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Nanofluids Flow and Heat Transfer through Isosceles Triangular Channels: Numerical Simulation

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Abstract–Nanofluids are a new class of heat transfer fluids that exhibit better thermal characteristics compared to the conventional heat transfer fluids. It is important to clarify various aspects of nanofluids behavior. In order to identify the thermal behavior of nanofluids flowing through non-circular ducts, in present study, laminar flow forced convective heat transfer of Al_2O_3/water nanofluid thorough channels with isosceles triangle cross section with constant wall heat flux was studied numerically. The effects of nanoparticle concentration, nanofluid flow rate and geometry of channels on the thermal behavior of nanofluids were studied. The single-phase model was used in simulation under steady state conditions. Results reveal that the local and average heat transfer coefficients of nanofluids are larger than those of the basefluid. Heat transfer coefficient enhancement of nanofluids increases with increase in nanoparticle concentration and Reynolds number. The local heat transfer coefficient of the basefluid and that of the nanofluids decrease with the axial distance from the channel inlet. Results also indicate that by increase in the apex angle of channel, Nusselt number and heat transfer coefficient decrease.

***Keywords*:** Nanofluids, Nanoparticle,Numerical simulation, Triangular duct

Nomenclature

C_p: Heat capacity (kJ/kg K)
D_h: Duct hydraulic diameter (m)
h: Average heat transfer coefficient (W/m^2K)
h_z: Local heat transfer coefficient (W/m^2K)
k: Thermal conductivity (W/m K)
L: Length of channel (m)
N: Number of nodes
Nu: Average Nusselt number (dimensionless)
p: Pressure (kPa)
T: Temperature (K)
V: Fluid velocity (m/s)
z: Axial distance from duct inlet(m)
z*: Dimensionless distance from duct inlet (= $\frac{z/D_h}{RePr}$)

Greek symbols

α: Half-apex angle of triangular duct(degrees)
φ: Particle volume percent (dimensionless)
ρ: Density(kg/m^3)
μ: Fluid viscosity(Pa.s)

Subscript

bf: basefluid
eff: effective
nf: nanofluid
np: nanoparticle
p: particle

1. Introduction

Since heat transfer has widespread applications in various industries, increasing the efficiency of heat transfer equipment is converted to a priority for industrial designers and researchers. The efforts of scientists in this regard have led to invent many different methods. Two of these methods that have been used in present study are the use of non-circular channels and the use of nanofluids as a new class of heat transfer fluids. Because of the size and volume constraints in heat transfer applications, such as aerospace, biomedical engineering and electronics, utilization of non-circular flow passage geometries is required, particularly in construction of compact heat exchangers. Relatively low thermal characteristics of heat transfer fluids, such as water, engine oil and ethylene glycol are limiting factors in improvement of the efficiency of heat transfer devices. Solids have larger thermal conductivity, therefore dispersing them in fluids can help to improve their thermal characteristics(Nasiri et al., 2011).Many investigations have been made on thermal behavior of nanofluids using computational fluid dynamics(CFD). Investigations have been accomplished considering mixed and forced convective heat transfer in laminar and turbulent flow regimes. The simulation can be performed using single phase model or two-phase model. In the single phase model, nanoparticles and basefluid behave as a homogeneous fluid. It requires less computational time in comparison to two-phase model. Lotfi et al. (Lotfi et al., 2010)compared these two approaches for nanofluids flowing in a circular tube in laminar regime. Their results show that the mixture model is a more precise model. It is illustrated that the single-phase model and the two-phase Eulerian model underestimate the Nusselt number. A similar study was also conducted by Saberi et al.(Saberi et al., 2013)on a vertical tube. According to their results, the mixture model have better agreement with experimental data, while the prediction of nanofluid mean bulk temperature distribution inside the tube by the single phase model is better than the mixture model.(Bianco et al., 2010, Bianco et al., 2011, Nazififard et al., 2012)investigated turbulent flow convective heat transfer of nanofluids. Their results show that heat transfer increases by increasing the nanoparticle volume concentrations and Reynolds number. Rostamani et al. (Rostamani et al., 2010)haveanalyzed the turbulent flow of nanofluids with different volume concentrations of nanoparticles through a two dimensional duct numerically. Their results are similar to that of Bianco and Nazififard results(Nazififard et al., 2012). Also they analyzed the effect of nanoparticles type on heat transfer. They found that for a constant volume concentration and Reynolds number, the effects of CuO nanoparticles on the heat transfer enhancement is more than those of Al_2O_3 and TiO_2 nanoparticles. Laminar flow heat transfer of nanofluids flowing in isosceles triangular duct with constant wall temperature was simulated numerically(Choi and Zhang, 2012). Results show that the heat transfer coefficient and the Nusselt number of nanofluids are enhanced by increasing the nanoparticle concentration and Peclet number. Ghavam et al. (Ghavam et al., 2012) studied the convective heat transfer of γ-Al_2O_3/water nanofluids flowing through horizontal straight isosceles triangular ducts with constant wall temperature boundary condition numerically by employing the single-phase model. The heat transfer of nanofluids through different geometrical configurations including horizontal straight circular tubes(Ebrahimnia-Bajestan et al., 2011, Mirmasoumi and Behzadmehr, 2008, Namburu et al., 2009, Maiga et al., 2004, Maiga et al., 2005, Tahir and Mital, 2012, Daschiel et al., 2013, Labiba et al., 2013),vertical tube ((Saberi et al., 2013)), annulus(Izadi et al., 2009), elliptic ducts (Shariat et al., 2011),square ducts (Etemad et al., 2010, Heris et al., 2012, Mashaei et al., 2012), rectangular ducts (Jou and Tzeng, 2006), triangular ducts (Choi and Zhang, 2012, Heris et al., 2011, Nasiri et al., 2011)have been investigated.

