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Numerical Prediction of Laminar Nanofluid Flow in Rectangular Microchannel Heat Sink

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Abstract – *This paper present the effect of using magnetic nanofluid inthree dimensional (3D) straight rectangular microchannel heat sink with single phase approach which is investigated numerically. In this study, the behaviour and effect of using pure water and magnetic nanofluid (Fe_3O_4-H_2O) as cooling fluids is examined. The Fe_3O_4 magnetic nano particles with an average diameter of 13nm dispersed in water with four volume fractions (0, 0.4, 0.6 and 0.8). The 3-D steady laminar flow and heat transfer governing equations are solved using ANSYS FLUENT. The result of the present work reveal that increasing the volume fractions provide a better heat transfer enhancement and increasing the Reynolds number caused an increase in volume fraction. Highestnusselt number has been observed as the fluid enters the channel inlet. This could be anticipated as a result of the development of thermal entry region at the channel and the value of the Nusselt number tend to stabilize after fully develop region has been achieved.*

Keywords: Nanofluid, Laminar fluid flow, Microchannel, Single phase flow

1.0 INTRODUCTION

Nowadays, cooling is one of the most top technical challenges facing high-tech industries. Since conventional ways/method of cooling such as forced convection air cooling fails to dissipate away the astronomical volumetric heats from the very small surfaces of electronic chips and circuits, new solution need to be present to overcome these matter. The advance cooling technology using microchannels were proposed by Tuckerman [1] for cooling very large scale integrated (VLSI) circuitry. The concept of microchannel heat sink applied in cooling system is important due to high-density electronics packaging requires new advancement in thermal management. The large surface area of MCHS enables the coolant to take away large amounts of energy per unit time per unit area while maintaining a considerably low device temperature. Using these MCHS, heat fluxes can be dissipated at relatively low surface temperatures [2].

Nanofluid was first proposed by Choi in 1995. He defined nanofluids as fluids containing particles of sizes below 100 nm. Many researchers have performed numerical studies of heat transfer with nanofluids. For example, Chein and Huang [3] investigated the heat transfer performance of nanofluid-cooled MCHS and found that they yielded better heat transfer performance than a water-cooled MCHS, and that a higher nanofluid bulk temperature could prevent particle agglomeration. Also a research conducted by Tsai and Chein [4] analyzed MCHS performance using Cu–H_2O and carbon nanotube-H_2O and a porous media approach and found that the nanofluid reduced the temperature difference between the bottom-heated

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wall and the bulk nanofluid. While, jang and Choi [5] numerically investigated a nanofluid-cooled MCHS using water-diamond nanofluidand discovered that the use of water–diamond nanofluid enhanced the performance by about 10% compared with that of an MCHS with pure water. Furthermore, Lee and Mudawar [6]studied experimentally the micro-channel cooling benefits of water-based nanofluids containing small concentrations of Al_2O_3 for single-phase and two-phase heat transfer. Promising result achieved by Bhattacharya et al [7] which presented numerical study of conjugate heat transfer in rectangular microchannel using Al_2O_3-H_2O nanofluid. The result confirmed that the useof Al_2O_3-H_2O nanofluid improved the thermal performance of a microchannel heat sink.. Chein and Huang [3] analyzed silicon microchannel heat sinks performance using nanofluids with a mixture of pure water and nanoscale Cu particles as coolants with various volume fractions. Koo and Kleinstreuer [8] used nanofluid flow in a representative microchannel, and conduction–convection heat transfer for different base fluids such as water and ethylene glycol with CuO-nanoparticles. They showed that using nanoparticles with high thermal conductivity are more advantageous, and a channel with high aspect ratio is desirable. Lee et al.[9] proposed 38.4 nm of Al2O3 and 23.6 nm of CuO particles to enhance the thermal conductivity of water and EG. They showed that the enhancement percentage in thermal conductivity was a function of concentration and conductivities of the particles material and liquid. Lee and Mudawar [6] studied the effectiveness of nanofluids for single-phase and two-phase heat transfer in microchannels.They showed that the higher single-phase heat transfer coefficients are achieved in the entrance region of microchannels with increased nanoparticle concentration.

