# HEAT TRANSFER ENHANCEMENT WITH ELLIPTICAL TUBE UNDER TURBULENT FLOW $TiO_2$-WATER NANOFLUID

by

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*Heat transfer and friction characteristics were numerically investigated, employing elliptical tube to increase the heat transfer rate with a minimum increase of pressure drop. The flow rate of the tube was in a range of Reynolds number between 10000 and 100000. The FLUENT software is used to solve the governing equation of CFD (continuity, momentum, and energy) by means of a finite volume method. The electrical heater is connected around the elliptical tube to apply uniform heat flux (3000 W/m$^2$) as a boundary condition. Four different volume concentrations in the range of 0.25% to 1% and different $TiO_2$ nanoparticle diameters in the range of 27 nm to 50 nm, dispersed in water are utilized. The CFD numerical results indicate that the elliptical tube can enhance heat transfer and friction factor by approximately 9% and 6% than the circular tube, respectively. The results show that the Nusselt number and friction factor increase with decreasing diameters but increasing volume concentrations of nanoparticles.*

Key words: *nanofluid, CFD, elliptical tube, friction factor, FLUENT*

## Introduction

Very small particles suspension in saturated liquids (water, ethylene glycol, engine oil) is defined as nanofluids may constitute a very interesting alternative for advanced thermal applications. It has been found that important heat transfer enhancement may be achieved while using nanofluids compared to the use of conventional fluids. Furthermore, some oxide nanoparticles exhibit an excellent dispersion properties in traditional cooling liquids [1].

Turbulent forced convection heat transfer of water and nanofluid inside tube under single phase approach was carried out by [2-6]. Pak and Cho [7] were investigated the effect of two different metallic oxide particles, $TiO_2$, and $Al_2O_3$, with mean diameters of 27 nm and 13 nm, respectively, on base fluid (water) experimentally. Results showed that significant heat transfer enhancement with nanofluids as compared with the base fluid. A theoretical model proposed by Sharma *et al.* [8] to predict friction and heat transfer coefficients for different nanofluids containing Cu, CuO, $TiO_2$, SiC, $ZrO_2$, and $Al_2O_3$ nanoparticles of different sizes, concentration and temperatures dispersed in water. Results showed deviation of theoretical with

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experimental data. The CFD modeling of heat transfer enhancement of $Al_2O_3$ nanofluid using low volume fraction under turbulent pipe flow with constant wall temperature has been studied by Kumar [9]. Nusselt number and friction factor have been predicted for the low volume fractions (*i. e.* 0.02%, 0.1% and 0.5%). Numerical study of turbulent forced convection flow of $Al_2O_3$ nanofluid through circular tube was subjected by Bianco [10]. Results showed that heat transfer increased with the particle volume concentration and Reynolds number. Experimental study of turbulent forced convection heat transfer through pipe employing twisted tape with and without $Al_2O_3$-water nanofluid was conducted by Sundar and Sharma [11], generalized correlation has been developed for the estimation of Nusselt number and friction factor in a pipe with and without inserts.

Wide variety of practical and industrial applications of forced convection in an elliptical tube between two horizontal cylinders such as heat exchangers, heating processes, power generation, chemical processes, microelectronics, and cooling processes have been led to the interest this type of studies [12-16].

Duangthongsuk and Wongwises [17] observed the heat transfer coefficient and friction factor of the nanofluid $TiO_2$-water flowing in a horizontal double tube counter flow heat exchanger under turbulent flow conditions, experimentally and showed that the heat transfer coefficient of nanofluid is higher than that of the base liquid. Forced convection turbulent flow of $Al_2O_3$-water nanofluid inside an annular tube with variable wall temperature has been investigated experimentally by Prajapati [18]. Results showed the enhancement of heat transfer due to the nanoparticle presence in the fluid. The forced convection flow between two corrugated cylinders has been studied by Kittur [19]. Results found friction factor and heat transfer on the boundaries. Horizontal double-tube heat exchanger counter turbulent flow was studied numerically by Bozorgan *et al*. [20], $Al_2O_3$-water nanofluid with particle size of 7 nm with volume concentrations up to 2% are selected as a coolant used, results showed that the pressure drop of nanofluid is slightly higher than water and increases with the increase of volume concentrations. Forced convection flow of nanofluids of $TiO_2$-water in a double-tube counter flow heat exchanger using CFD simulation FLUENT software has been investigated by Demir *et al*. [21]. A double tube coaxial heat exchanger heated by solar energy using $Al_2O_3$ nanofluid presented experimentally and numerically by Luciu *et al.* [22], results showed that nanofluids have a higher performance of heat transfer than the base fluid.

