A NUMERICAL STUDY USING MIXTURES OF WATER - ETHYLENE GLYCOL BASED NANOFLUIDS ON LAMINAR HEAT TRANSFER OF AN ANNULUS

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Abstract: In this study, developing laminar flow and heat transfer behaviour of ethylene glycol (EG) and water mixture based SiO2 nanofluids in an annulus have been numerically investigated. A constant heat flux was applied to the inner walls of the annulus with 100 W / m². Water 100% - EG 0%, water 50% - EG 50% and water 0% - EG 100% mixtures have been utilized as the base fluids. SiO2 nanoparticles have been used with d = 20 nm and volume fractions ϕ =0%-4%. The Reynolds number varies from 200 to 1000. The physical model of the test section mainly consists of two concentric horizontal cylinders that form an annular space ranging from two interconnected elliptical tubes with axis ratio (r1/r2=1/2) placed at the centre of a circular cylinder with major radius of 2r2 with the length of 1 m. Governing equations have been solved with Ansys Fluent programme. The velocity distribution, temperature contours, average Nusselt number and thermal-hydraulic performance have been analysed and presented. The effects of nanofluids have been examined on heat and flow fields and it has been observed that the heat transfer increases together with the nanoparticle volume concentration. When the nanofluid is used in a forced convection, the amount of heat transfer increases as the Reynolds number increases. The highest value of the average Nusselt number was obtained in the EG based nanofluid with ϕ=4% and Re=1000 as 29.14, and the lowest value was obtained in the water-based nanofluid with ϕ=4% and Re=200 as 5.61. Results show that the use of nanofluid in the annulus channel increases the thermal performance of systems.

Key words: Elliptic annulus, heat transfer, nanofluid, CFD.

