carried out experiments and three-dimensional numerical simulations of heat transfer behaviour and pressure loss in order to investigate the performance of the microchannel heat exchanger, in which water was used as a coolant through a rectangular microchannel with two different rectangular cross sections of 57 × 180 μm and 57 × 370 μm. The result shows that the silicon chip microchannel model gave very small thermal loss and the measured pressure loss showed good agreement with the analytical result obtained on the basis of the fully developed laminar pipe flow assumption.

Qu and Mudawar [6] analyzed numerically three-dimensional fluid flow and heat transfer in a rectangular MCHS using water as the cooling fluid. They found that the heat flux and Nusselt number are greater near the channel inlet and vary around the channel edge, approaching zero in the corners. Reynolds number affects the length of the developing region. For high Reynolds number of 1400, a fully developed flow may not be significant inside the heat sink. The result also showed that increasing the thermal conductivity of the solid substrate reduces the temperature at the heated base surface of the heat sink, especially near the channel outlet.

Wong [7] conducted a numerical simulation to investigate conjugate fluid flow in a microchannel for microelectronic cooling. A rectangular microchannel with hydraulic diameter of 86 mm together with water as a coolant was used in the simulation. The results indicated a large temperature gradient in the solid region near the heat source. The result also proved that silicon is a better MCHS material compared to copper and aluminium based on higher average heat transfer. A higher aspect ratio in a rectangular microchannel gave higher cooling capability due to the high velocity gradient around the channel when channel width decreases.

Patel and Modi [8] numerically investigated the optimization of heat sink for electronics cooling by using different pressures. They found that for constant pressure drop, as the heat flux increases, the average heat transfer coefficient and average Nusselt number also increases. The heat sink was compact and capable of dissipating a significant thermal load (heat fluxes of the order of 90 W/cm^2) with a relatively small increase in the package temperature.

Wang *et al*. [9] experimentally measured the effective thermal conductivity of mixtures of fluids and nanoparticles by using a steady-state parallel-plate method. Two types of nanoparticles, Al_2O_3 and CuO, dispersed in water, vacuum pump fluid, engine oil, and ethylene glycol were tested in the experiment. The results showed that the thermal conductivities of all nanofluids were higher than those of their base fluids. Also, comparison with various data indicated that the thermal conductivity of nanofluids increases with decreasing particles size.

Koo and Kleinstreuer [10] numerically investigated laminar nanofluid flow in microheat sinks. CuO-H2O and CuO-EG were used to investigate the conjugate heat transfer problem in MCHS. They suggested that a base fluid of high Prandtl number such as ethylene glycol and oil and

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using nanoparticles with high thermal conductivity are more advantageous, and a channel with high aspect ratio is desirable.

Chein and Huang [11] analyzed the performance of silicon microchannel heat sink using CuO-H2O nanofluid as the coolant with various particles' volume fraction. They found that the increase in thermal conductivity of coolant and nanoparticle thermal dispersion effect can greatly enhance the MCHS performance. Moreover, no extra pressure drop is produced since the nanoparticle is small and the particle volume fraction is negligible.

The present study deals with three-dimensional numerical simulation of laminar nanofluids and heat transfer characteristics through a rectangular MCHS using Al_2O_3, CuO, diamond and SiO_2 with nanoparticle volume fraction of 2%. The Reynolds number is in the range of 140 to 1400, and the heat flux is fixed at 100 W/cm^2. Results of interests such as velocity profile, temperature distribution, Nusselt number and effect of Reynolds number on Nusselt number are reported.

2.0 NUMERICAL ANALYSIS

2.1 MCHS Model

The physical configuration of MCHS is shown in Fig. 1. Taking advantage of symmetry, a unit cell consisting of only one channel and the surrounding solid is chosen as shown by the dashed lines to save the computational time. The result obtained can be extended to the entire MCHS. The heat sink is of silicon and various nanofluids are used as the cooling fluid. The heat is supplied at the heat sink top wall idealized as constant heat flux boundary condition.

