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Thermal Performance Analysis of Nanofluids in a Concetric Heat Exchanger Equipped with Turbulators

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***Abstract*—**Turbulent forced convection heat transfer and pressure drop characteristics of Al_2O_3–water nanofluid flowing through a concentric tube heat exchanger with and without coiled wire turbulators were studied experimentally. The experiments were conducted in the Reynolds number ranging from 4000 to 20000, particle volume concentrations of 0.8 vol.% and 1.6 vol.%. Two turbulators with the pitches of 25 mm and 39 mm were used. The results of nanofluids indicated that average Nusselt number increased much more with increasing Reynolds number compared to that of pure water. Thermal conductivity enhancement by the nanofluids resulted in heat transfer enhancement. Once the pressure drop of the alumina/water nanofluid was analyzed, it was nearly equal to that of pure water at the same Reynolds number range. It was concluded that nanofluids with the volume fractions of 0.8 and 1.6 did not have a significant effect on pressure drop change. However, the use of wire coils in heat exchanger enhanced heat transfer as well as the pressure drop.

***Keywords*—**Turbulators, heat exchanger, nanofluids, heat transfer enhancement.

I. Introduction

NANOFLUIDS are a new class of heat transfer fluids created by dispersing nanoparticles with smaller average diameter than 100 nm into traditional heat transfer fluids such as water, ethylene glycol, and engine oil. Suspending particles to enhance heat transfer is not a new technique. Maxwell first proposed this method. In these studies, rapid settling of these particles in base fluids was the major problem [1]. Also these suspensions can obstruct the narrow channels of miniaturized devices. Nanofluids have superior properties like high thermal conductivity, increased heat transfer coefficient and minimal clogging in flow, because of small sizes and very large specific surface areas of the nanoparticles. These characteristics of nanofluids lead to a wide range of application area such as energy supply, automotive industry, chemical industry, heating/cooling system, microelectronics, and so on. In spite of such benefits, the nanofluid technology is still restricted for commercial use because there is not accurate prediction of heat transfer characteristics of nanofluids. The stability of nanofluids, thermal conductivity, and viscosity are some of them.

Nanofluids have been expected to be the next-generation heat transfer fluids. Metallic or nonmetallic nanoparticles can be used in nanofluids such as Al_2O_3, CuO, CNTs and diamond [2]. There are many experimental and numerical studies focusing on the flow and heat transfer characteristics of nanofluids [1]-[16].

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Convective heat transfer of nanofluids in laminar flow have been experimented by some researchers [3]-[7]. But, heat transfer in turbulent flow is much more common in practices because heating or cooling systems usually work under turbulent flow regime conditions.

In literature survey, it is known that nanofluids have better thermal performance than the base liquids at the same flow conditions, and heat transfer increases with increasing nanoparticle volume concentrations [8], [9]. However, effective dynamic viscosity also increases with nanoparticle volume concentration, and this increase causes an increase in pressure drop resulting in extra pumping power. For that reason, there are restrictions on the volumetric rate increase.

There are numerous studies published on experimental convective heat transfer of nanofluids through heat exchanger with or without turbulators, and the results of these studies have been reviewed in this work. Duangthongsuk and Wongwises [10], [11] produced nanofluids with TiO_2 (21 nm) nanoparticles dispersed into pure water with the volume concentration range of 0.2–2%. They studied the convective heat transfer and pressure drop of TiO_2-water nanofluid experimentally. Nanofluid flow was analysed in a horizontal double tube counter flow heat exchanger under turbulent flow conditions. They found that the heat transfer coefficient of nanofluid was higher than that of the base liquid and increased with increasing Reynolds number and particle concentration. Contrary, the heat transfer coefficient of the nanofluids at a volume concentration of 2.0% was lower than that of base fluid. The heat transfer coefficient of nanofluids with 1 vol.% was approximately 26% greater than base fluid. As for the volume concentration of 2.0 vol.%, it was approximately 14% lower than that of base fluid. The pressure drop of nanofluids was slightly higher than the base fluid and increased with increasing volume concentrations.

Kayhani et al. [12] examined the forced convective heat transfer of TiO_2-water nanofluid in turbulent flow regime. They used the nanoparticles in the concentration range of 0.1-2 vol.%. Their results indicated that heat transfer coefficient increased with increasing nanofluid volume fraction and did not change with varying Reynolds number. The enhancement of the Nusselt number was about 8% for nanofluid with 2.0% nanoparticle volume fraction at Re=11800.

Pak and Cho [13] investigated the heat transfer and pressure drop of Al_2O_3 (13 nm) and TiO_2 (27 nm) nanoparticles in pure water at $10^4<Re<10^5$. Nanofluid flow was investigated in a horizontal circular tube under constant heat flux boundary condition. They obtained that the heat transfer rate increased with increasing Reynolds number and nanoparticle volume fraction up to 3%. For higher particle volumetrical concentration than 3 %, heat transfer decreased. He et al. [14] examined heat transfer and flow characteristics of TiO_2-water nanofluid through a vertical pipe in both laminar and turbulent flow. Nanoparticle volume fraction was lower than 1.18%. They indicated that the convective heat transfer coefficient increased with increasing nanoparticle concentration, and the pressure drop of the nanofluid was very close to that of the base fluid.

