CuO/water Nanofluid Convective Heat Transfer Through Square Duct Under Uniform Heat Flux

S. Zeinali Heris*, Taofik H. Nassan, S. H. Noie

Chemical Engineering Department, Engineering Faculty, Ferdowsi University of Mashhad, Mashhad, I. R. Iran

(*) Corresponding author: zeinali@ferdowsi.um.ac.ir
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Abstract:
Sometimes the need for non-circular ducts arises in many heat transfer applications because of lower pressure drop of non-circular cross section such as square duct compared to circular tube, particularly in compact heat. But square cross section has poor heat transfer performance and it is expected that using a nanofluid as a new heat transfer media may improve the heat transfer performance of this kind of duct. In this work, a nanofluid of CuO nanoparticles and distilled water has been prepared and its heat transfer characteristics have been studied through square cupric duct in laminar flow under uniform heat flux. Experiments revealed that a remarkable enhancement in heat transfer coefficient is achieved compared to the base fluid. Moreover, it has been reported that heat transfer coefficient enhances with increasing nanofluid flow rate as well as concentration of nanoparticles in the nanofluid especially at high flow rates. So, ultimate enhancement of 20.7% in Nu achieved at 1.5% volume concentration of CuO/water nanofluid. The basic reason for lower heat transfer rate of square ducts is existence of a static section for some part of fluid near corners of square duct and the results indicated that the presence of nanoparticles decrease this unmoved static section which consequently increase the heat transfer from the duct wall to the nanofluid.
Keywords: *Heat transfer enhancement; Square duct; Nanofluid; Uniform heat flux.*

1. INTRODUCTION

Low performance of conventional heat transfer fluids intensified the research for developing many techniques to increase heat transfer rate for cooling systems in a wide range of industry applications. The attention has been drawn to the heat transfer fluid as a key to increase heat transfer rate more than fifty years ago. Initial attempts to increase heat transfer properties of thermal fluids achieved by dispersing millimeter and micrometer-sized particles of better thermal conductivity materials since Maxwell presented a theoretical formula to predict thermal conductivity of suspensions [1]; however these coarse particles had major problems due to rapid settling, clogging small channels, high pressure drop, low heat transfer rates at low concentrations and erosion; therefore millimeter or micrometer-sized particles didn't find true applications in industry.

Fast advances in nanotechnology through the past two decades enabled to produce nanoparticles from different materials and the most amazing thing about these ultrafine particles is that their properties are different from the bulk materials.

Choi [2] at Argonne National Laboratory (USA) is considered the first who cast the term Nanofluid to describe nanoparticle fluid suspensions. He proposed that metallic nanoparticles can be suspended in traditional heat transfer fluids such as water, motor oil or ethylene glycol and this leads to increase the thermal conductivity of the new formed fluid (nanofluid) due to high thermal conductivity of the metallic nanoparticles. Since then, nanofluids are considered promised fluids for enhancing poor heat transfer properties of conventional fluids, so various kinds of nanomaterials have been tested such as metals, metallic oxides, nanotubes and so many others. For example, it is reported that a very small amount (less than1% in terms of volume fraction) of copper nanoparticles improved the measured thermal conductivity of the suspension by 40% [3-4], while over a 150% improvement of the effective thermal conductivity at a volume fraction of 1% was reported by Choi et al. [5] for multiwalled carbon nanotubes suspended in oil. Pak and Cho [6] measured the convective heat transfer coefficient with nanoparticles of γ-Al_2O_3 and TiO_2 dispersed in water. Their experimental results have revealed that heat transfer coefficients of the nanofluids increase with increasing the volume fraction of nanoparticles and the Reynolds number. Their heat transfer data showed Nusselt numbers up to 30% higher than predicted by the pure liquid correlation.

Numerical simulation has been used to investigate characteristics of heat transfer properties of nanofluids in laminar and turbulent flow regimes and results are available from many researchers [7-13] but to the best of my knowledge, there is only one numerical work published about nanofluid heat transfer in triangular duct [12]. Experimental investigations have also employed many types of nanoparticles with different diameters and a wide range of volume fractions in base fluid [14-21]. For example, Nassan et al. for the first time compared the heat transfer of Al_2O_3/water and CuO/water nanofluids through square duct with laminar flow and at constant wall heat flux. Their results showed that the convective heat transfer coefficient increases with nanofluids concentration and Peclet numbers [21].

Most of the results drawn from numerical simulations and experiments emphasize that the considerable augmentation in convective heat transfer coefficient can be achieved by suspending nanoparticles in different thermal fluids. However, the enhancement ratio depends on several factors such as the type of nanoparticle, base fluid, temperature and other factors which will be discussed later in this paper.

Several works has been reported on heat transfer in circular tubes for different applications and for all types of flow (laminar, transient and turbulent); however, few articles have discussed non-circular ducts like rectangular, triangular or other non-circular geometries.

