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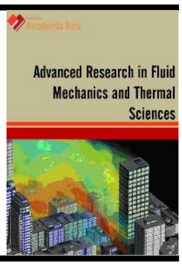

Heat Transfer Enhancement in Semicircle Corrugated Channel: Effect of Geometrical Parameters and Nanofluid

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ABSTRACT

Numerical investigations of forced turbulent convective flow and heat transfer in a symmetry semicircle-corrugated channel with SiO_2-water nanofluid as the working fluid are carried out in this investigation. The algebraic continuity, momentum, and energy equations were solved by means of a finite volume method (FVM). The top and bottom walls of the corrugated channel are heated at constant heat flux boundary conditions. The effects of geometrical parameters including corrugation height and pitch on the thermal and hydraulic characteristics are studied. The corrugated channel with four different corrugated heights of 0.0, 1.5, 2.0 and 2.5 mm with a different corrugation longitudinal pitch of 15.0, 25.0 and 35.0 mm are tested. This investigation covers Reynolds number and nanoparticles volume fraction in the range of 10000–30,000 and 0-8%, respectively. The results show that the average Nusselt number enhances with increase in Reynolds number and with the height of the corrugated channel, but this enhancement accompanied by increases in pressure drop. In addition, as the corrugation pitch decreases, the average Nusselt number increases and the pressure drop decreases. The numerical results indicate that the corrugated height of 2.5 mm with a corrugation longitudinal pitch of 15.0 mm are the optimum parameters and have shown significant improvement in heat transfer.

Keywords:
Enhancement, Turbulent flow, Corrugated channel, Nanofluids, Finite volume method, Geometrical parameters.

1. Introduction

In recent years, corrugated channels are a popular heat transfer augmentation device used in different heat- exchanging channels such as the internal cooling channels in gas turbine blades. Using this technique a significant enhancement in the flow mixing between hotter fluid layers near channel wall and cooler fluid layers in core region is demonstrated. In other words, the flow disturbance caused by the corrugations greatly increases the production of turbulent kinetic energy, which enhances turbulent heat transfer in the channel.

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For further augmentation in the thermal performance of heat exchangers, the nanofluids with different types can be used as coolant in these devices instead of conventional fluids. The flow and heat transfer in a corrugated channel with different configurations using traditional fluids as working fluids have been numerically and experimentally investigated by many researchers [1–10].

Xuan and Roetzel [11] analyzed the heat transfer mechanism in nanofluids based on single-phase and two-phase flow approaches. In this study, several correlations for heat transfer in nanofluids have been derived. Xuan and Li [12] experimentally investigated the heat transfer and flow of Cu–water nanofluid in a tube. The results showed that the enhancement in heat transfer increased as the concentration of solid particles increased.

Azwadi and Adamu [13] numerically studied the convective heat transfer over Reynolds numbers range of 10,000– 120,000 for hybrid nanofluid flow in a circular tube subjected to constant heat flux. In this study, a remarkable improvement was observed in using the hybrid nanofluid due to synergistic effect. At 1% volume fraction of Ag/HEG 34.34% and 38.72% enhancement was recorded at Reynolds number of 60,000 and 40,000 respectively.

Application of the alumina-water nanofluid in a counter flow corrugated plate heat exchanger was experimentally investigated by Pandey and Nema [14]. It was observed that the heat transfer performance improves with increasing both Reynolds number and Peclet number and decreasing the nanofluid concentration.

Sidik and Alawi [15] conducted numerical simulations for turbulent mixed convection heat transfer in concentric annulus using various nanorefrigerants. The results indicated that SiO_2 has the greatest Nusselt number followed by Al_2O_3, ZnO, CuO and lowest values for pure refrigerant.

The effect of nanoparticle's diameter on laminar mixed convection flow in a circular curved tube has been numerically investigated by Akbarinia and Laur [16]. The numerical results showed that as the diameter of the nanoparticle increases, Nusselt number as well as the secondary flow decrease. While the axial velocity augments when the diameter of nanoparticles increases.