Introduction
Heat transfer describes the exchange of thermal energy, between physical structures relying at the temperature and pressure, by means of dissipating heat. The essential modes of heat transfer are conduction, convection and radiation. Engineers also consider the transfer of mass of differing chemical species, either cold or hot, to attain heat transfer. Convection is concerned with the transfer of thermal energy in a moving fluid (liquid or gas). It’s far ruled by means of two phenomena: the movement of energy because of molecular vibrations and the massive-scale movement of fluid particles (2018). In preferred, convection is of sorts, forced convection and free convection. Forced convection takes place while a fluid is forced to flow. For example, a fan blowing air over a heat exchanger is an instance of forced convection. In free convection, the majority fluid movement is due to buoyancy effects. As an example, a vertical heated plate surrounded by using quiescent air causes the air surrounding it to be heated. Due to the fact hot air has a decrease density than cold air, the hot air rises. The void is crammed by using cold air and the cycle continues. Mixed convection heat transfer exists whilst natural convection currents are the identical order of importance as pressured flow velocities. The time period “mixed Convection” is also used, and the flows may be inner or external to a bounding floor (Joye, 2003). Annular pipe flow is regularly encountered in engineering applications which includes heat exchangers, combustion systems, and drilling operations inside the oil and gas industry. Furthermore, annular pipe flow gives a perception into the trouble of turbulent flows with curved walls. Commonly, flow in a flat channel generates a symmetrical velocity profile and makes the positions of zero shear stress and most velocity coincident. However, the flow in a concentric annular channel does not result in a symmetric velocity profile. The asymmetric velocity profiles end result from the interaction of flow zones with different Reynolds numbers primarily based at the outer and inner cylinder radii. In the case of annular pipe flow, boundary layers exist and each has a different distribution of turbulent quantities. Furthermore, pipe and channel flows are the restricting cases of annular pipe flow. For a high radius ratio, the
Behzadmehr (Mirmasoumi & Behzadmehr, 2008) investigated the laminar mixed convection heat transfer of in nanometre dimensions changed into progressed and a new type of strong–liquid mixture this is referred to as show that thermal conductivity of nanofluid is higher than the bottom fluids. Among them, Lee et al. (Lee, Choi, Li, & Eastman, 1999) established that oxide ceramic nanofluids including CuO or Al₂O₃ nanoparticles in water or ethylene-glycol show off more advantageous thermal conductivity. As an instance, the usage of Al₂O₃ nanoparticles having suggested diameter of thirteen nm at 4.3% quantity fraction expanded the thermal conductivity of water below stationary conditions via 30% (Masuda, Ebata, & Teramae, 1993). However, large particles with an average diameter of 40 nm led an increase of less than 10% (Choi, S. U. S., n.d.). Distinctive ideas were proposed to provide an explanation for this enhancement in heat transfer. Xuan and Li (Xuan, 2000) and Xuan and Roetzel (Xuan & Roetzel, 2000) have recognized two reasons of improved heat transfer with the aid of nanofluids: the increased thermal dispersion because of the chaotic motion of nanoparticles that accelerates energy exchanges inside the fluid and the enhanced thermal conductivity of nanofluid. Then again, Keblinski et al. (Keblinski, Phillpot, Choi, & Eastman, 2002) have studied 4 possible mechanisms that contribute to the increase in nanofluid heat transfer: Brownian motion of the particles, molecular-stage layering of the liquid/particle interface, ballistic heat transfer in the nanoparticle and nanoparticle clustering. Similarly to Wang et al., (Wang, Xu, & Choi, 1999) they showed that the effects of the interface layering of liquid molecules and nanoparticles clustering could offer paths for speedy heat transfer. These days, Izadi et al. (Izadi, Behzadmehr, & Jalali-Vahid, 2009) studied the hydrodynamic and thermal behaviours of an Al₂O₃/water nanofluid flowing through an annulus underneath a laminar glide regime. In their observe, a single-section version was used for nanofluid simulation. The effects indicated that the particle volume concentration has no considerable effect on the dimensionless axial speed, however affects the temperature subject and increases the heat transfer coefficient. Mirmasoumi and Behzadmehr (Mirmasoumi & Behzadmehr, 2008) investigated the laminar mixed convection heat transfer of Al₂O₃/water nanofluid flowing through a horizontal tube numerically. A two-phase aggregate model was used to explain the hydrodynamic and thermal behaviour of the nanofluid. The numerical results indicated that inside the completely developed region the particle concentration has insignificant results on the hydrodynamic parameters, whilst it has crucial results at the thermal parameters. Furthermore, the consequences showed that nanoparticle concentration is higher at the lowest of the take a look at tube and on the close to wall region. However, Akbarinia (Akbarinia, 2008) and Akbarinia and Behzadmehr (Akbarinia & Behzadmehr, 2007) numerically investigated the fully developed laminar mixed convection of Al₂O₃/water nanofluid flowing through a horizontal curved tube. In their research, 3-dimensional elliptic governing equations were used. The results of the buoyancy pressure, centrifugal pressure and particle concentration on the heat transfer performance have been provided. The results confirmed that the particle concentration has no direct impact on the secondary flow, axial velocity and pores and skin friction coefficient. However, while the buoyancy pressure is more important than the centrifugal pressure, the impact of particle concentration at the entire fluid temperature can affect the hydrodynamic parameters. Furthermore, the consequences also indicated that the buoyancy force decreases the Nusselt number while the...
particle concentration has a high quality impact on the heat transfer enhancement and on the skin friction reduction. In this paper, a numerical investigation on heat transfer performance and flow fields of different nanofluids flows through elliptic annulus in a laminar and turbulent flow regimes. The three-dimensional continuity, Navier–Stokes and energy equations are solved by using finite volume method (FVM) and the SIMPLE algorithm scheme is applied to examine the effects of laminar and turbulent flow on heat transfer characteristics. Dawood et al. (Dawood, Mohammed, Sidik, & Munisamy, 2015) evaluated the effects of four different types of nanoparticles, Al₂O₃, CuO, SiO₂ and ZnO, with different volume fractions (0.5–4%) and diameters (25–80 nm) under constant heat flux boundary condition using water as a base fluid. The Reynolds number of laminar flow was in the range of 200 ≤ Re ≤ 1500, while for turbulent flow it was in the range of 4000 ≤ Re ≤ 10,000. The results have shown that SiO₂–water nanofluid has the highest Nusselt number, followed by ZnO–water, CuO–water, Al₂O₃–water, and lastly pure water. They showed that the Nusselt number for all cases increases with the volume fraction but it decreases with the rise in the diameter of nanoparticles. In all configurations, the Nusselt number increases with Reynolds number. In their study, it is found that the glycerine–SiO₂ shows the best heat transfer enhancement compared with other tested base fluids.

