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Heat transfer enhancement and pressure drop characteristics of TiO_2–water nanofluid in a double-tube counter flow heat exchanger

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ABSTRACT

This article reports an experimental study on the forced convective heat transfer and flow characteristics of a nanofluid consisting of water and 0.2 vol.% TiO_2 nanoparticles. The heat transfer coefficient and friction factor of the TiO_2–water nanofluid flowing in a horizontal double-tube counter flow heat exchanger under turbulent flow conditions are investigated. The Degussa P25 TiO_2 nanoparticles of about 21 nm diameter are used in the present study. The results show that the convective heat transfer coefficient of nanofluid is slightly higher than that of the base liquid by about 6–11%. The heat transfer coefficient of the nanofluid increases with an increase in the mass flow rate of the hot water and nanofluid, and increases with a decrease in the nanofluid temperature, and the temperature of the heating fluid has no significant effect on the heat transfer coefficient of the nanofluid. It is also seen that the Gnielinski equation failed to predict the heat transfer coefficient of the nanofluid. Finally, the use of the nanofluid has a little penalty in pressure drop.

1. Introduction

New energy-efficient heat transfer equipment stands at the point of a miniature increase in heat flux on one hand and an astronomical one on the other. Heat transfer fluids such as water, minerals, oil and ethylene glycol play a vital role in many industry processes, including power generation, chemical processes, heating and cooling processes, transportation, microelectronics and other micro-sized applications. The poor heat transfer properties of these fluids compared with those of most solids are the primary hindrance of high compactness and the effectiveness of the heat exchanger. The key idea is to exploit the very high thermal conductivities of solid particles that can be several hundreds of times greater than all of the conventional fluids combined. As a result, an important need still exists to develop new strategies in order to improve the effective heat transfer behaviours of conventional fluids. An innovative way of improving the thermal conductivities of common fluids is to suspend small solid particles in the fluids. Various types of particles, such as metallic, non-metallic and polymeric, can be added into fluids to form slurries. However, slurries with suspended particles of the order of micrometre or even millimetre may cause some severe problems. The abrasive action of the particles causes clogging of flow channels, eroding of pipelines and a reduction in their momentum transfer increase in pressure drop in practical applications [1]. Furthermore, they suffer from stability and rheological problems. In particular, the particles tend to settle quickly out of the suspension. Although the slurries have higher thermal conductivities than those of conventional heat transfer fluids, they are still not appropriate to use as heat transfer fluids for practical applications.

A novel approach to engineering fluids with better heat transfer properties is based on the rapidly emerging field of nanotechnology. The use of particles of nanometre dimensions first materialized in a series of studies conducted at the Argonne National Laboratory about a decade ago, and Choi [2] was probably the first to call the fluids with suspended particles of nanometre dimensions "nanofluids", which has gained popularity.

Compared with millimetre or micrometre-sized particle suspensions, nanofluids possess better long-term stability and rheological properties, and can have dramatically increased thermal conductivities. Over the past 10 years, many researchers have studied the heat transfer characteristics of the various nanofluids. Focusing on the forced convective heat transfer experimentally, several existing published articles which involve the use of nanofluids are discussed in the following sections.

Pak and Cho [3] studied the heat transfer performance of γ-Al_2O_3 and TiO_2 nanoparticles dispersed in water flowing in a horizontal circular tube. Alumina (Al_2O_3) and titanium dioxide (TiO_2) nanoparticles with diameters of 13 nm and 27 nm, respectively, were used in their study. They found that the Nusselt number of nanofluids increased with an increase in the Reynolds number as

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Nomenclature

C_p	specific heat, J/kg K
d	nanoparticle diameter, m
D	tube diameter, m
f	friction factor
h	heat transfer coefficient, W/m^{2}K
k	thermal conductivity, W/mK
L	length of the test tube, m
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number
ΔP	pressure drop, Pa
Pe	Peclet number
Pr	Prandtl number
q	heat flux, W/m^2
Q	heat transfer rate, W
Re	Reynolds number
T	temperature, °C
u	mean velocity, m/s
V	volume, m^3
$\dot{V}_h$	hot water flow rate, LPM

Greek symblos

ϕ	volume fraction
ε	tube roughness
ρ	density, kg/m^3
α	thermal diffusivity, m^2/s
μ	viscosity, kg/ms

Subscript

ave	average
f	fluid
h	heating fluid
in	inlet
m	mean
out	outlet
p	particles
nf	nanofluid
w	water
wall	tube wall

well as the volume fraction. However, they still found that the convective heat transfer coefficient of the nanofluids with 3 vol.% nanoparticles was 12% lower than that of pure water at a given Reynolds number. This may cause the nanofluids to have larger viscosity than that of pure water, especially at high particle volume fractions. Finally, a new heat transfer correlation for predicting the convective heat transfer coefficient of nanofluids in a turbulent flow regime was proposed.

Li and Xuan [4] and Xuan and Li [5] investigated experimentally the convective heat transfer and flow characteristics for Cu–water nanofluid flowing through a straight tube with a constant heat flux under laminar and turbulent flow conditions. Cu nanoparticles with diameters below 100 nm were used in their study. The results of the experiment showed that the suspended nanoparticles remarkably enhanced the heat transfer performance of the conventional base fluid and their friction factor coincided well with that of the water. Furthermore, they also proposed the new convective heat transfer correlations for prediction of the heat transfer coefficients of the nanofluid for both laminar and turbulent flow conditions.

Tsai et al. [6] investigated gold–DI water nanofluid flowing in a conventional heat pipe with a diameter of 6 mm and a length of 170 mm. Gold nanoparticles of 2–35 nm and 15–75 nm in size were used in this study. Their data showed that the nanofluid causes a significant reduction in the thermal resistance of the heat pipe compared with DI water at given concentrations. The thermal resistance of the circular heat pipe ranged from 0.17 to 0.215 °C/W with various nanoparticle concentrations. The results indicated that the higher thermal potential of nanofluids means that they can be used as working fluids to replace the conventional fluids in vertical circular meshed heat pipes.