Practically, in compact heat exchangers may be necessary using non-circular flow-duct, to increase the heat transfer with a minimum increase of pressure drop Heris *et al.* [23]. The non-circular flow duct has been used in the study undertaken to achieve to this application with minimum pressure drop.

In this work, CFD simulation by FLUENT software is used to predict friction factor and Nusselt number, with turbulent forced convection of $TiO_2$-water nanofluid through elliptical tube under constant heat flux, then compared results of CFD with experimental data available in the literature.

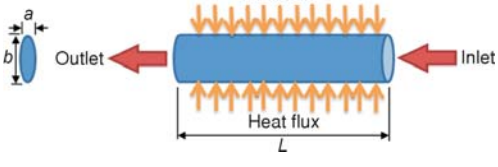


**Figure 1. Schematic of problem undertaken**

## Theoretical analysis

### *Physical model*

Cylindrical geometry co-ordinate of problem undertaken has been shown in fig. 1, Dimensions of elliptical tube are major and minor diameter ($a$ = 27 mm and $b$ = 9 mm), the length

($L$), and hydraulic diameter ($D_h$) of the elliptical tube are 1000 mm and 24 mm, respectively. Reynolds number has been calculated regarded hydraulic diameter ($D_h$):

$$D_h = 4\frac{\text{Area}}{\text{Perimeter}} \tag{1}$$

where $D_h$ is the hydraulic diameter of elliptical tube, area ($A = (\pi \times a \times b)/4$) and Perimeter = = $\pi[(a^2 + b^2)/2]^{1/2}$. Reynolds number is:

$$\text{Re} = \frac{\rho_{nf} D_h u}{\mu_{nf}} \tag{2}$$

The problem under taken is assumed to be 2-D, steady, incompressible and Newtonian turbulent fluid flow, constant thermo physical properties of nanofluid, no effect of gravity and heat conduction in the axial direction and wall thickness of tubes are neglected.

*Governing equations*

Infinitesimal (less than 100 nm) solid particles are considered to be able using single phase approach, so single phase approach is adopted for nanofluid modeling. The thermal properties of nanofluid is estimated by equations, [8]:

$$\rho_{nf} = \left(\frac{\phi}{100}\right)\rho_p + \left(1 - \frac{\phi}{100}\right)\rho_f \tag{3}$$

$$C_{nf} = \frac{\frac{\phi}{100}(\rho C)_p + \left(1 - \frac{\phi}{100}\right)(\rho C)_f}{\rho_{nf}} \tag{4}$$

$$k_r = \frac{k_{nf}}{k_f} = \left\{0.8938\left(1 + \frac{\phi}{100}\right)^{1.37}\left(1 + \frac{T_{nf}}{70}\right)^{0.2777}\left(1 + \frac{d_p}{150}\right)^{-0.0336}\left(\frac{\alpha_p}{\alpha_f}\right)^{0.01737}\right. \tag{5}$$

$$\mu_r = \frac{\mu_{nf}}{\mu_f} = \left(1 + \frac{\phi}{100}\right)^{11.3}\left(1 + \frac{T_{nf}}{70}\right)^{-0.038}\left(1 + \frac{d_p}{170}\right)^{-0.061} \tag{6}$$

where $\rho$, $C$, $k$, and $\mu$ are density, specific heat capacity, thermal conductivity, and viscosity, respectively, and subscripts nf, f, and p are represented nanofluid, fluid, and solid properties, respectively. On the other hand, thermal properties of solid particles are: $k_p$ = 8.4 W/m°C, $\rho_p$= 4175 kg/m$^3$, and $C_p$ = 692 J/kgK, [8].