In the present study, laminar flow forced convective heat transfer of Al_2O_3/water nanofluid thorough isosceles triangular cross sectional channel with constant wall heat flux was simulated numerically. Simulations were carried out based on the single-phase model. The effects of parameters such as nanoparticle concentration, flow rate of nanofluids and the geometry of channels on thermal behavior of nanofluids were studied.

2. Problems Statements

The geometry under consideration is shown in Figure 1. Since the geometry of channel is symmetric, simulations have been performed for half of the channels. Steady-state, laminar forced convective heat transfer of aqueous suspensions of Al_2O_3 with diameter of 25 nm flowing through horizontal isosceles triangular ducts with apex angles of 60°, 90° subjected to constant wall heat flux were considered. Simulations have been made for 0.1, 0.5, 1, 3, and 5 volume percentage of nanofluids. It was assumed that there is no difference between temperature and velocity of basefluid and nanoparticles, so single phase flow is used. The hydraulic diameter of all channels is chosen to be 0.01m. Non-uniform meshes were used near the walls and inlet of the channels; meshes are closer to each other because in these sections velocity and temperature gradients are more pronounced. To investigate the effect of flow rate, simulation has been done at four different Reynolds numbers of 400,800,1200,and 1600.

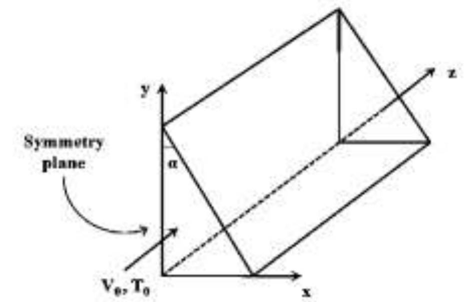

Fig. 1. Configuration of the triangular channel

Mathematical equations that describe the model are:

- Continuity equation:
  $$\nabla\cdot(\rho_{eff}V)=0 \tag{1}$$
- Momentum equation:
  $$\nabla\cdot(\rho_{eff}VV)=-\nabla p+\nabla\cdot(\mu_{eff}\nabla V) \tag{2}$$
- Energy equation:
  $$\nabla\cdot(\rho_{eff}VC_{p,eff}T)=\nabla\cdot(k_{eff}\nabla T) \tag{3}$$

The average heat transfer coefficient is calculated by Eq. (4),where N is the number of nodes in axial direction and Δz_i is shown in Figure 2.

$$h=\frac{1}{L}\int h_z dz=\frac{1}{L}\sum_{i=1}^{N}h_{zi}\Delta z_i \tag{4}$$

Fig. 2: Parameters of Eq. (4).

