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To cite this article: Kadhum Audaa Jehhef *et al* 2019 *IOP Conf. Ser.: Mater. Sci. Eng.* **518** 032004

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Convection Heat Transfer Enhancement in Square Cross-Section with Obstacle Using Nanofluids

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Abstract. The characteristics of heat transfer and pressure drop in turbulent flow of Al_2O_3 and CuO /water nanofluid that flowing in a horizontal straight square cross-section duct with obstacle has been studied experimentally and numerically. The experimental work is attempted by using a water and nanoparticles concentration of ($\phi = 0.5$ to 2.0 % vol.) that flows in an aluminium square duct with side dimensions of $a = 20$ mm and length of $L = 1000$ mm. The obstacle located at the middle of the duct with different obstacle height to pipe side ratio varied as ($h/a = 0.0, 0.25, 0.5$ and 0.75). The water entered to the duct with inlet Reynolds number of turbulent flow of 3500, 4300, 5200 and 6800. Constant heat flux of 6500 W/m^2 is applied upon the external walls of the duct. The numerical finite volume method with appropriate boundary conditions used with ANSYS-Fluent v.16.2 was employed to simulate the temperatures and velocity of the nanofluids. The experimental and numerical results showed that the average Nusselt number increases with increasing the obstacle height ratio and with using nanofluids as compared with pure water. Moreover, the Nusselt number increases with increasing the nanoparticles concentrations in the nanofluids. In addition, the results indicated that increasing in the obstacle height ratio and increasing the nanoparticles, concentration leads to increasing the friction factor and pressure drop. Finally, the numerical results give good indication to the presence of the recirculation cell zone behind the obstacle.

1. Introduction

Performance of many heat transfer equipment that use liquids (water, ethylene glycol or mineral oil) to transfer heat can be improved by manipulating its liquid efficiency. Physically known that solids have heat conductivity several times higher than that of liquids, so many studies had been accomplished by mixing solid particles with the liquid to increase its heat conductivity. Many researchers had used non-metallic particles for this purpose. This process exposed by some difficulties as the settlement of the solid particles inside the holding device as pipes or duct or any other type, pressure drop down, increasing friction between slurry and touched surface which increases erosion and/or corrosion. For specific liquid, this depends on the particle material and its physical properties, shape and size, its mixing percentage and the quantity of heat flux within the system in addition to cross section size, shape and alignment (vertical, horizontal, straight, helical or coiled etc...) of holding device. The challenges are to find sustainable or sophisticated heat transfer system that stands against these difficulties. The



magnesium oxide MgO nanoparticles was mixed in distilled water with different volume fractions (0.3, 0.4, 0.55 and 0.75%) [1], the result presented an enhancement of heat transfer as the volume fraction increases. The magnetic nanofluid was produced by mixing magnetic nanoparticles with liquids also used in this field [2]. He indicated in his review that many researchers used magnetic nanofluid and they found that it enhances heat transferred. The heat transfer of different volume concentration of Cu /ethylene glycol nanofluid (0.011, 0.44 and 0.171) that flowing inside concentric annular pipe with ethylene glycol fluid was correlated by [3]. They found that the coefficient of convective heat transfer of their study is higher than the base fluid. Moreover, the coefficient increases as the nanoparticle's volume fraction increases. Various concentrations of CuO/water nanofluid (0.3, 0.6, 1, 1.5 and 2% wt.) was used by [4], with stirrer speeds of (500, 1000 and 1500 rpm) and heating temperatures of (40, 45, 50 °C) flowing in heat exchanger as shell and then as helical coil. The result of the experiments was compared with heat transfer rate of water. Enhancement of heat transfer rate has been reported due to using nanofluid in all the experiments. The TiO₂/water nanofluid with overlapped dual twisted tapes was used by [5]. Different Reynolds numbers (5400, 15200) were used and different ratios of overlapped twist (1.5, 2, 2.5) and TiO₂ concentrations (0.07, 0.14, 0.21%). The results indicated that with 1.5 overlapped twist ratio heat transfer enhancement ratio is 89%. In addition, using twist ratio of 1.5 with nanofluid of TiO₂/water the heat transfer increases as the volume concentration of TiO₂ increases. The Ag-CuO/water hybrid nanofluid with heat generation, thermal radiation and chemical reaction was used by [6]. They had examined it on linearly striate surface. They found that hybrid nanofluid boosts both the heat transfer. The temperature distribution was investigated by [7]. The effect of the particle shape of Cu-CuO hybrid nanofluid on the heat transfer rate. Analysis of three different shape were carried out. The concluded indicated that brick shape causes low temperature distribution but particles having platelet shape induce efficient fluid flow. The experimental work was presented by [8] for different particle fractions (1-4) of hybrid aluminium nitride/ethylene glycol nanofluid, nanoparticle diameter, Reynolds number and heat flux within the strait tube are approximately 30 nm, 5,000 – 17,000, and 5000 w/m² respectively. They found that 1-3% nanoparticle fraction enhanced the heat transfer efficiently but 4% reduces the efficiency to less than 1 [9]. They investigated the effect of cross section's shape, nanoparticle's concentration and Reynolds number on laminar heat transfer of nanofluid consists of TiO₂/water. They found that addition of small amount of nanofluid particles affect positively the heat transferring in conduits. Moreover, they concluded that circular cross section performs better than triangular and square cross section. Ducts with different cross-sections that were studied by [10], to investigate the influence of Al₂O₃/water nanofluid on their thermal-hydraulic performance. The results they gate indicates that thermal performance of circular cross section is the better then 4:1 rectangular followed by 2:1 rectangular and then the square duct. In addition, they indicated that their pressure drop ordered as the best one is square ducts then 2:1 rectangular followed by circular and the last is the 4:1 rectangular. The behaviour of ZrO/water nanofluid in tubs with square cross section was studied by [11]. They consist on the effect of ZrO/water concentration, ZrO nanoparticle size and Reynolds number on thermal entropy generation. They concluded that at constant Reynolds number, thermal entropy increases as the nanoparticle size increased and decreases as the particles concentration increased.

As presented above there is a few works concerned with the effect of the longitude obstacle inserted in the square duct, thus, this study presented experimentally and numerically analyses of the effect with range of Reynolds number in turbulent flow and the effect of height of the obstacle and booth of Al₂O₃ and CuO nanofluid on the heat transfer parameters.

2. Experimental Part

The schematic draw of the experimental rig that employed in this study was represented in Figure 1. A centrifugal pump was located at the upstream of the test duct square section. This was used to recirculate the nanofluid between the rectangular storage tank and test section as

closed loop. Two adjusting valves were used to control the fluid flow rate, the first at the inlet of the tested duct and the second at the by-pass line. The nanofluid flow enters a rotameter to measure the nanofluid flow rate that entered the test section to obtain an inlet Reynolds number for turbulent flow with range of ($Re=3500, 4300, 5600$ and 6800). The pressure drop across the obstacle plate that is inserted in the middle of the square tube was measured by a two differential pressure transducer with a digital screen and of ± 1 Pa uncertainty was employed (manufactured by Endress Hauser) to measure the pressure loss inside and along the tested tube cross section. Experimental study was conducted to study the effect of the different obstacle height to pipe side ratio as (h/a) as (0, 0.25, 0.5 and 0.75). Eight T-type thermocouples were mounted on the external surface of the square duct walls at equal intervals about (10 cm) for measuring the wall temperature. While the bulk temperature was measured by two K-type thermocouples, which have been inserted at two locations, one in the entrance and the other at exit of the test section. The electrical heater with constant power of (400 W) was wrapped on the square test duct to give a uniform constant heat flux. Also, the square section test tube was thermally isolated from its beginning using fiberglass to minimize the heat loss and to guarantee hydro-dynamically full-developed condition. The reservoir tank was made of transparent acrylic plastic. This tank was intended to hold the nanofluid and to monitor the dispersion behavior and the stability of the nanofluid. In addition, the whole tank was thermally isolated with a fiberglass cover to minimize the heat loss from the nanofluid tank supplier to the surrounding area.