Wen and Ding [7] studied the convective heat transfer coefficients in which γ-Al_2O_3 nanoparticles were suspended in deionized water for laminar flow in a copper tube under a constant wall heat flux and focused in particular on the entrance region. Alumina nanoparticles of 27–56 nm in size were used in this study. The results show that the local heat transfer coefficient varied with the Reynolds number and particle concentration. In particular, it was found that the use of nanofluids at the entrance region resulted in a pronounced increase in the heat transfer coefficient, causing a decrease in the thermal boundary layer thickness which decreased with the axial distance. This behaviour implied that it might be possible to create a "smart entrance" region to meet the highest performance of nanofluids. Furthermore, the calculated Nusselt number using the Shah correlation for laminar flow and the Dittus–Boelter equation for turbulent flow did not coincide with the experimental results.

Yang et al. [8] reported on an experiment which studied the convective heat transfer coefficient of graphite nanoparticles dispersed in liquid for laminar flow in a horizontal tube heat exchanger. Disc-shaped nanoparticles of different sources (aspect ratio (l/d) of about 0.02) were used in this study. The experimental results showed that the heat transfer coefficient increased with the Reynolds number as well as the particle volume fraction. Furthermore, two graphite nanoparticle sources at the same particle loading gave different heat transfer coefficients. The correlation established by Li and Xuan [4] for laminar flow of nanofluids gave a larger heat transfer coefficient than that calculated from the experimental data. Following this, all of the experimental data were used to develop a new heat transfer correlation for the prediction of the heat transfer coefficient of laminar flow nanofluids in a more convenient form by modifying the Seider–Tate equation (1936).

Ding et al. [9] investigated the heat transfer performance of CNT nanofluids flowing through a tube with a 4.5 mm inner diameter. Their results showed that the heat transfer coefficient of CNT nanofluids is much larger than that of pure water and the enhancement depends on the flow conditions, CNT concentration and PH value. They suggested that possible reasons for enhancement cannot only be attributed to the augmented thermal conductivity. Particle rearrangement, shear-induced thermal conduction enhancement, reduction of the thermal boundary layer due to the presence of nanoparticles and a high aspect ratio of CNTs are all associated with the enhancement. These observations suggest that the aspect ratio should be associated with the high enhancement of heat transfer performance of CNT-based nanofluids [10].

Heris et al. [11,12] investigated the convective heat transfer coefficient of Al_2O_3–water and CuO–water nanofluids for laminar flow in an annular tube under a constant wall temperature boundary condition. This condition was rather different from the constant heat flux which was investigated by other researchers. The test section consisted of a 1 m annular tube in which an inner copper tube was constructed with a 6 mm inner diameter and a 0.5 mm thickness, while the outer tube was made from stainless steel with a 32 mm outer diameter. Saturated steam was circulated to create a constant wall temperature boundary condition. The results showed that the heat transfer coefficient increased with an increasing Peclet number as well as volume fraction and the Al_2O_3–water nanofluid showed larger enhancement than CuO–water nanofluid.

He et al. [13] reported the results of an experiment in which the heat transfer and flow behaviour of TiO_2–distilled water nanofluids were flowing in an upward direction through a vertical pipe in both the laminar and turbulent flow regimes under a constant heat flux boundary condition. The results showed that the convective heat transfer coefficient increased with an increase in nanoparticle concentration in both the laminar and turbulent flow regimes at a given Reynolds number and particle size. Similarly, at a given nanoparticle concentration and Reynolds number, the heat transfer coefficient did not seem to be sensitive to the average particle size under the conditions of the experiment. The small effect of particle size on the heat transfer coefficient could be due to migration of the nanoparticles. As for the pressure drop of the nanofluids, they found that this was approximately the same as that of the base fluid.

Nguyen et al. [14] investigated the heat transfer enhancement and behaviour of the Al_2O_3–water nanofluid flowing under a turbulent flow regime inside the cooling system of microprocessors or other electronic components. Their results showed that the nanofluid gave a larger heat transfer coefficient than the base fluid and that the nanofluid with a 36 nm particle diameter provided a higher heat transfer coefficient than the nanofluid which had particles of 47 nm in size.

As mentioned above, many researchers have used a wide variety of nanoparticles such as copper, aluminium and their oxides. However, titanium dioxide was not widely used in the above reviewed literature. In this study, the main reasons for choosing titanium dioxide as the nanoparticle are summarized as follows: (1) it is cheap because it is produced in commercially available products, (2) it is generally regarded as a safe material for human beings and animals (normally used in cosmetic products and water treatment) [13] and (3) it has excellent chemical and physical stability even without an additional stabilizer [15]. Moreover, it can be noted that the experimental investigations found in the literature described above focused on the constant heat flux and the constant wall temperature boundary condition, the heat transfer and flow characteristics of nanofluids in a double-tube heat exchanger remain unstudied. With this consequence, this article is aimed at studying the heat transfer enhancement and flow characteristics of TiO_2–water nanofluids at a low concentration flowing in a horizontal concentric tube-in-tube heat exchanger under a turbulent flow condition.

2. Experimental apparatus

As shown in Fig. 1, the apparatus used in this experiment consists of a test section, two receiver tanks, a magnetic gear pump, a hot water pump, a cooler tank, a hot water tank and a collection tank. The test section is a 1.5 m long counter flow horizontal double-tube heat exchanger with nanofluid flowing inside the tube while hot water flows in the annular. The inner tube is made from smooth copper tubing with a 9.53 mm outer diameter and an 8.13 mm inner diameter, while the outer tube is made from PVC tubing and has a 33.9 mm outer diameter and a 27.8 mm inner diameter. The test section is thermally isolated from its upstream and downstream section by plastic tubes in order to reduce the heat loss along the axial direction. The differential pressure transmitter and T-type thermocouple are mounted at both ends of the test section to measure the pressure

