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NUMERICAL INVESTIGATION ON HEAT TRANSFER COEFFICIENT ENHANCEMENT OF NON-NEWTONIAN NANOFLUID IN THE TURBULENT FLOW INSIDE A TUBE

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ABSTRACT

In this paper, attempt has been made to simulate the turbulent flow of a non-Newtonian nanofluid in a circular, horizontal tube with regard to constant heat flux on the tube wall, using Computational Fluid Dynamics (CFD). To this effect, using fluent software, the conservation equations of momentum and energy for turbulent flow of a non-Newtonian fluid and a nanofluid containing alumina particles in the non-Newtonian fluid is analysed. An aqueous solution of carboxymethyl cellulose (CMC) was used as the base non-Newtonian fluid. Temperature field of nanofluids was obtained and, analysing the results, the heat transfer coefficient of nanofluids was calculated. The results obtained were compared with the results of the base fluid. The effects of volume fraction or nanoparticle concentration and Reynolds number on heat transfer coefficient and Nusselt number were also investigated. The results indicated an increase in heat transfer coefficient and Nusselt number using non-Newtonian nanofluid compared to the base non-Newtonian fluid. This increase had a direct relationship with the volume fraction of the nanoparticles and the Reynolds number. The results showed good agreement with the results of laboratory research.

KEYWORDS: Nanofluids, non-Newtonian, heat transfer, turbulent flow, Nusselt.

Nowadays, the increase of convective heat transfer is an important issue in industries or heating & cooling equipments. In general, this increase is possible in three methods that consist of: Changes in flow geometry and boundary conditions of the problem and thermophysical properties of fluid.

The main property of the fluid in heat transfer is the thermal conductivity of the fluid. According to (Maxwell, 1873) and (Maxwell, 1881), the increase of thermal conductivity of the fluid is possible by adding the solid metal particles suspended in the fluid. Hence, the theory of Maxwell is the baseline of theories on nanofluids, which is obtained by suspension of solid metal particles with a dimension of nanometer scale in the base fluid.

Various studies have shown that heat transfer coefficient of nanofluids containing even very low concentrations of nanoparticles may also lead to an increase by more than twenty percent. Many studies have examined nanofluids convective heat transfer in both laminar flow (such as (Santra, 2009), (Wen & Ding, 2004), (Yang, 2005), (He, 2007), (buongiorno, 2009), (Chun, 2008)) and turbulent regimes (such as (Behzadmehr, 2007), (Bianco, 2011), (Akbari, 2012)). But the issue of convective heat transfer enhancement using nanoparticles suspended in non-Newtonian fluids has attracted much attention in recent years. Non-Newtonian fluids have wide application in various industries such as food processing, petrochemical and pharmaceutical etc. Thus, the study of behavior of non-Newtonian nanofluids in the possibility of enhancing the heat transfer in different processes of these industries is essential.

Soltani (2010) investigated the pool boiling heat transfer in non-Newtonian nanofluids. Their results showed that the coefficient of pool boiling heat transfer is enhanced with the increasing concentration of nanoparticles in non-Newtonian nanofluids. Kamali and Binesh (2010) using CFD method and finite volume method and also Fluent software, simulated the heat transfer rate used in non-Newtonian nanofluids based on carbon nanotubes in a circular, direct, horizontal tube under constant flux boundary conditions. They found that the heat transfer coefficient in the region adjacent to the wall, because of the non-Newtonian behavior of the carbon nanotube-based nanofluids, is most notable. Etemad (2010) investigated in an experimental study on convective heat transfer in the laminar flow regime for non-Newtonian nanofluids. The results showed that an enhancement in convective heat transfer of these nanofluids is possible with increasing particle concentration. Also, Etemad (2011) investigated, in another experimental investigation, on the forced convective heat transfer of three different non-Newtonian nanofluids inside a circular tube under turbulent flow regime and constant flux boundary condition. The results showed that the local and mean heat transfer coefficients of nanofluids was larger than the base fluid and that nanofluids heat transfer rate increases with increasing concentration of nanoparticles. In this study, an equation for the Nusselt number in non-Newtonian nanofluids was proposed in which the Nusselt was a function of Prandtl and Reynolds numbers. Moraveji (2012) observed numerically the heat transfer in laminar flow of non-Newtonian nanofluids. They used CFD and finite volume methods current in Fluent software to simulate the flow in the

