Numerical and experimental Study of Heat Transfer Enhancement in Contour Corrugated Channel Using nanofluid and Engine Oil

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Abstract. Research on improving heat transfer methods in heat exchangers has received wide attention in response to the increasing demand for high-efficiency devices in this field. Wherefore, the corrugated wall is one of the many suitable techniques used to enhance the heat transfer in heat exchangers. In the present investigation, laminar flow thermal and hydraulic efficiency of the trapezoidal, and straight counter heat exchanger with SiO2/water nanofluid and engine oil was carried out numerically and experimentally over Reynolds number ranges of 1100-2300 for nanofluid and 0.250 for engine oil. The results indicate that the numerical simulation of the laminar flow of nanofluids was performed with fractions of 0-4% volume. It should be noted that SiO2/water nanofluid has resulted in a significant improvement in heat transfer for all Reynolds numbers. Moreover, The error deviation of the Nusselt number is 0.0102% and 0.0208% between numerical and experimental work and the error deviation of the pressure drop is 0.0512% and 0.0325% between numerical and experimental work over the Reynolds number of 2300. The use of corrugated surfaces in reverse heat exchangers with nanofluid can improve heat transfer in many engineering applications, therefor the improving of the design and engineering parameters of a heat exchanger increases the heat transfer and makes it more compact, which in turn increases the efficiency of the thermal process, which leads to saving operating costs.