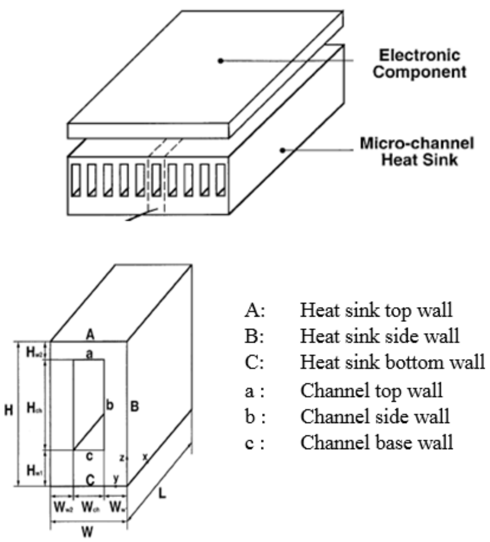

Figure 1: Schematic of rectangular microchannel heat sink and unit cell

The geometry consists of a rectangular channel 57 µm (W) × 180 µm (H) in cross-section, and 10 mm (L) in length. Heat transport in the unit cell is a conjugate problem that combines heat conduction in the solid and convective heat transfer to the cooling fluid. The dimension of MCHS is listed in Table 1.

2.2 Governing Equations

The major assumptions are:

1. Both fluid flow and heat transfer are at a steady-state.
2. Fluid is in a single phase, incompressible and laminar.
3. Properties of both fluid and heat sink material are temperature-independent.
4. All the surfaces of the heat sink exposed to the surroundings are assumed to be insulated except the top plate of heat sink where constant heat flux boundary condition simulating the heat generation from an electronic chip.
5. Uniform wall heat flux.
6. Uniform inlet velocity.
7. Negligible radiation heat transfer.

The continuity, momentum and energy equations for the current problem can be written as follows:

Continuity

$$\frac{dy}{dx}+\frac{dy}{dx}+\frac{dy}{dx}=0 \tag{1}$$

X-Momentum

$$\rho\left(u\frac{du}{dx}+v\frac{du}{dy}+w\frac{du}{dz}\right)=-\frac{dp}{dx}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \tag{2}$$

Y-Momentum

$$\rho\left(u\frac{dv}{dx}+v\frac{dv}{dy}+w\frac{dv}{dz}\right)=-\frac{dp}{dy}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right) \tag{3}$$

Z-Momentum

$$\rho\left(u\frac{dw}{dx}+v\frac{dw}{dy}+w\frac{dw}{dz}\right)=-\frac{dp}{dz}+\mu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \tag{4}$$

Energy

$$\rho C_P\left(u\frac{dT}{dx}+v\frac{dT}{dy}+w\frac{dT}{dz}\right)=k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \tag{5}$$

2.3 Boundary Condition

For hydraulic boundary condition, the velocity is zero at all boundaries except the channel inlet and outlet. A uniform velocity is applied at the channel inlet. The velocity is obtained from the Reynolds number.

$$u = \frac{\mathrm{Re}.\mu_f}{\rho.d_h} \tag{6}$$

Where d_h = hydraulic diameter. The flow is fully developed at the channel outlet.

$$\frac{du}{dx} = 0\,,\ \frac{dv}{dx} = 0\,,\ \frac{dw}{dx} = 0 \tag{7}$$

For thermal boundary condition, constant heat flux is assumed at the heat sink top wall.

$$q'' = -k_s \frac{dT}{dz} \tag{8}$$

Adiabatic boundary conditions are applied to all the boundaries of the solid region except the heat sink top wall. At the channel inlet, the liquid temperature is equal to a given constant inlet temperature.

$$T = T_{in} = 293K \tag{9}$$

The flow is also assumed thermally fully developed at the channel outlet because the change of temperature gradient along the flow direction at the channel exit is usually very small even for very large Reynolds number. Thus, large numerical error will not be introduced by the exit thermal boundary condition.