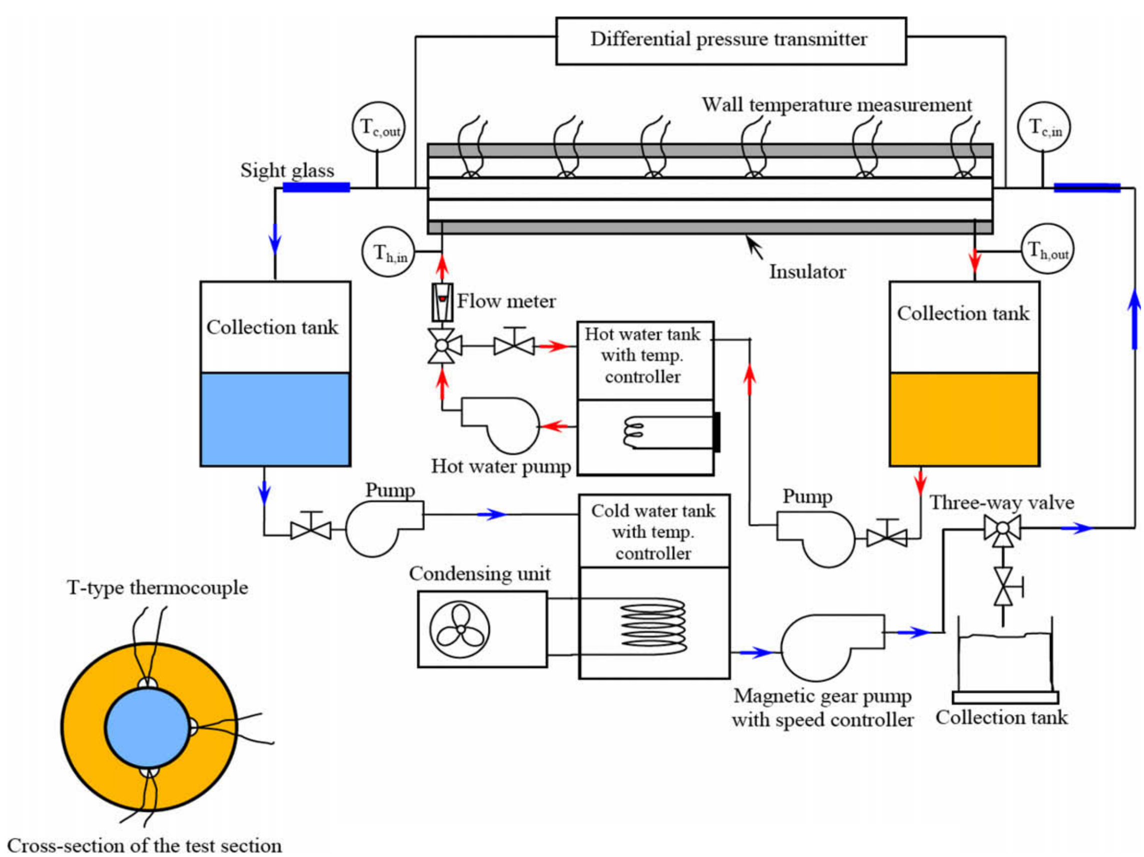

Fig. 1. Schematic diagram of the experimental apparatus.

drop and the bulk temperature of the nanofluid, respectively. Thermocouples are mounted at different longitudinal positions on the inner tube surface of the wall, each with three thermocouples equally spaced around the tube circumference. The inlet and exit temperatures of hot water are measured using T-type thermocouples which are inserted into the flow directly. The receiver tanks of 60 L are made from stainless steel to store the nanofluid and hot water leaving the test section. The cooler tank with a 4.2 kW cooling capacity and a thermostat is used to keep the nanofluid temperature constant. Similar to the cooler tank, a 3 kW electric heater with a thermostat was installed to keep the temperature of the hot water constant. The nanofluid flow rate is controlled by adjusting the rotation speed of the magnetic gear pump. The hot water flow rate is measured by a rotameter while the flow rate of the nanofluid is determined by the time taken for a given volume of nanofluid to be discharged.

All of the T-type thermocouples were calibrated with a portable programmable calibrator which has a maximum precision of 0.1 °C. The differential pressure transmitter was calibrated using an air operated dead weight tester. The uncertainty of the pressure measurement is ±0.030 kPa. Moreover, the nanofluid flow rates were determined by electronic balance. The uncertainty of the electronic balance is ±0.0006 kg. Therefore, the uncertainty of the heat transfer coefficient is around 5%.

During the test run, the inlet and exit temperatures of the hot water and nanofluids, wall temperatures of the test tube, mass flow rates of the hot water and nanofluid, and the pressure difference in the nanofluid will be measured.

3. Sample preparation

The term "nanofluid" does not mean a simple mixture of solid particles and base fluid. In order to prepare the nanofluids by dispersing the nanoparticles in a base fluid, proper mixing and stabilization of the particles is required. Normally, there are three effective methods used to attain stability of the suspension against sedimentation of the nanoparticles, which are summarized as follows: (1) control of the ph value of the suspensions, (2) addition of surface activators or surfactants and (3) use of ultrasonic vibration. All of these techniques aim at changing the surface properties of the suspended nanoparticles and suppressing the formation of clusters of particles in order to obtain stable suspensions. In this study, the ultrasonic vibrator and the additional surfactant methods were used for dispersing the nanoparticles into the base water. DEGUSSA P25 TiO_2 nanoparticles with mean diameters of 21 nm were used. Cetyltrimethylammoniumbromide (CTAB) with very low concentrations (about 0.01%) were used as surfactants and first mixed with water to ensure better stability and proper dispersion of the nanoparticles without affecting the thermo-physical properties of the nanofluid [15]. Nanofluids with a required volume fraction of TiO_2 were then prepared by dispersing the nanoparticles in specific amounts in the water base fluid. Following this, the nanofluids were sonicated continuously for 3–4 h using an ultrasonic vibrator in order to ensure complete dispersion. The transmission electron microscope was used to monitor the dispersion of the nanoparticles in the water, as shown in Fig. 2. The figure shows that a little agglomeration was observed 3 h after it was sonicated.

4. Data reduction

The heat transfer rate from the heating fluid is defined as:

$$Q_w = \dot{m}_w Cp_w (T_{in} - T_{out})_w \quad (1)$$

where Q_w is the heat transfer rate of the hot water and $\dot{m}_w$ is the mass flow rate of the hot water.

The heat transfer rate into the nanofluid is calculated from:

$$Q_{nf} = \dot{m}_{nf} Cp_{nf} (T_{out} - T_{in})_{nf} \quad (2)$$

where Q_{nf} is the heat transfer rate of the nanofluid and $\dot{m}_{nf}$ is the mass flow rate of the nanofluid.

The average heat transfer rate is defined as follows:

$$Q_{ave} = \frac{Q_w + Q_{nf}}{2} \quad (3)$$

where Q_{ave} is the average heat transfer rate between the hot water and the nanofluid.

