Proceedings of the 10th World Congress on Mechanical, Chemical, and Material Engineering (MCM'24)
Barcelona, Spain –August 22-24, 2024
Paper No. HTFF 286
DOI: 10.11159/htff24.286

Numerical Investigation of Heat Transfer Characteristics of Nanofluid Using Parallel and, Counter Flow Configurations

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Abstract **-** Heat transfer occurs between fluids at different temperatures using a device called a heat exchanger. Heat transfer devices with concentric double pipe heat exchangers are widely employed for many industrial applications. Nanoparticles embedded into the fluid have attracted many researchers for their ability to enhance the fluid thermal conductivity and surface area to conserve power. In this article, the flow properties and convective heat transfer of nanofluids containing 0.2%, 0.4% and 0.6% vol. TiO_2 nanoparticles distributed in water are numerically examined. Using 21 nm TiO_2 nanoparticles with pure water as fluid, are utilized for parallel and counter flow configurations in the presence of turbulent conditions with Reynolds numbers varying from 4,000 to 18,000. The results of the numerical model demonstrate that the convective heat transfer coefficient of nanofluids is enhanced when the Reynolds and volume fraction increase for both parallel and counter-flow configurations. While raising the temperature of the annular (hot water), the heat transfer rate would increase, however, it has minimal impact on the convective heat transfer coefficient of nanofluids. Using a counterflow configuration improves the convective heat transfer coefficient compared to parallel flow but has no meaningful influence on the heat transfer rate.

Keywords**: N**anofluids; TiO_2; Parallel; Counterflow; Heat transfer coefficient; CFD

1. Introduction

In various industrial and cooling applications, including HVAC systems, heat transfer fluids such as water, oil and ethylene glycol are commonly used. The double pipe heat exchanger is one of several heat exchanger devices. By using conduction through the double pipe wall separating the fluids and convection within the fluid, this device permits the exchange of heat between the two fluids. However, the low thermal conductivity of conventional fluids leads to poor heat transfer performance. To address this, many researchers are actively investigating the use of solid metallic nanoparticles in conventional fluids to enhance heat transfer within the double pipe heat exchanger. The double pipe heat exchanger was invented in the late 1940s, and various studies have consistently supported its use, resulting in significant gains [1]. While numerous experimental and numerical investigations have been conducted [2–5], the majority of these studies have focused on identifying the parameters that influence thermal performance, such as the operating temperature, inner and outer Reynolds numbers, volume fractions, thermal conductivity and dispersed nanoparticle shape. In addition to the particle shape, particle size was found to be dominant in improving the thermal conductivity of nanofluids [6]. Furthermore, increasing nanoparticle size can decrease the nanofluid viscosity [7 and 8]. On the other hand, heat exchanger performance can be affected by increasing nanoparticle volume fractions, which leads to an increase in nanofluid viscosity and thermal conductivity. Enhancing the thermal conductivity is advantageous, however, increasing the viscosity of the nanofluid reduces the fluid's heat transfer capability [9]. Notably, Pak and Cho [10] found that heat transfer performance was improved by selecting a larger size with higher thermal conductivity. Their study observed well-dispersed γ-Al_2O_3 and TiO_2 particles at pH values of 3 and 10, respectively. The Nusselt number increases with higher volume fraction and Reynolds number, indicating that thermal conductivity is the primary factor influencing heat transfer performance. Dayou et al. [11] experimentally compared the thermal performance of two nanofluids – namely multiwall carbon nanotube (MWCNT) and graphene nameplate (GnP). They found that GnP nanofluids have superior heat transfer coefficients compared to MWCNT nanofluids. In contrast, Ghazanfari et al. [12] numerically examined the mechanical limitations of twisted tubes and the use of (Al_2O_3, Cu, CuO, and TiO_2) nanofluids in heat exchangers. This study demonstrated that the implementation of twisted tubes and the use of nanofluids can improve the heat transfer performance of the heat exchanger. Another study showed increasing the nanoparticle volume concentration (Al_2O_3, CuO, TiO_2 and ZnO) enhances heat transfer and increase the pressure drop [13]. In this research, we employ numerical software such as ANSYS/Fluent® (i.e., ANSYS Fluent Release 15.0) and computational fluid dynamics (CFD) code [14] to discover the benefits of using nanofluid as a coolant to improve the thermal performance of double pipe heat exchanger in two different configurations, parallel and counter flow, under turbulent conditions.