$$\frac{\partial^2 T}{\partial x^2} = 0 \tag{10}$$

Table 1: Dimensions of unit cell of microchannel heat sink

H [μm]	W [μm]	H_{w2} [μm]	H_{w1} [μm]	H_{ch} [μm]	W_{ch} [μm]	$W_{w1/w2}$ [μm]	D_h [μm]
900	100	450	270	180	57	21.5	86

The modelling of the computational domain is done by GAMBIT in 3-D design of rectangular microchannel. Geometry drawing of microchannel heat sink is according to the dimensions given in Table 1. Finite volume method and the SIMPLE algorithm are applied and FLUENT

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solves the governing integral equations for conservation of mass and momentum. Also, for this case, energy equation is included. Double-precision solver is selected. Fluid properties are updated for every working fluid based on the current solution. The velocity inlet depends on Reynolds number. Convergence of equation is set at 1e-6 for continuity and momentum equations and for energy equation, the convergence is set at 1e-8.

Table 2: Properties of working fluid

	Density, ρ [kg/m^3]	Specific Heat Capacity, Cp [W/mK]	Thermal Conductivity, K [J/kgK]	Dynamic Viscosity, μ [kg/ms]
Silicon	2330	712	148	
		Base Fluid		
water	997.7	4178.9	0.6	0.000949
		Nanofluid		
CuO-H_2O	1108.236	3754.26	0.64772	0.00105315
Diamond-H_2O	1048.436	3935.28	0.65046	0.00105315
Al_2O_3-H_2O	1057.636	3925.48	0.64882	0.00105315
SiO-H_2O	1022.236	4032.25	0.62194	0.00105315

3.0 RESULTS AND DISCUSSION

3.1 Model Validation

The numerical model was verified in two ways to ensure the validity of the analysis. The computational domain was first tested for grid-independence test using several different mesh sizes for better result accuracy.

The numerical model was then validated by comparing the results with available analytical solutions by Shah and London. Based on the numerical results, the average peripheral Nusselt number, Nu was calculated and plotted in Fig. 2 along the MCHS. Nu is defined by:

$$\overline{Nu} = \frac{q'' D_h}{k\left(T_{T,m} - T_{in}\right)} \tag{11}$$

where

T-T_m=average temperature at the boundary.

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T-T_{in}=fluid bulk temperature.

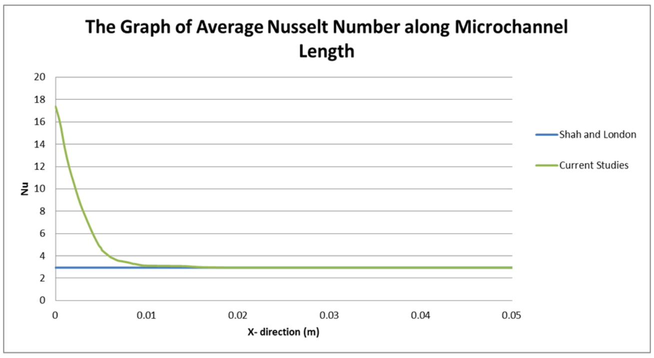

Figure 2: Graph of average Nusselt number along MCHS

3.2 Temperature Distribution

As a result of constant heat flux applied on the top of the silicon heat sink, other surfaces that are treated as a wall, including the working fluid that entered the channel were heated with a constant heat flux of 9 x 10^5 W/m^2.

From Fig. 3, the highest temperature is encountered at the heat sink top wall, which is around 315 K, where constant heat flux had been applied on that surface. Diamond-H_2O had been used in comparison with pure water and it can be seen that when Diamond-H_2O was used, the temperature of the surfaces was lower compared with when pure water as the working fluid was used. When Diamond-H_2O was used as the working fluid, it shows 0.03% of temperature reduction compared to pure water.

