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Journal of Advanced Research in Materials Science
ISSN (online): 2289-7992 I Vol. 23, No.1. Pages 15-24, 2016

The use of Fe_3O_4-H_2O_4 Nanofluid for Heat Transfer Enhancement in Rectangular Microchannel Heatsink

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Abstract – *The conjugate heat transfer in Three-dimensional (3D) rectangular microchannel heat sink was simulated using ANSYS FLUENT, constant heat flux of 9000000 W/m^2 was applied at the top of the silicon heat sink. In this study the effect of temperature on microchannel heat sink using pure water and Fe_3O_4-H_2O_4 as working fluids in the microchannel were examined. Fe_3O_4-H_2O_4 with volume fraction range of 0.4% - 0.8% were used in this simulation to evaluate the cooling performance of microchannel heat sink. The Results of present work shows that, fluid nearer to the wall have high temperature compared to the fluid flow at the core of the channel. These Phenomena reveal that the extraction of heat by the working fluid from the solid region is high near the channel wall compared to the middle part of the channel when using Fe_3O_4. Hence, the presence of nano particles has effect of reducing the tempreture of the surfaces as particle volume fraction of Fe_3O_4-H_2O_4 increases due to its higher dynamic viscosity and lower heat capacity compared to pure water.*

Keywords: nanofluid, laminar fluid flow, microchannel, temperature distribution, Fe_3O_4-H_2O_4

1.0 INTRODUCTION

The concept of the microchannel heat sinks was first put into used in 1981 by Tuckerman and Peas [1]. The duo demonstrated that the microchannel heat sinks, consisting of micro rectangular flow passages, have a higher heat transfer coefficient in laminar flow regime than in turbulent flow through macro size channels. Therefore, a significantly high heat flux can be dissipated by using such a microchannel heat sink. There are two main configurations for the application of microchannel cooling which are direct cooling and indirect cooling. Direct cooling requires a direct contact between the surface to be cooled and the coolant fluid. This scheme reduces the thermal resistance between the surface and the coolant and thus, enhances the cooling effectiveness. However, electrical and chemical compatibility between the coolant and device itself needs to be ensured for this system to work [2].

An alternative to the above configuration is the use of a metallic heat sink to conduct the heat away from the device to a coolant which is forced through circular or noncircular grooves in the heat sink. Such an indirect cooling configuration which allows for a greater flexibility in

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coolant selection at the cost of increased thermal resistance between the device and the heat sink due to the heat diffusion resistance in the heat sink itself [3]. Microchannels are very fine channels of the width of a normal human hair and are widely used for electronic cooling purposes. In a MCHS, multiple microchannel are stacked together as shown in which can increase the total contact surface area for heat transfer enhancement and reduce the total pressure drop by dividing the flow among many channels. Liquid or gas is used as a coolant to flow through these microchannels. 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 [4]. Cooling becomes one of the top technical challenges facing electronic and other high-tech industries. Since conventional ways of cooling such as forced convection air cooling fails to dissipate away the extremely large volumetric heats from the very small surfaces of electronic chips and circuits, new method need to be present to overcome these matter. The small physical size of electronic equipment and limitations of air cooling systems have caused an increase of interest in high-performance liquid cooling systems. A liquid coolant is pumped through the microchannels of the heat sink so as to extract the heat from the source such as electronic chip on which it is mounted. In most cases, water is used as a coolant. But water is well known as heat transfer fluids which have low thermal conductivity that greatly limits the heat exchange efficiency. Other base fluids like engine oil and ethylene glycol also have low thermal conductivity. One of the purpose of this study is to reduce the high temperature occurrence on the microchannel heat sink in laminar flow. Here we study and analyzed the effect of temperature on the microchannel heat sink using magnetic nanofluid (Fe_3O_4-H_2O_4).

2.0 METHODOLOGY

The method employed in this study is that, the micro-heat sink computational model is made up of a 10 mm long substrate and size of rectangular microchannels have a width of 57µm and a depth of 180µm and we consider a rectangular channel of dimension (900µm x 100µmx10mm) applied constant heat flux of 900000W/m2 from top of the heat sink, the modelling of computational domain is done by using GAMBIT software as depicted in Fig. 1 [5].

