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ORIGINAL RESEARCH PAPER

Experimental investigation on the heat transfer performance and pressure drop characteristics of γ-Al_2O_3/water nanofluid in a double tube counter flow heat exchanger

B. Raei[1], F. Shahraki[1,*], M. Jamialahmadi[2], S.M. Peyghambarzadeh[3]

[1]*Department of Chemical Engineering, University of Sistan and Baluchestan, Zahedan 98164-161, I. R.Iran*

[2]*Petroleum Engineering Department, Petroleum University of Technology, Ahwaz, I. R. Iran*

[3]*Department of Chemical Engineering, Mahshahr branch, Islamic Azad University, Mahshahr, I. R.Iran*

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ABSTRACT: In this paper, overall heat transfer coefficient and friction factor of water based γ-Al_2O_3 nanofluid in a double tube counter flow heat exchanger have been measured experimentally under turbulent flow condition. For better dispersion of γ-Al_2O_3 nanoparticles in distilled water, magnetic stirrer and ultrasonic vibrator (with a power of 240 kW and frequency of 35 kHz) were implemented. The stabilized γ-Al_2O_3 /water nanofluid have been examined at the concentrations of 0.05 and 0.15 vol. % with variation of flow rates in the range of 7–9 l/min. Nanofluid enters the inner tube of the heat exchanger at different temperatures including 45, 55, and 65 °C. Results demonstrated that increasing the nanofluid flow rate, concentration and inlet temperature can improve the overall heat transfer coefficient and heat transfer rate. Also, the ratio of the overall heat transfer coefficient of nanofluid to that of pure water decreased with increasing the nanofluid flow rate. Meanwhile, the maximum enhancements of the overall heat transfer coefficient and heat transfer rate and friction factor compared with those of base fluid (distilled water) are respectively equal to 19.3%, 10% and 25% which is occurred at the concentration of 0.15 vol. %.

KEYWORDS: *Double tube heat exchanger; Nanofluid; Overall heat transfer coefficient;*

Introduction

Increasing the heat transfer rate in various equipment used in microelectronics, industry, transportation, electronic, and etc. becomes a serious field of study for researchers and engineers. For decades, efforts have been done to enhance the heat transfer rate, reduce heat transfer time, minimize size of heat exchangers, and finally increase energy and fuel efficiencies. Heat transfer properties of conventional fluids are a major obstacle to the development of effective and compact heat transfer equipment. Heat transfer fluids are regularly used in industries including water, ethylene glycol, propylene glycol, and engine oil that generally, have low heat transfer coefficient. One way to increase heat transfer efficiency is improving thermal conductivity of working fluid, which this goal could be achieved by adding nano-sized particles to the base fluid. This type of fluid was called nanofluid for the first time by Choi [1]. Nanofluids as heat transfer fluid have a bright future because of better stability in comparison with microfluids, and increase the thermal conductivity even at low concentrations [2-8]. There are several published studies on the convective and overall heat transfer coefficients of nanofluids and most of them showed that these coefficients enhanced compared with that of the base fluid. Farajollahi et al. [9] conducted an experimental study for heat transfer characteristics of γ-Al_2O_3/water and TiO_2/water nanofluids in a shell and tube heat exchanger under turbulent flow condition. They observed that the overall heat transfer coefficient at a constant Peclet number increases with nanoparticle concentration for both nanofluids. The maximum enhancement of the overall heat transfer coefficients for γ-Al_2O_3/water and TiO_2/water nanofluids compared with that of the base fluids were approximately 20% and 24%, respectively. Pak and Cho [10] studied the heat transfer in turbulent flow regime using Al_2O_3/water and TiO_2/water nanofluids. They concluded that Nusselt number of nanofluid increases with increasing nanoparticles volume fraction and Reynolds number. They also provided the first relationship to calculate nanofluid's heat transfer. Jwo et al.[11] performed an investigation to analyze the effects of concentration, inlet flow temperature, and flow rates on the overall heat transfer coefficient of Al_2O_3/water nanofluid in a multichannel heat exchanger (MCHE). They observed that the overall heat transfer coefficient ratio was higher at higher nanoparticle concentrations and mass flow rates but the effect of temperature on the above mentioned

*Corresponding Author Email: *fshahraki@eng.usb.ac.ir*
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Nomenclature