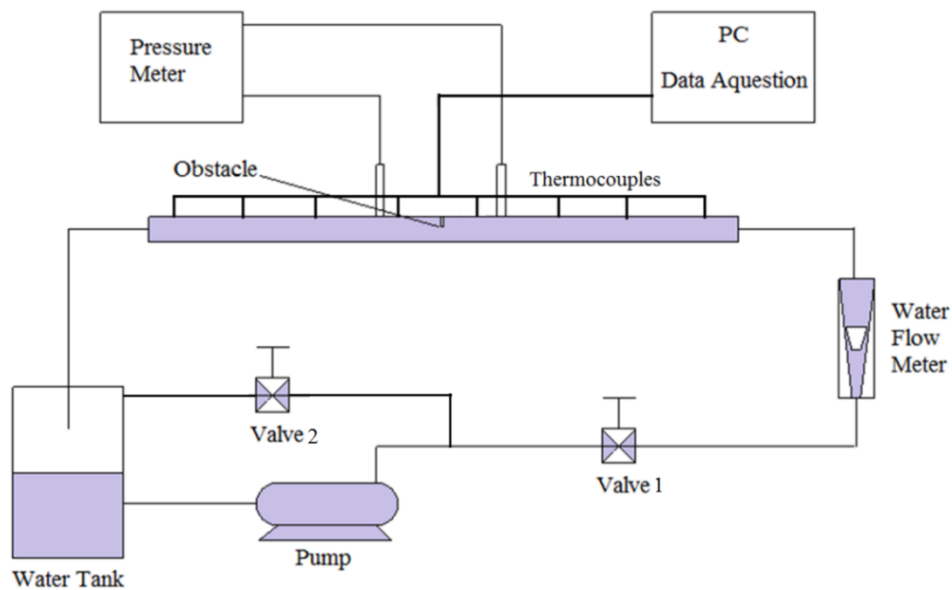


Figure 1. Simple schematic of the employed experimental rig.

While Figure 2 shows the straight horizontal square aluminium tube as a test section with 1 m (1000 mm) long. The tested nanofluid flowing into (20 mm) side length and (0.56 mm) thickness inner aluminium square duct.

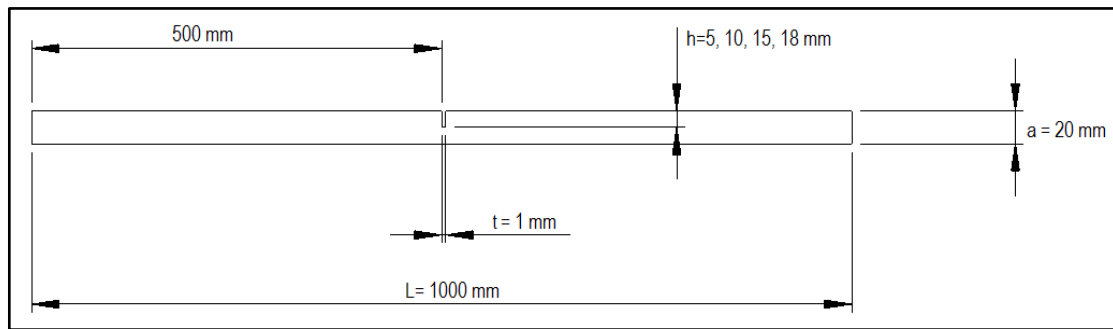


Figure 2. Test section dimensions used in the numerical work.

2.1 Nanofluid Preparation

Nanoparticles used in this study are CuO and Al₂O₃ with sizes of 30–50 nm respectively, supplied by (Nanostructured & Amorphous Materials, Inc., USA). Nanofluids are suspensions of nanoparticles in a liquid base, which is usually water. Two types of nanoparticles Al₂O₃ and CuO are used in this experimental study. The thermophysical properties of the two types of the nanoparticle and the base fluid are listed in **Table 1**. In this study, the two-steps method was used for nanofluid preparation **Karamallah and Jehhef [23]**. Using the electronic gram scale, the desired mass of nanopowder was fixed firstly. The weighted powder of Al₂O₃ or CuO then dissolved in the deionized water (DIW).

Table 1. The thermophysical properties of water and nanoparticle which used in this study, at T=300K.

Thermophysical Properties	Water	Al ₂ O ₃	CuO
Density, $\rho(\text{kg/m}^3)$	998.2	3970	6500
Specific heat $C_p(\text{J/Kg K})$	4182	765	535.6
Thermal conductivity, $K(\text{W/m K})$	0.6	40	20
Dynamic viscosity, $\mu(\text{Ns/m}^3)$	0.001003	0	0

The 3000ml volume of (CuO and Al₂O₃)/water nanofluid mixtures of was mixed by slowly stirring and ultrasonic vibration sonicator type **Yo Xun 3560** with power of 50W for 15-20 minutes' time period to break up any aggregates of nanoparticles. All the suspensions had an ink-like appearance. In this research during the experiments, four volume concentrations were used (0.5, 1.0, 1.5 and 2.0 % vol.) for each nanoparticle of Al₂O₃ and CuO. **Figure 3** shows the prepared samples of the employed nanofluids of CuO/W and Al₂O₃/W. The amount of nanoparticles required for a particular volume concentration can be calculated as bellow equation [12]:

$$\phi\% = \frac{(m_p / \rho_p)}{(m_p / \rho_p) + (m_f / \rho_f)} \quad (1)$$

where

$$m_p = \left(\frac{\rho_p}{\rho_f} \frac{\phi}{(1-\phi)} \right) m_f \quad (2)$$

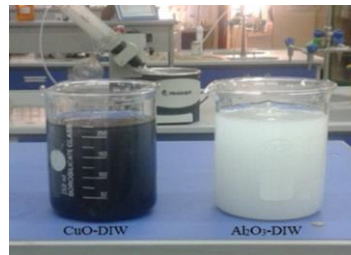


Figure 3. The nanofluid samples of (CuO/W, Al₂O₃/W) at $\phi=1.0\%$.

2.2 Data reduction

The heat flux applied on the four sides of square duct is given by the applied electrical power as:

$$Q = V \cdot I \quad (3)$$

where V and I are the voltage and current supplied by the heater respectively.

The amount of the heat transferred from the heating wire to the nanofluid is given by:

$$Q = \dot{m}_{nf} C_{p,nf} T_{bo} - T_{bin}$$

Also, the heat flux applied is determined by:

$$\dot{q}_w = \frac{Q}{A_s} \quad (4)$$

Where: A_s is the surface area of the tube, then

$$A_s = \pi DL$$

It is necessary to know the heat transfer coefficient of the nanofluid before any calculations.

The local heat transfer coefficient is given by:

$$h_z = \frac{\dot{q}_w}{\Delta T_z} \quad (5)$$

Where: ΔT_z is the difference between the tube wall temperature of tube (T_w)_z and the temperature of the nanofluid (T_b)_z at distance Z from the entrance of the tube. In addition, from the energy balance in the tube, the mean temperature of nanofluid can be expressed by, Hwang and Jang, 2009 [13]:

$$T_{b,z} = T_{bin} + \frac{\dot{q}_w P}{\dot{m}_{nf} C_{p,nf}} z \quad (6)$$

Thus, the local heat transfer coefficient becomes:

$$h_z = \frac{\dot{q}_w}{T_{w,z} - T_{b,z}} \quad (7)$$

The local Nusselt Number has been calculated from the following equation:

$$Nu_z = \frac{h_z D}{k_{nf}} \quad (8)$$

In general, the entrance region length is within 2% of the total tube length. Therefore, the fully developed laminar flow regime can be assumed. The average value of Nusselt number in the thermal fully developed region can be expressed by the integral:

$$\overline{Nu} = \frac{1}{L} \int_0^L Nu_z dz \quad (9)$$

A normalized Nusselt number is defined as the ratio of Nusselt number at any volume fraction of nanoparticles to that of distilled water or,

$$Nu_{avg} = \frac{Nu_{\phi}}{Nu_{\phi=0}} \quad (10)$$

And, the local prandtl number is

$$Pr_{nf_z} = \frac{\mu_{nf} C p_{nf}}{k_{nf}} \quad (11)$$

And, the Reynolds number is

$$Re_{nf_z} = \frac{\rho_{nf} \bar{w} D}{\mu_{nf}} \quad (12)$$

The average Nusselt number calculated by Eqs.(7) was compared with the following Dittus–Boelter equation [Dittus and Boelter (1930) [14] in a square duct with water flow for a constant heat flux condition according to the Reynolds number of 400 as shown in Figure 4.

$$Nu = 0.023 Re^{0.8} Pr^{1/3} \quad 0.7 \leq Pr \leq 160 \quad Re_h > 10000 \quad (13)$$

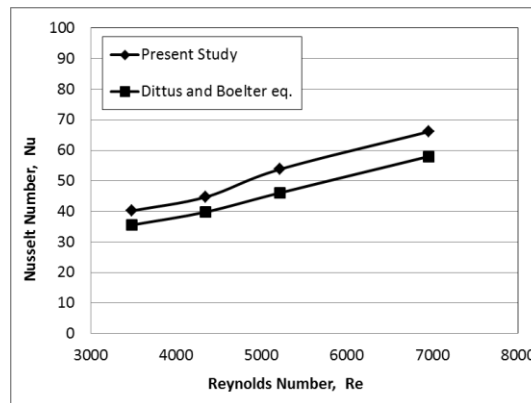


Figure 4. Comparison between Nusselt number for the present experimental results and that of Dittus and Boelter equation [14].