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circular, horizontal tube under constant flow boundary condition. In this analysis, they used single-phase fluid model for flow modeling and Herschel – Balkley model for non-Newtonian fluid, and then they studied the effect of the size and concentration of nanoparticles on heat transfer coefficient at Reynolds number of 500 to 2500. Their results showed that heat transfer coefficient and Nusselt number of non-Newtonian nanofluids increases with increasing particle concentration in the solution. For numerical simulation to study the characteristics of nanofluids heat transfer, two methods in various articles have been used The first method assumed that the assumption of continuity for fluids containing nano-scale particles dissolved in them (nanofluids) is still established. This method utilizes single phase fluid model and enriched characteristics of nanofluids for use in conservation laws. The second method utilizes multi-phase options, and in this regard, the use of two-phase models such as mixture models, dispersion models and discrete phase model (interactions of Euler – Lagrange) are the most common method among multiphase methods and are used to describe both liquid and solid phases. Although they are less common in most used literatures. That's because the single-phase method is the most simple but more efficient in terms of computation. Meanwhile, the results of using this method in various literatures have shown satisfactory compliance with the results of experimental researches.

In this study, the behavior of convective heat transfer of nanofluids using a non-Newtonian fluid in turbulent flow regime under constant heat flux boundary condition on tube walls is investigated using CFD tool and finite volume method and Fluent software. Aqueous solution of carboxymethyl cellulose (CMC) with a concentration of %0.5 wt. is used as non-Newtonian basic fluid. Carboxymethyl cellulose is a derivative of cellulose that is widely used in formulation, processing and production of various foods for owning many properties. Carboxymethyl cellulose solution in water is a non-Newtonian fluid, pseudo-plastic or taper. In most researches on non-Newtonian fluids and nanofluids, it is used as base fluid. Hence, in line with previous studies, it is also used in this study. Also, A12O3 nanoparticles in a size of 25 nm and volumetric concentrations of % 0.5 and %1.5 were used. Power-law model was used for the estimation of rheological behavior of non-Newtonian viscous fluids. The assumption of single-phase model was taken into consideration and modeling was used for steady-state fluid flow. This analysis encompasses flow and heat transfer along the tube from the developing zone to developed zone for velocity and temperature. In this paper, the effect of particles concentration and Reynolds number on heat transfer coefficient and Nusselt number (Nu) has been examined.

PROBLEM STATEMENT AND MATHEMATICAL MODELING

Fig. 1 shows the flow geometry and problem solving extent which is a circular tube 120 cm in length and 0.475 cm in diameter.

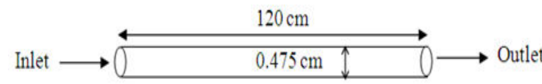

Figure 1: Problem solving extent and flow geometry

As outlined, in single phase models, nanofluids behave like a normal fluid, but with improved properties resulting from the use of nanoparticles. One of the main issues in the simulation of fluid flow and heat transfer in nanofluids is to find the thermo-physical properties of nanofluids. Among these properties, viscosity and thermal conductivity coefficient can be noted and the results of this study strongly depend on their actual values.

In this study, Fluent software was used for numerical analysis and the corresponding simulation. This software operates according to finite volume method. Conservation equations including respectively the mass conservation, momentum conservation and energy conservation are solved for the relevant flow geometry.

$$\nabla.(\rho_{nf} V_m) = 0 \quad (1)$$

$$\nabla.(\rho_{nf} V_m V_m) = -\nabla P + \nabla.(\mu_{nf} \nabla V_m) \quad (2)$$

$$\nabla.(\rho_{nf} C V_m T) = \nabla.(k_{nf} \nabla T) \quad (3)$$

Nanofluids thermophysical properties measurement

As it is obvious, to solve these equations, we need to measure nanofluids thermophysical properties such as the coefficient of specific heat and thermal conductivity coefficient. In this paper, to measure the coefficient of specific heat and thermal conductivity, the following relations which are so common in such measurements were used (Pak and Cho, 1998):

$$\rho_{nf} = \varphi \rho_p + (1-\varphi)\rho_f \quad (4)$$

$$(\rho C_p)_{nf} = \varphi (\rho C_p)_p + (1-\varphi)(\rho C_p)_f \quad (5)$$

Here, φ is particle volume fraction and *nf, f* and *p* indexes indicate respectively the nanofluid, basefluid and particle. To measure nanofluid thermal conductivity coefficient, the equation of Chon et al [18] has been used. This equation considers the impact of Brownian motion as well as the particle size to measure thermal conductivity coefficient. This relation is valid for particles of 11 to 150 nm:

$$\frac{k_{nf}}{k_f} = 1 + 64.7 \varphi^{0.7460} M \quad (6)$$

$$M = \left(\frac{d_f}{d_p}\right)^{0.3690} \left(\frac{k_p}{k_f}\right)^{0.7476} Pr^{0.9955} Re^{1.2321} \quad (7)$$