In the present study, the energy differences between the heating fluid and the nanofluid are around 3%.

The measured Nusselt number and heat transfer coefficient of the nanofluid are calculated from the following equations:

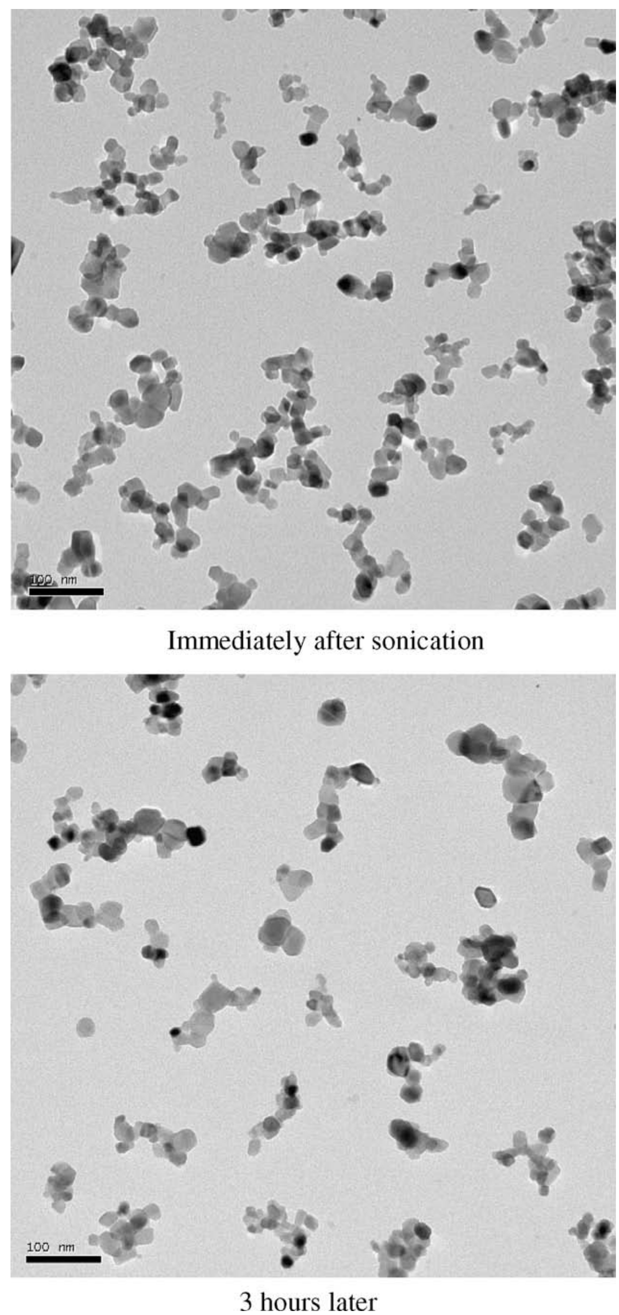

Fig. 2. TEM image of 0.2 vol.% dispersed TiO_2 nanoparticles in water.

$$h_{nf} = \frac{q_{ave}}{T_{wall} - T_{nf}} \tag{4}$$

$$Nu_{nf} = \frac{h_{nf} D}{k_{nf}} \tag{5}$$

where h_{nf} is the heat transfer coefficient of the nanofluid, q_{ave} is the average heat flux between the hot water and the nanofluid, T_{wall} is the average temperature of the wall, T_{nf} is the bulk temperature of the nanofluid, Nu_{nf} is the Nusselt number of the nanofluid, D is the inner diameter of the test tube and k_{nf} is the thermal conductivity of the nanofluid.

Similarly to the heat transfer coefficient, the friction factor of the nanofluid flowing through the test section is defined as:

$$f_{nf} = \frac{2D\Delta P_{nf}}{L\rho_{nf} u_m^2} \tag{6}$$

where f_{nf} is the friction factor of the nanofluid, ΔP_{nf} is the measured pressure drop of the nanofluid, L is the length of the tube, ρ_{nf} is the density of the nanofluid and u_m is the mean velocity of the nanofluid.

The physical properties such as the density, viscosity, and specific heat and thermal conductivity of the nanofluid are calculated using the following published correlations.

The density is calculated from Pak and Cho [3] using the following equation:

$$\rho_{nf} = \phi\rho_p + (1-\phi)\rho_w \tag{7}$$

where ϕ is the volume fraction of the nanoparticles, ρ_p is the density of the nanoparticles and ρ_w is the density of the base fluid.

Drew and Passman [16] suggested the well-known Einstein equation for calculating the viscosity, which is applicable to spherical particles in volume fractions of less than 5.0 vol.% and is defined as follows:

$$\mu_{nf} = (1 + 2.5\phi)\mu_w \tag{8}$$

where μ_{nf} is the viscosity of the nanofluid and μ_w is the viscosity of the base fluid.

In the present study, a very low concentration nanofluid with 0.2 vol.% is used. This equation can be applied to estimate the viscosity of the nanofluid [7].

The specific heat is calculated from Xuan and Roetzel [17] as follows:

$$(\rho Cp)_{nf} = \phi(\rho Cp)_p + (1-\phi)(\rho Cp)_w \tag{9}$$

where $(\rho Cp)_{nf}$ is the heat capacity of the nanofluid, $(\rho Cp)_p$ is the heat capacity of the nanoparticles and $(\rho Cp)_w$ is the heat capacity of the base fluid.

The thermal conductivity of the nanofluid is calculated from Yu and Choi [18] using the following equation:

$$k_{nf} = \left[\frac{k_p + 2k_w + 2(k_p - k_w)(1+\beta)^3\phi}{k_p + 2k_w - (k_p - k_w)(1+\beta)^3\phi}\right] k_w \tag{10}$$

where k_{nf} is the thermal conductivity of the nanofluid, k_p is the thermal conductivity of the nanoparticles, k_w is the thermal conductivity of the base fluid and β is the ratio of the nanolayer thickness to the original particle radius. Normally a value of $\beta = 0.1$ is used to calculate the thermal conductivity of the nanofluid [18].

The properties of the nanofluid shown in the above equations are evaluated from water and nanoparticles at average bulk temperature.

