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Thermal Performance Enhancement of a Corrugated Heat Exchanger Using Al_2O_3–Water Nanofluid: A CFD Study

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ABSTRACT

This research deals with a numerical study of turbulent convective flow of a nanofluid (Al_2O_3-water) circulating in a counter-current bi-tubular heat exchanger. The studied exchanger is characterised by a new design of the corrugated inner tube in elliptical shape, intended to optimize heat exchange. The aim of this research is to predict the influence of the thermophysical properties of the nanofluid and the geometric form of the inner tube on the exchanger's thermal performance. For this purpose, bidimensional numerical simulation was carried out using CFD code, with the implementation of the k-ε turbulence model. The regime examined corresponds to a range of Reynolds numbers varying between 5000 and 11000. The validity of the numerical results was verified by comparison with experimental data published by Sarmad. The obtained results reveal a significant improvement in Nusselt numbers at high nanoparticle concentrations. More specifically, the corrugated heat exchanger is more efficient than the conventional smooth-tube model, with a 33.6% increase in the Nusselt number. In addition, the adoption of geometry with a corrugated inner tube contributes to a significant improvement in the overall heat transfer coefficient, reaching values between 682.6 and 1148 $W/(m^2 \cdot K)$ for a volume fraction of 4% in Al_2O_3.

1. Introduction

Rapid technological progress makes energy efficiency crucial in all sectors. Improving the performance of energy-consuming systems and equipment is indispensable to ensure more efficient use of energy. For this purpose, thermal energy is a fundamental driver of the modern world, highlighting the importance of its optimization [1]. This optimization requires, in particular, a thorough understanding of heat transfer. As this phenomenon represents a key process in many engineering applications. Its classification depends on the types of fluids used, the intended application, the flow mode and the heat transfer mechanisms [2]. Heat exchangers are the preferred technical solutions for efficiently managing this process by transferring heat between two fluids at different temperatures[3].There are many types of heat exchanger technologies used in industry, The most common types include plate heat exchangers (PHEs), shell and tube heat exchangers, vertical shell heat exchangers, and micro heat exchangers [4] ,these devices play a crucial role in the

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functioning of thermal systems such as nuclear reactors, power plants, heat recovery units, and processing industries[5]. Chemical industry, oil refineries, agri-food industry[6]

The double-tube heat exchanger is a commonly adopted solution for improving thermal performance. Its design, which allows two fluids to circulate simultaneously, one in the inner tube and the other in the concentric annular space, it is characterized by its simplicity, robustness and efficiency[7]. These characteristics make it a suitable device for achieving the main objective of heat exchanger and thermal system design, involving the improvement of heat transfer. To achieve this goal, several studies have focused on strategies that optimize flow rate and transfer efficiency while reducing device size[8]. These studies have explored the integration of flow-modifying components, such as vortex generators[9], [10], fins[11]and baffles[12],[13], [14], [15] and turbulators [16]. Other approaches include the use of porous media in heat exchangers[17], [18] or the introduction of nanofluids as an additional means of improvement.[19], [20], [21].

The nanofluids can be defined as suspensions of nanoparticles (with an average size generally less than 100 nanometers) dispersed in conventional heat transfer fluids such as water, ethylene glycol or oil. This fundamental concept of improving the thermal conductivity of fluids by adding metallic particles dates back to Maxwell [6]. Accordingly, the integration of nanofluids into heat exchangers represents a preferred approach: their superior thermophysical properties compared to pure fluids significantly improve the thermal performance factor, making this technique an effective solution for heat transfer [8].The scientific literature deals extensively with nanofluids, but few studies combine their use in turbulent flow with elliptical corrugated tube designs. This study aims to fill this gap by examining the thermal performance of this specific configuration applied to counter-current double-tube heat exchangers.

In this context, the present study focuses on the numerical study of convective flow of a nanofluid composed of alumina (Al_2O_3) nanoparticles dispersed in water, circulating in a double-tube heat exchanger which is characterized by a corrugated elliptical architecture.

The main objective is to evaluate the combined influence of the nanofluid's thermophysical properties and the modified geometry on the heat transfer and flow dynamics in the exchanger, operating in countercurrent mode.

2. Geometric model

The physical model used is illustrated in Figure 1. It is a double-tube heat exchanger, consisting of a smooth outer tube and a corrugated inner tube. The dimensions characterising this configuration are: tube length L = 2 m, inner diameter D_{in} = 22 mm, external diameter D_{ex} = 61 mm, corrugation pitch p = 34 cm.