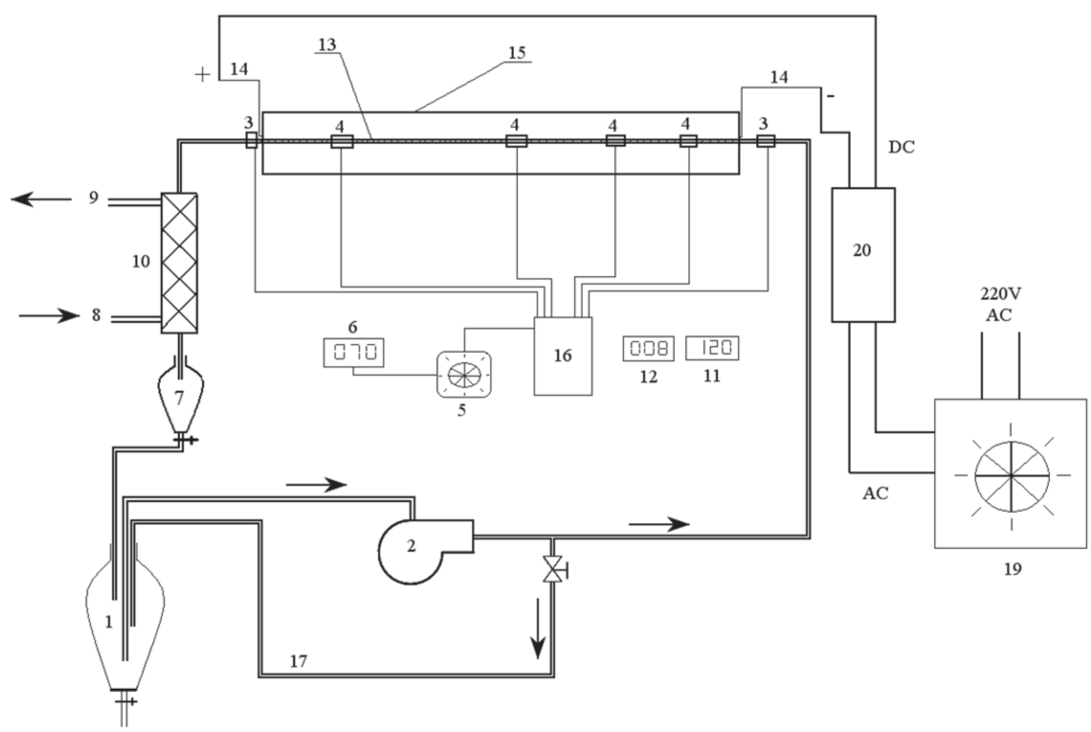
Friction between the fluid flowing through a conductor and its inner wall causes losses, which are quantified as pressure drop. Pressure drop in conductors is of important concern for the designer. Actually, the pressure drop in square cross section duct is less than that of circular tube (because of reducing contact with the wall) in laminar flow as the friction factor in square cross section duct is 56.92/Re and even less in triangular ducts, while in the circular tube the friction factor is 64/Re. Because of size and volume constraints in applications such as aerospace, nuclear, biomedical engineering and electronics, it may be required to use non-circular flow-passage geometries, particularly in compact heat exchangers [22]. Then it might be useful to benefit from the advantage of non-circular cross-section ducts in thermal engineering systems. However, heating exchange rates will decrease through these conduits (due to existence of a static section for some part of fluid near corners of square duct), but this could be compensated by using nanofluids in these systems, so this will enhance heat transfer rates. Nassan et al. were the first who compared the heat transfer of Al_2O_3/water and CuO/water nanofluids through square duct with laminar flow and at constant wall heat flux. Their results showed that convective heat transfer coefficient increased with nanofluids concentration and Peclet numbers [21].

Different criteria for selecting and optimizing the heat-exchanger passage geometries were outlined by Bergles [23]. Kays and London [24] showed that, a compact heat-exchanger with a triangular cross-sectional internal flow passage has a high ratio of heat-transfer area to flow-passage volume. Shah and London [25] studied heat transfer characteristics of laminar flow in a wide variety of conduit shapes, including for square, rectangular and triangular for an extensive range of thermal boundary conditions. Arsen’ev et al. [26] were reported an experimental investigation which considers the possibility to intensify heat transfer in a channel of a triangular cross section by installing 2 types of longitudinal turbulizing inserts. Kotcioglu et al. experimentally investigated the heat transfer, friction characteristics, and the second law analysis of the convective heat transfer for turbulent flow through a rectangular channel containing built-in wing-type vortex generator [27]. The heat transfer characteristics in a rectangular cross section duct using impingement jet technique for the purpose of heating and cooling are analyzed by Uysal et al. [28]. They investigated the effects of cross-flow on the overall flow characteristics in the housed channel and heat transfer distributions on both target surface and jet-issuing plate [28].

This paper aims to experimentally investigate forced convective heat transfer through square cross-sectional duct under laminar flow regime using CuO/water nanofluid and this research is a part of an integrated research project to study heat transfer characteristics through non-circular ducts and by utilizing many types of nanoparticles.