A	surface area (m^2)	T	temperature (°C)
CNT	Carbon Nano-tube	V	volt
Cp	specific heat (J/kg °C)	vol	volume
D	diameter (m)	W	mass concentration of nanofluids, wt.%
DWCNTs	Double-walled Carbon Nanotubes	x	Axial distance
Exp	experimental		**Greek Symbols**
EG	Etylene Glycol	φ	volume fraction
f	friction factor	ε	roughness (m)
h	hour	μ	viscosity (Pa.s)
ID	Inner diameter(mm)	ρ	density (kg/m^3)
k	thermal conductivity, (W/m.K)	Δ	difference
l	liter		**Subscripts**
L	length (m)	ave	average
Nu	Nusselt number	b	bulk
P	pressure (Pa)	bf	base fluid
PC	personal computer	c	cold
Pe	Peclet number	e	equivalent
Pr	Prandtl number	h	hot
PID	proportional–integral–derivative	i	inner
PVC	Polyvinyl chloride	LMTD	logarithmic mean temperature difference
q	heat transfer rate (kW)	m	mean
Q_h	volume flow rate of hot water (l/min)	nf	nanofluid
Q_c	volume flow rate of cooling water (l/min)	o	outer
Re	Reynolds number	p	particle
TEM	Transmission electron microscopy	w	wall
		x	local

coefficient was reverse. Yu et al. [12] obtained experimental heat transfer data for water/EG (45/55 vol.%) and Al_2O_3 nanoparticles. They observed that at 1 vol.% and 2 vol.% of nanoparticle, the rate of heat transfer increased 57 to 106 %, at Re = 2,000. Wen and Ding [13] studied the heat transfer in laminar flow regime under constant heat flux boundary condition using Al_2O_3/water nanofluid. They have reported that increasing the Reynolds number and concentration of nanoparticles, especially at the entrance region, increases the heat transfer coefficient of nanofluid.
Peyghambarzadeh et al.[14] studied the heat transfer coefficient of Al_2O_3/water nanofluid in a car radiator. They observed the heat transfer enhancement of about 45% compared with pure water. In other work, Peyghambarzadeh et al. [15] used different concentrations of water and ethylene glycol as a base fluid which is conventionally used in the car radiators. They have figured out that the convective heat transfer coefficient of Al_2O_3/ethylene glycol nanofluid showed an increase of about 40% compared with the base fluid in the best conditions. Leong et al. [16] studied the convective heat transfer coefficient and overall heat transfer coefficient of copper nanofluid in a shell and tube heat recovery exchanger. It was observed that about 16.9% and 9.5% enhancements were recorded for ethylene glycol with 1% copper nanoparticles compared with the base fluid, respectively. For 2 vol.% water based copper nanofluid, 33.4% and 10.11% enhancements of convective heat transfer coefficient and overall heat transfer coefficient were recorded compared with the base fluid in laminar flow. Sajadi and Kazemi [16] studied the heat transfer and rheological behavior of TiO_2/water nanofluid at 0.25 vol.% in a circular tubes at 5,000 <Re <30,000. Their results indicated that adding a small amount of nanoparticles to the base fluid considerably increases the heat transfer rate.Also,Nanofluid pressure drop was slightly more than the base fluid. Lotfi et al. [17] demonstrated that the overall heat transfer coefficient could be increased by using multi-walled carbon nanotube (MWCNT)/water nanofluid with low concentration of 0.015 wt% in a horizontal shell and tube heat exchanger instead of water as base fluid. Esfe and Saedodin [18] studied thermal conductivity, dynamic viscosity and Nusselt number of turbulent forced convection of MgO/water nanofluid in a circular straight pipe. The experimental results indicated that the existence of the nanoparticles in the pure water with all considered values of the nanoparticles volume fraction and diameters motivates the rate of heat transfer to increase. Peyghambarzadeh et al.[19] studied the heat transfer performance of the automobile radiator by calculating the overall heat transfer coefficient (U) according to the conventional ε-NTU technique using CuO/water nano-

fluids. In addition, double tube heat exchanger is probably one of the simplest configurations found in applications in which heat is transferred from hot fluid to cold fluid through a separating cylindrical wall. It consists of concentric tubes separated by mechanical closures. They are primarily adapted to high-temperature; high-pressure applications due to their relatively small diameters. Double tube heat exchangers have a simple construction. The amount of heat transfer per section is small which makes the double pipe heat exchangers a suitable heat transfer device in applications where a large heat transfer surface is not required. Some of experimental studies are summarized in Table 1 to investigate the thermal performance of nanofluids in double tube heat exchangers.

