

## Numerical Study of Turbulent Flows and Convective Heat Transfer of Al<sub>2</sub>O<sub>3</sub>-Water Nanofluids In A Circular Tube

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### ABSTRACT

The convective heat transfer of Al<sub>2</sub>O<sub>3</sub>-water nanofluids through a circular tube with a constant heat flux boundary condition is studied numerically. Turbulent flow conditions are considered with a Reynolds number ranging from 3500 to 20000. The numerical method used is based on the single-phase model. Four volume concentrations of Al<sub>2</sub>O<sub>3</sub>-water nanoparticles (0.1, 0.5, 1, and 2%) are used with a diameter of nanoparticle of 40 nm. A considerable increase in Nusselt number, axial velocity, and turbulent kinetic energy was found with increasing Reynolds number and volume fractions. However, the pressure losses were also increased with the rise of *Re* and nanoparticles concentration.

## 1. Introduction

Several investigations have been carried out to enhanced heat transfer characteristics of conventional heat transfer fluids by many researchers investigate the thermal properties of micro-sized particles dispersed fluids. In 1995, the first nanoparticle dispersed fluid was prepared by Choi [1]. In recent decades, the nanofluids have attracted a great interest owing to their greatly enhanced thermal properties. Some researchers interested in the study of the forced convective heat transfer in circular tubes, using both single-phase and multi-phase models [2].

Kim *et al.*, [3] examined by experiments the effect of nanofluids on the heat transfer inside a circular tube below the turbulent flow regime. They used the alumina oxide nanoparticles. They observed that the introduction of Al<sub>2</sub>O<sub>3</sub> nanoparticles in the base fluids yields an enhancement in the

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heat transfer rates. They obtained a maximum enhancement of 20% at the volume concentration of 3%.

Sundar and Sharma [4] investigated by experiments the turbulent flow through a uniformly heated pipe. Pouranfard *et al.*, [5] experimentally explored the drag reduction with nanofluids in a horizontal tube below the turbulent flow regime. The drag reduction in the rough tubes was more than that in the smooth tubes at the same conditions of flow. Pak and Cho [6] examined experimentally the heat transfer behavior of nanofluids under the turbulent flow regime in a circular pipe. They planted the following correlation depending only on Reynolds number ( $Re$ ) and Prandtl number ( $Pr$ ).

$$\overline{Nu} = 0.021Re^{0.8}Pr^{0.5} \quad (1)$$

Bianco *et al.*, [7-9] studied numerically the convective heat transfer of nanofluids inside a circular tube under turbulent flow conditions. They mentioned that the accuracy of models, the two-phase mixture and single-phase model could be improved by more precise nanofluid physical properties. Sekrani *et al.*, [10] investigated numerically the turbulent convective heat transfer of nanofluids in a circular tube subjected to a regular wall heat flux using different turbulence models.

The experimental study performed by Ahmad *et al.*, [11] for  $Al_2O_3$ /water and  $SiO_2$ /water nanofluids flowing through a circular tube under laminar conditions showed that the addition of small amounts of nano-sized  $Al_2O_3$  particles to the base fluid increases significantly the heat transfer rates. However, an adverse effect of silica nanofluids was observed, compared to the alumina nanofluids.

For laminar flow conditions, Heris *et al.*, [12] studied the thermal properties of turbine oil-based nanofluids through a circular tube. They prepared the nanofluids by dispersing  $Al_2O_3$ ,  $TiO_2$  and  $CuO$  nanoparticles in turbine oil with a volume fraction lower than 1%. For a circular tube, Lee *et al.*, [13] interested in the laminar flow of water-based nanofluids containing commercial Carbon nanotubes (CNTs). The surface area, the temperature difference between the inlet and outlet sections as well as the pumping power were used as criteria to determine the thermal performance for such systems. Based on the values of the figure of merit (FOM), they determined the most efficient water-based CNT nanofluids in terms of thermal performance. Manay and Mandev [14] studied experimentally the mixed convective heat transfer in a circular microchannel and used water and water based  $SiO_2$  nanofluids with 0.2 and 0.4% of volume concentrations. The overall heat transfer has been increased by 12-14% and 29-32% for the volume fractions of 0.2 and 0.4%, respectively.