2.3 Calculation of Pressure Drop of the Nanofluids

The pressure drop of Al_2O_3 and CuO / distilled water nanofluids flowing through a square tube was experimentally measured to investigate flow characteristics of the nanofluids. Based on the pressure drop of Al_2O_3 and CuO – distilled water nanofluids, one can express the Darcy friction factor, which is a dimensionless parameter defined as:

$$f = \frac{-\frac{dp}{dx} d_{tube}}{\frac{\rho u_m^2}{2}} \quad (14)$$

Along the fully developed flow the friction factor is given by Kays and London [15]:

$$f = \frac{2\Delta P D_h}{L \rho U_{av}^2} \quad (15)$$

The present experimental values of the friction factor results were validated with the friction factor for a fully developed turbulent flow in a circular pipe of Blasius formula, the results showed a good agreement as plotted in Figure 5, as:

$$f = \frac{0.316}{Re_h^{0.25}} \quad (16)$$

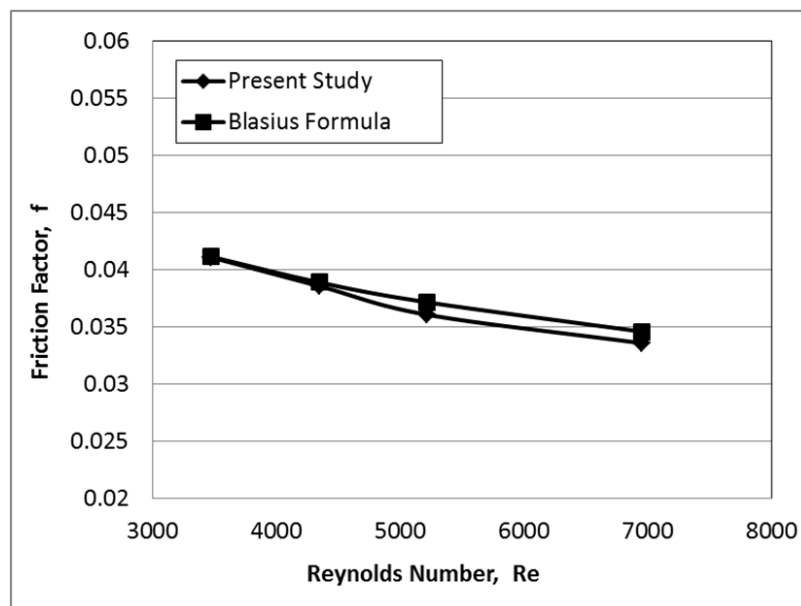


Figure 5. Comparison of the friction factor between the present experimental results and that of Blasius Formula.

3. Mathematical Modelling

3.1 Problem Formulation

The numerical solution analysis is considered by two-dimensional (2D) of the forced heat transfer convection in a heated square duct was carried out. Computational Fluid Dynamics (CFD) using ANSYS-Fluent v.16.2 software based on finite volume method was applied to investigate flow characteristics and the heat transfer within horizontal square duct having obstacle. In the finite volume method, the flow domain discretized into a finite set of control volumes called mesh or cells, and then the governing convection equations, momentum and energy were applied for each cell. A schematic diagram of the computational domain and physical problem and boundary conditions are shown in **Figure 6**. In the numerical section, the duct side is designed to be ($a=20$ mm) and the obstacle height (h) is of (0, 2.5, 5, 7.5, 10, 12.5 and 15 mm). Based on Reynolds number, the left end of the channel is designated as velocity inlet, while the other side of the channel is the exit which subjected to outlet fluid pressure, in the same time the bottom and top walls are exposed uniformly to heat flux q of (3000, 4000, 5000, 7000) in W/m^2 . At the outlet, the boundary condition set as atmospheric pressure at default value (0 Pa for gage pressure).

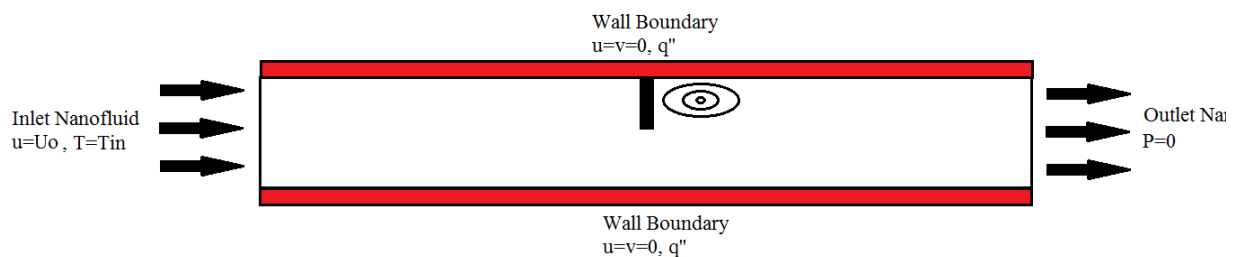


Figure 6. Problem geometrey and dimenssions of the present study.

3.2 Governing Equations

The forced convection heat transfer is carried out by the continuity, momentum, and energy equations which assumed at steady state, constant thermal and physical properties Incropera [16].

Using the bellow continuity equation:

$$\frac{\partial}{\partial x_i} \rho u_i = 0 \quad (17)$$

The momentum equation is:

$$\frac{\partial}{\partial x_j} \rho u_i u_j = \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \frac{\partial u_i}{\partial x_j} + \frac{\partial u_i}{\partial x_j} - \frac{2}{3} \delta_{ij} + \frac{\partial u_i}{\partial x_j} \right] + \frac{\partial}{\partial x_j} - \rho \overline{u'_i u'_j} \quad (18)$$

Reynolds-averaged approach of turbulence modeling requires Reynolds stresses $\overline{u'_i u'_j}$, also k- ϵ turbulence model was chosen for closure of the equations, the. A common method employs the Boussinesq hypothesis to relate the Reynolds stresses to the mean velocity gradients given by:

$$-\rho \overline{u'_i u'_j} = \mu_t \frac{\partial u_i}{\partial x_j} + \frac{\partial u_i}{\partial x_j} \quad (19)$$

The energy equation is:

$$\frac{\partial}{\partial x_i} \rho u_i T = \frac{\partial}{\partial x_i} \Gamma + \Gamma_t \frac{\partial T}{\partial x_i} \quad (20)$$

where Γ and Γ_t are molecular thermal diffusivity and turbulent thermal diffusivity, respectively and are given by:

$$\Gamma = \frac{\mu}{Pr} \quad \text{and} \quad \Gamma_t = \frac{\mu_t}{Pr_t} \quad (21)$$

3.3 Turbulence Model

In this study, the turbulent physical phenomenon of nanofluids was described by the of k- ϵ turbulence model Launder [17]. This model was presented by two additional equations were introduced from the standard k- ϵ turbulent model. The turbulent kinetic energy (k) given by:

$$\frac{\partial}{\partial x_i} \rho k u_i = \frac{\partial}{\partial x_i} \left[\mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_i} \right] + G_k - \rho \epsilon \quad (22)$$

Similarly the dissipation rate of turbulent kinetic energy, ϵ is given by the following equation:

$$\frac{\partial}{\partial x_i} \rho \epsilon u_i = \frac{\partial}{\partial x_i} \left[\mu + \frac{\mu_t}{\sigma_\epsilon} \frac{\partial \epsilon}{\partial x_i} \right] + C_{1\epsilon} \frac{\epsilon}{k} G_k - \rho C_{2\epsilon} \frac{\epsilon^2}{k} \quad (23)$$

And μ_t is the eddy viscosity and it is modeled as:

$$\mu_t = \rho C_\mu \frac{k^2}{\epsilon} \quad (24)$$

In the above equations, G_k represents the generation of turbulent kinetic energy due to mean velocity gradients and is written as:

$$G_k = -\rho \overline{u'_i u'_j} \frac{\partial u_i}{\partial x_j} \quad (25)$$