Table 1
Summary of researches of double tube heat exchangers using nanofluids.

Author(s)	**Base fluid**	**Nano Particle/mean diameter**	**Volume fraction %**	**Dimension**	**Flow regime, Re**	**Maximum enhancement of heat transfer coefficient**
Esfe et al.[20]	Water	COOH-functionalized DWCNTs	0.01,0.02,0.05, 0.1,0.2&0.4	Inner tube:ID:7.05 mm Outer tube:ID:37.9 mm L:110 cm	Turbulent	32% at φ=0.4%,
Darzi et al.[21]	Water	Al_2O_3 /20nm	0.25,0.5&1	Inner tube:ID:8.1 mm Outer tube:ID:150 mm L:220 cm	5,000-20,000	20% in Nu number at φ=1
Duangthongsuk et al.[22]	Water	TiO_2 /21nm	0.2,0.6,1,1.5&2	Inner tube:ID:8.13 mm Outer tube:ID:27.8 mm L:150 cm	Turbulent	26% at φ =1%
Zamzamian et al.[23]	EG	Al_2O_3 /20 nm CuO /20 nm	0.1, 0.5&1(wt %) 0.1, 0.3, 0.5, 0.7&1(wt %)	Inner tube:ID:12 mm Outer tube:ID:50.8 mm L:70 cm	Turbulent	26.2% at φ =1% 37.2% at φ=1%
Arani et al.[24]	Water	TiO_2 /30 nm	0.002,0.005,0.01 ,0.015&0.02	Inner tube:ID:8.18 mm Outer tubeID:26.02mm L:128.8 cm	8,000-51,000	82.47% in Nu number at φ=0.02%
Esfe et al.[25]	Water	MgO /40nm	0.0625,0.125 ,0.25,0.5&1	L:111 cm	Turbulent	35.93% at φ=1%
Chun et al.[26]	Transformer oil	Al_2O_3 /7,27,43nm	*0.25&0.5*	Inner tube:ID:6.35 mm Outer tube:ID:12.7 mm L:500cm	Laminar	25% at φ=0.5%
Aghayari et al.[27]	Water	γ-Al_2O_3 /20nm	0.1,0.2&0.3	Inner tube:ID:6 mm Outer tube:ID:14 mm L:120 cm	15,000-28,000	12% at φ=0.3%
Khalifa et al[28]	Water	γ-Al_2O_3 /10nm	0.25,0.5,0.75&1	Inner tube:ID:20 mm Outer tube:ID:50 mm L:76 cm	Turbulent	22.8% at φ=1%
Madhesh et al.[29]	EG	Ag/10-65nm	0.1,0.2,0.5,1&2	Inner tube:ID:4.4mm Outer tube:ID:10.7mm L:180 cm	Turbulent	54.3% at φ=1%
Sarafraz et al.[30]	EG (50%) Water(50%)	Ag/40-50nm	0.1,0.5&1	Inner tube:ID:6.35mm Outer tube:ID:12.7mm L:240 cm	Laminar & Turbulent	67% at φ=1%
Khedkar et al.[31]	Water	TiO_2	2&3	Inner tube:ID:8mm Outer tube:ID:16mm L:100 cm	Laminar	14%in Nu number at φ=3%

According to our knowledge, the overall heat transfer coefficient of γ-Al_2O_3/water nanofluid in a double tube heat exchanger has not been reported experimentally yet.

In this research,an experimental study has been performed to evaluate effective parameters on the overall heat transfer coefficient and pressure drop characteristics of stabilized γ-Al_2O_3/ water nanofluid in fully-developed turbulent flow regime using a double tube heat exchanger. Experiments performed at different concentrations of the nanofluid, several operating temperatures, and various nanofluid flow levels.

Experimental work

Nanofluid preparation and stabilization

In this study, aluminium(III) oxide (γ-Al_2O_3) nanoparticle of approximately 20 nm in diameter and +99% purity has been used.

A transmission electron microscope (TEM) was used to approximate the size of the primary nanoparticles. As shown in Figure 1, it is clear that the primary shape of nanoparticles is approximately spherical.

This method is commonly used by a wide range of researchers [32, 22, 33].