Arani and Amani [15] performed an experimental study to investigate the effect of nanoparticle volume fraction on the convection heat transfer characteristics and pressure drop of TiO_2–water nanofluids in the horizontal double tube counter-flow heat exchanger. Nanoparticle volume fraction was in the range of 0.002-0.02, and the Reynolds number range varied from 8000 to 51000. They observed that by increasing the Reynolds number or nanoparticle volume fraction, the Nusselt number increased. It was concluded that using nanofluids at high Reynolds numbers had lower benefits that of low Reynolds numbers.

TABLE I
NOMENCLATURE

Cp	specific heat of fluid (J/kgK)
Q	heat rate (W)
A	surface area (m^2)
m	mass flow rate (kg/s)
T	temperature (°C)
Nu	average Nusselt Number
Re	Reynolds Number
Pr	Prandtl Number
f	friction coefficient
v	velocity(m/s)
h	convection heat transfer coefficient (W/m^2K)
k	thermal conductivity (W/mK)
Dh	hydraulic diameter (m)
ΔP	pressure drop (Pa)
L	length of test pipe (m)
U	overall heat transfer coefficient (W/m^2K)
Greek symbols	
ρ	density (kg/m^3)
ϕ	volume fraction
μ	dynamic viscosity (kg/ms)
Subscripts	
nf	nanofluid
p	particle
w	water
ave	average
i	inner
o	outer
h	hot
c	cold
ln	logarithmic

Farajollahi et al. [16] examined γAl_2O_3-water and TiO_2-water nanofluids in a shell and tube heat exchanger under turbulent flow condition. They investigated the effects of Peclet number, particle volume concentration of suspended nanoparticles and particle type on heat characteristics. The maximum volume fractions of nanoparticles were 2 vol. % for Al_2O_3 and 0.75 vol. % for TiO_2. They obtained optimum nanoparticle concentration as 0.3 vol. % for TiO_2. Nanofluids had superior enhancement of heat transfer characteristics than base fluid. They explained that nanofluids had significant enhancement of heat transfer characteristics. Some physical properties of the Al_2O_3–water nanofluids were measured in this study. Results were compared with the conventional theories.

The aim of the present experimental work is to investigate the pressure drop and convective heat transfer of Al_2O_3–water nanofluid with two different particle volume concentrations in a concentric tube heat exchanger with and without coiled wire turbulators under turbulent regime conditions.

II. EXPERIMENTAL DESIGN AND PROCEDURE

A. Preparation of Nanofluids

Preparation of nanofluids is quite important since the stability of nanofluids depends on the preparation method. Nanofluid preparation is not only suspending nanoparticles into a base fluid. There are some techniques for preparing stable nanofluids. The Al_2O_3 nanoparticles having smaller average diameter than 50 nm have been supplied from Sigma-Aldrich Corporation. Al_2O_3-water nanofluids with two different volumetric concentrations of 0.8 and 1.6 % have been prepared by using two step method in which the particles are suspended into the base fluid.

The properties and the behaviour of a suspension depend on the base liquid, suspended particle type, shape, size and positions of the particles in the liquid. When the literature is investigated, de-ionised-water and distilled water have been used as the base fluid in numerous studies. Distilled water has been used as the base fluid in this study as the de-ionized water has deformation property to the aluminium test pipe [17].

The nanofluids were prepared with and without sodium dodecylbenzenesulfonte (SDBS) and cetyltrimethylammonium bromide (CTAB). Since the distilled water without any dispersant was the most stable and to avoid any effect of the dispersant on the heat transfer performance of the nanofluids, the distilled water was chosen as the base liquid. Therefore, no heat transfer experiment was conducted with the nanoparticles having SDBS and CTAB dispersants. Nanoparticles were dispersed into the distilled water with different concentrations of 0.8 and 1.6%, the mixture was sonicated continuously for 20-24 hours in an ultrasonic bath, which had a power of 600 peak/ 300 Watt and 28 kHz frequency. No sedimentation was observed for the produced nanofluids less than 2-4% for the first 14 days.

B. Thermo-Physical Properties

Thermal conductivity was measured by KD2 Pro thermal property meter (Decagon Devices, Inc.) dependent upon temperature. Thermal conductivity measurements data were taken with 5 °C intervals. The viscosity was measured by using a viscometer of Brookfield Inc. (DV-I Prime) from 5 °C to 30 °C. Thermodynamic properties are shown for 20°C at Table II.

The density (ρ_{nf}) of the nanofluid was obtained as mentioned below, and all the mass measurements were carried out by an AND sensitive balance. The mass of the nanoparticles was first determined for total nanofluid volume at a defined volume fraction [18].

