Evaluation of Heat Transfer Enhancement and Pressure Drop Penalty of Nanofluid Flow Through a \( \Delta \)-Channel

Saeideh Kermani

Department of Mechanical Engineering, University of Kashan, Kashan, Iran

Email address: saiedeh.kermani73@gmail.com

To cite this article:

Received: February 9, 2018; Accepted: February 21, 2018; Published: March 21, 2018

Abstract: In this research, turbulent flow of water based silicon-oxide nanofluid in a channel with \( \Delta \)-shaped wavy wall has been scrutinized. Governing equations have been solved by control volume method based on SIMPLE algorithm. To reach desirable geometry, optimization has been done by benchmarking the maximum amount of performance evaluation criteria (PEC) regarding Nusselt number and pressure drop, among five different phase shifts and three different wave amplitudes. After finding optimum phase shift and amplitude, flow field and heat transfer in compulsory displacement of water based silicon-oxide (SiO\(_2\)) nanofluid with volume fraction from 1% to 4% of nanoparticle has been investigated. Result of simulation showed that the function of wavy channels is almost related to phase shift and amplitude of wavy wall. The topmost function evaluation scale for phase shift 90°, 0.5 mm amplitude in 6,000 Reynolds number has been obtained. The results indicate that water-SiO\(_2\) nanofluid with 4% volume fraction has the highest PEC in comparison with the other studied cases.

Keywords: Nanofluid, Turbulent Flow, Wavy Wall, Silicon-Oxide, Performance Evaluation Criteria (PEC)

1. Introduction

In recent years due to have more useful and affordable heat exchanger, different methods have been examined. Making indentation and groove on the inner surface of heat exchanger is one of the frequent methods for breaking smooth substratum of flow and making turbulences of local walls that leads to decreasing thermal resistance and increasing the heat transfer noticeably. The mail goal of current work is to find a way to reach the minimum pressure drop and maximum rate of heat transfer.

Some numeral and experimental studies have been done on fluid flow and heat transfer in wavy channel by many experts. Rush et al. [1] investigated manner of flow and local heat transfer through a sinusoidal wavy-channel for region of transient flow regime. It has observed that Reynolds number and geometry of channel affect mixture situation and characteristic of flow noticeably. A numerical research has been done on turbulent compulsory convection in a channel with wavy wall by Wang and Vanka [2]. Results showed that by increasing wave amplitude, wave length and Reynolds number, local Nusselt number has been increased noticeably. Yin et al. [3] do some researches on heat and air hydraulic parameters in sinusoidal wavy channels for different phase shifts between upper and lower walls numerically. Results showed that by increasing phase shifts, friction coefficient and Nusselt number have been decreased. According to Choi [4] nanofluid is mixture of nanoparticles less than 10nm. Nanofluid has the better thermophysical properties than basic fluid such as water, oil and ethylene glycol. Many researches show that nanofluid makes more heat transfer than base fluids. Vanaki et al. [5] investigated the effect of nanofluid on heat transfer field and flow in channels with sinusoidal wavy-channel numerically. They realized that simultaneously use of nanofluid and wavy walls can increase remarkably. Khorasanizadeh et al. [6] investigated the effects of using corrugated absorber plate on heat transfer and turbulent flow in solar air-heater collectors numerically. In their paper choosing the proper geometry was carried out based on the best performance evaluation criteria (PEC) and increasing the air temperature from collector inlet to outlet (ITIO). Their results revealed that using corrugated absorber plate has a considerable influence on flow field and heat transfer. For the whole year the highest PEC was obtained for the sinusoidal corrugated model, however the highest ITIO was obtained for rectangular corrugated model. Also it was known that for the best ITIO and the highest PEC, the optimum Reynolds
numbers is 2,500. Sheikhzadeh et al. [7, 8], in their papers studied forced turbulent convection flow and heat transfer of air in a desert helicopter cabin. The main aim of their studies was providing human thermal comfort for a desert helicopter pilot in summer (usage of cooling system). In order to fulfill this demand, a body subdomain was considered around the pilot that includes the pilot's using area in cabin by them. The governing equations were numerically solved by the control volume approach based on the SIMPLE technique and standard k-ε turbulent method. The effects of different supply air performances (velocity and temperature) on pilot thermal comfort parameters were presented. Then, the optimization was carried out to reach the optimal case with the minimum predicted percentage dissatisfied (PPD).