The average Nusselt number is calculated as follows:

$$Nu=\frac{hD_H}{k} \tag{5}$$

3. Physical Properties of Nanofluids

Thermophysical properties of nanofluids at the inlet temperature are obtained according to the following equations(Maiga et al., 2004, Maiga et al., 2005,Pak and Cho, 1998):

$$\rho_{nf}=(1-\varphi)\rho_{bf}+\varphi\rho_{np} \tag{6}$$

$$C_{p_{nf}}=\frac{(1-\varphi)\rho_{bf}C_{p_{bf}}+\varphi\rho_{np}C_{p_{np}}}{\rho_{np}} \tag{7}$$

$$k_{nf}=k_{bf}(4.97\varphi^2+2.72\varphi+1) \tag{8}$$

$$\mu_{nf} = \mu_{bf} (123\varphi^2 + 7.3\varphi + 1) \tag{9}$$

4. Boundary Conditions

The governing equations were solved by considering the following boundary conditions:

- At the channel inlet (z=0) (10)
  $V_z=V_0, V_x=V_y=0$
  $T=T_0$

- At the channel outlet (z=L)
  Fully developed condition was considered, so there was no change in velocity and temperature distribution in the axial direction.

- At the channel walls:
  $V_x=V_y=V_z=0$ (no slip condition) (11)
  $q=q_w$(constant wall heat flux)

5. Numerical Procedure

Finite volume technique was used to solve the governing equations (Eqs. 1-3). This method is based on the spatial integration of the governing equations over finite control volumes. A segregated formulation was adopted to solve the coupled conservation equations sequentially. The convective and the diffusive terms were discretized using the second order upwind method. Pressure-velocity coupling was achieved using the SIMPLE algorithm.

6. Results and Discussion

In order to validate the present results, fully developed Nusselt numbers of water flowing through the triangular duct with different apex angles of 60˚ and 90˚, obtained from the simulations were compared with the values reported by Shah and London (Shah and London, 1978). Based on the comparison maximum and average differences between these numbers are 4.40% and 3.94%, respectively and the small differences indicate the accuracy of simulations.

Figure 3shows the local heat transfer coefficient of nanofluids as a function of the axial distance for the duct with apex angle of 60˚. The local heat transfer coefficient of fluids decreases by increasing the axial distance from the duct entrance. This is due to the increase in the boundary layer thickness. The local heat transfer coefficient of nanofluids increases with an increase in nanoparticle concentration. This enhancement is larger at the entrance of the ducts. For instance for the channel apex angle of 60˚, by increasing the nanoparticle concentrations from 0% to 5%, the local heat transfer coefficient increases about 31.8% at the entrance of channel while at the fully developed area it is about 14.8%. Similar trends were observed at other values of Reynolds numbers and the channel apex angle of 90˚.Improvement of the thermal conductivity of nanofluids is the main factor that causes the convective heat transfer coefficient of nanofluids to increase. It also seems that the boundary layer of nanofluids grows slower than that of the base fluid and that the rate of growth decreases by increase in nanoparticle concentration, i.e. at a given distance from the duct inlet the boundary layer thickness of nanofluid is less than that of the base fluid. The local heat transfer coefficient, h, can be approximately given as $k/\delta t$ with δt the thickness of thermal boundary layer (Lai et al., 2009). Thus the improvement of heat transfer coefficient is due to the enhancement of thermal conductivity of nanofluids and/or decreasing thermal boundary layer thickness. This is in agreement with results obtained by Ghavam et al. (Ghavam et al., 2012).

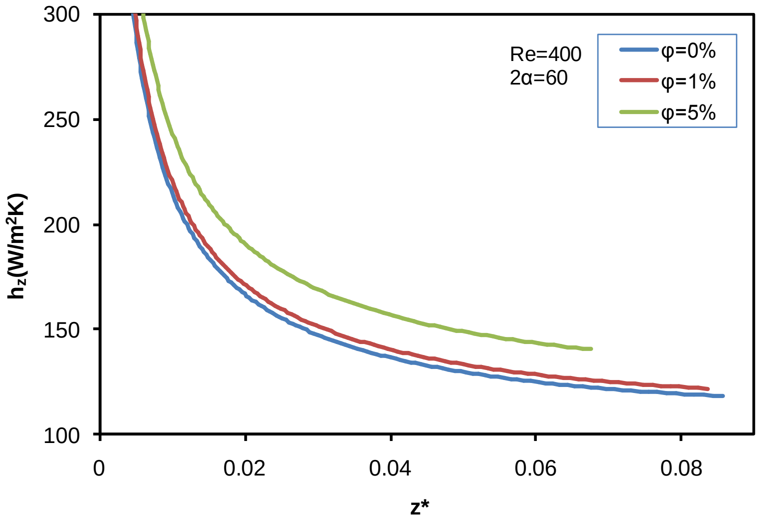

Fig. 3. Local heat transfer coefficient of nanofluids vs. axial position for different nanoparticle concentrations.