2. EXPERIMENTAL SET UP

The schematic of nanofluid heat transfer set up is shown in figure 1. This set up is built as a closed-loop system consisting of a reservoir tank (1 liter), a pump, a bypass line, a heat transfer test section, a water cooler and a flow meter. The heat transfer



1. Reservoir
2. Electric pump
3. Fluid inserted thermocouples
4. Surface soldered thermocouples
5. Thermo couple selector
6. Temperature indicator
7. Flow meter
8. Cooling water inlet
9. Cooling water outlet
10. Cooler
11. Volt meter
12. Amber meter
13. Square cross-section pipe 1000 mm
14. Heating element poles
15. Insulation Shell
16. Controller
17. Bypass
18. Drain
19. Controlled resistance
20. SCR unit

Figure 1: *Schematic diagram of the experimental set up*

section has a square cross-section area (1 cm^2) and was manufactured using copper paper (0.4mm thickness); the hydraulic diameter and the total length were 1cm and 100cm, respectively.
Two thermocouples (BT100-type) were inserted into the calming and mixing chamber at the inlet and the outlet of the test section, respectively for measuring the bulk temperature of the flowing fluid and another four thermocouples from the same type were inserted through the little soldered tubes on the surface of the test section at different points (distribution of thermocouples from the entrance side of flow is 4, 24, 54 and 96 cm, respectively) to check surface temperature variations during tests. The thermocouples have a precision of 0.1C° and were calibrated by the freezing and boiling points of distilled water before they are attached to the test section. All thermocouples were connected to a controller box and this in its turn is connected to a selector to indicate the thermocouple which is needed to be monitored on the temperature indicator.
To obtain a constant-heat flux boundary condition, the test section was heated by electric resistance which is fed by a constant DC power and the electric resistance works under 92.5 V and 8A to give a total heating power 740 W. The electric resistance is not directly attached to the surface of the test section; a very thin layer of uninflammable commercial material was used to avoid passing direct current to the surface of the test section and eventually to the whole system.
In order to minimize heat loss to the surroundings, the heat transfer section is insulated by a 10-cm-thick fiberglass blankets. The heat loss to the atmosphere was calculated and assumed negligible in the calculations due to its little amount. A water cooler after the test section was utilized to keep the temperature constant at the inlet of the test section.
The essential parameters measured during the test include electric power inputs, flow rate, bulk temperatures (inlet and outlet) and outer wall temperatures along the heat transfer section in 4 points. The flow rate through the loop was controlled with bypass line (see figure.1).
After filling the reservoir tank with nanofluid, the pump and the water cooler were switched on. Then electric resistance was turned on and the temperature of the test section started to increase. Through initial tests it has been found that the system needs 25-35 minutes to reach the steady state condition and after that the readings could be taken.

3. PREPARATION OF NANOFLUID

Distilled water was used for the suspending liquid medium. The mean diameters of CuO particles were 30-50nm (manufactured by Nanostructured & Amorphous Materials Inc., USA). Physical properties of nanoparticles were taken from the manufacturer data sheet (density ρs=6350 kg/m^3, heat capacity Cps= 535.6 J/kg. K, thermal conductivity Ks=69 W/m. K). In the present study no dispersant or stabilizer were used. This is because of the fact that the addition of any agent may change the fluid properties.
Nanofluids with different concentrations of CuO nanoparticles including 0.1%, 0.2 %, 0.5%, 0.8%, 1.0% and 1.5% volume fractions in distilled water were used to study heat transfer characteristics in laminar flow.
The volume fraction of the nanoparticles in suspension is defined as follows:

$$\Phi = \frac{V_s}{V_t} \tag{1}$$

While the density of the nanoparticles can be calculated from equation (2):

$$\rho_s = \frac{m_s}{V_s} \tag{2}$$

Then the required mass of nanoparticles for the required nanofluid suspension determined as follows:

$$m_s = \Phi.\rho_s.V_t \tag{3}$$

After preparing the required volume of powder using the equivalent weight of the solid, nanoparticles were mixed with the distilled water in a flask and then sonicated for 6-12hr by ultrasonic mixing system (model Parsonic 3600S). No sedimentation was seen for all volume factions, except for the volume fractions of 1.0% and 1.5% which a partial settling of nanoparticles was observed.

4. RESULTS ANALYSIS

The physical properties of the prepared nanofluids were calculated from water and nanoparticle characteristics at mean inlet and outlet bulk temperature using the following equations for density, viscosity, specific heat and thermal conductivity [29]:

$$\rho_{nf} = \phi.\rho_s + (1-\phi).\rho_w \tag{4}$$

$$\mu_{nf} = \mu_w.(1+2.5\phi) \tag{5}$$

Equation (5) which is known as Einstein equation [30] is applicable for dilute suspensions (ϕ<2%). This equation is also known as Stokes–Einstein formula for viscosity which can be a good approximation for nanofluids [31].

$$Cp_{nf} = \frac{\phi.(\rho_s.Cp_s) + (1-\phi).(\rho_w.Cp_w)}{\rho_{nf}} \tag{6}$$

Yu and Choi correlation [32] was used for determination of nanofluid effective thermal conductivity as follows:

$$k_{nf} = \left[\frac{k_s + 2k_w + 2(k_s - k_w)(1+\beta)^3\phi}{k_s + 2k_w - (k_s - k_w)(1+\beta)^3\phi}\right]k_w \tag{7}$$

In equation (7) β is the ratio of the nanolayer thickness to the original particle radius and β=0.1 was used to calculate the nanofluid effective thermal conductivity [32].

