EXPERIMENTAL STUDY ON TURBULENT HEAT TRANSFER, PRESSURE DROP, AND THERMAL PERFORMANCE OF ZnO/WATER NANOFLUID FLOW IN A CIRCULAR TUBE

by

Ahmad Reza SAJADI [a], Seyed Soheil SADATI [b], Masoud NOURIMOTLAGH [c], Omid PAKBAZ [a], Dariush ASHTIANI [a*], and Farshad KOWSARI [a]

[a] School of Mechanical Engineering, College of Engineering, University of Tehran, Tehran, Iran
[b] Department of Mechanical Engineering, Damavand Branch, Islamic Azad University, Tehran, Iran
[c] Young Researchers Club, Dareshahr Branch, Islamic Azad University, Iran

Original scientific paper
DOI: 10.2298/TSCI131114022S

In this experimental study heat transfer and pressure drop behavior of ZnO/water nanofluid flow inside a circular tube with constant wall temperature condition is investigated where the volume fractions of nanoparticles in the base fluid are 1% and 2%. The experiments' Reynolds numbers ranged roughly from 5000 to 30000. The experimental measurements have been carried out in the fully-developed turbulent regime. The results indicated that heat transfer coefficient increases by 11% and 18% with increasing volume fractions of nanoparticles, respectively, to 1 vol.% and 2 vol.% . The measurements also showed that the pressure drop of nanofluids were, respectively, 45% and145% higher than that of the base fluid for volume fractions of 1% and 2% of nanoparticles. However experimental results revealed that overall thermal performance of nanofluid is higher than that of pure water by up to 16% for 2 vol.% nanofluid. Also experimental results proved that existing correlations can accurately estimate nanofluids convective heat transfer coefficient and friction factor in turbulent regime, provided that thermal conductivity, heat capacity, and viscosity of the nanofluids are used in calculating the Reynolds, Prandtl, and Nusselt numbers.

Key words: *nanofluid, convective heat transfer, turbulent flow, friction factor, overall thermal performance*

Introduction

Nanofluid as a promising technology is supposed to enhance the heat transfer capabilities of conventional liquids both conductively and convectively [1-3]. Greater amount of working fluids' heat transfer will help thermal systems in many industries, such as energy, electronics, and transportation to be designed smaller and/or more efficient. Also the conventional working fluids which possess low thermal conductivity coefficients can no longer meet the requirements of high-intensity heat transfer duties.

The conventional methods to improve the heat transfer rate include passive techniques with application of extended or rough surfaces and swirl flow, and active techniques with sur-

* Corresponding author; e-mail: dariusha63@gmail.com

face or fluid vibration and mechanical aids [4-7]. However, these enhancing techniques have reached a bottleneck in regard to further improvement of the heat transfer rate. Taking this fact as granted, a search for high efficiency heat transfer fluids has started since [8] owing to the technologies which made it possible to produce solid particles of sizes smaller than 100 nm.

Nanofluids (nanoparticle fluid suspensions) is the term coined by Choi [9] to describe this new class of nanotechnology-based heat transfer fluids that exhibit thermal properties superior to those of their host fluids or conventional particle fluid suspensions. Since then many researchers investigated the heat transfer characteristics of nanofluids experimentally [10-13] as well as numerically [14-16]. They have achieved remarkable findings of up to 30% increase in heat transfer that brighten the future path for further studies in this field. It is noteworthy to mention that the majority of researches have converged to the point of nanofluids' superior performance compared to those of their corresponding base fluids.

Convective heat transfer researches of nanofluids are more recent than thermophysical properties ones. There is almost unanimous agreement in nanofluids' better heat transfer rate in comparison to their associated base fluids, especially for lower particle concentrations, say less than 0.5% by volume. Sajadi and Kazemi [17] studied turbulent convective heat transfer performance of TiO_2/water nanofluid with nanoparticles volume fractions less than 0.25%. They have observed that adding nanoparticles enhances heat transfer rate but there were no significant difference between results for various nanoparticles concentrations. Ashtiani *et al.* [18] have experimentally worked on heat transfer characteristics of MWCNT/HT-B oil nanofluid and observed that higher concentrations of this suspension shows far more heat transfer enhancement in a circular tube, while this enhancement in tubes with higher flattening degrees is weaker. Their volume fraction was just little above 0.25 at maximum. Fotukian and Nasr [19] studied turbulent convective heat transfer performance of dilute (less than 0.24 vol.%) Al_2O_3/water nanofluid. In their study, increasing the volume fraction of Al_2O_3 particles in nanofluid had negligible effect on the heat transfer enhancement.

For higher particle concentrations there are studies that claim higher heat transfer as well. Heyhat *et al*. [20] have worked on laminar flow convective heat transfer characteristics of alumina water nanofluids in fully developed flow regime. Their results showed that the heat transfer coefficient of nanofluid is higher than that of the base fluid and increases with increasing the Reynolds number and particle concentrations. They have reported a maximum heat transfer coefficient boostof approximately 32% at 2 vol.% nanofluid. In a similar study Heyhat *et al.* [21] have changed the flow condition from laminar to turbulent and drawn the conclusion that again heat transfer of nanofluid was better. The only difference between the two studies was the compliance of experimental data with existing correlations in each case where in turbulent flow the correlation could predict the results well.

