Numerical Simulation of Nanofluids for Improved Cooling Efficiency in Microchannel Heat Sink

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Abstract. Numerical simulation on 3-dimensional rectangular cross section of microchannel heat sink is conducted to investigate the effect of various type coolant consist of water and different type of nanofluids on the cooling performance of microchannel heat sink. FLUENT, a Computational Fluid Dynamic (CFD) is used as the solver of simulation. A rectangular microchannel with hydraulic diameter of 86um and length of 10mm under the boundary condition of constant heat flux and laminar flow with uniform inlet velocity with five sets of working fluid with different nanofluids. The defined model is validated with previous studies of numerical analysis. Results of present work show that using Diamond-H_2O as cooling lead to higher efficiency of heat transfer in microchannel heat sink in comparison to others nanofluid and base fluid. Numerical results show that increasing the thermal conductivity of working fluid can enhanced heat transfer. Nusselt number follows the incremental in Reynolds number.

Introduction

The severe need by user for greater IC speeds, functionality and minimization has fuelled an extraordinary acceleration in chip power dissipation. Increasing process speeds, decreasing product sizes and styling requirements cause higher and higher heat loads on the products. Bar Cohen in 1999 acknowledge that due to these enhancements, 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 super computers, 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 advance cooling technology using microchannels were first proposed by Tuckerman and Pease in 1981 [3], features 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 VLSI as well as electronic devices due to high-density electronics packaging requires new advancement in thermal management.

Conventional methods of cooling such as forced convection air-cooling fails to dissipate away the excessive volumetric heats from the very small surfaces of electronic chips and circuits have caused an increase of interest in high-performance liquid cooling systems.

MCHS is consists with many parallel microchannels containing the flow coolant (Fig. 1). 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 as heat transfer fluids which have low 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.

High thermal conductivity can be achieved by adding an appropriate amount of solid nanoparticles having high thermal conductivity to a base fluid for use as a coolant (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 to be next generation of heat transfer fluid since it improve the heat transfer performance. The potential benefit of using nanofluids as high performance coolants have been demonstrated by engineers and researchers.

Kawano et al. [5] carried out experiments as well as three dimensional numerical simulations on heat transfer behaviour and pressure loss in order to investigate the performance of the micro channel heat exchanger where water have been used as a coolant through rectangular microchannel with two different rectangular cross section which are (57 × 180) µm and (57 × 370) µm. The result show that silicon chip microchannel model gave very small thermal and measured pressure loss showed good agreement with analytical result obtained on the basis of fully develop laminar pipe flow assumption.

Qu and Mudawar [6] analysed 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 have much higher values near the channel inlet and vary around the channel edge, approaching zero in the corners. Flow Reynolds number affects the length of the flow developing region. For high Reynolds number of 1400, fully developed flow may not be significant inside the heat sink. 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] did a numerical simulation to investigate the conjugate fluid flow and heat transfer phenomena in a microchannel for microelectronic cooling. A rectangular microchannel with hydraulic diameter of 86 mm together with water as a coolant is used in the simulation. The investigation was conducted with consideration of temperature dependent viscosity and developing flow, both hydro dynamically and thermally. The results indicate a large temperature gradient in the solid region near the heat source. The result also prove that silicon is a better MCHS material compared to copper and aluminum, based on higher average heat transfer. A higher aspect ratio in a rectangular microchannel gives higher cooling capability due to high velocity gradient around the channel when channel width decreases.

Patel and Modi [8] numerically investigate optimization of heat sink for electronics cooling by using different type of pressure different. They found that, as the heat flux increase, the average heat transfer coefficient and average Nusselt number also increases for constant pressure drop. The heat sink is compact and is capable of dissipating a significant thermal load (heat fluxes of the order of 90 W/cm2) with a relatively small increase in the package temperature.

Wang et al. [9] experimentally measured the effective thermal conductivity of mixtures of fluids and nanometer-size particles 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. Experimental results show 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 investigate laminar nanofluid flow in microheat-sinks. CuO-H_2O and CuO-EG had been use to investigate the conjugated heat transfer problem for MCHS. They suggested that a base fluid of high-Prandtl number such as ethylene glycol and oil should be used compare to water, and using nanoparticles with high thermal conductivity are more advantageous, and a channel with high aspect ratio is desirable.

Chein and Huang [11] analysed the performance of silicon microchannel heat sink using CuO-H_2O nanofluid as a coolant with various particle volume fraction. They found that increase in thermal conductivity of coolant and nanoparticle thermal dispersion effect can greatly enhance the MCHS performance. Moreover, no extra pressure drop produced since the nanoparticle is small and particle volume fraction is low was an advantage of using nanofluid as coolant in MCHS.

