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RESEARCH NOTE

Experimental Investigation of Mixed Convection Heat Transfer in Vertical Tubes by Nanofluid: Effects of Reynolds Number and Fluid Temperature

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A B S T R A C T

An experimental investigation was carried out to study mixed convection heat transfer from Al_2O_3-water nanofluid inside a vertical, W-shaped, copper-tube with uniform wall temperature. The tests covered different ranges of some involved parameters including Reynolds number, temperature and particles volume fraction. The results showed that the rate of heat transfer coefficient improved with Reynolds number for average wall temperatures of 50 and 60ºC. Additionally, the heat transfer coefficient increased slightly with an increase of the Reynolds number. Interestingly, the pressure drop of nanofluid was very close to that of base fluid. Besides, a new correlation was proposed to calculate the Nusselt number in W-shaped tubes.

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NOMENCLATURE

A	Area , m^2
Cp	Specific heat of fluid, J kg^{-1} K^{-1}
D	Tube inside diameter, m
Gr	Grashof number
h	Heat transfer coefficient, W m^{-2} K^{-1}
k	Thermal conductivity, $Wm^{-1}K^{-1}$
L	Length of test tube, m
n	Shape factor
Nu	Nusselt number
NuF	forced convective Nusselt number
Pr	Prandtl number
Re	Reynolds number $\left(\mathrm{Re} = r\bar{U}D/m\right)$
Ri	Richardson number, Ri= Gr/Re^2
T	Temperature

Greek Symbols

μ	Fluid dynamic viscosity, kg m^{-1} s^{-1}
ν	Kinematic viscosity, m^2 s^{-1}
ñ	Density, kg m^{-3}
φ	Particle volume concentration (%)
$\bar{U}$	Average inlet axial velocity, m s^{-1}

Subscripts

b	Fluid bulk condition(water)
in	Inlet
nf	Nanofluid
out	Outlet
p	Particle
W	Wall

1. INTRODUCTION

Thermal loads are increasing in a wide variety of applications like microelectronics, heat exchanger and transportation device. Conventional fluids, such as water, engine oil, and ethylene glycol have inherently limited heat transfer capability. So, many efforts for dispersing small particles with high thermal conductivity in the liquid coolant have been conducted to enhance thermal properties of conventional heat transfer fluids. A new class of fluid consisting of uniformly dispersed and suspended nanometer sized

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particles or fibers in fluid for improving both thermal conductivity and suspension stability was by chol [1] in 1995. Increasing the thermal conductivity is a key to improve the heat transfer characteristics of conventional fluids. Since a solid metal has a larger thermal conductivity compared to base fluid, suspending metallic or metallic oxide solid nanoparticles into the base fluid is expected to improve the thermal conductivity of that fluid. Therefore, heat transfer can be improved using solid particles as an additive suspended into the base fluid. However, the enhancement of heat transfer rates of nanofluids reported in the literature is beyond the effect of increased thermal conductivity alone [2-6]. Nevertheless, thermal conductivity of nanofluids can be affected by different parameters such as volume fraction, type, size and shape of particles, type of base fluid, temperature, acidity, as well as additives. Besides changes in other thermo-physical properties of nanofluids such as viscosity, density, specific heat and agglomeration of nanoparticles can improve the heat transfer behavior of nanofluids. Therefore, some researchers examined the performance of convective heat transfer of nanofluids in different heat exchangers [7-13]. Wen and Ding [4] investigated the heat transfer performance of water-alumina mixture under the laminar flow regime in a copper tube. They found that the convective heat transfer coefficient increases with increasing Reynolds number and particles concentration. Similar to Wen and Ding, other studies that focused on forced convection heat transfer inside tubes, also observed an increase in the heat transfer coefficient [3, 5, 14, 15]. Convective heat transfer coefficient of Al_2O_3 - water and CuO - water nanofluids for laminar flow in the annular tube under a constant wall temperature boundary condition is presented by Heris et al. [15]. Their results showed that the heat transfer coefficient increases with Peclet number and particle volume concentrations, while Al_2O_3–water nanofluid showed larger heat transfer enhancement than CuO-water nanofluid. The conjugate problem of laminar mixed convection of Al_2O_3-water nanofluid in a uniformly heated inclined tube was studied by Mansour et al. [16] numerically. They found that the presence of nanoparticles intensifies the buoyancy-induced secondary flow, especially in the developing region. In recent years, The heat transfer enhancement for industrial application by using nanofluid has become increasingly important. For example, Hussein et al. [17] and Ebrahimi et al. [18] examined the performance of forced convection heat transfer in the car radiator with SiO_2- water nanofluid. They found that this nanofluid could enhance heat transfer rate compared to pure water. Laminar flow of CuO based oil nanofluid in a tube with wire coiled inserts have been estimated by Saeedinia et al. [19] experimentally. They found 45% enhancement heat transfer for 0.3% volume concentration. Similar results have been achieved by Chandrasekar et al. [20]. A number of researchers studied the performance of nanofluids within the heat pipe [7-9]. They reported significant reduction in thermal resistance of heat pipe with nanofluid compared to distilled water. However, most of the results show that nanofluids are useful for improving thermal performance [21]. The main reason of this enhancement is because of the thermal dispersion in which velocity slip induces a velocity and temperature perturbation [3, 5], particle migration and a non-uniform shear rate [10], Brownian diffusion and thermophoresis [1, 12], and nanoparticle random motion [2, 3, 11]. Experimental results show that the heat transfer behavior of nanofluids is not completely understood.