Equations (3)-(6) are valid for $TiO_2$ nanoparticles suspended in water with base temperature 25 °C and size diameter is 20-50 nm. For all these assumptions the dimensional conservation equations for steady-state mean conditions are: continuity, momentum, and energy equations [24]:

$$\frac{\partial u}{\partial x} + \frac{1}{r}\frac{\partial}{\partial r}(r\rho_{nf} u) = 0 \tag{7}$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial}{\partial r}(\rho_{nf} u) = -\frac{\partial P}{\partial x} + \frac{1}{r}\frac{\partial}{\partial r}\left[r(\upsilon + \varepsilon_H)\frac{\partial u}{\partial r}\right] \tag{8}$$

$$\frac{1}{r}\frac{\partial}{\partial r}(\rho u T) = \frac{1}{r}\frac{\partial}{\partial r}\left[r(\alpha_{nf} + \varepsilon_H)\frac{\partial T}{\partial r}\right] + \frac{1}{r^2}\frac{\partial}{\partial x}\left[\frac{k_{nf}}{C_p}\frac{\partial T}{\partial x}\right] \tag{9}$$

High Reynolds number was taken as input parameter, pressure treatment was adopted SIMPLE scheme and turbulent viscous $k$-$\varepsilon$ model has been employed, converged solutions were considered for residuals lower than $10^{-6}$ for all the governing equations. The results of simulation for nanofluid were compared with Blasius eq. (10) for friction factor and Dittus-Boelter eq. (11) for Nusselt number:

$$f = \frac{0.316}{\mathrm{Re}^{0.25}} \tag{10}$$

$$\mathrm{Nu} = \frac{h_\mathrm{f}}{k_\mathrm{f}} D_\mathrm{eff} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \tag{11}$$

### *Boundary conditions*

Volume concentration nanofluids (0.25, 0.5, 0.75, and 1%) at 25 °C base temperature were used for $TiO_2$-water as input fluids. For comparison purposes, water was also employed as working fluid. The CFD studies were carried out with uniform velocity profile at the inlet and pressure outlet condition has been used at the outlet of elliptical tube. Turbulent intensity ($I$), was specified for an initial guess of turbulent quantities ($k$ and $\varepsilon$). The turbulent intensity was estimated for each case based on the formula $I = 0.16\mathrm{Re}^{-1/8}$. The walls of the tube were assumed to be perfectly smooth. The constant heat flux condition has been specified on the inside tube wall with a value of 3000 W/m$^2$. Reynolds number was varied from $10^4$ to $10^5$ at each step of iterations as input data. The friction factor and Nusselt number were introduced as output data.

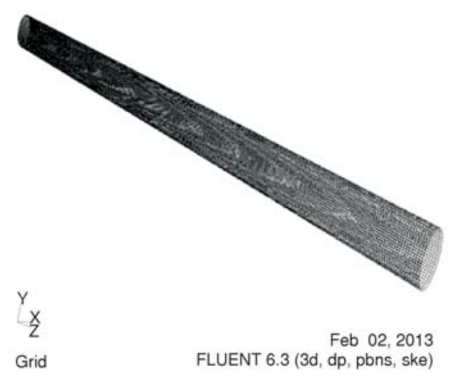


**Figure 2. Mesh generated by GAMBIT**

### *Grid independence test*

Grids independence have been chosen in GAMBIT software for elliptical tube narrow as 50000 cells and 1000 × 50, subdivisions in the axial, and radial directions, respectively. To find the most suitable size of mesh faces, grid independent test was performed for the physical model. In this study, rectangular cells were used to mesh the surfaces of tube wall but triangular cells were used to mesh the surfaces of gap as shown in fig. 2. The grid independence has been checked by using different grid systems and four mesh faces considered 50000, 40000, 30000, and 20000 for pure water. Friction fac-