These empirical constants for the turbulence model are arrived by comprehensive data fitting for a wide range of the turbulent flow Versteeg and Malalasekera [18]:

$$C_\mu = 0.09, C_{1\epsilon} = 1.47, C_{2\epsilon} = 1.92, \sigma_k = 1.0 \text{ and } \sigma_\epsilon = 1.3$$

3.4 Boundary Conditions

(a) At the inlet boundary:

$$u_x = u_{in}, \quad u_y = 0, \quad T = T_{in}, \quad k = k_{in}, \quad \epsilon = \epsilon_{in}$$

At the duct inlet the turbulent dissipation, i.e. ε_{in} and turbulent kinetic energy, i.e. k_{in} , are given by Versteeg and Malalasekera [18]:

$$k_{in} = \frac{3}{2} u_{in}^2$$

$$\varepsilon_{in} = \frac{C_{\mu}^{3/4} k^{3/2}}{L}$$

(b) At the upper and lower boundaries of the entrance section:

$$u_x = u_y = 0, \quad \frac{\partial T}{\partial x} = 0$$

(c) At the upper and lower boundaries of the test section:

$$u_x = u_y = 0, \quad -k \frac{\partial T}{\partial y} = q''$$

(e) At the outlet boundary

$$\frac{\partial u_x}{\partial x} = \frac{\partial u_y}{\partial x} = 0, \quad \frac{\partial T}{\partial x} = 0, \quad \frac{\partial k}{\partial x} = 0, \quad \frac{\partial \varepsilon}{\partial x} = 0$$

3.5 Thermophysical Properties of Nanofluids

Introducing the Nanofluid volume fraction (ϕ), the thermophysical properties as presented in Table 2 of the Nanofluid, namely the density and heat capacity, have been calculated from Nanoparticle and the pure fluid properties at the ambient temperature as follows. The thermophysical properties of nanofluids used in the present study are for water mixture and Al_2O_3 , and CuO is considered.

Table 2. The thermophysical properties equations are used in numerical simulations.

Thermophysical Properties	Equation	Reference
Density, ρ(kg/m³)	$\rho_{nf} = \phi \rho_p + 1 - \phi \rho_f$	Pak, and Cho 1998 [19],
Specific heat C_p (J/Kg K)	$C_{p,nf} = \frac{\phi \rho_p C_{p,p} + 1 - \phi \rho_f C_{p,f}}{\rho_{nf}}$	Xuan and Roetzel [20],
Thermal conductivity, K (W/m K)	$\mu_{nf} = \frac{\mu_f}{1 - \phi^{2.5}}$	Brinkman [21]
Dynamic viscosity, μ(Ns/m³)	$k_{nf} = \left[\frac{k_p + 2k_f - 2\phi k_f - k_p}{k_p + 2k_f + \phi k_f - k_p} \right] k_f$	Maxwell [22]

Maxwell's formula shows that the effective thermal conductivity of nanofluids relies on the thermal conductivity of the spherical particle, the base fluid and the volume fraction of the solid particles.

3.6 Numerical Solution

The schematic physical geometry of the present square pipe is shown in Figure 2. The square pipe has a total length of 1000 mm with a side dimension of 20 mm. Solid-Works 2014 was used to build 2-D of the square pipe. The inlet water velocity was varied as 0.04, 0.06, 0.08 and 0.12 m/s for laminar flow and 0.17, 0.21, 0.25 and 0.33 m/s. At the outlet, the boundary condition set as atmospheric pressure at default value (0 Pa for gage pressure). Quad mesh was used to generate the computational domain by using 73x270 rectangular cells. In order to analysis the grid size by using three different meshes intervals, i.e. 0.1 mm, 0.5 mm and 1 mm. In this study, pressure based solver was used as a default solver setting to solve the steady state problem. The second order upwind scheme based on multidimensional linear reconstruction approach is used. The present problem governing equations are solved by using the finite-volume method by using ANSYS-Fluent v.16.2 (Tan et al., 2012).

3.7 Grid Independent Test

Before conducting the simulation, the computational domain presented in above is tested for grid independence test for better result accuracy as well as time effectiveness. In the present paper, Four different mesh size models were modelled using Design Modular and only one suitable mesh will be selected for simulation model as shown in Figure 7. The model with very fine mesh size will be taken as reference for the other models. To make sure that the results are due to the boundary condition and physics used, not the mesh resolution, mesh independence should be studied in CFD. The standard method for testing grid independence is to increase the resolution and repeat the simulation.

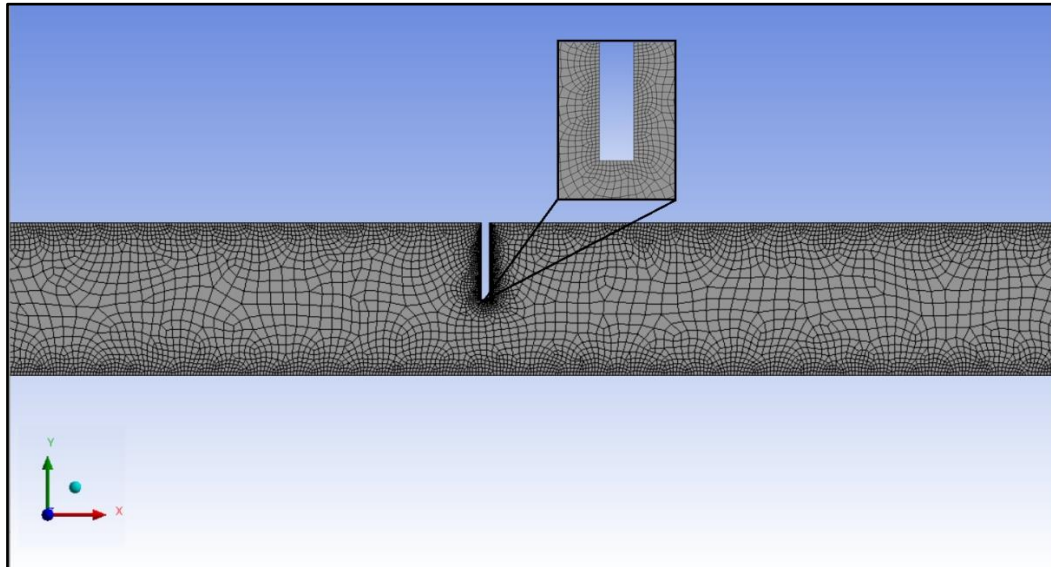


Figure 7. Meshed computational domain

If the results do not change appreciably, this means original grid is probably adequate. Computations have carried out for four selected node was shown in Table 3. It was observed that the 586092 and 1345142 nodes produce almost identical results with error percentage of 0.02%. Hence, a domain with 586092 nodes was chosen to increase the computational accuracy and reduce time of computations.

Table 3. Grid independent test.

Mesh Type	Number of Nodes	Heated wall average Nusselt number	Difference with previous coarse mesh (%)
Course	173801	71.22	-
Medium	424608	68.55	3.715
Fine	586092	67.23	1.912
Very Fine	1345142	66.91	0.425

4. Results and Discussion

4.1 Nusselt number

This work investigated the effect of water and nanofluids inlet Reynolds number and obstacle heights that inserted in a square duct on the heat transfer rate and pressure drop. The profile of the local Nusselt number along the length of the duct was plotted in Figure 8. The profile was changed along the length, it is increasing as the length of the duct increased, and the results showed that Nusselt number increases with increasing the Reynolds number for all cases of the obstacle. The results showed that the curves of Nusselt number was affected by the height of the obstacle, whereas, increasing the obstacle height found to cause increasing the values of the Nusselt number in the zone of the obstacle at (0.45 to 0.55m). In addition, the maximum value

of the Nusselt number is found at the obstacle height ratio of ($h^*=0.75$). The effect of obstacle height ratio was plotted in Figure 9, the results showed that the average Nusselt number increases as the obstacle height ratio increases, the fluid velocity increases upward after it pass the obstacle inside the pipe.

The effect of nanoparticles concentration on the Nusselt number was presented in Figure 10 and 11. The results showed that the Nusselt number increased by 32% when the nanoparticles concentration increased to 2% vol. of Al_2O_3 - water nanofluid at $\text{Re}=6900$ and $h^*=0$. but with obstacle ratio of 0.75 the Nusselt number increased by 37%. Figure 12 and 13 shows the relation of Nusselt number with the CuO nanoparticles concentration. The results declared that the Nusselt number increased by 29% as the nanoparticles concentration increasing to 2% vol. of CuO-water nanofluid at $\text{Re}=6900$ and $h^*=0$. However, with obstacle ratio 0.75 the Nusselt number increased by 33%. This means that Al_2O_3 - water nanofluid gives good enhancement with the aluminium square duct.