Naik *et al.*, [15] used water/propylene glycol based  $CuO$  nanofluids as a working medium to explore the characteristics of transient flow and convective heat transfer in a circular tube equipped with and without helical elements. They studied the effect of nanoparticle concentrations having an average diameter of less than 50 nm. With a concentration of 0.5%, the Nusselt number was higher by about 28% than that for a plain tube and it was increased by 5.4 times over the base fluid when helical elements with a twist ratio of 3 were inserted. Compared to the base fluid, increases of the friction factor by 10% and 140% for the tube without and with helical elements, respectively, were reached. However, these percentages of the friction factor were estimated to be very low compared to the heat transfer enhancement.

Dabiri *et al.*, [16] tested the influence of  $SiC$ /water and  $MgO$ /water nanofluids on hydrothermal properties in a circular tube with constant heat flux. The increase of volume fraction of nanofluids yielded an increase in their viscosity, density, and thermal conductivity. For turbulent flow conditions, lyahraja *et al.*, [17] addressed by experiments the forced convective heat transfer of silver-water nanofluids inside a circular tube with a uniform heat flux condition. A significant enhancement in thermal performance was obtained even with the addition of low concentrations of silver

nanoparticles. For a concentration of 0.1% and at  $Re = 2.1 \times 10^4$ , Nusselt number increased up to 32.6%.

Sasmal [18] used the two-phase Buongiorno's model to study the flow of water based  $Al_2O_3$  nanofluid in an inclined elliptic cylinder. They suggested an analytical correlation for predicting the Nusselt number for such fluids. Hassan *et al.*, [19] tested the effect of volume concentrations (1, 2.5, and 4.5%) of the ethylene glycol (EG) base cuprous oxide ( $Cu_2O$ ) nanofluids on the convective heat transfer. Compared with the base fluid, they found a thermal enhancement by about 74% at 4.5% of nanoparticle concentration and  $Re = 116$ . Manay and Mandev [20] studied experimentally the mixed convective heat transfer in a heat pipe heat exchanger. They used water and water based  $SiO_2$  nanofluids with 0.2 and 0.4% of volume concentrations. The overall heat transfer has been increased by 12-14% and 29-32% for the volume fractions of 0.2 and 0.4%, respectively.

Arunachalam *et al.*, [21] explored experimentally the hydrothermal characteristics of Alumina-water nanofluids and Cu-alumina-water hybrid nanofluids flowing in a straight tube with and without V-cut twisted tape inserts. The  $Al_2O_3$ -Cu/water hybrid nanofluid with a concentration of 0.01% has yielded a higher friction factor than those of the  $Al_2O_3$ /water nanofluid and base fluid. Dabiri *et al.*, [22] tested the influence of SiC/water and MgO/water nanofluids on hydrothermal properties in a heat pipe heat exchanger with constant heat flux. The increase of volume fraction of nanofluids yielded a rise in their viscosity, density, and thermal conductivity.

For a mixture of water/ethylene glycol, Monfared *et al.*, [23] explored the effect of boehmite alumina nanoparticle shapes (spherical, cylindrical, platelet, blade and brick shapes) on the entropy generation in a double-pipe heat exchanger. The spherical shape of nanoparticle yielded the lowest frictional entropy generation rate and maximum total entropy generation rates. However, the platelet shape provided the highest frictional entropy generation rate and minimum thermal and total entropy generation rates. Menni *et al.*, [24] inspected the hydrodynamic and thermal behavior of water, ethylene glycol and water-ethylene glycol as base fluids dispersed by aluminum oxide nano-sized solid particles. Boertz *et al.*, [25] used the single-phase approach to model the flow of  $SiO_2$ /water- Ethylene glycol nanofluids through a heat pipe heat exchanger under a constant heat flux. The mass ratio of base fluid was 60:40 EG/W, and the nanoparticle concentrations were changed from zero to 10%. For the same heat transfer rate, the increase of Reynolds numbers and particle concentrations yielded further pumping power with the use of nanofluids.