Researches considering aqueous ZnO nanoparticles, report higher thermal characteristics of it compared to pure water. For instance HaghshenasFard *et al.* [22] have studied ZnO/water nanofluid heat transfer in concentric tube and plate heat exchangers and obtained 14% and 20% increase of heat transfer coefficient compared to base fluid, respectively. In another study thermal conductivity of an ethylene glycol/ZnO nanofluid was investigated by Lee *et al*. [23] They have proposed a one-step physical method that enhances the nanofluid thermal conductivity more than the value predicted through Hamilton-Crosser correlation.

Looking up further in literature one would face contradicting reports about nanofluid performance as well. Some results show less heat transfer of nanofluids mostly with higher concentrations, while others present different behaviors.

Pak and Cho [24] studied heat transfer behavior of Al_2O_3 and TiO_2 nanofluids up to 4 vol.% concentration, they reported that at fixed Reynolds numbers heat transfer coefficient increases with increase of nanoparticles concentration. However, they have found that the convective heat transfer coefficient of nanofluid was smaller than that of pure water when compared under the condition of constant average velocity.

Williams and Buongiorno [25] studied nanofluids with 0.9 to 3.6 vol.% concentration of Al_2O_3 nanoparticles. They have found that heat transfer rate boosts with increasing nanoparticles concentration.

Duangthongsuk and Wongwises [26] experimentally studied TiO_2/water heat transfer characteristics in the range of 0.2 to 2 vol.% concentration of nanoparticles. They observed that heat transfer grows by increasing nanoparticles fraction up to 1 vol.% concentration. For higher concentrations, however, heat transfer coefficient growth stops and starts to decrease.

There are parameters other than nanoparticle concentration which affect thermophysical properties and hence heat transfer performance of nanofluids. For example, increasing zeta potential which is determined by pH value of fluid minimizes particle-particle interaction and then decreases viscosity and thermal conductivity of suspension [27, 28]. Indeed, agglomeration and clustering of nanoparticles should be taken into account to avoid undesirable settlement. The next important parameter is the shape of nanoparticles which traditionally affects thermal conductivity through Hamilton-Crosser correlation. However, Timofeeva *et al.* [29] have studied alumina nanoparticles shape effects on thermophysical properties of its ethylene glycol and water based nanofluids and have found that enhancements in the effective thermal conductivities due to particle shape effects expected from Hamilton-Crosser equation are strongly diminished by interfacial effects proportional to the total surface area of nanoparticles.

Since not much work is done directed to heat transfer performance of aqueous ZnO nanofluids and indeed there is lack of unanimous agreement in results about heat transfer characteristics of nanofluids with higher particle concentrations, this study aims at investigating the heat transfer and pressure drop of ZnO water-based nanofluids flow inside a circular tube with constant wall temperature. The research also targets at whether such a cost effective nanofluid can be of satisfactory thermal performance.

Nanofluid preparation and characterization

The diameter of the ZnO nanoparticles used ranges from 20 to 40 nm which were provided from Merch Company. The TEM (transmision electron microscope) picture of ZnO nanoparticles is shown in fig. 1. As can be seen nanoparticles are not spherical but are of a kind of hexagonal shape. Two volume concentrations of 1% and 2% were chosen for the tests and thermophysical properties of these nanofluids were measured at different temperatures experimentally. Working fluids density, heat capacity, viscosity, and thermal conductivity were measured in temperature interval of 30 °C to 90 °C. All measurements were made by precise apparatuses in Iran Research Institute of Petroleum Industry. Density was measured by SVM300 with 0.0005 kg/m^3 accuracy, viscosity with Brookfield DV-II+ Pro Programmable

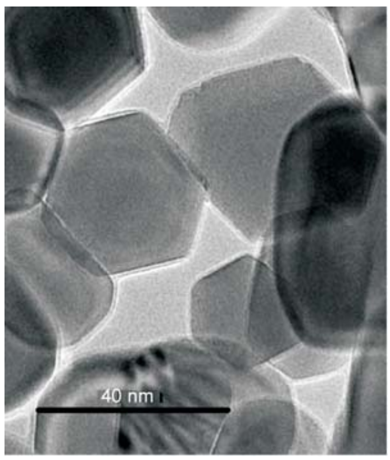

Figure 1. TEM of ZnO nanoparticles

Viscometer with accuracy of 1.0% of its full scale range, thermal conductivity with KD2 with 5% accuracy, and heat capacity with differential scanning calorimeter DSC F3 Maia by NETZSCH with 5% accuracy. The proper amount of ZnO nanoparticles were mixed into distilled water by a mixer for 30 minutes. Then ultrasonic cleaner (model UP400S-Hielcher) was used to disperse nanoparticles for thirty minutes. The resulting suspension did not tend to show any settlement for 4 days in aquiescent condition. Before each test this mixing and ultrasonication were done to ensure minimal clustering and/or settlement.

Experimental apparatus and procedure

In order to study the flow and convective heat transfer features of the nanofluids in a circular tube, an experimental apparatus was assembled to aid this process. The schematic shown in fig. 2 includes all necessary features of the experimental system consisting of a fluid reservoir tank, a pump, a flow loop, a test section, a cooler and a steam supplier tank. The transparent plastic fluid reservoir tank with capacity of 4 liter was applied to reserve the nanofluid and monitor the sedimentation rate and the height of nanofluid column which ensures NPSHR of the pump and in turn regular single-phase flow without any bubbles formed within the flow.