1. Introduction

In recent years, research on the methods for heat transfer enhancement in heat exchangers has received great attention in order to cater to the growing needs of higher efficiencies in these devices. There are diverse ways to improve heat transfer methods. One of the traditional procedures to decrease thermal resistance is using a heat exchanger with a larger surface area or reduce the thickness of the thermal boundary layer on the surface of the heat exchanger and this will be increased in the cost of materials and increase heat exchanger mass. To reduce the thickness of the boundary layer, the technique of generating vortices was invented. There are two categories or methods for applied techniques to expand heat transfer. Active methods required external power such as surface vibration, mechanical aids, electrostatic fields, and fluid vibration, and Passive methods that do not require to apply external power for example extended surfaces (fins), rough surface, displaced promoters, fluid additives (nanofluids), and vortex flow devices. The present work concentrations combined two passive methods to increase heat transfer efficacy which are fluid additives (nanofluids), and vortex flow devices (corrugated
surface). Several studies found an increase in the enhancement of heat transfer with a corrugated surface in the heat exchanger. Mohammed et al. [1] carried out a numerical study on the plate heat exchanger to investigate heat exchange and forced turbulent convective flow in a model of corrugated channel. This investigation was including measurement of performance of evaluation criterion in corrugated tilt angles effects of geometrical parameters using water as a working fluid in the smooth channel and wavy channel. The results illustrate the significant effect of the wavy channel on the enhancement of heat transfer compare to the smooth channel. Heidary et al., [2] Performed a parametric study to display the influence of corrugated flow channel on the improvement of heat exchange between the cold bottom wall and the core hot flow in the anode electrode of direct methanol fuel cells. They are observed that corrugated bottom wall with triangular, wavy, and trapezoidal shapes can significantly increase the efficiency of the heat exchange between the wall and the core flow. Tokgoz, Aksoy, and Sahin, [3] numerically and experimentally investigated the flow features and thermal efficacy in various ducts geometries. The entire studies were done for Reynolds numbers (Re) in the range of [3000-6000]. They found that improvement in heat transfer was significant at a larger amplitude wavelength ratio, especially at higher Re. Ghule and Soni, [4] have been analyzing numerically heat transfer of different cross-sections of wavy microchannels by different Re and the amplitude of waviness. The study has been considered four different cross-sections, circular, namely rectangular, notched circular, and notched rectangular in this study. The results show that enhancement in fluid mixing increasing by increasing the amplitude of waviness and yields an increase in the heat transfer coefficient. Zhang and Che, [5] investigated numerically the influence of corrugation profile for cross-corrugated plates on thermal-hydraulic performance using air as a working fluid. The result shows that the channel with sharp corrugations has a higher average Nusselt number and friction factor compared to the channel with the smooth corrugations. The average Nusselt number, as well as the friction factor, was about [1-4] times greater for the trapezoidal-corrugated channel than for the elliptic-corrugated channel. Dai et al., [6] carried out an experimental study of characteristics of the heat transfer and hydrodynamic of water flowing via microchannels for range 50 to 900 Re. The microchannels have consisted of semi-circular cross-sections with a diameter of 2 mm supported by sinusoidal or zigzag pathways with five repeating units as a minimum. By using wavy channels, the heat transfer was enhanced compared to the equivalent straight channel, albeit the pressure-drop penalty was increased. The bend leads to a rise in the heat transfer rates and induces recirculation of flow and secondary flow structures. Mabrouk, A., Y. Elhenawy, and G. Moustafa, [7] study the ability to enhance thermal energy efficiency and mass flux by using a new system of corrugated feed channel supported by the flat sheet. There was an enhancement in the thermal efficiency and mass flux, operating conditions, and gap height by using a corrugated feed channel. Because of the increased demand for heat removal from the thermal systems, the advance in nanotechnology allows a new class of heat transfer working fluids having far superior thermal conductivities called nanofluid. Nanofluids are expected to be one of the important future heat transfer fluids by dissolving Nano-size particles in the base fluids to elevate the level of the thermal conductivity and, therefore, thermal performance. As shown by the literature a large enhancement has been found in the thermophysical properties of the nanofluids compared to the conventional fluids even at very low solid concentrations. Ahmed et al., [8] studied numerically laminar flow and characteristics of nanofluid [CuO–water] for heat transfer in corrugated and straight channels for Reynolds number [100-800] and volume fraction [0-0.05]. The outcome of this study demonstrates that there were an increase in thermal-hydraulic performance factor and Nusselt number due to an increased volume fraction of nanoparticles and Reynolds number for all the shapes of the channel. Besides, when the nanoparticles volume fraction increased, the non-dimensional pressure drops were increased. Selime and Hakan, [9] examined numerically by using the finite volume method the laminar forced convection of pulsating nanofluid flow over a backward-facing step with a corrugated bottom wall. The researcher investigates the effect of a fraction of nanoparticle volume on the fluid flow and heat transfer. There was an increase in the rate of average heat transfer due to the inclusion of nanoparticles, however, the enhancement depends on the interval of the nanoparticle solid volume fraction. Khoshvaght, [10] were studied flow characteristics and heat transfer of the sinusoidal-corrugated channel supported by nanofluid, Al2O3-
water. They evaluate the influence of phase shift at different Re [6000–22,000], nanoparticle volume fractions [0–4%], channel height, wavelength, and amplitude of the channel. They found that the nanofluid flow through the sinusoidal-corrugated channels display higher results of Nusselt number compared to the base fluid, and the friction factor of nanofluid is higher compared to the base fluid. Abed et al., [11] studied numerically a developed turbulent flow and behavior of heat transfer in trapezoidal channels using nanofluids. They investigated the reliability of 4 types of nanoparticles, [Al₂O₃, CuO, SiO₂, and ZnO] with volume fractions of (0 to 4%) and diameters (20 nm to 80 nm) with constant heat flux (6 kW/ m²). The results approved that SiO₂ has the highest Nusselt number among the nanofluids. The heat transfer enhancement was seen with increased concentration of particle volume, however, a minor pressure loss with nanoparticle and less diameter is also detected. An increase in average Nusselt number by 10 % was found when the use of nanofluids in forced convection for nanoparticles diameter of [20 nm] at [4Vol %]. Ahmed et al., [12] studied numerically and experimentally convective heat transfer of SiO₂-water nanofluid flow in different shapes channels over Reynolds number ranges of [400–4000]. They show that increase the nanoparticle volume fraction leads to elevation of Nusselt number and the heat transfer enhancement while the use of nanofluid caused an increase in the pressure drop. Ahmed et al., [13] studied experimentally convective heat transfer of channels with different shapes with [SiO₂-water] nanofluid flow over Reynolds number ranges of [400–4000]. The current study was fabricated and test trapezoidal, sinusoidal, and straight channels. The nanofluid SiO₂-water with 0%, 0.5% and 1.0% volume fractions made and tested. The results showed that with increase nanoparticle volume fraction there was an increase in the average Nusselt number and enhancement in the heat transfer, however, there was an increase in pressure drop.