Under turbulent flow conditions, Ny *et al.*, [26] used silver/ graphene (Ag/Heg) nanofluid to enhance the forced convective heat transfer in pipes. When increasing the Reynolds number from  $10^5$  to  $2 \times 10^5$ , the highest enhancement in Nusselt number by about 17.97% was reached at 0.1% volume concentration. Zainal *et al.*, [27] used various volume concentrations of Ag/Heg water nanofluids to enhance the heat transfer in a circular pipe. Their results revealed a decrease in the Nusselt number and heat transfer coefficient with the augmentation of volume fraction. Other interesting works may be found in the literature [28-34].

The purpose of this work is to add a more contribution to nanofluids turbulent convection in circular tubes. We have employed the "single-phase fluid" to achieve all investigations. The  $Al_2O_3$ -water mixture with a diameter of 40 nm of the alumina nanoparticles and low volume fractions (less than 2%) of the solid particles is used. The finite volume mode is employed to resolve the governing equations of the problem. The results acquired by the model are accomplished in terms of friction, velocity distributions, pressure drop, and Nusselt number profiles. Moreover, a comparison with theoretical and experimental data offered in the literature is made.

## 2. Mathematical Modeling

In the present study, for the fluid flow and heat transfer in the "single-phase fluid" with constant physical properties which the fluid flow is Newtonian and incompressible, the steady-state governing equations can be expressed as follows

Continuity equation

$$\nabla \cdot (\rho \vec{v}) = 0 \quad (2)$$

Momentum equation

$$\nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \cdot \left[ \mu (\nabla \vec{v} + \nabla \vec{v}^T) - \frac{2}{3} \nabla \cdot \vec{v} I \right] + \rho \vec{g} \quad (3)$$

Energy equation

$$\nabla \cdot (\rho \vec{v} C_p T_{nf}) = \nabla \cdot (k \nabla T_{nf}) \quad (4)$$

To renormalize the Navier-Stokes and the generation of the turbulence kinetic energy due to the velocity gradients, the  $k-\varepsilon$  turbulent model with enhanced wall treatment is used to simulate the turbulent flows. Other authors have used this model and they obtained good results [35].

### 3. Thermal and Physical Properties of Nanofluids

By supposing that the particles are well dispersed inside the base-fluid, the particle fraction may be considered as uniform all over the computational domain. For the single-phase model, the following formulas were used to calculate the physical and thermal properties of the considered nanofluid. The indices "p", "bf", "nf" assign, in this order, to the particles, the base-fluid and the nanofluid, at the same time as "r" refers to the "nanofluid base fluid" ratio of the physical quantity under consideration.

The density and the specific heat [35]

$$\rho_{nf} = (1 - \varphi) \rho_{bf} + \varphi \rho_p \quad (5)$$

$$(C_p)_{nf} = (1 - \varphi) (C_p)_{bf} + \varphi (C_p)_p \quad (6)$$

In order to evaluate the nanofluid dynamic viscosity, based on experimental data available, Maiga *et al.*, [20] leading to the following equation of the dynamic viscosity

$$\mu_{nf} = (123\varphi^2 + 4.3\varphi + 1) \mu_{bf} \quad (7)$$

For the thermal conductivity, the same criteria as that used for the dynamic viscosity is considered, which conduct to the following formula, set in [36]

$$k_{nf} = (4.97\varphi^2 + 2.72\varphi + 1) k_{bf} \quad (8)$$

### 4. Boundary Conditions

On the tube wall, the usual non-slip conditions are imposed with a uniform heat flux condition,  $q'' = 11 \text{ kW/m}^2$ . Enhanced wall treatment is used in the present investigation. At the pipe inlet, the inlet temperature of fluid is fixed to 293 K and the hydraulic diameter ( $D_H$ ) is set to 0.02. For the turbulent case, the turbulent intensity ( $I$ ) is determined by  $I = 0.16 \cdot Re^{-1/8}$ . At the pipe outlet, a static gauge pressure  $p_{\text{gauge}} = 0$  is specified.