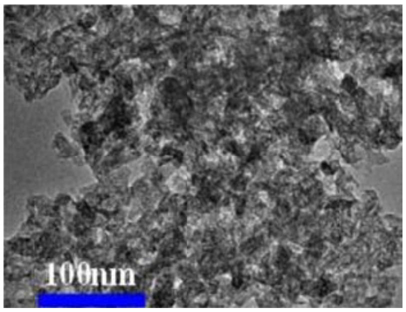

Fig.1. Image of TEM nanoparticles used in this study

The physical properties and some characteristics of the nanoparticle are listed in Table 2.

Preparation of nanofluids is the first key step in experimental studies using nanoparticles to improve the thermal efficiency of fluids. Two methods including single-step and two-step existed for nanofluid production. As the nanoparticles are commercially available, many researchers used two-step procedure for preparing nanofluids.

Table 2

Characteristics and physical properties of γ -Al_2O_3 nanoparticle.

Aluminum Oxide (gamma)	Nanoparticle
20	Average particle size (nm)
+99%	Purity
3890	Density (kg/m^3)
White	Color
Nearly spherical	Morphology
>138	Specific area (m^2/g)
880	Specific heat (J/kg K)
46	Thermal conductivity (W/m K)

In the two-step method, providing stable nanofluid is a challenge.

Various methods such as changing in pH of the nanofluid, addition of surface activators (surfactants), and ultrasonic vibration were used to achieve stable nanofluids. In this study, nanofluid with the concentrations of 0.05 and 0.15 of vol.% were prepared.

A certain amount of γ-Al_2O_3 nanoparticle is weighed (accurate to three decimal places), and added to distilled water as a based fluid.

After half an hour of mixing with magnetic stirrer, fluid was placed in an ultrasonic vibrator (BANDELIN Company- with a power of 240 kW and frequency of 35 kHz) for 3 h. Figure 2 shows the nanofluid stability after 24 h. It should be noted that no surfactant was used during the preparation of nanofluid due to the changes of its thermo-physical properties. In addition; sedimentation of nanoparticles has less importance in the turbulent flow regime because of the higher imposed shear which breaks down the possible agglomerated particles.

Therefore, turbulent flow regime helps to produce stable nanofluid_in the experiment. This point of view was also presented by Nasiri et al. [34].

Fig.2. Stability photograph of γ -Al_2O_3 nanofluids after 24 h

Experimental setup and procedure

Schematic view and realistic photo of the experimental setup are shown in Figures 3a and 3b.

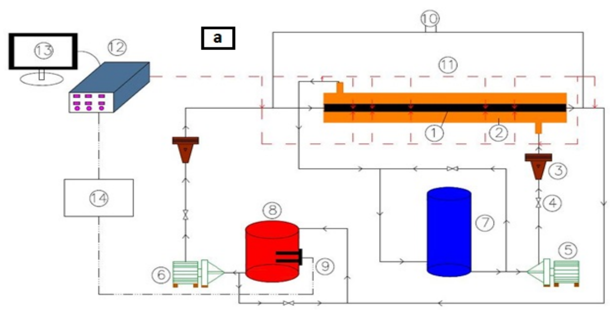

Fig.3a. Schematics of the experimental setup: (1) inner tube, (2) annulus, (3) rotameter, (4) control valve, (5) cold water pump, (6) hot water pump, (7) cold reservoir tank, (8) hot reservoir tank, (9) electrical heater, (10) differential pressure gauge, (11) thermocouples, (12) data logger, (13) PC, (14) PID controller

Fig. 3b. Photograph of the experimental setup

The test loop consists of two reservoir tanks, a heater, a digital thermostat with PID controller, two flow meters, one flange, temperature sensors, two centrifugal pumps, data logger, control box, a U-shaped manometer, metal valves for opening and closing of flow passes, and a personal computer. The test section includes a double tube heat exchanger that composed of two concentric tubes.

The hot fluid (nanofluid) passes through the inner tube which is made of stainless steel (316 SS) with 12.7 mm inner diameter and 6 mm thickness. The cold fluid (distilled water) passes through the outer tube that surrounds the inner tube and is made of carbon steel with 63.5 mm inner diameter and 6 mm thickness. The total length of the test section is 60 cm. According to the following equation: ($\frac{L_e}{D}$~$4.4Re^{\frac{1}{6}}$) [35] the length of tube needed to create a fully developed turbulent flow is calculated about 30 cm at Reynolds number 40,000 (maximum Reynolds number in this study). Therefore, considering the length of the heat exchanger, 60 cm, it assures that the flow would be developed for all experiments. The outer surface of the test section was thermally insulated by glass wool with 7 cm thickness to minimizing the heat loss to the surrounding.

