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The Effects Of Corrugated Geometry On Flow And Heat Transfer In Corrugated Channel Using Nanofluid

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ABSTRACT
In this paper, a numerical investigation is performed to study the effects of corrugated geometry and nanoparticles volume fractions on heat transfer and pressure drop through corrugated channel under constant heat flux boundary condition. The governing equations for laminar flow are discretized using finite volume method and solved iteratively using SIMPLE algorithm. The investigation covers three different corrugated geometries with Reynolds number and nanoparticles volume fractions in the ranges 50-500 and 0-0.1 respectively. The results indicate that the Nusselt number increased with increasing nanoparticles volume fraction and Reynolds number and pressure drop increased with Reynolds number. It is found that case 3 shows the best thermal-hydraulic enhancement factor in all ranges of Reynolds number compared with other cases.

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INTRODUCTION

The subject of enhancement in heat transfer is very important for heat transfer devices. The corrugated plate is a suitable technique to improve the thermal performance and provide higher compactness in heat exchangers. There are many experimental and numerical studies on the heat transfer and pressure drop in corrugated channel. Laminar forced convection flow and heat transfer studied numerically in a wavy-wall channel by Wang and Chen (2002). The flow and heat transfer in a channel with one-sided corrugated surface investigated numerically by Naphon *et al.* (2007). The flow and heat transfer in sinusoidal and arc-shaped channel were numerically investigated by Bahaidarah *et al.* (2005). The governing equations were solved using finite volume method (FVM). In their study, they found that the enhancement in heat transfer increases with increasing Reynolds number for sinusoidal and arc-shaped channel configurations.

Naphon (2007) studied experimentally the heat transfer characteristics and pressure drop in channel with V corrugated upper and lower plates. In this study investigated corrugated plates with three different corrugated tile angles of 20, 40 and 60 degrees. Result showed that the corrugated surface has significant effects on the enhancement of heat transfer and pressure drop. Islamoglu and Parmaksizoglu (2003 and 2004) investigated numerically and experimentally the effect of channel height on the heat transfer enhancement in a corrugated channel. Mohammed *et al.* (2013) studied numerically forced turbulent convective flow and heat transfer in a corrugated channel. In this study, the corrugated channel walls are heated at constant heat flux boundary conditions. They found that the channel with corrugated angle of 60° and corrugated height of 2.5 mm with channel height of 17.5 mm are the optimum parameters.

All the experimental results have demonstrated the enhancement of the thermal conductivity by addition of nanoparticles. Xuan and Li (2003) experimentally studied the flow and heat transfer characteristics for cu-water based nanofluids through a plain tube with a constant heat flux boundary condition . They found that the nanofluids give substantial enhancement of heat transfer rate compared to pure water. For an up to date review of heat transfer in nanofluids one may refer to Das *et al.* (2003). Ahmed *et al.* (2013) investigated numerically laminar forced convection heat transfer of cu–water nanofluid in trapezoidal corrugated channel under constant walls temperature boundary condition. The effect of geometrical parameters, nanoparticle volume fraction and Reynolds number on the velocity vectors, temperature contours, pressure drop and average Nusselt number investigated. They found that with increasing nanoparticle volume fraction and the amplitude of corrugated channel increases average Nusselt number and increases pressure drop. Santra *et al.* (2009) investigated numerically the heat transfer of cu–water nanofluid in a two dimensional parallel plat. The governing equations for laminar flow were discretized using finite volume method. They found that the average Nusselt number and average wall shear stress increases with the increase in nanoparticle volume fraction. Raisi *et al.* (2011)

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investigated numerically laminar forced convection heat transfer of cu–water nanofluid flow in a microchannel. They found that the velocity and volume fraction of nanoparticle had great influence on Nusselt number at high Reynolds numbers. Experiments on heat transfer due to natural convection with nanofluid have been studied by Putra *et al*. (2003) and Wen and Ding (2006). They have observed that rate of heat transfer decreases with increase in volume fraction of nanoparticles. The viscosity of this nanofluid increases rapidly with inclusion of nanoparticles as shear rate decreases. Ahmed *et al*. (2011) studied numerically heat transfer and pressure drop characteristics of cu-water in corrugated channel with triangular corrugations under constant walls temperature boundary condition. The governing equations solved by Finite Difference (FD) method. They found that the heat transfer enhancement increases with increase in the nanoparticle volume fraction and Reynolds number, while there is slight increase in pressure drop.

