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Numerical and experimental investigations on the heat transfer enhancement in corrugated channels using SiO$_2$-water nanofluid

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Abstract

In this paper, convective heat transfer of SiO$_2$-water nanofluid flow in channels with different shapes is numerically and experimentally studied over Reynolds number ranges of 400-4000. Three different channels such as trapezoidal, sinusoidal and straight were fabricated and tested. The SiO$_2$-water nanofluid with different volume fractions of 0%, 0.5% and 1.0% were prepared and examined. All physical properties of nanofluid which are required to evaluate the flow and thermal characteristics have been measured. In the numerical aspect of the current work, the governing equations are discretized by using the collocated finite volume method and solved iteratively by using the SIMPLE algorithm. In addition, the low Reynolds number k-ε model of Launder and Sharma is employed to compute the turbulent non-isothermal flow in the present study. The results showed that the average Nusselt number and the heat transfer enhancement increase as the nanoparticles volume fraction increases, however, at the expense of increasing pressure drop. Furthermore, the trapezoidal-corrugated channel has the highest heat transfer enhancement followed by the sinusoidal-corrugated channel and straight channel. The numerical results are compared with the corresponding experimental data, and the results are in a good agreement.

Keywords: Nanofluids, Corrugated channels, Turbulent flow, Heat transfer, Finite volume method

1. Introduction

In spite of using corrugated channels to provide a significant enhancement in thermal performance of the compact heat exchangers, this improvement was insufficient to meet all the industrial requirements. Therefore, research on enhancement technique in such channels have become very prominent. For this purpose, using nanofluids as a cooling fluids in corrugated channels instead of traditional fluids can enhance thermal conductivity of the base fluids and thereby a further improvement in thermal performance of heat exchangers with a more compact design.

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Nomenclature

\( a \) amplitude of corrugated channel, mm

\( A_\text{s} \) surface area, m\(^2\)

\( C_1, C_2, C_\mu \) empirical constant for turbulence model

\( C_p \) specific heat, J/Kg k

\( D_h \) hydraulic diameter, mm (\( D_h = H_{\text{max}} + H_{\text{min}} \))

\( h \) heat transfer coefficients, (W/m\(^2\).\(^o\)C)

\( H \) Height of channel, mm

\( L_c \) Length of corrugated channel, mm

\( L_w \) wavelength of corrugated channel, mm

\( f \) friction factor

\( f_1, f_2, f_\mu \) damping function

\( J \) Jacobian of transformation

\( k \) turbulent kinetic energy, m/s\(^2\)

\( K \) thermal conductivity, W/m. \(^o\)C

\( m \) mass flow rate, kg/s

\( \text{Nu} \) Nusselt number

\( p \) pressure, pa

\( Q_f \) heat received by fluid, W

\( q_{11}, q_{12}, q_{22} \) geometry factors

\( q_w \) heat flux, W/m\(^2\)

\( \text{Pr} \) Prandtl number

\( \text{Pr}_t \) Turbulent Prandtl number

\( \text{Re} \) Reynolds number

\( T \) temperature, \(^o\)C

\( u, v \) velocities components, m/s

\( U^c, V^c \) contravariant velocities components, m/s

\( x, y \) 2D Cartesian coordinates, mm

\( W \) width of channel, mm

Greek Symbols

\( \zeta, \eta \) body-fitted coordinates

\( \sigma_k, \sigma_x \) empirical constant for turbulence model

\( \phi \) general function

\( \varphi \) volume fraction of nanoparticles,\%

\( \varepsilon \) dissipation rate of turbulent kinetic energy, m\(^2\)/s\(^3\)

\( \mu \) dynamic viscosity, N s/m\(^2\)

\( \mu_t \) turbulent dynamic viscosity, N s/m\(^2\)

\( \rho \) density, kg/m\(^3\)

\( \Delta p \) pressure drop, pa

\( \Gamma \) Diffusion coefficient

Subscripts

\( \text{av} \) average

\( b \) bulk fluid

\( \text{eff} \) effective
Several experimental and numerical studies were conducted on the flow and thermal characteristics of conventional fluids in different corrugated channels [1-9]. The researchers have found that the convective heat transfer in such channels could enhance the heat transfer rate at the expense of higher pressure drop.