An experimental study on the forced convective flow of different nanofluids through a corrugated channel was performed by Khoshvaght-Aliabadi *et al.,* [17] at the constant wall temperature condition. Effects of different factors including nanoparticles weight fraction (0.1–0.4%), type of nanoparticles (Cu, SiO_2, TiO_2, ZnO, Fe_2O_3, Al_2O_3, and CuO), and base fluid material (water-ethylene glycol mixture) were examined.

Santra *et al.,* [18] numerically investigated the heat transfer and flow characteristics of copper–water nanofluid in a horizontal duct using the finite volume method. It was observed that the rate of heat transfer increases with the Reynolds number and with the concentration of the nanoparticles as well.

Fotukian and Nasr Esfahany [19] experimentally conducted on the turbulent forced convection of a γ-Al_2O_3/Water nanofluid in a circular tube. Their results showed that the addition of small amounts of solid particles to the base fluid enhanced the heat transfer, but accompanied by increasing the pressure drop.

Ajeel and Salim [20] examined the impact of Al_2O_3- water nanofluid on the heat transfer and friction factor in semi-circular corrugated channel numerically. Symmetry configuration for corrugation profile of semicircle has been employed. They reported from the obtained results that adopted geometry of semicircle corrugated profile with nanofluid can contribute to improving the efficiency of heat transfer devices. In addition, the outcomes of the study showed that the increased ratio in Nusselt number was 2.07 at Re= 30000 and volume fraction 6%.

Raisi *et al.,* [21] numerically studied the laminar forced convection of Cu–water nanofluid flow in a microchannel. Results showed that the coefficient of slip velocity and nanoparticle volume fraction had great influence on the rate of heat transfer at high Reynolds numbers. The laminar forced

convection flow of nanofluid in a sinusoidal-wavy channel was numerically studied by Heidary and Kermani [22] using finite volume method. Results displayed that the heat transfer rate increased with an amplitude of the wavy channel, nanoparticles volume fraction and Reynolds number.
Ajeel *et al.,* [23] performed a numerical study to test four types of trapezoidal corrugated channel by employing four types of nanofluid. The authors claimed that corrugation profile has a significant impact on the thermal performance as well as the latter has been increased with using SiO_2-water nanofluid compared with another tester nanofluid.

A numerical study on the forced convection of Al_2O_3-water nanofluid in the ribbed channel has been conducted by Manca *et al.,* [24]. It was found that the heat transfer rate enhanced with the concentration of nanoparticles as well as Reynolds number. But this enhancement in heat transfer was accompanied by increasing the pressure drop penalty.

Ahmed *et al.,* [25] investigated numerically the laminar forced convection flow in a triangular-corrugated channel using copper–water nanofluid. Results indicated that the enhancement in heat transfer increased with the nanoparticle volume fraction and with Reynolds number.

Recently, Ajeel *et al.,* [26] numerically studied the heat transfer enhancement of different types of nanofluid in a trapezoidal corrugated channel. The study tested symmetry and zigzag forms of trapezoidal channel. The numerical results showed that the symmetry profile of trapezoidal-corrugated channel has a great effect on the thermal performance compared with a straight profile and zigzag profile. Also, the best improvement in heat transfer among the nanofluids types was by SiO_2-water.

As summarized above, many researchers have investigated various aspects of corrugated channel geometry to enhance heat transfer. However, the heat transfer performances of many other corrugated shapes have not yet been reported. In addition, very little studies have been done to investigate the impact of geometrical parameters through the corrugated channels using nanofluids. Thus, the current study attempts to full fill the existing gap by studying the influence of the geometrical parameters on heat transfer rate and pressure drop as well as thermal performance factor in semicircle corrugated channel using SiO_2-water nanofluid. The numerical simulations are carried out by solving the governing equations using the finite volume approach for Reynolds number and nanoparticle volume fractions with ranges 10000–30,000 and 0–8.0%, respectively. The outcomes from this study on the influence of geometrical parameters should find its employ in many industrial and natural processes in which the knowledge on the heat transfer behavior is of uttermost importance.

2. Problem Description

Figure 1 illustrated the 2-dimensional geometrical model of the present study (a), and (b) geometry of the semicircle- corrugated channel (test section).