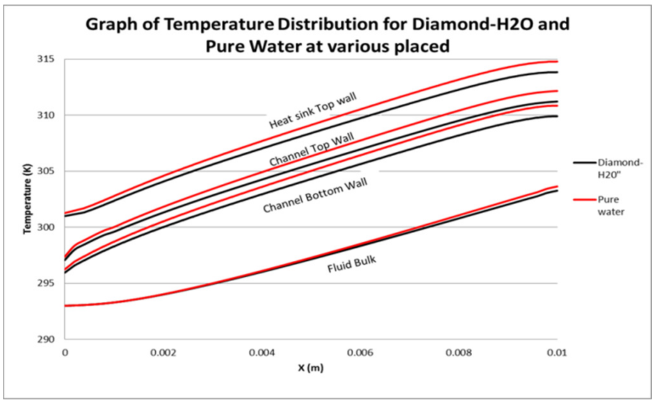

Figure 3: Graph of temperature distribution for Re-140

3.3 Nusselt Number

In this study, heat transfer performance in a microchannel heat sink was analyzed based on Nusselt number. From Fig. 4, the variation of Nusselt number values is observed along the length of microchannel in X direction. Higher Nusselt number was found where fluid enters

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the channel inlet. This is due to the development of thermal entry region at the channel and the values of Nusselt number tend to stabilize after a fully developed region has been achieved.

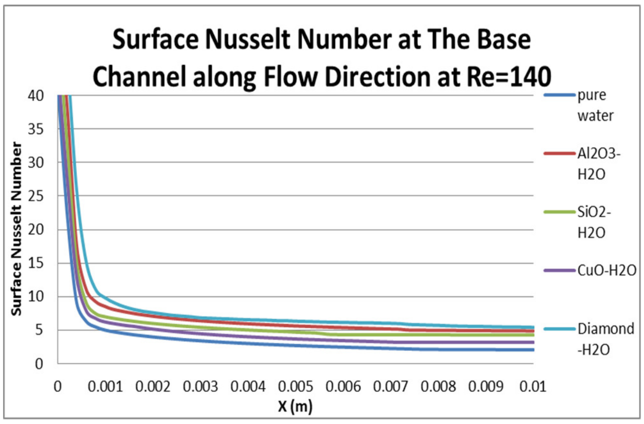
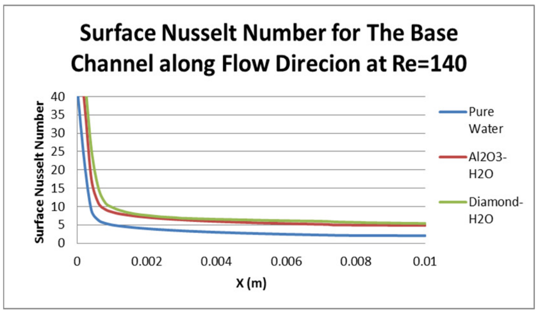


Figure 4: Graph of Nusselt number at Re= 140

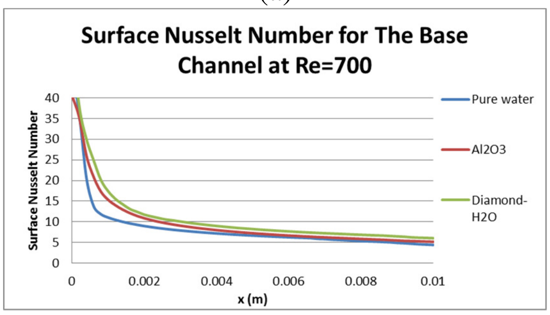
For Reynolds 140, Diamond-H_2O has the highest Nusselt number followed by Al_2O_3-H_2O, CuO-H_2O, SiO_2-H_2O and pure water have a low Nusselt number, which indicates low heat transfer performances. In this case, Diamond- H_2O has the highest thermal conductivity among other nanofluids as shown in Table 2. Higher Nusselt number indicates better heat transfer removal.