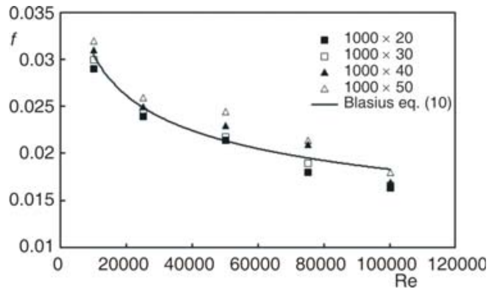


**Figure 3. Optimum mesh grid size for friction factor with Reynolds number at 25 °C**

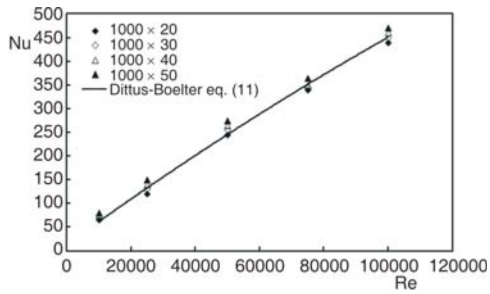


**Figure 4. Optimum mesh grid size of Nusselt number with Reynolds number at 25 °C**

tor and Nusselt number was determined for all four mesh faces and results were proper. However, any number of mesh faces for these four cases can be used, but in this study, mesh faces with 50000 has been adopted as the best in terms of the accuracy as shown in figs. 3 and 4.

## The CFD simulation

The CFD simulations were used FLUENT software with solver strategy. To analyze problems Gambit software has been used. To make possible numerical solution of governing equations, single phase conservation equations were solved by control volume approach then converting them to a set of algebraic equations. Simulation results were tested by comparing the predicted results of Pak and Cho [7] who used circular heated tube in experimental work. The FLUENT software was used in CFD analysis in the literature and a detailed description of the mathematical model can be found in the fluent user's guide [25]. The CFD modeling region could be classified into few major step: preprocess stage, the geometry of problem undertaken was constructed as elliptical narrow and computational mesh was generated in GAMBIT. It followed by the physical model, boundary conditions and other parameters appropriate were defined in models set-up and solving stage. All scalar values and velocity components of the problem are calculated at the center of control volume interfaces where the grid schemes are used intensively. Throughout the iterative process accurately monitor of the residuals has been done. When the residuals for all governing equations were lower than $10^{-6}$, all solutions were assumed to be converged.

Finally, the results could be obtained when FLUENT iterations have been led to converged results defined by a set of converged criteria. The friction factor and Nusselt number inside the elliptical tube could be obtained throughout the computational domain in the post-process stage.