Based on the above reviews, it was observed that many numerical and experimental investigations were done on the convective heat transfer in the corrugated channel using conventional fluids. There are a few numerical and experimental researches regarding nanofluid in the corrugated channel for heat transfer and flow characteristics. There is no numerical or experimental study has been conducted on nanofluids and oil engine in counter trapezoidal corrugated channels for the convective heat transfer.

Figure 1. basic geometry of the corrugated channel of the present study.
2. Mathematical model

2.1. Problem description

The basic geometry of the corrugated channel is shown in Figure 1. It consists of one corrugated stainless steel wall with an amplitude (a) and wavelength (Lw). There are ten corrugations (pitches) along the corrugated wall. To create proper boundary conditions for both outlet and inlet of the corrugated channel of water and oil, four adiabatic smooth sections are taken into account. The top and the bottom of the test section (Lc), Upstream the corrugated section (Ld), and the other downstream (Le).

The upstream of the corrugated section has an axial length of Ld= 200 mm and the downstream has an axial length of Le=600 mm. The average height (Hav) of all these sections was 10mm. The test section Lc=200mm However, the following geometric parameters are considered in the current study: the amplitude of corrugated channel (a) of 0,1,2,3 and 4 mm and the wavelength of the corrugated wall (Lw) of 20,30 and 40mm. In the current study and to get the numerical solution and final form of governing equations, several expectations have to be taken into account.

1. The flow is assumed steady.
2. Two-dimensional and incompressible.
3. the mixture of base fluid (water) and the solid spherical nanoparticles (MgO and SiO2) are in thermal equilibrium and both run through the channel at the same velocity.
4. Nanofluid is considered a Newtonian fluid.

2.2. Governing equations and Assumptions

To complete the numerical solution and obtain the final form of governing equations for the current study, some assumptions should be considered. The flow is adopted to be steady-state conditions, fully developed, and two-dimensional. Sidewalls and all the upstream walls are considered to be adiabatic surfaces. The base fluid is assumed to have a thermal equilibrium and the no-slip condition occurs. The fluid flow is considered to be Newtonian and incompressible. The SiO2/water nanofluid and oil in thermal equilibrium and run through the channel at the same velocity.

In this model, the governing equations were discretized using the finite volume approach. Continuity equation, Momentum equation, and Energy equation can be written as (Moraveji et al.) [14]:

**Continuity equation:**
\[ \nabla \cdot (\rho \mathbf{v}) = 0 \]  \hspace{1cm} (1)

**Momentum equation**
\[ \nabla \cdot (\rho \mathbf{v} \mathbf{v}) = -\nabla P + \nabla \cdot (\mu \nabla \mathbf{v}) \]  \hspace{1cm} (2)

**Energy equation:**
\[ \nabla \cdot (\rho C_v T) = \nabla \cdot (k \nabla T) \]  \hspace{1cm} (3)

2.3. Thermophysical Properties of nanofluid Water-based Suspension

Although some studies are designed to investigate thermophysical properties, such as classical models. Indeed, experimental outcomes allow us to choose a suitable model for a specified feature. The influencing features of nanofluid MgO/water and SiO2/water are described as below:

Equation (4) was originally presented in Ben-Mansour and Habib, [15], for determining the density.

Equation (5) that is used for specifying heat capacity was first utilized in (Moraveji and Ardehali, [14]):

\[ \rho_{nf} = \phi_p \rho_p + (1 - \phi_p) \rho_{bf} \]  \hspace{1cm} (4)

\[ C_{nf} = \frac{\phi_p \rho_p C_p + (1 - \phi_p) \rho_{bf} C_{bf}}{\rho_{nf}} \]  \hspace{1cm} (5)

to achieve the most precise numerical results in the current study viscosity values and thermal conductivity for different concentrations of nanofluid have been reported by Corcione, [16]. They produced empirical correlation with a standard deviation of error equal to 1.86%.