The basic geometry of the present study is shown in Figure 1. It consists of two walls which are the upper and lower walls and the average spacing between these walls (H) is 10 mm while the channel width (W) is (5H). The channel is composed of three sections; upstream-adiabatic section, corrugated-heated section, and downstream-adiabatic section. The length of the upstream section is 4times of the downstream section while the length of the corrugated wall is double length of the downstream wall. The corrugated height and the axial length of each pitch are (h) and (p), respectively. It can be assumed that the flow is fully developed, turbulent, incompressible, three-dimensional and steady. Furthermore, nanofluid is assumed as a Newtonian fluid. Moreover, the mixture of water and nanoparticles of SiO_2 is homogenous and enters the channel at the same flow and thermal conditions.

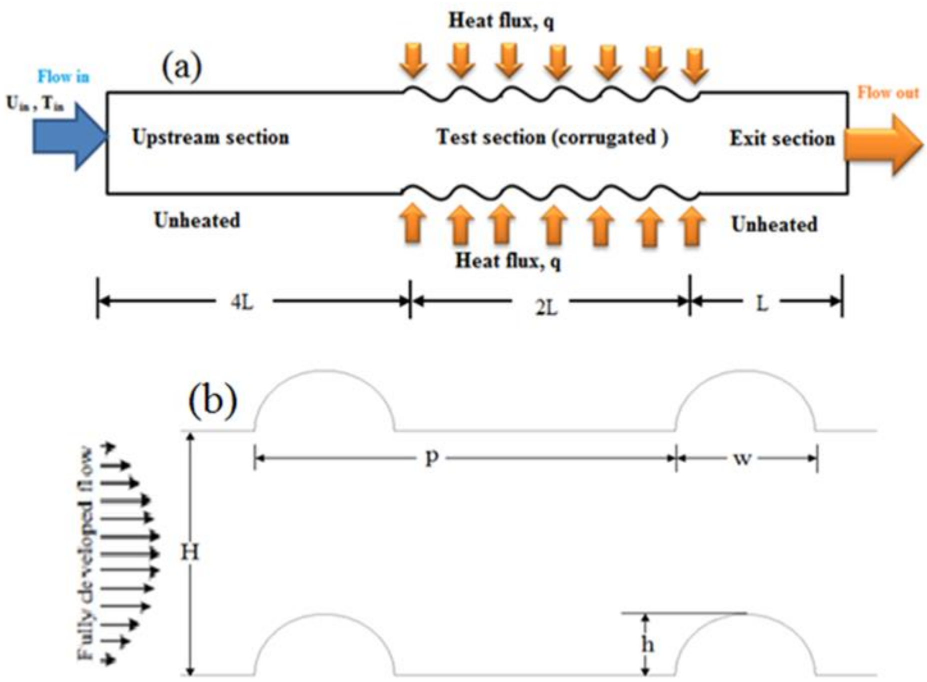

Fig. 1. Schematic diagram (a) Computation domain, (b) Test section

3. Governing Equations

The governing equations of the flow problem are solved for three-dimensional flow as follows

Continuity equations

$$\nabla.(\rho_f V) = 0 \tag{1}$$

Momentum equation

$$\nabla.(\rho_f\, VV) = -\nabla p + \nabla.\tau \tag{2}$$

Energy equation

$$\nabla.\left(\rho_f V C_{p,f} T\right) = \nabla.(k_f \nabla T - C_{p,f}\rho_f\ \overline{\mathrm{v}}\mathrm{t}) \tag{3}$$

The current study used the k − ε turbulence model suggested by Launder and Spalding [27]. Also, the model takes into account the impact of mean velocity gradients which lead to generate turbulent kinetic energy and can symbolized G. It can be illustrated by the equations below

$$\nabla.(\rho_m V k) = \nabla.\left(\frac{\mu_{t,m}}{\sigma_k}\nabla k\right) + G_m - \rho_m \varepsilon \tag{4}$$

$$\nabla.(\rho_m V \varepsilon) = \nabla.\left(\frac{\mu_{t,m}}{\sigma_\varepsilon}\nabla \varepsilon\right) + \frac{\varepsilon}{k}(C_1 G_m - C_2 \rho_m \varepsilon) \tag{5}$$

where,

$\mu_{t,m} = \rho_m C_\mu \frac{K^2}{\varepsilon}$, $C_1 = 1.44$, $C_2 = 1.92$, $C_\mu = 0.09 = 0.09$, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$.