Going through the existing pertinent literature, limited papers investigate heat transfer characteristics of nanofluid flow inside vertical tubes, especially W-shaped tubes, under free and forced convection flow. In addition, it is essential to measure the heat transfer performance of nanofluid under the given flow condition and specific applications directly in order to provide helpful guidelines to the compact heat exchangers.

Therefore, in the present work, our goal is to investigate experimentally, flow and convective heat transfer characteristics of water-based Al_2O_3 nanofluid flowing through a W-shaped tube with the constant wall temperature under free and forced convection flow.

At the same time, the effects of Reynolds number and the wall temperature on the mixed convection of this nanofluid are investigated and the results are presented. In addition, pressure drop of working fluid are measured experimentally.

2. EXPERIMENT

2. 1. Experimental Set Up and Procedure Figure 1 shows a schematic illustration of the present experimental system that was used. The fluid, distilled water or a nanofluid, flows from a flow control valve and enters the heated section. The latter part is a w-shaped copper tube with an internal diameter of D = 6 mm, a total length of 940mm and 0.9mm thickness. The test section is placed in a water bath. The wall of the tube has been constant and uniform temperature. The water in the bath is heated using one 1500 Watt electrical element and one thermo control, which was employed to preserve the constant temperature of the bath. Furthermore, a mixer is used to keep a uniform fluid temperature in the bath. In order to minimize the heat loss to the ambient, a thick layer of fiber-glass insulating blanket is wrapped around the water bath system. A tube-in-shell type heat exchanger was used to cool the heat transfer fluid to help attaining steady state, where cooling water was used as the coolant. Flow rates are measured by a rotameter (which is calibrated using a classical and reliable 'stop-watch-and-weighting' technique, with an accuracy of ±0.1 gr). A centrifugal pump is used for pumping the collected fluid in the reservoir to the test section. In order to control the flow rate, a reflux line with a valve is used.

Temperature is measured at various places in the system (Figure 1). Ten thermocouples with an accuracy of ±0.1℃ are installed as follows:

Three PT100 thermocouples are placed at different positions of the bath to ensure constant and uniform temperature in all locations. Two PT100 thermocouples are used to measure fluid temperatures at the tube inlet and outlet. To ensure an isothermal condition at the outer surface of the test tubes, five PT100 thermocouples are used at different locations.

In addition, 2 pressure sensors are placed at inlet and outlet of the tube for measuring the pressure loss along the test section. These sensors have an accuracy of ±1 mbar. All the thermocouples and sensors are connected to a data-acquisition system and the collected data is saved on a computer.

During the experimental run, bath temperature was initially set to a specified value (starting from 40 ℃ and then a 10-degree increase). The volume flow rate was then controlled to have the needed Reynolds number, including 1.93, 3.996, 7.024, 9.970, 13.387cc/s. Measurements were carried out after 20-30 min when the system reached steady state condition. Each measurement was repeated at least three times.