In a nutshell, many interesting results suggesting the potential of nanofluids to enhance the thermal performance of an MCHS have been reported. However very few literatures study the effect of using magnetic nanofluid in microchannels and this has motivated the present study. Thus, the present study presents the heat transfer enhancement in 3D MCHS using magnetic nanofluid.

2.0 METHODOLOGY

In this study, the micro-heat sink model consists of a 10 mm long substrate and dimension of rectangular microchannels have a width of 57 µm and a depth of 180 µm and we consider a rectangular channel of dimension (900µm x 100µmx10mm) applied constant heat flux of 90W/m^2 from top of the heat sink as Figure (1a) [10].

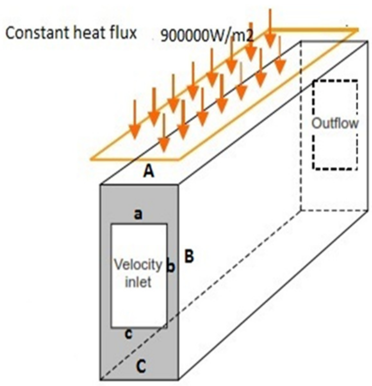

Figure 1(a): Schematic of unit cell rectangular microchannel heat sinks

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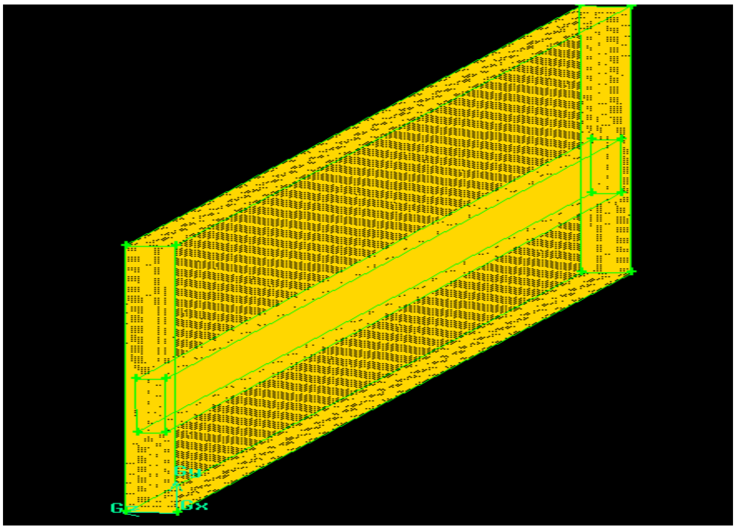

Figure 1(b): Geometry modelling of computational domain

The structure of the heat sink is illustrated in Fig. 1. The modelling of computational domain is done by using GAMBIT in 3D design of rectangular microchannel as depicted in figure (1b). The dimension of the physical model is tabulated in Table 1.

Table 1: Dimensions of the physical model

W_{w1}	W_{ch}	W_{w2}	H_{w1}	H_{ch}	H_{w2}	L
21.5µm	57µm	21.5µm	270µm	180µm	450µm	10mm

2.1 Governing Equation

In order to determine the distribution of pressure, velocity and temperature to predict convective heat transfer rate, the principles of conservation of mass, conservation of momentum and conservation of energy are applied. These conservation principles lead to the so-called continuity, Navier-Stokes and energy equations. The continuity, momentum and energy equations for the problem can be written as [11]:

For continuity equations:

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

For Momentum Equations:

X-Momentum Equation

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

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Y-Momentum Equation

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

Z-Momentum Equation

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

For Energy Equation

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

2.2 Grid Independent Test

The present work consists of a rectangular straight microchannel with constant heat flux at the top wall heat sink. Before the carrying out of complete simulation, the computational domain was tested for better result accuracy and time effectiveness. In this case, three different mesh size models were modeled using GAMBIT software, of all the three meshes only one suitable mesh is selected for simulation model. The model with fines mesh size is taken as reference for other models. Table 4-1 shows the mesh type, number of element and maximum deviation from channel outlet temperature for the three mesh size.Model with medium mesh size was selected for simulation model since the maximum deviation of the channel outlet temperature is approximately 0.015% which is very small and not significant to the result of the numerical analysis.