On other hand, the convective heat transfer of nanofluids in straight channels have been numerically and experimentally investigated by many researches. Rostamani et al. [10] presented the numerical investigation of the flow and heat transfer characteristics of nanofluids in straight channel. Three different types of nanofluids (CuO, Al2O3 and TiO2-water) have been examined. Reynolds number range of 20000-100000 and nanoparticles volume fraction of 0-6% were considered. The average Nusselt number for CuO-water nanofluid was higher than the Al2O3 and TiO2-water but the shear stress was higher as well. Bianco et al. [11] have numerically studied on the turbulent forced convection of Al2O3-water nanofluid flow in a circular tube. The study was conducted for Reynolds number range of 10^4-10^5 and particles volume fraction of 1%, 4% and 6%. Results showed that the heat transfer coefficient for nanofluid was higher than that of the base fluid. The enhancement in heat transfer increased with Reynolds number and particles volume fraction. Namburu et al. [12] have investigated numerically the turbulent convective heat transfer of nanofluids in a circular tube. Different types of nanofluids such as CuO, Al2O3 and SiO2 in the mixture of water and ethylene glycol have been examined. It was found that the convective heat transfer coefficient increases with increase of Reynolds number and nanoparticles volume fraction and decrease of the size of particles. However, the pressure drop increases with increase of nanoparticles volume fraction. Bayat and Nikseresht [13] have carried out a numerical investigation on the heat transfer enhancement and pressure drop of convective turbulent flow in circular tube using Al2O3 in water and ethylene glycol. It was found that at a particular Reynolds number, Nusselt number increases with the increase of nanoparticles volume faction but the pumping power increases as well.

Furthermore, there are only few numerical and experimental studies have been done on the heat transfer and flow characteristics of laminar nanofluid in corrugated channel. For example, Heidary and Kermani [14] have numerically studied on the heat transfer enhancement of
nanofluid in a sinusoidal channel. They observed that the enhancement in heat transfer increases with respect of the Reynolds number, the amplitude of channel as well as the particles volume fraction. Ahmed et al. [15] have investigated numerically of the laminar forced convection flow of copper-water nanofluid in triangular-corrugated channel. Results showed that the average Nusselt number increases when the volume fraction of nanoparticles as well as Reynolds number increase. Ahmed et al. [16] have conducted a numerical investigation of the laminar forced convective of copper-water nanofluid in a sinusoidal channel using finite difference method (FDM). They observed that the heat transfer enhancement increases with the amplitude of channel, nanoparticle volume fraction and Reynolds number. Pandey and Nema [17] have experimentally studied the convective heat transfer of \( \text{Al}_2\text{O}_3 \)-water nanofluid flow in wavy-plate heat exchanger. Their results showed that the heat transfer rate increases with the increase of Reynolds and Peclet numbers. Also, the required pumping power increases as the nanoparticles volume fraction increase. Recently, Ahmed et al. [18] have numerically investigated the heat transfer enhancement and pressure drop of laminar nanofluid flow in trapezoidal-corrugated channel using finite volume method (FVM). Their results reveal that the heat transfer rate enhances when the nanoparticle volume fraction, Reynolds number and the amplitude of channel increase at the expense of increasing pressure drop. Khoshvaght-Aliaabadi [19] have experimentally investigated the effects of the plate-fin channels geometries on thermal-hydraulic performance using copper-water nanofluid. Their experiments were performed for different geometries of plate-fin channels, including plain, vortex generator, pin, offset strip, louvered, perforated and wavy. It was observed that heat transfer coefficient and the pressure drop for all channels increase with increasing the nanoparticles volume fraction and volumetric flow rate. Also, heat transfer coefficient and the pressure drop for the plain channel are lower than the other channels.

To the best knowledge of authors, all the numerical studies on the convective heat transfer of nanofluids in corrugated channels were only focused on the laminar flow regime. In addition, the convective heat transfer of nanofluid in trapezoidal-corrugated channels has never experimentally studied. Therefore, this paper aims to investigate numerically and experimentally the turbulent forced convection flow of \( \text{SiO}_2 \)-water nanofluid in corrugated channels.

2. Nanofluid preparation

In this paper, nanoparticles of \( \text{SiO}_2 \) with the average diameter of 30 nm (Purchased from Beijing Deke Daojin Science And Technology Co., Ltd.) was used to prepare the nanofluid. The scanning electron microscope (SEM) of \( \text{SiO}_2 \) nanoparticles is depicted in Fig. 1. The amount of nanoparticles required to prepare the nanofluid with two different volume fractions (0.5% and 1.0%) is calculated. Then the nanoparticles were mixed with distilled water. The mixture of nanoparticles and distilled water were
continuously sonicated with ultrasonic bath (Fisher Scientific, Model FB15051). The SiO$_2$-water nanofluid prepared in this study was stable and uniform during all the experiments. Furthermore, no surfactant was added to nanofluid because of may have some effects on the physical properties of nanofluid [20].