In the above equations, according to various studies, the physical properties of alumina such as density, specific heat and thermal conductivity coefficient are respectively 3700 kg/m3, 880 J/kg.K and 46 W/m.K. (Zeinali Heris, 2007).

In general, the experimental study of nanofluids is an extensive research with a long range of several variables the most important of which are thermal conductivity coefficient and fluid viscosity. As mentioned before, considering different laboratory conditions in various studies, different values have been reported in measuring such variables and also in relation with their mutual dependency or their dependency to other variables. Several various mathematical relations or models introduced in this regard and for the definition of these variables indicate the same diversity of results of conducted researches. Now if we add the non-Newtonian condition to other conditions of this literature, new variables will be added such as the method of defining fluids rheological behavior and values including power-law coefficient and index. The dependence of each of these variables to one or more other variables (for example, n and K dependence to the concentration of nanoparticles and temperature (Etemad, 2011) or the dependence of thermal conductivity coefficient to rheological behavior of fluids (Gomez diaz, 2003)) and their mutual effectiveness and also the dependence of these variables to temperature and other physical or geometrical characteristics of problem, broaden more the scope of these studies and researches.

In this paper, power-law model and equation (8) were used to define the rheological behavior of fluids:

$$\tau_{xy} = K^*(\dot{\gamma}_{xy})^n \quad (8)$$

Here, τ_{xy} is the shear stress, K is the stability constant, $\dot{\gamma}_{xy}$ is the shear rate and n is the subscript or index of power-law. K and n values, namely, power-law coefficient and index are obtained from a graph in the experimental investigation conducted by Etemad (2011) for 0.5 and 1.5 volume percent of alumina suspended in non-Newtonian fluid derived from 0.5% wt solution of carboxymethyl cellulose in water. It is noteworthy that according to the investigation conducted by Etemad (2011), n and K values depend also on temperature. But since the temperature difference of the fluid at the beginning and the end of the tube in this project is not high, the power-law coefficient and index can be assumed constant according to temperature with an acceptable accuracy. But we must bear in mind that if there is an extensive temperature range in the outlet and inlet of the tube, this assumption may lead to incorrect results in the analysis of the problem.

Also in this simulation, the flow regime is turbulent and the analysis is performed for Reynolds numbers of 4500, 6000 and 8000. According to the definition of Reynolds number for non-Newtonian fluids, the role of power-law coefficient and index in measuring Reynolds numbers and the initial velocity of fluids flow in the tube inlet is very crucial. Reynolds and Prandtl numbers for non-Newtonian fluids are calculated through euqations (9) and (10):

$$Re = \frac{\rho.u^{2-n}.D^n}{k} \quad (9)$$

$$Pr = \frac{C_p.k.(\frac{u}{D})^{n-1}}{K} \quad (10)$$

Here, K is the fluid thermal conductivity and k is the power-law coefficient.

Considering the symmetry of the problem geometry in this study, the analysis is performed in a two-dimensional space. To mesh the flow geometry by using Gambit software, 10 meshes in radial direction and 1200 meshes along the tube were considered and the independence of results based on the above gridding was investigated. The present survey was conducted taking into consideration the constant heat flux boundary condition on the tube walls, bearing in mind that this boundary condition is commonly used in similar investigations. The ratio of this heat flux was assumed to be 100,000 W/m^2 or 100 kilowatts per square meter.

It is noteworthy that due to high Reynolds number and turbulent regime of the flow, the fluid has a high velocity inside the tube and, as a result, heat exchange is less achievable. Thus, as it can be seen in the investigation carried out by Bianco (2011), in turbulent flow, the amount of heat flux in order to reach developed zone of the temperature was considered higher than that in low Reynolds numbers and laminar flow. Inlet fluid temperature is uniformly 300 K and fluid velocity in the inlet of the tube is uniform widthwise. So, the velocity will be different according to the Reynolds number.