Average heat transfer coefficient of nanofluids at different nanoparticle concentrations as a function of Pe, for two apex angles of 60˚ and 90˚are shown in Figure 4.Heat transfer coefficient of nanofluids increases by increasing Peclet number, because the flow rate has a direct influence on the boundary layer thickness. The effect of Peclet number on the heat transfer coefficient is the same for the base fluid and nanofluids. However, the influence of Peclet number on the heat transfer coefficient is more pronounced in channel with higher apex angel. When the Peclet number of 3 vol.% nanofluid flowing through channels with apex angles of 60 and 90 changes from 2600 to 10500, the heat transfer coefficient enhances about 76 and 84 percent respectively. This is in agreement with the results obtained by Ghavam et al. (Ghavam et al., 2012) for similar ducts with the thermal boundary condition of constant wall temperature. These results also indicate that the influence of flow rate on the enhancement of heat transfer coefficient is larger in the case of ducts that subjected to the constant wall heat flux.

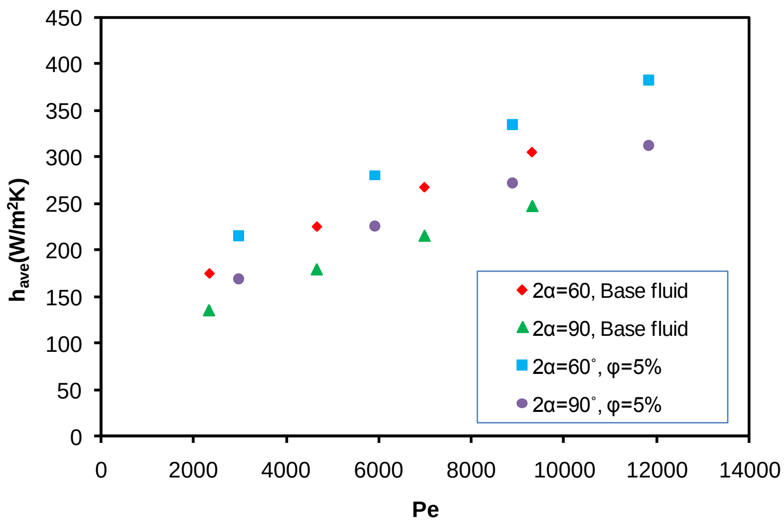

Fig. 4. Average heat transfer coefficient of nanofluids vs. Pe for different values of channel apex angle and different nanoparticle concentrations

Figure 5 indicates the effect of nanoparticle concentration and apex angle on average heat transfer coefficient. The value of h_{ave} increases by nanoparticle concentration. For example the average heat transfer coefficient of nanofluids with nanoparticle volume concentration of 5% is about 23%, and 24% greater than those of the base fluid for channels with apex angles of 60˚ and 90˚, respectively. This indicates that the effects of nanoparticles on the heat transfer enhancement of both channels are almost the same. Similar results have obtained by Ghavam et al. (Ghavam et al., 2012).

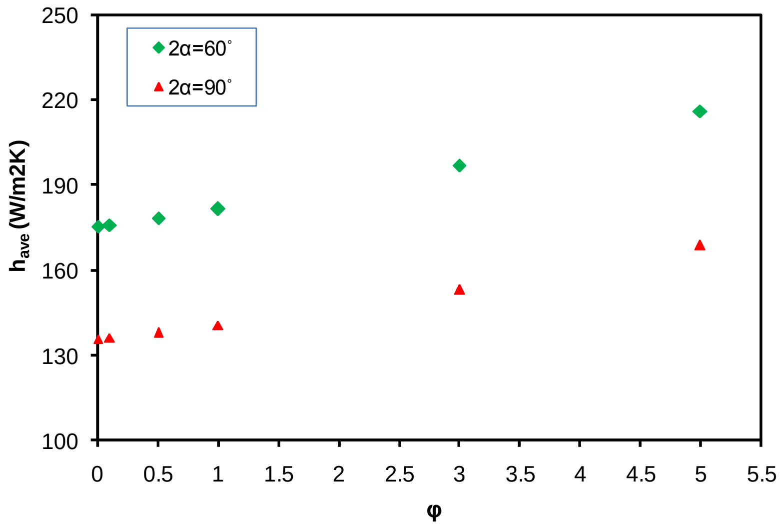

Fig. 5. Average heat transfer coefficient of nanofluids vs. nanoparticle concentration for channels with different apex angles.