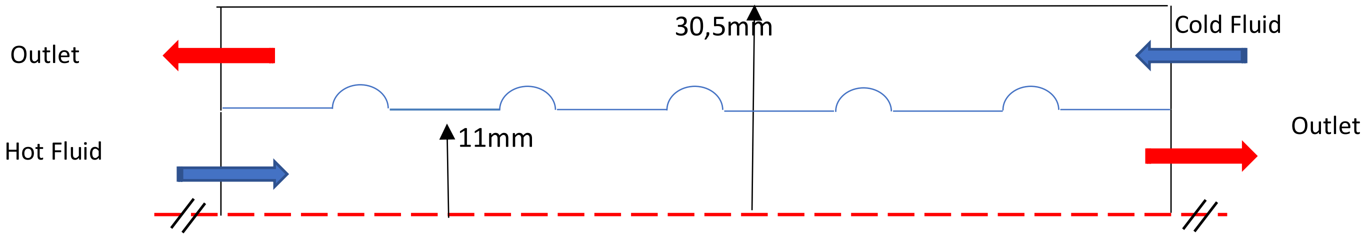

Fig. 1. Schematic of the heat exchanger studied

The corrugated tube has an elliptical cross-section (figure 2), with a semi-major axis a=20mm and a semi-minor axis b=2.5mm. The heat transfer fluid (hot water) circulates in the inner tube, while the cold fluid (nanofluid) flows in the annular space.

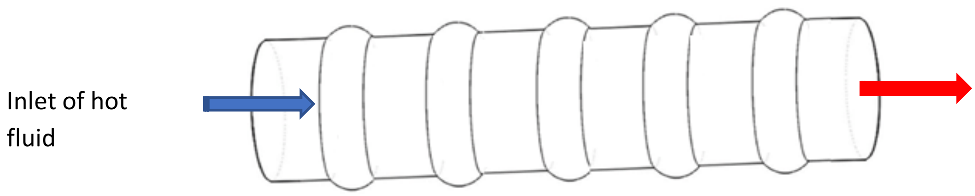

Fig. 2. Internal tube of the heat exchanger studied

3. Boundary conditions

The boundary conditions applied to the calculation domain are as follows: The inlets of the two heat exchanger tubes are characterized by imposed velocity and temperature conditions. The hot fluid enters at a temperature of 333 K with a variable velocity corresponding to Reynolds numbers ranging from 5000 to 11000. The cold fluid is introduced at 298 K, with its inlet velocity adjusted to cover the same range of Reynolds numbers. At the outlets, atmospheric pressure is imposed. An axial symmetry condition was used for the inner tube, and the outer wall of the external tube is treated as adiabatic.

4. Hypotheses

To solve the transport equations, our numerical simulations were performed based on the following simplifying assumptions. The flow is considered steady. The fluid is treated as incompressible and isotropic. The thermophysical properties (ρ, C_p, λ, μ) are assumed to be constant throughout the study domain.

5. Governing equation

The generalized transport equation is given by [22]:

$$div\left(\rho U\phi\right)=\frac{\partial}{\partial x}\left(\Gamma\frac{\partial\phi}{\partial x}\right)+\frac{\partial}{\partial y}\left(\Gamma\frac{\partial\phi}{\partial y}\right)+\frac{\partial}{\partial z}\left(\Gamma\frac{\partial\phi}{\partial z}\right) \quad (1)$$

Where ρ represents the fluid density, U the fluid velocity vector, and ϕ the generalized diffusion coefficient. In the context of the governing equations: for the continuity equation, $\phi = 1$; for the momentum equation, ϕ represents the velocity vector U; and for the energy equation, $\phi = T$, where *T* is the temperature.