RESULTS

Local, convective heat transfer coefficient is determined by the following equation:

$$h(x) = \frac{q''}{T_w(x) - T_f(x)} \quad (11)$$

In the above equation, q" is the constant heat flux on the tube walls and its value was assumed to be 100 kilowatts per square meter. $T_w(x)$ and $T_f(x)$ respectively are local wall temperature along the tube and local mean temperature of fluid. $T_w(x)$ can be obtained directly from the Fluent output curves. To calculate $T_f(x)$, the following equation is used:

$$T_f(x) = \frac{2}{D}\sum_{r=0}^{R} T(r)\, d_r \qquad (12)$$

In the above equation, D is the tube diameter and T(r) is fluid temperature in radial direction in each point of the tube length. So, using the outputs of the software and the above equations, the convective heat transfer coefficient can be measured. Local Nusselt number also can be calculated by the following equation:

$$Nu_{nf} = \frac{h_{nf}\, D}{k_{nf}} \qquad (13)$$

Figures (2) to (5), in the form of graphs, indicate the fluid temperature in the vicinity of the wall along the tube and the fluid mean or bulk temperature along the tube and also the difference between the above temperatures. These graphs are plotted for base non-Newtonian fluids and non-Newtonian nanofluids containing alumina particles with a volumetric concentration of %1.5 and a size of 25 nm.

As it can be seen, the fluid temperature besides the tube is increased in a curved form with an abrupt steep prior to temperature developed zone. Then, it becomes linear in the developed zone. The line steep, as can be seen, is equal to that of fluid mean temperature changes. Hence, the difference of temperature on the tube walls and the local, mean temperature of the fluid stay unchanged in the developed zone. This is shown in a better way in figure 5.

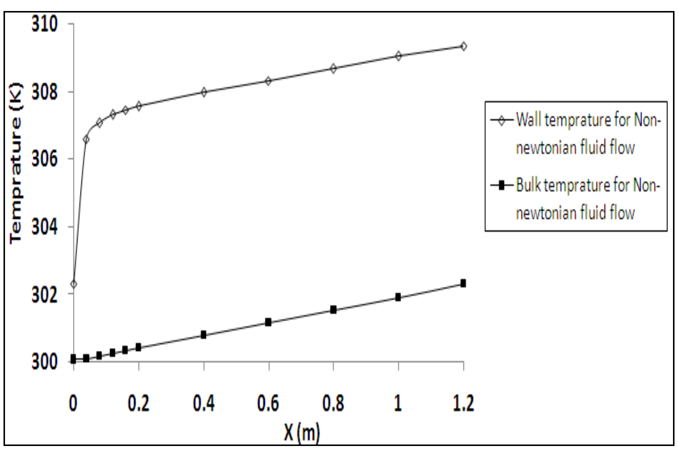

Figure 2:Variation of wall and bulk temperature of base non-Newtonian fluid in the Reynolds number of 8000

Another important point is deductible in figure 4 which is a comparison between figures 2 and 3. This figure shows clearly that the fluid temperature on the wall in the base non-Newtonian fluid is more than the similar value for the non-Newtonian nanofluid. Also, this figure along with figure 5 indicate that the difference of temperature on the wall and local, mean temperature of the base non-Newtonian fluid is more than that in the similar non-Newtonian nanofluid.

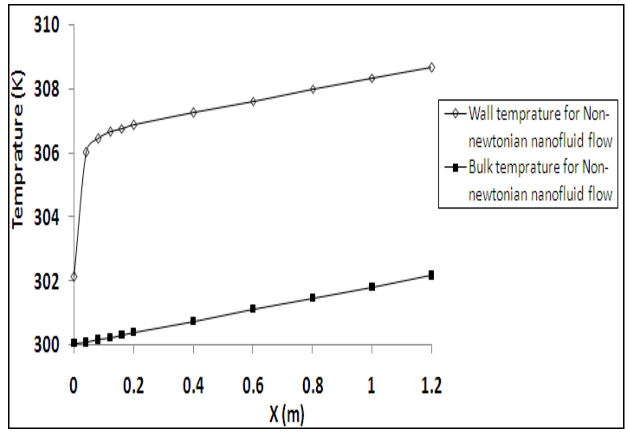

Figure 3: Variation of wall and bulk temperature of non-Newtonian nanofluid containing alumina particles with a volumetric concentration of %1.5 and a size of 25 nm