4. Boundary Conditions

Computational domains and boundary conditions were applied at the semicircle corrugated channel which included velocity inlet condition and temperature of 300K, pressure outlet condition while slip velocity was ignored. Additionally, there was uniform heat flux on the corrugated walls whereas adiabatic condition applied for the remaining walls which are straight. The specific thermal conditions for the complex flow field as well as the boundary conditions can be illustrated as below

The boundary conditions at the inlet

$u = u_{in}$, $v = w = 0$, $T_{in} = 300k$

$k_{in} = \frac{3}{2} (\mathrm{I} u_{in})^2$, $\varepsilon_{in} = {C_\mu}^{3/4} \frac{K^{3/2}}{Lt}$

Outlet boundary: In current study, fully developed for the properties are assumed at the outlet.

$\frac{\partial T_f}{\partial x} = 0, \frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial w}{\partial x} = 0$, and $\frac{\partial k}{\partial x} = \frac{\partial \varepsilon}{\partial x} = 0$

At the wall

$u = v = w = 0$, $q = q_{wall}$

and, average Nusselt number can be defined as below

$$\overline{\mathrm{Nu}} = \frac{\bar{h} D_h}{k_f} \tag{6}$$

And, average heat transfer coefficient as

$$\bar{h} = q'' \cdot \frac{\ln\left(\frac{T_w - T_{m,in}}{T_w - T_{m,out}}\right)}{(T_w - T_{m,in}) - (T_w - T_{m,out})} \tag{7}$$

$$q'' = \dot{m}\, C_p\, (T_{m,in} - T_{m,out})/A \tag{8}$$

where, $T_{m,in}$, $T_{m,out}$ are the average inlet and outlet temperatures of the working fluid while A is the corrugated surface area. Additionally, can obtain the inlet velocity depending on the Reynolds number as below

$$u_{in} = \frac{R_e\, \mu}{\rho D_h} \tag{9}$$

In the corrugated house shape channel, the hydraulic diameter is computed based on cross section area (A_{cross}) and the perimeter of wetted (P) as [25, 28]

$$D_h = \frac{4A_{cross}}{P} \tag{10}$$

From definition of Fanning friction factor as

$$C_{fx} = \frac{2\tau_s}{\rho u_{in}^2} \tag{11}$$

and friction factor is defined [25,28]

$$f = 4C_{fx} \tag{12}$$

The pressure drop can also be obtained as [25,28]

$$\Delta p = f\,\frac{\rho L_{corr} u_{in}^2}{2D_h} \tag{13}$$

For a better evaluation system, the thermal performance factor was computed based on the ratio between the heat transfer enhancements into the increase in pressure drop. The thermal performance factor is given by [28]

$$PEC = \frac{(\overline{Nu}/\overline{Nu_0})}{(f/f_0)^{1/3}} \tag{14}$$

5. Properties of Nanofluid

In present study, the thermophysical properties of SiO_2–water nanofluid are defined as follows

i. Density and heat capacity
The density and heat capacity of nanofluid are given by [24]

$$\rho_{nf} = (1-\phi) + \phi\rho_{np} \tag{15}$$

$$(\rho C_P)_{nf} = (1-\phi)(\rho C_P)_f + \phi(\rho C_P)_{np} \tag{16}$$

ii. Thermal conductivity
In order to compute the effective thermal conductivity by using nanoparticles in corrugated channel, the effect of Brownian motion will be taken into consideration by utilizing the empirical correlation below [29]

$$k_{eff} = k_{static} + k_{Brownian} \tag{18}$$

$$k_{static} = k_f\left[\frac{(k_{np}+2k_f)-2\phi(k_f-k_{np})}{(k_{np}+2k_f)+\phi(k_f-k_{np})}\right] \tag{19}$$

$$k_{Brownian} = 5 * 10^4\beta\,\phi\rho_f C_{p,f}\sqrt{\frac{KT}{2}}\,f(T,\phi) \tag{20}$$