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

The specific heat is calculated by using following equation proposed by [19]:

$$\rho(Cp)_{nf} = \phi(\rho Cp)_p + (1-\phi)(\rho Cp)_w \tag{2}$$

TABLE II
THERMODYNAMIC PROPERTIES FOR 20 °C

	ρ (kg/m³)	Cp (J/kgK)	k (W/mK)	μ (kg/ms)	Pr
Water	998	4182	0.598	0.001002	7.01
0.80%	1022.02	4075.46	0.613	0.00126	8.459
1.60%	1040.03	3998.78	0.627	0.00158	10.256

C. Experimental Set-up

The experimental system designed for this study is presented schematically in Fig. 1. It consists of a flow loop, heating unit, cooling section and measuring unit. The flow loop includes a pump, a reservoir, a collection tank and a test section. The test section consists of aluminium test pipe, thermocouples, coil heaters, a differential pressure transmitter (Huba Control Inc.) and isolation. The inner diameter of the aluminium test pipe with a length of 770 mm is 11.7 mm and the thickness is 1.5 mm. The test pipe has been heated by coiled heaters, which provided a constant heat flux boundary condition. The heater output power is 2000 W at 187 V, and the measurement current is 10.7 A. The electrical power input to the heater is controlled by a variac transformer to obtain constant heat flux along the test tube.

Fig. 1 Experimental system

The steady state inlet and outlet bulk temperatures of the nanofluids, and the surface temperatures of the pipe at seven stations were measured by copper-constantan thermocouples with a 0.25 mm inner diameter. The copper-constantan thermocouples were calibrated in a thermostat (Polyscience) within ±0.1 ºC deviation before using them in the experiments. The readings of the thermocouples were recorded using a computer via a data acquisition card (ADVANTEC, HG818 and 789D multiplexer), and the average of these readings was taken. The temperature of the pipe surface was taken as the average of the readings from seven thermocouples. In the experiments, 50-60 minutes elapsed in order to reach steady state. The Reynolds number of the Al_2O_3-water nanofluid ranging from 4000 to 20000 was based on the bulk mean properties and the pipe inner diameter. Whenever the thermal equilibrium was reached, the measured data was recorded.

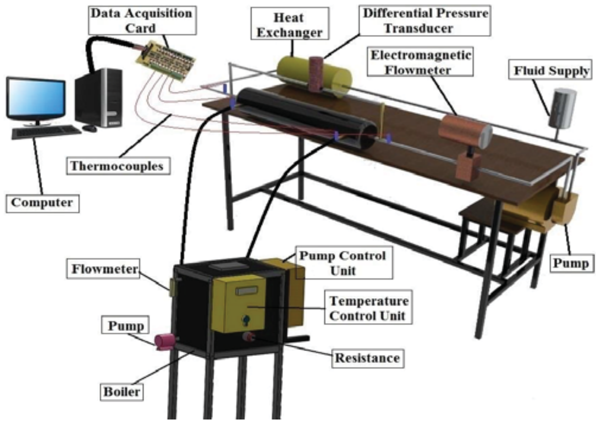

Fig. 2 Schematic diagram of the experimental system

Counter flow coaxial double-tube heat exchanger located in the test section is the most important part of the test unit, and it consists of two tested tubes. The inner pipe through which nanofluid flows has been made from aluminium. The inner diameter of aluminium tube is 12 mm and wall thickness is 2 mm. The outer tube having 33 mm inner diameter is made of polyethylene. The total length of test section is 1300 mm. Two T-type thermocouples with 0.25 mm diameter are mounted onto both ends of aluminium pipe while two pieces of them are mounted onto both ends of polyethylene pipe in order to determine the input and output temperature of nanofluid and hot water, respectively. The test section is thermally isolated to reduce the heat transfer losses.

The temperature of the nanofluid increases due to the heat transfer from hot water. For that reason, the temperature of the hot water decreases. Integrated water bath section is used in order to set the temperature of the nanofluid to the inlet value and to keep it at constant temperature. Integrated water bath section has a heater, a volumetrical circulation pump, a flow meter and a PID control unit. Water tank has a volume of 95 L (51x36x52 cm) and 3 mm wall thickness. The outside of the water tank is covered with an insulating material to ensure the stability of the water temperature by preventing heat transfer. The hot water in the bath is heated by electrical resistance, and the temperature is kept constant at 70 ºC by the aid of the PID temperature control unit. Differential pressure transducer is used to measure the pressure drop along the test section. Hot water flows between the inner and the outer tubes. Nanofluid motion is provided with the volumetric pump, and the flow rate of nanofluid is measured by an electromagnetic flowmeter supplied from EuroMag Corporation.

Two turbulators which have 25 mm and 39 mm pitches and the length of 1300 mm have been used in the concentric heat exchanger in order to investigate the effect on heat transfer and pressure drop characteristics (Figs. 3 and 4). Turbulators can be added to the test area with the help of removable elements on both ends of the concentric tube heat exchanger.