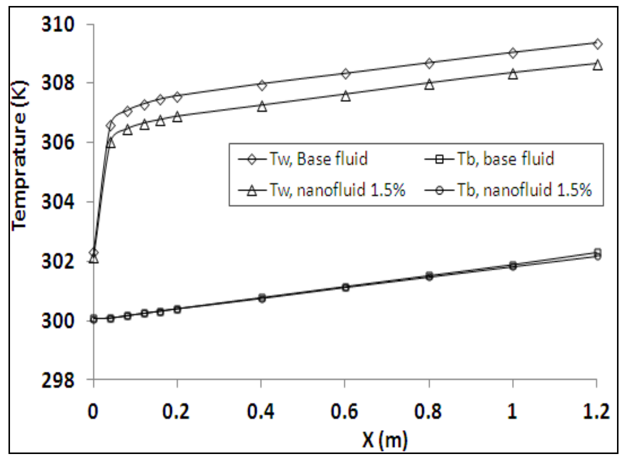

Figure 4: Comparison between variation of wall and bulk Temperature in both the base non-Newtonian fluid and non-Newtonian nanofluid containing alumina particles with a volumetric concentration of %1.5 and a size of 25 nm

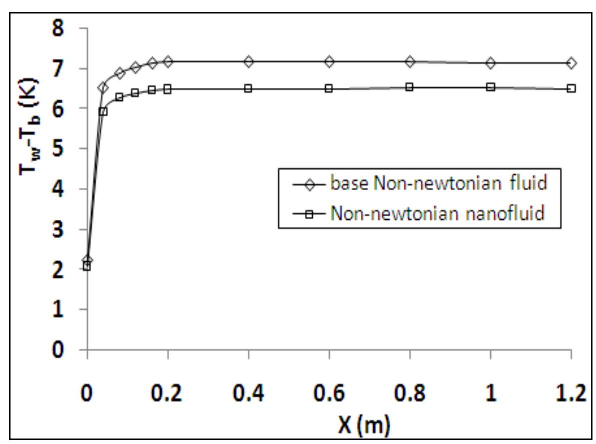

Figure 5: Difference of wall and bulk temperature in both the base non-Newtonian fluid and non-Newtonian nanofluid containing alumina particles with a volumetric concentration of %1.5 and a size of 25 nm

In figure 6, local, convective heat transfer coefficient in the mid-section of the tube in the developed zone are plotted for pure water Newtonian fluid, the non-Newtonian fluid of carboxymethyl cellulose solution in water and non-Newtonian nanofluid containing 25 nm alumina particles with two volumetric concentrations of 0.5% and 1.5% . As outlined, the conversion of fluid from Newtonian to pseudo-plastic non-Newtonian results in considerable increase of convective heat transfer coefficient. This is essentially one of the effective methods in the increase of heat transfer coefficient (Etemad,1994). The fluid heat transfer coefficient enhancement with the addition of nanoparticles to the fluid can also be observed in figure 6.

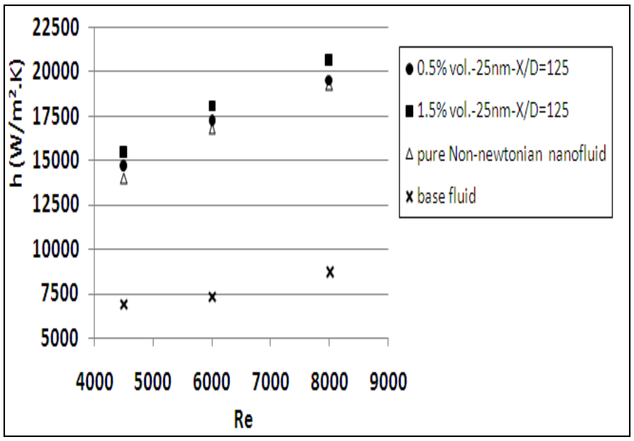

Figure 6: The effect of particles concentration and Reynolds number on local, convective heat transfer coefficient for 25 nm particles