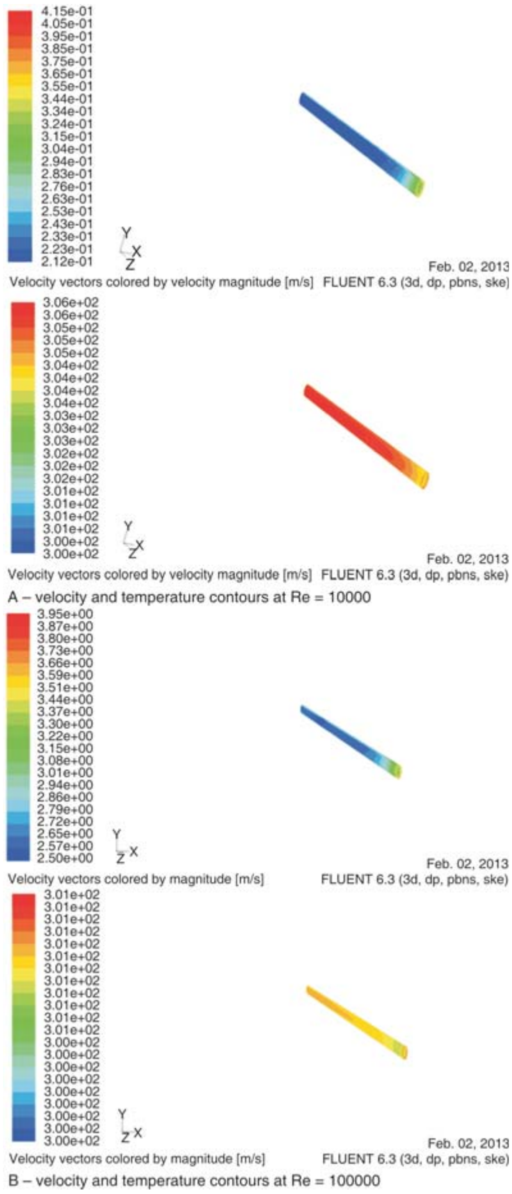


**Figure 5. Contour of velocity and temperature**

## Results and discussion

### *Velocity and temperature*

Velocity and temperature have been represented as contour in fig. 5. It shows that the difference between minimum and the maximum values of temperature is depended on velocity inlet, as the maximum value of velocity 0.415 m/s and 3.95 m/s have led to temperature deference 6 K and 1K, respectively. It seems that same behavior of velocity and temperature with different values depending on Reynolds number.

### *The effect of nanofluid volume concentrations*

This section includes the effects of different nanoparticle concentrations of $TiO_2$ nanofluid on the friction factor and Nusselt number. The effect of different volume concentration on friction factor is shown in fig. 6, at the range of (0-1%) nanoparticle volume concentration. The results are revealed that $TiO_2$ nanofluid with highest concentration of volume 1% has the highest friction factor at all Reynolds numbers. The reason of increasing of friction factor with the increasing of concentration of volume of nanofluid is the increasing of fluid viscosity which reduced the moving of fluid. Also, validation of friction factor results from simulation for nanofluid with Blasius equation for water is studied. On the other hand, fig. 7 shows that Nusselt number is increasing with increasing in volume concentrations of nanofluid because of the high thermal properties of $TiO_2$ which enhancement of heat transfer and hydrodynamic flow. The results are revealed that $TiO_2$ nanofluid with highest concentration of volume 1% has the highest Nusselt number at all Reynolds numbers. The irregular and random movements of particles increase the energy exchange rates in the fluid with penalty on the wall shear stress and consequently enhance the thermal dispersion of the flow. It is also found that the friction factor is decreased while Nusselt number is increased with the increase of Reynolds number. Also, validation of Nusselt number results from simulation for nanofluid with Dittus-Boelter equation for water is showed. Similarly, increasing in volume concentrations of nanofluid was led to increase in Nusselt number and heat transfer enhancement.