The average Nusselt number of nanofluids as a function nanoparticle concentration for channels with different apex angles at constant Reynolds number of 400 is shown in Figure 6.The Nusselt number of nanofluids also increases with increase in nanoparticle concentration. For example the average Nusselt numbers of nanofluid with 3% by volume concentration through the channels with apex angles of 30° and 60° are about 3.5 and 4 percent greater than those of the base fluid respectively. It is obvious that the enhancement of the average Nusselt number is less than that of the heat transfer coefficient. This is due to the fact that both the effective thermal conductivity and the heat transfer coefficient of base fluid enhance as a results of adding nanoparticles. It is obvious that the improvement of the heat transfer coefficient is more than that of the thermal conductivity.

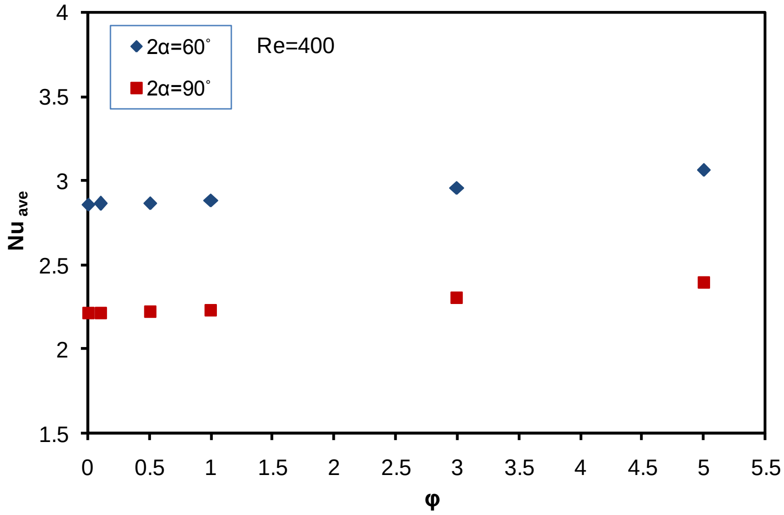

Fig. 6.AverageNusselt number vs. nanoparticle concentration for channels with different apex angles.

Laminar forced convection heat transfer of Al_2O_3/water nanofluids through an equilateral triangular duct was studied numerically using dispersion model (Zeinali Heris et al., 2011). They used nanoparticles with 10, 20, 30, 40, and 50 nm in diameter to identify the effect of particle size on thermal behavior of nanofluids. Their results show that the heat transfer coefficient of nanofluid is remarkably higher than that of the base fluid. The enhancement in heat transfer resulted from present study is much less than that resulted from their simulation. For example their results show that the Nusselt number of nanofluids with 1% by volume of nanoparticles of 20nm in diameter at Re=400 is about 10% higher than the base fluid, while the corresponding value obtained by present study is about 1%. This may be due to the models used for these two studies. The other reason is the difference between the sizes of nanoparticles.

4. Conclusions

In this study laminar convection heat transfer of Al_2O_3/water nanofluids flowing through horizontal isosceles triangular ducts subjected to constant wall heat flux was studied numerically. Simulations carried out according to the single-phase model. It is found that by increasing the distance from the duct inlet the local heat transfer coefficients of nanofluids decreases. Both the local heat transfer coefficient and the Nusselt number of nanofluids are enhanced by increasing of nanoparticle concentration. The enhancement of both is larger at the entrance of the ducts. As expected, average Nusselt number of nanofluids increases by increasing the Peclet number. The effect of Peclet number on the enhancement of nanofluids heat transfer is larger for ducts with higher apex angle. Results show that the average heat transfer coefficient and the Nusselt number of nanofluids are higher than that of the base fluid and increase by increasing the nanoparticle concentration. The enhancement of the heat transfer coefficient is much higher than that of the Nusselt number. Results also show that the influence of nanoparticle on the heat transfer enhancement of nanofluids is almost independent of the apex angle of the channels.

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