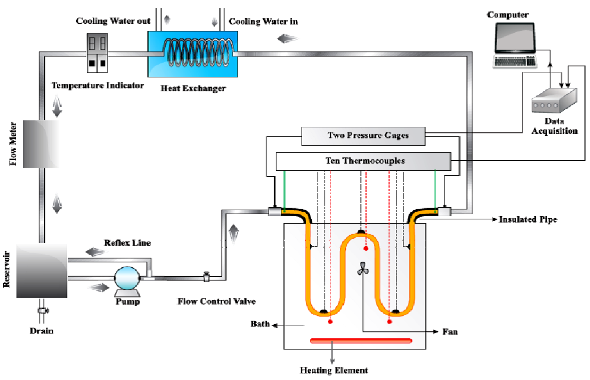

Figure 1. Schematic representation of the experimental system

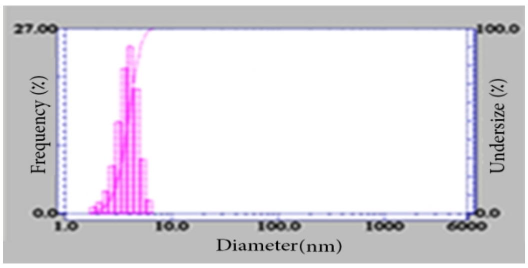

Figure 2. Particle size distribution

3. PREPARATION OF NANOFLUIDS

Nanofluids of aluminum oxide nanoparticles with different concentrations were synthesized by chemical vapor deposition (CVD) instrument. Briefly, the production line is a quartz tubing (0.5 cm diameter) with a total length of about 3 meters. The quartz line passed through two furnaces, an ultraviolet (UV) irradiator, and a microwave oven. Several vibrators were attached to the production line. Two tubing furnaces, with 40cm length and power of 2000 W, were used for preheating and production of aluminum nanoparticles. A 450 W UV irradiator and a 1500 W Samsung appliance type microwave oven were used to prevent coagulation of aluminum oxide nanoparticles.

Before startup, a casting agent such as several ml of aqueous solution of CTAB was poured inside the cooled vessels which were located at the end part of the instrument. Argon gas was bubbled through a solution of aluminum acetate and then introduced into the production line. A piezoelectric vibrator located at the bottom of the aluminum acetate solution was applied to promote formation of aluminum acetate aerosols. The temperature of the aluminum solution was controlled using a mantle which is located around the reagent cell. Trace flow of oxygen in argon was run through the production line for aluminum oxide nanoparticles. The UV synthesized was finally flowed into the cooled vessel containing CTAB solution. The solenoid valves automatically controlled the flow of all gases. Gas flow furnaces, solenoid valves, UV irradiator and all parameters of the microwave oven were controlled with computer software written in Visual Basic. Figure 2 shows particle size distribution for nanoparticles which is specified by light scattering particle size analyzer. Particles sizes are between 3 to 8 nanometers, with an average size of 4.5nm (these values are measured before the test).

4. NANOFLUID PROPERTIES AND DATA REDUCTION

To investigate the heat transfer performance of flowing nanofluids, the knowledge of the transport properties of nanofluids is required. Based on classical formulas that were presented for a two-phase mixture, the thermal and physical properties of the nanofluid under consideration are computed by Equations (1) to (4). All fluid properties are calculated by using the mean fluid temperature between the inlet and the outlet. Nanofluid density:

$$\rho_{nf} = (1-\varphi)\rho_b + \varphi\rho_p \tag{1}$$

where, φ is the particle volume fraction and ρ_{nf} , ρ_p and ρ_b are the nanofluid, particle material and base fluid densities, respectively.

Nanofluid specific heat and dynamic viscosity can :

$$c_{p_{nf}} = (1-\varphi)c_{p_b} + \varphi . c_{p_p} \quad (2)$$

$$\frac{\mu_{nf}}{\mu_b} = \frac{1}{(1-\varphi)^{2.5}} \quad (3)$$

where, $C_{p_{nf}}$, μ_{nf} are the heat capacity and dynamic viscosity of nanofluid [2, 22]. Nanofluid thermal conductivity: Effective thermal conductivity values can be evaluated by the use of Hamilton and Crosser formula [23], which is given by the following equation:

$$\frac{k_{eff}}{k_b} = \frac{k_p + (n-1)k_b - (n-1)\varphi\,(k_p - k_b)}{k_p + (n-1)k_b + (k_p - k_b)\varphi} \quad (4)$$

In Equation (4), k_b and k_p are thermal conductivity of the base fluid and particles, respectively. n is the empirical shape factor given by $n = 3/\psi$ and Ψ is the sphericity (Ψ = 1 for spherical particles).

