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# Numerical Simulation of the Flow in a Concentric Double-Pipe Heat Exchanger with a Square Inner Pipe and a Circular Outer Pipe

Hamoon Pourmirzaagha[a], Alireza Yekrangi Sendi[b,*], Saeed Janzadeh Karat Mahalleh[c]

[a] *Department of Mechanical and Aerospace Engineering, Faculty of Mechanical and Aerospace Engineering, Islamic Azad University, Ramsar Branch, Ramsar, Iran.*

[b] *Department of Mechanical and Aerospace Engineering, Faculty of Mechanical and Aerospace Engineering, Islamic Azad University, Ramsar Branch, Ramsar, Iran.*

[c] *Department of Mechanical and Aerospace Engineering, Faculty of Mechanical and Aerospace Engineering, Islamic Azad University, Ramsar Branch, Ramsar, Iran.*

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Double-pipe heat exchangers are commonly used in various industries, including power plants, solar cells, refineries, and automotive. The double-pipe heat exchanger is one of the most used simple exchangers in the industry. In this study, we employ computational fluid dynamics to investigate the flow characteristics of nanofluids within a double-pipe heat exchanger featuring a square inner tube and a circular outer tube (SC). The simulations are conducted under constant heat flux conditions, exploring laminar and turbulent flow regimes. Numerical results for water flow under forced convection are compared with reference results for validation. The results indicate that as the Reynolds number increases, particularly in turbulent flow regimes, the Nusselt number in nanofluid flow increases more than in water flow. For instance, in the case of aluminum oxide nanofluid flow at a Reynolds number of 500, the Nusselt number demonstrates a nearly 5% enhancement over water flow. In contrast, at a Reynolds number of 20000, this enhancement escalates to approximately 20%. Three types of nanoparticles are considered to investigate the effect of nanoparticle type on heat transfer and pressure drop. The results show that the use of nanoparticles has a slight effect on the friction factor while significantly enhancing heat transfer.

## 1. Introduction

*Corresponding Author: yekrangi_ali@yahoo.com*

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The rapid expansion of industrial activities has led to the significant release of greenhouse gases, contributing to numerous environmental challenges for humanity.Scientists have prioritized enhancing efficiency, particularly in the functioning of thermal systems, as these systems directly involve energy usage. Improving the effectiveness of thermal systems yields advantages for the environment, economy, and operations. Among the many heat transfer equipment, engineers find heat exchangers particularly intriguing.

A heat exchanger is a device that facilitates the transfer of heat between two fluids or between a fluid and a solid surface, creating favorable conditions for heat exchange. Heat exchangers are commonly used to either cool a hot fluid, heat a low-temperature fluid, or both simultaneously. One of the most important applications of heat exchangers in renewable systems, especially in solar cells is. For example, solar water heating systems use heat exchangers to transfer solar energy absorbed in solar collectors to potable (drinkable) water. The double-pipe heat exchanger, composed of two concentric pipes, is a commonly employed and straightforward type of heat exchanger in several industries. This heat exchanger is used in various operating conditions and offers several advantages. These include simple construction, cost-effective installation, straightforward calculations and design, easy fluid flow control in two paths, convenient maintenance and cleaning, and suitability for high-pressure applications. Double-pipe heat exchangers are commonly used in industries to handle fluids susceptible to fouling.

Thermal scientists and engineers focus on achieving the most efficient design of heat exchangers while minimizing their size and construction costs. Ding et al. [1] conducted an experimental study on the laminar flow of nanofluids in a copper pipe. They observed a substantial enhancement in the heat transfer coefficient at the inlet section using an aluminum oxide-water nanofluid. Vajjha and Das [2] conducted an experimental study to investigate the impact of thermal conductivity on three distinct nanofluids: aluminum oxide, copper oxide, and zinc oxide. These nanofluids were combined with ethylene glycol and water, which served as the base fluid. The findings indicated an increase in the thermal conductivity of nanofluids compared to the base fluids. They demonstrated that the ethylene glycol-aluminum oxide nanofluid had a 29% increase in thermal conductivity at a 10% volume concentration compared to a 1% volume concentration. Furthermore, for a 7% volume concentration, the thermal conductivity of the ethylene glycol-zinc oxide nanofluid increased by 48.5% when compared to the base fluid. In the temperature range of 298 kelvin to 363 kelvin, a 21.4% improvement in thermal conductivity for ethylene glycol-copper oxide nanofluid was reported at a 6% volume concentration.