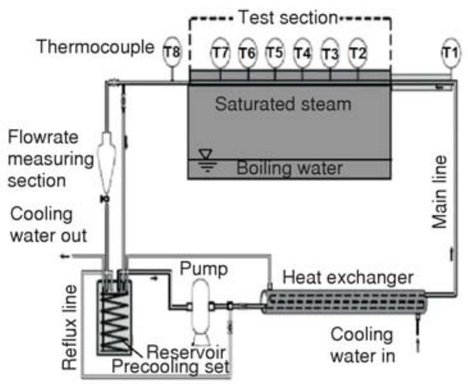

Figure 2. Schematic diagram of experimental set-up

The test section is a straight copper tube with the inner diameter, thickness and length of about 9, 0.5, and 1800 mm, respectively. Six K-type thermocouples were mounted on the copper tube wall at equal intervals to measure the wall temperature. Two other K-type thermocouples were inserted at the inlet and outlet of the test section to measure the fluid bulk temperature. All the thermocouples were connected to separate thermometers (of type Hyundai) for comfortable and accurate temperature read. The first 50 cm of the copper tube at upstream was left outside the steam tank which means before the application of the thermal condition to guarantee hydrodynamically fully-developed flow condition and indeed thermally isolated by fiberglass to ensure minor heat loss which helps the accuracy of inlet flow temperature read at the beginning of the test section. The flow rate was controlled with two adjusting valves, one at the end of the test section and the other at the by-pass line. The cooler includes a shell and tube heat exchanger which was used to reduce the inlet nanofluid temperature to the test section. This means that the cooler's function is to dissipate the heat gained by the working fluid in the steam tank in order for the fluid to not become so hot while it circulates in the loop. The 50-liter steam supplier tank contains water as well as 8 kW electric element heaters to generate fully saturated vapor which surrounds the next 120 cm of tube outside wall to keep constant its surface temperature. The conventional method of isolation by fiberglass cover was used to minimize the heat loss from the steam tank supplier to the surrounding area, so sufficient water vapor condensation occurs on the tube outside wall and its excess amount is discharged through steam tank infiltration. By this way water evaporation rate will not fall and then constant wall temperature condition is more consistent in all test cases with different flowrates. A precise differential pressure transducer (manufactured by Endress Hauser) with an uncertainty of ±1 Pa was employed for measuring the pressure loss along the test section tube. It is of great importance to mention that all of the measuring instruments were calibrated prior to data collection. The classical method of bucket-and-stopwatch was used to measure flowrates due to simplicity and no need for calibration. Also the flowrates encountered in the experiments are rather low, therefore this method fits well. A 1-liter calibrated glass vessel as the bucket and a regular digital chronometer of ±0.01 s accuracy as the stop-watch were utilized to yield data for flow rate calculation.

Data reduction

In order for heat transfer and pressure drop calculations to be carried out from raw measured experimental data, four main fluid properties are required: density, viscosity, thermal conductivity, and specific heat capacity. To ensure the accuracy of the results for nanofluids, these properties are preferred to be measured experimentally.

One would see that tab.1 contains measured amounts of the four necessary properties in temperature range of 30 °C to 90 °C with 20 °C steps for 1 vol.% and 2 vol.% nanofluids, respectively. The measured experimental data of density and heat capacity fall within 1% and 7% of well-known Pak and Cho correlations [24], respectively.

Table 1. Measured thermophysical properties of 1% and 2% ZnO/water nanofluid

	Nanofluid 1 vol.%				Nanofluid 2 vol.%			
T [K]	ρ [kgm^{-3}]	k [Wm^{-1}K^{-1}]	μ [Pa·s]	C_p [Jkg^{-1}K^{-1}]	ρ [kgm^{-3}]	k [Wm^{-1}K^{-1}]	μ [Pa·s]	C_p [Jkg^{-1}K^{-1}]
30	1044.2	0.662	0.00103	3957.4	1087.8	0.709	0.00132	3783.4
50	1031.2	0.693	0.00067	3955.2	1083.9	0.741	0.00092	3784.5
70	1025.5	0.714	0.00056	3963.4	1072.3	0.762	0.00064	3798.2
90	1011.6	0.727	0.00046	3967.7	1052.7	0.775	0.00047	3802.1

For each nanofluid case a curve fitting of all properties were derived and the amount of a specific property at an arbitrary temperature was interpolated linearly or of greater degree where applicable according to its fitting. Then the rheological and thermophysical properties of the nanofluids were calculated at the mean fluid temperature.