In this work, the relevant thermo-physical properties of the solid nanoparticles (Al_2O_3) are k=36 W/m K, C_P=773 J/kg K and ρ=3880 kg/m^3. The convective heat transfer coefficient and the corresponding Nusselt number are given as follows:

$$\bar{h}_{\exp} = \frac{c_p\, \rho A (T_{out} - T_{in})\bar{U}}{\pi DL\,(T_W - T_b)_{L.M.}} \quad (5)$$

$$N\bar{u}_{\exp} = \bar{h}_{\exp} . D/k \quad (6)$$

where, ρ is the fluid density and A, $\bar{U}$ and D are the cross sectional area, average fluid velocity and the tube diameter, respectively, T_{out} and T_{in} are outlet and inlet fluid temperature, T_w and T_b are the wall and fluid temperatures and L is the length of pipe. $(T_W-T_b)_{L.M.}$ is logarithmic mean temperature difference.

5. VALIDATION TESTS OF THE EXPERIMENTAL DATA

In order to establish the reliability and accuracy of the present experimental set up, the pressure drop and the heat transfer coefficient are experimentally measured using distilled water as a working fluid. The Reynolds number varies from 350 to 3800. Because L/D>>1, it is reasonable to neglect the entrance effect, so the flow inside the tube is considered fully developed. The total pressure loss includes major loss is because of friction and minor loss. It is also because of changing velocity in bends. The total head loss is,

$$H_f = (K_f + K_m)\frac{V^2}{2g} \quad (7)$$

The energy loss factor (K_m) is due to the curvature of the tube which depends on radius and angle of curvature. K_f is the friction loss coefficient which is related to friction coefficient of length L and diameter (D) of the pipe and it can be achieved as follows [24]:

$$k_f = f\,\frac{L}{D} \quad (8)$$

f is the friction factor, the friction coefficient depends on the type of flow (laminar, transient, turbulent) and the roughness of the tube. For fully developed laminar flow, the friction coefficient depends only on the flow Reynolds number; the following Hagen-poiseuille equation is used:

$$f = 64/\mathrm{Re} \quad (9)$$

For turbulent flow, the friction coefficient depends on the flow Reynolds number and the roughness of the pipe. This coefficient can be expressed as [24]:

$$f = 0.0055\left\{1 + \left[20000\left(\frac{e}{D}\right) + \frac{10^6}{\mathrm{Re}}\right]^{1/3}\right\} \quad (10)$$

where, e is the tube roughness. Figure 3 shows the head loss of distilled water based on the laboratory data of this study versus the head loss resulting from theoretical relations (Equations (7-10)) which illustrates an adequate agreement between experimental results and the empirical formula. By calculating Richardson number, it was found that the flow inside the tube is a mixed of free and forced convection. Figure 4 shows the relation between Reynolds number and the products of Grashof number with Prandtl number (Rayleigh number) times diameter divided by length. In this figure, the present results are compared with those of Metais and Eckert [25] for the vertical tube. Demonstrated in Figure 4, the experimental data is mixed convection flow at low Reynolds number and force convection at high Reynolds number (average linear velocity between 0.06827and 0.473389m/s).

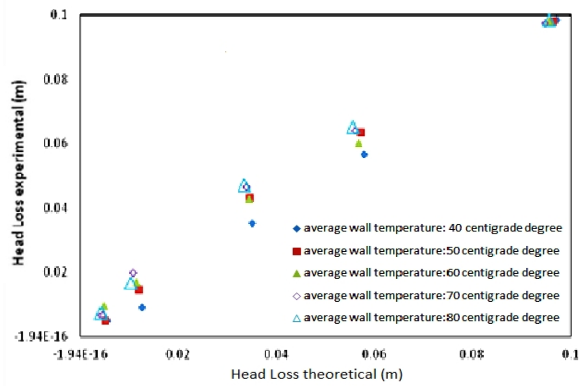

Figure 3. Parity plot comparing the theoretical and experimental results.