6. Turbulence model

To simulate the flow, the k–ε turbulence model is used, a standard method in CFD that approximates turbulence through two governing equations. These equations respectively describe the turbulent kinetic energy and the turbulent dissipation rate [23]

$$\rho\left(U.\nabla\right)k = \nabla.\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\nabla k\right] + P_k - \rho\varepsilon \quad (2)$$

$$\rho\left(U.\nabla\right)\varepsilon = \nabla.\left[\left(\mu + \frac{\mu_t}{\sigma_e}\right)\nabla \varepsilon\right] + C_{e1}\frac{\varepsilon}{k}P_k - C_{e2}\rho\frac{\varepsilon^2}{k} \quad (3)$$

Where $C_{e1} = 1.44, C_{e2} = 1.92, \sigma_k = 1.0$ and $\sigma_e = 1.3$

The term P_k represents the generation of turbulence kinetic energy

7. Nanofluid properties

The density and specific heat of the nanofluid are evaluated using the following formulas: [22]

$$\rho_{nf} = \left(1-\varphi\right)\rho_f + \varphi\ \rho_p \quad (4)$$

$$Cp_{nf} = \frac{\left(1-\varphi\right)\rho_f Cp_f + \varphi\ \rho_p Cp_p}{\left(1-\varphi\right)\rho_f + \varphi\ \rho_p} \quad (5)$$

The following equation, which is based on certain experimental relationships, can be used to determine the thermal conductivity of nanofluids[23].

$$\lambda_{nf} = \lambda_f\left[\frac{\lambda_p + \left(n-1\right)\lambda_f - \left(n-1\right)\varphi\ \left(\lambda_f - \lambda_p\right)}{\lambda_p + \left(n-1\right)\lambda_f + \varphi\ \left(\lambda_f - \lambda_p\right)}\right] \quad (6)$$

To calculate the dynamic viscosity, we use the flowing formula [24]

$$\mu_{nf} = \left(1 + 7,3\varphi + 123\varphi^2\right)\ \mu_f$$

(7)

The thermophysical properties of the Nanoparticles and Water used in this simulation are given in Table 1.

Table 1 Thermophysical properties of Nanoparticles and Water [24], [25]

Medium	Density [Kg/m^3]	Specific Heat [J/(kg.K)]	Thermal conductivity [W/(m.K)]	Dynamic viscosity [Kg/(m.s)]
Al_2O_3	3970	765	40	
Pure water (20°C)	998	4182	0.598	0.001002
Pure water (60°C)	983,3	4185	0.654	0.000467

8. Calculations

The intensity of heat exchange between the solid surface and the circulating nanofluid is quantified by the local convective heat transfer coefficient *h (x),* whose value is determined by the formula: [26]

$$h(x) = \frac{\varphi_{th}}{(T_w - T_\infty)} \tag{8}$$

Where T_w and T_∞ represent the temperatures of the wall and the bulk fluid, respectively.

To determine the average heat transfer coefficient $\bar{h}$, the local heat transfer coefficient must be integrated over the entire length of the exchanger L, which gives: [27]

$$\bar{h} = \frac{1}{L}\int_0^L h(x)\,dx \tag{9}$$

The average Nusselt number transfer Nu is calculated using the following expression:

$$\overline{Nu} = \frac{\bar{h}.D_{hy}}{\lambda} \tag{10}$$

9. Numerical resolution

The conservation equations were solved using computational fluid dynamics code, which is based on the finite volume method. The turbulence model used in this simulation is $k-\varepsilon$. Additionally, the coupling between the pressure and velocity terms was performed using the SIMPLE algorithm. The momentum and energy equations were discretized with a second-order upwind scheme, extrapolation of the pressure has been performed using the standard method.

10. Results and discussions

10.1 Thermal validation

In order to validate our numerical simulation, we performed a comparative analysis between the values of average Nusselt number (Nu) obtained and the experimental data [3]. These results were obtained using a smooth double-tube heat exchanger. Water was used as the working fluid for both the hot and cold sides. The hot fluid circulated in the annular space, while the cold fluid flowed in the inner tube. The inlet conditions were set at 60°C for the hot fluid and 20°C for the cold fluid. The mass flow rate of the cold fluid was varied between 0.06 kg/s and 0.18 kg/s. As shown in Figure 3.