Due to constant wall temperature condition the experimental convective heat transfer coefficient of nanofluid is defined as:

$$h_{\mathrm{nf}}(\mathrm{exp.}) = \frac{(\rho C_p)_{\mathrm{nf}} A u (T_{\mathrm{b,out}} - T_{\mathrm{b,in}})}{\pi D L \Delta T_{\mathrm{LMTD}}} \tag{1}$$

$$\Delta T_{\mathrm{LMTD}} = \frac{(T_{\mathrm{s}} - T_{\mathrm{b,in}}) - (T_{\mathrm{s}} - T_{\mathrm{b,out}})}{\ln\left(\dfrac{T_{\mathrm{s}} - T_{\mathrm{b,in}}}{T_{\mathrm{s}} - T_{\mathrm{b,out}}}\right)} \tag{2}$$

where ΔT_{LMTD} is the logarithmic mean temperature difference and defined as eq. (2), and T_{s} – the wall temperature that is the average of six measured temperatures on tube wall at different positions. The inlet and outlet bulk temperatures are also measured as is discussed in the section *Ex-*

perimental apparatus and procedure. The convective heat transfer coefficient is usually expressed in the form of Nusselt number:

$$\mathrm{Nu}_{\mathrm{nf}}(\exp.) = \frac{h_{\mathrm{nf}}(\exp.)D}{k_{\mathrm{nf}}} \tag{3}$$

Examining the apparatus reliability

The reliability and accuracy of the experimental apparatus and the obtained data was investigated prior to handling measurements of convective heat transfer and pressure drop of nanofluids. This was done by using pure distilled water as the working fluid and then its comparison to valid correlations.

The well-known Dittus-Boelter correlation [30] (eq. (4)) was used for comparing its calculated values with the heat transfer results of this test:

$$\mathrm{Nu} = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{4}$$

In fact the Re-Nu line drawn by this equation should be compared to experimental scattered data in the same figure.

The experimental friction factors were compared with obtained values from the Petukhov correlation [31] (eq. (5):

$$f = [0.79\ln(\mathrm{Re}) - 1.64]^{-2} \tag{5}$$

Again the equation line and experimental data in the same figure should be plotted. If both cases show reasonable agreement, the experimental apparatus might be construed as reliable.

In fig. 3 the Re-Nu plot for distilled water flow in circular tube under constant wall temperature condition is given. The solid line is according to eq. (4) and the scattered data is obtained through experiment. Figure 4 is the same for friction factor with eq. (5) and experimental data. As is witnessed in figs. 3 and 4, well coincidence between the experimental results and the calculated values for pure distilled water revealed the good agreement between the experimental data and the prediction of the correlation (±8% and ±9% for heat transfer coefficient and friction

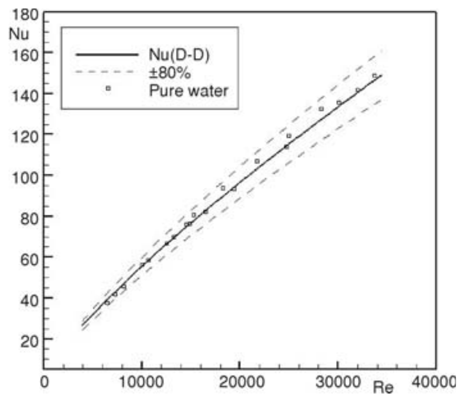

Figure 3. Comparison of experimental Nusselt number with data obtained by Dittus-Boelter correlation *vs*. Reynolds number

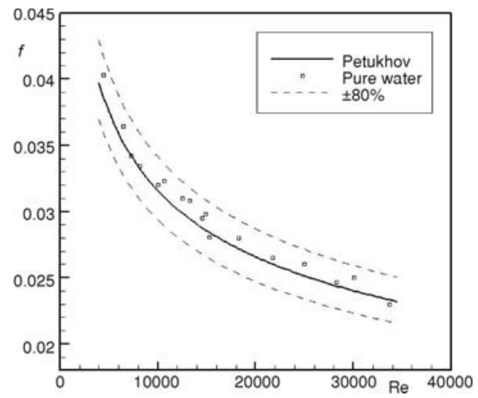

Figure 4. Comparison of experimental friction factor with data obtained by Petukhov correlation *vs*. Reynolds number

factor, respectively). This could mean that experimental data obtained by current system can be trusted as valid.

Sufficient care was taken to maintain the experimental set-up condition as consistent as possible in the other two fluid cases (nanofluids 1% and 2% by volume). Measures like preventing tube vibration, smooth pump rotation, and supplying NPSHR persistently were taken and owing to manual data collection, monitoring change of conditions was not far to reach. This could help to ensure the same level of accuracy for the results associated with nanofluids as for results of pure water.

Uncertainty analysis

There is a degree of uncertainty in calculation of flow and heat transfer parameters from raw measured data due to the errors associated with the instruments used.

Table 2. Accuracy of measuring instruments

Description	No.	Model	Accuracy in the study range
Inlet and/or outlet flow temperature	2	Type K (chromel-Alumel) thermocouple connected to Hyundai thermometer	0.2 °C
Temperature of the tube surface	6	Type K (chromel-Alumel) thermocouple connected to Hyundai thermometer	0.2 °C
Flowmeter	1	Bucket and stop watch (including calibrated glass vessel and simple digital cell phone chronometer application)	0.05 lit/s at maximum including human error*

* Accuracy was estimated by repeating measurements several times for different cases

Table 2 presents the accuracy of measuring instruments included in the assembly. Based on the method proposed by Kline and Mcclintock [32] and substituting the data from tab. 2 in it, the uncertainty analysis results are given in tab. 3.These amounts are the maximum for each parameter in all cases of experiments.