In this study, the forced convection heat transfer and pressure drop of cu–water nanofluid in three configurations of triangular corrugated channel is numerically studied for nanoparticle volume fractions and Reynolds number with ranges 50–500 and 0–10%, respectively. The governing equation solved by Finite volume method (FVM). The effects of Reynolds number, nanoparticles volume fraction and configurations of corrugated channel on Nusselt number and pressure drop are investigated.

Mathematical modeling:

Physical model:

The geometries of the present problem have been shown in Figure 1. The geometries consist of two dimensional symmetric triangular corrugated plates with minimum height $(H_{\min})$ and the maximum height $(H_{\max})$ and the ratio of height is $H_{\max}/H_{\min}=6$. The width of the duct is very large compared to the height. The wall of channel is consisting of a flat wall (adiabatic wall) and a corrugated wall (constant heat flux wall). The axial length of each cycle is (S) and the length of each smooth adiabatic wall section is $(2S)$.The total length of corrugated wall is six triangular corrugations. It can be assumed that the flow is steady, fully developed, laminar, incompressible, and two-dimensional. The nanofluid in the channel is Newtonian and assumed that the fluid phase and nanoparticles are in the thermal equilibrium state and they flow with the same velocity.

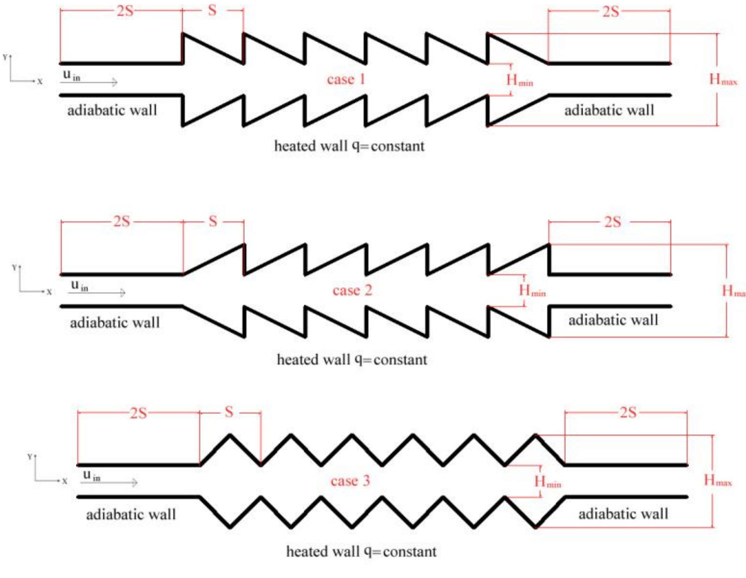


Fig. 1: Schematic diagram of the corrugated channels studied in the present computation

Governing equations and Boundary condition:

The continuity, momentum, and energy equations for the laminar and steady state forced convection in the two-dimensional corrugated channel can be written in dimensional form as follows (Santra *et al*. 2009):

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \qquad (1)$$

$$\rho_{nf}\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\mu_{nf}\nabla^2 u \qquad (2)$$

$$\rho_{nf}(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y})=-\frac{\partial p}{\partial y}+\mu_{nf}\nabla^2 v \tag{3}$$

$$(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y})=\alpha_{nf}\nabla^2 T \tag{4}$$

where u and v are the velocity components in the x and y directions, respectively. p is the pressure, T is the temperature. The thermo-physical properties (Table. 1) of the nanofluid are obtained from the following relations (Khanafer *et al*. 2003):

$$\rho_{nf}=(1-\chi)\rho_f+\chi\rho_s \tag{5}$$

$$(\rho c_p)_{nf}=(1-\chi)(\rho c_p)_f+\chi(\rho c_p)_s \tag{6}$$

$$\frac{k_{nf}}{k_f}=\frac{k_s+2k_f-2\chi(k_f-k_s)}{k_s+2k_f+\chi(k_f-k_s)} \tag{7}$$

$$\mu_{nf}=\frac{\mu_f}{(1-\chi)^{2.5}} \tag{8}$$

$$\alpha_{nf}=\frac{k_{nf}}{(\rho c_p)_{nf}} \tag{9}$$

Table 1: Thermophysical properties of pure fluid and nanoparticles

Property	Fluid phase(water)	Solid phase (Cu)
$C_P\ (J/kg\,K)$	4179	385
$\rho\ (kg/m^3)$	997.1	8933
$k\ (W/m\,K)$	0.613	400