2. Modelling
2.1. Geometry
A heat exchanger with a double pipe was previously modelled and validated [15] using the ANSYS/Fluent® CFD code (i.e., ANSYS Fluent Release 15.0), considering both inner and outer flows. All measurements were taken as per the experimental system [16]. The inner and outer copper tubing had an 8.13 mm and a 9.53 mm diameter, respectively. In contrast, the inner and outer PVC pipe had diameters of 27.8 mm and 33.9 mm, respectively. The test section, 1.5 m in length, allowed hot water flow in the annular region and nanofluid flow inside the tube, as shown in Figure 1.

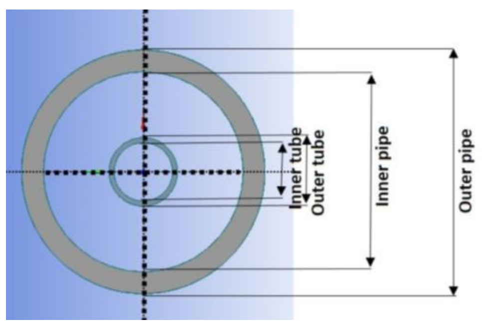

Fig. 1: Geometry design of double pipe heat exchanger.

2.2. Mesh Development
For both parallel and counter flow configurations, we created a model of the double pipe heat exchanger using a quadrilateral structured mapped mesh type (as shown in Figure 2). A smaller element mesh was required in some areas to capture crucial flow behaviour, such as the viscous and thermal boundary layers near the outer and the inner walls, with the progression of the grid spacing size employed as needed to avoid any artificial numerical effect. In connection with the current investigation, a sequence of grid refinements was carried out to verify that the outcome was grid-independent. The final mesh consisted of 106,555 nodes and 105,840 elements.

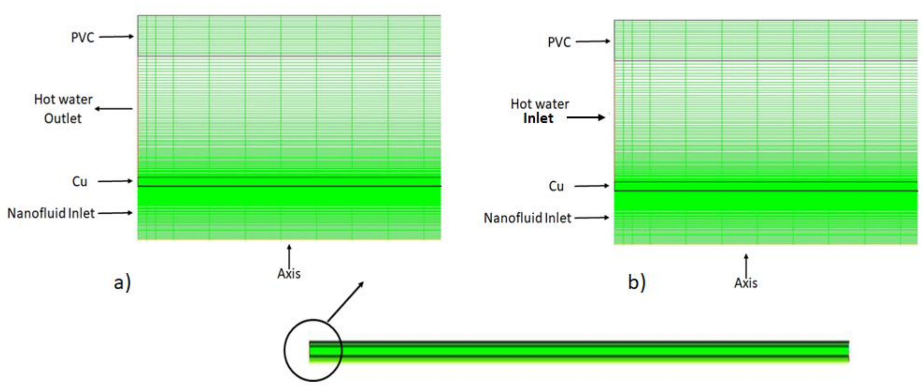

Fig. 2: Structured mapped mesh of double pipe heat exchanger a) Counter flow and b) Parallel flow.

2.3. Numerical Method

As for the double pipe heat exchanger's axisymmetric geometry, which was defined earlier. ANSYS/Fluent® CFD (i.e., ANSYS Fluent Release 15.0) is likely to be used to solve the governing partial differential equations in two dimensional, along with Shear Stress Transport (SST) k-ω turbulence model with a coupled explicit solver and determine the thermal flow field within the inner tube and annular region for both parallel and counterflow configurations. These calculations were made subject to the boundary conditions defined earlier [15]. The solver then ran until convergence, with residuals reaching 10^{-6} for mass, momentum and energy, ensuring no backflow occurred at any cell on the pressure outlet boundaries.

3. Data Reduction Equation

3.1. Thermal and Physical Properties of Nanofluids

To begin, after imposing the boundary conditions, the effective thermal properties need to be determined – such as specific heat, density, thermal conductivity and viscosity – at various temperatures for both nanofluids and hot water [17].

$$\rho_f = -3\times10^{-3} T_{avg}^2 + 1.505\, T_{avg} + 816.781 \tag{1}$$

$$C_{pf} = -4.63\times10^{-5} T_{avg}^3 + 0.0552\, T_{avg}^2 - 20.86\, T_{avg} + 6719.637 \tag{2}$$

$$\mu_f = 2.414\times10^{-5} \times 10^{247.8/(T_{avg}-140)} \tag{3}$$

$$k_f = 0.6067\left(-1.26523 + 3.704\left(\frac{T_{avg}}{298.15}\right) - 1.43955\left(\frac{T_{avg}}{298.15}\right)^2\right) \tag{4}$$

Once the thermal properties have been determined for the base fluid and hot water. Then, the thermophysical properties of nanofluids (pure water containing TiO_2 nanoparticles with a diameter of 21 nm) can be defined based on known thermophysical data and the volume concentration of TiO_2 nanoparticles.