## 5. Numerical Method and Validation

The computational fluid dynamic software Ansys-Fluent was used to resolve the present problem. The results of the Nusselt number acquired by the numerical simulations are compared with different well-known and strong experimental data and correlations, widely used in the literature, especially Pak and Cho [6], Dittus-Boelter [37], and Gnielinski [38].

Dittus–Boelter correlation [37]

$$Nu = 0.023Re^{0.8}Pr^{0.4} \quad \text{for } Re > 10000 \quad \text{and } 0.7 \leq Pr \leq 160 \quad (9)$$

Gnielinski correlation [38]

$$Nu = \frac{\frac{f}{8}(Re-1000).Pr}{1+12.7\sqrt{\frac{f}{8}}(Pr^{2/3}-1)} \quad \text{for } 3000 \leq Re \leq 5 \times 10^6 \quad (10)$$

The computer tool Gambit was used to create and mesh the computational domain. An increased mesh density was used near the pipe walls to capture the details of the flow boundary layer. Mesh tests were performed (Table 1) by checking that additional cells did not change the  $Nu$  values by more than 2.5%.

To verify the grid independency, the number of cells was increased by a factor of about 2. The original 3D mesh of the model had 45,123 computational cells. The number of cells was increased from 45,123 cells to 90,246 cells. The additional cells changed the  $Nu$  values by more than 3%. Thus, the number of cells was changed from 90,246 cells to 180,492 cells. The additional cells did not change the  $Nu$  values by more than 2.5%. Therefore, 90,246 cells were employed for the next computations.

**Table 1**

Details on mesh tests for  $\varphi = 2\%$  and  $Re = 20,000$

	M1	M2	M3
Number of cells	45,123	90,246	180,492
$Nu$	173.51	176.13	176.15
Time required [second]	8512	16845	25128

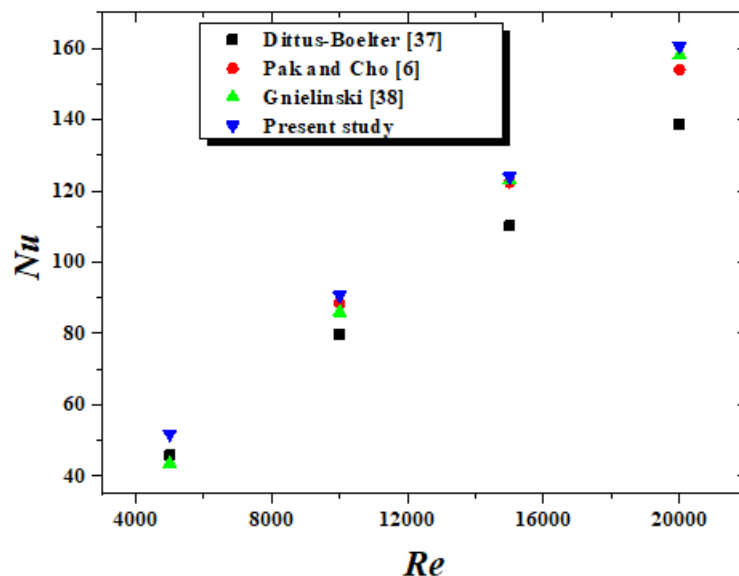
## 6. Results and Discussion

The numerical simulations have been carried out for  $\text{Al}_2\text{O}_3$ -water mixture and various particle concentrations  $\varphi$  varying from 0 to 2%. The considered geometrical configuration consists of a tube has a diameter of 0.02 m and a length of 1.5 m. The flow Reynolds number has varied from  $3.5 \times 10^3$  to  $2 \times 10^5$ .