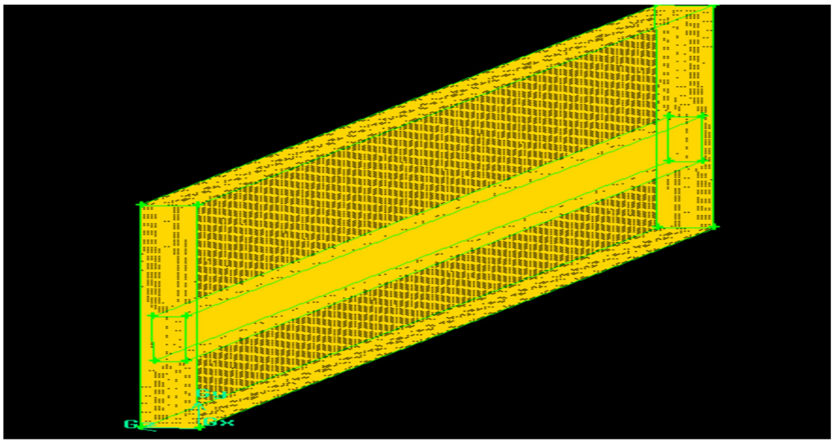

Figure 1: Geometry modelling of the computational model

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2.1 Governing Equation

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 [6]

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

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

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 Boundary conditions

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 dh = 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 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.

$$q^{"}=-k_s\frac{\partial T}{\partial z} \tag{8}$$

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At the channel inlet, the liquid temperature is equal to a given constant inlet temperature.

$$T = T_{in} = 293k \quad (9)$$

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.

$$\frac{\partial^2 T}{\partial x^2} \quad (10)$$

2.3 Thermo physical Properties of Nanofluid

Nano fluid as heat transfer enhancement fluid exhibited a unique property of enhanced thermal conductivity and the convective heat transfer coefficient compared to the base fluid. Presented below are the thermos-physical properties of nanofluids culled from literature [7].

Density:

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

Effcetive heat capacity:

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

Thermal conductivity:

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

Effective viscosity:

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

where φ is the particle volume fraction

2.4 Data Validation

The data was validated against Shah and London with the Average Nusselt number evaluated and plotted in Fig. 2 as a function of longitudinal distance x. The validation correlation of average Nusselt number is given by[8]

$$\bar{N}_u = \frac{q'' d_h}{k_f(T_{\Gamma,m} - T_m)} \quad (15)$$

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

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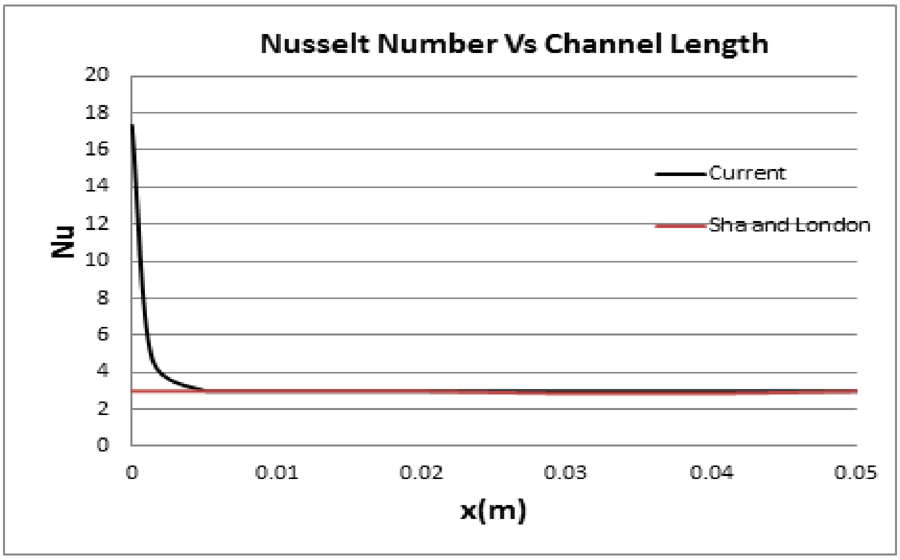

Figure 2: Validation of average Nusselt number along micro channel length

The average Nusselt number from Shah and London [8] 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

The temperature contours from the simulation results of pure water and Fe_3O_4-H_2O as working fluids were taking as plotted in Fig. 3a-3c. Fluid nearer to the wall have high temperature compared to the fluid flow at the core of the channel. These Phenomena shows that the extraction of heat by the working fluid from the solid region is high near the channel wall compare to the middle part of the channel.