Sadripour et al. [9] investigated numerically the thermal comfort parameters and energy saving inside the room with specified dimensions using a ceiling fan with central heating systems during the winter. The flow was turbulent in all models and k-ε model was used to simulate turbulence. Rayleigh and Reynolds numbers were in the range of $1.15 \times 10^{12} \leq Ra \leq 1.55 \times 10^{13}$ and $6,480 \leq Re \leq 19,440$, respectively. The finite volume method (FVM) and SIMPLE algorithm were used to solve the governing equations. Based on the results, using the ceiling fan during the winter had a considerable effect on improving the thermal comfort and energy saving inside buildings. By using the ceiling fan, the effective room temperature increased by 0.35°C that can be used to reduce the radiators temperature, thereby reducing energy consumption. Additionally, the study results indicated that the location of ceiling fan did not have any effect on room effective temperature and residents' thermal comfort. Sheikhzadeh et al. [10] numerically studied the effects of speed and place of ceiling fans on thermal comfort parameters (PMV and PPD) and energy spending in two different office rooms with certain geometry in winter using district heating system. Based on results, using ceiling fans in winter (heating system) has a considerable influence on improvement of buildings thermal comfort and reducing energy. By using ceiling fans the effective temperature of room increases and therefore radiator energy consumption decreases.

Arani et al. [11] in their paper studied forced convection flow and heat transfer of boehmite alumina in ethylene glycol and water mixture nanofluid in sinusoidal-wavy mini-channel with phase shift and variable wavelength. Their optimization was carried out by using different nanoparticle shapes (spherical, spheroidal, platelets, blades, cylindrical and bricks) to reach the optimal nanoparticle shape with the maximum performance evaluation criterion (PEC). From this study, it is concluded that the thermal-hydraulic performance of channel is greatly influenced by changing the shape of nanoparticles. Using spherical and spheroidal nanoparticles improves the thermal-hydraulic performances of channel, while using non-spherical nanoparticle shapes (platelets, blades, cylindrical and bricks) leads to lower PEC in channel than the base fluid. Sadripour et al. [12] in their study investigated the effects of corrugated absorber plate and using aerosol-carbon black nanofluid on heat transfer and turbulent flow in solar collectors with double application and air heating collectors, numerically. In this investigation all the simulations were done for two different angles of tilt of collector according to horizon, that these angles were the optimum ones for the period of six months setting. As a result the corrugated absorber plate was inspected in the case of triangle, rectangle and sinusous with the wave length of 1mm and wave amplitude of 3mm in turbulent flow regime and Reynolds number between 2,500 to 4,000. Choosing the proper geometry was carried out based on the best performance evaluation criteria (PEC), for collectors with dual usage and increasing the air temperature from collector inlet to outlet for air heating collector. The results revealed that using corrugated absorber plate has a considerable influence on flow field and heat transfer. For all times of the year the highest PEC was obtained for corrugated Sinusoidal model, however the highest temperature increase from inlet to outlet was obtained for rectangular corrugated model. Sadripour [13] studied forced convection flow and heat transfer of MWCNTs-water nanofluid in heat sink collector equipped with mixers. The optimization was carried out by comparison of different parameters to reach the optimal case with the maximum exergy efficiency. From this study, it is concluded that in the case of using heat sink, instead of shell and tubes, the time that the fluid is inside the collector increases and leads to outlet temperature increase from the collector the exergy efficiency increases. Also, it is realized that using mixers enhance the outlet fluid temperature, energy efficiency and exergy efficiency. Generally, while the trend of exergy efficiency variation with effective parameters is increasing, applying the mixers precipitate the efficiency increment. Sadripour et al. [14] investigated numerically complex heat transfer (turbulent natural convection, conduction and surface thermal radiation) in a rectangular enclosure with a heat source has been carried out. The finite volume method based on SIMPLEC algorithm has been utilized. The effects of Rayleigh number in a range from $10^6$ to $10^9$, internal surface emissivity $0 \leq \varepsilon < 1$ on the fluid flow and heat transfer have been extensively explored. Detailed results including temperature fields, flow profiles, and average Nusselt numbers have been presented. In this investigation it has been tried to study the shape of heat source influence on heat transfer and fluid field in the considered domain. According to results in low emissivity values usage of circular obstacles is recommended. Although in high emissivity values using rectangular obstacles lead to more efficiency.