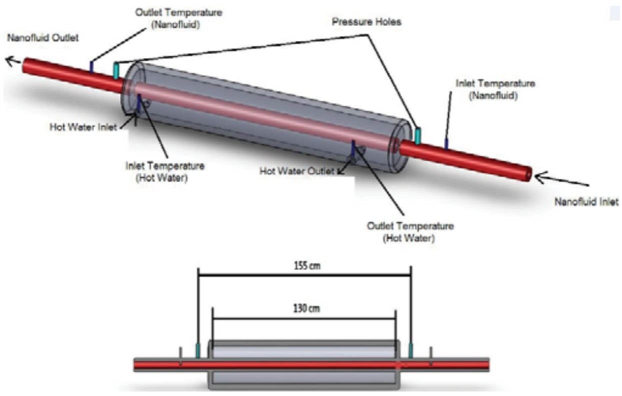

Fig. 3 (a) Schematic of concentric heat exchanger

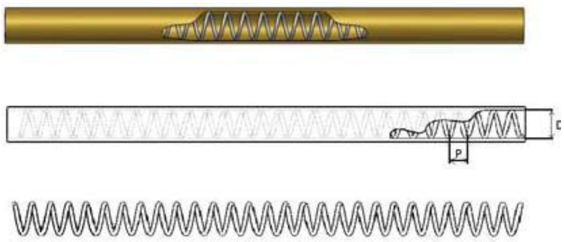

Fig. 3 (b) Schematic of details of coiled wire inserts

D. Data Reduction

The surface of the outer tube of the heat exchanger is covered with insulating material with the aim of preventing heat losses to the environment. By neglecting the radiation heat losses and assuming that the test section is well insulated the heat transfer taken by the nanofluid is equal to the heat transfer taken by hot water.

$$Q_{nf} = Cp_{nf} m_{nf} (T_{out,nf} - T_{in,nf}) \tag{3}$$

$$Q_w = Cp_w m_w (T_{out,w} - T_{in,w}) \tag{4}$$

The average heat transfer rate is defined as given in (5):

$$Q_{ave} = \frac{Q_w + Q_{nf}}{2} \tag{5}$$

The overall heat transfer coefficient of U is calculated from;

$$Q_{ave} = UA_i \Delta T_{lm} \tag{6}$$

Heat transfer surface area given in (6) is calculated from (7) and logarithmic temperature difference is calculated from (10):

$$A_i = \Pi D_i L \tag{7}$$

$$\Delta T_1 = T_{h,i} - T_{c,o} \tag{8}$$

$$\Delta T_2 = T_{h,o} - T_{c,i} \tag{9}$$

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\frac{\Delta T_2}{\Delta T_1}} \tag{10}$$

Gnielinski [20] correlation given in (11) for Nusselt number and friction factor given in (12) are used in order to validate the experimental facility. After calculating the Nusselt number, it is written in (12) to calculate the hot water convection coefficient of ho. D_h given in (12) is hydraulic diameter.

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

$$f = [1.58\ln(\mathrm{Re}) - 3.82]^{-2} \tag{12}$$

$$h_0 = \frac{Nuk}{D_h} \tag{13}$$

Because of inner tube is made from 2 mm thick aluminium materials, the conduction heat transfer is neglected, and by writing the overall heat transfer coefficient of U and hot water convection coefficient of h_o in (14), heat transfer coefficient of the nanofluid of hi is calculated as;

$$h_i = \left[\frac{1}{U} + \frac{1}{h_0}\right]^{-1} \tag{14}$$

The Nusselt number of nanofluid is calculated from (15):

$$Nu = \frac{h_i D_h}{k} \tag{15}$$

Darcy friction factor calculated by using the measured pressure drop is defined as;

$$\Delta P = f\frac{L}{D}(\frac{1}{2}\rho v^2) \tag{16}$$

In this study, Al_2O_3 nanoparticles were dispersed into the water by 0.8 vol. % and 1.6 vol. %, the effect of the particle volumetric concentration, Reynolds number and turbulators on heat transfer and pressure drop characteristics. Total heat transfer enhancement of nanofluid is given in (17):

$$THTE = (\frac{Nu_{nf}}{Nu_w})(\frac{f_w}{f_{nf}})^{\frac{1}{3}} \tag{17}$$

E. Experimental Uncertainties

Kline and McKlintock equation [21] is used to calculate the uncertainty in the Nusselt number, friction factor and Reynolds number. The results of the uncertainty analysis are shown in Table III.

$$w_R = \left[\left(\frac{\partial R}{\partial X_1} w_1\right)^2 + \left(\frac{\partial R}{\partial X_2} w_2\right)^2 + \ldots\ldots + \left(\frac{\partial R}{\partial X_n} w_n\right)^2 \right]^{(1/2)} \quad (18)$$

TABLE III
THE UNCERTAINTY RESULTS

	Re	f	U
Uncertainty (%)	8.74	9.05	8.38

III. RESULTS AND DISCUSSIONS

A. Results of Heat Transfer

For verifying the accuracy of the experimental set-up used in this study, the results obtained for pure water were compared with the data obtained from well-known correlations for turbulent heat transfer. Colburn correlation given by (19), Dittus-Boelter correlation given by (20) [24] and Gnielinski [25] correlation given by (21) were used for Nusselt number comparison. As for the friction factor, the correlations of Blasius [22] and Pethukov [23] given by (22) and (23) were used:

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3} \quad \text{(Colburn)} \quad (19)$$

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \quad \text{(Dittus-Boelter)} \quad (20)$$

$$Nu = \frac{\left(\frac{f}{8}\right)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \quad \text{(Gnielinski)(21)}$$

$$f = 0.316\mathrm{Re}^{-0,25} \quad \mathrm{Re} \le 20000 \quad \text{(Blasius)} \quad (22)$$

$$f = [0.79\ln(\mathrm{Re}) - 1.64]^{-2} \quad \text{(Petukhov)} \quad (23)$$

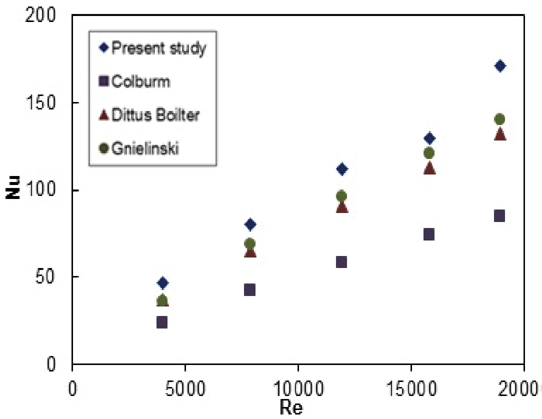

Fig. 4 Comparison of the Nusselt number of pure water with the well-known correlations