The nanofluid is placed in a cylindrical 16 l carbon steel reservoir tank (the inner layer is corrosion protected). At the bottom of the tank, an electrical heater with 3 kW power is embedded which is capable of heating the fluid up to the boiling temperature.

This heater is connected to the thermostat with temperature control and a digital display (BR6- FDMP4 models with accuracy of ±0.1 °C) that indicates and controls the temperature of the hot fluid. The required energy is supplied by 220 V electrical heater. After reaching the required temperature, the nanofluid is pumped in to the test section by a centrifugal pump (HAPPY Company with the maximum capacity of 35 l/min, 0.5 hp, and maximum head of 35 m). The fluid flow can be adjusted by a valve on the recycle line or a valve that is mounted before the flow meter.

Then, the distilled water is poured in the cold cylindrical tank which is made of PVC with capacity of 100 l. It should be noted that the cold reservoir tank temperature was always kept constant at a temperature of 6 °C by a mixture of water and ice. Throughout the test, the cooling fluid flow rate was constant at 13 l/min.

After switching the pump on, the cold fluid passes through the valves. Depending on the experimental conditions, co-current or countercurrent flow can be prepared in this setup. The specification of cold fluid pump is exactly the same as the hot fluid pump. For measuring flow rates, two flow meters (Technical Groups Model sp.gr.1.0) with operating temperature range between 0 to 90 °C, and 1.8-18 l/min flow rate were used. Precision of the flow meters is 0.1 l/min. Both flow meters were calibrated by the time taken for a given volume of fluid to be discharged. Four temperature sensors have been used to measure the bulk temperature of the flow at the inlet and outlet of the inner tube and annulus. The accuracy of all temperature sensors is ±0.1 °C. Data logger (TIKA Company and model TM-1202) was also used for recording the temperature data. All temperature measuring devices were calibrated before testing.

Nanofluid physical properties

Addition of nanoparticle to the base fluid changes its density, specific heat, thermal conductivity, and viscosity. For better understanding, Figure 4 depicts the variations of dimensionless physical properties of γ-Al_2O_3 nanofluid, i.e. the ratios of physical properties of the nanofluid to those of pure water as a function of nanoparticle concentration. It is obvious that the addition of small amount of γ-Al_2O_3 nanoparticle can change more or less all the physical properties of the base fluid.

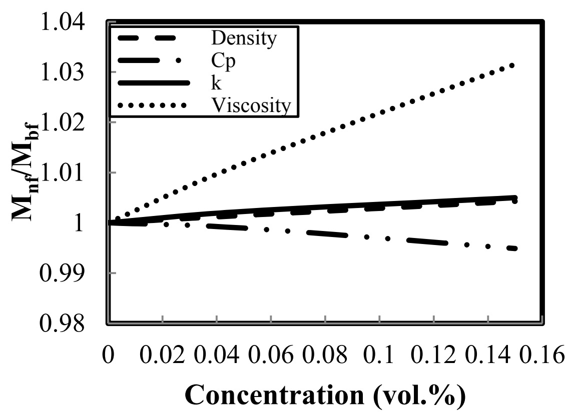

Fig.4. The effect of nanoparticle addition on the physical properties of γ-Al_2O_3 /water nanofluid

Therefore, before the study on the heat transfer performance of the nanofluid,the properties of nanofluid must be known accurately. By assuming that the nanoparticles are well dispersed in the base fluid, the concentration of nanoparticles may be considered uniform throughout the tube.

Although this assumption may not be true in reality because of some physical phenomena such as particle migration, it can be a useful tool to evaluate the physical properties of a nanofluid.