By analyzing former works collection of this issue obtained that numerical study for displacement turbulent nanofluid with different phase shifts between upper and lower walls has not been done. In current work nanofluid water-silicon oxide with volume fraction of 0 to 4% of nanoparticles and diameter of 25nm has been used.

2. Numerical Method

2.1. Physical Model

Schematic schema of studied channel with inlet height of
10mm and wave length of 11mm has been shown in Figure 1. The geometry indicating of two-dimensional wavy-wall channel includes six waves through test section. Length of test section is \( L_2 = 66 \text{mm} \) and length of upstream section for certainty of entering completely development flow to test section is \( L_1 = 150 \text{mm} \). Due to prevent returning flow to measurable outlet section has length of \( L_3 = 24 \text{mm} \). For left section of channel inlet velocity boundary condition has been considered in range of Reynolds number \((6,000 \text{ to } 8,000)\) and for outlet section of channel the outlet pressure boundary condition has been considered. Upper and lower walls have been kept in test section with \( \Theta \) and constant temperature of 400K. It has been assumed that flow fluid in inlet section of channel is steady and turbulent and with environment temperature of 300K.

### 2.2. Governing Equations

This study is done by using a commercial computational fluid dynamics package FLUENT 15.0. The governing equations for flow and heat transfer in the cavity can be written in the Cartesian tensor system as [11]:

\[
\frac{\partial}{\partial x_i} \left( \rho u_i \right) = 0
\]  

(1)

\[
\frac{\partial}{\partial x_i} \left( \rho u_i u_j \right) = \frac{\partial}{\partial x_j} \left( \rho u_i \right)
\]

+ \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] - \frac{\partial}{\partial x_i} \left( \rho u_i \overline{u_j} \right)
\]

(2)

\[
\frac{\partial}{\partial x_i} \left( \rho u_i T \right) = \frac{\partial}{\partial x_j} \left[ \mu + \frac{\mu_s}{P_r} \frac{\partial T}{\partial x_j} \right]
\]

(3)

where \( \rho \) is the fluid density and \( u_i \) is the axial velocity, \( \mu \), \( \mu_s \) and \( u_j \) are the fluid viscosity, fluctuated velocity and the axial velocity, respectively, and the term \( \rho u_i u_j \) is the turbulent shear stress.

For closure of the equations, the \( k-\epsilon \) turbulence model was chosen. A common method employs the Boussinesq hypothesis to relate the Reynolds stresses to the mean velocity gradient as (4) [11]. Also the turbulent viscosity term \( \mu_s \) is to be computed from an appropriate turbulence model. The expression for the turbulent viscosity is given as (5) [11].

\[
\mu_s = \rho C_{\mu} \frac{k^2}{\epsilon}
\]

(5)

where \( k \), named as turbulence kinetic energy (TKE), is obtained from the following equation [11]:

\[
\frac{\partial}{\partial x_i} \left( \rho u_i u_j \right) = \frac{\partial}{\partial x_j} \left[ \mu + \frac{\mu_s}{\sigma_k} \frac{\partial k}{\partial x_j} \right] + G_k - \rho \epsilon
\]

(6)

Similarly, in the dissipation rate of TKE, \( \epsilon \) is given by the following equation [45, 46]:

\[
\frac{\partial}{\partial x_i} \left( \rho u_i \overline{u_j} \right) = \frac{\partial}{\partial x_j} \left[ \mu + \frac{\mu_s}{\sigma_\epsilon} \frac{\partial \epsilon}{\partial x_j} \right]
\]