The above equations can be converted to nondimensional forms using the following nondimensional parameters.

$$X=\frac{x}{D_h},\quad Y=\frac{y}{D_h},\quad U=\frac{u}{u_{in}},\quad \theta=\frac{T-T_{in}}{\frac{q''L}{k_{nf}}},\quad P=\frac{p}{\rho_{nf}u_{in}^2} \tag{10}$$

The governing equations are written in the following dimensionless form:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{11}$$

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{1}{\mathrm{Re}_{D_h}}\frac{\rho_f}{\rho_{nf}}\frac{1}{(1-\chi)^{2.5}}\nabla^2 U \tag{12}$$

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{1}{\mathrm{Re}_{D_h}}\frac{\rho_f}{\rho_{nf}}\frac{1}{(1-\chi)^{2.5}}\nabla^2 V \tag{13}$$

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\frac{k_{nf}}{k_f}\frac{(\rho c_P)_f}{(\rho c_P)_{nf}}\frac{1}{\mathrm{Re}_{D_h}\,\mathrm{Pr}}\nabla^2 T \tag{14}$$

The boundary conditions for equations (11)–(14) in the dimensionless form are given by:

At inlet: $U=1, V=0, \theta=0$ (15)

At outlet: $\frac{\partial U}{\partial X}=0,\ \frac{\partial \theta}{\partial X}=0$

At smooth walls: $U=0,\ V=0,\ \frac{\partial \theta}{\partial Y}=0$

At corrugated walls: $U=0,\ V=0,\ q=q_W$

The local Nusselt number at the corrugated walls of channel is defined as:

$$Nu_x = \frac{h(x) D_h}{k_{nf}} \tag{16}$$

Local heat transfer coefficient $h(x)$ can be evaluated as follow::

$$h(x) = \frac{q''}{T_w(x) - T_b(x)} \tag{17}$$

where q'' represents the heat flux, $T_b(x)$ and $T_w(x)$ are the bulk temperature and local wall temperature, respectively.

The average Nusselt number (Nu) at corrugated wall is determined by integrating Nu along the corrugated wall.

$$Nu = \frac{1}{6S}\int_{2S}^{8S} Nu_x \, dx \tag{18}$$

The Reynolds number is defined as:

$$\mathrm{Re}_{D_h} = \frac{\rho_{nf}\, u_{in}\, D_h}{\mu_{nf}} \tag{19}$$

The hydraulic diameter is computed as (Incropera and Dewitt., 2011):

$$D_h = H_{\min} + H_{\max} \tag{20}$$

The average value of Fanning friction factor and friction factor for corrugated channel is (Incropera and Dewitt., 2011):

$$C_f = \frac{2\tau_w}{\rho u_m^2} \tag{21}$$

$$f = 4C_f \tag{22}$$

where the ΔP is pressure drop in the corrugated channel is defined as:

$$\Delta P = f\,\frac{(6S)\rho_{nf} u_{in}^2}{2D_h} \tag{22}$$

To estimate the enhancement of heat transfer between pure fluid and nanofluid, a augmentation average Nusselt number $(Nu,_a)$ is defined as the ratio of Nusselt number at any volume fraction of nanoparticles to that of pure Water that is:

$$Nu,_a = \frac{Nu(\chi)}{Nu(\chi = 0)} \tag{23}$$

The effectiveness of using corrugated channels was evaluated by studying by heat transfer performance ratio because the improvements in heat transfer are also accompanied by increase in the frictional losses. The thermal-hydraulic enhancement factor (η) for pure water can be calculated using the predicted Nusselt numbers and friction factors as follows:

$$\eta = (Nu / Nu_o)(f / f_o)^{-\frac{1}{3}} \tag{24}$$

The value of Nusselt number in smooth channel (Nu_o) and friction factor in smooth channel (f_o) as given by many authors (Incropera and Dewitt., 2011).

Numerical method:

The governing equations, equations (11)-(14), and the associated boundary conditions are solved numerically using the Finite volume method (FVM) and solved iteratively using SIMPLE algorithm (Patankar, 1980). The governing equations are non-linear and coupled; the solution loop must be carried out iteratively in order to obtain a converged numerical solution. Discretization of the momentum and energy equations is performed by a second order upwind scheme and pressure interpolation is provided by Standard scheme (Versteeg and Malalasekera., 1995). Convergence criterion considered as residuals is admitted for momentum, continuity and energy equations it is lower than 10^{-6}.

