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A Numerical Investigation of Turbulent Magnetic Nanofluid Flow inside Square Straight Channel

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Abstract – *A numerical study using computational fluid dynamics method with a single-phase approach has been presented in order to determine the effects of the concentration of nanoparticles and flow rate on the convective heat transfer and friction factor in turbulent regime flowing through a square duct with Reynolds number range of $10000 \leq Re \leq 1000000$ using constant applied heat flux around the duct. The nanofluid used consisted of Fe_3O_4 magnetic nanoparticles with an average diameter of 13 nm dispersed in water with six volume fractions (0, 0.6, 0.8, 1, 1.5 and 2%). The results revealed that as volume fraction and Reynolds number increased, Nusselt number increased, and friction factor decreased as Reynolds number increased.*

Keywords: Magnetic Nanofluid, Turbulent Fluid Flow, Convection Heat Transfer, Square Duct, Single Phase Flow

1.0 INTRODUCTION

Nanotechnology has opened new challenges in various streams of fundamental science and engineering applications. The common applied heat carrier fluids like oil, water and ethylene glycol have a low thermal conductivity compared to most solids [1,2]. Improving heat exchangers, changing boundary conditions and enhancing the thermophysical properties of the fluid are some effective ways to decrease energy consumption [3,4].

The first effort in establishing a solid-liquid mixture was by Maxwell [5,6], which was to increase the thermal conductivity of the base fluid using micrometer or millimeter size. This lead to so many problems like fouling, erosion and high losses due to pressure drop [7,8].

A new generation of fluids has been introduced, known as nanofluid and have been used in heat exchangers. The terms have been used first time by Choi [9] in 1995. The new fluids consist of a base fluid (water) and dispersed nanometer-sized particles (1-100 nm) such as copper, aluminum and other oxides. An improvement has been noticed in the thermal properties of the base fluid when nanoparticles with a high thermal conductivity have been mixed with the base fluid [3,10].

Xuan and Li [11] investigated experimentally the convective heat transfer and flow characteristics for a CuO-water nanofluid flowing through a straight tube with a constant heat flux under laminar and turbulent flow conditions. The results of the experiment showed that the suspended nanoparticles remarkably enhanced the heat transfer performance of the conventional base fluid, and their friction factor coincided well with that of the water.

Furthermore, they also proposed new convective heat transfer correlations for the prediction of heat transfer coefficients of the nanofluid for both laminar and turbulent flow conditions.

Tsai et al. [12] investigated gold-deionized water nanofluid flowing in a conventional heat pipe with a diameter of 6 mm and a length of 170 mm. Their data showed that the nanofluid caused a significant reduction in the thermal resistance of the heat pipe compared with deionized water at given concentrations.

Moraveji and Hejazian [13] numerically studied a fully developed turbulent of heat convection nanofluid flow inside a straight channel using CFD method without the influence of magnetic field with Reynolds number ranging between 3000 and 22000. They used magnetic nanofluid consisted of Fe_3O_4 as the nanoparticles with the average diameter of 36 nm and water as a base fluid with nanoparticles volume concentrations of 0.02, 0.1 and 0.6%. They proposed a correlation to estimate Nusselt number and friction factor, which was in good agreement with the experimental work.

Forced convection heat transfer with turbulent nanofluid flow inside a tube was experimentally studied by Sundar et al. [7]. The nanofluid used was magnetic nanoparticles Fe_3O_4 suspended in water with nanoparticles volume concentrations from 0 to 0.6% and the range of Reynolds number from 3000 to 22000. New correlation was proposed to estimate Nusselt number and friction factor. The results revealed that the heat transfer was enhanced by 30.93%.

Heris et al. [14] numerically analyzed laminar flow-forced convective heat transfer of Al_2O_3 water nanofluid in a triangular duct, where a constant wall temperature was applied to the triangle walls. Their results showed that the average Nusselt number increased with increasing nanoparticles concentration and decreased nanoparticles diameter with increasing Reynolds number.

In this paper, the modeling of 3D turbulent flow containing magnetic nanofluid with average nanoparticle diameter of 13 nm having different volume fractions was numerically investigated using CFD tools (FLUENT) in a horizontal straight square duct. The essential target of this research is to examine the effect of flow rate, nanofluid volume fraction and change of Reynolds number in the range from 10^3 to 10^6 without the effect of magnetic field on the thermal and flow field.