5. Results and discussion

Before starting to determine the convective heat transfer coefficient and friction factor of the nanofluid, the reliability and accuracy of the experimental system are estimated by using water as the working fluid. The results of the experimental heat transfer coefficient and friction factors are compared with those obtained from the Gnielinski equation [19] and Colebrook equation [20] which are defined as follows:

The Gnielinski equation is defined as:

$$Nu = \frac{(f/8)(Re - 1000)Pr}{1 + 12.7(f/8)^{0.5}(Pr^{2/3} - 1)} \tag{11}$$

where Nu is the Nusselt number, Re is the Reynolds number, Pr is the Prandtl number and f is the friction factor.

The Colebrook equation is defined as:

$$\frac{1}{\sqrt{f}} = -2.0\ell og\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{Re\sqrt{f}}\right) \tag{12}$$

where ε is the roughness of the test tube.

Moreover, the Pak and Cho [3] and Xuan and Li [5] correlations for predicting the Nusselt number for nanofluid are compared with the results which are defined as follows:

The Pak and Cho correlation is defined as:

$$Nu_{nf} = 0.021 Re_{nf}^{0.8} Pr_{nf}^{0.5} \tag{13}$$

The Xuan and Li correlation is defined as:

$$Nu_{nf} = 0.0059(1.0 + 7.6286\phi^{0.6886} Pe_d^{0.001}) Re_{nf}^{0.9238} Pr_{nf}^{0.4} \tag{14}$$

The Reynolds number of the nanofluid is defined as:

$$Re_{nf} = \frac{\rho_{nf} u_m D}{\mu_{nf}} \tag{15}$$

The Prandtl number of the nanofluid is defined as:

$$Pr_{nf} = \frac{\mu_{nf} Cp_{nf}}{k_{nf}} \tag{16}$$

and the Peclet number of the nanofluid in Eq. (14) is defined as:

$$Pe_{nf} = \frac{u_m d_p}{\alpha_{nf}} \tag{17}$$

where d_p is the diameter of the nanoparticles.

In order to calculate the Peclet number, the thermal diffusivity of the nanofluid (α_{nf}) is defined as:

$$\alpha_{nf} = \frac{k_{nf}}{\rho_{nf} Cp_{nf}} \tag{18}$$

As shown in Figs. 3 and 4, an agreement between the experimental results and the calculated values for pure water can be seen.

In the present study, TiO_2 nanoparticles mixed with the water by 0.2 vol.% are used to investigate the effect of the Reynolds number and temperature of the flowing nanofluid and mass flow rate and temperature of the hot water on the heat transfer and the pressure drop characteristics of the nanofluid. The experimental conditions that are used in this study are as follows:

(1) the Reynolds number of the nanofluid varies in the approximate range of 4000–18,000,
(2) the temperature of the nanofluid is 15 °C, 20 °C and 25 °C,
(3) the mass flow rates of the hot water are 3 LPM and 4.5 LPM,
(4) the temperature of the hot water is 35 °C, 40 °C, 45 °C and 50 °C.

The results are reported and discussed in the following subsection.

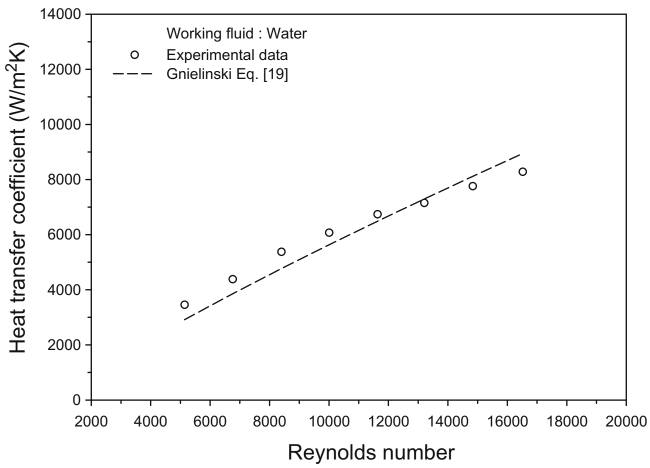

Fig. 3. Comparison between measured heat transfer coefficient and that calculated from Gnielinski equation [19].

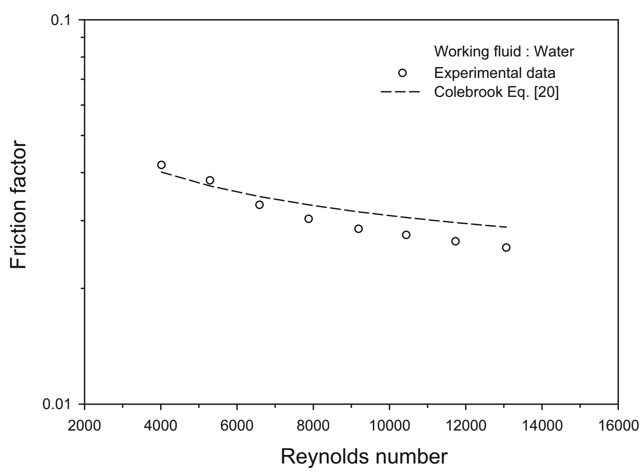

Fig. 4. Comparison between measured friction factor and that calculated from Colebrook equation [20].

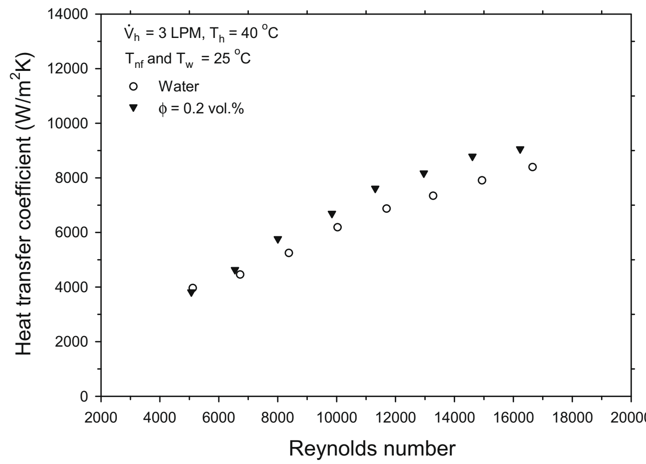

Fig. 5. Comparison of heat transfer coefficient obtained from water and that from the 0.2 vol.% of TiO_2 nanoparticles dispersed in water.