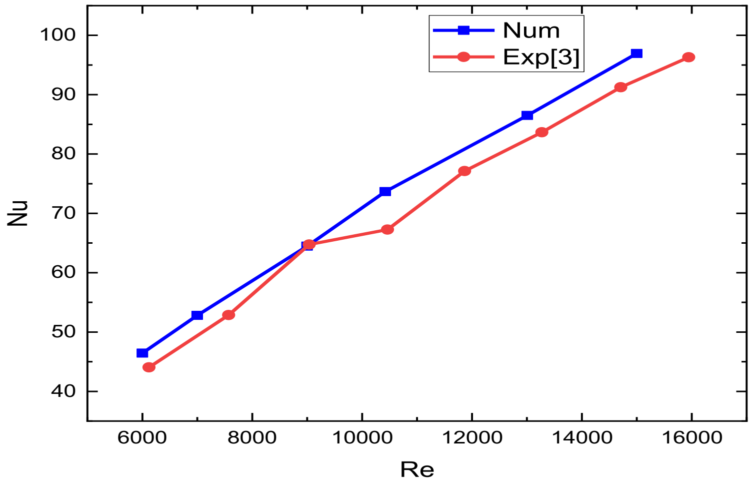

Fig. 3. Validation of results

The results of our current numerical simulations are in strong agreement with those reported by Sarmad [3], with a slight error of 4.24 %. The slight difference observed between the numerical and experimental results may be due to the turbulence model chosen or the accuracy of the numerical scheme used for our calculation.

Parametric study:

A comparison of Nusselt numbers was conducted for two heat exchangers: one with a smooth configuration and the other with a corrugated configuration. As shown in Figure 4,

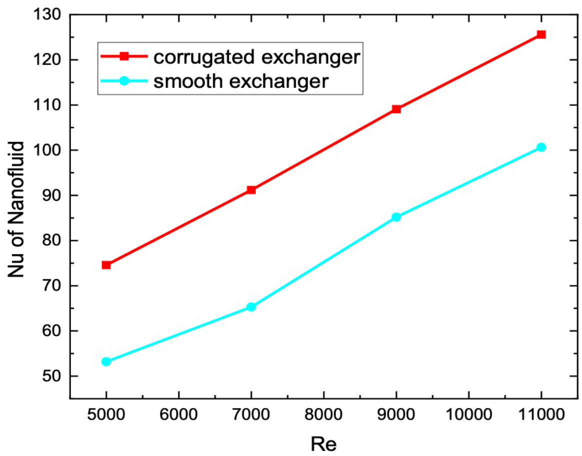

Fig 4. Evolution of the Nusselt number as a function of the Reynolds number for two heat exchangers, with corrugation and without corrugation (ϕ =1%)

A comparative analysis reveals that the use of a corrugated heat exchanger leads to a sustained 33.6% increase in the Nusselt number relative to a conventional smooth tube exchanger.

This improvement in heat transfer is principally due to the increased turbulence and better mixing induced by the corrugations, which leads to more efficient convective heat transfer. This impact increases with increasing Reynolds number of the nanofluid, which indicates that the corrugated geometry is highly efficient.

As corrugated heat exchangers provide better performance than their non-corrugated counterparts, the results presented in the section below will focus on analysing the results obtained for the corrugated heat exchanger.

In Figure 5, we plotted the Nusselt number variation as a function of volume fraction for Reynolds number values ranging from 5000 to 11000 with a step of 2000.

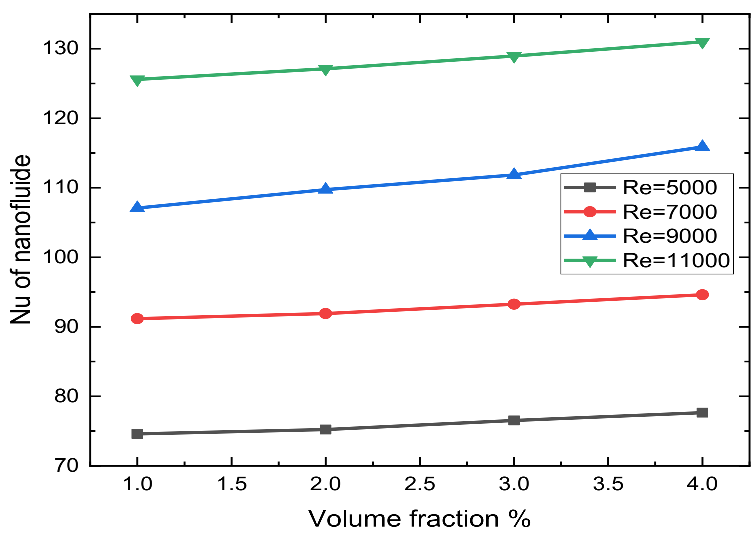

Fig. 5. Influence of the Reynolds number on the Nusselt number variation as a function of the volume fraction

From this figure, we can clearly see that an increase in the volume fraction leads to an increase in the Nusselt number. This tendency is observed whatever the value of the Reynolds number chosen. This result is logical, as the addition of nanoparticles increases the nanofluid's thermal conductivity, favouring better heat transfer, which directly translates into an increase in the Nusselt number.