2.0 MATHEMATICAL MODELING

In this work, a 3D computational fluid dynamics model was developed for nanofluid turbulent flow inside three geometries based on a single phase approach and steady state. The nanofluid as a single phase fluid with relatively different physical properties such as density, thermal conductivity and viscosity was used. The fluid phase was assumed to be as a continuity phase, and the governing equations (continuity, momentum and energy) are presented as follows:

Conservation of mass:

$$\nabla \cdot (\rho v) = 0 \tag{1}$$

Conservation of momentum:

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$$\nabla \cdot (\rho v) = -\nabla p + \nabla.(\tau) \tag{2}$$

Conservation of energy:

$$\nabla.(\rho V C_p T) = \nabla.(\lambda \nabla T - C_p \rho v) \tag{3}$$

3.0 THERMOPHYSICAL PROPERTIES OF NANOFLUID

The effective properties of the nanofluid are defined as follows [1]:

Density:

$$\rho_{nf} = (1-\phi)\rho_{bf} + \phi\rho_p \tag{4}$$

Heat capacitance:

$$C_{nf} = (1-\varphi)C_{bf} + \varphi C_{Fe3O4} \tag{5}$$

where ρ_{nf} and C_{nf} are the density and specific heat of the nanofluid respectively. The thermal conductivity and viscosity of nanofluid are computed as follows [15]:

Thermal conductivity:

$$k_{nf} = k_{bf}(1+10.5\varphi)^{0.1051} \tag{6}$$

Viscosity:

$$\mu_{nf} = \mu_{bf}(1+\frac{\varphi}{12.5})^{6.356} \tag{7}$$

The turbulence model $k-\varepsilon$ that was proposed by Launder and Splading [16] was used. The $k-\varepsilon$ model introduces two new equations; one for the turbulent kinetic energy and the other is for the turbulent dissipation rate. The two equations are expressed as the following form:

$$\nabla.(\rho_m V_m k) = \nabla.(\frac{\mu_{t,m}}{\sigma_k}\nabla k) + G_{k,m} - \rho_m \varepsilon \tag{8}$$

$$\nabla.(\rho_m V_m k) = \nabla.(\frac{\mu_{t,m}}{\sigma_k}\nabla \in) + \frac{\varepsilon}{k}(C_1 G_{k,m} - C_2 \rho_m \varepsilon) \tag{9}$$

where

$$\mu_{t,m} = \rho_m C_\mu \frac{k^2}{\varepsilon} \tag{10}$$

$$G_{k,m} = \mu_{t,m}(\nabla V + (\nabla V_m)^T) \tag{11}$$

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with $C_1 = 1.44, C_2 = 1.92, C_\mu = 0.09, \sigma_k = 1$ and $\varepsilon_k = 1.36$.

3.0 Physical model

A 3D square duct that has been examined in this study with w= 0.01 m and L= 1.8 m with the applied heat flux of 50 w/cm^2 is shown below.

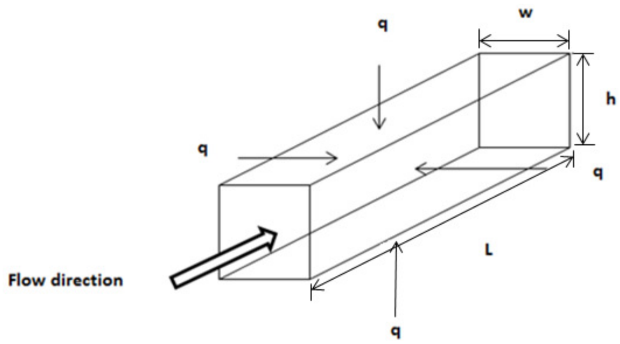

Figure 1: Geometry of the present study

4.0 BOUNDARY CONDITIONS

At the channel inlet, the profiles of uniform axial velocity, V_0 and temperature $T_0 = 293k$ were assumed. At the channel exit section, the fully developed conditions were considered, i.e., all axial derivatives are zero. On the channel wall, the non-slip conditions and uniform heat flux were imposed, while both turbulent kinetic energy and dissipation of turbulent kinetic energy were equal to zero.

5.0 NUMERICAL METHOD IMPLEMENTATION AND VALIDATION

The computational fluid dynamic code FLUENT [17] was used to solve the problem. The governing equations (1)-(3) were solved by control volume approach. This method is based on the spatial integration of conservation equations over finite control volumes. The conversion of the governing equations to a set of algebraic "discretized equations" resulting from this spatial integration process was sequentially solved throughout the physical domain. FLUENT [17] solves the systems resulting from the discretization schemes using a numerical method. In order to ensure the accuracy and consistency of numerical results, several uniform grids have been subjected to an extensive testing procedure for each case considered.

The results have shown that for the problem under consideration, the $60 \times 60 \times 151$ uniform grid appears to be satisfactory in ensuring the accuracy of the numerical results, as well as their independency with respect to the number of nodes used.The results are successfully validated with the correlations reported for thermally and hydraulically developed flow.