Fig. 4 shows the comparison between the results of this study and the well-known correlations. As shown in Fig. 4, the results with respect to the Nusselt number are in good agreement. The experimental results of the friction factor are higher than the values calculated by Petukhov and Blasius correlations. The experimental results at high Reynolds numbers are closer to the results obtained by correlations Fig. 5. The reason is that the pressure difference at low Reynolds number cannot be read very accurately.

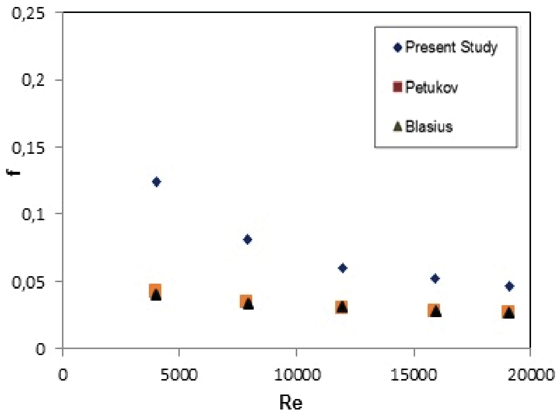

Fig. 5 Comparison of the friction factor of pure water with the well-known correlations

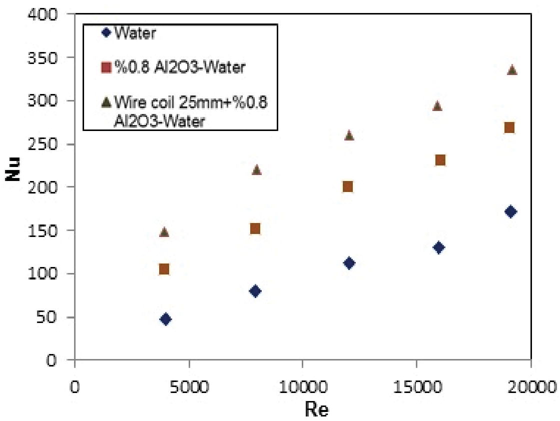

Fig. 6 Nusselt number variations for Al_2O_3-water nanofluid with 0.8 vol. % and without turbulators

Nusselt number of Al_2O_3-water nanofluid with 0.8 vol. % increases much more than that of pure water with increasing Reynolds number as seen from Fig. 6. The Nusselt number of nanofluids with 0.8 vol. % and 1.6 vol. % is 125.23% and 271.91% and greater than that of the base fluid, respectively. The Nusselt number of the nanofluids with 0.8 vol. % and 1.6 vol. % is 57.55% and 111.19% greater than that of the base fluid, respectively. The Nusselt number increases much more when the turbulators are used. It is seen that the addition of the nano sized particles into the pure water caused a remarkable increase in heat transfer. The reason of this behaviour is that thermal conductivity increases by the addition of the nano sized

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particles and the nano sized particles makes Brownian motion in the base fluid.

Average heat transfer coefficient variations versus the Reynolds number is shown in Fig. 7. It is evident from the figure that inserting a ring turbulator into the test pipe increases the average heat transfer coefficient. This is because the turbulence intensity become greater by the presence of the ring turbulator. Furthermore, the average heat transfer coefficient of U increases by increasing Reynolds number. The average heat transfer coefficient of 25 mm coil 8.64% higher at Re=4000 and 2.81% higher at Re=20000 than that of the case without turbulator.

Fig. 8 indicates that using Al_2O_3-water nanofluid with 0.8 vol. % and without turbulator has higher Nusselt number value and average heat transfer coefficient value than that of the case with turbulators in pure water. Al_2O_3-water nanofluid with 0.8 vol. % has higher increase in the Nusselt number approximately as 12% than that of the wire coil with 25mm pitch approximately as 48.50%. The reason of this behaviour is that the thermal conductivity increase achieved by the nanoparticles is more influential on heat transfer than that of the turbulence intensity increase caused by the turbulators used in the pure water.

The Nusselt number and average heat transfer coefficient of Al_2O_3-water nanofluid with 1.6 vol. % has slight increase as compared to that of the pure water as seen in Figs. 9 and 10. Al_2O_3-water nanofluid with 1.6 vol. % and turbulator has higher increase in the Nusselt number as 75.48% than that of the nanofluid with 1.6 vol. % and without turbulator. The reason of this behaviour is that the combined effect of the thermal conductivity increases and the turbulent intensity increase is more dominant than that of the only thermal conductivity increase.

