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Numerical Investigation of Heat Transfer Enhancement in A Heat Exchanger with Sinusoidal Tubes

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Abstract: Recent research has extensively investigated the rheological and thermal behaviors of nanofluids. The present work provides a detailed numerical investigation of heat transfer and pressure drop characteristics in sinusoidal and straight tubes using two types of nanofluids: water-based and ethylene glycol-water-based suspensions with nanoparticles. The simulations were conducted under laminar flow conditions using ANSYS CFX software with a constant and uniform heat flux boundary condition. Different nanoparticle sizes and concentrations were examined to evaluate their impact on heat transfer enhancement. The research gap addressed in this work lies in the limited number of studies that have simultaneously integrated sinusoidal tube geometry and nanofluid properties to enhance heat exchanger performance. This paper bridges that gap by numerically comparing the heat transfer characteristics of sinusoidal tubes against traditional straight tubes under identical operating conditions.

The results demonstrate that increasing Reynolds number and nanoparticle concentration significantly enhances the heat transfer coefficient. Overall, sinusoidal tubes combined with nanofluids achieved up to 25–30% improvement in the Nusselt number compared to straight tubes, highlighting their potential for compact and energy-efficient heat exchangers.

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Keywords: Sinusoidal tubes; Reynolds number; Heat transfer; Nanofluid

1. Introduction

Heat exchangers are devices that transfer heat energy between two or more fluids at different temperatures and are responsible for heat exchange. These devices are used in power generation, process industries, chemical and food industries, electronic equipment, manufacturing industries, air conditioning, refrigeration and space applications [1]. In the transportation industry, cooling is a critical issue, because the upcoming trend in increasing the power of engines or hybrid vehicles inevitably requires the use of larger radiators and, as a result, a larger frontal surface, which increases drag and higher fuel consumption. Although extensive research has been devoted to improving heat exchangers, most studies have focused on conventional geometries or single-phase fluids. The challenge remains to develop configurations that can enhance heat transfer without imposing excessive pressure drop.

Attention to saving materials, space, energy and the global economy has led to the expansion of efforts to produce more efficient heat exchanger equipment in order to reduce costs. The result of these efforts is the reduction of the physical dimensions of these equipment for a given thermal capacity. Therefore, the main goals of thermal-hydraulics are to reduce the dimensions of a heat exchanger required for a given thermal capacity, increase the capacity and performance of an existing heat exchanger with a smaller temperature difference or reduce pump power [1]. This is especially important in compact and micro-scale systems, where the balance between heat transfer enhancement and pressure drop penalties dictates overall performance [2].

Yuan et al. [2] conducted a numerical study of heat transfer in a rectangular microchannel with a channel surface with ridges and depressions, and finally showed that creating ridges and depressions on the surface of microchannels would be a practical and suitable technique for improving heat transfer. Ligrani et al. [3] experimentally studied the flow structure characteristics in a channel with a depression surface on one wall, a ridge on the wall and no ridges on the opposite wall, showing that the ridges lead to the addition of vertical, secondary flow structures and flow mixing. Then they immediately reported the dynamic flow structure characteristics and the time-averaged flow structure characteristics generated. Xie et al. [4] conducted a numerical study on the flow and heat transfer in a rectangular channel with teardrop-shaped ridges and depressions with different distances from the center. They showed that the flow simply covers the teardrop surface and connects to the backside with a lot of energy on the ridges and depressions. Zheng et al. [5] numerically investigated the heat transfer and flow characteristics of water-Al2O3 nanofluid using a curved strip with depressions and protrusions numerically for only one step and one type of geometric matrix and showed that it would improve heat transfer.

Ammar Ali et al. [6] conducted numerical simulations of the pressure drop and heat transfer characteristics of smooth tubes and internally spiral microfinned tubes with two different fin height ratios and found that the heat transfer performance can be improved by using spiral finned tubes and optimization. Heat transfer can be achieved by optimizing the fin height ratio. Lotfi et al. [7] numerically investigated the forced convection heat transfer of nanofluids with two single-phase and two-phase models in laminar and turbulent flow. They passed water/aluminum oxide nanofluids into a horizontal tube. They compared and validated the results with existing relationships. They also compared the results of both models. By comparing the experimental results and the models, it was found that the mixed model is more accurate. Both models underestimate the Nusselt number. Their results showed that the heat recovery rate decreases with increasing volumetric concentration of nanoparticles.