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

$$Cp_{nf} = (1-\varphi)Cp_{bf} + \varphi Cp_p \tag{6}$$

The TiO_2/water nanofluid thermal conductivity and the viscosity can be approximated by the following:

$$\frac{k_{nf}}{k_{bf}} = 0.8938\,(1+\varphi)^{1.37}\,(1+\frac{T_{nf}}{70})^{0.2777}\,(1+\frac{d_p}{150})^{-0.0336}\,(\frac{\alpha_p}{\alpha_{bf}})^{0.01737} \tag{7}$$

$$\frac{\mu_{nf}}{\mu_{bf}} = (1+\varphi)^{11.3}(1+\frac{T_{nf}}{70})^{-0.038}(1+\frac{d_p}{170})^{-.0.061} \tag{8}$$

where Equations (7) and (8) are developed using the experimental data of various researchers by Sharma et al. [18] to estimate the thermal conductivity and viscosity of water-based nanofluids.

3.2. Heat Transfer Coefficient, Heat Transfer Rate and Nusselt Number

To evaluate the thermal performance of the double pipe heat exchangers and to better understand the physical processes involved in heat transfer, one needed to investigate parameters such as the heat transfer coefficient. The following are the main parameters that have a direct influence on the heat transfer process [19]:

$$h_{nf} = \frac{Q_{avg}}{A(T_{wall} - T_{nf})} \tag{9}$$

where $A = \pi D_{tube} L$ and Q_{avg} is the average heat transfer rate between hot water and nanofluid, which can be defined follows:

$$Q_{avg} = \frac{Q_w + Q_{nf}}{2} \tag{10}$$

The heat transfer rates from the heating fluid to the nanofluid can be calculated by the following:

$$Q_w = \dot{m}_w Cp_w (T_{in} - T_{out})_w \tag{11}$$

$$Q_{nf} = \dot{m}_{nf} Cp_{nf} (T_{out} - T_{in})_{nf} \tag{12}$$

Once the average heat transfer rate (Q_{avg}), wall temperature (T_{wall}) and nanofluids bulk temperature (T_{nf}) are defined:

$$T_{nf} = \frac{T_{in} + T_{out}}{2} \tag{13}$$

The convective heat transfer coefficient of the nanofluids can be calculated as well as the Nusselt number (Nu) using the following equation:

$$Nu_{nf} = \frac{h_{nf} D_{tube}}{k_{nf}} \tag{14}$$

4. Results and Discussion

The main objective of this study is to conduct a numerical investigation of the heat transfer coefficient for (water–TiO_2) nanofluids in two different configurations: parallel and counter flow. To initialize the numerical model and estimate the value of the heat transfer coefficient of the nanofluid, thermophysical properties of the nanofluid must be determined as can be seen from Figure 3. The results reveal that by increasing the volume concentration of nanoparticles the specific heat of the nanofluid decreases, while its thermal conductivity increases. This is because of the lower specific heat and higher thermal conductivity of the nanoparticles compared to pure water. Furthermore, the viscosity of the nanofluids increases with higher volume fractions due to the random Brownian motions of nanoparticles [20], which interrupt the moving of fluid particles and have a tendency to increase the viscosity of the entire system. The numerical model's findings are discussed in the subsection that follows.

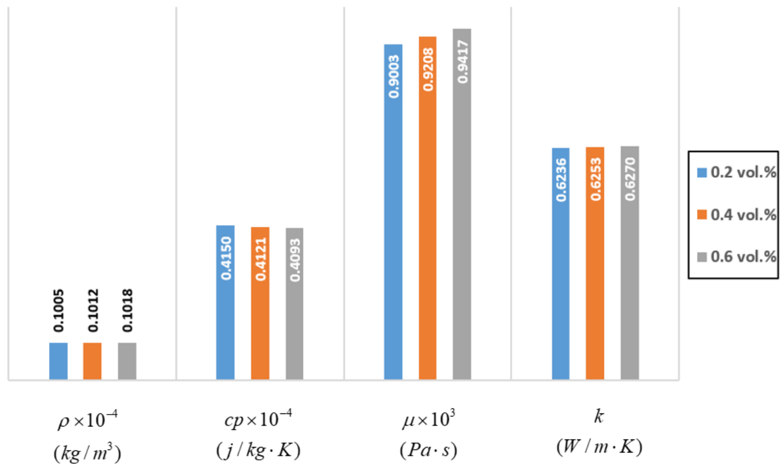

Fig. 3: Effective thermophysical parameters of the nanofluid @ 298.15K.