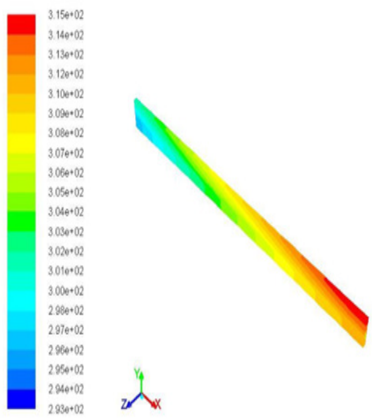

Figure 3a: Temperature contour at the microchannel with water as working fluid

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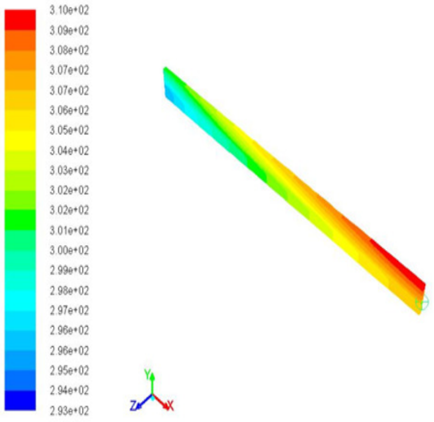

Figure 3b: Temperature contour at the microchannel with Fe_3O_4-H_2O as working fluid

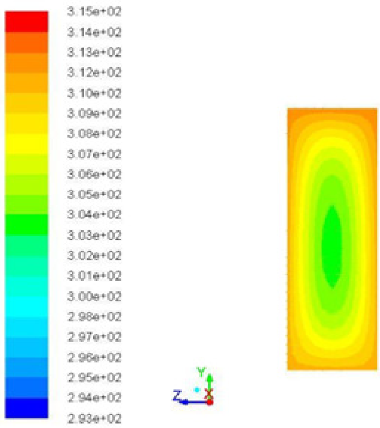

Figure 3c: Temperature contour at the channel oulet

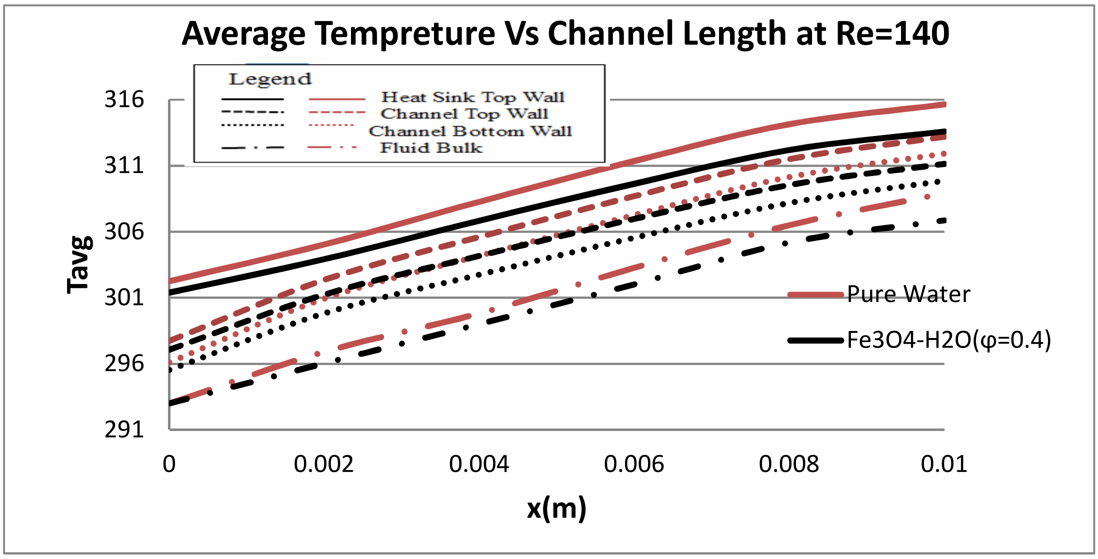

Figure 5a: Average temperature distribution versus channel length for Re=140, ϕ=0.4