On the other hand, it can be seen that the increase in the Nusselt number (Nu) is accompanied by an increase in the Reynolds number (Re). This is due to the increased turbulence in the flow. Indeed, a higher Reynolds number characterises a more turbulent regime, which favours the mixing of fluids. This phenomenon intensifies heat exchange by convection between the surface of the exchanger and the nanofluid, and consequently the heat flux and the Nusselt number.

The influence of the volume fraction on the convective transfer coefficient variation of the nanofluid (Al_2O_3-water) as a function of the Nusselt number is shown in Figure 6.

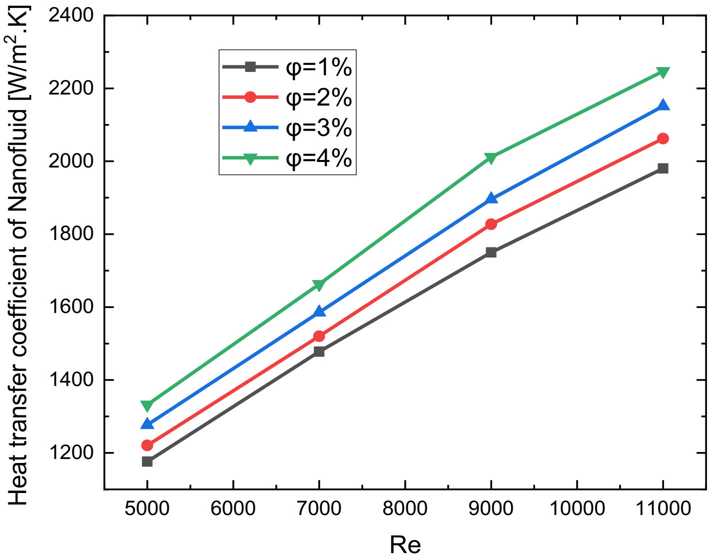

Fig. 6. Influence of the volume fraction on the convective transfer coefficient variation for various Reynolds numbers

It is clear from this figure that the convective transfer coefficient of the Al_2O_3-water nanofluid increases when the Reynolds number increases, specifically for a volume fraction between 1 and 4%. This increase is physically expected and it can be attributed to the direct dependence of the convective transfer coefficient on the Nusselt number, which in turn depends on the Reynolds number.

Furthermore, it can be seen that the volume fraction of the nanofluid has a significant effect on the variation of the convective transfer, i.e. any increase in the volume fraction leads to an increase in the coefficient of the nanofluid's convective transfer. This increase is due to the improved thermal connectivity of the nanofluid. Indeed, the Al_2O_3 has a much higher thermal conductivity than water. Dispersing these nanoparticles in water creates a nanofluid whose thermal conductivity is greater than that of pure water. This allows us to increase heat transfer and consequently, the convective transfer coefficient.

Figure 7 demonstrates the temperature variation of the hot fluid (water) as a function of the axial position (x) for a Reynolds number ranging from 5000 to 11000 with a constant volume fraction of 4%.

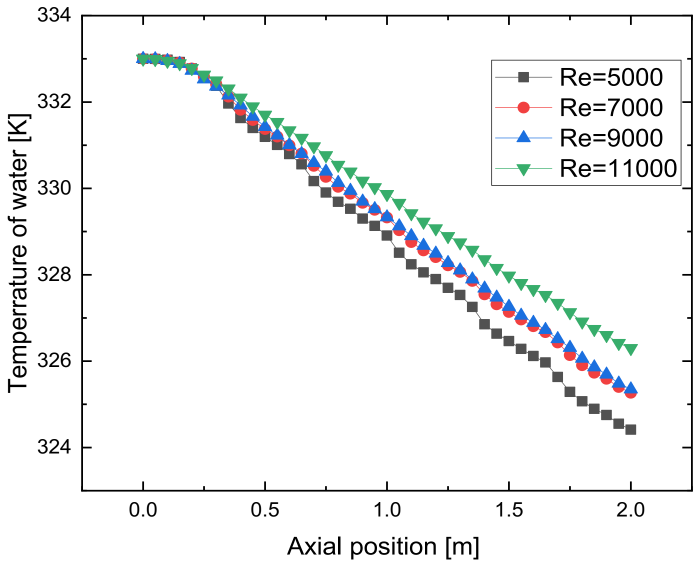

Fig. 7. Effect of Reynolds number on the water temperature variation as a function of axial position. (ϕ=4%)