Table 2: Grid independent test

Mesh Type	Course	Medium	Fine
Number of Element	28000	127500	780000
Maximum Deviation From Channel Outlet Temperature	0.03%	0.016%	References

2.3 Boundary Conditions

To achieve solutions to the flow and temperature fields, boundary conditions are employed and specified. The following are the boundary equations under study [10].

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.

$$u=\frac{R_e * \mu_f}{\rho * d_h} \tag{6}$$

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

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

For thermal boundary conditions, adiabatic boundary conditions are applied to all the boundaries of the solid except the heat sink top wall, where a constant heat flux is assumed.

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

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.

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

2.4 Thermo physical Properties of Nanofluid

The thermo physical properties of nanofluids listed in this study are calculated using the following equations [12] , [13] and [14]

Density:

$$\rho_{nf}=(1-\varphi)\rho_{bf}+\varphi\rho_p \tag{11}$$

Effcetive heat capacity:

$$\left(\rho c_p\right)_{nf}=(1-\varphi)\left(\rho c_p\right)_{bf}+\varphi\left(\rho c_p\right)_p \tag{12}$$

Thermal conductivity:

$$\left(\frac{k_p+2k_b+2(k_p-k_b)\varphi}{k_p+2k_b-2(k_p-k_b)\varphi} k_{bf}\right) \tag{13}$$

Effective viscosity:

$$\mu_{nf}=\mu_{bf}(1+2.5\varphi) \tag{14}$$

where, φ= particle volume fraction

2.5 Data Validation

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To ensure the validity of this simulation analysis of three dimensional rectangular straight microchannel, data validation was done for the accuracy of the result. The data is validated against Shah and London with the Average Nusselt number evaluated and plotted in Figure 2 as a function of longitudinal distance x. The validation correlation of average Nusselt number is given by [15]:

$$\overline{N}_u=\frac{q^{''}d_h}{k_f(T_{\Gamma,m}\text{-}T_m)} \tag{15}$$

$T_{\Gamma,m}$ is the average temperature at the boundary and $T_m$is bulk temperature.

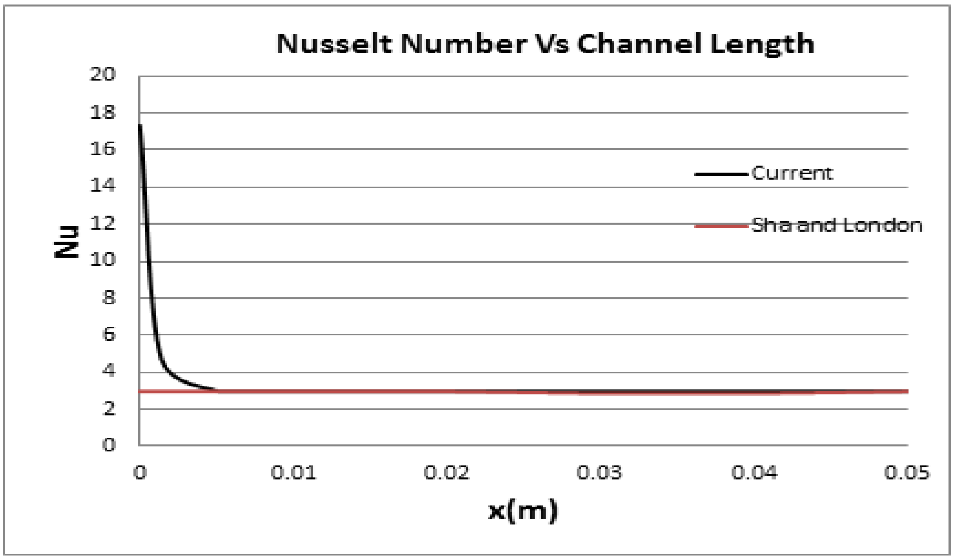

Figure 2: Validation of average nusselt number along micro channel length

The average Nusselt number from Shah and London [15] have constant straight line graph while the current study have an average Nusselt number with high value at the entrance and eventually approaches the fully developed value with good agreement with Shah and London result as the length increase. The discrepancies between Sha and London and the current study was due to the fact that, Shah and London applied fully developed flow before the working fluid enter the inlet whereas current study used uniform inlet velocity which take a while to become fully developed 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.