Table 3. Uncertainty of parameters

Parameter	Uncertainty
Re	±2.1%
Nu	±6.4%

Results and discussion

Heat transfer results

The heat transfer data of 1% nanofluid in terms of Nusselt number *vs.* Reynolds number is depicted in fig. 5 and Dittus-Bolter correlation is included as well for the sake of comparison. It is evident from the graph that the experimental figures are located within ±7% of the considered correlation. Taking into account the fact that measured thermophysical properties are used in Dittus-Bolter correlation for this case (1% nanofluid), one would not go astray to take this correlation as valid for this specific nanofluid. The same can be argued about 2% nanofluid as is obvious form fig. 6 due to its similarity to fig. 5.

One important conclusion that can be drawn from figs. 5 and 6 is that heat transfer enhancements of ZnO/distilled water nanofluids (1 and 2% by volume) are attributed to their

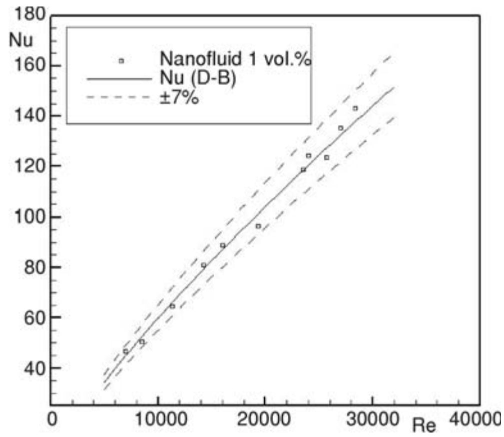

Figure 5. Comparison of experimental Nusselt numbers of 1 vol.% fraction nanofluids with the evaluated line of Dittus-Boelter correlation in the same Reynolds number range

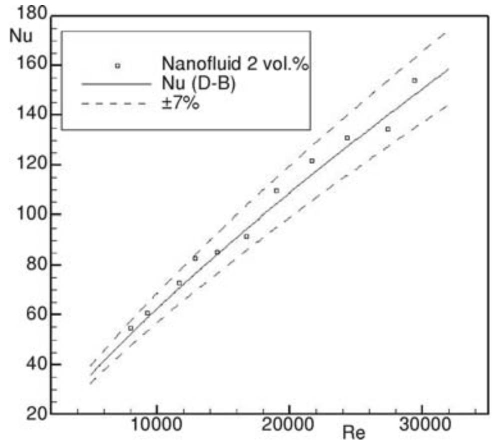

Figure 6. Comparison of experimental Nusselt numbers of 2 vol.% fraction nanofluids with the evaluated line of Dittus-Boelter correlation in the Reynolds number range

thermophysical properties' boosts and no other helping or deteriorating factor does exist or at least their effects cancel out. This argument's justification can be put forward with the support of the proximity of current experimental results and the existing proved correlation.

For the natural convection of nanofluids there are contradictory discussions about different nanoparticles in the literature [1, 14] about whether it increases or decreases heat transfer rate, however here the flow regime is turbulent and the effect of natural convection is minor.

To obtain a better understanding of heat transfer variation due to nanoparticle fraction change, the whole Nusselt number data of the three fluid types (one pure plus two nanofluids) are gathered in fig. 7. The trend of the experimental data is quite regular and so strengthens its validity and makes interpretations straightforward too. The increase in Nusselt number for higher concentrations of nanoparticles in the host fluid is apparent and hence heat transfer enhancement of nanofluids is not far to claim. Heat transfer coefficient amplified 11% and 18% relative to pure water for nanofluid with 1 vol.% and 2 vol.% fractions of ZnO nanoparticles, respectively. These amounts of heat transfer enhancements are almost the same throughout the studied Reynolds numbers range. This fact is along with the applicability of Ditus-Bolter correlation to the two nanofluids with updated properties, for this equation with enhanced values of properties will present a constant augmentation in Nusselt number throughout the Reynolds number range of the study.

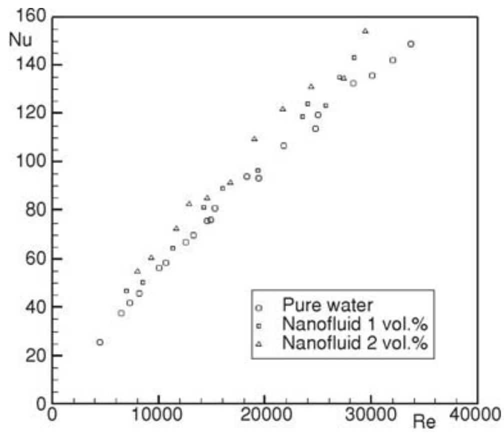

Figure 7. Nusselt number variations *vs.* Reynolds number for pure water and nanofluids with 1 vol.% and 2 vol.% fractions on nanoparticles

Pressure drop results

Friction factor is a dimensionless parameter which is a normalized form of pressure drop that is not affected by viscosity and other properties. Therefore if no other factor influences the pressure drop of a certain type of flow in a specific path (say a circular tube), changing the fluid will not tend to alter this parameter. This is conceived the same as the condition in the present experimental study. The three types of fluids possess different amounts of viscosity and hence will definitely show varying levels of pressure drops. However their friction factors seems to be of the same amount in the study range as is witnessed in fig. 8 where they are located within a ±10% margin of Petukhov correlation. The reason can be put in this way that the only effective parameter in nanofluid pressure drop is viscosity, while if ever any other increasing or decreasing factor was present, their effects would cancel out. It is quite clear that higher nanoparticle concentration will give rise to the nanofluid viscosity as is observed in tabs. 1 and 2. Hence to conclude finally one can say that the friction factor values will remain the same despite of nanoparticle concentration variation.