As seen in those and/or similar works, heat transfer mechanisms in annulus can be very complicated and this geometry might be regarded in many commercial set up. Consequently, the existing work targets to investigate some behaviours of nanofluid flow in an elliptic annulus. As a result, the results of volume fraction and specific base fluids comprised of water and ethylene glycol mixtures on the thermodynamics and hydrodynamics parameters of a 3-D laminar forced convection through an elliptic concentric annulus were studied. Effect of water and ethylene glycol volume fractions were addressed as EG fractions ranges from 0% to 100% with a 50% increment. This research covers Reynolds range within the range of 200 ≤ Re ≤ 1000 (laminar) with a diameter of 20 nm silicon oxide (SiO₂) nanoparticle. Different volume fractions SiO₂ nanoparticles within the base fluids ranged from 0% to 4% were also considered. Outcomes of pursuits which include Nusselt number, heat transfer enhancement in an elliptic annulus were stated to illustrate the impact of nanofluids on these parameters.

**Physical Model**

The physical model of the test section mainly consists of two concentric horizontal cylinders used to form an annular space ranging from an integrated double elliptical tube placed at the center of a circular cylinder. The outer cylinder was made from aluminum of 20 mm outer diameter, 1 mm thickness, and 500 mm length. The inner elliptic cylinder was made from aluminum with a major radius (r₂) of 10 mm and a length of 500 mm that had an axis ratio (r₁/r₂=1/2). The internal wall of the annular space (elliptic tube surface) was maintained under constant heat flux (qₜ). Whereas the external wall of the annular space (circular cylinder surface) was kept insulated (T_c). The schematic diagram of the annular space under consideration and coordinate system are shown in Figure 1.

![Figure 1. Schematic diagram of the computational domain of annulus.](image)
Table 1: The thermophysical properties of different nanoparticles and different base fluids at \( T = 300 \text{ K} \).

<table>
<thead>
<tr>
<th>Thermo-physical property</th>
<th>Unit</th>
<th>100%W-0%EG</th>
<th>50%W-50%EG</th>
<th>0%W-100%EG</th>
<th>SiO₂</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density, ( \rho )</td>
<td>kg/m³</td>
<td>997,1</td>
<td>1071,1</td>
<td>1132</td>
<td>2200</td>
</tr>
<tr>
<td>Specific heat, ( c_p )</td>
<td>J/kgK</td>
<td>4180</td>
<td>3300</td>
<td>2349</td>
<td>703</td>
</tr>
<tr>
<td>Dynamic viscosity, ( \mu )</td>
<td>kg/ms</td>
<td>0,0009</td>
<td>0,0034</td>
<td>0,0151</td>
<td>-</td>
</tr>
<tr>
<td>Thermal conductivity, ( k )</td>
<td>W/mK</td>
<td>0,613</td>
<td>0,37</td>
<td>0,258</td>
<td>1,2</td>
</tr>
<tr>
<td>Thermal expansion coefficient, ( \beta )</td>
<td>1/K</td>
<td>0,00021</td>
<td>0,00039</td>
<td>0,00057</td>
<td>0,0000055</td>
</tr>
</tbody>
</table>

Geometry and the governing equations

The phenomenon under consideration is governed by the steady three-dimensional form of the continuity; the time-averaged incompressible Navier–Stokes equations and energy equation are used to describe the heat transfer in the annulus. Heat is transferred between the fluids through the wall which is separating them. Several assumptions were made on the operating conditions of the annulus: (i) the annulus operates under steady-state conditions and three-dimensional; (ii) the nanofluid is Newtonian and incompressible; (iii) the fluid is in single phase and the flow is laminar; (iv) the external heat transfer effects are ignored; (v) the outer walls of the annulus are adiabatic; and (vi) constant thermophysical properties are considered for the nanofluid.

The governing equations for flow and heat transfer in the annulus are as follows (Edition, Ashgriz, & Mostaghimi, 2002):

Continuity equation:

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{v}) = 0
\]  (1)

Momentum equation:

\[
\rho \frac{D \mathbf{v}}{D t} = \nabla \cdot \mathbf{\tau} + \mathbf{p} + \rho \mathbf{F}
\]  (2)

Energy equation:

\[
\rho \frac{D e}{D t} + p (\nabla \cdot \mathbf{v}) = \frac{\partial Q}{\partial t} - \nabla \cdot \mathbf{q} + \Phi
\]  (3)

Where \( \mathbf{v} \) is the fluid velocity vector, \( \mathbf{F} \) is the body forces, \( \mathbf{q} \) represents heat transfer by conduction and \( \Phi \) is the dissipation term. These governing equations along with the given boundary conditions are solved to obtain the fluid temperature distribution and pressure drop along the annulus. These data were then used to examine the thermal and flow fields along the annulus.