Fotukian et al. [3] investigated the pressure drop and heat transfer in a circular pipe with a constant temperature boundary condition for the turbulent flow of a water-aluminum oxide nanofluid. The volume fraction of nanoparticles was below 0.3%. They demonstrated that the increase in nanoparticle concentration does not always lead to an increase in pressure drop. Additionally, finding the optimal nanoparticle concentration for each nanofluid is essential to achieving improved heat transfer and reduced pressure drop. Zhang et al. [4] conducted an experimental study on heat transfer in a double-pipe heat exchanger equipped with helical fins on the inner pipe. They incorporated several vortex generators on the inner pipe to enhance the heat transfer performance. Experimental results using air as the working fluid demonstrated the pivotal significance of the vortex generators in augmenting heat transfer. By comparing the heat transfer enhancement using a heat exchanger equipped with vortex generators and helical fins, heat transfer in the heat exchanger with helical fins was greater under similar conditions.

In another study by Sheikholeslami et al. [5], angular fins were installed on the surface of the pipe in a double-pipe heat exchanger. During the experimental investigations, where hot water flowed inside the inner pipe and cold air circulated between the two pipes, they studied pressure drop and heat transfer in the heat exchanger. Their findings revealed that incorporating angular fins increased friction factor and Nusselt number. As a result, the heat exchanger's overall performance improved. Han et al. [6] performed an empirical investigation to assess the influence of aluminum oxide nanofluid, with volume fractions of 0.5% and 0.25%, on the water base fluid at different inlet temperatures. They found that higher temperatures and nanoparticle concentrations led to improved heat transfer.

Qi et al. [7] conducted experiments to examine the thermal performance and pressure drop of titanium dioxide ($TiO_2$) nanofluid in both smooth and wavy double-tube heat exchangers. They found that the wavy double-tube heat exchanger exhibited significantly superior thermal performance to the smooth double-tube heat exchanger. Nevertheless, the nanofluid experiences a significant decrease in pressure within the wavy double-tube heat exchanger.

Karimi et al. [8] applied a numerical investigation on a mixed-phase model to replicate the flow of aluminum oxide-water nanofluid within a double-tube heat exchanger equipped with twisted tapes at various pitch ratios. The primary results revealed that using twisted tapes in the tube enhances heat transfer, pressure drop, and friction factor.

Heat transfer can be increased by up to 10% by choosing tapes with the highest roughness. Nevertheless, the roughness of tapes within the tube increased pressure drop and friction factor by up to 28%. A numerical analysis was used by Gnanaval et al. [9] to examine the impact of incorporating twisted tapes with rectangular cuts on their ribs in a circular double-pipe heat exchanger in thermal and flow fields. The results demonstrated the higher performance of the heat exchanger equipped with a specially designed twisted tape than the one without it (smooth pipe). Moreover, the titanium dioxide ($TiO_2$) nanofluid demonstrated superior thermal efficiency compared to the analyzed nanofluids. The nanofluids containing beryllium oxide (BeO), zinc oxide (ZnO), and copper oxide (CuO) are ranked in the following categories, respectively.

Sinaga et al. [10] examined how air and water passage in a two-phase flow affect the thermomechanical performance of a horizontal double-pipe heat exchanger. The air and water streams were mixed in an exterior three-way junction before entering the inner pipe. The findings indicated that the overall heat transfer coefficient positively correlated with the volume fraction, regardless of the hot water flow rates.

El Maakoul et al. [11-13] studied the characteristics of a double tube with split longitudinal and helical fins on the annulus side. The results showed that the helical fin enhances the annulus side heat transfer significantly with the pressure drop penalty compared with the double tube heat exchanger without a helical fin under the same unit weight.