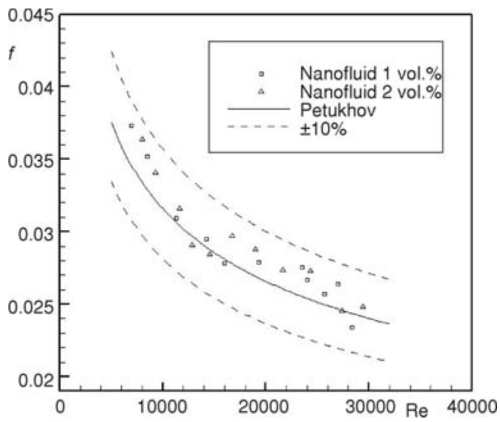

Figure 8. Comparison of experimental friction factors of working nanofluids and evaluated ones from Petukhov correlation

To explore more on pressure drop, in fig. 9 compared pressure drops of nanofluids with pure water is given. It shows that at a given Reynolds number pressure drop of nanofluids increases significantly as nanoparticles concentration rises. For example for Reynolds number around 20000, pressure drop of nanofluids are 45% and 145% higher than that of pure water for 1% and 2% volume fractions of nano-particles, respectively.

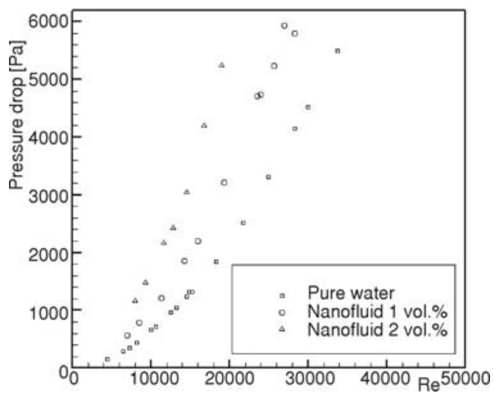

Figure 9. Pressure drop of nanofluids and pure water *vs.* Reynolds number

Overall thermal performance of nanofluids

From the previous discussions just given, it is evident that heat transfer and pressure drop both are higher for greater concentrations of nanoparticles in the host fluid. However considering system performance, these two parameters contradict each other. In order for better comparison of nanofluids heat transfer behavior to be carried out, the overall thermal performance of nanofluids is defined as the ratio of thermal efficiency rise to pumping power rise as:

$$\eta = \frac{\dfrac{\varepsilon_{\mathrm{nf}}}{\varepsilon_{\mathrm{w}}}}{\dfrac{\dot{W}_{\mathrm{nf}}}{\dot{W}_{\mathrm{w}}}} \tag{6}$$

In which thermal efficiency of each fluid can be calculated through eq. (7):

$$\varepsilon = \frac{q}{q_{\max}} = \frac{\dot{m}C_p(T_{\mathrm{b,out}} - T_{\mathrm{b,in}})}{\dot{m}C_p(T_{\mathrm{s}} - T_{\mathrm{b,in}})} \tag{7}$$

and the pumping power is:

$$\dot{W} = Q\Delta P \tag{8}$$

That is proportional to pressure drop in each case.

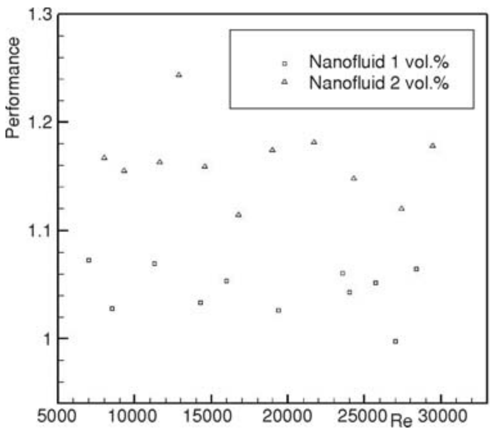

Figure 10. Overall thermal performance of nanofluid *vs.* Reynolds number

Figure 10 shows the variation of the overall thermal performance with Reynolds numbers for 1 vol.% and 2 vol.% fractions of nanoparticles. Although the trend seems erratic and irregular for each case, generally higher thermal performance of thicker nanofluids is apparent. Actually the overall thermal performance of 1% nanofluid varies between just above 1 to less than 1.08, while the figure for 2% nanofluid is always higher than that of the former by around 10% on average which may even exceed 1.2. Unfortunately in each case there is no meaningful variation of overall thermal performance with Reynolds number, however the data fluctuate around an average number for each case (approximately 1.04 and 1.16 for 1% and 2% nanofluids, respectively).

Conclusion

The following conclusions have been drawn from the present study.