It should be noted from the above literature review, to the best knowledge of the authors' that there is no work investigates the effect of various types of nanofluids on the MCHS performance has been reported in the past. The case of rectangular MCHS using different nanofluids seems had not been investigated in the past and this has motivated the present study. Thus, the present study deals with three dimensional numerical simulations of laminar nanofluids flow and heat transfer characteristics through rectangular MCHS using Al_2O_3, CuO, diamond and SiO_2 with nanoparticle volume fraction of 2%. The Reynolds number is the range of 140–1400, 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 in this paper.

Numerical Analysis

MCHS Models

The physical configuration of MCHS is schematically shown in figure1. Taking advantage of symmetry, a unit cell consisting of only one channel and the surrounding solid is chosen as shown by the dashed lines in Fig. 1 to save the computational time. The result obtained can be extended to the entire MCHS.The heat sink is made from silicon and various nanofluid is used as the cooling fluid. The heat is supply at the heat sink top wall idealized as constant heat flux boundary condition.

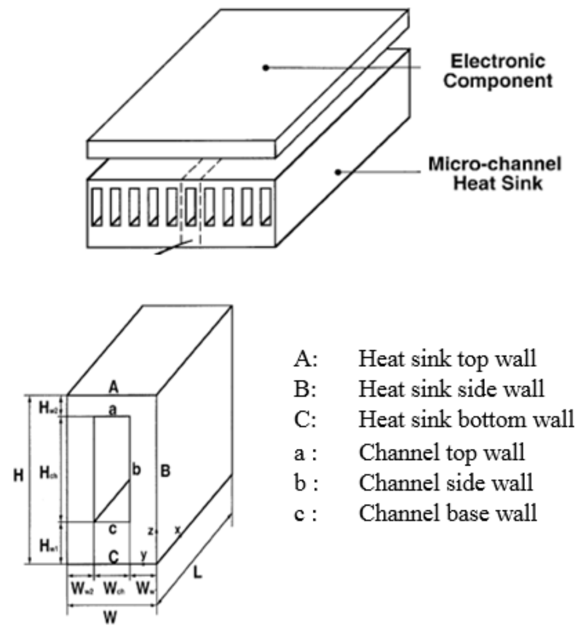


Fig. 1 Schematic of rectangular micro-channel heat sink and unit cell

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

Governing Equation

Some simplified assumptions are required to apply the modelling conditions to the heat transfer process in the MCHS. The major assumptions are:

1) Both fluid flow and heat transfer are in steady-state and three-dimensional
2) Fluid is in single phase, incompressible and the flow is laminar
3) Properties of both fluid and heat sink material are temperature-independent

4) All the surfaces of 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 electronic chip is specified.
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:

Continuity

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{1}$$

X-Momentum

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

Y-Momentum

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

Z-Momentum

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

Energy

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

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 channel outlet.

$$\frac{\partial u}{\partial x} = 0, \frac{\partial v}{\partial x} = 0, \frac{\partial w}{\partial x} = 0 \tag{7}$$

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

$$q'' = -k_s\frac{\partial T}{\partial z} \tag{8}$$

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 numbers. Thus, large numerical error will not be introduced by the exit thermal boundary condition.

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

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

Table 1 Dimensions of unit cell of micro-channel heat sink

Method of Solution

The modelling of computational domain is done by using GAMBIT in 3-dimensional design of rectangular microchannel. Geometry drawing of microchannel heat sink is according to the dimension given in the table??. The finite volume method and the SIMPLE algorithm are applied and FLUENT will solve governing integral equations for conservation of mass and momentum, and for this case include energy equation. Double-precision solver will be selected since it more accurate result. Fluid properties are updated for every working fluid, based on current solution. The velocity inlet is dependents on the Reynolds number. Convergence of equation is set at 1e-6 for continuity and momentum equation and for energy the convergence is set at 1e-8

	Density, $\rho(kg/m^3)$	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

Table 2 Properties of working fluid

Results & Discussion

Model Validation

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

The numerical model is 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 is calculated and plotted in figure 2 along the MCHS. Nu is defined as:

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

where
T-$_{(T,m)}$ = average temperature at the boundary
T-$_{in}$ = fluid bulk temperature