Grid independent study and validation:

For the grid independence test, five different grid arrangements into the three case corrugated channels are tested. They have mesh layout with approximately 4900, 8000, 13500, 19500 and 29500 nodes, respectively. The forth grid configuration (19500 nodes) confirms the grid-independence and it is therefore used throughout the present study to get an acceptable compromise between the computational time and the result accuracy (see Figure 2).

To validate the numerical algorithm used in the present study, the local Nusselt number for cu-water flow with 5% nanoparticls volume fraction through a case3 was calculated and compared with the numerical results of Ahmed *et al*. (2011).

The comparison shows that the results are in a good agreement as shown in Fig. 3a. Also, Fig. 3b shows the comparison of the present numerical results for Nusselt number through a smooth channel for pure water and cu-water flow with 5% nanoparticls volume fraction with the results of Santra *et al*. (2009). The comparison shows that the results are also in a good agreement.

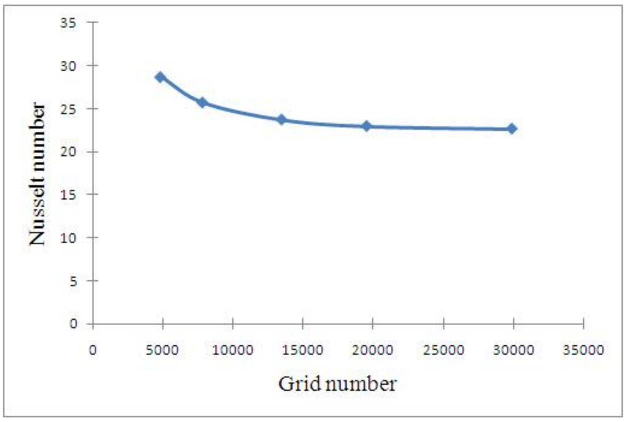


Fig. 2: Grid point independence study for Re=500

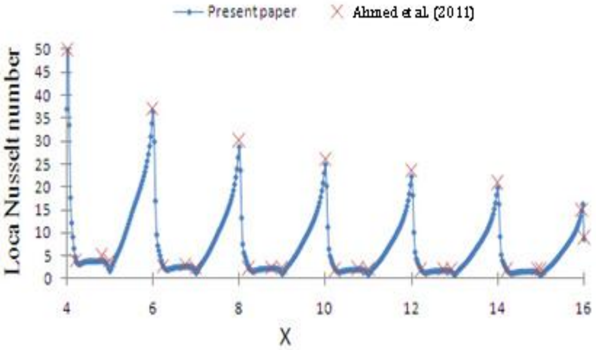


Fig. 3a: Comparison of the local Nusselt number versus length with the results of Ahmed *et al*(2011) for case3 and nanoparticles volume fraction 5%.

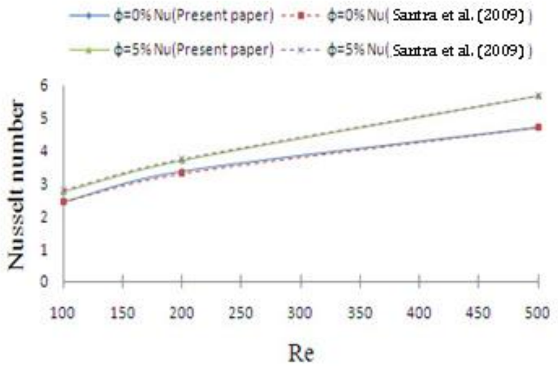


Fig. 3b: Comparison of the Nusselt number versus Reynolds number with the results of Santra *et al*. (2009) for smooth channel.

Results:

Figure 4 shows Comparison of the corrugated geometries of corrugated channel on the streamlines and isotherms. The figure shows that geometry of corrugated surface of channel has a significant effect on the change in the flow structure. Also it can be seen that temperature gradient increases in case 2 than other cases thus the enhancement in heat transfer is decreases in this case. In addition recirculation zone near the wall in case 2 lower than other cases, so the heat transfer in cases 1 and 3 is increases because recirculation zone near the wall in these cases is bigger than case 2.