- For 1 vol.% fraction of ZnO, an increase of about 11% in the heat transfer coefficient was occurred in comparison with purewater. In case of 2 vol.% nanofluid, heat transfer enhancement was about 18%. These amounts of heat transfer enhancements were almost the same throughout the studied Reynolds numbers range.
- Common correlations for heat transfer coefficient and friction factor were applicable for nanofluids when their measured thermophysical properties were used in correlations. Dittus-Boelter correlation estimates nanofluid Nusselt number with maximum deviation of 7% from experimental results and Petukhov correlation also predicts friction factor of nanofluids properly (with 10% maximum deviation).
- Nanoparticles concentration change does not affect friction factor.
- Pressure drop however will leap by adding ZnO nanoparticles to water. More nanoparticles concentration results in higher pressure drop. This fact along with no change in friction factor means nanofluid pressure drop can be predicted just by knowing viscosity.
- Suspending ZnO in water as the base fluid increases its overall thermal performance. Higher nanoparticles concentration results in greater overall thermal performance. This parameter, being erratic in the range of study, does not show meaningful growth or decline while Reynolds number increases, however the data fluctuate around an average number (approximately 1.04 and 1.16 for 1% and 2% nanofluids, respectively) for each case.

Acknowledgments

The authors would like to express their thanks to Iran Research Institute of Petroleum Industry (IRIPI), engineering college of University of Tehran, and also Islamic Azad University for financial supports through the set-up construction, properties measurement, and research implementation.

Nomenclature

A – tube cross-section area, [m^2]
C_p – specific heat capacity, [kJkg^{-1}K^{-1}]
D – diameter of the tube, [m]
f – friction factor (= $2\Delta P/\rho u^2$), [–]
h – heat transfer coefficient, [Wm^{-2}K^{-1}]
k – thermal conductivity, [Wm^{-1}K^{-1}]
L – length of the tube, [m]
Nu – Nusselt number (= hD/k), [–]
ΔP – pressure drop, [Pa]
Pr – Prandtl number (= $\mu C_p/k$), [–]
Q – flowrate, [m^{3}s^{-1}]
q – heat transfer rate, [W]
Re – Reynolds number, (= uD/ν), [–]
T – temperature, [K]
u – mean fluid velocity, [ms^{-1}]

Greek symbols

ε – convection thermal efficiency, [–]
η – overall thermal performance, [–]
μ – viscosity, [Pa·s]
ν – kinematics viscosity, [m^{2}s^{-1}]
ρ – density, [kgm^{-3}]
φ – nanoparticle volume fraction, [–]

Subscripts

b – bulk
exp. – obtained experimentally
in – inlet
max – maximum
nf – nanofluid
Np – solid nanoparticle
out – outlet
s – tube surface
w – water (base fluid)

References

[1] Ternik, P., Rudolf, R., Conduction and Convection Heat Transfer Characteristics of Water-Based Au Nanofluids in a Square Cavity with Differentially Side Walls Subjected to Constant Temperatures, *Thermal Science, 18* (2014), Suppl. 1, pp. S189-S200
[2] Ternik, P., Rudolf, R., Laminar and Natural Convection of Non-Newtonian Nanofluids in a Squre Enclosure with Differentially Heated Side Walls, *Int. J. Simul. Model, 12* (2013), 1, pp. 5-16
[3] Soltanipour, H., *et al*., Numerical Analysis of Heat Transfer Enhancement with the Use of γ-Al_2O_3/Water Nanofluid and Longitudinal Ribs in a Curved Duct, *Thermal Science, 16* (2012), 2, pp. 469-480
[4] Bejan, A., Kraus, A. D., *Heat Transfer Handbook*, John Wiley and Sons Inc., New York, N. J., USA, 2003, pp. 1029-1130
[5] Webb, R. L., *Principles of Enhanced Heat Transfer*, John Wiley and Sons Inc., New York, N. J., USA, 1994
[6] Webb, R. L., Kim, N. H., *Principles of Enhanced Heat Transfer*, Taylor and Francis, New York, USA, 2005
[7] Bergles, A. E., ExHFT for Fourth Generation Heat Transfer Technology, *Exp. Therm. Fluid Sci., 26* (2002), 2-4, pp. 317-323
[8] Das, S. K., *et. al.*, *Nanofluids Science and Technology*, John Wiley and Sons Inc., New York, N. J., USA, 2007
[9] Choi, S. U. S., Enhancing Thermal Conductivity of Fluids with Nanoparticles, Developments and Applications of Non-Newtonian Flows, FED-vol. 231/MD-vol. 66, *ASME,* 1995, pp. 99-105
[10] Lee, J. H., *et al*., Effective Viscosities on Thermal Conductivities of Aqueous Nanofluid Containing Low Volume Concentrations of AL_2O_3 Nanoparticles, *Int. J. Heat Mass Transfer., 51* (2008), 11-12, pp. 2651-2656
[11] Fotukian, S. M., Nasr Esfahany, M., Experimental Study of Turbulent Convective Heat Transfer and Pressure Drop of Dilute Cu/Water Nanofluid Inside a Circular Tube, *Int. Commun. Heat Mass, 37* (2010), 2, pp. 214-219