3. Thermophysical properties of nanofluids

The properties of nanofluids such as density, viscosity, thermal conductivity and the specific heat were measured in present study. Therefore, the thermal conductivity and the viscosity of the nanofluids were measured using KD2 Pro thermal properties analyzer (Decagon devices, Inc., USA) and Brookfield LVDV-III Ultra Rheometer, respectively. Furthermore, the density was measured using density meter (DA-130N, Kyoto Electronics). Moreover, a differential scanning calorimeter (PerkinElmer model DSC 4000) was used to measure the specific heat of nanofluids. All the measured properties of nanofluids and their base fluid are presented in Table 1.

4. Experimental procedure and theoretical analysis

4.1 Experimental setup

Fig. 2 shows the photograph of the experimental setup of the present study. It is mainly includes the water chiller, test section, thermocouples, plate heater, flow meter, differential pressure transducer, data logger, power regulator and multi meter. However, the chiller consists of pump, condensing unit and working fluid tank.

A 0.4 HP pump was used to driven the working from tank of 8 liter capacity to flow through the test section. Two electrical heater plates with maximum power for each heater was 320 W were used to heat the top and lower walls of the test section. These heaters were attached to the rear faces of the top and bottom walls of the test section. In order to prevent heat transfer from the test section to environment, two layers of fibre-glass insulation (with 5cm thick) surrounded the test section. However, the electrical heaters were connected to AC power regulator unit (W5 SERIES, SPINE) which was used to control the input voltage and current to the heaters. A digital multimeter (BK PRECISION, 2831C) was connected to the circuit of electrical heaters to measure the current (with accuracy of 1% FS) and the voltage (with accuracy of 0.5% FS) supplied to the electrical heaters.

A data logger (simex, MultiCon CMC-99) was connected to thermocouples (k-type with accuracy of 0.2 °C) to measure the wall and the bulk fluid temperatures. Six of these thermocouples were fixed at the rear face of the upper wall of channel to measure the temperature distribution along the wall of the test section. These thermocouples were inserted in the holes, which were drilled with 2 mm diameter from the back side of the channel walls, and fixed using thermal epoxy. The holes were centered on
the channel walls and located with 1.5 mm apart from the inner walls of channel. The holes were located at 30, 70, 110, 150, 190 and 230 mm from the inlet of the test section. Furthermore, two thermocouples inserted in the fluid flowing were used to inlet bulk fluid temperature and other two thermocouples were inserted in the exit section to measure outlet temperature of the fluid. All the thermocouples used in current study were pre-calibrated over the different temperature range.

A flow meter (FC-SD70-R15, TOFCO) was connected between the pump and the inlet of the developing section to measure the flow rate of the fluid. It can measure the flow rate over the range of 1-15 LPM with accuracy of ±5.0% FS. A bucket and stopwatch method for measuring the flow rate of the fluid was used for a second check of the flow meter calibration. A bypass line with a valve was used to adjust the flow rate of the pump. A chiller with 1 kW cooling capacity was immersed inside the tank to adjust the temperature of the working fluid enters the test section. Therefore, the temperature of the working fluid, in this study, was 25 °C with deviation of ± 0.2 °C. A differential pressure transducer (PX409-10WDWUI) together with panel meter (DP24-E-230) have been used to measure the pressure drop across the test section. The differential pressure range of the pressure transducer was 0-10 in of water with accuracy of ± 0.08% FS.

In experiments, when the flow reached a steady state condition, the pressure drop across the test section, the flow rate, the bulk fluid temperature at the inlet and the outlet of the test section as well as the wall temperature of the test section were recorded. After performed the experiments for nanofluid with one volume fraction, the system was washed (cleaned) with the pure water, in order to completely remove the nanofluid from system.

4.2 Test section
It consists of the top and bottom (main) walls and two side walls. The top and bottom (corrugated) walls of test section were fabricated from copper plates with dimensions of 8 mm thick, 50 mm wide and 240 mm long. However, the form of corrugations were accomplished by using wire electrical discharge machining (WEDM). The side walls of test section were fabricated from acrylic, 8 mm thick, to reduce heat losses to the environment. Each of them has two axial grooves (along the length of side wall) to prevent the fluid leakage from the test section. The test section was assembled and corrugated wall-side wall junction were sealed using thermal epoxy. Three different shapes of channels such as trapezoidal, sinusoidal and straight channels were fabricated and tested in this study, as shown in Fig.3. However, the average spacing, H_m, between the top and bottom wall was 10 mm, the width of channel, W, was 50 mm, the axial length of the test section was 240 mm, the wavelength of corrugated channels, L_m, was 20 mm and the amplitude of corrugated channel was 2.0 mm. Two adiabatic straight ducts, which are upstream acrylic duct of 800 mm and downstream acrylic duct of 200 mm length, were used in order to create appropriate conditions for the inflow and outflow of the test section.
4.3. Data reduction