+ \( C_{\epsilon} \frac{\epsilon}{k} \) \( G_k \) + \( \frac{\epsilon}{k} \) \( \epsilon^2 \)

(7)

where \( G_k \) is the rate of generation of the TKE while \( \rho \epsilon \) is its destruction rate. \( G_i \) is written as [11]:

\[
G_k = -\rho u_i \overline{u_j} \frac{\partial u_j}{\partial x_i}
\]

(8)

The boundary values for the turbulent quantities near the wall are specified with the enhanced wall treatment method. These boundary values such as \( C_{\mu} = 0.09 \), \( C_{\epsilon} = 1.44 \), \( C_{\sigma_k} = 1.92 \), \( \sigma_k = 1.00 \), \( \sigma_\epsilon = 1.30 \) and \( P_r = 0.90 \) are chosen as empirical constants in the turbulence transport equations [11, 15]. The fluid is considered to be Newtonian, and the physical properties of the fluid are temperature dependent. Since the temperature variation is higher than 10°C [13], the following polynomial expressions are used to calculate these parameters:

\[
\rho(T) = 5.3738 \times 10^{-10} T^5 - 9.59976 \times 10^{-7} T^4
\]

+ 6.93809 \times 10^{-6} T^3 - 0.255822 T^2

+ 47.8074 T - 2584.53

(9)
To analyze and compare the flow characteristics and heat transfer of different suspended nanoparticles volume fractions and different nanoparticle sizes and shapes in wavy mini-channels, some definitions are given as follows [11]:

The Reynolds number is defined as:

$$Re = \frac{\rho_f u_m D_h}{\mu_f}$$

(13)

where $u_m$ is the mean velocity of fluid over the cross section. The hydraulic diameter of channel is defined as:

$$D_h = 2H + 2\alpha$$

(14)

The average Nusselt number is defined as:

$$Nu_{av} = \frac{h_f D_h}{k_f}$$

(15)

where $k_f$ and $h_f$ are the thermal conductivity and average heat transfer coefficient of base fluid, respectively.

The pressure drop between inlet and outlet is defined as:

$$\Delta P = P_{av,inlet} - P_{av,outlet}$$

(16)

The friction factor for fully developed flow is expressed as:

$$f = \frac{2}{\left(\frac{L}{D_h}\right)} \frac{\Delta P}{\rho_f u_m^2}$$

(17)

The performance evaluation criteria index (PEC) is used to compare the thermal and fluid-dynamic performances of wavy mini-channels with nanofluid to evaluate heat transfer enhancement. It is calculated using the predicted Nusselt numbers and friction factor as follows:

$$PEC = \left(\frac{Nu_{av,nf}}{Nu_{av,f}}\right) \left(\frac{f_{nf}}{f_f}\right)^{\frac{1}{3}}$$

(18)

where $Nu_{av,nf}$ and $Nu_{av,f}$ are the averaged Nusselt number for nanofluid and the base fluid, respectively. Also, $f_{nf}$ and $f_f$ are the friction factor for nanofluid and the base fluid, respectively.

To calculate the thermophysical properties of nanofluid with spherical nanoparticle, the following equations are proposed. The effective density $\rho_{nf}$ and specific heat $(c_p)_{nf}$ of the nanofluid at the reference temperature ($T_{in}$) are as follow [17]:

$$\rho_{nf} = (1-\phi)\rho_f + \phi\rho_{np}$$

(19)

$$\left(c_p\right)_{nf} = \left(1-\phi\right)\left(c_{p_f} + \phi\left(c_{p_{np}}\right)\right)$$

(20)

By using Brownian motion of nanoparticles in a wavy channel, the effective thermal conductivity can be obtained by using Corcione correlation [11]:

$$k_{eff} = \frac{1}{k_f} + 4.4Re_0^{0.4}Pr_f^{0.66} \left(\frac{T}{T_{fr}}\right)^{10} \left(\frac{k_{np}}{k_f}\right)^{0.03} \phi^{0.66}$$