Majidi et al. [14] experimentally worked on the air heat transfer in a helical fin in the double tube heat exchanger and presented an equation for increasing the overall heat transfer coefficient due to the fin. The results showed that the presence of an annulus fin increases the overall heat transfer coefficient. Asadi et al. [15] considered Ag-$MoS_2$ and $Fe_3O_4$-$SiO_2$ hybrid nanofluids in a double pipe Heat Exchanger and observed heat transfer coefficient enhancement of 62.21% for 1% vol. of Ag-$MoS_2$ hybrid nanofluid numerically. Sharifi et al. [16] numerically reported that *the Nu* of a double tube heat exchanger with wire-coiled inserts increased by 1.77 times. Alhusseny et al. [17] proposed a double tube heat exchanger with a rotating metal foam structure that significantly improves the heat transfer and reduces the pumping power. Andrzejczyk et al. [18] experimentally reported that the heat transfer of a double tube heat exchanger with wire inserts increases up to 280%. Arjmandi et al. [19] numerically found that combined twisted tape with a vortex generator can improve the efficiency of a double tube heat exchanger by five times. Córcoles et al. [20] considered a helically corrugated tube to enhance the performance of a double-pipe heat exchanger. Different combinations of corrugation height and pitch were studied, and their performance was compared with a plain tube.

Kola et al. [21] examined the heat transfer performance of a double pipe heat exchanger with twisted tapes varying mass flow rate, cur geometry, etc. Nakhchi and Esfahani [22] incorporated a two-phase CuO nanofluid in a heat pipe, inserting the louvered strip. They observed that the optimum angle of the strip is 25%, and the maximum thermal augmentation is found to be 1.99 at a certain velocity.

Using nanofluid, Lokhande et al. [23] investigate the heat transfer in a shell and tube heat exchanger. They observed that the overall heat transfer coefficient rose by 1.27 times for 0.01% concentration. Rahman et al. [24] worked on a rectangular channel with square ribs to assess the heat transfer characteristics. They noticed that when the p/k ratio is eight, the highest amount of heat transfer is found compared to other ratios. Ghazanfari et al. [25] evaluated the thermal acts of a shell and tube heat exchanger incorporating twisted tubes using alumina nanofluid. They discovered that the twisted tube ensures a 20% increase in thermal performance and a 14% increase in pressure drop.

Tusar et al. [26] numerically compared the performances between plain and helical strip-fitted annulus. They confirmed that helical strip inserts can increase the heat transfer with an additional penalty of pressure drop, and the helical strip-fitted annuli can be more energy-saving at lower Re. Hangi et al. [27] numerically studied a double tube heat exchanger equipped with helical inserts on the tube side and helical strips on the annulus side of different configurations.

Chaurasia and Sarviya [28] experimentally compared the thermo-hydraulic performance between tubes with single and double helical strip inserts. The double insert has a more significant Nusselt number and pressure drop than the single insert and has excellent thermal performance compared to the single insert in most cases.

As mentioned in the previous research, all research has predominantly concentrated on the effect of nanofluids on heat transfer and pressure drop in simple or wavy double-pipe heat exchangers without considering the effects of various parameters along with geometric variations. This study examines heat transfer and pressure drop in a proposed heat exchanger with a square inner tube and a simple circular outer tube using a numerical method in both laminar and forced convection regimes, considering geometric dimensional changes. The primary objectives of this research are as follows:

- Investigating the impact of structural and geometric dimensions on heat transfer and pressure drop in the proposed double-pipe heat exchanger.
- Adding various nanofluids to the base fluid at different volume percentages and examining their effects on heat transfer and pressure drop in the double-pipe heat exchanger.

## 2. Geometry and Governing Equations

Figure 1 illustrates a three-dimensional view of the double-pipe heat exchanger under study. Briefly, the geometry of the two pipes with a square inner wall and a circular outer wall is referred to as SC.