The applied current and volt through the whole experiments were 8 ampere and 92.5 volts respectively, so by measuring wall temperature at the cupric duct and the temperature of the fluid at the inlet and outlet, Peclet number, convective heat transfer coefficient, Nusselt number and Prandtl number of nanofluids under laminar flow are calculated using the following equations:

$$Pe_{nf} = Re_{nf}.Pr_{nf} \tag{8}$$

$$Pe_{nf} = \frac{\rho_{nf}.Cp_{nf}.\overline{U}.D_h}{k_{nf}} \tag{9}$$

$$\overline{h}_{nf}(\exp) = \frac{q}{A.(T_w - T_b)_M} \tag{10}$$

Where $(T_w\text{-}T_b)_M$ is the mean temperature difference and can be calculated from equation (11):

$$(T_w - T_b)_{nf} = \frac{(T_{w1} - T_{b1}) + (T_{w4} - T_{b2})}{2} \tag{11}$$

$$\overline{Nu_{nf}}(\exp) = \frac{\overline{h_{nf}}(\exp).D_h}{k_{nf}} \tag{12}$$

After computing $\overline{Nu_{nf}}(\exp)$, the values are compared with $\overline{Nu_{nf}}(theory)$ which can be calculated from Seider-Tate equation (13), this correlation is for laminar flow of single phase fluid [31,33]:

$$\overline{Nu_{nf}}(th) = 1.86\left(Re_{nf}\,Pr_{nf}\,\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_{nf}}{\mu_{wnf}}\right)^{0.14} \tag{13}$$

Where μ_{wnf} is calculated at mean wall temperature of the duct. In equations (8) and (13) Re_{nf} and Pr_{nf} are nanofluid Reynolds and Prandtl numbers respectively, which are defined as follows:

$$Re_{nf} = \frac{\rho_{nf}.\overline{U}.D_h}{\mu_{nf}} \tag{14}$$

$$Pr_{nf} = \frac{Cp_{nf}.\mu_{nf}}{k_{nf}} \tag{15}$$

Many independent variables (x_i) need to be measured in this test .The aim is to analyze how errors in the interval x_i *s* propagate into the calculation of any parameter such as R from measured values.

The uncertainty in *R*,(u_R), due to the combined effects of uncertainty intervals in all the x_i *s* (u_{xi})

$$u_R = \pm\left[\left(\frac{x_1}{R}\frac{\partial R}{\partial x_1}u_{x1}\right)^2 + \left(\frac{x_2}{R}\frac{\partial R}{\partial x_2}u_{x2}\right)^2 + \ldots + \left(\frac{x_n}{R}\frac{\partial R}{\partial x_n}u_{xn}\right)^2\right]^{1/2} \tag{16}$$

could be calculated by following equation [34,35]: The uncertainty calculated by the above procedure for heat transfer coefficient, Nusselt number, Peclet number and Reynolds number were ± 3%, ±3.5%, ±4.4% and ±5% respectively.

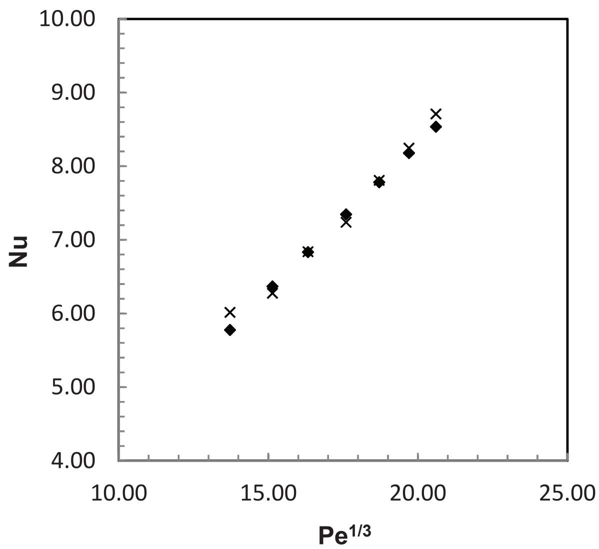


Figure 2: *Nusselt number versus $Pe^{1/3}$ for distilled water (♦Nu(th), × Nu(exp))*