The heat received by the nanofluids from the test section can be determined as follows [19]:

\[
Q_f = m_{nf} C_{p,nf} (T_{b,o} - T_{b,in})
\]  

(1)

Therefore, the average heat transfer coefficient can be expressed as follows [19]:

\[
h_{av} = \frac{Q_f}{A_c (T_{w,av} - T_{b,av})}
\]  

(2)

Then, the average Nusselt number is calculated as follows [21]:

\[
Nu_{av} = \frac{h_{av} D_h}{K_{nf}}
\]  

(3)

Where \( D_h \) is the hydraulic diameter of corrugated channel which can be defined as [9]:

\[
D_h = H_{min} + H_{max}
\]  

(4)

The friction factor \( (f) \) can be expressed as [20]:

\[
f = \frac{\Delta p D_h}{L_c \rho_{nf} u_i^2}
\]  

(5)

4.4 Uncertainty analysis

The experimental uncertainties of dependent parameters, such as Reynold number, friction factors and Nusselt number, were estimated in current study based on the Kline and McClintock method [19]. For example, given a dependent parameter, \( R \), as:

\[
R = R(X_1, X_2, ..., X_n)
\]  

(6)

Where \( X_1, X_2 \) and \( X_n \) are independent measured parameters. Therefore, the uncertainty of \( R \) can be calculated as follows:

\[
W_R = \pm \sqrt{\left( \frac{\partial R}{\partial X_1} W_{X_1} \right)^2 + \left( \frac{\partial R}{\partial X_2} W_{X_2} \right)^2 + ... + \left( \frac{\partial R}{\partial X_n} W_{X_n} \right)^2}
\]  

(7)
Therefore, the uncertainty in Reynolds number, friction factor and average Nusselt number were within ±5.1%, ±10.2% and ±6.19%, respectively.

5. Mathematical formulation

5.1. Problem description

The basic channels used in present study are trapezoidal, sinusoidal and straight channels, as depicted in Fig. 3. The top and bottom walls of these channels are subjected to uniform heat flux conditions. The average spacing between these walls is $H_{av}$. The corrugated channel consists of ten corrugation units with amplitude of $a$ and the wavelength of $L_w$. It is assumed that the flow is steady, fully developed, incompressible and two-dimensional. Furthermore, it can also be assumed that the mixture of base fluid (water) and the nanoparticles ($\text{SiO}_2$) are in thermal equilibrium and they flow at the same velocity. The mixture is also assumed as Newtonian fluid.

5.2. Governing equations

In this study, the single-phase approach has been used in the modeling of nanofluid. Therefore, the two-dimensional governing for steady, incompressible flow in terms of Cartesian coordinates are [22]:

Continuity equation:

$$\frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial y}(\rho v) = 0$$

(8)

u-momentum equation:

$$\frac{\partial}{\partial x}(\rho u u) + \frac{\partial}{\partial y}(\rho u v) = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}\left[(\mu + \mu_t)\frac{\partial u}{\partial x}\right] + \frac{\partial}{\partial y}\left[(\mu + \mu_t)\frac{\partial u}{\partial y}\right]$$

$$+ \frac{\partial}{\partial x}\left[(\mu + \mu_t)\frac{\partial u}{\partial x}\right] - \frac{2}{3} \rho k$$

(9)

v-momentum equation:

$$\frac{\partial}{\partial x}(\rho u v) + \frac{\partial}{\partial y}(\rho v v) = -\frac{\partial p}{\partial y} + \frac{\partial}{\partial x}\left[(\mu + \mu_t)\frac{\partial v}{\partial x}\right] + \frac{\partial}{\partial y}\left[(\mu + \mu_t)\frac{\partial v}{\partial y}\right]$$

$$+ \frac{\partial}{\partial x}\left[(\mu + \mu_t)\frac{\partial v}{\partial x}\right] + \frac{\partial}{\partial y}\left[(\mu + \mu_t)\frac{\partial v}{\partial y}\right] - \frac{2}{3} \rho k$$

(10)

Energy equation:
\[
\frac{\partial}{\partial x} (\rho u T) + \frac{\partial}{\partial y} (\rho v T) = \frac{\partial}{\partial x} \left( \frac{K}{\alpha} + \mu_t \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{K}{\alpha} + \mu_t \frac{\partial T}{\partial y} \right)
\]

(11)

In order to determine the turbulent dynamic viscosity (\(\mu_t\)), the Launder-Sharma k-\(\varepsilon\) model is adopted in this study as follows:

Turbulent kinetic energy equation [23]:
\[
\frac{\partial}{\partial x} (\rho u k) + \frac{\partial}{\partial y} (\rho v k) = \frac{\partial}{\partial x} \left( \Gamma_k \frac{\partial k}{\partial x} \right) + \frac{\partial}{\partial y} \left( \Gamma_k \frac{\partial k}{\partial y} \right) + P_k - \rho (\varepsilon + \varepsilon_w)
\]

(12)

Where the dissipation rate at the wall (\(\varepsilon_w\)) is given by:
\[
\varepsilon_w = 2 \mu \frac{\left( \frac{\partial \sqrt{k}}{\partial x} \right)}{\rho} + \left( \frac{\partial \sqrt{k}}{\partial y} \right)^2
\]

(13)

Turbulent kinetic energy dissipation equation [23]:
\[
\frac{\partial}{\partial x} (\rho \mu \varepsilon) + \frac{\partial}{\partial y} (\rho v \varepsilon) = \frac{\partial}{\partial x} \left( \Gamma_\varepsilon \frac{\partial \varepsilon}{\partial x} \right) + \frac{\partial}{\partial y} \left( \Gamma_\varepsilon \frac{\partial \varepsilon}{\partial y} \right)
\]

\[
+ (C_1 f_i P_k - \rho C_2 f_2 \varepsilon) \frac{\varepsilon}{k} + \phi
\]

(14)

Where
\[
\phi = 2 \mu_t \frac{\mu}{\rho} \left[ \left( \frac{\partial^2 u}{\partial x^2} \right)^2 + \left( \frac{\partial^2 v}{\partial x^2} \right)^2 + 2 \left( \frac{\partial^2 u}{\partial x \partial y} \right)^2 + 2 \left( \frac{\partial^2 v}{\partial x \partial y} \right)^2 + \left( \frac{\partial^2 u}{\partial y^2} \right)^2 + \left( \frac{\partial^2 v}{\partial y^2} \right)^2 \right]
\]

(15)

In the above equations, the production rate of the turbulent kinetic energy (\(P_k\)) is defined as:
\[
P_k = \mu_t \left\{ 2 \left[ \left( \frac{\partial u}{\partial x} \right)^2 + \left( \frac{\partial v}{\partial y} \right)^2 \right] + \left( \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 \right\} - \frac{2}{3} k \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} \right)
\]

(16)

Also, the turbulent eddy viscosity is given by [23]:
\[
\mu_t = C_\mu f_\mu \rho \frac{k^2}{\varepsilon}
\]

(17)

The empirical constants as well as the turbulent Prandtl number that appear in the above equations are defined as [23]:
\[
C_\mu = 0.09 \cdot C_i = 1.44 \cdot C_2 = 1.92 \cdot \alpha_k = 1.0 \cdot \alpha_\varepsilon = 1.3 \cdot \text{Pr}_t = 0.9
\]

(18)

Furthermore, the wall-damping functions can be defined as [24]:
\[
f_i = 1.0 \cdot f_2 = 1 - 0.3 \exp (-Re_\tau^2) \cdot f_\mu = \exp \left[ \frac{-3.4}{(1 + 0.02 \text{Re}_\tau)^2} \right]
\]

(19)
And, the turbulent Reynolds number is given by:

\[ \text{Re_t} = \frac{\rho k^2}{\varepsilon \mu} \]  

(20)

The above governing equations are transformed from Cartesian coordinate system \((x, y)\) into body-fitted coordinate system \((\zeta, \eta)\) due to the complex geometry used in this study. So, the transformed governing equations can be written in general form as follows:

\[
\frac{1}{J} \left[ \frac{\partial}{\partial \zeta} (\rho \Phi U') + \frac{\partial}{\partial \eta} (\rho \Phi V') \right] = \frac{1}{J} \left[ \frac{\Gamma_\phi}{J} \left( q_{11} \frac{\partial \phi}{\partial \zeta} - q_{12} \frac{\partial \phi}{\partial \eta} \right) \right] \\
+ \frac{1}{J} \left[ \frac{\Gamma_\phi}{J} \left( q_{22} \frac{\partial \phi}{\partial \eta} - q_{12} \frac{\partial \phi}{\partial \zeta} \right) \right] + S_\phi(\zeta, \eta) 
\]

(21)

Where \(q_{11}, q_{12}\) and \(q_{22}\) are geometry factors, \(J\) is the Jacobian of the transformation these factors can be defined as:

\[
q_{11} = x_\eta^2 + y_\eta^2, \quad q_{22} = x_\zeta^2 + y_\zeta^2, \quad q_{12} = x_\zeta x_\eta + y_\zeta y_\eta, \quad J = x_\zeta y_\eta - x_\eta y_\zeta 
\]

(22)

And \(U'\) and \(V'\) are the contravariant velocity components in \(x\) and \(y\) direction, these velocities can be defined as [25]:

\[
U' = u y_\eta - v x_\eta, \quad V' = v x_\zeta - u y_\zeta 
\]

(23)

In above equations, \(\phi\) is the general variable, \(\Gamma_\phi\) is the diffusion coefficient and \(S_\phi(\zeta, \eta)\) is the source terms. All of these parameters are defined in Table 2.