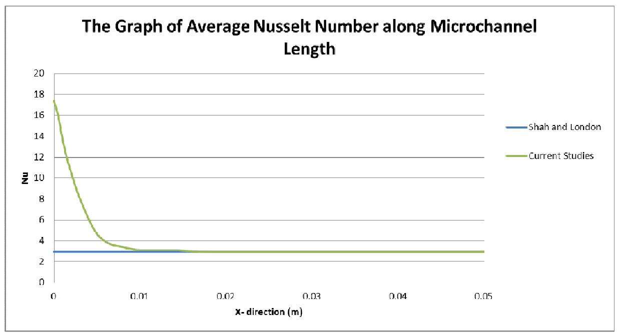


Fig. 2 Graph of average Nusselt Number along MCHS

Temperature Distribution

As a result of constant heat flux applied on the top of the silicon heat sink, other surfaces which are treated as wall including the working fluid entered the channel were heated with constant heat flux of 9 x 105 W/m^2.

From the figure 3, 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 using Diamond- H_2O, the temperature of the surfaces is lower compare when using pure water as working fluids. When using Diamond- H_2O as working fluid, it shows 0.03 % of temperature reduction compare to pure water.

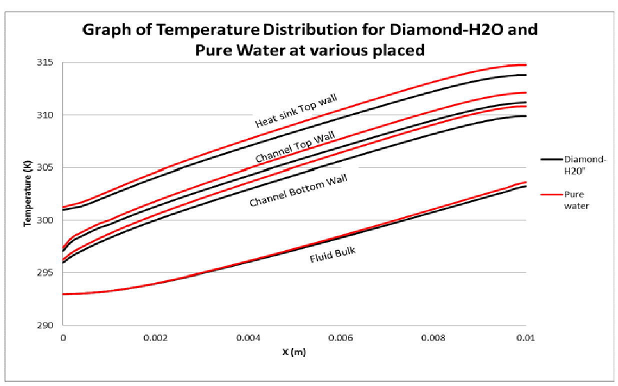


Fig. 3 Graph of temperature distribution for Re-140

Nusselt Number

In this study, heat transfer performance in microchannel heat sink is analysed based on Nusselt number. From the figure 4 variation of Nusselt number values are observed along the length of microchannel in x direction. Higher Nusselt number is found as fluid enters the channel inlet. This is because of the development of thermal entry region at the channel and the values of Nusselt number tend to stabilize after fully develop region has been achieved.

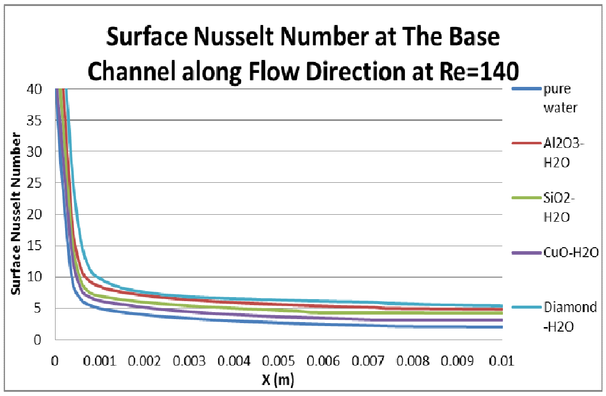
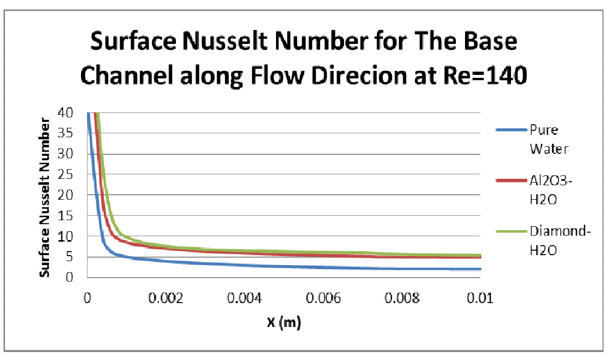


