Numerical and CFD analysis of a heat transfer enhancement in turbulent flow through a circular pipe using nanofluid

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Abstract. This paper presents numerical and CFD simulation of an Al₂O₃-water nanofluid turbulent flow in a circular pipe which has a 36 mm diameter with a constant heat flux at the pipe wall using ANSYS FLUENT 2020 R2. The turbulent flow under different Reynolds numbers, from Re = 10000 to 50000 was used to process the numerical experiments. Different concentrations of nanoparticles, ranging from 1% to 4%, were used. The results were computed utilizing the single-phase approach. The coefficient of transfer of heat of nanofluids is found to be greater than that of base liquid. Enhancement in transfer of heat is also observed with rising in volume concentration of particles.

1. Introduction

In today's industry, heat exchangers are quite vital and their main role is to transfer heat at a maximum rate. Many various strategies are already being employed previously to improve the rate of transfer of heat [1] and achieve a desirable degree of thermal efficiency. Changes in flow design, boundary conditions, or thermophysical parameters, such as fluid thermal conductivity, can all passively improve heat transfer rates.

Traditional fluids for heat transfer, like ethylene glycol, water, and oil, have insufficient thermal properties [2]. To boost the overall heat transfer rate, a new novel fluid named Nanofluid has been first introduced by Choi et al. [3] which has piqued researchers' interest because their thermal conductivities are high when compared to the traditional fluids for heat transfer. It is feasible to reduce the size of equipment for the transfer of heat by using nanofluids.

Nanofluids comprise nanoparticles in a usual fluid and these particles are in colloidal suspension. Carbides, metals, oxides, carbon nanotubes are nanoparticles that are most commonly employed in Nanofluids. TiO₂, CuO, Al₂O₃, SiO₂, and CeO₂ are examples of common nanoparticles.

In this field of heat transfer, there has been a significant rise in research activity recently. To explain the improvement of heat transfer, various models have been developed. Several publications published in the last few years have focused on the experimental investigation of the convection of nanofluid in both turbulent and laminar regimes.

Pak and Cho [4] reported the first research on heat transfer via turbulent convection of TiO₂ and Al₂O₃ nanofluids within a circular tube, showing an increment in Nusselt number in association with a concentration of particles and developing a Nusselt number relation. Xuan and Li [5] did another key study on Cu-based nanofluids undergoing turbulent convection, introducing a novel Nusselt number correlation that takes particle concentration into account directly. The forced convection in a laminar
motion of nanofluids in a tube that is circular was experimentally evaluated by S. Zenali et al. [6] with a fixed temperature on the surface. The outcomes indicated that nanoparticles that are there in fluid enhanced transfer of heat.

Numerical simulations of turbulent and laminar convection of nanofluids have been the subject of numerous articles. To carry out the simulations, several methods are used.

The thermodynamic characteristics of nanofluid convection were first considered by Palm et al. [7], and they discovered significant differences between nanofluids with characteristics that are consistent and nanofluids with thermodynamic characteristics. Maiga et al. [8] investigated the feasibility of convection, both turbulent and laminar, within a tube that is uniformly heated, finding that nanoparticles’ existence improved transfer of heat. In a tube that is circular, Vincenzo et al. [9] tested a motion of turbulence nanofluids using single as well as mixture models. The results showed that conventional coefficient of heat transfer of base fluid is smaller than nanofluid and even Reynolds number as well as fraction of particle volume rises, enhancement in heat transfer increases.

In this current paper, turbulent transfer of heat flow inside a pipe which is circular using Al$_2$O$_3$/water nanofluid with constant heat flux on pipe wall is developed and explored numerically utilizing a single-phase technique. The results obtained are achieved in terms of volume concentration, temperature, length, Nusselt number, and Reynolds number profiles.

2. Mathematical modeling

2.1. Geometry

The two-dimensional geometrical configuration as shown in figure 1 consists of a circular pipe having D means diameter of about 0.01 meter and an L means length of about 1 meter. The nanofluid is composed of Al$_2$O$_3$ nanoparticles having a 36 nm diameter and water.

2.2. Numerical approach and Meshing

The current problem was solved using the code for CFD ANSYS FLUENT 20R2. The volume-control scheme was utilized as a tool for resolving governing equations. Under this procedure, the conservation equations are transformed into a series of algebraic problems by spatial integration. For single-phase, pressure-velocity coupling was approached by the SIMPLE algorithm.

To ensure that numerical findings are accurate and consistent, meshing is done. Meshing divides the domain into parts, each representing a different element as shown in figure 2. A default meshing is also performed, which produces results that are not better than the one chosen. In the problem under consideration, the domain is divided into 400×22 elements.