3.0 RESULTS AND DISCUSSION

In this study, heat transfer performance in microchannel heat sink is analysed based on Nusselt number. Nusselt number is defined as dimensionless parameter that provides a comparison rate

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for how fast heat is transferred between materials where convection is taking place, as compared to basic heat transfer by conduction where little internal movement of matter is occurring.

Figure 3(a) to Figure 3(c), show the graphs of Nusselt number versus length x, taken at the base channel wall of microchannel heat sink for Pure water and Fe_3O_4- H_2O. From the plots 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 could be anticipated as the result 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. The energy equation for the entry length is complicated which is why the simple solution to explain the thermal entry length problem is based on assuming that thermal condition develop in the presence of fully develop profile. It would never be the case that thermal conditions are fully developed while the hydrodynamic condition is still developing. 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.

Figure 3(d) 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 themolecular of fluid will also increase. The interruption of the particle of fluid will increase thus increase the heat being transfer. However, the 0.8% volume fraction provides the maximum increase in Nusselt number compared to 0.6% and 0.4%.

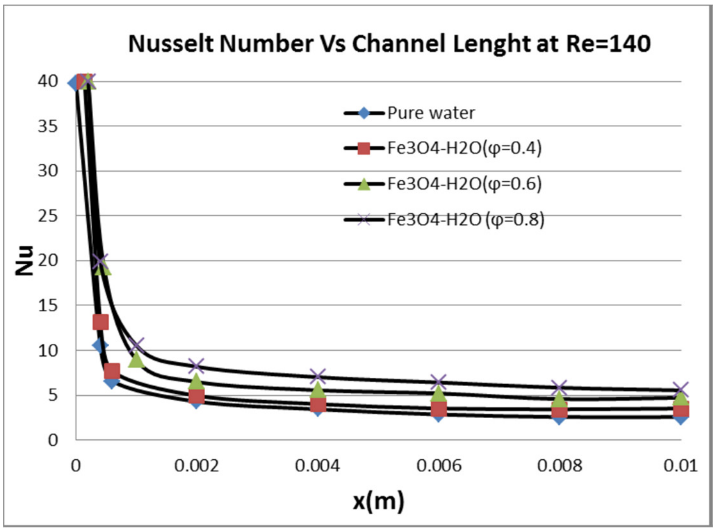

Figure 3(a): Nusselt Number (Nu) Vs Channel Length(x) at Re=140

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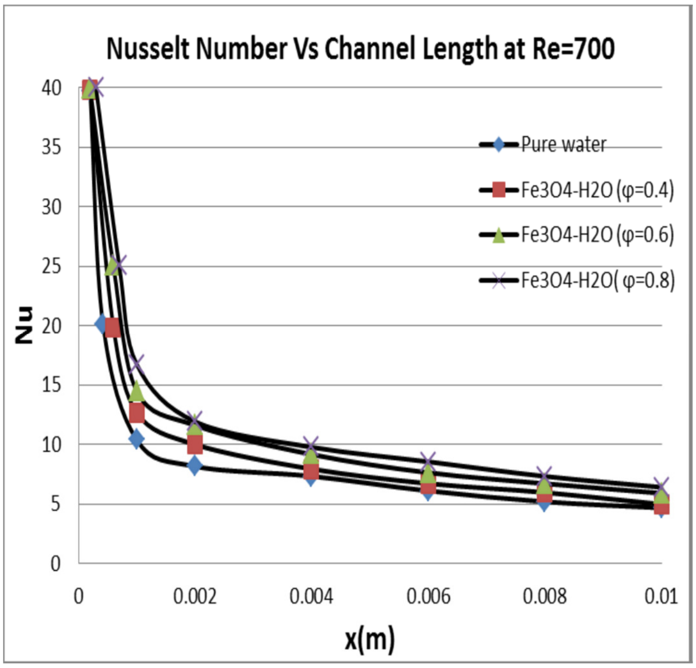