Boundary conditions

At the elliptic inlet, different velocities depending on the values of Reynolds number were used, and the outlet temperature was taken as \( T_{\text{in}} = 300 \text{ K} \). The constant heat flux used was 100 W/m² to heat up the inside walls. At the domain outlet the flow and heat transfer are assumed to be fully developed. The boundary condition can be expressed as follows:

At the inlet of annulus:

\[
u_x = u_0 = u_z = 0 \text{ and } T = T_{\text{in}}
\]  (4)

At the fluid wall interface:
\[ u_r = u_\theta = u_z = 0 \text{ and } q_{\text{wall}} = -k_e \frac{\partial T}{\partial r} \]  \hspace{1cm} (5)

At the outlet of annulus free pressure outlet is applied:
\[ p = p_0 \]  and an overall mass balance correction is applied.

**Thermophysical properties of nanofluids**

In order to carry out simulations for nanofluids, the effective thermophysical properties of nanofluids must be calculated first. Basically the required properties for the simulations are effective thermal conductivity \( (k_{\text{eff}}) \), effective dynamic viscosity \( (\mu_{\text{eff}}) \), effective mass density \( (\rho_{\text{eff}}) \), effective coefficient of thermal expansion \( (\beta_{\text{eff}}) \) and effective specific heat \( (c_{\text{p,eff}}) \) are given in Table 1. The effective properties of mass density, specific heat and coefficient of thermal expansion are actually calculated according to the mixing theory.

By using Brownian motion of nanoparticles in three-dimensional horizontal concentric annulus, the effect thermal conductivity can be obtained as following mean empirical correlation (Ghasemi & Aminossadati, 2010):

\[ k_{\text{eff}} = k_{\text{static}} + k_{\text{brownian}} \]  \hspace{1cm} (6)

\[ k_{\text{static}} = k_f \left( \frac{(k_r + 2k_\theta)}{(k_p + 2k_f)} - 2\phi (k_r - k_p) \right) \]  \hspace{1cm} (7)

\[ k_{\text{brownian}} = 5 \times 10^4 \beta \phi \rho_f c_p f(T, \phi) \]  \hspace{1cm} (8)

where:

Boltzmann constant: \[ \kappa = 1.3807 \times 10^{-23} \text{ J/K} \]

Value of the base fluid fraction goes with the nanoparticle, \( \beta \) is calculated as following:

\[ \beta_{\text{SiO}_2} = 1.9526(100\phi)^{-1.4594} \quad 1\% \leq \beta \leq 10\% \quad 298 \text{ K} \leq T \leq 363 \text{ K} \]  \hspace{1cm} (9)

Modelling function, \( f(T, \phi) \),

\[ f(T, \phi) = \left( 2.8217 \times 10^{-2} \phi + 3.917 \times 10^{-3} \right) \left( \frac{T}{T_0} \right)^{(-3.0669 \times 10^{-2} \phi - 3.3911123 \times 10^{-3})} \]  \hspace{1cm} (10)

for \( 1\% \leq \beta \leq 4\% \quad 300 \text{ K} \leq T \leq 325 \text{ K} \)

By using Brownian motion of nanoparticles the effective viscosity can be obtained by using the following empirical correlation (Mohammed, Abbas, & Sheriff, 2013):

\[ \mu_{\text{eff}} = \mu_f \left( 1 - \frac{34.87 (d_p/d_f)^{0.3} \phi^{1.03}}{1 - \phi} \right) \]  \hspace{1cm} (11)

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

where \( M \) is the molecular weight of base fluid, \( N \) is the Avagadaro number, \( N=6.022 \times 10^{23} \text{ mol}^{-1} \), \( \rho_f \) is the mass density of the based fluid calculated at temperature \( T_0=293 \text{ K} \).

The effective density of the nanofluid can be calculated using (Ghasemi & Aminossadati, 2010):

\[ \rho_{\text{eff}} = (1-\phi) \left( \rho_f \right)^{1+\phi \rho_p} \]  \hspace{1cm} (12)

where \( \rho_{\text{eff}} \) and \( \rho_f \) are the nanofluid and base fluid densities respectively and \( \rho_p \) is the density of nanoparticle.
The effective specific heat at constant pressure of the nanofluid \( c_{\text{peff}} \) is computed using the following equation (Ghasemi & Aminossadati, 2010):

\[
\left( \rho c_p \right)_{\text{eff}} = (1-\phi)\left( \rho c_p \right)_{f} + \phi \left( \rho c_p \right)_{p}
\] (13)

where \( c_{ps} \) and \( c_{pf} \) are the heat capacity of solid particles and base fluid respectively.