(21)

where $Re_{np}$ is the nanoparticle Reynolds number, $Pr$ is the Prandtl number of the base liquid, $T$ is the nanofluid temperature, $T_{fr}$ is the freezing point of the base liquid, $k_{np}$ is the nanoparticle thermal conductivity, and $\phi$ is the volume fraction of the suspended nanoparticles. In more detail, the nanoparticle Reynolds number is defined as:

$$Re_{np} = \frac{\rho_f u_B d_{np}}{\mu_f}$$

(22)

where $\mu_f$ and $\rho_f$ are the mass density and the dynamic viscosity of the base fluid, respectively, and $d_{np}$ and $u_B$ are the nanoparticle diameter and mean Brownian velocity, respectively. Assuming absence of agglomeration, the nanoparticle Brownian velocity $u_B$ is calculated as the ratio between $d_{np}$ and the time $\tau_B$ required to cover such distance, that, according to Keblinski et al. [11], is:

$$\tau_B = \frac{d_{np}^2}{6\mu}$$

(23)

where $D$ is the Einstein diffusion coefficient and $k_B$ is the Boltzmann’s constant. Hence:

$$u_B = \frac{2k_B T}{\pi \mu_f d_{np}^3}$$

(24)

By substituting Eq. (22) in Eq. (20), we obtain:

$$Re_{np} = \frac{2\rho_f k_B T}{\pi \mu_f d_{np}^3}$$

(25)

Note that in the preceding equations all the physical properties are calculated at the nanofluid temperature $T$.[11].

$$\frac{\mu_{nf}}{\mu_f} = \left(1 - 34.87\left(\frac{d_{np}}{d_f}\right)^{-0.3} \phi^{0.3}\right)^{-1}$$

(26)
where $d_f$ is the equivalent diameter of a base fluid molecule, given by:

$$
 d_f = 0.1 \left( \frac{6M}{N \pi \rho_f} \right)^{1/3} 
$$

(27)

in which $M$ is the molecular weight of the base fluid, $N$ is the Avogadro number, and $\rho_f$ is the mass density of the base fluid calculated at temperature $T_0=293$K. The thermophysical properties of SiO$_2$ and water are listed in Table 1.

The governing equations were solved by finite volume method with SIMPLEC algorithm. For diffusivity and convective terms, the second-order upwind difference method is used. The convergence criterion is $10^{-6}$ and the under relaxation actors for velocity and temperature are 0.8 and 0.6, respectively. Since the temperature difference is insignificant, by considering the buoyancy force effects, dependence of energy and flow equations on temperature the Boussinesq model has been used. The time step used to solve the differential equations has been chosen to be $\Delta \tau = 10^{-3}$.

### Table 1. The thermophysical properties of water and SiO$_2$ at $T=300$ K [16].

<table>
<thead>
<tr>
<th>Material</th>
<th>$\rho$ (kg/m$^3$)</th>
<th>$c_p$ (J/kg·K)</th>
<th>$k$ (W/m·K)</th>
<th>$\mu$ (N·s/m$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>998.2</td>
<td>4182</td>
<td>0.6</td>
<td>0.001003</td>
</tr>
<tr>
<td>SiO$_2$</td>
<td>2200</td>
<td>703</td>
<td>1.2</td>
<td>-</td>
</tr>
</tbody>
</table>

2.3. Validation

![Figure 2](image-url)

(a) Grid independence test, and (b) Code validation.

The independence of the solution with respect to the grid size has been studied for a Reynolds number range of 6,000 to 18,000. Four different grid sizes with 58,793, 61,723, 64,416 and 67,583 nodes have been utilized. Figure 2a shows the variation of Nusselt number with different Reynolds numbers and grid sizes. Taking into account the conducted grid independence test the uniform grid with 64,416 nodes has been adopted to get an acceptable compromise between the computational time and the result accuracy.