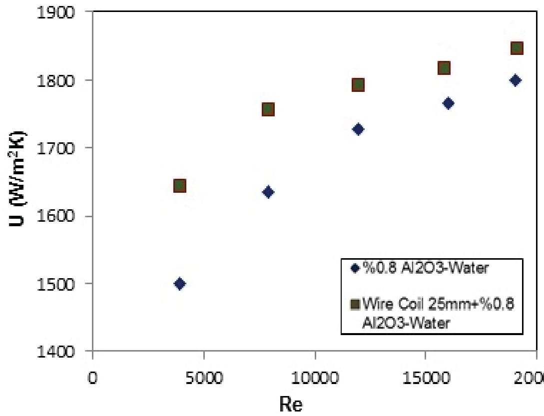

Fig. 7 Average heat transfer coefficient variations for Al_2O_3-water nanofluid with 0.8 vol. % with and without turbulators

In Fig. 10, the average heat transfer coefficient changes with the Reynolds number is presented. It is obvious from the figure that inserting a ring turbulator into the test pipe increases the average heat transfer coefficient. This is because the turbulence intensity become greater by the presence of the ring turbulator. Furthermore, the average heat transfer coefficient of U increases by increasing Reynolds number. The average heat transfer coefficient of 25 mm coil 5.3% higher at Re=4000 and 2.8% higher at Re=20000 than that of the case without turbulator.

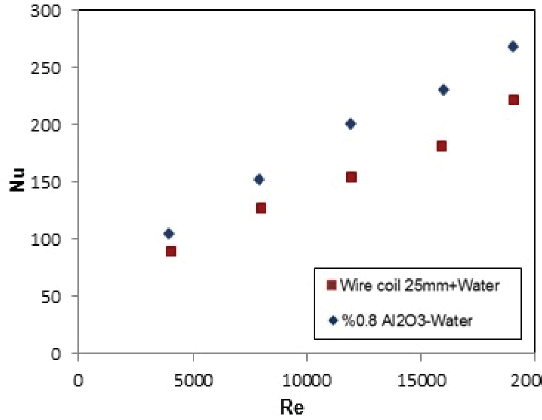

Fig. 8 Comparison of the Nusselt number for Al_2O_3-water nanofluid with 0.8 vol % and wire coils

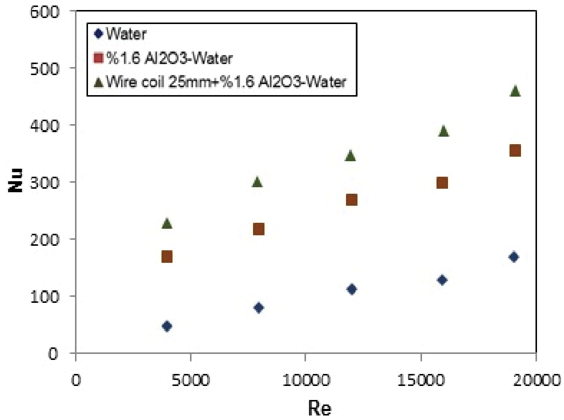

Fig. 9 Nusselt number variations for Al_2O_3-water nanofluid with 1.6 vol. % with or without turbulators

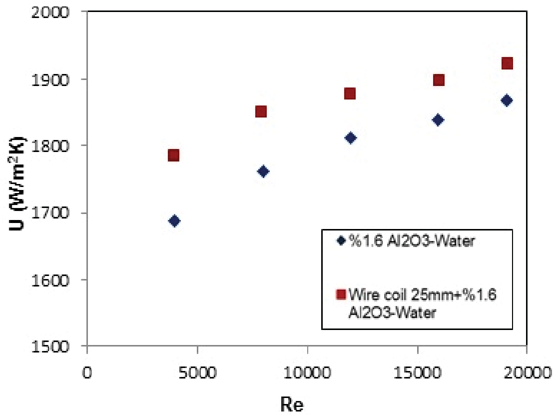

Fig. 10 Average heat transfer coefficient variations for Al_2O_3-water nanofluid with 1.6 vol. % with or without turbulators

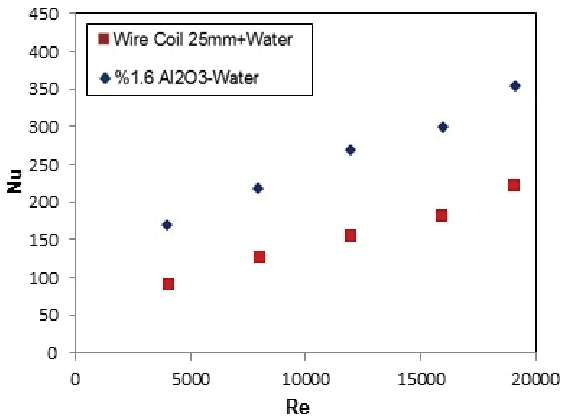

Fig. 11 Comparisons Nusselt number enhancement of Al_2O_3-water nanofluid with 1.6 vol. % with wire coils in water