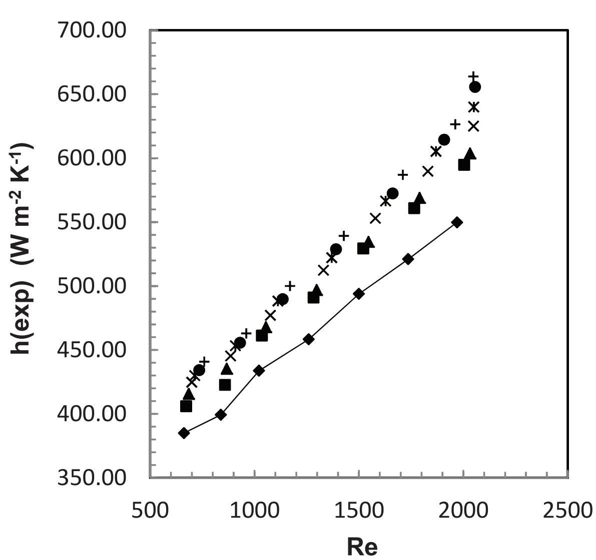


Figure 3: *Experimental convective heat transfer coefficient vs. Reynolds number for distilled water and different concentration of CuO/water nanofluid (-♦-Distilled water, ■0.1% CuO, ▲0.2% CuO, ×0.5%CuO, * 0.8%CuO, ●1.0%CuO, +1.5% CuO)*

5. RESULTS AND DISCUSSION

To provide a baseline for comparison and to check the reliability and accuracy of the nanofluids measurements, the tests were first performed on distilled water. Experimental results for water were compared with the prediction of Seider-Tate equation for laminar flow. Figure 2 illustrates this comparison, which indicates that a very good agreement was achieved and maximum discrepancy between experimental results and prediction of Seider-Tate equation is which confirm the reliability of the experiment procedure.

Heat transfer tests were performed on the base fluid and nanofluid at different volume fractions (0.1%, 0.2%, 0.5%, 0.8%, 1.0% and 1.5%) and every test has been repeated for two times at least, between Reynolds numbers 660 and 2050 and all the tests were performed under the same heat flux.

Figure 3 shows convective heat transfer coefficient as a function of Reynolds number and various volume concentrations. It is clear that the convective heat transfer coefficient increases with both volume concentrations and Reynolds number.

Moreover, Figure 3 illustrates that heat transfer characteristics of nanofluid is better than that of distilled water. The reasons are due to the extensive large surface area and the interactions among the nanoparticles themselves on one hand, and between nanoparticles and the inner surface of the duct on the other hand during flowing. Since heat transfer between nanoparticles and bulk fluid is performed on the surface of nanoparticles the higher specific surface area of these nanoparticles causes the better heat exchanges.

Figure 4 demonstrates the ratio of experimental Nusselt number to theoretical one which was calculated from Seider-Tate equation, versus Peclet number. It is noticed that the ratio increases with Reynolds number and with higher nanoparticles' concentration. For instance, for nanoparticles concentration of 0.1%, the ratio increases from 1.0197% to 1.091% and at 1.5% concentration the ratio increases from 1.0812% to 1.1864%.

It has been found that heat transfer enhancement is greater when Reynolds number increases; this may be due to better distribution, dispersion and

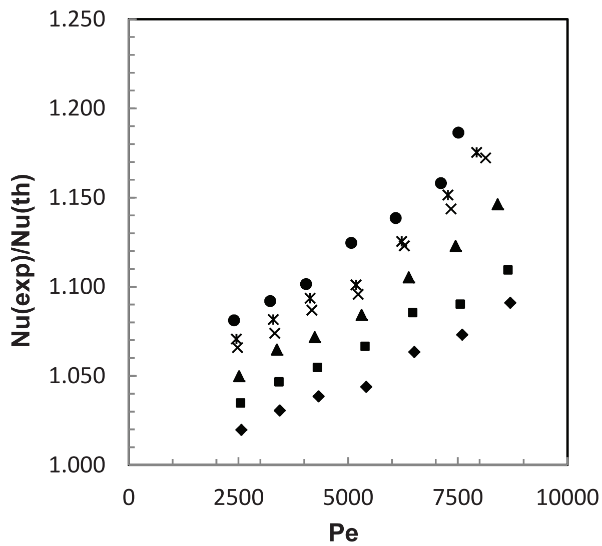


Figure 4: *The ratio of experimental Nusselt number of CuO/water nanofluid to Seider-Tate equation results vs. Peclet number at different concentrations of nanoparticles (♦0.1%CuO, ■0.2% CuO, ▲0.5%CuO, × 0.8%CuO, *1.0 %CuO, ●1.5%CuO)*