where, Boltzmann constant: k = 1.3807*10^{-32} J/K and β =1.9526$(100\phi)^{-1.4594}$.
Modeling, $f(T,\phi)$

$$f(T,\phi) = (2.8217 \times 10^{-2}\phi + 3.917 \times 10^{-3})\left(\frac{T}{T_0}\right) + (-3.0669 \times 10^{-2}\phi - 3.391123 \times 10^{-3}) \quad (21)$$

iii. Dynamic viscosity
The effective dynamic viscosity of nanofluid is [26]

$$\mu_{eff}=\mu_f * \left(\frac{1*\phi^{-1.03}}{1-34.87(\frac{d_p}{d_f})^{-0.3}}\right) \quad (22)$$

Equivalent diameter of based molecule

$$d_f = [\frac{6M}{N\pi\rho_{f0}}]^{1/3} \quad (23)$$

Table 1 illustrates the thermo-physical properties of SiO_2-water nanofluid $\phi = 0.08$ at T=300K.

Table 1
Thermo-physical properties of nanofluid ϕ=0.08

Thermo-physical properties	SiO_2
Density ρ (kg/m^3)	1094.344
Dynamic viscosity, μ(Ns/m$^{2)}$	0.004795
Thermal conductivity, k(w/m.K)	0.643072
Specific heat, c_p (J/kg.K)	3622.483

6. Implementation of Numerical Solution
6.1 Numerical Method

The finite volume method is used to solve the governing equations with corresponding boundary conditions by employing the CFD commercial software ANSYS-FLUENT-V16.1. The SIMPLE algorithm is employed to join the pressure-velocity system and a 2ndorder upwind scheme was also adopted for the convective terms. The k-ε turbulent model with standard wall function was selected while the diffusion term in the momentum and energy equations is approximated by 2nd order upwind. In the current investigation, the convergence criterion is considered as 10^{-5} for continuity, momentum and turbulence equations while 10^{-10} for energy equation.

6.2 Code Validation and Grid Testing

Figure 2 shows a comparison of the average Nusselt number which is obtained in present study for turbulent air flow in straight and corrugated channels with the experimental work of Elshafei *et al.,* [9]. It is observed that the results are in good agreement.

Five sets of grid distributions are tested on the symmetry semicircle-corrugated channel using water as working fluid to analyze the effects of grid size on the results. They are 162632, 267696, 329668, 413520 and 544235 elements, respectively. By comparing the third, fourth and fifth mesh configurations, in terms of average Nusselt number, the corresponding percentage relative errors are 0.008%. Therefore, the fourth grid case has been adopted to obtain an acceptable compromise between the computational time and the results accuracy.

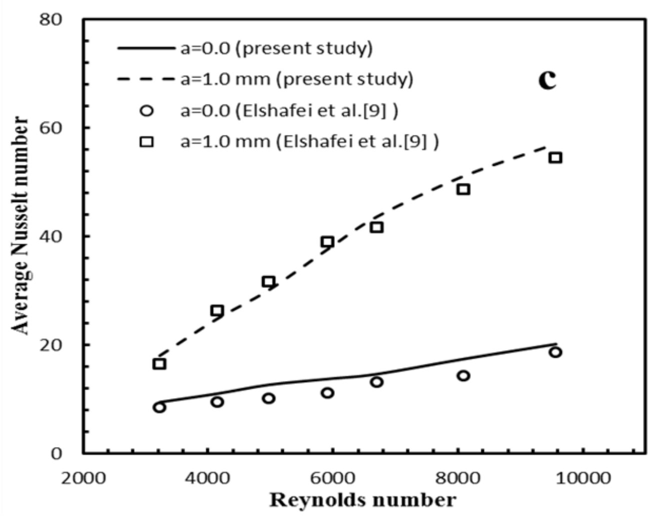

Fig. 2. Comparison between the current work and the outcomes of Elshafei *et al.,* [9]

7. Results and Discussions

Turbulent forced convection flow of SiO_2–water nanofluid in symmetry semicircle-corrugated channel has been numerically investigated over Reynolds number ranges of 10,000–30,000 and nanoparticles volume fraction of 0–8%. Three different values of corrugated height (h=0, 1.5, 2.0 and 2.5) and axial pitch (p=15.0, 25.0 and 35.0) have been considered in the present study.