![Figure 1. 2D Circular pipe geometry](image1)

![Figure 2. Meshing](image2)

2.3. Equation that directing

A dynamic fluid, as well as thermal behavior of investigated nanofluid, is studied utilizing a one-phase strategy as in this approach, the fundamental assumption behind it says combination acts as a one-
phase fluid (nanofluid behaves like a homogenous fluid), but thermophysical properties get enhanced owing to the presence of nanoparticles.

The directing equations of a one-phase strategy are mathematically described in the following way [10]

Mass conservation:
\[ \nabla \cdot (\rho_m \vec{V}) = 0 \]

Momentum conservation:
\[ \nabla \cdot (\rho_m \vec{V} \vec{V}) = -\nabla P + \nabla \cdot (\tau - \tau_t) \]

Energy conservation:
\[ \nabla \cdot (\rho \vec{V} C_p T) = \nabla \cdot \left( \lambda \nabla T - c_p \rho m \dot{v} \right) \]

In the equation for energy, viscous dissipation and compression work are considered negligible.

2.4. Turbulence modeling
To account for turbulence processes, estimate models are required to complete the Thermo fluid dynamic field's governing equations. This work considers the k- \( \varepsilon \) model, which was developed by Launder et al. [11], which provides two equations.

For turbulent kinetic energy
\[ \nabla \cdot (\rho_m \vec{V} m_k) = \nabla \cdot \left( \frac{\mu_{t,m}}{\sigma_k} \nabla k \right) + G_{k,m} - \rho_m \varepsilon \]

For dissipation rate
\[ \nabla \cdot (\rho_m \vec{V} m \varepsilon) = \nabla \cdot \left( \frac{\mu_{t,m}}{\sigma_k} \nabla \varepsilon \right) + \frac{\varepsilon}{\kappa} (C_{1,k} G_{k,m} - C_{2} \rho m \varepsilon) \]

2.5. Thermophysical characteristics of nanofluids
The simulation technique necessitates the nanofluid thermophysical characteristics calculation since its outcomes have a big impact. The thermophysical properties of nanofluids with various concentrationsof particle were calculated using classical theory of mixtures as shown in equations 6-9 and used for CFD simulation [10,12,13]. The following are the equations that were employed in this work.

For density
\[ \rho_{nf} = (1 - \phi) \rho_f + \phi \rho_{np} \]

For specific heat
\[ C_{p,nf} = \frac{\phi (\rho C_p)_{np} + (1 - \phi) (\rho C_p)_f}{\rho_{nf}} \]

For viscosity
\[ \mu_{nf} = (123\phi^2 + 7.3\phi + 1)\mu_f \]

For thermal conductivity
\[ k_{nf} = \left[ \frac{k_{np} + 2k_f + 2(k_{np} - k_f)\phi(1 + \beta)^3}{k_{np} + 2k_f + (k_f - k_{np})\phi(1 + \beta)^3} \right] k_f \]

In the above equation,
\[ \beta = \frac{t}{r_p} \]

**Table 1.** Thermophysical properties at T=293k for Al₂O₃ nanoparticles and water[14]

| Material   | \( \rho \) (kg/m³) | \( C_p \) (J/kgK) | \( \mu \) (kg/ms) | \( k \) (W/mK) |
|------------|---------------------|-------------------|------------------|---------------|
| Water, \( \phi=0\% \) | 998.20              | 4182.0            | 9.98\times10^{-4} | 0.597         |
| Al₂O₃      | 3880                | 773               | -                | 36            |

**Table 2.** Thermophysical properties at T=293k for Al₂O₃/water nanofluid

| \( \phi \) (%) | \( \rho \) (kg/m³) | \( C_p \) (J/kgK) | \( \mu \) (kg/ms) | \( k \) (W/mK) |
|----------------|---------------------|-------------------|------------------|---------------|
| 1              | 1027.01             | 4053.21           | 1.023\times10^{-3} | 0.620         |
| 2              | 1055.83             | 3931.45           | 1.048\times10^{-3} | 0.643         |
| 3              | 1084.65             | 3816.16           | 1.073\times10^{-3} | 0.667         |
| 4              | 1113.47             | 3706.84           | 1.098\times10^{-3} | 0.692         |

2.6. *A Boundary conditions*

2.6.1. **Inlet.** The uniform axial velocity, \( V_0 \), for considered Reynold numbers are applied at the inlet. A temperature, \( T_0 = 293 \) K, and turbulence intensity, \( I \), is imposed.

For initial velocity
\[ V_0 = \frac{Re \times \mu_{nf}}{D \times \rho_{nf}} \]

For the intensity of turbulence
\[ I = 0.16 \times Re^{-1/8} \]

2.6.2. **Pipe wall.** While kinetic energy of turbulence, as well as its dissipation, is zero, the non-slip requirements and consistent heat flux are enforced. The surface is heated to a constant temperature (100 °C).