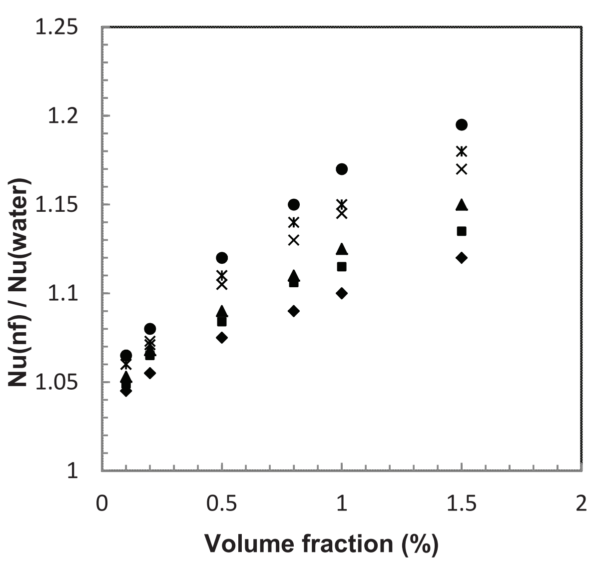


Figure 5: *CuO/ water nanofluid experimental Nusselt number ratio to distilled water vs. volume fraction of nanoparticles at different Peclet number (●Pe = 7500, *Pe=6500, ×Pe= 5500, ▲Pe= 4500, ■ Pe=3500, ♦Pe=2500)*

migration of nanoparticles through the flow. At higher flow rates of nanofluid at large Re the dispersion effects of the nanoparticles intensifies the mixing fluctuations and changes temperature profile to a flatter profile similar to semi-turbulent flow and causes an increase in heat transfer coefficient. The basic reason for lower heat transfer rate of square ducts is the existence of a static section for some part of fluid near corners of square duct and it seems that the presence of nanoparticles causes to decrement this unmoved static section. Increasing Re with nanofluid flow rate increment caused the better nanoparticles random movement and migration especially near duct corner which tends to enhance the heat transfer. Tests at turbulent flow should also be achieved in the future for better understanding of heat transfer properties through non-circular ducts.

Figure 5 shows the ratio of Nusselt number of the nanofluid to that of water as a function of the volume fraction of nanoparticles at constant Peclet numbers from 2500 to 7500. It is obvious that the maximum enhancement is achieved at the maximum concentration of 1.5% where the enhancement increases from 1.117 to 1.195, while at the minimum concentration the enhancement increases from 1.0439 to 1.071. Moreover, the enhancement of convective heat transfer coefficient of water-based CuO nanofluids is much higher than that of effective thermal conductivity at the same volume fraction of 1.5 % vol, predicted by equation (7).

By comparing the results obtained by Zeinali Heris et al. [36] through circular tube with the same nanofluid CuO/water in laminar flow at the same concentrations (0.2%, 0.5%, 1.0% and 1.5%) it has been found that the enhancement achieved by Zeinali Heris et al. [36] is 27% over the base fluid while our experiments give maximum enhancement of 20.7%.

Asirvatham et al. [37] experimentally investigated the heat transfer properties of CuO/de-ionized water nanofluid through copper tube under laminar flow and the results have shown 8% enhancement of the convective heat transfer coefficient of the nanofluid at 0.003% volume concentration of CuO nanoparticles, they also reported that the heat transfer enhancement was considerably increased as the Reynolds number increased. This is quite interesting results and if it is compared with the results obtained from this paper, it can be seen that the results achieved at 0.1% volume fraction are almost the same as the augmentation at this concentration, which is 8.2%.

Recently, Fotukian and Esfahany [38] reported a work on CuO/water nanofluid in circular tube under turbulent regime and the results indicate that at a low concentration of cupric oxide between 0.015% to 0.236% volume fractions, the convective heat transfer coefficient was enhanced by 25% and the enhancement ratio did not show any significant variation with concentration of CuO nanoparticles, so similar test could be achieved through square duct in the future to compare results and choose the optimum concentration for CuO nanoparticles in water.

Equation (7) was considered the interface of solid nanoparticles and the base fluid for heat exchange between the solid and liquid as a parameter for calculating nanofluid thermal conductivity under the static condition. Therefore, other factors such as dispersion of nanoparticles, Brownian motion and particle migration, especially near the duct wall, must be considered in the interpretation of heat transfer performance of nanofluids.

The diffusion and collision intensification of nanoparticles in nanofluid near duct wall due to increase in concentration of nanoparticles leads to rapid heat transfer from wall to nanofluid. At high flow rates the migration and dispersion effects of the nanoparticles intensifies the mixing fluctuations and changes temperature profile to a flatter profile similar to turbulent flow and causes increase in heat transfer coefficient. Because of random and chaotic motion, collision and migration of nanoparticles inside nanofluid suspension, the local turbulence and transient fluctuation with the transverse temperature gradient in the bulk of the fluid is produced (this mechanism is known as dispersion effect) and caused to enhanced heat transfer. In low flow rates clustering and agglomeration of nanoparticles may exist in nanofluid flow [39] and therefore at low Reynolds number, less heat transfer enhancement could be observed.

One of the possible reasons for enhanced nanofluid heat transfer coefficient could be nanoparticles migration due to shear action, Brownian motion, and viscosity gradient in the cross section of the square duct. Heat transfer between nanoparticles and fluid is performed on the surface of nanoparticle, thus, these nanoscale particles increase the heat transfer. Movements, interaction between particles and heat transfer surfaces, increases by adding the nanoparticles. Brownian movement of nanoparticles increases local turbulence of fluid flow. Dispersion of nanoparticles decreases thermal boundary layer thickness, because of temperature gradient change. By decreasing the thermal boundary layer thickness, length of development increases. Convective heat transfer coefficient is proportional to the thermal conductivity coefficient and it is inversely proportional to boundary layer thickness. Thus heat transfer coefficient increases by adding nanoparticles to the base fluid.