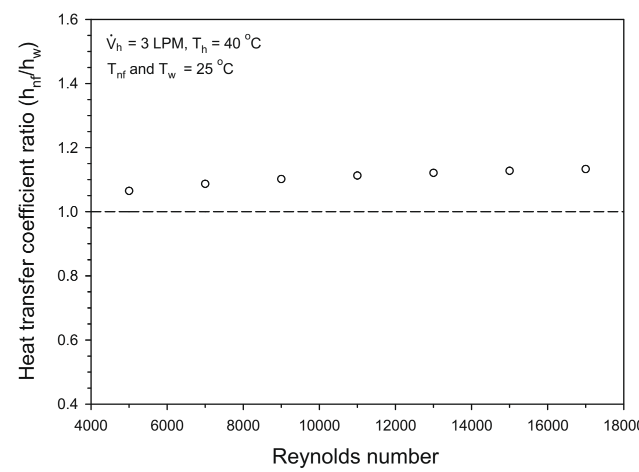

Fig. 6. Heat transfer coefficient ratio (h_{nf}/h_w) against Reynolds number.

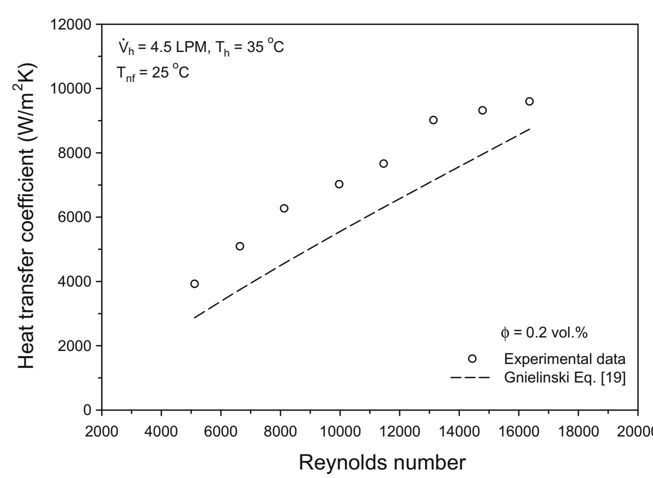

Fig. 7. Comparison of heat transfer coefficient between measured data and calculated values from Gnielinski equation [19].

5.1. Convective heat transfer coefficient

As shown in Fig. 5, the heat transfer coefficient increases with an increasing Reynolds number. It can be clearly seen that the convective heat transfer coefficient of the nanofluid is higher than that of the base fluid (water) at a given Reynolds number. The results complied with those obtained from Pak and Cho [3], Xuan and Li [5] and He et al. [13]. The possible reason for this enhancement may be associated with the following: (1) the nanofluid with suspended nanoparticles increases the thermal conductivity of the mixture and (2) a large energy exchange process resulting from the chaotic movement of nanoparticles [17].

As shown in Fig. 6, the ratio of the convective heat transfer coefficient of the TiO_2–water nanofluid to that of water varies from 1.06 to 1.11 under the same Reynolds number. This means that the nanofluid had a higher heat transfer coefficient than that of water in the range of approximately 6–11%.

Fig. 7 shows that the Gnielinski equation fails to predict the convective heat transfer coefficient of the nanofluid. This may be due to the fact that the correlation was established from single-phase fluid data and was valid only for the single-phase flow.

As shown in Fig. 8, the calculated values from the Pak and Cho [3] correlation are closer to the results of the experiment than the values calculated using the Xuan and Li [5] correlation. The correlation established by Xuan and Li [5] for turbulent flow of nanofluids gives a lower heat transfer coefficient than that which is calculated using the experimental data and the Pak and Cho

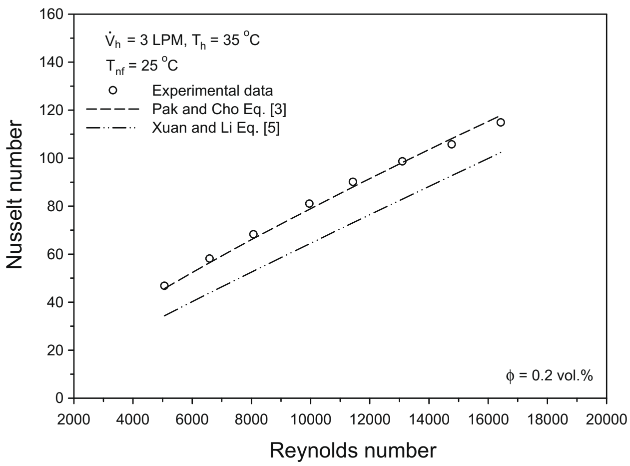

Fig. 8. Comparison of Nusselt number between the measured data and calculated values from nanofluid correlations.

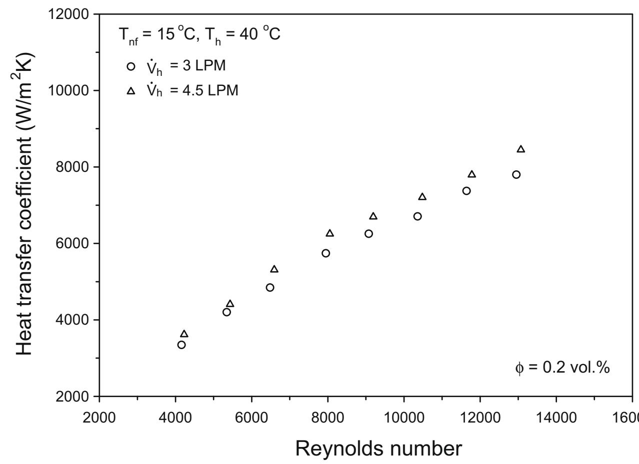

Fig. 10. Convective heat transfer coefficient as a function of Reynolds number with various hot water flow rates.

correlation [3]. This may be because the Pak and Cho correlation was formed from the data of TiO_2–water nanofluids; however, the Xuan and Li correlation was established from the data of Cu–water nanofluids.