This figure reveals that all the curves have similar shapes and that the water temperature decreases continuously along the axial length of this exchanger. The reason for a decrease is that the hot fluid, water, transfers its thermal energy to the cold fluid with which it is exchanged. This heat transfer occurs along its entire length, causing a gradual decrease in its temperature. On the other hand, we can observe that the temperature differences of the hot fluid between the inlet (x=0) and the outlet (x=2m) decrease with increasing Reynolds number. This is perfectly logical, as an increase in Reynolds number leads to a higher fluid flow velocity, which means that the hot fluid remains less time in the heat exchanger to transfer its heat to the cold fluid. It therefore has less time to cool down, causing a decrease in the temperature difference between the inlet and outlet.

In Figure 8, we have shown the Reynolds number effect on the temperature variation of the cold fluid (nanofluid) along the length of the exchanger.

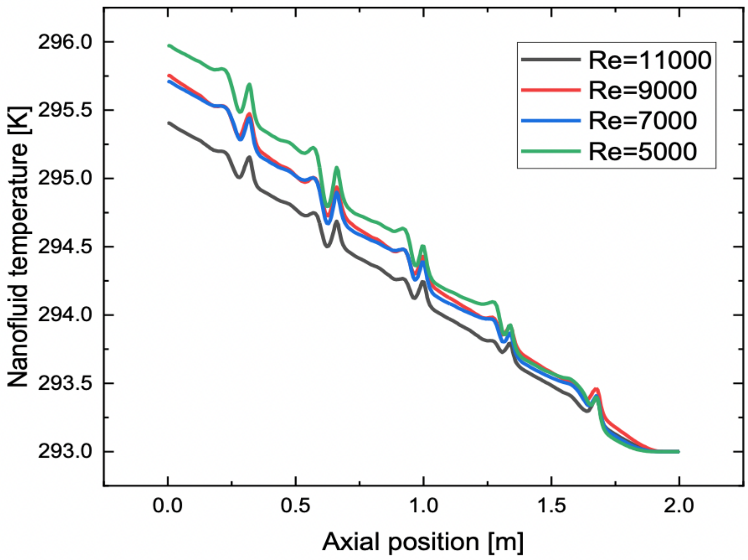

Fig. 8. Influence of Reynolds number on the variation of nanofluid temperature as a function of axial position (ϕ=4%)

From this figure, it is evident that any increase in the Reynolds number leads to a decrease in the temperature difference of the nanofluid. This is perfectly normal, because an increase in the Reynolds number corresponds to an increase in the nanofluid's flow velocity, which reduces the time the fluid remains in the heat exchanger. Consequently, the cold fluid has less time to absorb heat, which limits the increase in its temperature. On the other hand, a more turbulent flow improves instantaneous heat transfer, but the fluid's average temperature varies less along the exchanger.

The influence of the Reynolds number on the variation of the overall convective transfer coefficient as a function of the volume fraction is presented in Figure 9.

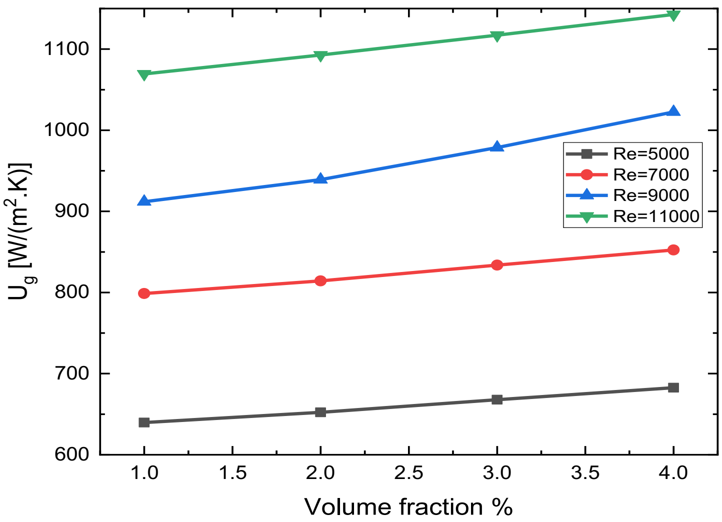

Fig. 9. Evolution of the overall convective transfer coefficient as a function of the volume fraction for various values of the Reynolds number