5.3. Boundary conditions

In order to solve the governing equations, appropriate boundary conditions for all dependent variables must be prescribed on all the boundaries of the computational domain. These boundary conditions are presented as follows [8]:

i. Inlet flow:

\[
u = u_m, v = 0, \quad T = T_m, \quad k = k_m = \frac{2}{3}(I_\phi u_m)^2, \quad \varepsilon = \frac{C\mu^{3/4} k_m^{3/2}}{(0.07D_k)} \]

(24)

ii. At the outlet section:

\[
\frac{\partial u}{\partial \zeta} = 0, \quad \frac{\partial v}{\partial \zeta} = 0, \quad \frac{\partial T}{\partial \zeta} = 0, \quad \frac{\partial k}{\partial \zeta} = 0, \quad \frac{\partial \varepsilon}{\partial \zeta} = 0 
\]

(25)
iii. At the walls of channel:

\[ u = 0, \ v = 0, \ k = 0, \ \varepsilon = 0 \]  \hspace{1cm} (26)

\[ \frac{\partial T}{\partial n} = - \frac{q_w}{k_{\text{eff}}} \]  \hspace{1cm} (27)

5.4. Implementation of numerical solution

In this study, the finite volume method method (FVM) is used for discretization of governing equations. The upwind scheme is used to discretise the convection terms of governing equations, while diffusion terms were discretised using the central differencing scheme. The SIMPLE algorithm was used, for coupling of the velocity and pressure equations, to determine pressure field [27]. The collocated grid arrangement was used in current study, in which all dependent variables are stored at the same control volume. This results in a weak coupling between the velocity components and the pressure. Therefore, Rhie and Chow momentum interpolation method [26] was used to provide a direct link between the velocity and the pressure nodes to avoid the unreal pressure oscillation. Moreover, Poisson equations are employed to develop the computational mesh of the present study. In order to achieve a better convergence behavior, under-relaxation is applied. The computation is terminated when the sum of absolute residual for each variables over computational domain is less than \(1 \times 10^{-5}\).

6. Validation of numerical methods and grid independence test

In order to validate the results obtained from CFD code developed in present study, the average Nusselt number for turbulent convective heat of air flow in triangular-corrugated channel are calculated and compared with the previous experimental results of Elshafei et al. [7] as shown in Fig. 4(a). According to this figure, the results are in good agreement. Moreover, the average Nusselt number for copper-water nanofluid flow in sinusoidal channel was compared with the numerical results of Heidary and Kermani [14]. From Fig. 4(b), it is found that the present results are very close to the previous results. To estimate the required grid size of the present study, the non-dimensional temperature and streamwise velocity at the trough of the eighth wave of the sinusoidal channel have been investigated for different grid sizes at \(Re=2000\) and \(\varphi = 1\%\), as depicted in Fig. 5. It is found that the grid size of 995×101 ensures the grid-independent solution.
7. Results and discussion

Fig. 6 depicts the velocity vectors and isotherms contours for SiO-water nanofluid flow in trapezoidal, sinusoidal and straight channels at Re=2000 and $\varphi = 1\%$. Generally, the velocity vectors and the isotherms contours for all channel geometries are symmetric about the axial direction. From the velocity vectors, it can be clearly seen that the reversal flow (i.e. recirculation flow regions) appears in the troughs of trapezoidal and sinusoidal channels, while the flow in the straight channel is regular. In other word, there is no reversal flow in straight channel. However, the re-circulation regions that appear in corrugated channels can improve the mixing of the hot fluid near to the channel walls with the core (cold) fluid. As a results, the thermal boundary layer in corrugated channels is thinner than that for the straight channel and hence the temperature gradients near the heated-walls is higher, as shown in Fig. 6 (b).