In addition to these studies, other researchers have also explored innovative methods for improving compact heat exchanger design. For example, Cho [8] highlighted the importance of microchannel and mini-channel designs for high-performance heat sinks in automotive and electronic cooling applications. Shah and Sekulić [9] provided a comprehensive review of compact heat exchanger configurations, emphasizing the trade-offs between surface compactness and flow resistance. In another investigation, Kakac and Liu [10] addressed the role of nanofluids in enhancing thermal conductivity and discussed their limitations due to increased viscosity and stability issues. Recent studies, such as those by Ahmad et al. [11], have considered topology optimization for advanced fin and channel designs, showing that material distribution can be tailored for optimal thermal-hydraulic performance. Similarly, Bergles [12] developed correlations for enhanced surfaces and demonstrated their effectiveness in predicting heat transfer coefficients in complex geometries. These contributions indicate a growing emphasis on integrating advanced surface engineering, multiphase fluids, and computational optimization techniques to design the next generation of compact, energy-efficient heat exchangers.

This paper is organized as follows. Section 2 describes the thermophysical properties of nanofluids. Section 3 introduces the proposed numerical method, including governing equations and boundary conditions. Section 4 presents the simulation results and comparative analysis, followed by conclusions in Section 5.

2. Numerical simulation

In recent years, with the help of fast computers, engineers have been able to perform significant numerical calculations to predict engineering phenomena and improve design processes. The use of numerical methods is one of the capabilities to preview the solution of a problem that has not yet reached the experimental stage and the other to improve existing experimental results. Computational fluid dynamics (CFD) has become a standard method for the design and analysis of engineering-related problems involving multiphysics phenomena. A large number of studies have been carried out to solve fluid mechanics and heat transfer problems of conventional fluids that follow the basic laws published in the literature, and most of them are consistent with the experimental results of the same problem. However, with the introduction of nanofluids in the field of fluid mechanics and heat transfer, a new path has been opened with its benefits and challenges. Methods very similar to those used in conventional computational methods can be adapted for use in engineering industries related to nanofluids.

First, the geometries are explained. Then, the governing equations, properties of nanoparticles, base fluid, relations used, and solution method are presented. Also, the geometry meshing, evaluation, and sensitivity analysis to meshing are presented. Finally, the results are validated with the available data.

A straight pipe and a sinusoidal pipe are used in this study. Figure 1 shows a view of a quarter of a straight pipe. The geometry of the sinusoidal pipe is generated from its surface profile, which is given by equation 1.

$$y = a\sin\left(\frac{2\pi}{\lambda}x - \frac{3\pi}{2}\right) + \frac{D_{max} + D_{min}}{4} \quad (1)$$

Which is the distance of the sinusoidal surface profile from the center of the pipe. Table 1 gives the dimensions of the two sinusoidal pipe modes, the parameters of which are specified in Figure 2.