The Nusselt number, the Reynolds number and the friction factor are dimensionless parameters which are calculated, respectively, as follows (Mohammed et al., 2013):

\[
\text{Nu} = \frac{hD_h}{k_{\text{eff}}}
\] (14)

where \( k \) and \( h \) are the thermal conductivity and average heat transfer coefficient of fluid, respectively.

The Reynolds number is defined as:

\[
\text{Re} = \frac{\rho_{\text{eff}} u_m D_h}{\mu_{\text{eff}}}
\] (15)

where \( \rho_{\text{eff}}, u_m, \) and \( \mu_{\text{eff}} \) are nanofluid density, mean fluid velocity over the cross section and dynamic viscosity of the nanofluid, respectively.

The hydraulic diameter (\( D_h \)) is defined as:

where \( A \) is the cross area and \( P \) is the wetted perimeter of the cross section.

The friction factor, \( f \), for fully developed flow is expressed as follows:

\[
f = \frac{2\Delta p D_h}{L \rho_{\text{eff}} u_m^2}
\] (16)

and the power required to pump, \( P \), the nanofluid is calculated as follows:

\[
P = Q \Delta p
\] (17)

where \( Q \) is the volumetric flow rate of the nanofluid.

**Numerical Solution Method**

**Grid testing and code validation**

The computational domain resulted from the subtraction of the elliptical cylinder section from the circular cylinder section. The grid is made up of triangular elements to improve the quality of the numerical prediction near the curved surfaces.
As shown in Figure 2 the computational grid of the elliptic annulus, built through the mesh generation, three grids types with elements count 153636, 1164822 and 8846771 show no much difference in the values of average Nusselt number and average fanning friction factor. Thus, the grid with 153636 elements is selected in this study as it is found to provide a more stable grid independent solution and due to the fact that resulting in a lower computational cost.

The code validation was done based on the geometry and boundary conditions which were used by (Shah & London, 1978). They studied the thermal characteristics of laminar and turbulent convection heat transfer in a concentric annulus with constant heat flux boundary condition. In this case, the results of the Nusselt number variation were compared with the predictions of the following well-known Shah equation for laminar flows under the constant heat flux boundary condition in the fully developed region as shown in Figure 3. To validate the accuracy of the numerical solutions, the Nusselt number (Nu) and the friction factor times Reynolds number (fRe) of the concentric annular is compared with the theoretical data. It is clearly seen that the deviation between the numerical results and the theoretical data is very low. Therefore, the present numerical predictions have reasonable accuracy.

**Figure 3.** Model comparison Nu (upper) and fRe (lower)

**Numerical implementation**

A numerical steady-state simulation of the flow field through 3D elliptic concentric annulus is considered to investigate and solve complex fluid flow and heat transfer model. The commercial available CFD software, FLUENT was used to solve the governing equations of continuity, momentum and energy. The numerical computations were performed by solving the governing conservations along with the boundary conditions using the finite volume method (FVM). It is based on the control volume method; COUPLED algorithm is used to deal with the problem of velocity and pressure coupling. The pressure staggering option (second order) scheme is used to solve pressure equations. The diffusion term in the momentum and energy equations was approximated by second-order central difference which gives a stable and more accurate solution. In addition, a second-order
upwind differencing scheme was adopted for the convective terms (John & Anderson, 1995). The numerical model was developed in the physical domain, and dimensionless parameters were calculated from the computed velocity and temperature distributions. The residual sum for each of the conserved variables is computed and stored at the end of each iteration. The convergence criterion required that the maximum relative mass residual based on the inlet mass be smaller than $1 \times 10^{-5}$. Also average Nusselt number and fanning friction factor values were watched through the simulations to decide the convergence of the solution as their value not to change after a certain value.

**Results and Discussion**

The simulations are performed of laminar forced convection heat transfer and fluid flow for different types of base fluids in a three-dimensional through a double-integrated elliptic annulus. Effect of water and ethylene glycol volume fractions are addressed as EG fractions ranges from 0% to 100% with a 50% increment. Different values of Reynolds number were used in the range of $200 \leq \text{Re} \leq 1000$ for laminar flow and with volume friction of SiO$_2$ nanoparticles in the range of $0 \leq \phi \leq 0.04$ with a diameter of 20 nm. Nanofluids are proven to enhance the heat transfer characteristics. However, there is no research done on finding the effect of the base fluid. To get the best base fluid, each base fluid is compared in terms of average surface Nusselt number and pumping power.