4.1. Nusselt number, Heat Transfer Coefficient and Heat Transfer Rate

Since the numerical model was previously validated [15] against experimental work conducted by Duangthongsuk et al. [16], this study will present the numerical result of the Nusselt number variation for both parallel and counter flow configurations against the Reynolds number, as shown in Figure 4.

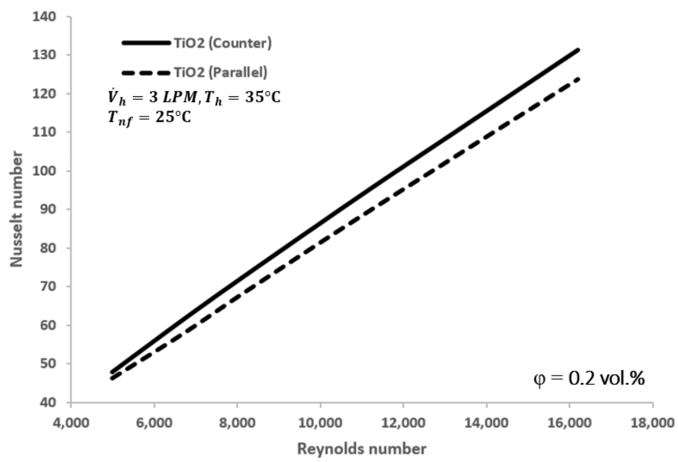

Fig. 4: The variation of Nusselt's number versus the Reynolds number for both directions

Figure 4 illustrates the comparison of the nanofluid Nusselt number containing 0.2% volume of TiO_2 nanoparticles dispersed in water for both parallel and counter-flow configurations. This figure demonstrates that the Nusselt number increases as the Reynolds number rises in both configurations. The numerical results revealed a 6.1% elevation in the nanofluid Nusselt number for the counter flow compared to the parallel flow at a higher Reynolds number (Re=16200). Heat transfer performance improves with a low volume concentration of 0.2% volume of TiO_2 nanoparticles dispersed in water. This study will be extended to explore convective heat transfer coefficients for volume concentrations of 0.4% and 0.6% TiO_2 nanoparticles dispersed in water, considering both parallel and counter flow configurations at higher Reynolds numbers (as shown in Figure 6).

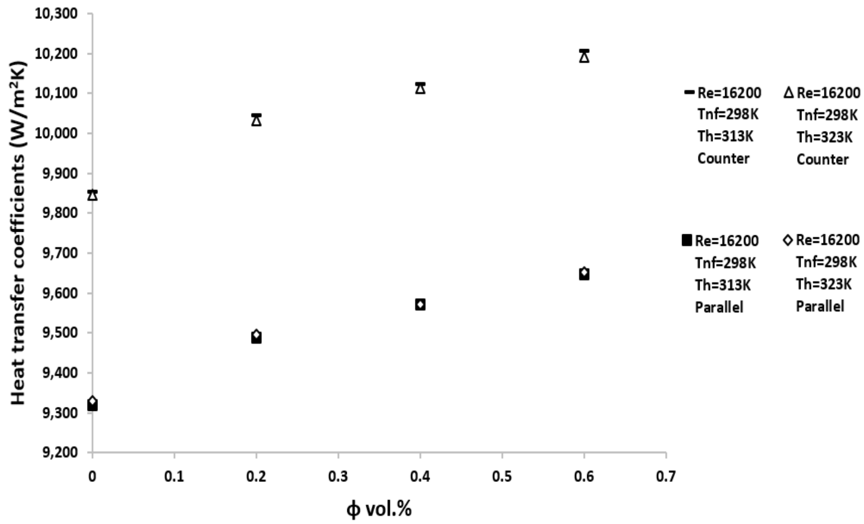

Fig. 5: Comparison of heat transfer coefficients with different volume concentrations for counter and parallel flow direction.