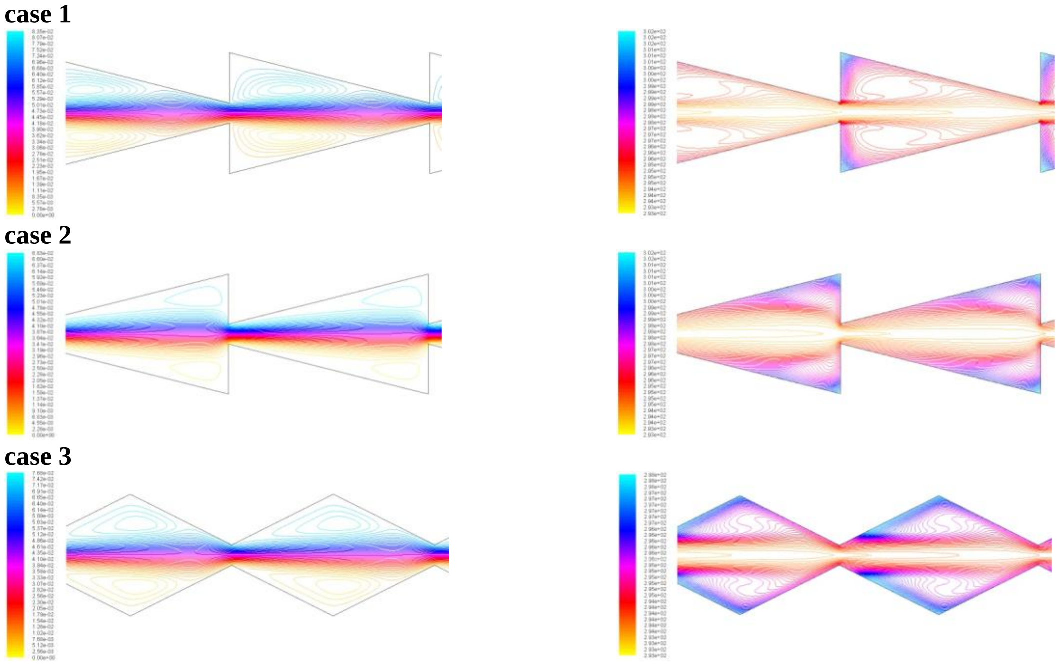
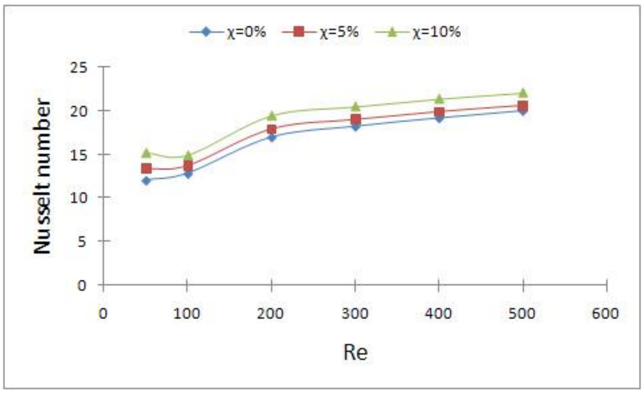


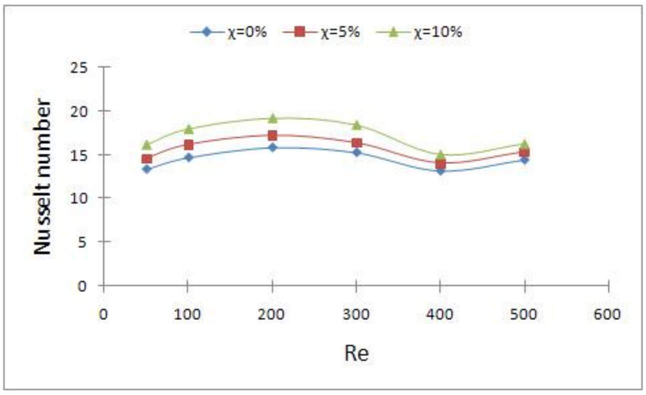
Fig. 4: Streamlines (left) and isotherms (right) for different cases at Re=400 and χ=5%