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6.0 RESULTS AND DISCUSSIONS

The results were obtained using a single phase approach for volume fraction of $\varphi =$ 0,0.6,0.8,1,1.5 and 2% with the range of Reynolds number of $10000 \leq Re \leq 1000000$. The average heat transfer coefficient and the average Nusselt number for turbulent flow were computed using the following formulas [17]:

$$h = \frac{q}{T_{wall} - T_b} \quad (12)$$

$$Nu = \frac{h \times D}{k} \quad (13)$$

where q, h, D, k, T_{wall} and T_b are the heat flux, average heat transfer coefficient, hydraulic diameter, thermal conductivity of the fluid, average wall temperature and bulk temperature respectively.

6.1 Effect of Nanoparticle Volume Fraction on Nusselt Number

Fig. 2 represents a plot of Nusselt number versus Reynolds number for various volume fractions of the nanoparticles.

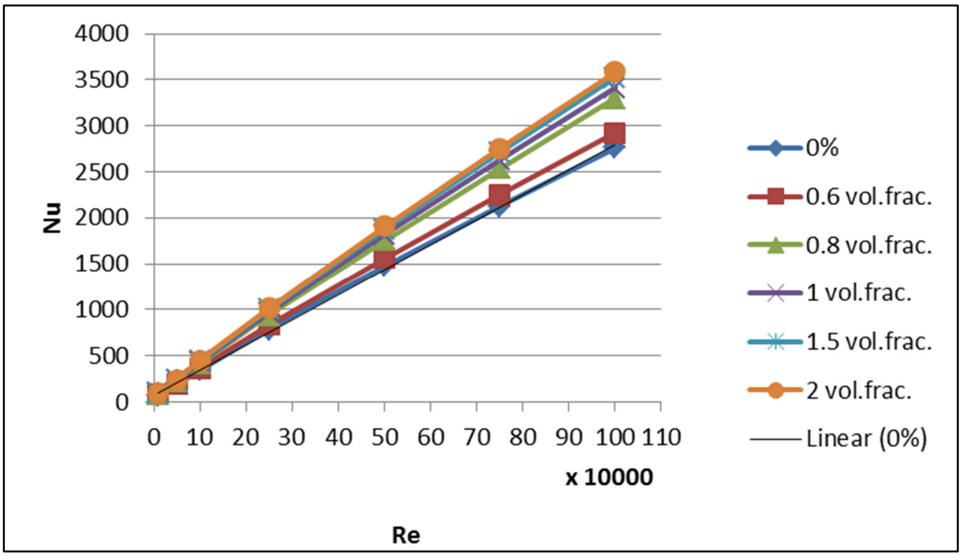

Figure 2: Effect of nanofluid concentrations and Reynolds number on average Nusselt number.

It is obvious that increased Reynolds number and volume fraction of nanoparticles lead to the increase of the value of Nusselt number. According to Eq. 5, nanofluid with higher volume fraction has higher thermal conductivities, which in turn increases the heat transfer coefficient. For example, in the case of 0.8% volume fraction and Reynolds number of 50000, the mean heat transfer coefficient is about 19.15% larger than the base fluid (pure water). The obtained

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results are in good agreement with Pak & Choi's correlation [18] for the prediction of Nusselt number more than Gilinisk's correlation [19].

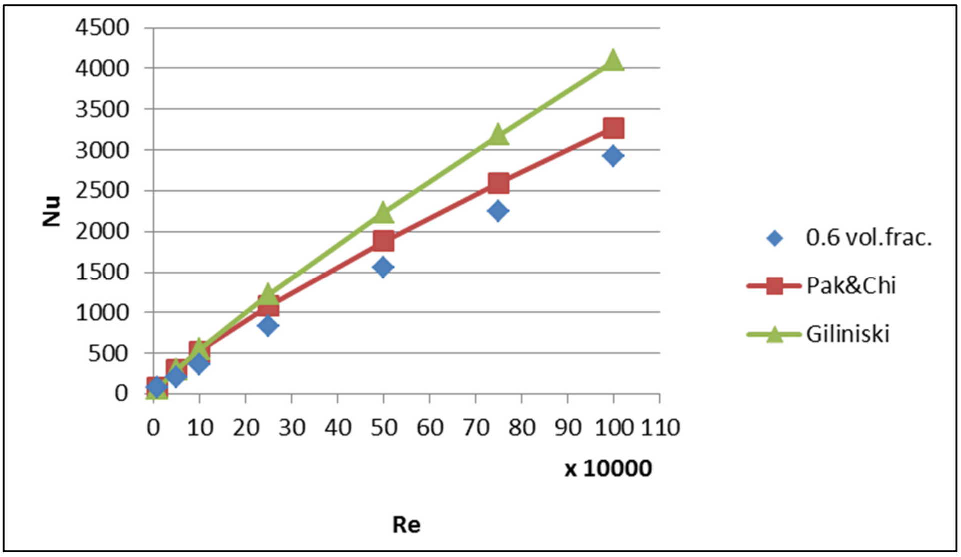

Figure 3: Comparison of computed results with the correlations at 0.6% volume fraction