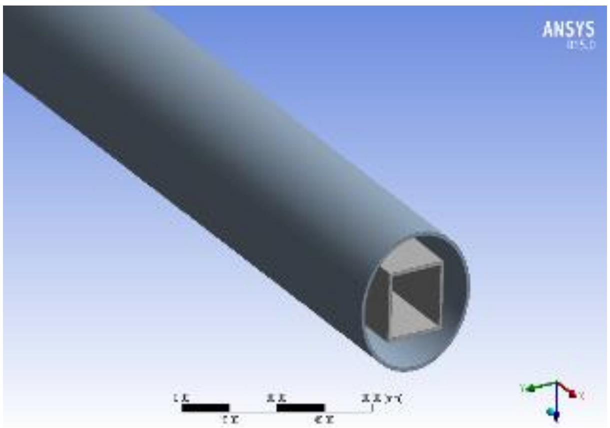


Figure 1. The geometry under investigation

Due to the investigation of the effect of pipe dimensions on the thermo-hydraulic characteristics, multiple sizes of pipes have been provided. Table 1 presents information related to the geometric dimensions.

Table 1. Geometric Specifications

| Geometry abbreviations | Heat exchanger length L (mm) | Internal wall size $D_i$ (mm) | External wall size $D_o$ (mm) | Hydraulic diameter size $D_h$ (mm) |
|---|---|---|---|---|
| SC1 | 1500 | 24 | 41 | 13 |
| SC2 | 1500 | 26 | 41 | 11 |
| SC3 | 1500 | 20 | 41 | 17 |
| SC4 | 1500 | 20 | 51 | 27 |
| SC5 | 1500 | 26 | 68 | 37 |

In fluid mechanics and heat transfer, fundamental laws such as the continuity equation, the Navier-Stokes equation, and the energy equation are employed to describe velocity and temperature fields. In the context of steady-state incompressible flow, these equations are expressed as follows [29]:

$$\nabla . V = 0 \tag{1}$$

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial(p)}{\partial x_i} + \mu \frac{\partial}{\partial x_j}\left(2s_{ij} - \frac{2}{3}\frac{\partial u_k}{\partial x_k}\delta_{ij}\right) \tag{2}$$

$$\frac{\partial(\rho H u_i)}{\partial x_i} = \frac{\partial}{\partial x_i}\left(u_j \tau_{ij} + K\frac{\partial T}{\partial x_i}\right) \tag{3}$$

The Navier-Stokes equations represent a comprehensive mathematical framework for describing fluid motion, yet their inherent complexity renders analytical solutions impractical. Consequently, numerical methods, augmented by computational power, have emerged as the most effective means of tackling these equations. The remarkable advancements in computer technology over recent decades have facilitated the widespread adoption of computational fluid dynamics (CFD) for numerically resolving fluid flow problems. Given the inherently turbulent nature of the flow under investigation in this study, it becomes imperative to analyze the equations within the context of turbulent flow. The k-epsilon turbulence model is well recognized as one of the most prominent models for simulated turbulent flow. It encompasses several models, such as Standard, RNG, and Realizable. These models incorporate two additional turbulence transport equations into the existing flow equations, specifically for turbulent kinetic energy and dissipation rate. Research has demonstrated that out of the several numerical approaches available, the standard k-epsilon model yields the most accurate outcomes for non-Newtonian power-law fluids [30, 31].

The Reynolds stress in the standard k-epsilon model is calculated using the Boussinesq technique, as explained below. The Kronecker delta is indicated as $\delta_{ij}$, and the strain rate tensor as $S_{ij}$, represented by the equation (4) [32]:

$$S_{ij}=\frac{1}{2}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right) \quad (4)$$

$\kappa$ is the turbulent kinetic energy obtained from equation (5) [32]:

$$\frac{\partial(\rho\kappa)}{\partial t}+\frac{\partial(\rho\kappa u_i)}{\partial x_i}=\frac{\partial}{\partial x_j}\left[(\mu+\frac{\mu_t}{\sigma_\kappa})\frac{\partial\kappa}{\partial x_j}\right] +G_\kappa+G_b-\rho\varepsilon \quad (5)$$

$\varepsilon$ is the rate of viscous dissipation of turbulent kinetic energy, and according to equation (6), it is obtained as [32]:

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\frac{\partial(\rho\varepsilon u_i)}{\partial x_i}=\frac{\partial}{\partial x_j}\left[(\mu+\frac{\mu_t}{\sigma_\varepsilon})\frac{\partial\varepsilon}{\partial x_j}\right] +C_{1\varepsilon}\frac{\varepsilon}{\kappa}(G_\kappa+C_{3\varepsilon}G_b)-\rho C_{2\varepsilon}\frac{\varepsilon^2}{\kappa} \quad (6)$$

In these equations, the production of turbulent kinetic energy ($G_k$) is due to the gradients of the mean velocity, and the production of turbulent kinetic energy by turbulence ($G_b$) is a result of the buoyancy effect. The constants in the equation are $C_{1\varepsilon}$, $C_{2\varepsilon}$, and $C_{3\varepsilon}$, and $\sigma_k$ and $\sigma_\varepsilon$ represent the turbulent Prandtl numbers for k and ε, respectively. These constants are obtained from experimental results.

This study used water as a Newtonian fluid, assuming nanofluids to be homogenized (single phase). Tables 2 and 3 show the thermophysical characteristics of the water and nanofluid.

Table 2. Thermophysical properties of water [5]

| $\rho\left(kg/m^3\right)$ | $C_P\left(J/kgK\right)$ | $K\left(W/mK\right)$ | $\mu\left(kg/ms\right)$ | $\beta\left(1/K\right)$ |
|---|---|---|---|---|
| 998.2 | 4182 | 0.597 | 9.93E-4 | 2.1E-4 |

Table 3. Thermophysical properties of nanoparticles [33]

| $\rho\left(kg/m^3\right)$ | $C_P\left(J/kgK\right)$ | $K\left(W/mK\right)$ | $B(nm)$ | |
|---|---|---|---|---|
| 3900 | 880 | 42.34 | 8.5E-6 | $Al_2O_3$ |
| 4250 | 686.2 | 8.9 | 9E-6 | $TiO_2$ |
| 6350 | 535.6 | 69 | 1.67E-6 | CuO |

## 3. Results and Discussion

The numerical results were compared with the experimental results of Bhadouriya et al. [34] to validate the numerical study in both laminar and turbulent flows. This validation was conducted where water flows inside the exchanger, using a double-pipe geometry with square inner walls and circular outer walls (SC) for the comparative analysis. The results for the Nusselt number (Nu) and the Reynolds-friction factor ($f_{Re}$) product were presented in Tables 4 and 5, respectively. The comparison showed that the difference in the friction factor and Nusselt number in laminar flow is less than 4%, with the maximum difference observed in turbulent flow. The Reynolds-friction factor product in turbulent flow has less than an 11% difference, indicating the accuracy of the numerical solution.

Table 4. Comparison of the Nusselt number in the present study with the experimental study by Bhadouriya et al. [34]

| Re | Nu | | Difference percentage |
|---|---|---|---|
| | Numerical solution | Experimental results | |
| 1000 | 4.28 | 4.41 | -3% |
| 2000 | 4.32 | 4.46 | -4% |
| 11000 | 40.05 | 43.31 | -8% |

Table 5. Comparison of the friction factor with the experimental study by Bhadouriya et al. [34]

| Re | $f_{Re}$ | | Difference percentage |
|---|---|---|---|
| | Numerical solution | Experimental results | |
| 1000 | 22.14 | 22.7 | -3% |
| 2000 | 21.64 | 22.4 | -4% |
| 11000 | 96.23 | 87.4 | 11% |

### 3.1. Investigating the Effect of SC Geometry on Pressure Drop and Heat Transfer in Forced Convection

A double-pipe heat exchanger with a square inner tube and circular outer tube (SC) was investigated in five different geometries. The results for the water flow's friction factor and Nusselt number in these SC geometries are presented in Figures 2 and 3, respectively. Examinations were conducted across a range of Reynolds numbers from 500 to 20000. The 500–20000 Reynolds number range is the subject of the investigation. As depicted in Figure 2, the Nusselt number demonstrates an increase with rising Reynolds numbers. Notably, turbulent flow exhibits a significantly higher Nusselt number compared to laminar flow. SC5 exhibits the highest heat transfer capability. Furthermore, a change in geometry leads to a notable alteration in pressure drop, even when the Nusselt numbers for different geometries are closely aligned.