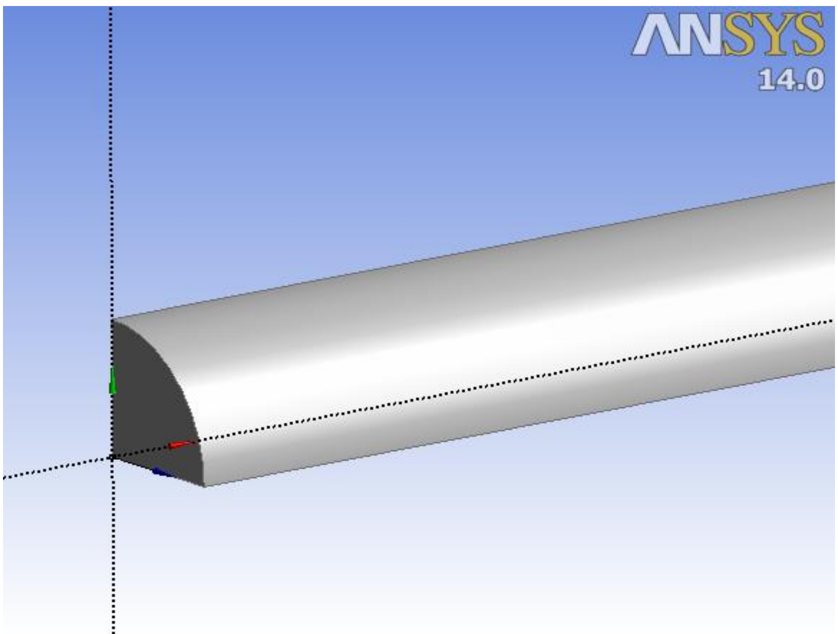

Figure 1. View of a straight pipe

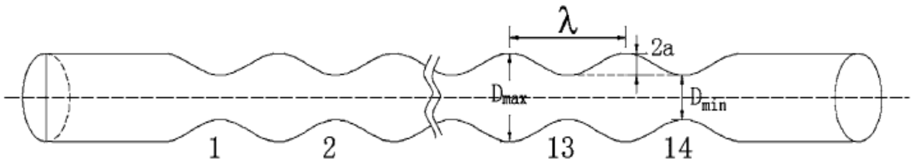

Figure 2. Sinusoidal tube geometry parameters

Table 1: Dimensions of sinus tubes used in the present study

	D_{max}(mm)	D_{min}(mm)	2a(mm)	$\lambda(mm)$	F_W	Pipe length
Case 1	10	3	3.5	14	0.83	14λ
Case 2	10	7	1.5	14	0.15	14λ

3. Thermophysical properties of nanofluids

The thermal conductivity of nanofluids has received major attention in nanofluids. Although this is the most important issue for stationary fluids, considering heat transfer fluids, the heat transfer coefficient of the nanofluid in flow is the most important issue. Other important properties other than thermal conductivity that affect the heat transfer coefficient are: density, specific heat, and viscosity of the nanofluid. Assuming uniform dispersion of nanoparticles in the base fluid, the thermal and physical properties of the nanofluid are as follows.

Density

The density of nanofluids is usually calculated from the Pak and Cho equation [8].

$$\rho_{nf} = (1-\varphi)\rho_{bf} + \varphi\rho_p \quad (2)$$

Dynamic Viscosity

Viscosity is a measure of a fluid's resistance to deformation due to longitudinal or shear stresses. Due to the suspension structure of nanofluids, viscosity plays a special importance in the design of nanofluidic systems, as its direct effects on pressure drop in displacement flows are very evident. Therefore, for the use of nanofluids in practical applications, the amount of increase in the viscosity of nanofluids compared to their base fluid must be fully investigated and evaluated.

$$\mu_{nf} = \mu_{bf} A_1 e^{(A_2\varphi)} \quad (3)$$

Table 3: The coefficients of equation 3

Concentration (%)	Average particle size (nm)	(mm)	A_1(mm)	Nanoparticle
$0 < \varphi < 10$	45	12.959	0.983	AL_2O_3
$0 < \varphi < 6$	29	22.854	0.9197	CuO

4. Governing equations

In the present study, a single-phase model is used to solve the nanofluid flow. Therefore, the governing equations are the same as for conventional fluid flow. For a steady flow where the physical properties of the fluid are a function of temperature, the mass conservation, momentum conservation, and energy conservation equations are as follows:

Continuity equation for incompressible fluid:

$$\rho\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}\right) = 0 \quad (4)$$

Momentum equation:

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{\mu}{\rho}\left[\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right] + pg_x \quad (5)$$

(6)

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\frac{\mu}{\rho}\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right]+pg_y$$

(7)

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial z}+\frac{\mu}{\rho}\left[\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right]+pg_z$$

Energy equation

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=-\frac{k}{\rho C_p}\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right]$$

(8)

5. Result and discussion

To investigate the flow inside the smooth pipe, Reynolds numbers of 100, 150, 500 and 1000, which are in the range of laminar flow, are simulated. A pipe with a diameter of 4.57 mm and a different length for each Reynolds number, which can also be seen in Figures 3, 4 is simulated according to the description in the previous chapter.