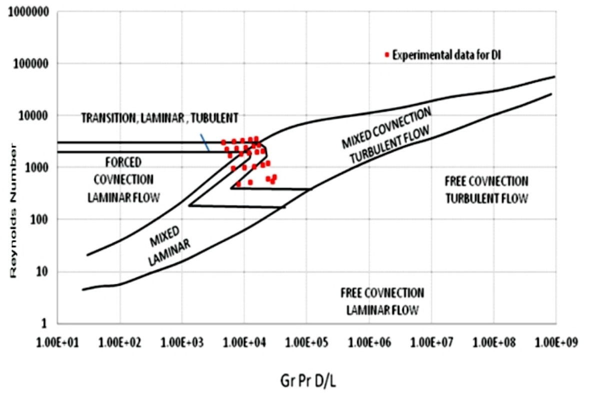

Figure 4. Regime of free forced and mixed convection for flow through vertical tubes and experimental data in this work [25]

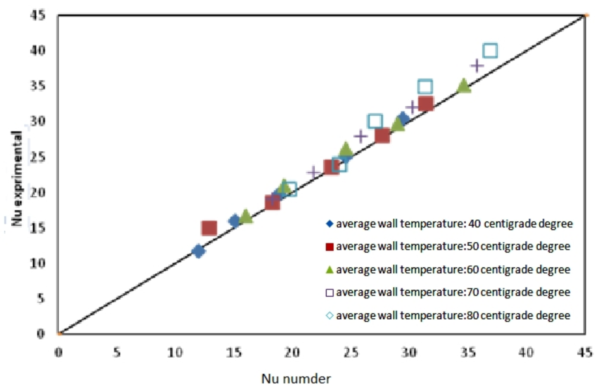

Figure 5. Plot of experimental Nusselt number versus theoretical Nusselt number

For downward flow of water in pipe with mixed convection, the experimental results of the convective heat transfer coefficient computed by Equations (5) and (6) which are compared with those calculated by Jackson and Fewster equation [25] based on the Nusselt number:

$$\frac{Nu}{Nu_F}=\left[1+4500\,{Gr_b}\Big/{Re_b^{2.625}Pr_b^{0.5}}\right]^{0.31}$$
$$10^{-5}\le {Gr_b}\Big/{Re_b^{2.625}Pr_b^{0.5}}\le 0.1 \qquad (11)$$

Another comparison is performed for upward flow in water tube by Hall and Jackson equation [25];

$$\frac{Nu}{Nu_F}=\left[1\pm 10^{4}\,{Gr_b}\Big/{Re_b^{2.7}Pr_b^{0.5}}\right]^{0.46}$$
$$0.1\le 10000\,{Gr_b}\Big/{Re_b^{2.7}Pr_b^{0.5}}\le 100 \qquad (12)$$

where, Nu_F denotes Nusselt number for forced convection with the following Sieder-Tate equations [26] for the laminar and turbulent flows under the constant wall temperature boundary conditions:
For laminar flows:

$$N\bar{u}_F(th)=1.86\left(Re_f \cdot Pr_f \cdot \frac{D}{L}\right)^{1/3}\left(\frac{m_f}{m_{wf}}\right)^{0.14} \qquad (13)$$

For turbulent flows:

$$N\bar{u}_F(th)=0.027\ Re_f^{0.8}\ Pr_f^{1/3}\left(\frac{m_f}{m_{wf}}\right)^{0.14} \qquad (14)$$

The measured data of the Nusselt number versus theoretical results are presented in Figure 5. There is a good agreement between the present experimental results and the theoretical values computed based on Equations (11)-(14).

6. RESULTS AND DISCUSSION

The present experiments were carried out using Al_2O_3 -water mixtures (average diameter of particles is 4.5 and 24nm). Tests were performed for flow Reynolds number between 350 to 4000, the particle volume fraction φ from 0% to 4%, and average wall temperature from 40 to 80℃. The Effects of flow Reynolds number and average wall temperature are investigated and presented in the subsequent sections. Note that, heat transfer behavior of the nanofluid is presented in the fully developed region.

6. 1. Effect of Reynolds Number in Convective Heat Transfer

Inlet and outlet fluid temperatures, wall temperature and fluid velocity parameters are measured. Then, the heat transfer coefficient is computed by Equation (5). Figure 6 shows the heat transfer coefficient as a function of the flow Reynolds number, for a particle volume concentration of φ=0.4% and five different average wall temperatures (T_{wall}=40 to 80℃). It is obvious that for all average wall temperatures, the heat transfer coefficient increases significantly with flow Reynolds number. The heat transfer coefficient of nanofluids increases as a result of the mixing effect of particles near the wall, Brownian motion of particles, reduction in boundary layer thickness and delay in boundary layer development [4, 13, 27].

Values of the convective heat transfer coefficient of nanofluid (with 4nm particle size) with respect to distilled water for different flow Reynolds number is presented in Figure 7. It is clear that the maximum enhancement takes place in a minimum Reynolds number and maximum particle volume concentration.