In the laminar flow regime, the Nusselt number exhibits a sharp increase, whereas in turbulent flow, the rise in Nusselt number with increasing Reynolds number is less pronounced. Figure 3 illustrates that the SC2 geometry experiences a substantial friction factor compared to the other geometries. This can be attributed to the relationship between hydrodynamic diameter and pressure drop in the flow. A decrease in hydrodynamic diameter corresponds to a significant increase in pressure drop. Consequently, geometries with smaller hydrodynamic diameters demonstrate higher pressure drops, as evidenced in Figure 3, where the friction factor of SC2 is nearly three times that of SC1.

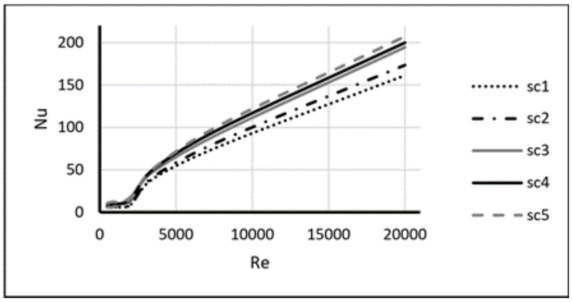


Figure 2. Changes in the Nusselt number concerning Reynolds number for different SC geometries and forced convection.

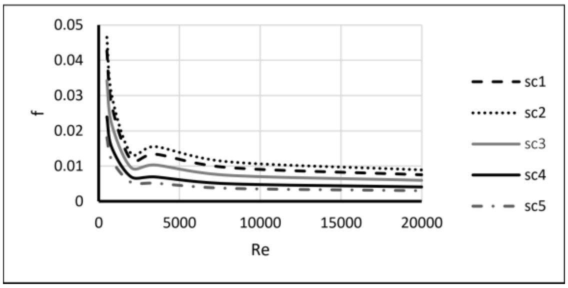


Figure 3. Changes in the friction coefficient concerning Reynolds number for different SC geometries and forced convection.

### 3.2. Investigating the Effect of Nanoparticle Type on Pressure Drop and Heat Transfer in Forced Convection in SC Geometry

Three types of nanoparticles, Aluminum Oxide ($Al_2O_3$), Titanium Oxide (TiO), and Copper Oxide (CuO), with a volume fraction of 5.1% and base fluid (water), were taken into consideration in order to examine the impact of nanoparticle type on heat transfer and pressure drop in the heat exchanger with square inner walls and circular outer walls (SC). Figures 4 and 5 illustrate the results for the Nusselt number and friction factor for the SC3 geometry as functions of the Reynolds number.

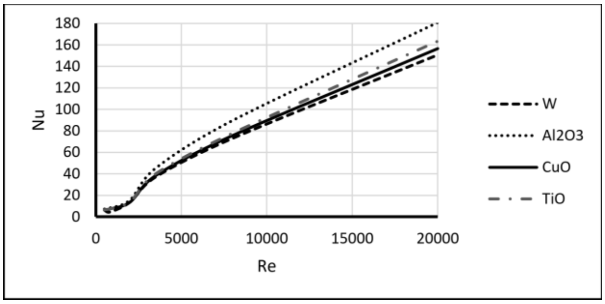


Figure 4. Changes in the Nusselt number concerning Reynolds number (Re) in SC3 geometry for different nanoparticle fluids and forced convection.

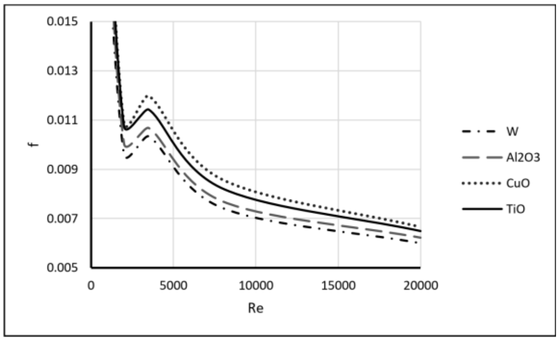


Figure 5. Changes in the friction factor concerning Reynolds number in SC3 geometry for different nanoparticle fluids and forced convection.