Figure 5 illustrates that increasing the volume concentrations of TiO_2/water nanofluids improves the heat transfer coefficient for both parallel and counter flow configurations compared to pure water. Specifically, at a Reynolds number of 16,200, the heat transfer coefficients for counter flow and parallel flow are found to be ≈ 1.9%, 2.7% and 3.5% for volume concentrations of 0.2%, 0.4% and 0.6%, respectively. Interestingly, increasing the annular temperature had no significant impact on the convective heat transfer coefficient of nanofluids for both flow directions. However, as demonstrated in Figure 7, these variations can only be seen for heat transfer rates at the same operating conditions.

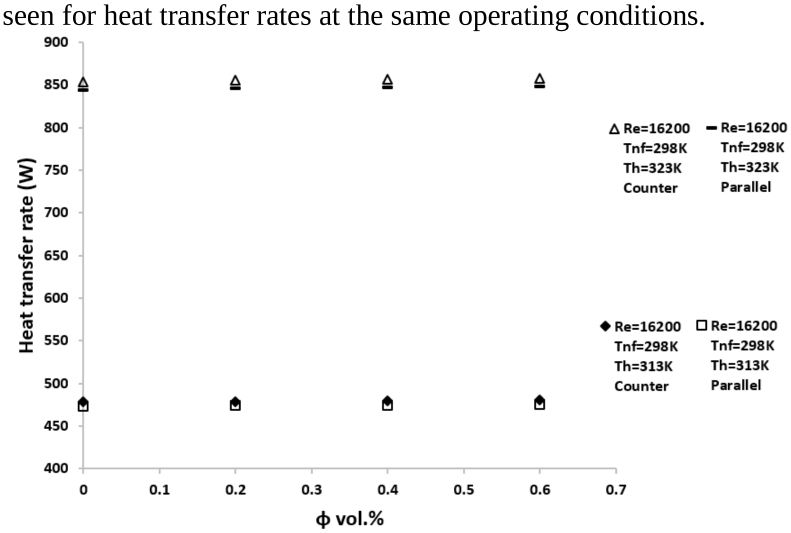

Fig. 6: Comparison of heat transfer rate with different volume concentrations for counter and parallel flow directions.

Figure 6 reveals that increasing the annular temperature will only increase the heat transfer rate. Therefore, increasing the heat transfer rate has no significant impact on the nanofluid convective heat transfer coefficient. Equation (9) highlights that multiple parameters have a direct influence on the convective heat transfer coefficient. Furthermore, the fluid direction has no major effect on the heat transfer rate compared to the convective heat transfer coefficient. In summary, counter flow proves significantly more efficient than parallel flow for heat transfer performance

4. Conclusion

In this research, a numerical investigation was conducted to study the effect of using nanofluid as a cooling medium instead of pure fluid on the performance of parallel and counter flow double pipe heat exchangers. From the results, the following conclusions can be drawn:

1. Heat performance, as represented by the Nusselt number, increases with an increasing nanofluid Reynolds number for both counter flow and parallel flow double pipe heat exchangers.
2. Increasing the volume fraction of TiO_2-water nanofluid enhances nanofluid thermal conductivity, leading to higher nanofluid convective heat transfer coefficients in both counter flow and parallel flow double pipe heat exchangers.
3. While the enhancement of the heat transfer rate is associated with increasing the annular temperature (hot water), it has no significant impact on the nanofluid convective heat transfer coefficient for either flow configuration.
4. Flow direction significantly enhances the convective heat transfer coefficient, favouring counter flow over parallel flow. However, it has no significant effect on the heat transfer rate.
5. Overall, counter flow significantly more efficient than parallel flow for heat transfer performance.

Nomenclature

A	heat transfer surface area m^2	**Subscript**	
Cp	specific heat, $\mathrm{J/kg}$ K	ave	average
d	nanoparticle diameter, m	bf	base fluid
D	tube pipe diameter, m	f	Fluid
h	heat transfer coefficient, $\mathrm{W/m^2K}$	h	hot fluid
k	thermal conductivity, $\mathrm{W/mK}$	in	inlet
L	length of the test tube, m	out	outlet
$\dot{m}$	mass flow rate, kg/s	p	particles
Nu	Nusselt number	nf	nanofluid
Q	heat transfer rate, W	w	water
Re	Reynolds number	$wall$	tube wall
T	temperature, C	**Greek symbols**	
$\dot{V}_h$	hot water flow rate, (LPM) litres per minute	$\emptyset$	volume fraction
		ρ	density, $\mathrm{kg/m^3}$
		α	thermal diffusivity, $\mathrm{m^2}/s$
		μ	viscosity, $\mathrm{kg/ms}$

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