As Figure 7 shows, convective heat transfer coefficient (expect 4 %vol) increases with flow Reynolds number. This trend is almost true for all particles volume concentration and Reynolds number except 4% particle volume concentration. In this case, it is possible that, high volume concentration and low velocity flow in the pipe lead to agglomeration of particles. This behavior increases the thermal conductivity and heat performance.

6. 2. The effect of Tube Wall Temperature on the Heat Transfer

Figure 8 exhibits the heat transfer coefficient versus the temperature of the tube wall for 1.93cc/s volume flow rate and four different particle volume concentrations. As Figure 8 demonstrates, heat transfer coefficient improved by increasing particle volume concentration and increasing the temperature of the tube wall. Therefore, maximum heat transfer coefficient occurs at the maximum particle volume concentration.

The particle volume concentration of 0.5% and 1%, enhancement in the heat transfer coefficient for base fluid and nanofluid is almost equal.

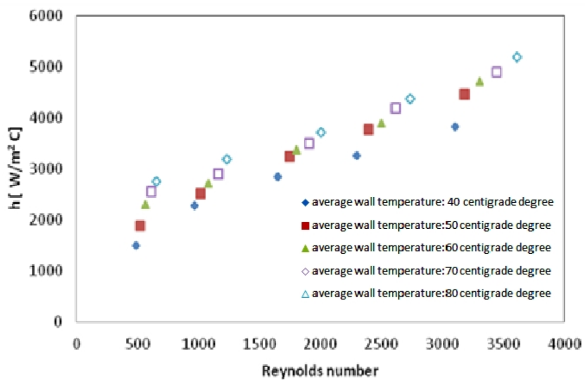

Figure 6. Plot of coefficient of heat transfer versus Reynolds number in 4% volume fraction of nanofluid

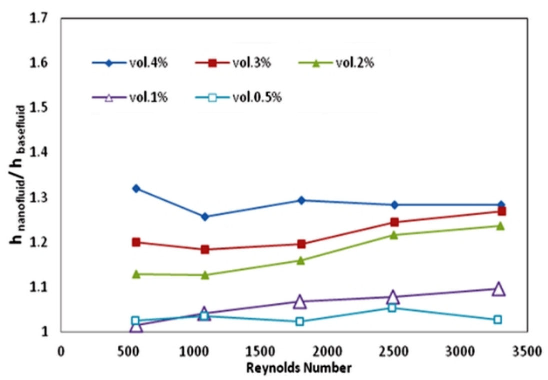

Figure 7. The ratio of the heat transfer coefficient of water-based Al_2O_3nanofluids to that of base fluid as a function of the Reynolds number.

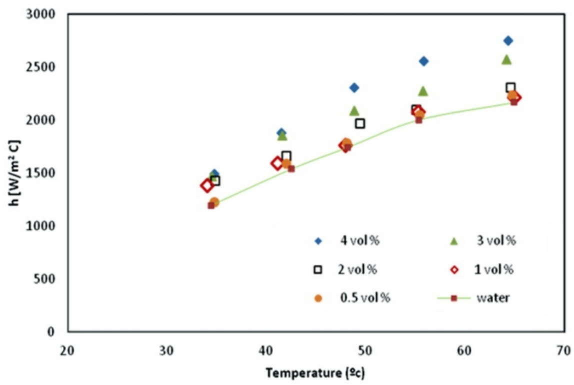

Figure 8. Plot of heat transfer coefficient versus temperature at constant volume flow rate 1.93cc/s

TABLE 1. Y=ax+b (y: heat transfer coefficient, x: Reynolds number)

Average wall temperature	a	b	R^2
40 (4%vol)	0.8135	1202.2	0.9765
50 (4%vol)	0.9593	1474.9	0.9943
60 (4%vol)	0.8371	1853.7	0.9948
70 (4%vol)	0.7899	2011.2	0.9947
80 (4%vol)	0.7394	2234.5	0.9906

Using the relationship between convective heat transfer coefficient and flow Reynolds number for constant tube wall temperature (Figure 6), a correlation for h, given by $h=aR+b$ is developed, where R is flow Reynolds number. Values of constants a and b for each tube wall temperature are presented in Table 1. For tube wall temperature of 50 and 60°C and 4 vol% we have maximum heat transfer coefficient rates. This effect is observed for other particle volume concentrations such as 0.5, 1, 2 and 3 vol%. For example, rate of enhancement at 50℃ is 95% whenever at 80℃ the rate is 75%. This effect can be explained by the hysteresis phenomena on viscosity, which revealed the existence of a critical temperature. Nanofluid viscosity reaches its lowest level at a critical temperature. Beyond this temperature, the viscosity increases by increasing temperature. Critical temperature depends on various parameters such as particle size and particle volume concentration [28]. Moreover, Yang et al. [29] show 22% increase in heat transfer coefficient over the base fluid at 50 °C. He also revealed that for 70℃ the heat transfer coefficient increases 15%.