4.2 Pressure drop and Friction factor

Practically, reversed flow will occur at the backside of the obstacle plate, and this will generate an eddy that appears because of the sharp static pressure drop behind the obstacle plate. The results show that the maximum fluctuating in the pressure level and friction factor appears at the obstacle height ratio of ($h^*=0.75$) when the obstacle plate height is ($h=15$ mm). The experimental results of the effect of Reynolds Number on the friction factor were plotted in Figure 14. The results show that increasing the inlet Reynolds number lead to increasing the friction factor for Reynolds number Re greater than 2000, this results give good agreement with the co-relation reported in literature by (Moody, 1944). Moreover, the friction factor increased with using a high obstacle height ratio. In addition, the experimental results showed that Al_2O_3 -water nanofluid with different nanoparticles concentration at $h^*=0.75$ leads to increasing the values of the friction factor as plotted in Figure 15.

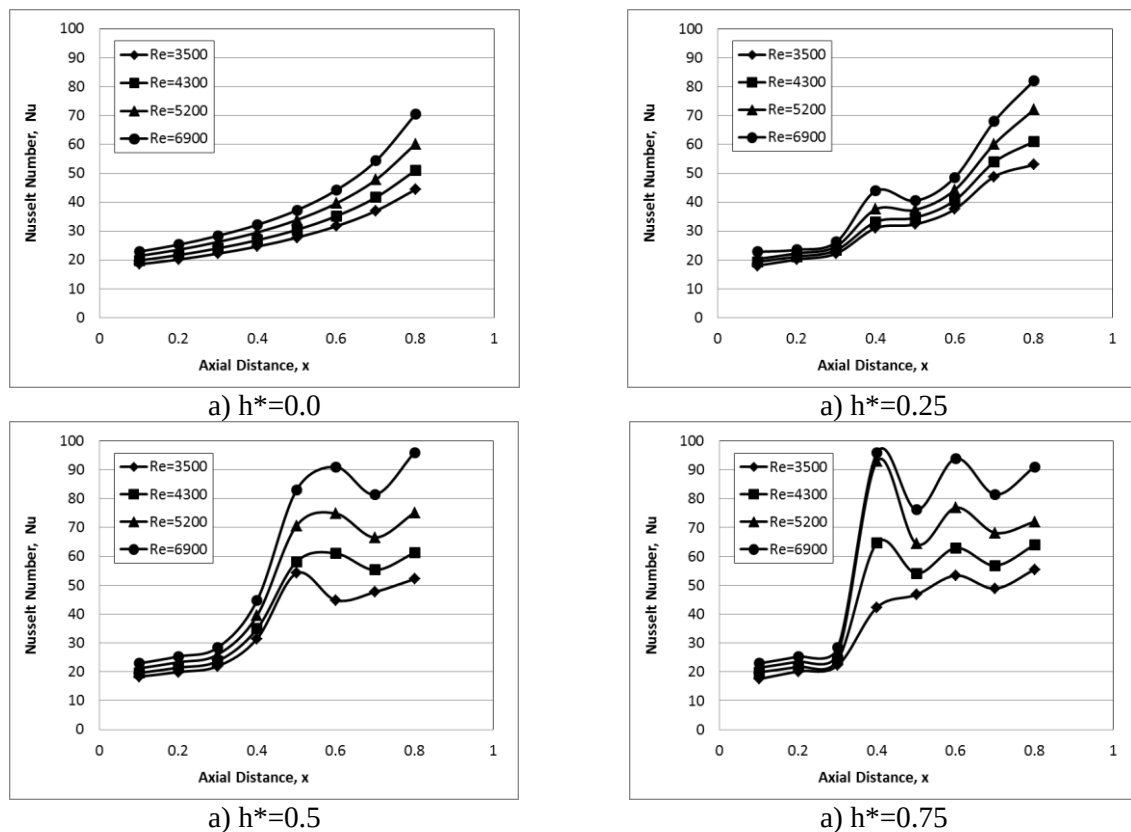


Figure 8. Variation of water Nusselt number via axial distance at different obstacle height ratio.

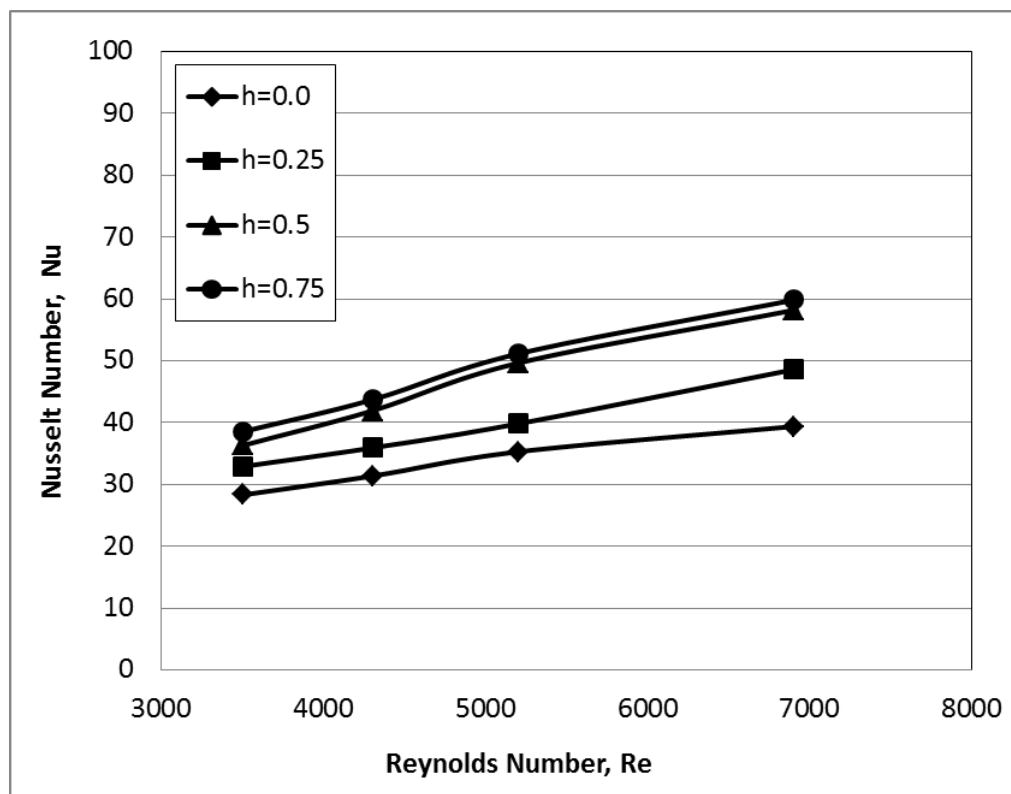


Figure 9. Variation of Nusselt number via Reynolds number of water at different obstacle height ratio.

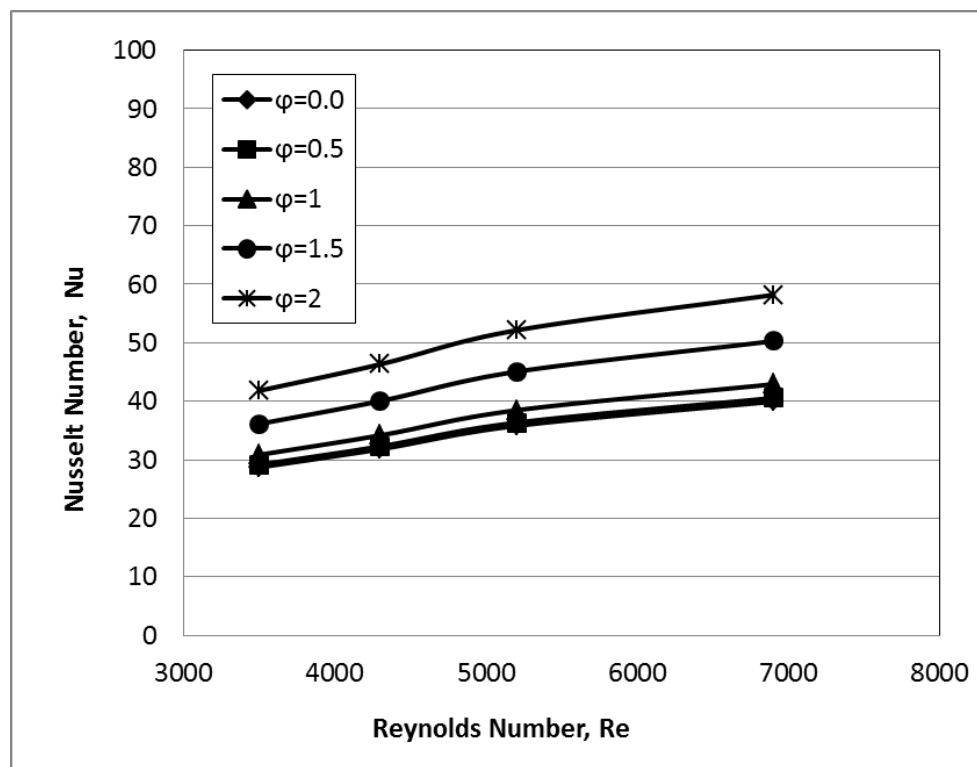


Figure 10. Variation of Nusselt number via Reynolds number for different nanoparticles concentration of Al₂O₃/ water nanofluid at h*=0.