Figure 5 shows variation of Nusselt number with Reynolds number for different nanoparticles volume fraction values in every case. It can be seen that increasing nanoparticles volume fraction enhances the Nusselt number in all cases. In cases 1 and 3, with increasing the Reynolds number, the Nusselt number increases however in case 2, the Nusselt number gradually increase with increasing Reynolds number up to nearly Re=200, after which it trend to decrease up to nearly Re=400, after that it trend to increase.

case 1



case 2



case 3

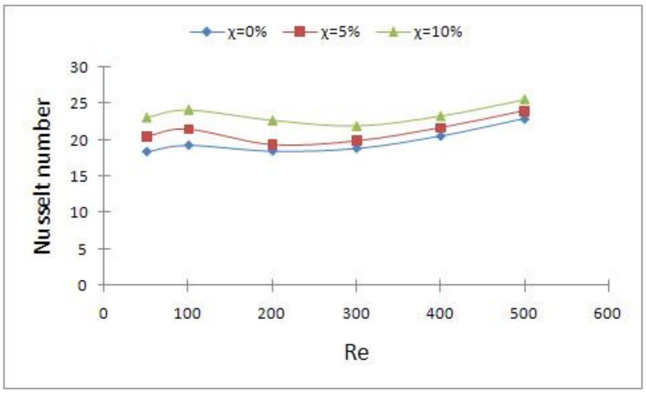


Fig. 5: Variation of Nusselt number with Reynolds number for different nanoparticles volume fraction values

Figure 6 shows variation of Nusselt number with Reynolds number for different cases and nanoparticles volume fraction 5%. It can be seen that in all Reynolds number, the enhancement in heat transfer is increases in case 3, because the temperature gradient in this case lower than other cases and also recirculation zone is bigger than other cases (see figure 4).

Figure 7 shows the influence of the Reynolds number and the nanoparticles volume fraction on the augmentation Nusselt number along the corrugated channel for case 3. It is clearly observed that the addition of nanoparticles causes the values of augmentation average Nusselt number to increase. Also, the percentage of heat transfer enhancement decreases with increasing of Reynolds number for all nanoparticles volume fractions. For example in case 3, at Re=100, the addition of 10% nanoparticles by volume, augmentation average Nusselt number enhance about 25% however for Re=400 enhance about 13%.

Figure 8 shows Variation of friction factor with Reynolds number for different cases at nanoparticles volume fraction 5%. It can be seen that the friction factor decreases with increasing the Reynolds number in all cases and the case 3 has maximum values in all Reynolds number.

Figure 9 shows variation of pressure drop with Reynolds number for different cases at nanoparticles volume fraction 5%. It can be seen that the pressure drop increases with increasing the Reynolds number in all cases because the drag force increases. The case 3 has the highest pressure drop, this is because the recirculation zone in this case is the highest and the drag has maximum value.

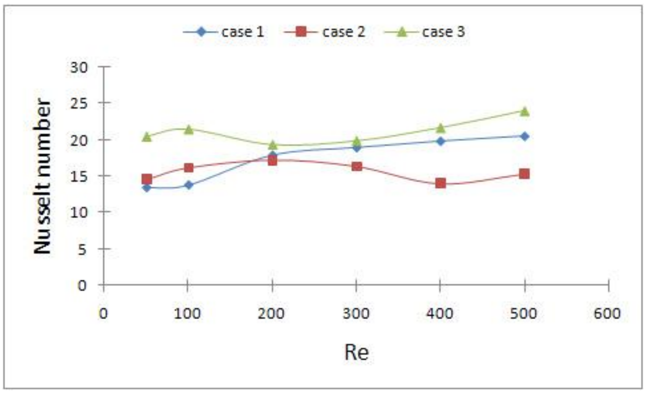


Fig. 6: Variation of Nusselt number with Reynolds number for different cases and χ=5%

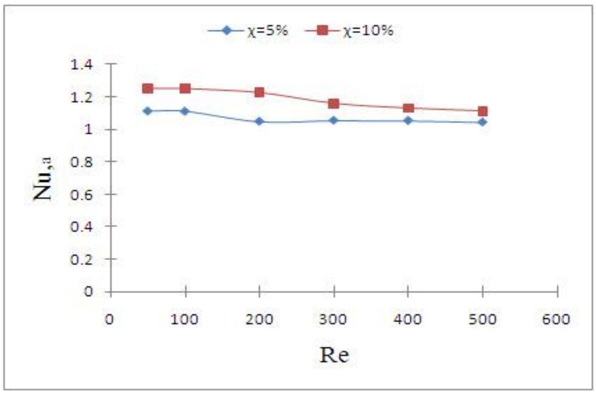


Fig. 7: Variation of the augmentation Nusselt number with Reynolds number for case 3