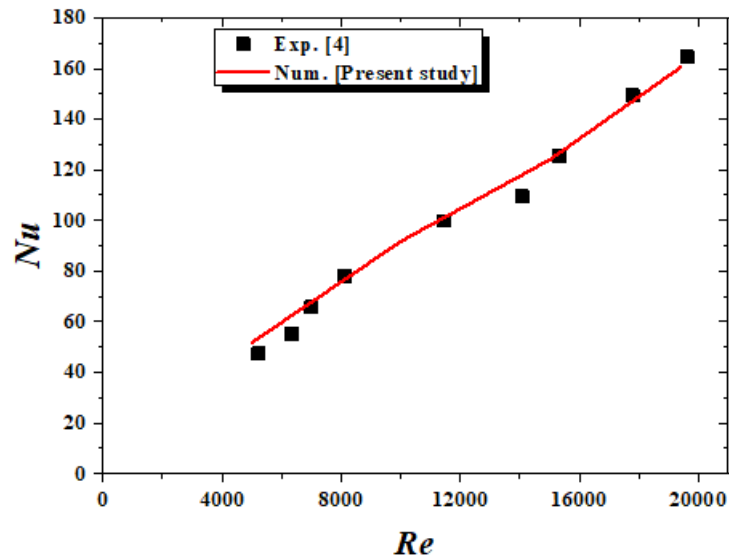
In the first part of our study, we focused on the validation of our predicted results against experimental data. Figure 1 shows the values of Nusselt number vs. Reynolds number for  $\varphi = 0.1\%$ . In this figure, a comparison is made between the results of this investigation and the correlation of Pak and Cho [6], the correlation of Gnielinski [38], and the Dittus-Boelter's equation [37]. As observed, a good agreement between our simulation and both correlations of Pak-Cho and Gnielinski. For high Reynolds numbers, a large difference is observed with the Dittus-Boelter's equation. As the correlation of Dittus works well for the base fluid, its role is of a lower limit for the measured nanofluid results.

In Figure 2, the results of this study display a similar behavior of Nusselt with a volume concentration of  $\text{Al}_2\text{O}_3$ -water  $\varphi = 0.1\%$ . Compared to the experimental data of Sundar and Sharma [4], a good agreement is observed.

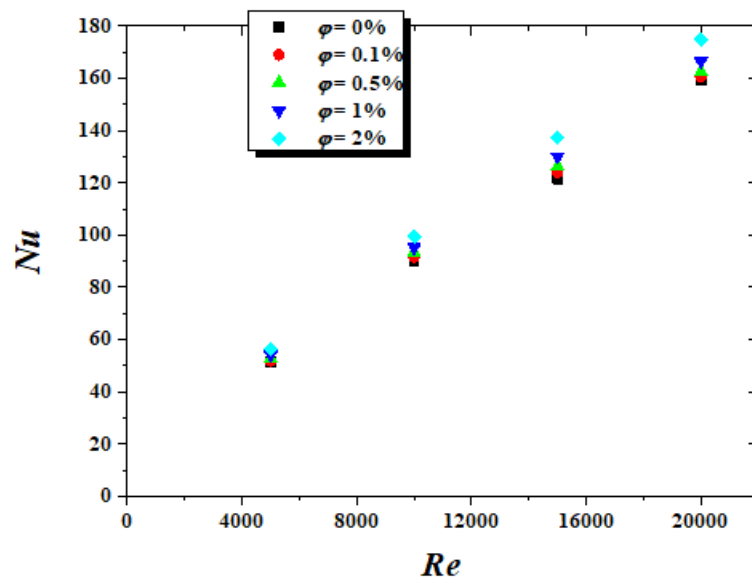
Figure 3 shows the Nusselt number distribution for  $\varphi = 0, 0.1, 0.5, 1, \text{ and } 2\%$ . For all Reynolds numbers and all concentrations of nanoparticles, the Nusselt number increases with increasing values of  $Re$ . It is also seen that the average Nusselt number of nanoparticles is significantly higher than that for pure water ( $\varphi = 0\%$ ). In addition, a good enhancement in  $Nu$  is observed with increasing volume fractions of nanoparticles.