Thermal conductivity of nanofluid:
\[
\frac{k_{nf}}{k_{bf}} = 1 + 4.4Re^{0.4}Pr^{0.66} \left( \frac{T}{T_f} \right)^{10} \left( \frac{k_p}{k_{bf}} \right)^{0.03} \phi^{0.66} 
\]

where \(Re\) represents the Reynolds number of nanoparticles, \(Pr\) is the Prandtl number of the base liquid, \(T\) is the temperature of nanofluid, \(T_f\) is the freezing point of the base liquid, \(k_p\) is the thermal conductivity of the nanoparticle, and \(\phi\) is the volume fraction of the suspended nanoparticles. The nanoparticle Reynolds number in detail is defined as:

\[
Re = \frac{\rho_{bf}u_B d_p}{\mu_{bf}}
\]

Where \(\rho_{bf}\) is a mass density and \(\mu_{bf}\) is the dynamic viscosity of the base fluid and \(d_p\) is nanoparticle diameter and \(u_B\) is the mean Brownian velocity. In case of the absence of agglomeration, \(u_B\) equation is:

\[
u_B = \frac{2k_{bf}T}{\pi\mu_{bf}d_p^2}
\]

the viscosity of nanofluids is dependent on Corcione, [16] equation.

\[
\mu_{nf} = \frac{1}{1 - 34.87(d_{bf}^{-0.3} + 0.03) \phi^{0.66}} \mu_{bf} 
\]

where

\[
d_{bf} = 0.1 \left( \frac{6M}{N\pi\rho_{bf}} \right)^{0.33}
\]

Where

\(M\): Molecular weight of water, \(M = 80 \times 10^{-2} (kg/mol)\)

3. Experimental Setup

The experimental setup for measuring the convective heat transfer and pressure drop characteristics in a corrugated channel using nanofluid and oil engines are shown in Figure 2. It mainly includes water chiller, developing section, test section, exit section, flow meter, differential pressure, data logger, and heater. The detailed descriptions of these components are given below.

![Figure 2. Photograph and schematic diagram of the experimental setup.](image)
The experimental setup, figure 3, and test section consist of a counter corrugated heat exchanger as depicted in Figure 3.b. The top and bottom walls of the test section were fabricated from acrylic with dimensions of 8 mm thick, 50 mm wide, and 200 mm long as shown in figure 3c. However, the form of a sidewall of the two channels was accomplished by using wire electrical discharge machining (WEDM). The wall between the two channels of the test section was fabricated from stainless steel at 1 mm thick, figure 3c. The test section was assembled and the corrugated wall-side wall junction was sealed using thermal epoxy and silicon.

![Figure 3](image1.png)  
**Figure 3.** Experimental setup and the test section.

3.1. Preparation of Nanofluids

In the current study, the Nanopowder of SiO$_2$ with an average diameter of 20 nm and the purity of 99.99% (Purchased from schmitz Science and Technology Co., Ltd.) was used to prepare the nanofluid. The scanning electron microscope (SEM) of SiO$_2$ Nanopowder is depicted in figure 4.

![Figure 4](image2.png)  
**Figure 4.** SEM photograph of SiO$_2$ nanoparticles.

In the current study, the nanofluid was prepared by using ultrasonic equipment based on the two-step method. This method was previously used by many researchers such as Pandey and Nema [17], Kalteh
et al. [18], and Heris et al. [19] to prepare different types of nanofluids. Therefore, the ultrasonic bath (Fisher Scientific, Model FB15051) with the ultrasonic frequency of 37 kHz and the maximum power of 320 W was used in the current study. The amount of SiO$_2$ Nanopowder required for nanofluid preparation can be given by the following relation equation (Syam et al.,[20]):

$$\varphi = \frac{m_p}{\rho_p} \left( \frac{m_p + m_f}{\rho_p + \rho_f} \right)$$

(8)

The photograph of the nanofluids samples with volume fractions 4% is depicted in figure 5. It was observed that the SiO$_2$-water nanofluid prepared in the current study was stable and uniform during all the experiments.

Figure 5. SiO$_2$-water nanofluids at 4% particle volume fractions.

a. Data Reduction

The experimental data recorded in the current study, such as bulk fluid temperature, wall temperatures, pressure drop, volume flow rate, voltage, and current, can be used to estimate the Nusselt number, friction factor as well as the thermal-hydraulic performance of the corrugated channels. In this investigation, the nanoparticles were added to the water-based fluid with volume concentrations of 4%. The heating water gave its thermal energy to the nanofluid during the experimental test. Equation (9), is used to calculate the heat transfer rate from the heat transfer fluid. Khoshvaght-Aliabadi, et al.[21].

$$Q_{oil} = \dot{m}_{oil} C_{P_{oil}} (T_{out} - T_{in})_{oil}$$

(9)