As can be deducted by figures 7 to 9, the effect of nanoparticles low concentration in higher Reynolds number is decreased. The addition of 0.5% of 25 nm alumina particles to base non-Newtonian fluid with Reynolds number of 4500, leads to an increase of almost 5% of heat transfer coefficient. But this number is decreased by 1.5% for Reynolds number of 8000. Therefore, it seems that in higher Reynolds numbers, higher concentrations of nanoparticles are needed. Considering the turbulent regime and higher flow velocity, the enhancement of nanoparticles concentration is more feasible, because it diminishes the probability of deposition or settling in channels. In addition, these figures show the changes of local, convective heat transfer coefficient along the tube for the relevant non-Newtonian nanofluid with 0.5% and 1.5% concentration, base non-Newtonian nanofluid and also pure water. These graphs are plotted respectively for Reynolds number in a range of 4500, 6000 and 8000. The difference between the curves of non-Newtonian fluid and nanofluid with a concentration of 0.5% is lessening from figure 7 to 9; so that in figure 9, the two curves are quite adjacent. However, the enhancement of nanofluid convective heat transfer coefficient with a concentration of 0.5% has been significantly preserved in all three graphs.

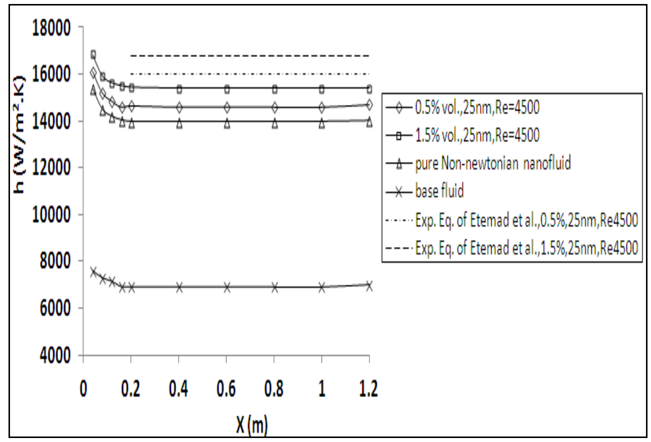

Figure 7: The effect of 25 nm particles concentration on convective heat transfer coefficient with Reynolds number of 4500

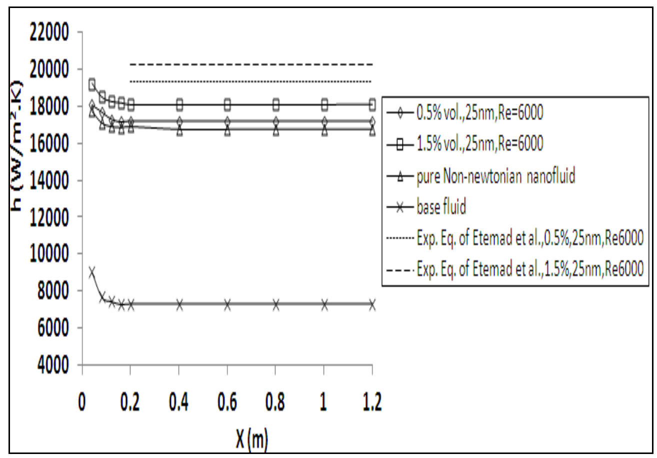

Figure 8: The effect of 25 nm particles concentration on convective heat transfer coefficient with Reynolds number of 6000

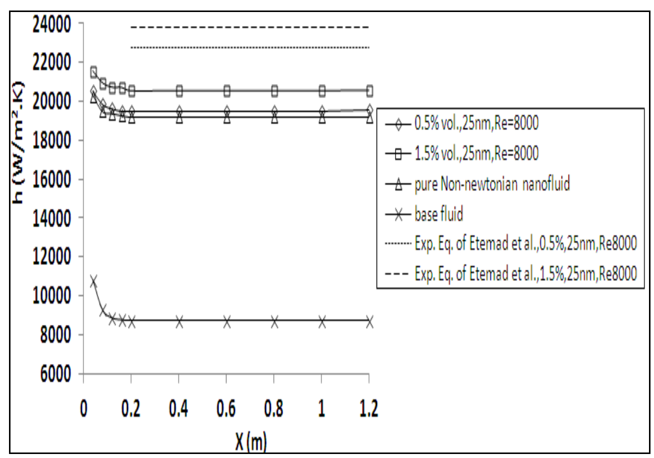

Figure 9: The effect of 25 nm particles concentration on convective heat transfer coefficient with Reynolds number of 8000

Accordingly, the study of the above figures suggests that with the increase of 25 nm alumina nanoparticles concentration in the base non-Newtonian fluid from 0.5% to 1.5%, the nanofluid convective heat transfer coefficient will be increased. This increase is more noticeable in lower Reynolds numbers. And this level is decreased gradually with the increase of Reynolds number and turbulent regime of fluid.