[12] Sundar, L. S., Sharma, K. V., Turbulent Heat Transfer and Friction Factor of AL_2O_3 Nanofluid in Circular Tube with Twisted Tape Inserts, *Int. J. Heat Mass Transfer, 53* (2010), 7-8, pp. 1409-1416
[13] Suresh, S., *et al*., Experimental Studies on Heat Transfer and Friction Factor Characteristics of CuO/Water Nanofluid under Turbulent Helically Dimpled Tube, *Exp. Therm. Fluid Sci., 35* (2011), 3, pp. 542-549
[14] Ternik, P., *et al.*, Numerical Study of Heat Transfer Enhancement of Homogeneous Water-Au Nanofluid under Natural Convection, *Mate. Technol., 46* (2012), 3, pp. 257-261
[15] Ternik, P., Rudolf, R., Heat Transfer Enhancement for Natural Convection Flow of Water-Based Nanofluids in a Square Enclosure, *Int. J. Simul. Model, 11* (2012), 1, pp. 29-39
[16] Sourtiji, E., Hosseinizadeh, S. F., Heat Transfer Augmentation of Magnetohydrodynamics Natural Convection in L-Shaped Cavities Utilizing Nanofluids, *Thermal Science, 16* (2012), 2, pp. 29-39
[17] Sajadi, A. R., Kazemi, M. H., Investigation of Turbulent Convective Heat Transfer and Pressure Drop of TiO_2/Water Nanofluid in Circular Tube, *Int. Commun. Heat Mass, 38* (2011), 10, pp. 1474-1478
[18] Ashtiani, D., *et al*., An Experimental Investigation on Heat Transfer Characteristics of Multi-Walled CNT-Heat Transfer Oil Nanofluid Flow Inside Flattened Tubes under Uniform Wall Temperature Condition, *Int. Commun. Heat Mass, 39* (2012), 9, pp. 1404-1409
[19] Fotukian, S. M., NasrEsfahany, M., Experimental Investigation of Turbulent Convective Heat Transfer of Dilute-Al_2O_3/Water Nanofluid Inside a Circular Tube, *Int. J. Heat Fluid Fl., 31* (2010), 4, pp. 606-612
[20] Heyhat, M. M., *et al*., Experimental Investigation of Laminar Convective Heat Transfer and Pressure Drop of Water-Based Al_2O_3 Nanofluids in Fully Developed Flow Regime, *Exp. Therm. Fluid Sci., 44* (2013), 1, pp. 483-489
[21] Heyhat, M. M., *et al*., Experimental Investigation of Turbulent Flow and Convective Heat Transfer Characteristics of Alumina Water Nanofluids in Fully Developed Flow Regime, *Int. Commun. Heat Mass, 39* (2012), 8, pp. 1272-1278
[22] HaghshenasFard, M., *et al*., Numerical and Experimental Investigation of Heat Transfer of ZnO/Water Nanofluid in the Concentric Tube and Plate Heat Exchangers, *Thermal Science, 15* (2011), 1, pp.185-196
[23] Gyoung-Ja Lee, *et al*., Thermal Conductivity Enhancement of ZnO Nanofluid Using a One-Step Physical Method, *Thermochim. Acta, 542* (2012), 20, pp. 24-27
[24] Pak, B. C., Cho, Y. I., Hydrodynamic and Heat Transfer Study of Dispersed Fluids with Submicron Metallic Oxide Particles, *J. Exp. Heat Transfer, 11* (1998), 2, pp. 151-170
[25] Williams, W., Buongiorno, J., Experimental Investigation of Turbulent Convective Heat Transfer and Pressure Loss of Alumina/Water and Zirconia/Water Nanoparticle Colloids (Nanofluids) in Horizontal Tubes, Transactions (ASME), *Journal of Heat Transfer, 130* (2008), ID 042412
[26] Duangthongsuk, W., Wongwises, S., An Experimental Study on the Heat Transfer Performance and Pressure Drop of TiO_2-Water Nanofluids Flowing under a Turbulent Flow Regime, *Int. J. Heat Mass Transfer, 53* (2010), 1-3, pp. 334-344
[27] Prasher, R., *et al*., Effect of Aggregation on Thermal Conduction in Colloidal Nanofluids, *Appl. Phys. Lett., 89* (2006), 14, 143119, DOI: 10.1063/1.2360229
[28] Eapen, J., *et al.*, The Classical Nature of Thermal Conduction in Nanofluids, *J. Heat. Transf., Trans ASME, 132* (2010), 10, 102402, DOI: 10.1115/1.4001304
[29] Timofeeva, E. V., *et al.*, Particle Shape Effects on Thermophysical Properties of Alumina Nanofluids, *J. Appl. Phys, 106* (2009), 014304, DOI: 10.1063/1.3155999
[30] Dittus, F. W., Boelter, L. M. K., *Heat Transfer in Automobile Radiators of the Tubular Type*, University of California Publications in Engineering, *2* (1930), 13, pp. 443-461
[31] Petukhov, B. S., Popov, V. M., Theoretical Calculation of Heat Exchange and Frictional Resistance in Turbulent Flow in Tubes of an Incompressible Fluid with Variable Physical Properties, *J. High Temp., 1* (1963), 1, pp. 69-83
[32] Kline, S. J., Mcclintock, F. A., Describing Uncertainties in Single-Sample Experiments, *J. Mech. Eng., 75* (1953), 1, pp. 3-8

Paper submitted: November 14, 2013
Paper revised: February 10, 2014
Paper accepted: January 18, 2014