Figure 3 displays the variation of pressure drop with Reynolds number for different values of corrugated height (p=15.0 and ϕ= 8%). It should be noted that the pressure drop increases as the Reynolds number increases for all values of h. Furthermore, at particular Reynolds number, the pressure drops increases with an increase (h) of the corrugated channel. This is because the flow becomes more disturbed when the corrugated height of corrugated channel increased. On other hand, when the corrugated height of corrugated channel decreases, the spacing between the top and bottom walls of the corrugated channel in throat regions, which have a maximum spacing between the walls, increases and hence the pressure drop will decrease (straight channel).

The variation of the average Nusselt number with Reynolds number for different values of corrugated height at p=15.0 and ϕ=8% is shown in Figure 4. It should be noted that Reynolds number and (h) have a significant effect on the average Nusselt number. At any value of h, the average Nusselt number increases with Reynolds number due to the increases of the convection effect. In addition, the average Nusselt number increases with the corrugated height of corrugated channel at a given Reynolds number. This is because the mixing of fluid in the corrugated channel is improved and the temperature gradient near the wall increases as the value of h increases, especially at high Reynolds number.

The variation of pressure drop with Reynolds number for different axial pitch (p) of the corrugated channel at h=2.5 and ϕ=8% is shown in Figure 5. As expected, at a given Reynolds number, the pressure drop increases as the (p) of corrugated channel increases. This is because when the (p) of the corrugated channel increases, the total length of the channel increases and hence increases the pressure drop. It can be clearly seen that there is a significant effect of axial pitch on pressure drop at Re=30,000 and this effect increases as Reynolds number increases. Moreover, when Reynolds number increases, the pressure drop increases for all values of axial pitch.