5.1.1. Effect of the temperature of the nanofluid

Fig. 9 shows the variation of the heat transfer coefficient with the Reynolds number at a hot water flow rate of 4.5 LPM and a hot water temperature of 45 °C with variations in the temperature of the nanofluid. In Fig. 9, the effect of the nanofluid temperature can be seen at a higher Reynolds number, i.e. the heat transfer coefficient is higher for a lower nanofluid temperature than for a higher nanofluid temperature, because decreasing the nanofluid temperature leads to an increase in the heat transfer rate which results in an increase in the heat transfer coefficient.

5.1.2. Effect of the hot water flow rate

Fig. 10 shows the effect of the hot water mass flow rate on the heat transfer coefficient of the nanofluid at a hot water temperature of 40 °C and a nanofluid temperature of 15 °C with different hot water flow rates. The results show that the heat transfer coefficient of the nanofluid increases with increasing hot water flow rates. At the Reynolds number, the heat transfer coefficient at the water flow rate of 3 LPM is lower than at 4.5 LPM across the range of Reynolds numbers. Similar to the effect of the nanofluid temperatures, increasing the hot water flow rate leads to an increase in the heat transfer rate and results in an increase in the heat transfer coefficient of the nanofluid.

5.1.3. Effect of the hot water temperature

Fig. 11 shows the convective heat transfer coefficient as a function of the Reynolds number at a hot water flow rate of 3 LPM and a nanofluid temperature of 15 °C at different hot water temperatures. It can be clearly seen that the hot water temperature has no significant effect on the heat transfer coefficient. In the case of forced convective heat transfer, this is due to the effect of heating the fluid temperature on the heat transfer rate being very small compared with the effect of mass flow rate.

5.2. Pressure drop

In order to apply the nanofluid for practical application, it is important to study simultaneously the flow feature and heat transfer performance of the nanofluid. In this study, nanofluid with 0.2 vol.% suspended nanoparticles is used in a pressure test. As shown in Fig. 12, the friction factors of the nanofluid agree well with those of water data under the same Reynolds number. This may be because the small additional nanoparticles in the base

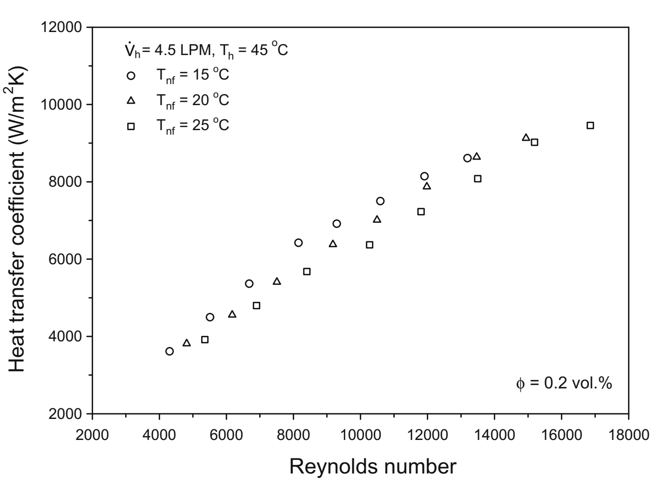

Fig. 9. Convective heat transfer coefficient as a function of Reynolds number with different nanofluid temperatures.

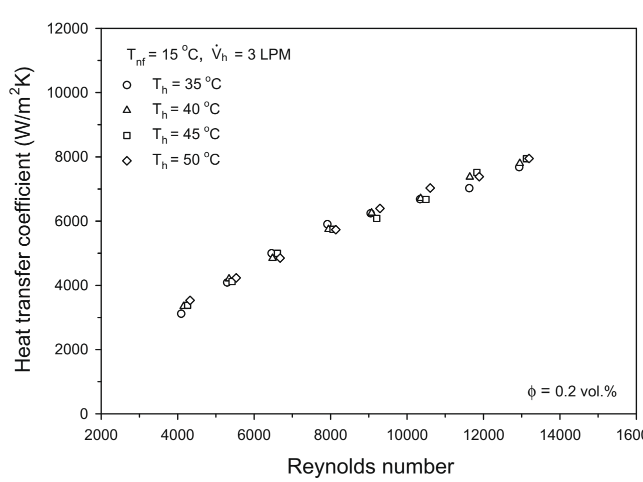

Fig. 11. Convective heat transfer coefficient as a function of Reynolds number with different hot water temperatures.

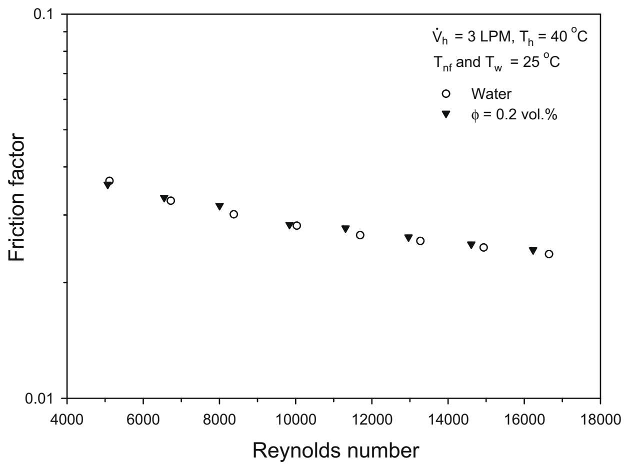

Fig. 12. Comparison of friction factor for water and 0.2 vol.% nanofluid.

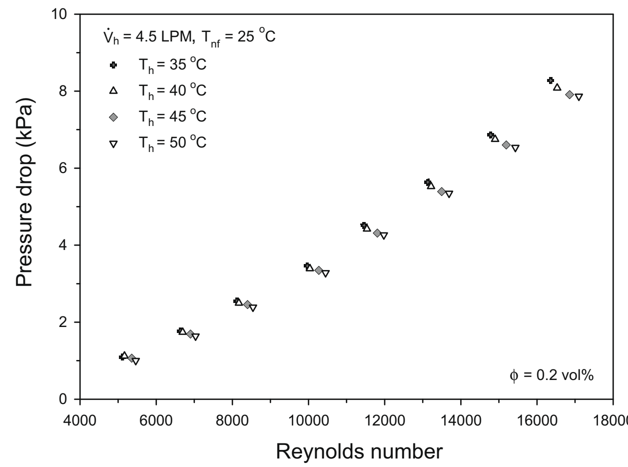

Fig. 14. Comparison of 0.2 vol.% nanofluid pressure drop with different hot water temperatures.