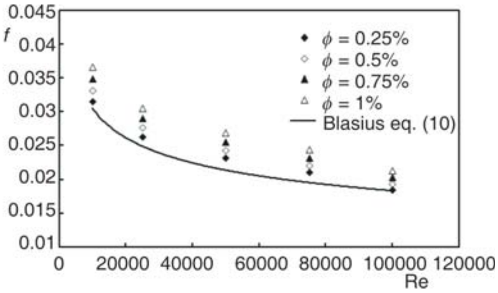


**Figure 6. Effect of volume concentration of nanofluid on fiction factor**

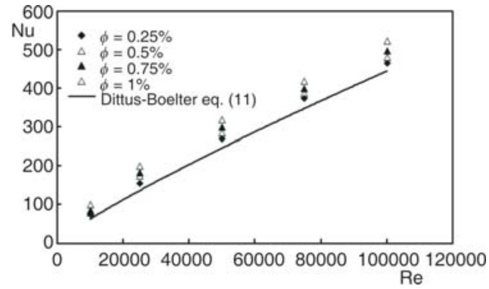


**Figure 7. Effect of nanofluid volume concentration on heat transfer enhancement**

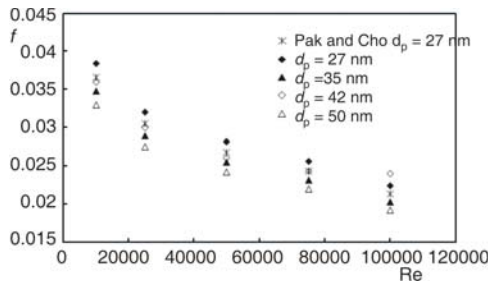


**Figure 8. Effect of particle size diameter on friction factor with Reynolds number**

### *The effect of nanofluid size diameter*

The effect of nanoparticle size diameters of $TiO_2$-water nanofluid on the friction factor and Nusselt number with different Reynolds numbers was examined in this section. The ranges of nanoparticle diameter are varied from 27 nm to 50 nm with 1% concentration of $TiO_2$ nanofluid. Figure 8 shows the effect of nanoparticle size diameters of $TiO_2$ on the friction factor with different Reynolds numbers. It appears that the friction factor has minor change when nano-particle diameters of $TiO_2$ nanofluid are varied. As shown in this figure the nanofluid with 27 nm nanoparticle size diameter has the highest friction factor, whereas, the nanoparticle with a diameter of 50 nm has the lowest friction factor. The 27 nm size $TiO_2$ particles give rise high friction factor over the same particles of 50 nm size related to two reasons which are the high specific surface area of the nanoparticles and the Brownian motion. As the particle size reduces, the surface area per unit volume increases, the heat transfer is being dependent on the surface area, and thus the effectiveness of nanoparticles in transferring heat to the base liquid increases. However, reducing the particle size means increasing the Brownian motion velocity, which again adds up to the contribution by the nanoparticles to the total heat transfer by continuously creating additional paths for the heat flow in the fluid.

Figure 9 revealed that Nusselt number has lower value with higher particle size diameter of $TiO_2$ nanofluid. May be the high specific surface area of the nanoparticle is the reason of the effect of size diameter, the heat transfer is being dependent on the surface area. Also figs. 8 and 9 are concluded that comparisons between CFD analysis and experimental data of Pak and Cho [7] for both the friction factor and Nusselt number of size diameter 27 nm of $TiO_2$ nanofluid and there is good agreement.

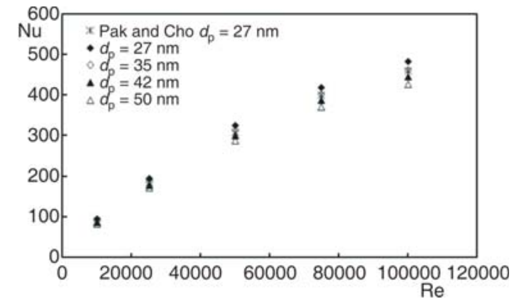


**Figure 9. Effect of particles size diameter on Nusselt number with Reynolds number**

### *Comparison CFD results with experimental data*

This section analyzes the comparison between CFD data and experimental data for $TiO_2$-water nanofluid with particle size is 27 nm and base temperature is 25 °C. Figure 10 shows the CFD analysis of friction factor *vs.* Reynolds number compared with Pak and Cho [7] experimental data. There is same behavior and good agreement with deviation 2%. On the other hand, Nusselt number *vs.* Reynolds number compared with Pak and Cho [7] experimental data as shown in fig. 11. There is good agreement with deviation 1.4%. Results show that elliptical tube has enhancement of heat transfer and hydrodynamic flow more than circular tube.

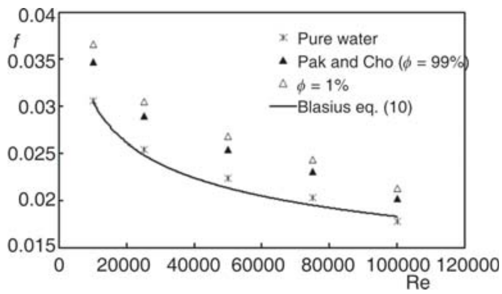


**Figure 10. Comparison CFD results of friction factor with Pak and Cho [7] experimental data**

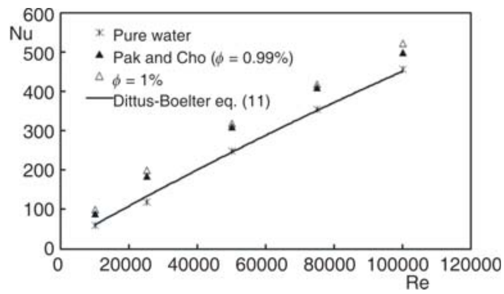


**Figure 11. Comparison CFD results of Nusselt number with Pak and Cho [7] experimental data**

## Conclusions

In this paper, forced convection heat transfer inside elliptical tube under turbulent flow by numerical simulation with uniform heat flux boundary condition around elliptical tube has been studied. The heat transfer enhancement resulting from various parameters such as nanoparticle concentration of volume, and Reynolds number is reported. The finite volume methods have been used to solve the governing equations with certain assumptions and appropriate boundary conditions. The Nusselt number, friction factor were obtained through the numerical simulation.