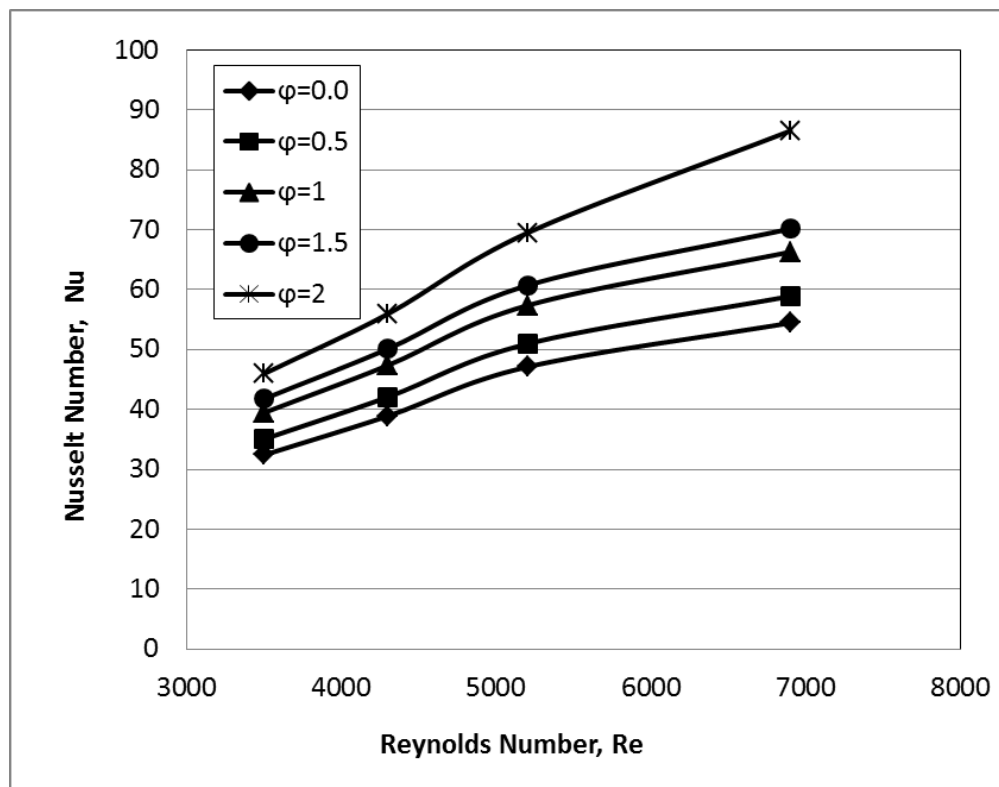


Figure 11. Variation of Nusselt number via Reynolds number for different nanoparticles concentration of Al₂O₃/ water nanofluid at $h^*=0.75$.

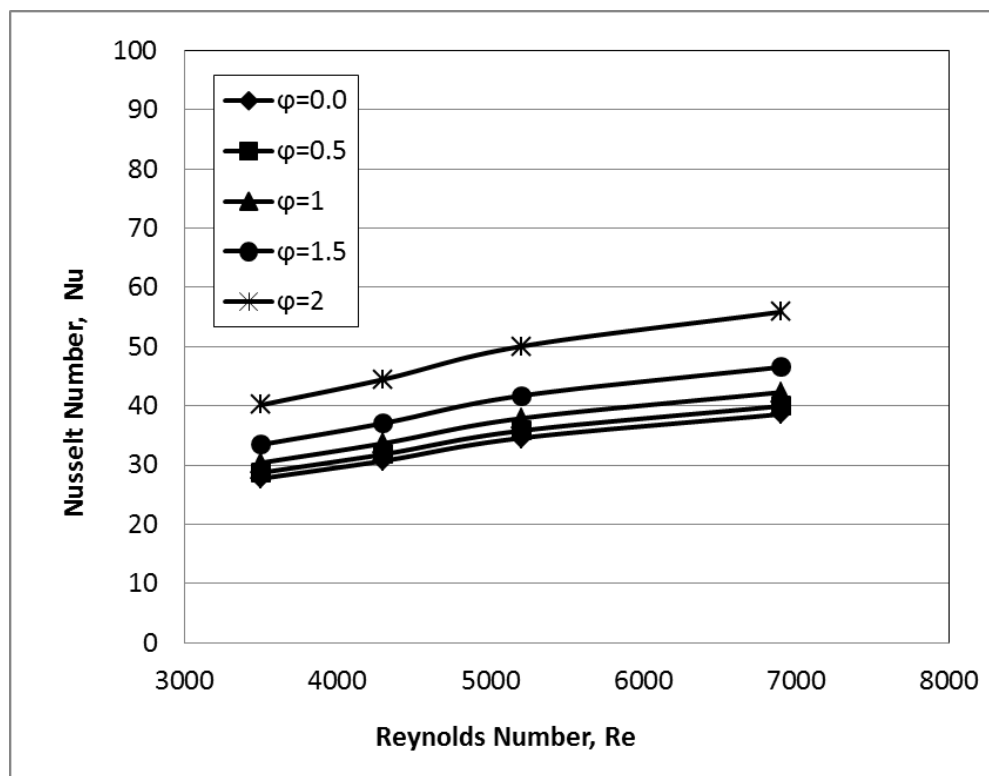


Figure 12. Variation of Nusselt number via Reynolds number for different nanoparticles concentration of CuO/water nanofluid at $h^*=0$.

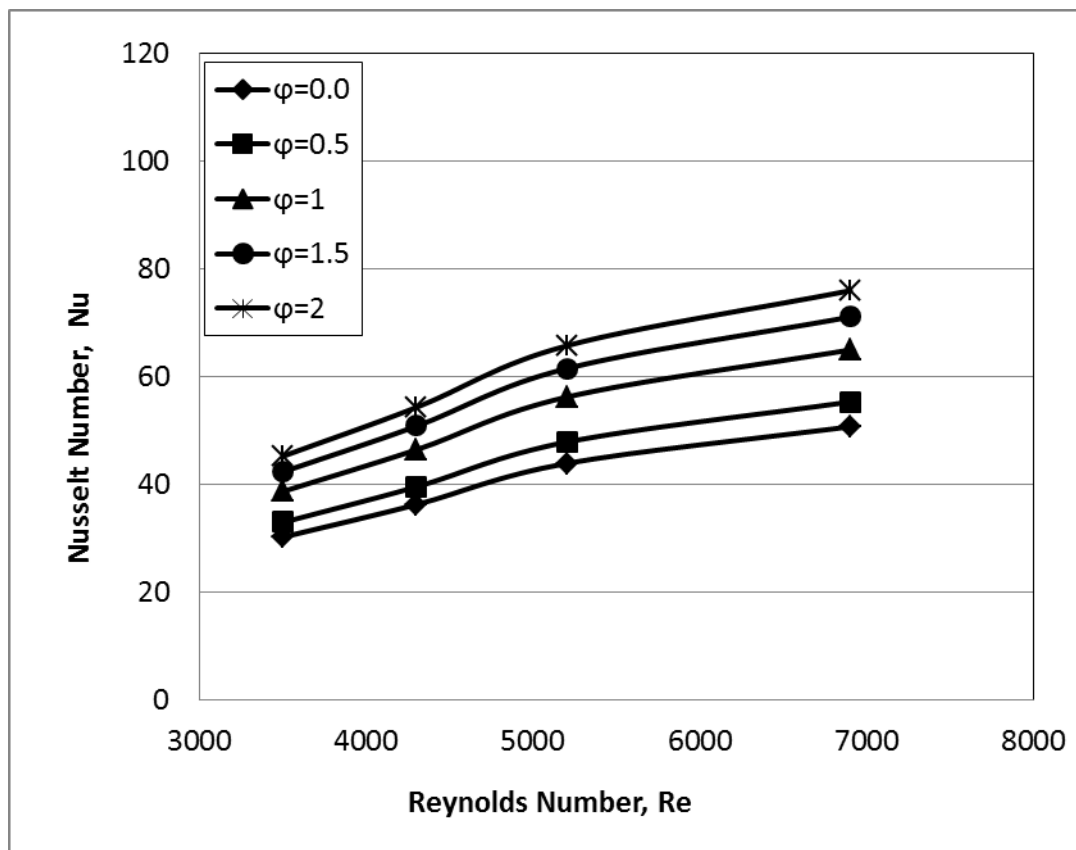


Figure 13. Variation of Nusselt number via Reynolds number for different nanoparticles concentration for CuO- water nanofluid at $h^*=0.75$.