Figure 10 shows the comparison between the Nusselt numbers obtained from this study and the results obtained from the correlation of the experimental investigation conducted by Etemad (2011), (by inserting Reynolds and Prandtl numbers in the mentioned correlation). Since the above experimental investigation has been conducted only for $A1_2O_3$ nanoparticles of size 25 nm, the mentioned experimental correlation can be used only for 25 nm particles. As a result, figure 10 is plotted only for nanoparticles of the same size but with different concentrations. Based on figure 10 above, it can be said that the Nusselt numbers obtained in this study have acceptable accuracy compared with the abovementioned experimental investigation, for non-Newtonian nanofluids used, namely, the aqueous solution of carboxymethyl cellulose containing $A1_2O_3$ nanoparticles of size 25 nm.

The main difference between numerical investigations and simulations in nanofluids and laboratory results is about the computational difference of thermophysical properties including the most important of them namely the nanofluids heat transfer coefficient: none of these equations for measuring the nanofluids heat transfer coefficient is individually sufficient to predict accurately this property under different, actual conditions. However, due to the relatively satisfactory results of this study in comparison with the results derived from the experimental investigation conducted by Etemad (2011), it seems that the use of the correlation, employed by Chon et al (equation 6) to determine nanofluids thermal conductivity coefficient in this study, has led to accurate and adequate results.

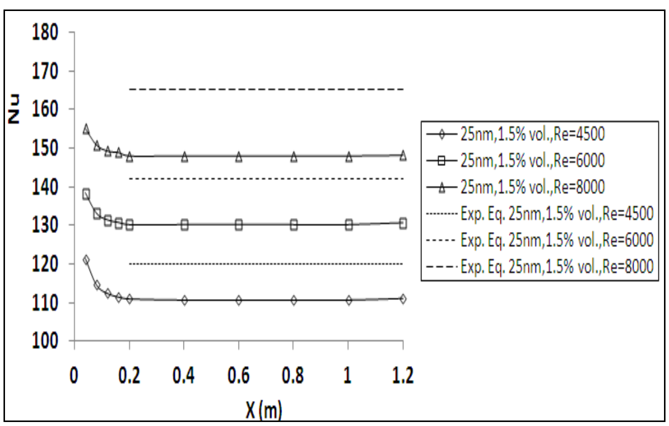

Figure 10: The effect of Reynolds number on Nusselt number in non-Newtonian nanofluid with particles of constant concentration and size

COCLUSION

In this study, we simulated and analyzed the convective heat transfer of a non-Newtonian nanofluid in the turbulent flow of a circular tube using Fluent software. The effect of particles and Reynolds number on local, convective heat transfer and Nusselt number was investigated and the following results were obtained:

- The change of rheological behavior of a fluid from Newtonian to non-Newtonian increases noticeably the base fluid convective heat transfer coefficient
- The non-Newtonian nanofluids convective heat transfer coefficient is higher than pure non-Newtonian fluid.
- The increase of nanofluids concentration leads to the increase of convective heat transfer coefficient and the Nusselt number of non-Newtonian nanofluids.
- Reynolds number has a significant impact on convective heat transfer coefficient and Nusselt number of non-Newtonian nanofluid and its increase will lead to the increase of this coefficient.

REFERENCES

Maxwell, J. C. (1873): A Treatise on Electricity and Magnetism, Clarendon Press, Oxford.

Maxwell, J.C. (1881): A Treatise on Electricity and Magnetism, Oxford University Press.

Santra, A.K., Sen, S., Chakraborty, N. (2009): Study of heat transfer due to laminar flow of copper–water nanofluid through two isothermally heated parallel plates, International Journal of Thermal Sciences, **48** : 391–400.

Wen, D., and Ding, Y. (2004): Experimental Investigation into Convective Heat Transfer of Nanofluids at the Entrance Region under Laminar Flow Conditions, International Journal of Heat and Mass Transfer, **47**(24) : 5181-5188.

Yang, Y., Zhang, Z.G., Grulke, E.A., Anderson, W.B., Wu, G. (2005): Heat transfer properties of nanoparticle-in-fluid dispersions (nanofluids) in laminar flow, International Journal of Heat and Mass Transfer, **48** : 1107–1116.