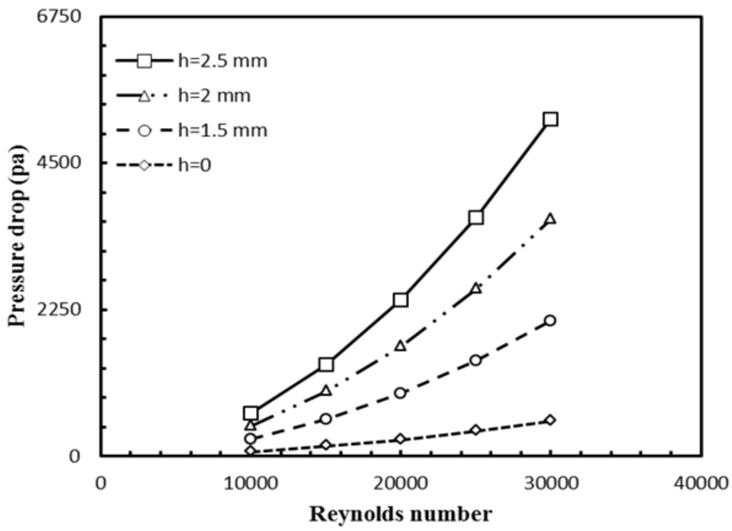

Fig. 3. Pressure drop for various corrugated height

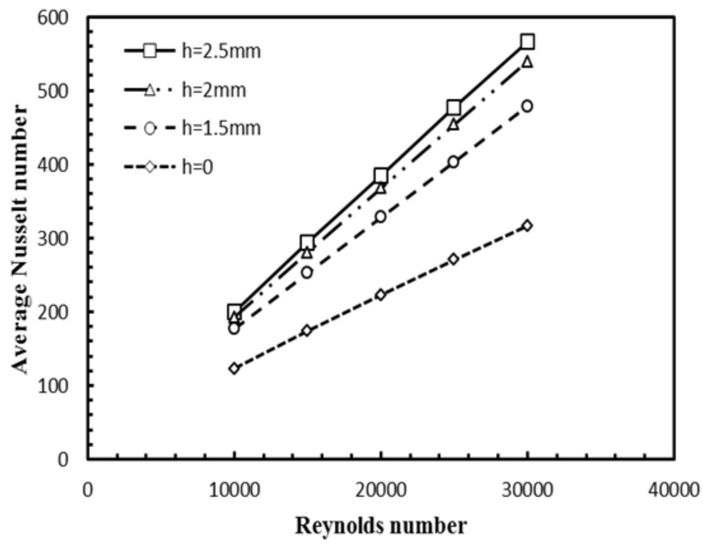

Fig. 4. Average Nusselt number for various corrugated height

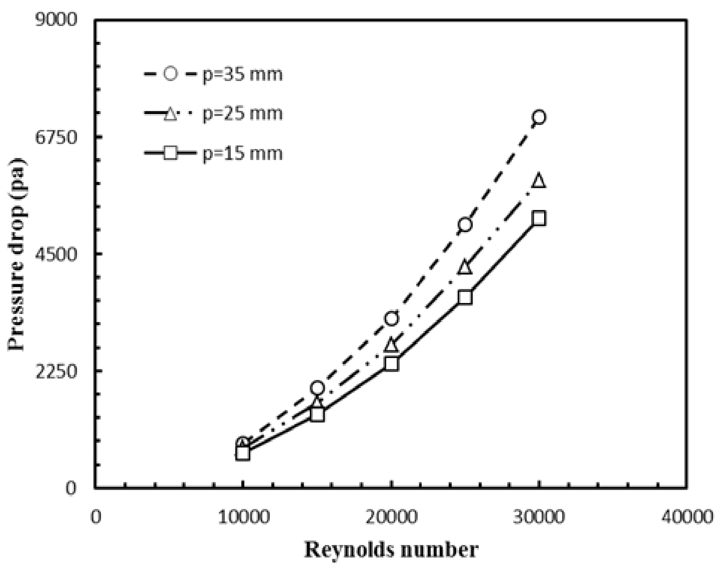

Fig. 5. Pressure drop for various axial pitches

Figure 6 displays the variation of the average Nusselt number with Reynolds number for the different axial pitch of corrugated channel at h=2.5 and ϕ=8%. Generally, the average Nusselt number increases with the Reynolds number for all values of p. Furthermore, the average Nusselt number decreases, when the axial pitch of corrugated channel increases. Because the flow becomes less disturbed when the (p) increases and less fluid mixing in the core with the fluid near the walls of the corrugated channel and consequently, decreases the heat transfer rate.

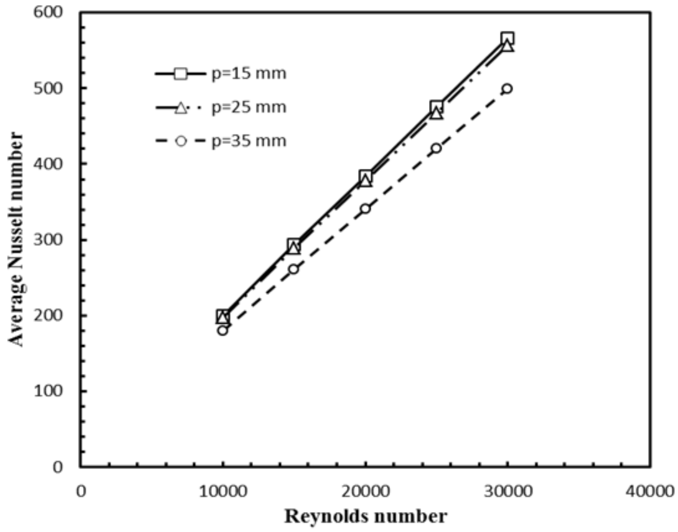

Fig. 6. Average Nusselt number for various axial pitches

The PEC of the three axial pitches in operations with various Reynolds numbers is shown in Figure 7. The result indicates that for each test of axial pitch the values of PEC have a quite similar trend in the considered range of Reynolds number. It is seen that the PECs for the (p) decrease with increasing Reynolds number, which means that an optimum Reynolds number is corresponding to the maximum PEC for a fixed corrugated height. The optimum Reynolds numbers for this case at Re= 10,000. So, it is evident from this figure that, the PECs increase with decreasing axial pitches. Because the increase in heat transfer is more effective than the increase in friction factor for the case of p= 15.0. Therefore, the use of this pitch over the entire range of Reynolds number is the best overall enhancement.