It can be seen from this figure that the Reynolds number has a very significant influence on the variation of the overall heat transfer coefficient as a function of the volume fraction, which varies from 1 to 4%. Indeed, for instance, at a Reynolds number of 5000 and with a volume fraction ranging from 1% to 4%, the overall heat transfer coefficient values range between 639.6 and 682.61 W/(m²·K). However, increasing the Reynolds number to 11000 significantly enhances these values, reaching 1069 to 1148 W/(m²·K). This gain of approximately 68.17% confirms that the interaction between the corrugated exchanger configuration and the thermal conductivity of Al_2O_3 is optimized when inertial forces dominate the flow. Indeed, this high increase can be explained by the intensification of turbulence in the flow when the Reynolds number increases. Greater turbulence favours mixing of the fluid, reduces the thermal boundary layer. This leads to improved heat exchange between the two fluids and, consequently, a considerable increase in the overall heat transfer coefficient.

Figure 10 shows the graphs illustrating the volume fraction's impact on the dynamic pressure variation of the cold fluid with radius.

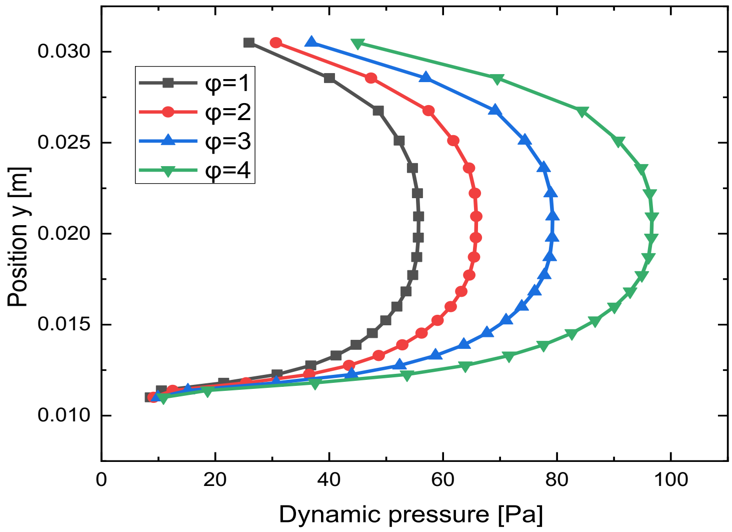

Fig. 10. Impact of the volume fraction on the dynamic pressure variation of the cold fluid along the radius (Re=11000)

In this figure, we can see that the various curves have a similar behavior, characterized by a horizontal parabolic shape It is also clear that increasing the nanofluid's volume fraction leads to an increase in dynamic pressure.

This phenomenon can be explained by the increase in the effective density of the nanofluid as the volume fraction of nanoparticles increases. Indeed, the addition of solid particles to the base fluid has the effect of increasing the effective density of the nanofluid, which leads to an increase in dynamic pressure, in accordance with the relationship $P_{dy} = \frac{1}{2}\rho V^2$.

11. Conclusions

In this study, the thermal and dynamic behavior of nanofluids in a double-tube heat exchanger is simulated numerically using CFD code. The aim of this study is the influence study of nanofluid (Al_2O_3-water) on the performance of a heat exchanger, which is configured with a corrugated inner tube and operates in counter-current mode.

The numerical results obtained and validated by experimental data demonstrate that the use of an Al_2O_3-water nanofluid, with a corrugated inner tube, led to a significant improvement in heat transfer, as evidenced by the increase in the Nusselt number and the overall heat transfer coefficient. In addition, the corrugated elliptical configuration induces stronger flow mixing and boundary layer destabilization, which contributes to enhanced convective heat transfer performance. These results also demonstrate that parameters such as the volume fraction of nanofluids and the Reynolds number have a strong influence on the efficiency of the heat exchanger. Furthermore, they prove that these techniques are effective in designing more efficient heat exchangers and pave the way for new research into other types of nanofluids or innovative geometric shapes.

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