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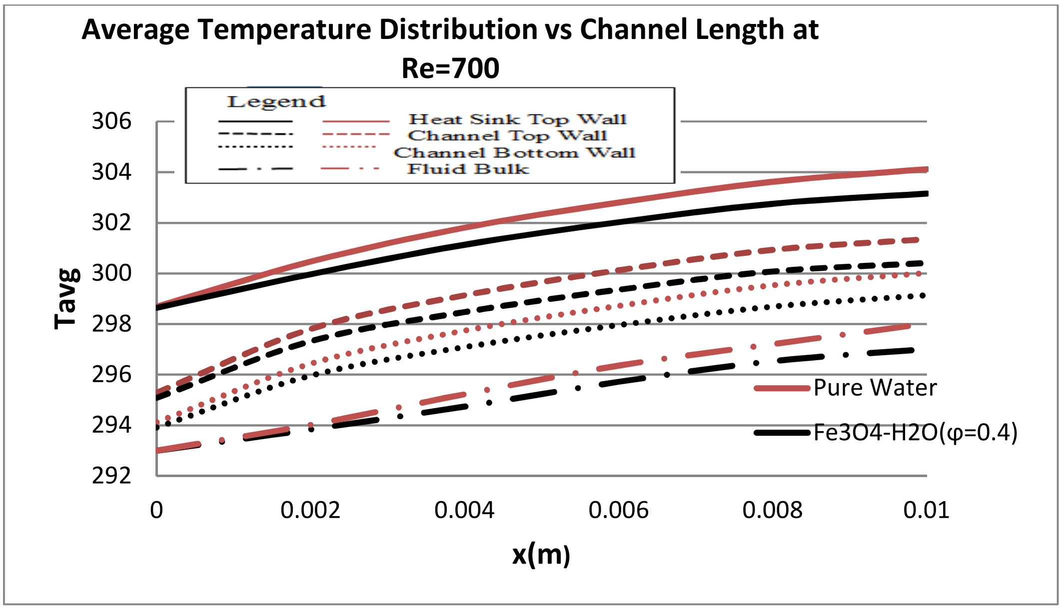

Figure 5b: Average temperature distribution versus channel length for Re=700, ϕ=0.4