**Effect of different volume fractions of nanoparticles**

In this section, the effect of nanoparticles volume fraction on the average Nusselt number was investigated in the range of $0 – 4\%$ with different values of the Reynolds number and diameter of particle $d_p = 20$ nm for SiO$_2$ nanofluid. As shown clearly in Figure 4, increasing nanoparticle volume fraction enhances the Nusselt number. The Nusselt number is not very sensitive to the volume fraction of nanoparticles at lower Reynolds number and in all cases with increasing the Reynolds number, the Nusselt number increases. It can be seen that the highest

![Figure 4](https://www.tojsat.net)
Figure 5. Average Nusselt Number variations due to effect of nanoparticle volume fraction concentration of nanoparticles has the highest Nusselt number profiles. This is due to the enhanced effective thermal conductivity of the nanofluid which is accompanied by an increase in the thermal diffusivity. Heat transfer enhancement is increased when volume fraction is increased; it can be observed that 4% volume fraction has the highest heat transfer enhancement, while 0% concentration has the lowest enhancement as shown in Figure 4. This is because the physical properties of nanofluid vary with the volume fraction. Thus, transfers more energy in the fluid, because of the momentum energy is much higher than the thermal energy in higher volume fraction.

As illustrated in Figure 5, the pumping power increases with the increase of Reynolds number for different volume fractions of nanoparticles. In general, the increase of nanoparticles volume fraction results in an increase of fluid viscosity which diminishes the fluid movement.
Figure 6. Local Nusselt Number variations due to effect of different volume fractions

Figure 6 shows the variation of the local Nusselt numbers through the annulus with Reynolds number chosen as 1000 and pure EG is the base fluid with different SiO\textsubscript{2} particle volume fractions. Local heat transfer coefficients thus the local Nusselt numbers values are very high at the entrance since the flow is developing. Because the thickness of the thermal boundary layer is zero at the entrance and it decreases continuously in the axial direction due to the thermal boundary layer that develops. As illustrated, when the nanoparticle concentration increases, local Nusselt number values also increases since the Brownian motion that dissipates heat is higher at high nanoparticle concentrations.

Velocity distribution and isotherms contours at the outlet of the annulus for pure EG nanofluid with different SiO\textsubscript{2} particle volume fraction at Reynolds number value equals to 1000 have been shown in Figure 7. The left hand side of the figure shows the isoterms while right hand side shows the velocity distribution. In this case local Nusselt Numbers varies as 18.87, 19.35, 20.08, 20.97 and 22.11 for volume fractions %0, 1, 2, 3 and 4, respectively. Thus there is no much difference and this is also supported by isoterms since they seem to be nearly the same for all concentrations. By adding the nanoparticle, that is by increasing the nanoparticle volume concentration, both density and dynamic viscosity of nanofluid increases. However ratio of density and dynamic viscosity decreases with increasing concentration. Velocities at the annulus walls are zero for all cases due to the boundary layer that develops. Therefore for the same Reynolds number values, the nanofluid with the highest concentration has the highest velocity in the annulus core. This fact is also supported by the velocity contours which are shown. As the intensity of red color increases that means that the velocity also increases. There are also local increments in the velocities where the length is lower between the inner and outer walls due to narrowing crosssection in this regions.
Figure 7. Isotherms (left) contours and velocity distribution (right) and for varying volume fractions
Conclusion
Numerical simulations for laminar forced convection heat transfer and fluid flow characteristics in a double integrated elliptic annulus using various nanofluids as the working fluids were presented. A three dimensional grid setup was built in order to simulate the geometry using Computational Fluid Dynamics (CFD) software. Using finite volume method (FVM), the governing equations were deciphered and correlated to case study, provided with some particular assumptions. The emphasis is given on the heat transfer enhancement resulting from various parameters, which include base fluid types and volume fraction of nanoparticle. The results were obtained through the numerical simulation that gives the highest Nusselt number. It is found that SiO$_2$ EG nanofluid gives the highest Nusselt number while pure water gives the lowest Nusselt number. The Nusselt number is remarkably increased with the increments of nanoparticle volume fraction and Reynolds number. However use of EG in such microchannels sharply increases the required pumping power. This may be a drawback for this nanofluid if there is no available space for larger size pumps.

References