In order to validate the CFD code which is developed in this study, the average Nusselt number of forced convection of SiO$_2$-water nanofluid flow with volume fraction of $\varphi=4\%$ through heated sinusoidal-wavy channel with amplitude of $\alpha=0.5$mm, wavelength of $\lambda=11$mm and phase shift of $\theta=30^\circ$ were calculated and compared with previous numerical results of Vanaki et al. [16]. According to Figure 2b, the results are in good agreement.

3. Results and Discussion

3.1. The Effect of Different Phase Shifts

In this section, the effect of phase shift between the upper and lower walls of wavy channel on fluid flow and heat transfer has been studied. For this propose five different phase shifts for configurations with wave amplitude of $\alpha=1$mm are prepared. The effect of different phase shifts at different Reynolds numbers for \(\Delta\)-channel with wave amplitude of $\alpha=1$mm on the Nusselt number, pressure drop and friction factor is showed in Figure 3. Also the effect of different phase shifts at different Reynolds numbers on PEC is reported in Figure 4.

As it can be seen in Figure 3a, as the Reynolds number increases, the average Nusselt number also increases. The large Reynolds number is attributed to the higher velocity which can lead to disturb the flow and thus, the heat transfer is strengthened. In all cases, the wavy-wall channel flows gave higher values of Nusselt number than that for smooth channel flow due to the induction of high disturbance and thin boundary layer in the wavy channels, leading to higher temperature gradients. It is found that the wavy channel with
A phase shift of $\theta=180^\circ$ has the highest average Nusselt number at all Reynolds numbers and can promote the heat transfer by approximately 126% in comparison with smooth channel at the lowest value of Reynolds number. $\theta=90^\circ$, $0^\circ$, $60^\circ$ and $30^\circ$ had the rates of approximately 115%, 112%, 105% and 101%. The pressure drop for various types of wavy-wall channels over the considered range of Reynolds numbers is plotted in Figure 3b. It is clearly seen that the wavy-wall channels provide a high amount of pressure drop in comparison with the smooth channel. This behavior is due to the fact that wavy corrugations disturb the entire flow field and cause more pressure drop. However, it is observed that $\theta=180^\circ$ phase shift channel has the highest pressure drop among all geometries, which is followed by $\theta=0^\circ$, $\theta=30^\circ$, $\theta=60^\circ$ and $\theta=90^\circ$, respectively. In addition, the pressure drop rises sharply with the increment of Reynolds number. Figure 3c shows the variation of friction factor along the channel over the investigated Reynolds number. The value of friction factor for $\theta=180^\circ$ phase shift is the highest compared to other configurations, while the lowest value is related to $\theta=90^\circ$ phase shift. The maximal friction factor of the wavy-wall channel is about 4 times that of the smooth channel. It is obvious that the friction factor decreases gradually for all configurations with the increase of Reynolds number. Figure 4 depicts the PECs calculated using the computed Nusselt numbers and friction factors. The result indicates that for each test channel the values of PEC have quite similar trend in the considered range of Reynolds number. It is seen that the PECs for the channels decrease with increasing Reynolds number, which means that an optimum Reynolds number is corresponding to the maximum PEC for each type of geometry. The optimum Reynolds number is related to $Re=6,000$ for all tested geometries. The PEC of wavy channel with $\theta=90^\circ$ phase shift is found to be the best among all geometries and is about 1.217 at the lowest value of Reynolds number.
3.2. The Effect of Different Wave Amplitudes

The combined effect of wavy amplitude of channel on the heat transfer performance of channels is analyzed. The range of wavy amplitude varies from 0.5 to 1.5 mm.