Figure 3(b): Nusselt Number (Nu) Vs Channel Length(x) at Re=700

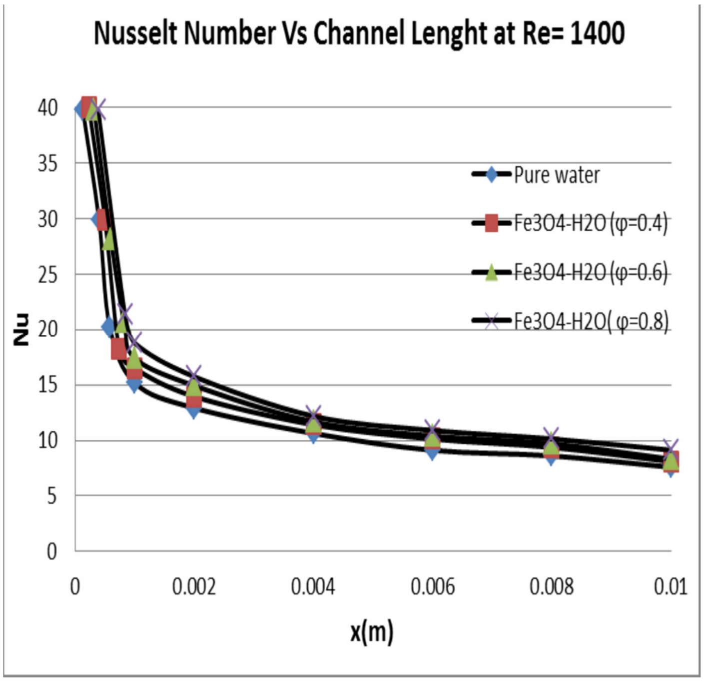

Figure 3(c): Nusselt Number (Nu) Vs Channel Length(x) at Re=1400

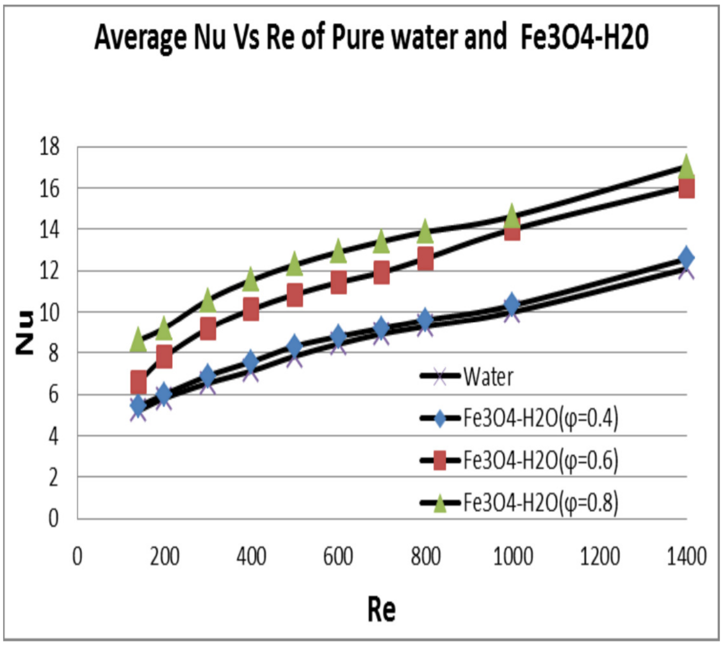

Figure 3(d): Average Nusselt Number (Nu) Vs Reynolds Number(Re)

4.0 CONCLUSSION

Numerical simulation on laminar magnetic nanofluid flow in MCHS is presented in this paper. The effects of using magnetic nanofluids with different volume fractions on the heat transfer enhancement of rectangular MCHS are studied. Based on the presented results, the following conclusions can be drawn:

i) as uniform inlet velocity is applied at the channel inlet, it takes an interval of time for the flow to be fully developed flow inside the microchannel the phenomena behind this is because of the developing boundary layer at the channel inlet.
ii) Fe_3O_4-H_2O with volume fraction 0.8% provides the maximum temperature reduction of 0.4% compare to using pure water as working fluid.
iii) The simulated results showed that the heat transfer performance of Fe_3O_4- H2O with 0.8% was better than that of Fe_3O_4-H2O with 0.4%, of Fe_3O_4-H2O with 0.6%, and pure water. Increasing the thermal conductivity of working fluid enhanced the heat transfer performance of microchannel heat sink. Fe_3O_4-H20 is recommended to achieve overall heat transfer enhancement.

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