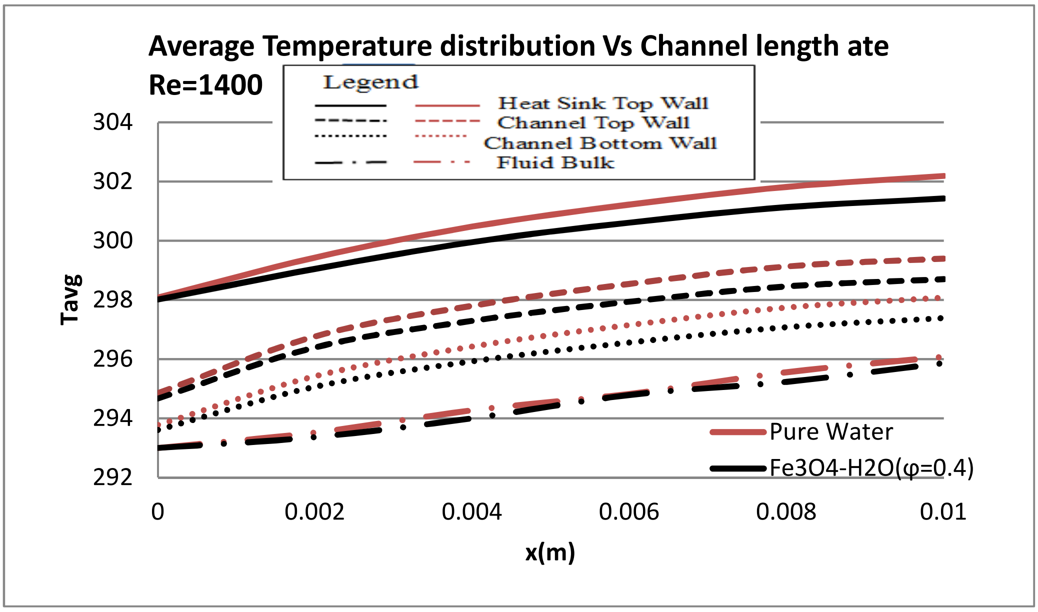

Figure 5c: Average tempreture distribution versus channel length for Re=1400, ϕ=0.4

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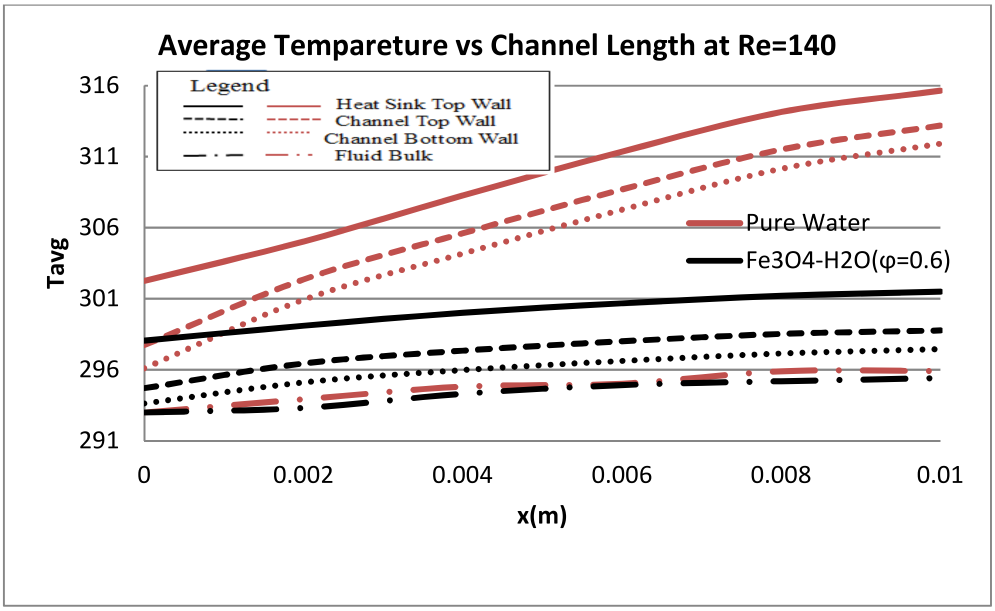

Figure 5d: Average temperature distribution versus channel length for Re=140, ϕ=0.6