The following conclusions of the study are:

- the enhancement of friction factor and Nusselt number are 4% and 9% for enhanced tube than that of the circular tube at all Reynolds numbers,
- the concentration of volume (1%) of $TiO_2$ nanofluid has the highest Nusselt number and friction factor values, followed by (0.75, 0.5, and 0.25%). Finally, pure water has the lowest values of them, and
- there is good agreement between simulation results and experimental data of Pak and Cho [7] with deviation 2% and 1.4% for Nusselt number and friction factor, respectively.

## Acknowledgment

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## Nomenclatures

$C$ – specific heat capacity, [Jkg$^{-1}$K$^{-1}$]
$D$ – diameter, [m]
$E$ – energy, [W]
$f$ – friction factor
$h$ – heat transfer coefficient, [Wm$^{-2}$°C$^{-1}$]
$k$ – thermal conductivity, [Wm$^{-1}$°C]
Nu – Nusselt number, (= $hDk_{\mathrm{nf}}^{-1}$), [–]
$P$ – pressure, [Nm$^{-2}$]
Pr – Prandtl number, (= $C\mu k_{\mathrm{nf}}^{-1}$), [–]
Re – Reynolds number, (= $\rho_{\mathrm{nf}}D_{\mathrm{h}}u\mu_{\mathrm{nf}}^{-1}$), [–]
$u$ – velocity, [ms$^{-1}$]

*Greek symbols*

$\alpha$ – thermal difusivity, [m$^2$s$^{-1}$]
$\varepsilon$ – turbulent viscosity, [m$^2$s$^{-1}$]
$\mu$ – viscosity, [Nsm$^{-2}$]
$\rho$ – density, [kgm$^{-3}$]
$\tau$ – shear stress, [Nm$^{-2}$]
$\phi$ – volume concentration

*Subscripts*

f – liquid phases
h – hydraulic
nf – nanofluid
p – solid particle

## References

[1] Das, S. K., *et al.*, *Nanofluids Science and Technology*, John Wiley and Sons, Inc., New York, USA, 2007
[2] Moghari, R. M., Two Phase Mixed Convection $Al_2O_3$-Water Nanofluid Flow in an Elliptical Tube, *Int J. of Multiphase Flow, 37* (2011), 6 pp. 585-595
[3] Shariat, M., *et al.*, Numerical Study of Two Phase Laminar Mixed Convection Nanofluid in Elliptic Ducts, *Applied Thermal Engineering, 31* (2011), 14-15, pp. 2348-2359
[4] Puli,U., Rajvanshi, A. K., An Image Analysis Technique for Determination of Void Fraction in Sub Cooled Flow Boiling of Water in Horizontal Elliptical Tube at High Pressures, *Int. J. of Heat and Fluid Flow, 38* (2012), Dec., pp. 180-189
[5] Jawarneh, A. M., Heat Transfer Enhancement in Swirl Elliptical Tube Flows, *Proceedings*, 5th WSEAS Int. Conf. on Environment, Ecosystems and Development, Tenerife, Spain, 2007
[6] Heris, S. Z., Convective Heat Transfer of a Cu/Water Nanofluid Flowing through a Circular Tube, *Experimental Heat Transfer, 22* (2009), 4, pp. 217-227
[7] Pak, B. C., Cho, Y. I., Hydrodynamic and Heat Transfer Study of Dispersed Fluids with Submicron Metallic Oxide Particles, *Experimental Heat Transfer, 11* (1998), 2, pp. 151-171
[8] Sharma, K. V., *et al.*, Correlations to Predict Friction and Forced Convection Heat Transfer Coefficients of Water Based Nanofluids for Turbulent Flow in a Tube, *IJMNTFTP, 3* (2010), 4, pp. 1-25