The average Nusselt number versus Reynolds number for trapezoidal and sinusoidal-corrugated channels at different volume fractions (0, 0.5 and 1%) is shown in Fig. 7. As expected, the average Nusselt numbers for both trapezoidal and sinusoidal channels increase with increasing Reynolds number, at a given volume fraction. Also, it is found that the average Nusselt number increases as the volume fraction of nanoparticles increase due to the addition nanoparticles to the base fluid which can improve thermal conductivity of base fluid (hence heat transfer rate). Furthermore, it is found that for trapezoidal and sinusoidal-corrugated channels, the average deviation between numerical and experimental results are approximately 7.8% and 7.3%, respectively, which display good agreement between these results.

Fig. 8 depicts the pressure drop obtained from experiment and simulation at different Reynolds number and volume fractions for both trapezoidal and sinusoidal-corrugated channels. According to this figure, the pressure drop increases as Reynolds number increases, at a particular value of nanoparticles volume fraction. It is also observed that the pressure drop increases with increasing volume fractions of nanoparticles. This is due to the fact that when the volume fraction of nanoparticles increases, the viscosity of nanofluids increases and therefore it leads to increase the pressure drop. The results are similar to those obtained by Fotukian and Nasr Esfahany [28]. Furthermore, the effect of the volume fraction on the pressure drop increases with Reynolds number. Moreover, the average deviation between the experimental and numerical pressure drop for trapezoidal and sinusoidal channel are 7.4% and 7.1, respectively. The results are in a good agreement.
The ratio of the average Nusselt number of nanofluid flow in corrugated channels to that of the distilled water (\( \varphi = 0\% \)) flow in straight channel at different Reynolds numbers and volume fractions is given in Fig. 9. It should be noted that at \( \varphi = 0\% \), the enhancement ratio for both trapezoidal and sinusoidal channels increase with increasing Reynolds number because the fluid mixing in corrugated channels strongly depends on Reynolds number. Also, it can be clearly seen that enhancement ratio increases with the volume fraction of nanoparticles due to enhance the thermal conductivity of the base fluid. Therefore, at \( \varphi = 1\% \), both trapezoidal and sinusoidal channels display the highest enhancement in heat transfer over Reynolds number range.

Fig. 10 illustrates the variation of the average Nusselt number with Reynolds number for trapezoidal, sinusoidal and straight channels at \( \varphi = 1\% \). In general, the average Nusselt number increases with increasing Reynolds number for all channels shapes because of the temperature gradient on the walls of channel increases when Reynolds number increases. It can also be seen that that the trapezoidal channel has the highest average Nusselt number, while the straight channel has the lowest Nusselt number. This is to be expected since the re-circulation regions appeared in corrugated channels are able to improve the fluid-mixing within these channels. Similar trend is observed for both numerical and experimental average Nusselt numbers.

Fig. 11 presents the pressure drop for trapezoidal, sinusoidal and straight channels with different Reynolds numbers at \( \varphi = 1\% \). It should be noted that the pressure drop, obtained from both experiment and simulation, increases with increasing in Reynolds number for all channels shapes. Furthermore, the trapezoidal channel has the greatest pressure drop followed by the sinusoidal channel due to intensity of the re-circulation regions that appear in such channels in addition to the effect of the sharp edges of the corrugated channel. Also, it found that the straight channel has the lowest pressure drop because of there is no reverse flow in such channel.

Fig. 12 shows the ratio of the average Nusselt number for nanofluid flow in various channels shapes in comparison to that for the distilled water flow in a straight channel at different Reynolds number and \( \varphi = 1\% \). It can be observed that the enhancement ratio for all shapes of channels increase with Reynolds number up to 3000. While, the enhancement ratio is slightly decrease when Reynolds number is beyond 3000. It is also found that the trapezoidal-corrugated channel provides the highest enhancement ratio followed by sinusoidal and straight channels. Therefore, the peak values of the enhancement ratio for trapezoidal channel obtained from computation and experiment are 5.5 and 5.8, respectively.
8. Conclusion

In this paper, heat transfer enhancement and pressure drop of SiO$_2$-water nanofluid flow in different channels shapes are numerically and experimentally investigated over Reynolds number range of 400-4000. The effect of nanoparticles volume fraction and channel shape on the average Nusselt number, pressure drop as well as the heat transfer enhancement are presented and analyzed for different Reynolds number. Both numerical and experimental results show that the average Nusselt number and the enhancement in heat transfer increase as the nanoparticles volume fraction increases, but the pressure drop penalty also increases. In addition, it is observed that the trapezoidal-corrugated channel has the highest average Nusselt number, pressure drop and heat transfer enhancement followed by the sinusoidal-corrugated channel and straight channel. Thus, the trapezoidal-corrugated channel is numerically and experimentally recommended as the best channel to achieve the highest thermal performance with more compact design.

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Caption of Figures:

Fig. 1. SEM photograph of SiO₂ nanoparticles.