6. 3. Correlations for Nusselt Number

The heat transfer behavior of the nanofluid is characterized by various factors such as thermal conductivity, heat capacity, thermal expansion coefficient, viscosity and particle volume concentration.

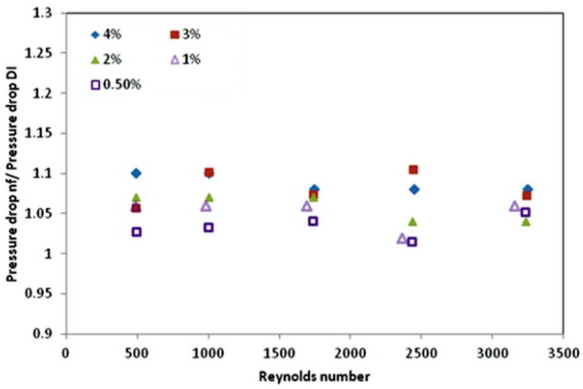

Figure 9. The ratio of pressure drop of nanofluids to that of distilled water versus Reynolds number for different volume fraction

Using the experimental data, the following correlation was determined for the Nusselt number in a W-shaped tube as a function of the particle volume concentration, Peclet, Reynolds and prandtle number based on the relation proposed by Xuan and Li [30].

$$Nu_{nf} = 0.48\left(1 + 5.24 f^{1.191} Pe_d^{\ 0.001}\right) \mathrm{Re}_{nf}^{\ 0.447} \ \mathrm{Pr}_{nf}^{\ 0.4} \tag{15}$$

(This equation is applicable for $450 < \mathrm{Re} < 4000$).

6. 4. Pressure Drop To use nanofluids in industry, pressure drop must be investigated, as well. Figure 9 shows the ratio of pressure drop of nanofluid to distilled water as a function of Reynolds number.

The current result shows that pressure drop does not increase significantly for fluid with suspended nanoparticles.

7. CONCLUSIONS

In this study, mixed and forced convection flow of water and water-Al_2O_3 mixture inside a vertical w-shaped tube with a constant wall temperature boundary condition has been investigated experimentally. The results confirmed that heat transfer improved with the addition of alumina to the base fluid (water), also the ratio of convective heat transfer coefficient of nanofluid to that of distilled water slightly increased with Reynolds number. Moreover, the rate of heat transfer coefficient with Reynolds number in mean wall temperatures was greater than the other wall temperatures tested in the present study. Furthermore, it was observed that the pressure drop of nanofluids was very close to the base fluid for a given Reynolds number. However, this difference between nanofluid and base fluid increased by raising the volume fraction of nanoparticles.
From the result of this research, a new correlation is proposed for computing the Nusselt number for W-shaped vertical tubes as a function of the particle volume concentration, Peclet, Reynolds and prandtle number.

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Experimental Investigation of Mixed Convection Heat Transfer in Vertical Tubes by Nanofluid: Effects of Reynolds Number and Fluid Temperature

RESEARCH NOTE

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چکیده

در این مطالعه، انتقال حرارت جابجایی نانو سیال آلومینا، درون یک لوله مسی عمودی و W شکل با شرط مرزی دمای دیوار ثابت مورد مطالعه قرار گرفته است . آزمایشات انجام شده محدوده ای از پارامترها نظیر عدد رینولدز، دما و غلظت نانو ذرات را در بر می گیرد. نتایج نشان دهنده بهبود نرخ ضریب انتقال حرارت با عدد رینولدز در دمای متوسط دیواره 50 و 60 درجه سانتیگراد است. علاوه بر آن نرخ انتقال حرارت با افزایش عدد رینولدز اندکی افزایش می یابد. همچنین افت فشار نانو سیال بسیار نزدیک به سیال پایه است و در نهایت رابطه ای برای محاسبه عدد ناسلت درون لوله W شکل پیشنهاد شده است.

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