3. **Results**

The transfer of heat in turbulent motion through a circular pipe utilizing Al₂O₃-water nanofluid was investigated using several numerical simulations in this paper. The results were computed using the single-phase approach for \( \phi = 0\% \) to 4\%, \( Re = 10 \times 10^3 \) to 50 \times 10^3, \( q = 500000 \) W/m².
For all particle concentration, the case of \( \text{Re} = 20 \times 10^3 \) and \( \text{Re} = 50 \times 10^3 \) are considered. Figures 3 and 4 show bulk and wall temperature curves along the pipe axis for both Reynolds numbers. Figure 3 illustrates that the bulk temperature increases along the pipe axis but when particle concentration rises, the result remains the same. Figure 4 shows that when the concentration of particles rises, the temperature of the wall goes down which indicates that adding nanoparticles has a favorable influence on the temperature of the wall. In comparison to base fluid, for a value of \( \phi = 1\% \), the temperature of the wall is lower.

The temperature difference between base fluid and nanofluid at pipe exit is around 10 K, resulting in a significant particle effect.

Figure 3. For i) \( \text{Re} = 20 \times 10^3 \) and ii) \( \text{Re} = 50 \times 10^3 \), the effect of particle concentration on bulk temperature along pipe axis.

Figure 4. For i) \( \text{Re} = 20 \times 10^3 \) and ii) \( \text{Re} = 50 \times 10^3 \), the effect of particle concentration on wall temperature along pipe axis.

For \( \text{Re} = 10 \times 10^3 \) to \( 50 \times 10^3 \), the case of \( \phi = 0\% \) (base fluid) and \( \phi = 4\% \) are considered. Figures 5 and 6 show bulk and wall temperature curves along the pipe axis for both particle concentrations. Figure 5 illustrates that the bulk temperature increases along the pipe axis but drops as \( \text{Re} \) increases. It’s clear from figure 6 that when \( \text{Re} \) rises, temperature of wall reduces it depicts that changing \( \text{Re} \) has a positive impact on wall as well as bulk temperatures. In comparison to \( \text{Re} = 20 \times 10^3 \), temperature of wall for \( \text{Re} = 50 \times 10^3 \) is lower.
Figure 5. For i) $\phi = 0\%$ and ii) $\phi = 4\%$, the influence of varying Re on temperature of bulk along pipe axis.

Influence of concentration of particles on coefficient of transfer of heat at surface throughout axis of tube for $Re = 20 \times 10^3$ and $Re = 50 \times 10^3$ is shown in figure 6. It reduces to a minimum value very immediately after entering the pipe where temperature differential exists amid bulk as well as wall is greatest and then slightly decreases along the pipe axis but with accordance to particle concentration, surface heat transfer coefficient increases.

Figure 6. For i) $\phi = 0\%$ and ii) $\phi = 4\%$, the influence of varying Re on wall temperature along pipe axis.

Figure 7. i) $Re = 20 \times 10^3$ and ii) $Re = 50 \times 10^3$, the influence of $\phi$ on coefficient of transfer of heat at surface on along pipe axis.
Average coefficient of transfer of heat rises as particle concentration increases for Re = 20 × 10^3 and Re = 50 × 10^3 as shown in figure 8. The increment in the ratios of the average coefficients of transfer of heat from nanofluid to base fluid is given in Tables 3 and 4. As a result, a nanofluid has a greater thermal conductivity, allowing heat transfer rates to also be expanded. But for increasing particle concentration and increasing average heat coefficient, the Nusselt number decreases.

\[
\text{Figure 8. For i) Re} = 20 \times 10^3 \text{ and ii) Re} = 50 \times 10^3, \text{ the influence of } \phi \text{ on average coefficient of transfer of heat.}
\]

| ϕ (%) | h_{av} (W/mK) | h_{nf}/h_{f} | Nu_{av} | Nu_{nf}/Nu_{f} |
|-------|----------------|--------------|----------|-----------------|
| 0 (bf)| 8052.95        | 1.00         | 134.89   | 1.00            |
| 1     | 8227.21        | 1.02         | 132.70   | 0.98            |
| 2     | 8396.93        | 1.04         | 130.59   | 0.97            |
| 3     | 8569.27        | 1.06         | 128.47   | 0.95            |
| 4     | 8743.95        | 1.09         | 126.36   | 0.94            |

\[
\text{Table 3. Average Heat transfer coefficients and Nusselt Number for Re} = 20000
\]

| ϕ (%) | h_{av} (W/mK) | h_{nf}/h_{f} | Nu_{av} | Nu_{nf}/Nu_{f} |
|-------|----------------|--------------|----------|-----------------|
| 0 (bf)| 17665.2        | 1.00         | 295.90   | 1.00            |
| 1     | 18051.2        | 1.02         | 291.15   | 0.98            |
| 2     | 18427          | 1.04         | 286.58   | 0.97            |
| 3     | 18808.7        | 1.06         | 281.99   | 0.95            |
| 4     | 19195.4        | 1.09         | 277.39   | 0.94            |

\[
\text{Table 4. Average coefficients of heat transfer as well as Nusselt Number for Re} = 50000
\]