The addition of nanoparticles has a negligible effect on the friction factor, as demonstrated in Figure 5. However, Figure 4 illustrates a considerable enhancement in heat transfer attributed to the presence of nanoparticles. Among the various nanoparticles examined, Aluminum oxide exhibits the most substantial increase in the Nusselt number, owing to its higher thermal conductivity.

Relative to the base fluid, the Nusselt number of nanoparticle-laden flow increases with rising Reynolds numbers, particularly in turbulent flow. For instance, at a Reynolds number of 500, the Nusselt number is approximately 5% higher in nanoparticle-laden flow compared to the base fluid, and this difference exceeds 20% at a Reynolds number of 20,000. This phenomenon arises due to the heightened turbulence within the flow upon the addition of nanoparticles to the base fluid.

Compared to laminar flow, heat transfer improves dramatically with increasing flow turbulence because it makes it easier for the flow to penetrate through the boundary layers and into the core. Following titanium oxide nanofluid, copper oxide nanofluid resulted in the most significant pressure decrease and the slightest improvement in heat transfer among the different nanofluids.

### 3.3. Investigating the Effect of Nanoparticle Volume Fraction on Pressure Drop and Heat Transfer for Flow in SC Geometry and Forced Convection

Three volume fractions of aluminum oxide ($Al_2O_3$) nanoparticles and the base water fluid—0.5%, 1.5%, and 3%—were taken into consideration in order to examine the impact of nanoparticle volume fraction on heat transfer and pressure drop in the heat exchanger with square inner walls and circular outer walls (SC). Figures 6 and 7 illustrate the friction factor and Nusselt number results for SC3 geometry concerning the Reynolds number. The Nusselt number rises when Aluminum Oxide nanoparticles are added to the base fluid, as seen in Figure 6. It can be concluded that this increase is directly correlated with the volume fraction of nanoparticles. For instance, a fluid with a 3% volume fraction has a Nusselt number about 20% higher than a fluid with a 1.5% volume fraction. This improvement in heat transfer is attributed to enhanced convective heat transfer and increased flow turbulence in the fluid with a higher volume fraction of nanoparticles. Consequently, as the volume fraction of nanoparticles increases, the Nusselt number experiences a more significant enhancement.

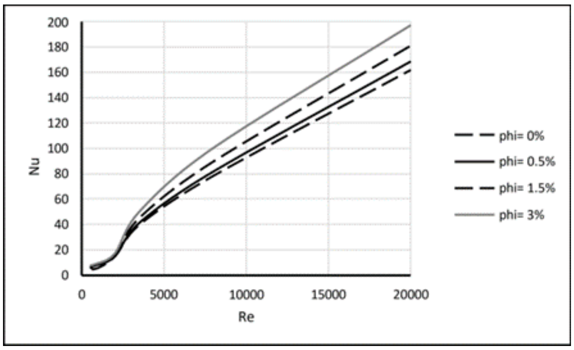


Figure 6. Changes in the Nusselt number concerning Reynolds number in SC3 geometry for different volume fractions of Al2O3 nanoparticle fluids and forced convection.

As illustrated in Figure 7, the addition of nanoparticles to the base fluid leads to a slight increase in pressure drop, with this effect becoming more pronounced at higher volume fractions. The friction coefficient of nanoparticle-laden flow exhibits only minor variations compared to that of the base fluid. The addition of nanoparticle fluid increases pressure drop, which the fluid's increased viscosity can explain. Furthermore, adding nanoparticles to the base fluid raises the flow's turbulence level. As turbulence intensifies, a substantial portion of fluid pressure and energy is expended in turbulence generation. Consequently, increasing nanoparticle volume fraction in turbulent flow conditions results in a more significant pressure drop. Also, it is observed in the laminar flow regime, the friction coefficient decreased sharply while in the turbulent flow regime decrease in the friction coefficient with the increase in the Reynolds number was not as high.