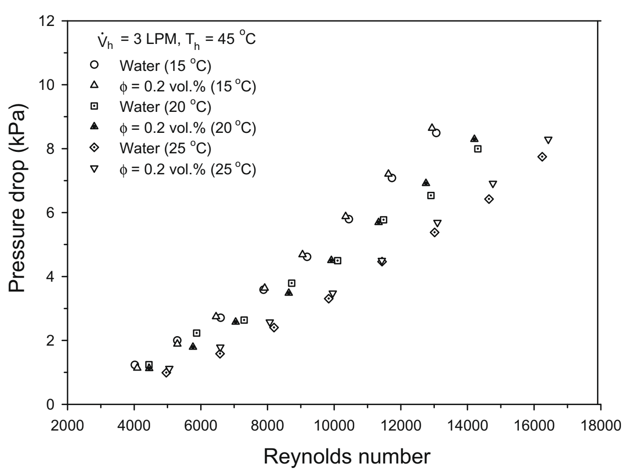

Fig. 13. Comparison of 0.2 vol.% nanofluid pressure drop and water pressure drop.

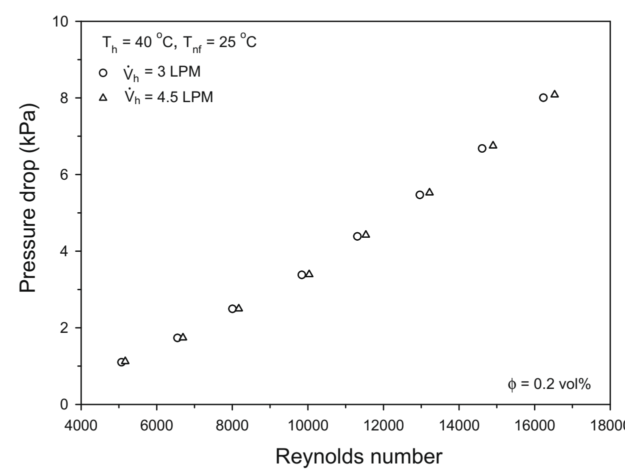

Fig. 15. Comparison of 0.2 vol.% nanofluid pressure drop with different hot water flow rates.

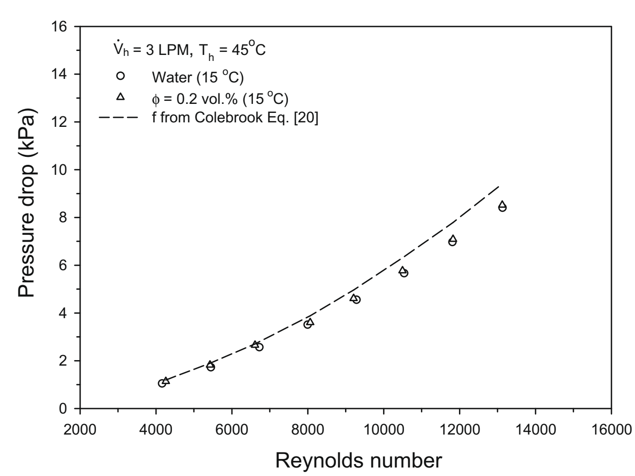

Fig. 16. Comparison of the measured pressure drop data and the calculated values obtained from Colebrook equation [20].

liquid do not cause the change in the flow behaviour of the fluid. This means that the nanofluid will not cause a penalty drop in pressure and there is no need for additional pump power.

Figs. 13 and 14 show the pressure drop as a function of the Reynolds number under different experimental conditions. These figures show that the pressure drop in the nanofluid is very close to that in the water for the given conditions and that it decreases with increases in its temperature and hot water temperatures. An increase in the nanofluid temperature and hot water temperature leads to a decrease in the viscosity of the nanofluid which results in a reduction in the pressure drop.

Fig. 15 shows the pressure drop as a function of the Reynolds number at a hot water temperature of 40 °C and a nanofluid temperature of 25 °C at different hot water flow rates. It can be clearly seen that the hot water flow rate has an insignificant effect on the pressure drop of the nanofluid. This is due to the increase in the hot water flow rate having a slight effect on the viscosity of the nanofluid, which leads to a tiny change in the measured pressure drop of the nanofluid.

The experimental results show the conventional single-phase pressure drop correlation may be used to predict a pressure drop in the very low concentration nanofluid. The prediction value agrees well with the measured data, as shown in Fig. 16.

Compared with all the results mentioned above, similar trends are observed from the rest of the experimental data. Although the nanofluid is actually a two-phase fluid in nature, the results show that the nanofluid behaves more like a pure fluid than a liquid–solid mixture.

6. Conclusions

The convective heat transfer performance and flow characteristic of a TiO_2–water nanofluid flowing in a horizontal double-tube counter flow heat exchanger was experimentally investigated. Experiments were carried out under turbulent flow conditions. The effects of the flow Reynolds number and the temperature of the nanofluid and the temperature and flow rate of the heating fluid on the heat transfer coefficient and flow characteristics were investigated. The following conclusions have been obtained.

- The use of TiO_2–water nanofluid significantly gives higher heat transfer coefficients than those of the pure base fluid.
- The Gnielinski correlation for predicting the heat transfer coefficient of pure fluid is not applicable to a nanofluid. Vice versa, the Pak and Cho correlation [3] for predicting the heat transfer coefficient of a nanofluid agreed better with the results of this experiment than the Xuan and Li correlation [5].
- The convective heat transfer coefficient increases with an increasing Reynolds number and an increasing mass flow rate of the heating fluid, and increases with a decreasing nanofluid temperature.
- The pressure drop and friction factor of the nanofluid are approximately the same as those of water in the given conditions. This implies that the nanofluid incurs no penalty of pump power and may be suitable for practical application.

Additional work is required to investigate the effects of different particle concentrations on the convective heat transfer coefficients and flow features of nanofluids. Moreover, the heat transfer correlation in its simplest form will predict the heat transfer coefficient of nanofluids flowing in a horizontal double-tube counter heat exchanger accurately.

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