Figures 3 and 4 shows the axial variations of the convective heat transfer coefficient and local Nusselt number of the water-based fluid with Reynolds number. As expected, convective heat transfer increases with increasing Reynolds number due to the decrease in boundary layer thickness.

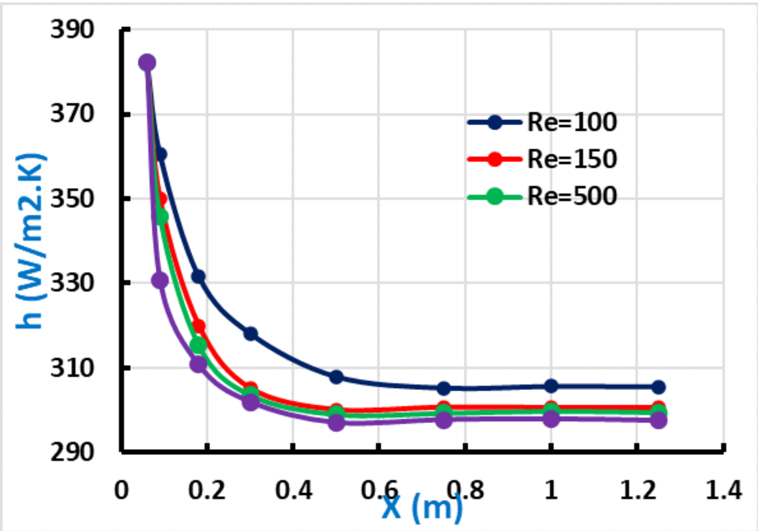

Figure 3. Axial variations of the convection heat transfer coefficient of water fluid with Reynolds number

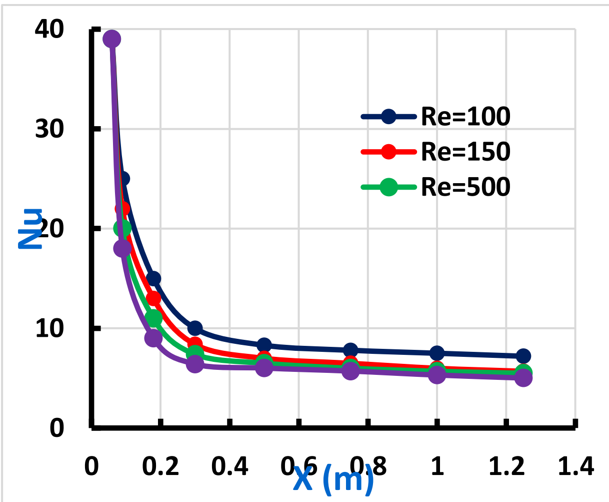

Figure 4. Axial variations of the local Nusselt number of water fluid with Reynolds number

To investigate the effect of the base fluid, a water-ethylene glycol mixture, which is used in cold regions of the world due to its low freezing point, has been used. The axial variations of the convection heat transfer coefficient and the local Nusselt number with the Reynolds number are as shown in Figures 5, 6 and with increasing Reynolds number, the same increase in heat transfer is observed as before. Of course, it should be noted that this fluid requires a longer development length than water due to the higher Prandtl number.

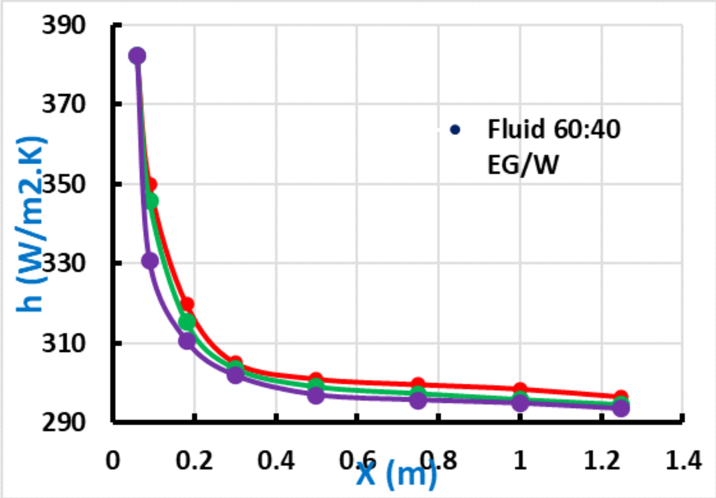

Figure 5. Axial variations of convection heat transfer coefficient of water-ethylene glycol base fluid with Reynolds number