**Fig. 1.** Comparison of the numerical results of  $\text{Al}_2\text{O}_3$ -water and the proposed correlations



**Fig. 2.** Variation of Nusselt number vs. Reynolds number for  $Al_2O_3$ -water nanofluid, at  $\varphi = 0.1\%$  (validation)



**Fig. 3.** Effect of nanoparticles volume fraction of  $Al_2O_3$ -water on the Nusselt number at different Reynolds numbers

Figure 4 illustrates the variations of the pressure drop vs. Reynolds number for different volume concentrations. The pressure loss of the nanofluid is higher than that of pure water. The rise of nanofluid volume concentration increases the pressure drop. It illustrates that with the further increase of the Reynolds number (higher than 12,000), the rising volume concentration has a clear effect on the pressure drop.

At  $Re = 10^4$ , Figure 5(a) and 5(b) present the variation of the dimensionless mean velocity along the pipe length and pipe radius, respectively. In addition, in Figure 6, the values of dimensionless turbulent kinetic energy for four different volume fractions are presented. It is clearly illustrated that the interaction of fluid particles is intensified with increasing volume concentrations of nanoparticles, giving thus further kinetic energy.

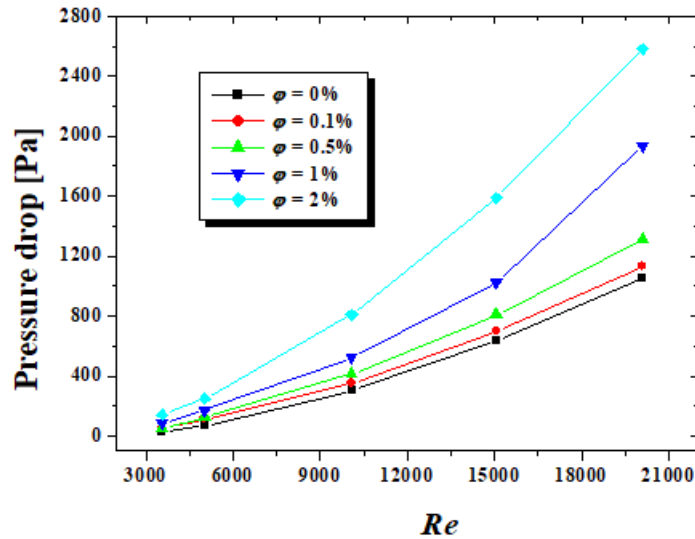
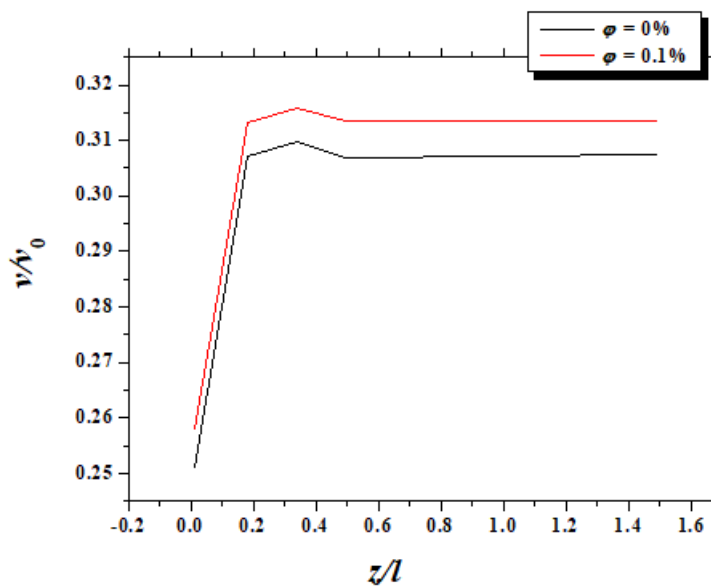
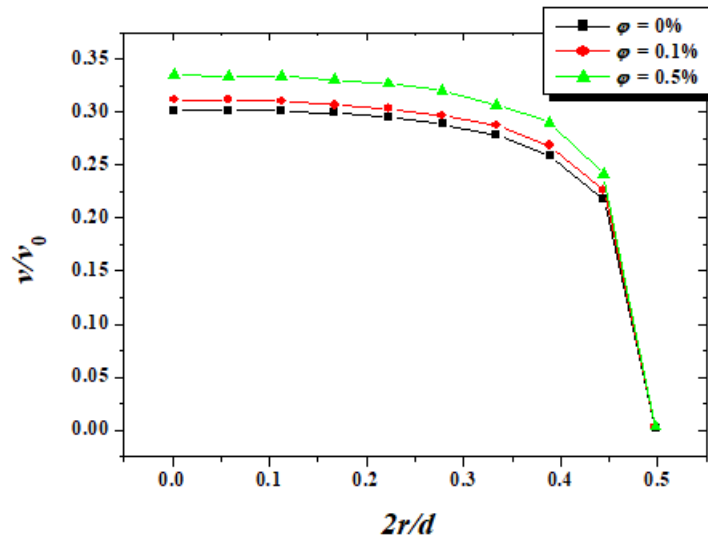