The heat transfer rate into the cooling nanofluid was obtained by utilizing the equation below:

$$Q_{nf} = \dot{m}_{nf} C_{P_{nf}} (T_{out} - T_{in})_{nf}$$

(10)

Where $m_{nf}$ is the mass flow rate of the nanofluid, $C_{P_{nf}}$ is the specific heat of nanofluid and $T_{b\,in}$ and $T_{b\,o}$ are the average bulk temperatures at the inlet and the outlet of the test section, respectively. So, the average heat transfer rate can be calculated by equation (10).

$$Q_{ave} = \frac{Q_{oil} + Q_{nf}}{2}$$

(11)

Therefore, the average heat transfer coefficient can be expressed as follows in equation (12).
\[ h_{ave} = \frac{q_{ave}}{A_s(T_{wall} - T_{b,ave})} \]  

(12)

Where, \(A_s\) is the surface area of the corrugated wall, \(T_{wall}\) represents the mean temperatures of the pipe wall surface measured by six thermocouples located alongside the inlet and outlet of the test section by equation (13).

\[ T_{wall} = \frac{\sum T_{wall}}{6} \]  

(13)

\(T_{b,nf}\) is the mean bulk nanofluid temperature from equation (14).

\[ T_{b,nf} = \frac{T_{in} + T_{out}}{2} \]  

(14)

Then, the average Nusselt number is calculated as follows.

\[ Nu_{ave} = \frac{h_{ave}D_h}{k_{nf}} \]  

(15)

Where \(k_{nf}\) is the thermal conductivity of the nanofluid and \(D_h\) is the hydraulic diameter of the corrugated the channel which can be defined as from equation (16).

\[ D_h = \frac{4A_c}{p} \]  

(16)

Where, \(A_c\) is the cross-sectional area of corrugated channel and \(p\) is the channel circumference, respectively.

The friction factor \((f)\) can be estimated as follows (Pandey and Nema,[17]).

\[ f = \frac{2D_hA_c^2\mu_{nf}}{L_cH_{ave}^2} \]  

(17)

Also, the Reynolds number can be defined as follows.

\[ Re = \frac{mD_h}{\mu_{nf}A_c} \]  

(18)

Where, \(m\) is the mass flow rate which was measured using the flow meter, \(\rho_{nf}\) is the density of nanofluid, \(\mu_{nf}\) is the viscosity of nanofluid and \(A_c\) is the cross-sectional area of the test section which is defined as:

\[ A_c = H_{ave}W \]

Here, \(H_{ave}\) is the average height of the channel and \(W\) is the width of the channel.

4. Result and discussion

Recently, researchers have paid great attention to nanofluid as an alternative to conventional fluids such as water, oil, etc, as these conventional fluids are unable to meet the requirements of development in industrial applications as a tool to transfer heat in heat exchangers. For this reason, the current study aimed to prove this fact. This work has been divided into two parts, numerical and experimental work. The numerical part dealt with SiO\(_2\) nanofluid, and its effect on the rate of heat transfer and pressure drop in the trapezoidal heat exchanger then the comparison of the nanofluid with the traditional fluid water has been done. The purpose of the numerical study is to demonstrate the best fluid to be studied.
experimentally. The second part studied experimentally the best fluid in the heat exchanger and its effect on improving heat transfer taking into account the pressure drops.

4.1. Effect of Nanoparticles Volume Fraction

Numerical investigation on the effect of nanofluid (SiO$_2$/water nanofluid) in volume fractions of (0, 0.8%, 2%, 3%, and 4%) and dp=20 nm in the trapezoidal-corrugated channel at a=4.0 mm and Lw=20 mm on the Nusselt number average and drop in pressure has been done.

4.1.1 Effect of volume fraction of nanofluid with the Nusselt number

Figure 6 depicts the different volume fractions of SiO$_2$-water nanofluid and variation of the average Nusselt number versus Re within the trapezoidal-corrugated channel. As expected, the average Nusselt number of trapezoidal-corrugated channel, at a given volume fraction, increases as the Re increases. This increase in Re is associated with a decrease in the thermal boundary layer thickness and an increase in the temperature gradient near the channel walls due to the increased intensity of the turbulent near the wall. Also, the increase in the volume fraction of the nanoparticles is accompanied by an increase in Nusselt number at a given Re. This increase is caused by the Brownian movement of the nanoparticles and the improved thermal conductivity of the primary fluid. Besides, the figure shows that the SiO$_2$-water nanofluid has led to a higher Nusselt number than water. The reason for this is attributed to the fact that the thermal conductivity of SiO$_2$-water nanofluid is higher than water. This means that the rate of heat transfer with SiO$_2$-water nanofluid is higher than water.