3.4 Effect of Reynolds Number

The effects of Reynolds number on the heat transfer process in the microchannel heat sink are illustrated in Fig. 5a- c for three different Reynolds numbers of 140, 700 and 1400, where the inlet temperature and the heat flux applied are still the same.



(a)



(b)

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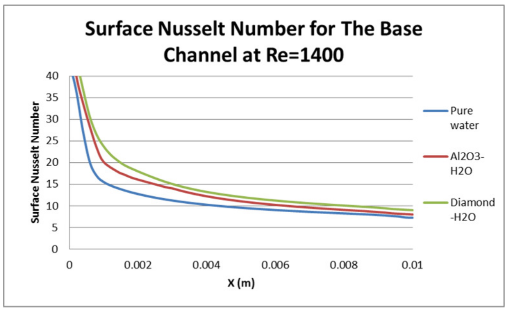

(c)

Figure 5: Graph of surface Nusselt number at various Reynolds numbers; (a) Re=140 (b) Re=700 and (c) Re=1400

It can be seen that as Reynolds number increases, the value of Nusselt number also increases. At the channel outlet, the same trend is found, indicating that the length of the thermal developed region is larger than the channel length. For relatively high Reynolds number, which in this case is 1400, a fully developed flow may not be achieved inside the heat sink even if there is a very small gradient of the average Nusselt number near the channel outlet.

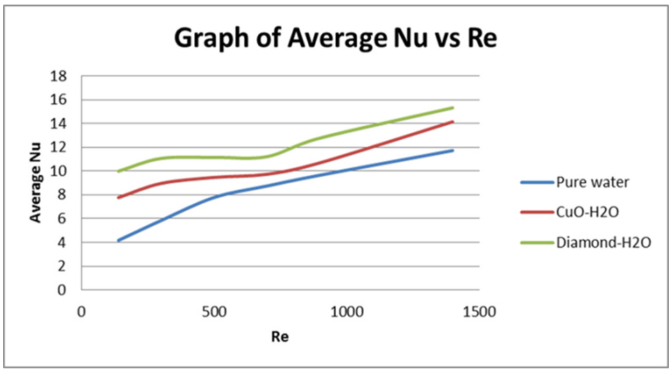

Figure 6: Graph of average Nusselt number against Reynolds number

Figure 6 shows that average Nusselt number increases as Reynolds number increases. This is because the Reynolds number is the function of velocity. By increasing the Reynolds number, the velocity will increase and the movement of the molecular of fluid will also increase. The interruption of the particle of fluid will increase, thus will increase the heat being transferred.

4.0 CONCLUSION

A three-dimensional rectangular silicon microchannel heat sink was analyzed numerically under conjugate heat transfer condition to investigate the fluid flow and heat transfer performance via various types of working fluids, i.e. water and different types of nanofluids. The microchannel heat sink performance was evaluated in terms of temperature profile, Nusselt number and effect of Reynolds number on Nusselt.

As uniform inlet velocity was applied at the channel inlet, it took a while for the flow to fully develop inside the microchannel because of the developing boundary layer at the channel inlet. The highest temperature was encountered at the heat sink top wall where the constant heat flux was applied. Diamond-H_2O provided 0.3% temperature reduction compared to other

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nanofluids. Increased thermal conductivity reduced the temperature at the heated surface of heat sink, especially near the channel outlet.

Nusselt number was used to determine the heat transfer performance. The results of the present work show that the highest heat transfer enhancement is obtained by Diamond-H_2O, which has a higher thermal conductivity and Nusselt number. Indeed, the calculated results showed that the heat transfer performance of Diamond-H_2O is better than that of pure water. Diamond-H_2O is recommended to achieve overall heat transfer enhancement in this case. For low Reynolds numbers, the entrance region was relatively short. However, at high Reynolds numbers, the effects of developing region became more significant. Fully developed flow may not be achieved inside the microchannel heat sink for high Reynolds number.

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