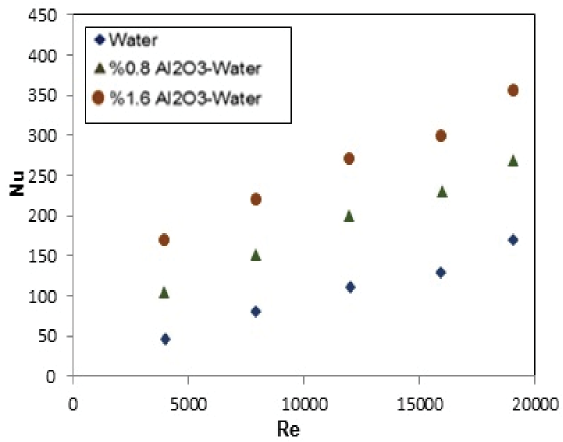

Fig. 12 Nusselt number variations for Al_2O_3-water nanofluids with different volume concentrations as a function of Reynolds number

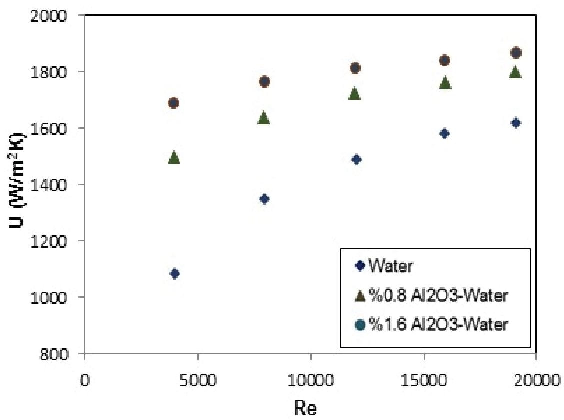

Fig. 13 Average heat transfer coefficient variations for Al_2O_3-water nanofluids at different volume concentrations as a function of Reynolds number

It is evident from Fig. 11 that Al_2O_3-water nanofluid with 1.6 vol. % and without turbulator has higher Nusselt number value and average heat transfer coefficient value than that of the case with turbulators in pure water. Al_2O_3-water nanofluid with 1.6 vol. % has higher increase in the Nusselt number approximately as 23-26% than that of the wire coil with 25mm pitch.

In Figs. 12 and 13, the effect of the particle volume concentration on heat transfer performance of the used nanofluids. The Nusselt number increases with increasing Reynolds number as shown from Fig. 12. An increase in the particle volume fraction causes heat transfer to increase. By the increase of the particle volume concentration, the increase in the thermal conductivity continues incrementally. The highest Nusselt number value is obtained for the case without turbulator is obtained at Re=20000 and 1.6 vol. %. The average heat transfer coefficient of U in Fig. 13 exhibits a similar trend to that of the Nusselt number variations presented in Fig. 12. It can be concluded from the figure that higher volume concentrations provide higher average heat transfer coefficients. The highest value of U is achieved at Re=20000 and 1.6 vol. %.

B. Results of Pressure Drop

In Fig 14, the friction factor variations versus the Reynolds number is presented for different volume fraction values. It can be concluded from the figure that adding nano-sized particles into the base fluid causes a slight increase in the friction factor. This is because the viscosity of the fluid increases by increasing particle volume fraction. Therefore, it also can be concluded that higher particle volume fractions cause higher pressure drops, and thus pumping power. Furthermore, the friction factor decreases with increasing Reynolds number.

In Fig. 15, the friction factor variations of the case with turbulators for both the pure water and the nanofluids are presented versus the Reynolds number. A remarkable increase occurs in the fraction factor by the insertion of the 25 mm wire coil in comparison to the smooth tube. The reason of this behaviour is because the presence of the wire coil exhibits more flow resistance than that of the pure water. Due to the relatively blocked, higher pressure drop occurs than that of the smooth tube. The highest friction factor value is obtained at Re=4000 and 1.6 vol. %. The combined effect of the increasing flow resistance and the fluid viscosity causes higher pressure drop values.

C. Results of Total Heat Transfer Enhancement

In heat transfer enhancement studies, it is important to evaluate the total effect of both heat transfer and the friction factor. According to the general heat transfer knowledge, it can be said that there is a net energy gain for the cases in which heat transfer enhancement is greater than unity. Therefore, this case is useful with respect to both heat transfer and the friction factor [12]. In Figs. 16 and 17, total heat transfer enhancement of the all studied cases is presented versus the Reynolds number. As seen from Figs. 16 and 17, total heat transfer enhancement decreases with increasing Reynolds number. It is evident from Fig. 16 that total heat transfer enhancement is obtained for the nanofluids with 0.8 vol. % and 1.6 vol. %. This means that using Al_2O_3 nanofluids inside the pure water is appropriate for heat transfer enhancement applications. This is because thermal conductivity increase is more influential than that of the viscosity increase on heat transfer. Higher particle volume fractions provide higher total heat transfer enhancement. Al_2O_3 nanofluids with 1.6 vol. % provide approximately 40% higher heat transfer than that of the 0.8 vol. %.

Fig. 17 represents the total heat transfer enhancement of the nanofluids with turbulator at different Reynolds numbers. Inserting wire coil into the test pipe also provides total heat transfer enhancement. But, lower total heat transfer enhancement is obtained than that of the smooth pipe at the same Reynolds number values. This expected because increasing flow resistance caused by the wire coil results in higher pressure drop than that of the case without turbulator.