He, Y., Jin, Y., Chen, H., Ding,Y., Cang, D., Lu, H. (2007): Heat transfer and flow behavior of aqueous suspensions of TiO2 nanoparticles (nanofluids) flowing upward through a vertical pipe, International Journal of Heat and Mass Transfer, **50**: 2272–2281.

Rea, U., McKrell, T., Hu, L.W., Buongiorno, J. (2009): Laminar convective heat transfer and viscous pressure loss of alumina-water and zirconia-water nanofluids, International Journal of Heat and Mass Transfer, **52** : 2042–2048.

Chun, B.H., Kang, H.U., Kim, S.H. (2008): Effect of alumina nanoparticles in the fluid on heat transfer in double-pipe heat exchanger system, Korean Journal of Chemical Engineering, **25**:966–971.

Behzadmehr, A., Saffar-Avval, M., and Galanis, N. (2007): Prediction of Turbulent Forced Convection of a Nanofluid in a Tube with Uniform Heat Flux Using a Two Phase Approach, International Journal of Heat and Fluid Flow, **28**(2):211-219.

Bianco, V., Manca, O., Nardini, S. (2011): Numerical investigation on nanofluids turbulent convection heat transfer inside a circular tube, International Journal of Thermal Sciences, **50**:341-349.

Akbari, M., Galanis, N., Behzadmehr, A. (2012): Comparative assessment of single and two-phase models for numerical studies of nanofluid turbulent forced convection, International Journal of Heat and Fluid Flow, **37**:136–146.

Soltani, S., Etemad, S.Gh., Thibault, J. (2010): Pool boiling heat transfer of non-Newtonian Nanofluids, International Communications in Heat and Mass Transfer **37** : 29–33.

Kamali, R., Binesh, A.R. (2010): Numerical investigation of heat transfer enhancement using carbon nanotube-based non-Newtonian nanofluids, International Communications in Heat and Mass Transfer **37**: 1153–1157.

Hojjat, M., Etemad, S.Gh., Bagheri, R. (2010): Laminar heat transfer of non-Newtonian nanofluids in a circular tube, Korean Journal of Chemical Engineering, Volume 27, Issue **5**:1391-1396.

Hojjat, M., Etemad, S.Gh., Bagheri, R. Thibault, J. (2011): Convective heat transfer of non-Newtonian nanofluids through a uniformly heated circular tube, International Journal of Thermal Sciences **50**:525–531.

Moraveji, M.K., Haddad, S.M.H., Darabi, M. (2012): Modeling of forced convective heat transfer of a non-Newtonian nanofluid in the horizontal tube under constant heat flux with computational fluid dynamics, International Communications in Heat and Mass Transfer, **39**:995-999.

Pak, B.C. and Cho, Y.I. (1998): Hydrodynamic and Heat Transfer Study of Dispersed Fluids with Submicron Metallic Oxide Particles, Exp. Heat Transfer, **11**(2) : 151-170.

Chon, C. H., Kihm, K. D., Lee, S. P., and Choi, S. U. S. (2005): Empirical Correlation Finding the Role of Temperature and Particle Size for Nanofluid (Al_2O_3) Thermal Conductivity Enhancement, Applied Physics Letters, **87**(15).

Zeinali Heris, S., Nasar Esfahany, M., Etemad, S.-G. (2007): Numerical investigation of nanofluid laminar convective heat transfer trough a circular tube, Numerical Heat Transfer Part A **52** : 1043-1058.

Hojjat, M., Etemad, S. Gh., Thibault, J. (2011): Rheological characteristics of non- Newtonian nanofluids: Experimental investigation, International Communications in Heat and Mass Transfer, **38** : 144–148.

Gomez diaz, D., Navaza, J.M. (2003): Rheology of aqueous solutions of food additives Effect of concentration, temperature and blending, Journal of Food Engineering, **56** : 387–392.

Etemad, S. GH., Mujumdar, A.S., Huang, B. (1994): Viscous dissipation effects in entrance region Heat transfer for a power law fluid flowing between parallel plates, International Journal of Heat and Fluid Flow, **15**:122.

Hojjat, M., Etemad, S.GH., Bagheri, R.,Thibault, J. (2011): Turbulent forced convection heat transfer of non-Newtonian nanofluids, Experimental Thermal and Fluid Science, **35**: 1351-1356.