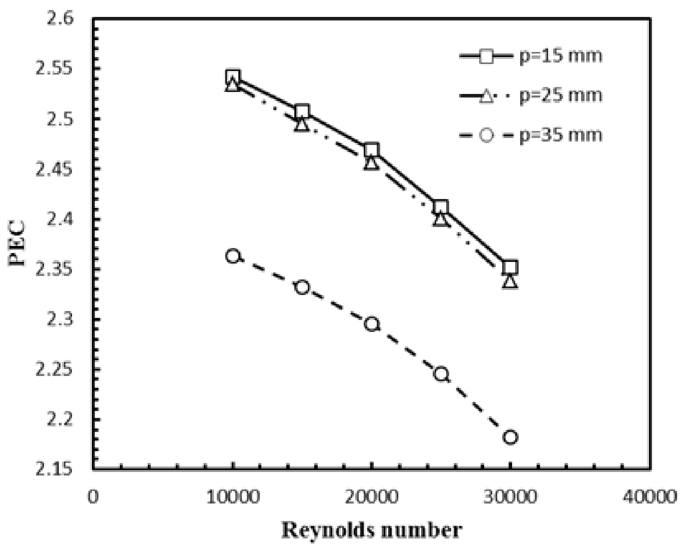

Fig. 7. PEC for various axial pitches

Figure 8 indicates that the PEC variance with the corrugated heights for different Reynolds numbers. It can be seen that the PECs increase with increasing the corrugated height. So there is a best (h) for a fixed Reynolds number. In addition, the result indicates that for each test of corrugated height the values of PEC have a quite similar trend in the considered range of Reynolds number. It is seen that the PECs for the (h) decrease with increasing Reynolds number. The maximum value of PEC recorded at h=2.5 and Re=10,000.

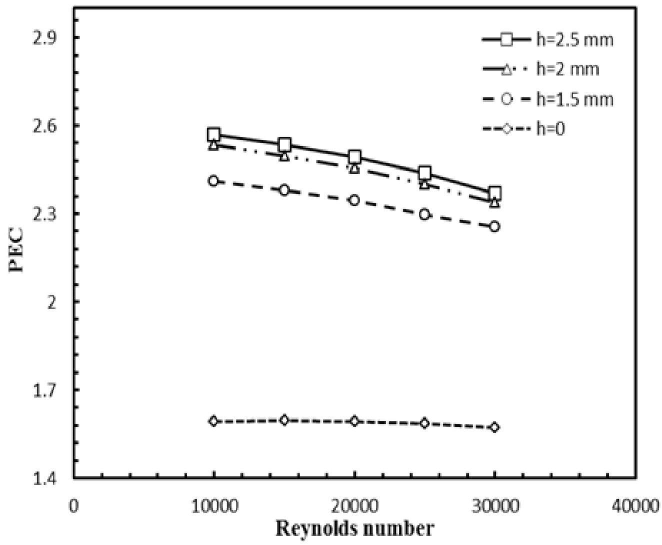

Fig. 8. PEC for various corrugated height

8. Conclusions

In this paper, turbulent forced convection heat transfer of SiO_2– water nanofluid in symmetry semicircle-corrugated channel has been numerically studied using finite volume method. The range of Reynolds number is 10,000–30,000 and nanoparticle volume fraction in the range of 0–8%. The effects of Reynolds number, axial pitch and corrugated height of corrugated channel on the average Nusselt number, pressure drop, and PEC are presented and discussed. Numerical results show that the average Nusselt enhanced with increasing Reynolds number and corrugated height of the corrugated channel, but the pressure drop will also increase. In addition, when the axial pitch of the corrugated channel increases, the average Nusselt number decreases, and the pressure drop will increase. The results show that the corrugated height of 2.5 mm with a corrugation longitudinal pitch of 15.0 mm is the optimum parameters and they have a significant effect on the heat transfer enhancement. Therefore, it can be proposed it to design more compact heat exchangers with a higher thermal performance.

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