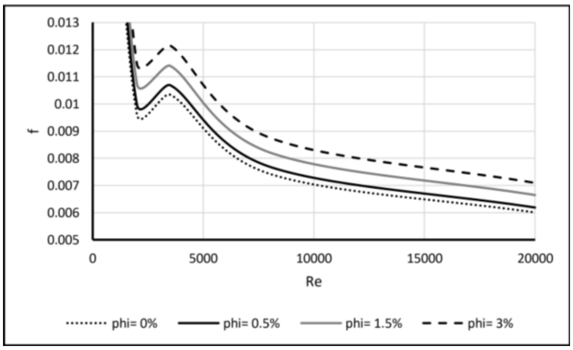


Figure 7. Changes in the friction coefficient concerning Reynolds number in SC3 geometry for different volume fractions of Al2O3 nanoparticle fluids and forced convection.

## 4. Conclusions

This study employed a numerical method based on finite volume simulation to investigate the influence of geometric parameters and nanoparticle addition on heat transfer and pressure drop in a double-pipe heat exchanger operating under constant heat flux conditions and across laminar and turbulent flow regimes. The key findings of our study are as follows:

- An increase in the hydraulic diameter of a heat exchanger leads to improved heat transfer performance; however, it also results in a simultaneous increase in pressure drop within the system.
- To evaluate the impact of different nanoparticle types on heat transfer and pressure drop in a double-pipe heat exchanger featuring a square inner tube and a circular outer tube, we

investigated the effects of aluminum oxide (Al2O3), titanium oxide (TiO), and copper oxide (CuO) nanoparticles at a volume fraction of 1.5%, in addition to the base fluid (water). Our findings indicate a substantial enhancement in heat transfer when utilizing nanoparticles, with minimal effect on the friction factor.

- In this study, flow with Reynolds numbers ranging from 500 to 20,000 was examined. The results revealed that for Reynolds numbers below 5000, nanoparticle utilization has a more significant effect on heat transfer in the exchanger.
- Among various nanoparticles, aluminum oxide exhibited the most significant increase in the Nusselt number, attributed to its higher thermal conductivity compared to other nanoparticles. Additionally, copper oxide resulted in the highest pressure drop and the slightest improvement in heat transfer among different nanofluids.
- To investigate the influence of nanoparticle volume fraction on heat transfer and pressure drop within the heat exchanger with a square inner tube and a circular outer tube, three-volume fractions of 0.5%, 1.5%, and 3% were considered for aluminum oxide nanoparticles (Al2O3) and the base fluid (water). The results showed that the Nusselt number for the fluid with a volume fraction of 3% is approximately 20% higher than that for the nanofluid with a volume fraction of 1.5%.
- In the laminar flow regime, a sharp increase in the Nusselt number was observed, whereas in the turbulent flow regime, the rate of increase in the Nusselt number with rising Reynolds numbers was comparatively lower.

Additionally, the following aspects could be considered as complementary and relevant backgrounds to the current study:

- Experimental investigation of various nanofluids in double-pipe heat exchangers and comparison with numerical results.
- Experimental investigation of non-Newtonian fluids in double-pipe heat exchangers.
- Investigation of porous geometry on heat transfer and pressure drop.

**Nomenclature**

| | |
|---|---|
| Cp | specific heat capacity, J/(kg K) |
| Ci | coefficients in k _ ε model |
| Di | internal annulus diameter, m |
| $D_0$ | external tube diameter, m |
| $D_h$ | hydraulic diameter, m |
| k | turbulent fluctuation kinetic energy, $m^2/s^2$ |
| f | friction factor |
| Pr | Prandtl number |
| Re | Reynolds number |
| Nu | Nusselt number |
| u | average velocity, m/s |
| ε | turbulent kinetic energy dissipation rate, , $m^2/s^2$ |
| $\mu$ | dynamic viscosity, kg/(m s) |
| $\rho$ | density, $kg/m^3$ |
| $\sigma$k | Prandtl number for k |
| $\sigma$ ε | Prandtl number for ε |

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