The variation of Nusselt number with Reynolds number for wavy channel with \( \theta=90^\circ \) phase shift is presented in Figure 5a. As shown in this figure the Nusselt number increases with the increase of Reynolds number. It is observed that the wavy amplitude of \( \alpha=1.5\text{mm} \) has the best heat transfer compared with other amplitudes of \( \alpha=1\text{mm} \) and \( \alpha=0.5\text{mm} \) respectively. This can be explained by a strong turbulence intensity generated by greater wavy amplitude, leading to a rapid mixing of the flow especially at higher Reynolds number. Figure 5b the higher wavy amplitude would cause more turbulent flow which consequently increases the pressure drop. Figure 5c shows the variation of friction factor with Reynolds number for different wavy amplitudes. It is seen that the wavy amplitude of \( \alpha=1.5\text{mm} \) has the highest friction factor and it is followed by \( \alpha=1\text{mm} \) and \( \alpha=0.5\text{mm} \) respectively. Figure 6 illustrates the effect of different wavy amplitudes on PEC index in the investigated range of Reynolds numbers. It is seen that the wavy amplitude of \( \alpha=0.5\text{mm} \) has the highest PEC index in all ranges of Reynolds numbers and the highest value of that is around 1.396 at \( \text{Re}=6000 \). As a result, it is concluded that by using an aspect ratio of \( \alpha=0.5\text{mm} \) would lead to achieve better heat transfer augmentation. Based on the obtained results, the wavy wall channel with the phase shift of \( \theta=90^\circ \) and the wavy amplitude of \( \alpha=0.5\text{mm} \) is the most efficient geometry, thus it is adopted for the rest of studies.

3.3. The Effect of Using Nanofluid

The effect of SiO\(_2\)-water nanofluid with 25 nm nanoparticle diameter and volume fractions of 0, 1, 2, 3 and 4% on the heat transfer enhancement in the terms of average Nusselt number is plotted in Figure 7. The results indicate that the average Nusselt number of SiO\(_2\)-water nanofluid increases with increasing particle volume concentration. According to Eq. (21), nanofluids with higher particle concentration have higher static and dynamic thermal conductivities, which in turn increase the average Nusselt number. For example, in the case
of 4% volume fraction, the average Nusselt number is about 36.5% larger than pure water at the lowest Reynolds number in testing range through the wavy channel.

Figure 8 shows the variation of local Nusselt number along the lower wavy wall for different volume fractions in the range of 0 to 4% and Re=6,000 through wavy channel with $\theta=180^\circ$ and $\alpha=0.5\text{mm}$. It is found that as the nanofluid volume fraction increases, the peak value of the local Nusselt number increases as well. In addition, at the start point of each hump a considerable velocity gradient increase can be observed which leads to a sudden increase in local Nusselt number over that point. On the other hand, the local skin friction coefficient is independent of the nanoparticle volume concentration as shown in Figure 9. However, with increasing the Reynolds number, irregular and random movements of the nanoparticles enhance the energy exchange rates in the fluid with penalty on the wall shear stress.

In their paper choosing the proper geometry was carried out based on the best PEC. Therefore based on the obtained results, the wavy wall channel with the phase shift of $\theta=90^\circ$ and the wavy amplitude of $\alpha=0.5\text{mm}$ using $\text{SiO}_2$-water nanofluid with volume fraction of $\phi=4\%$ at Re=6,000 has the most value of PEC and is the most efficient geometry between all different investigated configurations and models.

4. Conclusion

Numerical study was performed to investigate the thermal and hydraulic characteristics of forced convection nanofluid flow in wavy wall channels for turbulent regime with Reynolds number of 6000 to 18,000. The emphasis was given on the heat transfer augmentation resulting from various factors, which include different phase shifts between the upper and lower walls, different wavy amplitudes, volume fractions of nanoparticle and Reynolds number. Finite volume method was adopted to solve the governing equations throughout this study. According to the results, the wavy wall channel with the phase shift of $\theta=90^\circ$ and the wavy amplitude of $\alpha=0.5\text{mm}$ have the best PEC index with the value of 1.396 at Re = 6000. However, the channel with wavy amplitude of $\alpha=1.5 \text{mm}$ and $\theta=180^\circ$ gives the highest Nusselt number which is accompanied by increased friction factor and pressure drop penalty. It is also found that $\text{SiO}_2$-water nanofluid with 25nm particle diameter and 4% concentration provides the highest values of average Nusselt number compared to pure water due to its higher thermal conductivity. As the flow Reynolds number increases, the peak value of the local Nusselt number and local skin friction coefficient increase considerably.

References