Fig. 2. Schematic diagram of the experimental setup.

Fig. 3. Physical domain of the present study (a) trapezoidal-corrugated channel (b) sinusoidal-corrugated channel.(c) straight channel.

Fig. 4. Comparison of the results for (a) air flow in triangular-corrugated channel, (b) copper-water nanofluid flow in sinusoidal-corrugated channel.

Fig. 5. Grid independence test (a) non-dimensional streamwise velocity, (b) non-dimensional temperature.

Fig. 6. Velocity vectors (left) and isotherms contours (right) at Re=2000, φ = 1.0% for (a) trapezoidal-corrugated channel, (b) sinusoidal-corrugated channel, (c) straight channel.
Fig. 7. Average Nusselt number vs. Reynolds for (a) trapezoidal-corrugated channel; (b) sinusoidal-corrugated channel.

Fig. 8. Pressure drop vs. Reynolds for (a) trapezoidal-corrugated channel, (b) sinusoidal-corrugated channel.

Fig. 9. Average Nusselt number enhancement ratio vs. Reynolds for (a) trapezoidal-corrugated channel, (b) sinusoidal-corrugated channel.

Fig. 10. Average Nusselt number vs. Reynolds for various shapes of channels at $\phi = 1\%$.

Fig. 11. Pressure drop vs. Reynolds for various shapes of channels at $\phi = 1\%$.

Fig. 12. Average Nusselt number enhancement ratio vs. Reynolds for various shapes of channels at $\phi = 1\%$.

Table 1. Thermophysical properties of distilled water and SiO$_2$-water nanofluid with different volume fractions at 298 K.

| $\phi$ | $\rho$ (kg/m$^3$) | $C_p$ (J/kg K) | $K$ (W/m K) | $\mu$ (kg/m s) |
|-------|------------------|----------------|-------------|---------------|
| 0     | 998.4            | 4144.0         | 0.608       | 0.001         |
| 0.5   | 1003.1           | 4001.2         | 0.615       | 0.000103      |
| 1.0   | 1007.8           | 3938.5         | 0.615       | 0.000107      |

Table 2. Parameters of general transport equation (Eq.21).

| Equations            | $\phi$ | $\Gamma_{\phi}$ | $S_{\phi}(\zeta, \eta)$ |
|----------------------|--------|-----------------|------------------------|
| continuity           | $\rho$ | 0               | 0                      |
| u-momentum           | $u$    | $\mu + \mu_t$   | $S_u$                  |
| v-momentum           | $v$    | $\mu + \mu_t$   | $S_v$                  |
| Energy               | $T$    | $k / C_p + \mu / \nu_v$ | 0                      |
| Turbulent kinetic energy | $k$ | $\mu + \mu_t / \sigma_k$ | $P_k - \rho (\varepsilon + \varepsilon_v)$ |
| Energy dissipation rate | $\varepsilon$ | $\mu + \mu_t / \sigma_\varepsilon$ | $(C_1 f_1 P_k - \rho C_2 f_2 \varepsilon) \varepsilon / k$ |
Fig. 1.

Fig. 2.
Fig. 3.

(a) Reynolds number
Average Nusselt number
3000 4500 6000 7500 9000
10
20
30
40
50
60
70
80

(a) $a = 0.0$ (present study)

(a) $a = 1.0$ mm (present study)

(a) $a = 0.0$ (Elshafei et al. [7])

(a) $a = 1.0$ mm (Elshafei et al. [7])

(b) $H_{\text{avg}}$

(c) $H_{\text{max}}$ $H_{\text{min}}$

L_w

2a

2a

H_{\text{avg}}
Fig. 4.

(b) Reynolds number
Average Nusselt number
200 400 600 800 1000
5
10
15
20
25
30
35
0% (present study)
10% (present study)
0% (Heidary and Kermani [14])
10% (Heidary and Kermani [14])

(a) Non-dimensional streamwise velocity
y (m)
-0.5 0 0.5 1 1.5 2 2.5
-0.006
-0.004
-0.002
0
0.002
0.004
0.006
597X51
697X71
797X91
995X101
1193X111

Fig. 4.
Fig. 5.

Non-dimensional temperature

y (m)

0.006
0.004
0.002
0
-0.002
-0.004
-0.006

0 0.25 0.5 0.75 1

(b)

597X51
697X71
797X91
995X101
1193X111

Fig. 6.

Non-dimensional temperature

(b)

597X51
697X71
797X91
995X101
1193X111

(a)

(b)

(c)
Fig. 7.
Fig. 8.
Fig. 9.
Fig. 10.

Fig. 11.
Fig. 12.