Fig. 4. The variations of the pressure drop vs. Reynolds number for different volume concentrations



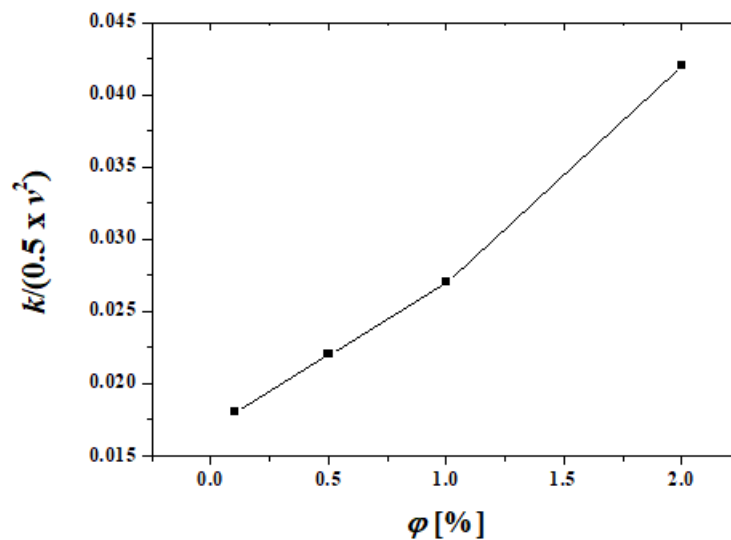
(a)





(b)

**Fig. 5.** Dimensionless mean velocity, (a) comparison for pure water and  $\text{Al}_2\text{O}_3$ -water with  $\phi = 0.1\%$ , (b) radial profiles of the mean velocity



**Fig. 6.** Variation of the dimensionless turbulence kinetic energy with different volume fractions, at  $Re = 10^4$ ,  $Z/L = 0.5$

## 7. Conclusions

The hydrothermal characteristics of  $\text{Al}_2\text{O}_3$ -water nanofluid through a circular tube has been numerically studied. Under turbulent flow conditions, four values of volume fractions were considered, namely:  $\phi = 0.1, 0.5, 1, \text{ and } 2\%$ . The single-phase approach was used to perform the investigations. The comparison of numerically predicted Nusselt numbers over a wide range of turbulent Reynolds numbers, with corresponding experimental results and literature correlations, revealed a satisfactory agreement.

The obtained results have revealed an enhancement in the heat transfer rates, with respect to the base fluid. The enhancement has been more significant with the increase of the particle volume fraction, with relatively a higher thermal conductivity.

Moreover, a more noticeable increase in Nusselt number with the particle volume fraction was obvious at high Reynolds numbers. Increasing Reynolds number for all volume fractions caused an increase in the pressure drop. The velocity and turbulent kinetic energy of nanofluids often possess upper values than that of the base fluid.

As well-known, the insertion of baffles within heat exchanger pipes creates further interaction between the fluid particles, which provides a significant enhancement in heat transfer rates. However, pressure losses will be also increased. From this viewpoint and for future works, the combination of the baffling technique with nanofluids seems to be an efficient method to optimize the overall performances of heat exchangers. The lower heat transfer areas that are formed behind the baffles and the pressure losses may be reduced by using perforations in baffles.

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