Fig. 4 Graph pf Nusselt number at Re= 140

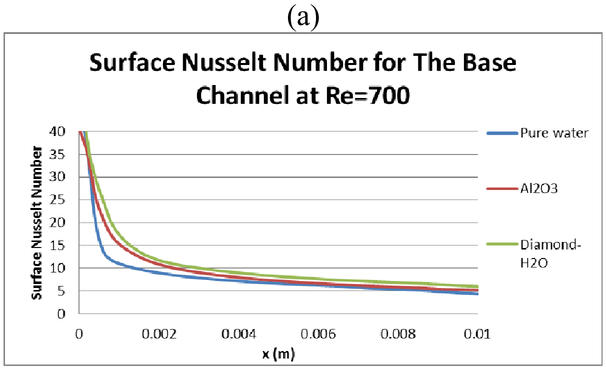
For Reynolds number of 140, Diamond-H_2O has the highest Nusselt number followed by Al_2O_3-H_2O, CuO- H_2O, SiO_2- H_2O and pure water having low Nusselt number which indicates low heat transfer performances. By using nanofluids which have higher thermal conductivity than base fluid (pure water), give greater Nusselt number compared to using pure water as working fluid. In this case, Diamond- H_2O has higher thermal conductivity among other nanofluids as shown in table 2.Higher Nusselt number indicate better heat transfer enhancement.

Effect of Reynolds Number

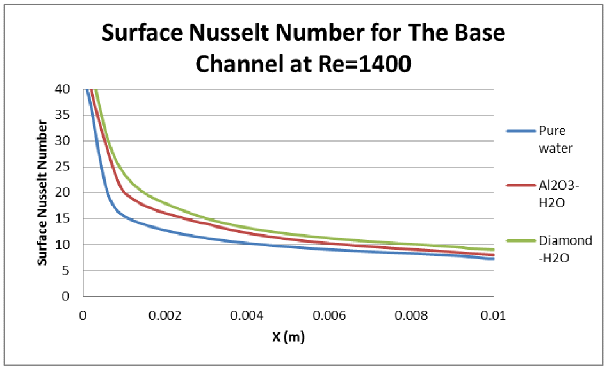
The effects of Reynolds number on the heat transfer process in the micro-channel heat sink are illustrated in Fig. 5a, b and c for three different Reynolds numbers 140, 700 and 1400 where the inlet temperature as well as the heat flux applied is still the same.



(a)



(b)



(c)

Fig. 5 Graph of surface Nusselt number at various Re. (a) Re=140 (b) Re=700 (c) Re=1400

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

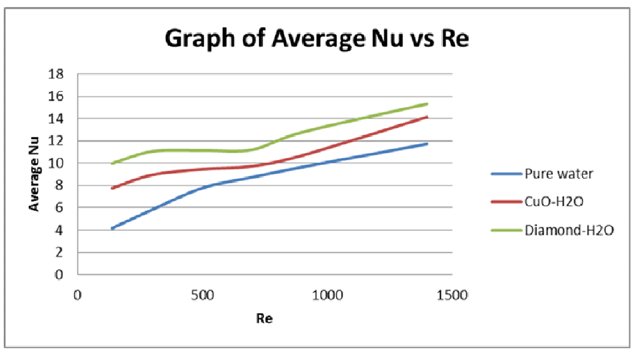


Fig. 6 Graph of Average Nusselt Number Against Reynold Number

Figure 6 shows that average Nusselt number increase as Reynolds number increase. This is because the Reynolds number is the function of the 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 increase the heat being transfer.

Conclusions

The three-dimensional rectangular silicon microchannel heat sink were analysed numerically for combine conduction and convection heat transfer to investigate the fluid flow and heat transfer performance via various type of working fluid used which consist of water and different type of nanofluids. The microchannel heat sink performance is evaluated in term of temperature profile Nusselt number and effect of Reynolds number on Nusselt number.

As uniform inlet velocity is applied at the channel inlet, it takes a while for the flow to be fully developed flow inside the microchannel because of the developing boundary layer at the channel inlet. Highest temperature is encountered at the heat sink top wall where the constant heat flux is applied. Diamond-H_2O provides 0.3 % temperature reduction compare to using other nanofluid. Increasing the thermal conductivity reduces the temperature at the heated surface of heat sink especially near the channel outlet.

Nusselt number has been used to determine the heat transfer performance for microchannel heat sink. The results of present work show that the highest heat transfer enhancement is expected for

Diamond-H_2O which have higher thermal conductivity and have the higher Nusselt number. Indeed, the calculated results showed that the heat transfer performance of Diamond-H_2O was better than that of pure water. Increasing the thermal conductivity of working fluid enhanced the heat transfer performance of microchannel heat sink. Diamond-H_20 is recommended to achieve overall heat transfer enhancement.

For low Reynolds number, the entrance region is relatively short. However at high reynold number, the effects of developing region become more significant. Fully develop flow may not be achieved inside the microchannel heat sink for high Reynolds number.

This study is practical and can be apply in the real life situation and nanofluids can be considered to be the next-generation heat transfer fluid.

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