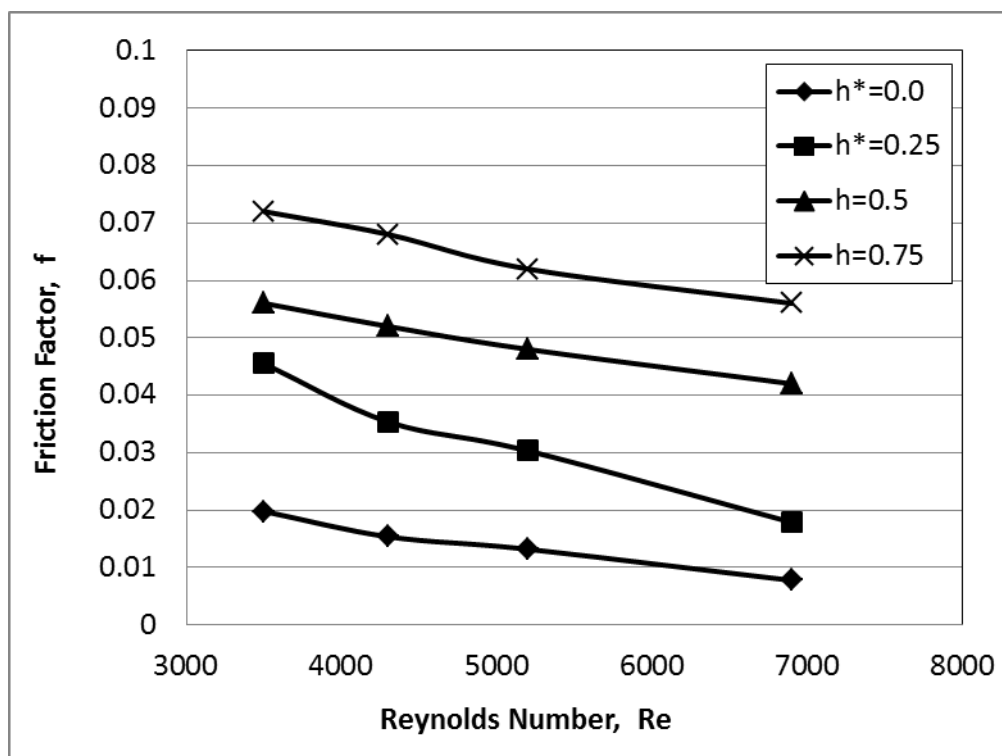


Figure 14. Inlet Reynolds number via Friction factor inside the square duct for different obstacle height ratio.

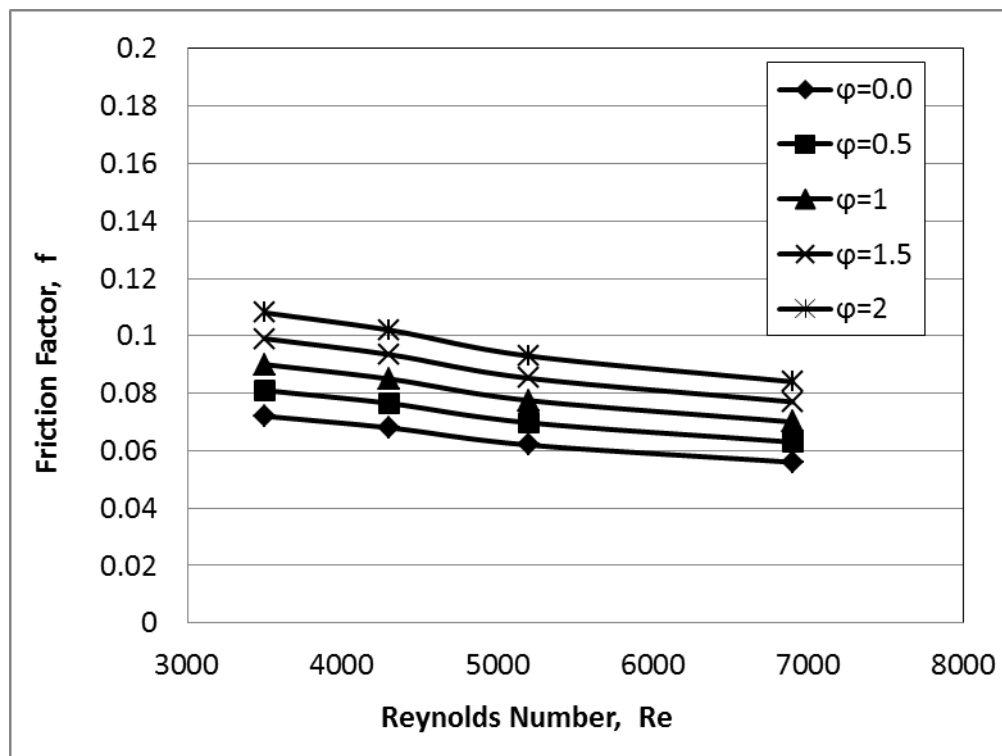


Figure 15. Inlet Reynolds number via Friction factor inside the square pipe of Al_2O_3 /water nanofluid for different obstacle height ratio at $h^*=0.75$.

4.3 Numerical Velocity and Vorticity

In this section, the velocity vectors, velocity contour, and temperatures contour are used to present the vortices region of the rotational flow that caused by presence of the obstacle in the square duct, and are discussed based on CFD ANSYS-Fluent v.16.2 simulation results only. The obstacle is located exactly at mid-way of the 1-m testing pipe length. Figure 16 shows the effect of presence and the height of the obstacle on the nanofluid velocity and the existence of vortices. The simulation results showed that the fluid flows smoothly along the downstream region of the obstacle without any vortices or recalculation zones. However, the zone that in the upstream of the obstacle, the flow shows a circulation cell behind the obstacle. The appearance of the circulation cells behind the obstacle is because of viscous force that has maximum effect at the inner surface of the pipe. Also, it is observed that the shape and the size of the vorticity is changing as Reynolds' s number and/or the obstacle height ratio changes. At high Reynolds number of 6800, the vorticity is formed very close to the obstacle plate while at low Reynolds number of 4000, a small vorticity will generated near the obstacle plate creating a region of low pressure. Increasing the obstacle plate height to ($h^*=0.5$) lead to increase the diameter of the circulation vorticity cell behind the obstacle plate.

The effect of using nanofluids on the temperatures and velocity contour up and downstream of the obstacle when using 2% vol. of (Al_2O_3 or CuO)/ water nanofluid at $h^*=0.75$, $\text{Re}=6800$ was plotted respectively in Figure 17 and 18. The numerical results showed that using Al_2O_3 / water nanofluid will decrease the temperatures inside the core of the circulation zone in better performance compared with CuO /water nanofluid due to higher thermal conductivity of Al_2O_3 nanofluid that flowing in the duct.