Generally, only very few correlations are available to exactly predict the heat transfer performance of nanofluids, and correlations which include the effect of volume fraction, particle shape and particle size are not suffice. Therefore, further research on convective heat transfer of nanofluids, and more theoretical and experimental research works are needed in order to clearly understand and accurately predict their hydrodynamic and thermal characteristics especially in non-circular ducts.

6. CONCLUSIONS

Experimental investigation on laminar convective heat transfer of CuO/water nanofluid was performed through square cross section cupric duct with constant uniform heat flux boundary conditions. Experiments' results have clearly revealed that the addition of nanoparticles has remarkably increased the heat transfer compared to the distilled water (base fluid). Such heat transfer enhancement appears to be more related to the increase of nanoparticles' volume concentration. Maximum enhancement in the convective heat transfer coefficient is 20.7 % over the base fluid. Moreover, it has been proved that Seider-Tate correlation underestimates heat transfer properties of CuO/water nanofluid. The

basic reason for lower heat transfer rate of square ducts is existence of a static section for some part of fluid near corners of triangular duct and it seems that the presence of nanoparticles causes to decrement this unmoved static section.
It is discovered that using nanofluids as a thermal fluid medium can be used as a solution to overcome poor heat transfer performance of non-circular ducts. Consequently, nanofluid flow through square conduits has benefits of both low pressure drop and high heat transfer rate. But further theoretical and experimental investigations are needed to understand heat transfer characteristics of nanofluids in non-circular ducts like triangular ducts, rectangular ducts with different aspect ratios and other possible non-circular ducts with different nanofluids.

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REFERENCES

1. J. C. Maxwell, A Treatise on Electricity and Magnetism, second ed., Oxford University Press, Cambridge, UK, 1904.
2. S.U.S. Choi, Enhancing thermal conductivity of fluid with nanoparticles developments and Applications of Non-Newtonian Flows, FED-V. 231/ MD-V, vol. 66, ASME, New York, 1995.
3. S. Lee, S.U.S. Choi, S. Li, J. A. Eastman, Measuring thermal conductivity of fluids containing oxide nanoparticles, ASME J. Heat Transfer 121 (1999) 280.
4. J.A. Eastman, S.U.S. Choi, S. Li, W. Yu, L. J. Thompson, Anomalously Increased Effective Thermal Conductivities of Ethylene Glycol-Based Nanofluids Containing Copper Nanoparticles, Appl. Phys. Lett. 78 (2001) 718.
5. S.U.S. Choi, Z.G. Zhang, W. Yu, F.E. Lockwood , E.A. Grulke, Anomalous thermal conductivity enhancement in nanotube suspensions, Appl. Phys. Lett. 79 (2001) 2252.
6. B.C. Pak, Y.I. Cho, Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles, Exp. Heat Transfer 11 (1998) 15.
7. H.F. Oztop, E. Abu-Nada, Numerical study of natural convection in partially heated rectangular enclosures filled with nanofluids, Int. J. Heat Fluid Flow, 29 (2008)1326.
8. P.K. Namburu, D.K.Das, K.M. Tanguturi, R.S. Vajjha (2009). Numerical Study of Turbulent Flow and Heat Transfer Characteristics of Nanofluids Considering Variable Properties, Int. J. Thermal Sciences 48 (2009) 290.
9. R.Y. Jou, S.C. Tzeng, Numerical research of nature convective heat transfer enhancement filled with nanofluids in rectangular enclosures, Int. Comm. Heat Mass Transfer 33(2006)727.
10. S. Mirmasoumi, A. Behzadmehr, Numerical study of laminar mixed convection of a nanofluid in a horizontal tube using two-phase mixture model, Appl. Therm. Eng. 28 (2008)717.
11. S. Zeinali Heris, M. Nasr Esfahany, S. Gh. Etemad, Numerical investigation of nanofluid laminar convection heat transfer through a circular tube, Num. Heat Transfer, A 52(11)(2007)1043.
12. S. Zeinali Heris, S.H. Noie, E. Talaii, J. Sargolzaei, Numerical investigation of Al2O3/water nanofluid laminar convective heat transfer through triangular ducts, Nanoscale Research Letters, 6 (2011) 179.
13. R. Lotfi, Y., Saboohi, A.M., Rashidi, Numerical study of forced convective heat transfer of nanofluids: Comparison of different approaches, Int. Commun. Heat Mass Transfer (2009), doi:10.1016/j.icheatmasstransfer. 2009.07.013
14. W. Duangthongsuk, S. Wongwises, Effect of thermo-physical properties models on the prediction of the convective heat transfer coefficient for low concentration nanofluid. Int. Comm. Heat Mass Transfer, 35 (2008)1320.
15. S. Zeinali Heris, S.Gh. Etemad , M. Nasr Esfahany, Experimental investigation of oxide nanofluids laminar flow convection heat transfer, Int. Commun. Heat Mass Transfer 33 (2006) 529.
16. S. Zeinali Heris, Experimental investigation of pool boiling characteristics of low-concentrated CuO/ ethylene glycol–water nanofluids, Int. Commun. Heat Mass Transfer 38 (2011) 203.