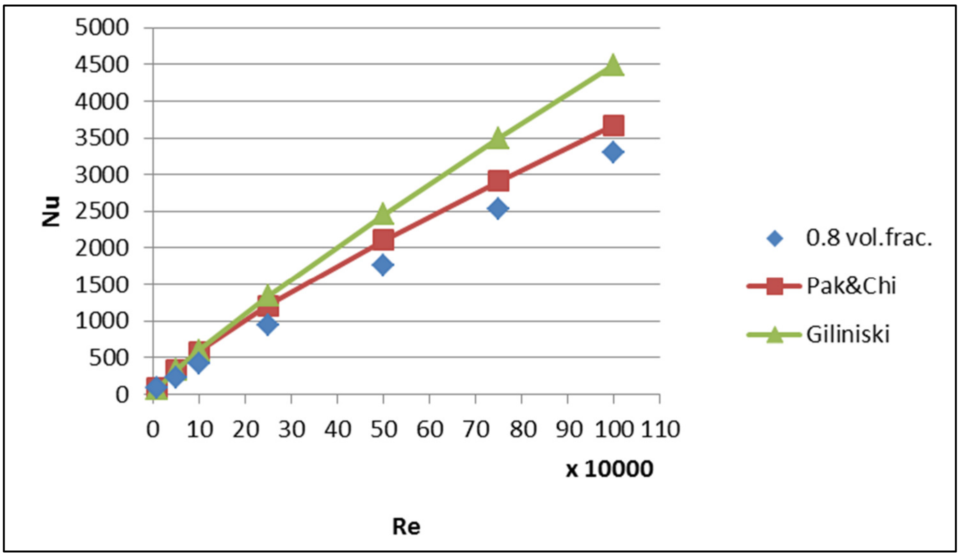

Figure 4: Comparison of computed results with the correlations at 0.8% volume fraction

6.2 Effect of Nanoparticle Volume Fraction on Friction Factor

Fig. 5 shows the friction factor for pure water compared to Blasius formula [20]. It is obvious from the figure that the friction factor for square channel decreased as Reynolds number increased.

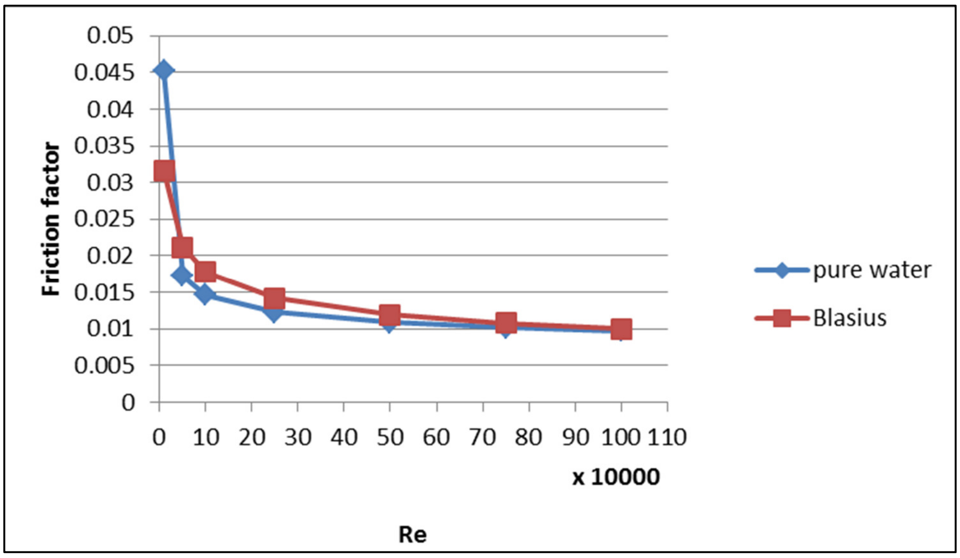

Figure 5: Comparison of computed results of friction factor of pure water with the correlation of Blasius

7.0 CONCLUSIONS

In this article, numerical investigations were carried out to investigate the effect of flow rate and nanofluid concentration on the average Nusselt number and friction factor using commercial software. Important conclusions drawn from this work include:

1. The Nusselt number and the amount of heat transfer increase with the increasing of Reynolds number and volume concentration of nanoparticles.

2. Friction factor decreases as Reynolds number increases.

3. By applying the simulation results, an equation of Nusselt number prediction based on the dimensionless numbers was correlated.

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