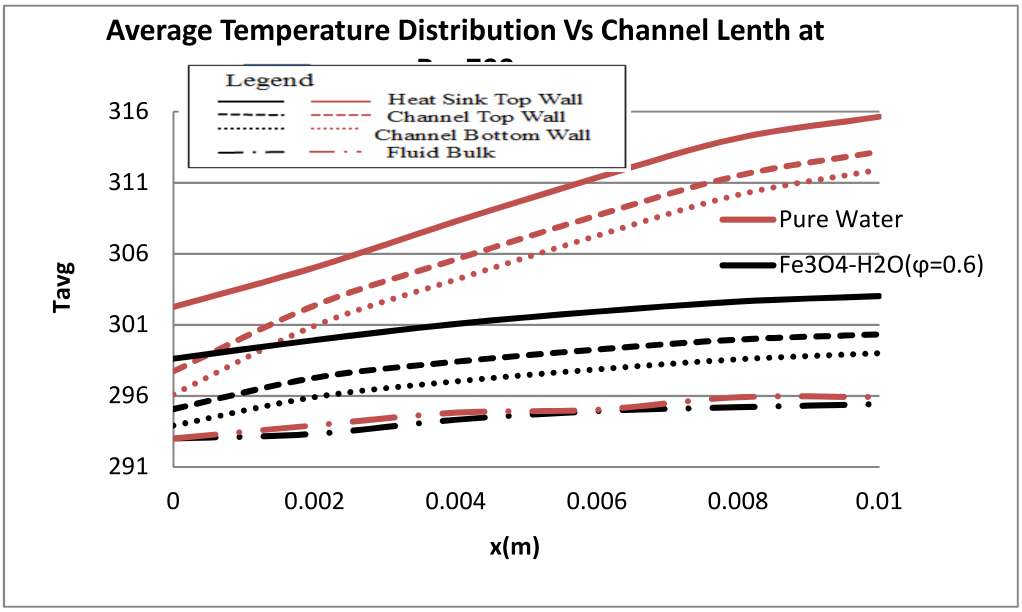

Figure 5e: Average temperature distribution versus channel length for Re=700, ϕ=0.6

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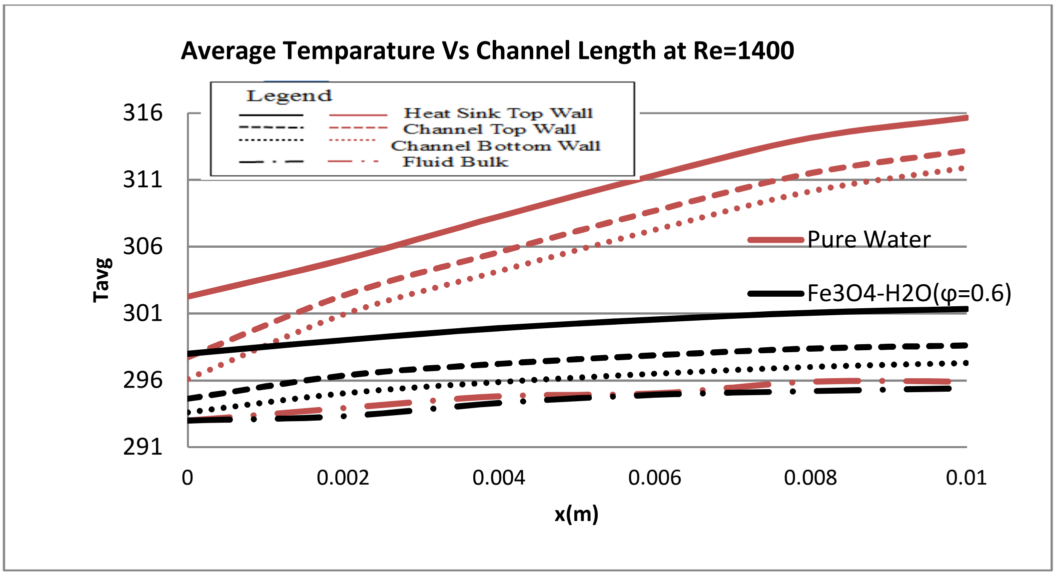

Figure 5f: Average temperature distribution versus channel length for Re=1400, ϕ=0.6

Figures 5a – 5f above shows the average temperature distribution for pure water and Fe_3O_4-H_2O as working fluids on several surfaces including the fluid bulk in the MCHS. The maximum average temperature can be seen at the heat sink top wall for Reynolds number (140, 700 and 1400) and volume fractions (ϕ=0.4, ϕ=0.6 and ϕ=0.8). It is interesting to note that the presence of nanoparticles has effect of reducing the temperature of the surfaces as particle volume fraction of Fe_3O_4-H_2O increases due to its higher dynamic viscosity and lower heat capacity compared to pure water. When using Fe_3O_4-H_2O with volume fraction of ϕ=0.4 as working fluid it shows 0.04% temperature reduction in comparison with pure water, at ϕ=0.6 volume fraction the temperature reduces to 0.07%, while using ϕ=0.8 volume fraction there is more reduction in temperature to about 0.08%. However, we can deduce that as the volume fraction increases has an effect in reducing the temperature due to an increase in thermal conductivity and lower heat capacity.

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