17. S. Zeinali Heris, S.Gh. Etemad, M. Nasr Esfahany, Convective heat transfer of a Cu/water nanofluid flowing through a circular tube, Exp. Heat Transfer 22 (2009) 1470.

18. K.V. Sharma , L. Syam Sundar, P.K. Sarma, Estimation of heat transfer coefficient and friction factor in the transition flow with low volume concentration of Al2O3 nanofluid flowing in a circular tube and with twisted tape insert, Int. Commun. Heat Mass Transfer 36 (2009) 503.

19. H. Salehi, S. Zeinali Heris, M. Koolivand Salooki, S.H. Noei, Designing a neural network for closed thermosyphon with nanofluid using genetic algorithm, Brazilian J. Chem. Eng., 28(1) (2011)157.

20. D. Zhu, Sh. Wu, N. Wang, (2010) Thermal Physics and Critical Heat Flux Characteristics of Al2O3–H2O Nanofluids, Heat Transfer Engineering, 31(14) (2010) 1213.

21. Taofik H. Nassan, S. Zeinali Heris, S.H. Noie, A comparison of experimental heat transfer characteristics for Al2O3/water and CuO/water nanofluids in square cross-section duct, , Int. Commun. Heat Mass Transfer 37 (2010) 924.

22. A.Z. Sahin, Irreversibility in various duct geometries with constant wall heat flux and laminar flow, Energy 23 (1998) 465.

23. A. E. Bergles, Heat transfer enhancement-the encouragement and accommodation of high heat fluxes, J. Heat Transfer 119 (1997) 8.

24. W.M. Kays, A.L. London, Compact Heat Exchangers, third ed., McGraw-Hill, New York, 1984.

25. R.K. Shah, A.L. London, Laminar flow forced convection in ducts, first ed., Academic press Inc., New York, 1978.

26. L.A. Arsen'ev, N.P. Sokolov, V.G. Polishchuk , Intensification of Heat Transfer in a Triangular Channel with Turbulizing Inserts, Heat Transfer Research , 31(1-2) (2000) 68.

27. I. Kotcioglu, S. Caliskan, T. Ayhan, S. Baskaya, An Experimental Investigation of Heat Transfer Enhancement in a Rectangular Duct with Plate Fins, Heat Transfer Research, 40 (3) (2009) 263.

28. U. Uysal , F. Sahin, M. Chyu, Experimental and Theoretical Analysis of Heat Transfer Characteristics in a Rectangular Duct with Jet Impingement, Heat Transfer Research , 41 (8) (2011) 901.

29. D.A. Drew, S.L. Passman, Theory of Multi Component Fluids, first ed., Springer, Berlin, 1999.

30. A. Einstein, Investigation on Theory of Brownian motion, first ed., Dover, New York, 1956.

31. Sarit K. Das, S. U. S. Choi, Wenhua Yu, T. Pradeep, Nanofluids: Science and Technology, first ed., John Wiley & Sons Inc., New Jersey, 2008.

32. W. Yu, S.U.S. Choi, The role of interfacial layers in the enhanced thermal conductivity of nanofluids: A renovated Maxwell model, J. Nanopart. Res. 5 (2003) 167.

33. E.N. Seider, G.E. Tate, Heat transfer and pressure drop of liquid in tubes, Ind. Eng. Chem. 28 (1936) 1429.

34. H.D. Young, Statistical Treatment of Experimental Data, first ed., McGraw Hill, New York, 1962.

35. J. P. Holman, Experimental Methods for Engineers, fifth ed., McGraw-Hill, New York, 1989.

36. S. Zeinali Heris, M. Nasr Esfahany,S.Gh. Etemad, Investigation of CuO/water nanofluid Laminar convective heat transfer through a circular tube, J. Enhanced Heat Transfer 13 (2006)279.

37. L.G. Asirvatham, N. Vishal, S.K. Gangatharan, D.M. Lal, Experimental study on forced convective heat transfer with low volume fraction of CuO/water nanofluid, Energies 2 (2009) 97.

38. S. M. Fotukian, M. Nasr Esfahany, Experimental study of turbulent convective heat transfer and pressure drop of dilute CuO/water nanofluid inside a circular tube, Int. Commun. Heat Mass Transf. (2009),doi:10.1016/j.icheatmasstransfer.2009.10.003

39. Y. Yang, Z.G., Zhang, E.A. Grulke, W.B. Anderson, and G., Wu, Heat transfer properties of nanoparticle-in-fluid dispersions (nanofluids) in laminar flow, Int. J. Heat Mass Transfer 48 (2005) 1107.