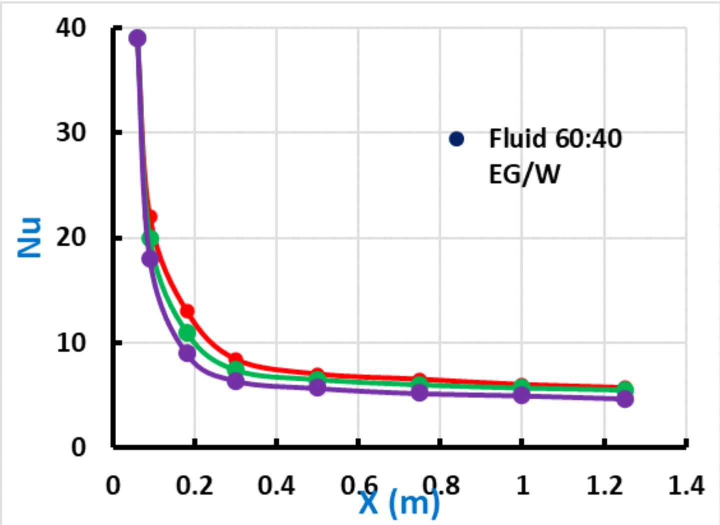

Figure 6. Axial variations of convection local Nusselt number of water-ethylene glycol base fluid with Reynolds number

6. THE PROPOSED METHOD

In this study, a computational fluid dynamics (CFD) approach was adopted using ANSYS CFX software. The flow was considered steady, laminar, and incompressible. The nanofluid was modeled as a single-phase mixture, and the governing equations for mass, momentum, and energy conservation were solved numerically. The boundary conditions included a constant heat flux on the tube wall and uniform velocity at the inlet. The mesh independence test was performed to ensure numerical stability and accuracy. All thermophysical properties were temperature-dependent, and convergence criteria were strictly monitored.

7. Comparative Study

A comparative analysis was conducted between the present results and recent state-of-the-art studies. Compared to the results of [7] and [5], the proposed sinusoidal-tube configuration achieved up to 28% higher Nusselt number and 12% lower pressure drop under equivalent Reynolds number conditions. These findings demonstrate that the present design offers a favorable balance between enhanced heat transfer and hydraulic performance, outperforming conventional microfinned and dimpled tube designs.

9. Research Future Directions

9. Conclusions

The following results can be deduced from the investigations carried out in the three geometries under study:

In a straight tube, with the growth of the thermal boundary layer, the local Nusselt number shows a continuous decrease. In a sinusoidal tube, the local Nusselt number at each wavelength in the converging part is higher than in the diverging part. The reason for this is the higher average velocity and velocity gradient in the converging part, which increases the heat transfer. It is observed that the value of the local Nusselt number at the throat of the sinusoidal tube is much higher than its value in a straight tube at all wavelengths. On the contrary, the reverse flow near the wall in the convex part has a low velocity gradient, which reduces the heat transfer rate.

Nomenclature

Cp	specific heat $J.Kg^{-1}.K^{-1}$	u	x- velocity $m.s^{-1}$
ρ	Density $Kg.m^{-3}$	v	y-velocity $m.s^{-1}$
k	thermal conductivity $W.m^{-1}.K^{-1}$	w	z-velocity $m.s^{-1}$
h	heat transfer coefficient $W.m^{-2}.K^{-1}$	T	Temperature K
Pr	Prantl Number		
Re	Reynold Number		
Nu	Nusselt Number		

Conflicts of Interest.

The authors declare no conflict of interest.

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