[9] Kumar, P., A CFD Study of Heat Transfer Enhancement in Pipe Flow with $Al_2O_3$ Nanofluid, *Int. Jur. of Mechanical, Aerospace, Industrial, Mechatronic and Manufactoring Eng., 5* (2011), 9 pp. 1850-1854
[10] Bianco, V., Numerical Investigation on Nanofluids Turbulent Convection Heat Transfer Inside a Circular Tube, *Int. J. of Thermal Sciences, 50* (2011), 3, pp. 341-349
[11] Sundar L. S., Sharma, K. V., Turbulent Heat Transfer and Friction Factor of $Al_2O_3$ Nanofluid in Circular Tube with Twisted Tape Inserts, *Inter. J. Heat and Mass Transfer, 53* (2010), 7-8, pp. 1409-1416
[12] Durmus, A., *et al*., Investigation of Heat Transfer and Pressure Drop in a Concentric Heat Exchanger with Snail Entrance, *Applied Thermal Engineering, 22* (2002), 3, pp. 321-332
[13] Bergles, A. E., Heat Transfer Enhancement the Encouragement and Accommodation of High Heat Fluxes, *Transaction ASME, Journal Heat Transfer, 119* (1995), 1, pp. 8-19
[14] Yildiz, C., *et al.*, Effect of Twisted Strips on Heat Transfer and Pressure Drop in Heat Exchangers, *Energy Conversion and Management, 39* (1998), 3-4, pp. 331-336
[15] Manglik, R. M., Bergles, A. E., Heat Transfer and Pressure Drop Correlations for Twisted-Tape Inserts in Isothermal Tubes: Part II-Transition and Turbulent Flows, Enhanced Heat Transfer, *Transaction ASME, Journal Heat Transfer, 115* (1992), 4, pp. 890-896
[16] Sarma, P. K., *et al*., Laminar Convective Heat Transfer with Twisted Tape Inserts in a Tube, I*nt. J. of Thermal Sciences, 42* (2003), 9, pp. 821-828
[17] Duangthongsuk, W., Wongwises, S., An Experimental Study on the Heat Transfer Performance and Pressure Drop of $TiO_2$-Water Nanofluids Flowing under a Turbulent Flow Regime, *Int. J. of Heat and Mass Transfer, 53* (2010), 1-3, pp. 334-344
[18] Prajapati, O. S., Effect of $Al_2O_3$-Water Nanofluids in Convective Heat Transfer, *Int. J. of Nano science, 11* (2012), 3, p. 1240005
[19] Kittur, B. M. G., On the Forced Convective Flow between Two Corrugated Cylinders, *Adv. Theor. Appl. Mech., 3* (2010), 9, pp. 491-506
[20] Bozorgan,N., *et al*., Performance Evaluation of $AI_2O_3$/Water Nanofluid as Coolant in a Double-Tube Heat Exchanger Flowing under a Turbulent Flow Regime, *Hindawi Publishing Corporation Advances in Mechanical Engineering, 2012* (2012), ID 891382
[21] Demir, H., *et al*., Numerical Investigation on the Single Phase Forced Convection Heat Transfer Characteristics of $TiO_2$ Nanofluids in a Double-Tube Counter Flow Heat Exchanger, *Int. Comm. in Heat and Mass Transfer, 38* (2011), 2, pp. 218-228
[22] Luciu, R. S., *et al*., Nusselt Number and Convection Heat Transfer Coefficient for a Coaxial Heat Exchanger Using $Al_2O_3$-Water ph=5 Nanofluid, *Bulletin of the Polytechnic Institute of Jassy, LV(LIX)* (2009), 2, pp. 71-80
[23] Heris, S. Z., *et al*., CuO-Water Nanofluid Heat Transfer through Triangular Ducts, *Iranian Journal of Chemical Engineering, 9* (2012), 1, pp. 23-32
[24] Bejan, A., *Convection Heat Transfer*, 3rd edition, John Wiley & Sons, Inc., New York, USA, 2004
[25] ***, Fluent Incorporated, "Fluent 6.2 User Manual," 2006

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