4.1.2 Effect of volume fraction of nanofluid with the pressure drop.

Figure 7 displays the difference of the pressure drop versus Re for SiO$_2$-water nanofluid at the different volume fractions of 0%, 0.8%, 2%, 3%, and 4%, with dp=20 nm in the trapezoidal-corrugated channel at a=4.0 mm and Lw=20 mm. It is also observed from the same aforementioned figure that with the increase of Re, the pressure increases dramatically at a given volume fraction of the nanoparticles due to the high-velocity gradient at the channel walls. The results showed that a pressure drop will increases with nanoparticle volume fractions due to increasing nanofluid viscosity. In addition, the intensity of recirculation regions becomes stronger due to the increase of the nanoparticles volume fraction, as pointed out before, resulting in a further increase in the pressure drop.
Figure 7. Variation of pressure drops vs Reynolds number for different volume fractions, $\phi$ of SiO$_2$ at dp=20nm in the trapezoidal channel.

4.2 Experimental work

Choosing an industrial product, such as a heat exchanger, usually undergoes several design tests in order to obtain the product within the required specifications. Nowadays most industries focus on the cost of production for this reason and in order to avoid the high costs, simulation programs have been used to choose the required design. In the current study, this principle was applied to select the optimal design for the corrugated channel heat exchanger, as well as the optimal type of fluid used in the heat exchanger. It is unreasonable to perform virtually all operations in order to choose the final design.

4.2.1 Effect of nanofluid on Average Nusselt number

Nusselt number is considered one of the most important parameters that determine the efficiency of the heat exchanger because it shows the ratio of heat transfer by convection to the ratio of heat transferred by conduction. For this reason, it was studied experimentally in the current work. Figure 8 shows the relation between Nusselt number and Reynolds number for SiO$_2$-water nanofluid and water flow in a trapezoidal channel and water flow in the straight channel. It is observed that the Nusselt number of the nanofluid and water flow increases with increasing Reynolds number. The reason for this increase is due to the increase in the velocity of the fluid, which leads to an increase in a temperature gradient, and thus the increase in heat transfer rate. As we have shown previously in the numerical study that the Nusselt number was greater in the case of nanofluid compared to water because of the high thermal conductivity that resulting from the addition of nanoparticles to the base fluid. Besides, the fig shows also that the trapezoidal channel for all the fluids has given a higher Nusselt number than the straight channel due to the fact that the flow becomes more disturbing because of the corrugated surface.
4.2.2 Pressure drop
To demonstrate the success of any heat exchanger, a drop in pressure must take into consideration that resulting from the optimization made on the heat exchanger. Figure 9 shows that the pressure drops resulting from the flow of the SiO$_2$-water nanofluid in the corrugated channel is higher than the pressure drops resulting from the flow of water, and for all cases of Reynolds number. The reason for this is that the viscosity and density of SiO$_2$-water nanofluid are higher than the viscosity and density of water, as the pressure increases with the increase in viscosity and density of the fluid. Moreover, from the figure, the pressure drop in the trapezoidal channel is higher than the pressure drops in the straight channel because of the increasing resistance of the corrugated surface to the flow in the trapezoidal channel.

4.2.3 performance evaluation criteria (PEC)
As mentioned earlier in the numerical study that the success of improving heat transfer also depends on the friction loss, as there is no benefit in improving heat transfer through various techniques if the friction loss is higher than the rate of improving heat transfer. Researchers have found many ways to demonstrate the viability of a heat exchanger. One of the most important methods or terms used by researchers is the
term performance evaluation criteria. This parameter gives an accurate description of the ratio of improved heat transfer to the ratio of friction loss, which should be higher than one. Figure 10 shows that the performance evaluation criteria of SiO$_2$-water nanofluid is higher than one and for all Reynolds number in the trapezoidal channel.

**Figure 10.** Performance evaluation criteria vs Reynolds number for SiO$_2$-water nanofluid in trapezoidal and water in a smooth channel.