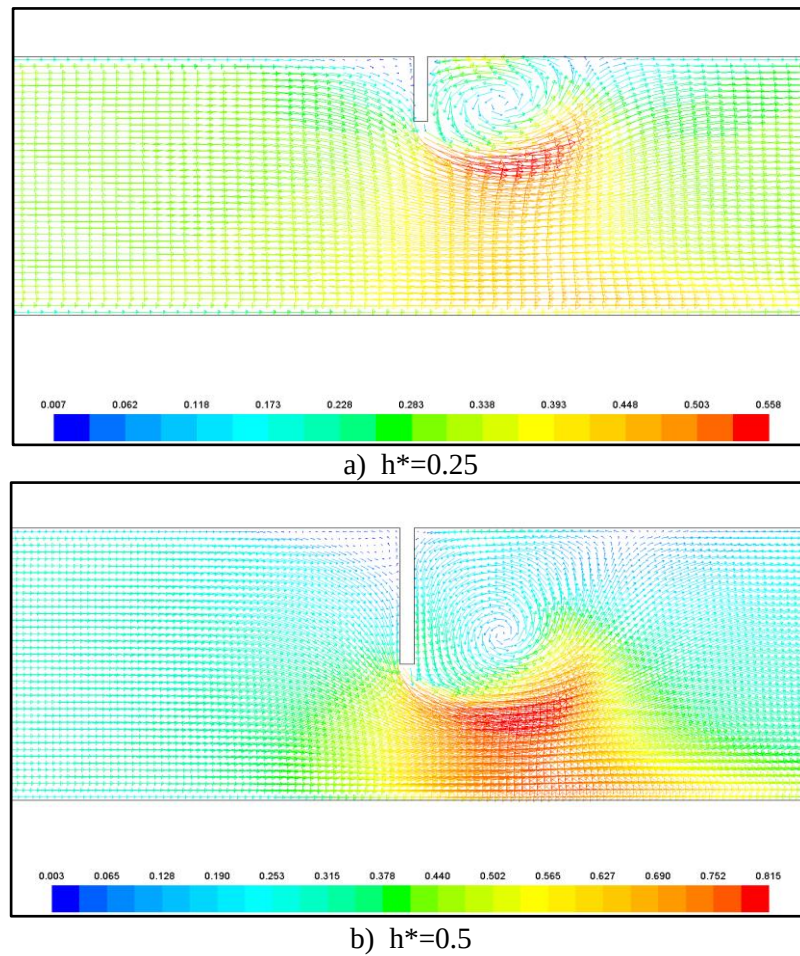


Figure 16. The flow velocity vector up and downstream of the obstacle with height ratio of a) $h^*=0.25$, b) $h^*=0.5$ at Reynolds number $Re=6800$.

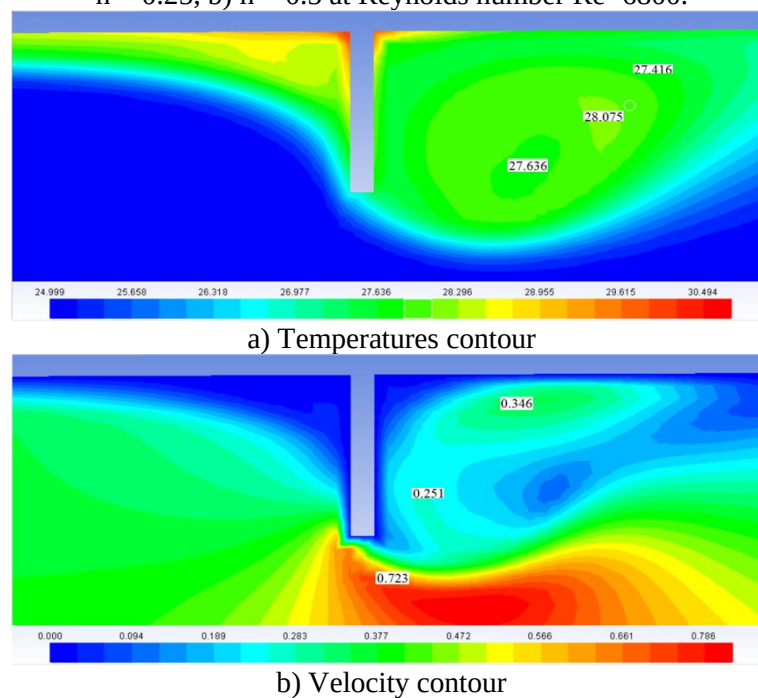


Figure 17. The a)Temperature and b)velocity contour up and downstream of the obstacle for 2% vol. of Al_2O_3 /water nanofluid at $h^*=0.75$, $Re=6800$.

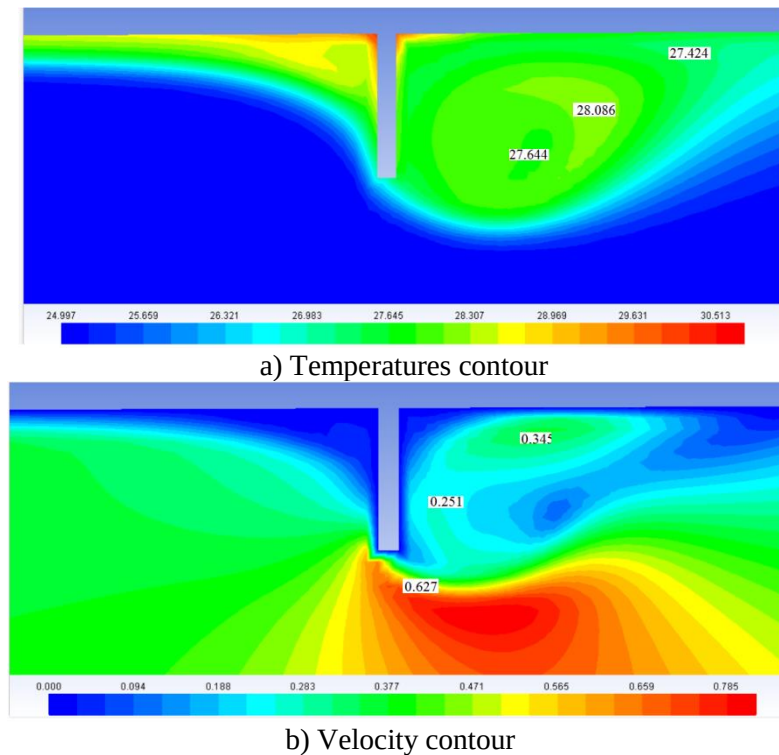


Figure 18. The a)Temperatures and b)velocity contour up and downstream of the obstacle for 2% vol. of CuO/water nanofluid at $h^*=0.75$, $Re=6800$.

5. Conclusions

Experimental and numerical investigation of water based nanofluid flow and the heat transfer rate inside an aluminium square duct with obstacle, turbulent flow condition was considered in this study. The results showed that the average Nusselt number increases with increasing the Reynolds number, with increasing the obstacle height ratio and with using nanofluids as compared with pure water. Moreover, the Nusselt number increases with increasing the nanoparticles concentrations in the nanofluids. In addition, the results indicated that increasing in the obstacle height ratio and increasing the nanoparticles, concentration leads to increasing the friction factor and pressure drop, due to the vortex generation in the upstream region of the obstacle. The numerical results give good indication to the presence of the recirculation cell zone behind the obstacle.

Nomenclatures

a	side length of the square pipe, m
h	obstacle height, m
h^*	obstacle height to square pipe side length ratio, ($h^*=h/a$)
D_h	hydraulic diameter of the square pipe by ($D_h=4P/A$)
P	square pipe perimeter, m
A	area of the square pipe, m^2
u	velocity of the fluid, ms^{-1}
p	dimensional pressure, Nm^{-2}
Re	Reynolds number, $Re = u_i D_h / \nu$.
x, y	Cartesian coordinates, m
g	gravitational acceleration, ms^{-2}
G_k	turbulence kinetic energy generation due to the mean velocity gradients

G_b	turbulence kinetic energy generation due to buoyancy
Y_M	fluctuating dilatation rate
σ_k	Turbulent Prandtl numbers for k
σ_ε	Turbulent Prandtl numbers for ε_0
S_m	Mass added to the continuous phase from the dispersed second phase
L	length of the square pipe, m
S_k, S_ε	source terms
ΔP	Pressure drop across the obstacle, pa
I	turbulent intensity
$\dot{m}_{nf}$	mass flow rate of the nanofluid, kg/sec
$C_{p,nf}$	specific heat of the nanofluid, J/kg.K
T_{bo}, T_{bin}	inlet and out let temperature of the nanofluid, K
L	channel length, m
ΔP	pressure drop between the entering get and the exit, Pa
u_i	velocity in x-direction, m/sec
u'	fluctuated velocity, m/sec
σ_k and σ_ε	effective Prandtl numbers for turbulent kinetic energy and rate of dissipation.
$C_{1\varepsilon}$ and $C_{2\varepsilon}$	constants,

Greek Letters

μ	fluid dynamic viscosity, Pa.sec
ν	fluid kinematic viscosity, m ² /sec
ρ	density of the air, kg/m ³
ε	dissipation rate, m ² /sec
k	kinetic energy of turbulence, m ² /sec
ϕ	Particle Volume Fraction

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