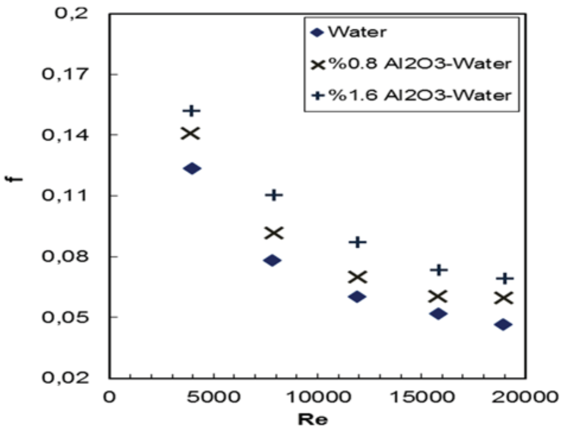

Fig. 14 The effect of adding different volume concentrations of Al_2O_3 particles into the water on pressure drop at different Reynolds numbers

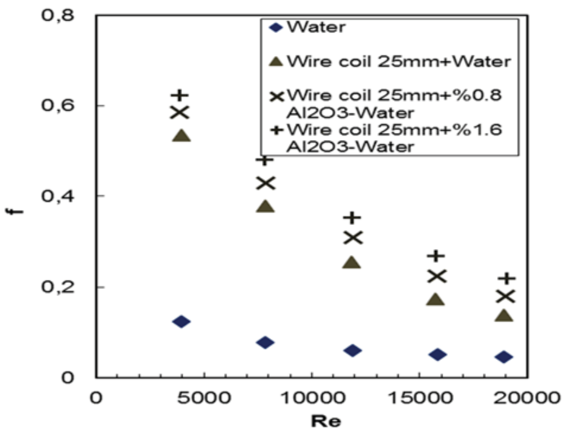

Fig. 15 The effect of using 0.8 vol. % and 1.6 vol. % Al_2O_3nanofluids with wire coil (25mm pitch) on pressure drop at different Reynolds numbers

IV. Conclusions

The use of Al_2O_3-water nanofluids with 0.8 vol. % and 1.6 vol. % provided higher heat transfer coefficient than that of the pure water. The Nusselt number of nanofluids with 0.8 vol. % and 1.6 vol. % increased approximately as 85.66% and 168.26% compared to the pure water, respectively.

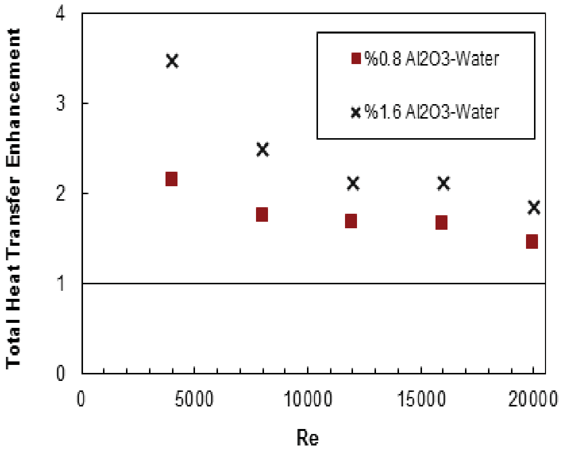

Fig. 16 Total heat transfer enhancement with using 0.8 vol% and 1.6 vol. % Al_2O_3-water nanofluids

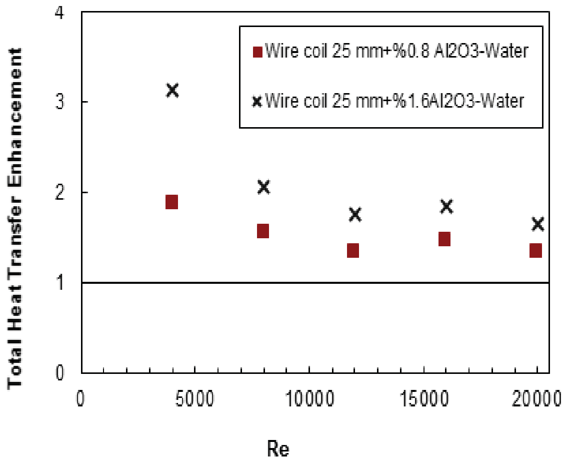

Fig. 17 Total heat transfer enhancement with using 0.8 vol. % and 1.6 vol. % Al_2O_3-water nanofluids with wire coil (25mm pitch)

Nusselt number of nanofluids provided much more increase by using turbulators having different pitches. But, an increase in the pressure drop occurred too. For that reason, total heat transfer enhancement was investigated for all cases. Using turbulators with nanofluids had heat transfer enhanced higher than that of pure water, but using nanofluids without turbulators provided better heat transfer performance.

Nanofluids were improved in order to increase the heat transfer without causing an increase in pressure drop. The pressure drop and friction factor of the nanofluids were nearly equal to that of pure water at the same Reynolds number. But, inserting wire coils with different pitches increased the pressure drop. The rate of the increase in heat transfer and the reduction in the pumping power is remarkably high in heat exchangers because even the smallest increase in heat transfer can be achieved without causing an increase in pumping power will be great benefit for industry. For that reason, nanofluids may be suitable for practical applications. However, some available problems must be solved for using nanofluids in thermal systems. For example, nanoparticles are very expensive. There is also a need for additional measurements in the preparation for maintaining the stability much longer time. After overcoming such difficulties nanofluids may be suitable for most of the heat exchange systems.

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