4.3. **Validation numerical with experimental work**

In order to validate the numerical output, the variation of the Nusselt number of the trapezoidal channel has been firstly compared with experimental data from current work for conventional fluids and nanofluid in a trapezoidal channel. Figure 11 shows that the agreement of present numerical data of the average Nusselt number for distilled water and SiO$_2$-water nanofluid with a 4% volume fraction with the Nusselt number of the experimental work. The inaccuracy of temperature gauges and thermocouple caused little deviation between experimental Nusselt number and numerical Nusselt number values.

**Figure 11.** Comparison between numerical results and measured data of the average Nusselt number for SiO$_2$-water nanofluid at $\phi=4\%$ and $d_p=20$ nm: trapezoidal channel with $a=4$ mm and $L_w=20$ mm.

Figure compares the average pressure drop obtained from experimental results with the corresponding
numerical results for distilled water and SiO$_2$-water nanofluid with a 4% volume fraction. Single-phase modeling results are presented in this figure. Trapezoidal shapes of the corrugated channel have been studied. From the observation of the figure, it is clear that the pressure values for the experimental study are almost identical to the pressure drop values of the numerical study. The reason for the deviation between the experimental and numerical study is the inaccuracy of pressure gauges in the practical study.

![Figure 12](image_url)

**Figure 12.** Comparison between numerical results and measured data of the pressure drop for SiO$_2$-water nanofluid at $\phi=4\%$ and $dp=20$ nm: trapezoidal channel with $a=4$ mm and $L_w=20$ mm.

5. **Conclusion**

In this work, a numerical verification was performed to study the laminar forced convective heat transfer of SiO$_2$/water nanofluids in corrugated channels. Ansys's (Fluent) R19 program has been used to simulate the single-phase laminar flows of nanofluids at different nanoparticles concentration. The experiments were conducted for 4% volume fractions of SiO$_2$ nanoparticles dispersed in the distilled water and different shapes of channels which are trapezoidal, and straight channels over Reynolds’s number range of 1100-2300. Based on the experimental results discussed in the result section, the following conclusions can be summarized:

1. The average Nusselt number, as well as the pressure drop in all channels, considerably increased with the increasing Reynolds number.
2. The average Nusselt number, pressure drop, and the heat transfer enhancement for the corrugated channel increased with the addition of nanoparticles. About 9.9-15.7%, heat transfer enhancement was observed in all channels with the addition of 4% SiO$_2$ nanoparticles to the distilled water.
3. The trapezoidal-corrugated channel has the highest average Nusselt number, pressure drop, and heat transfer enhancement than the straight channel.
4. The average Nusselt number, pressure drop, heat transfer enhancement, thermal-hydraulic performance increased, as the volume fraction of nanoparticles increased. For SiO$_2$-water nanofluid with 4% volume fraction and 20 nm nanoparticles diameter, the maximum heat transfer enhancement was 135% in the trapezoidal channel compared to water in the straight channel.
5. The trapezoidal channel at the amplitude of 4 mm was 11.4%, times the average Nusselt number for the pure water ($\phi=0\%$) in a trapezoidal channel.

In general, the numerical and experimental results of the current study show that using nanofluid instead of traditional heat transfer fluids as well as using corrugated channels instead of the conventional (straight) channel can potentially achieve considerable improvement in the heat transfer performance, which can lead to the design of more compact heat exchangers.
Nomenclature

| Symbols | Description |
|---------|-------------|
| P       | Pressure Drop, Pa |
| a       | Amplitude, mm |
| H       | Height of the trapezoidal channel, mm |
| Lw      | Wavelength, mm |
| N       | Avogadro number |
| Nu      | Nusselt number |
| Nu_{ave} | average Nusselt number |
| Pr      | Prandtl number |
| Re      | Reynold Number |
| X       | Distance, m |
| Cp      | Heat capacity, J/kg K |
| T       | Temperature, K |
| k       | Thermal conductivity, W/m. K |
| p       | Pressure, Pa |
| Q       | Heat transfer W |
| U, V    | Velocity components, m/s |

Subscripts

| Symbols | Description |
|---------|-------------|
| av      | average |
| fr      | Freezing Point |
| bf      | Base fluid |
| np      | Nanoparticle |
| nf      | Nanofluid |
| W       | Wall |

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