Similarly, influence of increasing Re on coefficient of heat transfer at surface along pipe axis as evidenced in figure 9. for ϕ = 0% and ϕ = 4%.

Table 5. shows average coefficient of transfer of heat at surface for all considered cases. The table clearly shows that average coefficient of transfer of heat increases as concentration of particles as well as Re rises. Therefore, the influence of particle concentration appears to be more significant at higher Re. Nusselt no. rises in parallel with coefficient of average heat transfer.
Figure 9. For i) $\phi = 0\%$ and ii) $\phi = 4\%$, influence of different Re on coefficient of transfer of heat at surface along pipe axis

Table 5. Average coefficients of transfer of heat for all considered cases

| Re    | Base Fluid | $\phi = 1\%$ | $\phi = 2\%$ | $\phi = 3\%$ | $\phi = 4\%$ |
|-------|------------|---------------|---------------|---------------|---------------|
| 10000 | 4522.58    | 4619.01       | 4712.96       | 4808.43       | 4905.26       |
| 20000 | 8052.95    | 8227.21       | 8396.93       | 8569.27       | 8743.95       |
| 30000 | 11636.7    | 11893         | 12142.5       | 12395.7       | 12652.1       |
| 40000 | 14732.9    | 15054.6       | 15367.9       | 15686.1       | 16008.6       |
| 50000 | 17665.2    | 18051.2       | 18427         | 18808.7       | 19195.4       |

Based on the simulation findings, the Nu numbers are computed as follows:

$$Nu = \frac{h \cdot D}{k}$$

The average Nu for considered Re and concentration of particles evidenced in Figure 10. Nu rises as Re number rises, but falls as concentration of particle rises.

Figure 10. For varying $\phi$, compare the Nusselt number to the Reynolds number.

4. Conclusion

The numerical and CFD simulation of coefficient of transfer of heat of turbulent convective motion of alumina-based nanofluid within a circular pipe is presented in this study using a single-phase model.
1. These results have demonstrated that in contrast to base fluid, incorporating nanoparticles in a base fluid made a considerable contribution to coefficient of transfer of heat.

2. The data clearly illustrates that as concentration of particle as well as Re rises, so does coefficient of transfer of heat.

3. For all concentrations of particles, maximal rates of transfer of heat were found in correlation with the highest Reynolds.

4. Tables 3 and 4 exhibits average coefficient of transfer of heat & Nusselt number as a function of concentration of nanoparticle, with proportionate rises in total rates of heat transfer and decrease in Nusselt number but with rising Re, average coefficient of heat transfer and Nusselt number increases.

Nomenclatures

\begin{itemize}
\item $D$ diameter of the pipe, m
\item $d$ diameter of nanoparticles, m
\item $r$ radius of nanoparticle, m
\item $V$ velocity of flow, m/s
\item $C_p$ fluid's specific heat, J/kg K
\item $L$ length of pipe, m
\item $h$ coefficient of heat transfer, W/m$^2$K
\item $T$ temperature, K
\item $k$ kinetic energy of turbulence, m$^2$/s$^2$
\item $t$ thickness of the nanolayer
\item $I$ turbulence intensity
\item $Nu$ Nusselt number,
\item $q$ heat flux of wall, W/m$^2$
\item $Re$ Reynolds number
\end{itemize}

Greek Symbols

\begin{itemize}
\item $\rho$ density of the fluid, kg/m$^3$
\item $\phi$ volume concentration of particle
\item $\varepsilon$ dissipation of turbulent kinetic energy, m$^2$/s$^3$
\item $\lambda$ fluid thermal conductivity, W/mK
\item $\tau$ wall shear stress, Pa
\item $\mu$ dynamic viscosity of the fluid, kg/ms
\end{itemize}

Subscript

\begin{itemize}
\item $m$ mixture
\item $t$ turbulent
\item $b$ bulk
\item $np$ nanoparticle
\item $nf$ nanofluid
\item $av$ average
\item $s$ surface
\item $0$ inlet condition
\item $f$ base fluid
\end{itemize}

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