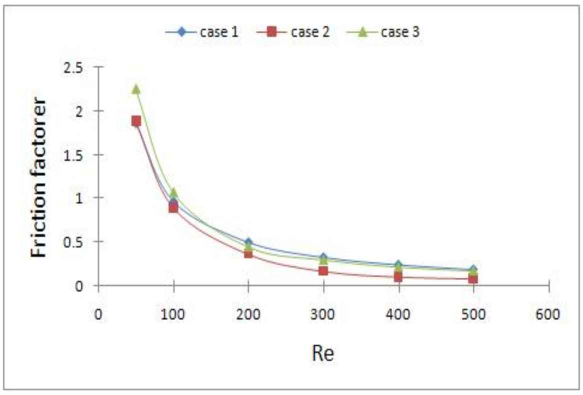


Fig. 8: Variation of friction factor with Reynolds number for different cases at χ=5%

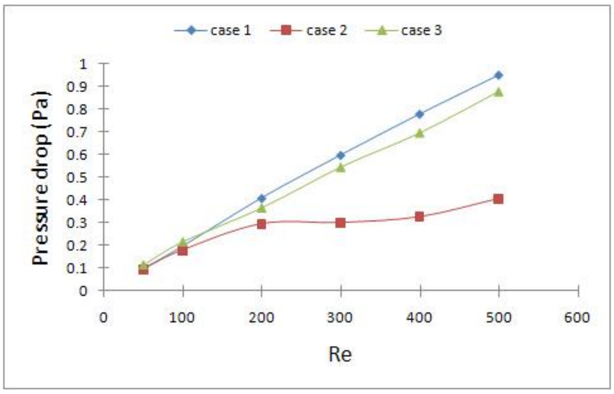


Fig. 9: Variation of pressure drop with Reynolds number for different cases at χ=5%

Figure 10 shows variation of thermal-hydraulic enhancement factor with Reynolds number for different cases. It can be seen that the thermal enhancement factor increases with increasing the Reynolds number in all cases and the case 3 has the highest the thermal-hydraulic enhancement factor.

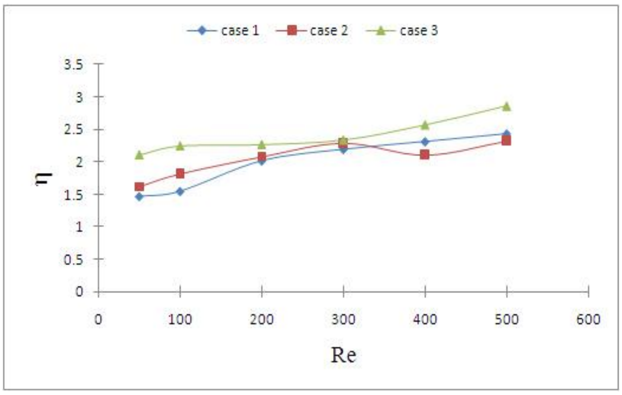


Fig. 10: Variation of thermal-hydraulic enhancement factor with Reynolds number for different cases

Conclusion:

In this paper, Numerical simulations of laminar forced convection heat transfer of Cu-water nanofluid in three different geometry of the corrugated channel under constant heat flux boundary conditions were carried out. The nanoparticles volume fractions and Reynolds number are in the ranges of 0–0.1 and 50–500 respectively. The governing equations were solved numerically using finite volume method (FVM) with the SIMPLE algorithm. The following results can be de derived from this study:

The Nusselt number increased with increasing Reynolds number however in case 2, the Nusselt number gradually increase with increasing Reynolds number in the ranges 50-200 and decreasing between Re=200-400, after that it trend to increase.

The Nusselt number increased with increasing nanoparticles volume fractions in all cases. However, the percentage of heat transfer enhancement decreases with increasing of Reynolds number for all nanoparticles volume fractions.

The friction factor decreased with increasing Reynolds number and pressure drop increased with increasing Reynolds number in all cases.

The thermal-hydraulic enhancement factor increased with increasing Reynolds number in all cases and the highest value of thermal-hydraulic enhancement factor related case 3 in all ranges of Reynolds number.

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