The following correlation proposed by Pak and Cho [10] is used to estimate the nanofluid density:

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

C_{pnf} is the effective specific heat of the nanofluid which can be calculated from Xuan and Roetzel [36] relation:

$$C_{pnf} = \frac{(1-\varphi)\rho_{bf}C_{pbf} + \varphi\rho_p C_{pp}}{\rho_{nf}} \tag{2}$$

So far, various theoretical and experimental studies have been conducted and various correlations have been proposed for the dynamic viscosity and thermal conductivity of nanofluids. However, any general correlation has not been established due to the lack of common understanding on the mechanism of nanofluid.The effective dynamic viscosity and thermal conductivity of nanofluid can be usually calculated by existing formulas that have been obtained for two-phase mixtures, i.e, the well-known Einstein equation [37] for dynamic viscosity, and the Maxwell model [38] for thermal conductivity. Some previous studies showed the aforementioned correlations are not made for nowadays nanofluids. They have weak approaches which can be employed to characterize the nanofluid's viscosity and thermal conductivity. Therefore, in this study, the viscosity and thermal conductivity of the nanofluid have been obtained from Williams et al. [39] that proposed dynamic viscosity and thermal conductivity equations based on limited experimental data for the γ-Al_2O_3-water nanofluid.

$$\mu_{nf} = \mu_{bf}(T)exp[\frac{4.91\varphi}{(0.2092-\varphi)} \tag{3}$$

$$k_{nf} = k_{bf}(T)(1+4.5033\varphi) \tag{4}$$

Data reduction

In this paper, the experimental data was used to calculate the rate of heat transfer, overall heat transfer coefficient, and pressure drop of the γ-Al_2O_3 /water nanofluids. The significance of these parameters in improving the heat transfer rate and thermal performance of the heat exchanger are discussed in the forthcoming sections. The circulating fluid inside the heat exchanger rejects the heat to the cold fluid and absorbs the heat from the hot fluid. The rate of heat transfer can be calculated from the following equations.

$$q_{bf} = \dot{m}_{bf}C_{pbf}\left(T_{bf,out} - T_{bf,in}\right) \tag{5}$$

where,q_{bf},$\dot{m}_{bf}$,C_{pbf},$T_{bf,out}$,$T_{bf,in}$, are the heat transfer rate, mass flow rate, specific heat capacity, and temperature at the outlet and inlet of the base fluid (cold fluid), respectively.

The rate of heat transfer of the hot fluid (nanofluid) was computed using the following equation:

$$q_{nf} = \dot{m}_{nf}C_{pnf}\left(T_{nf,in} - T_{nf,out}\right) \tag{6}$$

where,q_{nf},$\dot{m}_{nf}$,C_{pnf},$T_{nf,in}$,$T_{nf,out}$, are the heat transfer rate, mass flow rate, specific heat capacity, and temperature at the inlet and outlet of the nanofluid (hot fluid), respectively. The average rate of heat transfer calculated as:

$$q_{ave} = \frac{q_{bf} + q_{nf}}{2} \tag{7}$$

The overall heat transfer coefficient was calculated, using the following equation:

$$U_{exp} = \frac{q_{ave}}{A_i.\Delta T_{LMTD}} \tag{8}$$

Where, U_{exp} is the experimental overall heat transfer coefficient, A_i is the surface area of the inner tube, and ΔT_{LMTD} is the logarithmic mean temperature difference. The logarithmic mean temperature difference was obtained as follows:

$$\Delta T_{LMTD} = \frac{\left(T_{nf,in} - T_{bf,out}\right) - \left(T_{nf,out} - T_{bf,in}\right)}{ln\left(\frac{T_{nf,in} - T_{bf,out}}{T_{nf,out} - T_{bf,in}}\right)} \tag{9}$$

where $T_{nf,in}$ and $T_{nf,out}$ are the inflow and outflow temperatures of the nanofluid in the heat exchanger, respectively. $T_{bf,in}$ and $T_{bf,out}$ are the inflow and outflow temperatures of the base fluid in the heat exchanger, respectively.

Experimental friction factor was determined from the measurements of pressure drop along the length of the test section using the following equation;

$$f_{nf} = \frac{2D\Delta P_{nf}}{L\rho_{nf}u_m^2} \tag{10}$$

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, D is the diameter of the tube, ρ_{nf} is the density of the nanofluid, and u_m is the mean velocity of the nanofluid. In all the calculations, the values of the thermo physical properties of the nanofluids were obtained at the average bulk temperature, which is $T_b = (T_{in} + T_{out})/2$.

Uncertainty analysis

The range of the operating variables and their relevant uncertainty in the measurement which were calculated according to Moffat [40] are shown in Table 3.

Table 3
The range of operating conditions and their measurement uncertainty.

Uncertainty	Unit	Range	Condition
±0.1	l/min	7-11	Hot liquid flow rate
±0.1	°C	45-65	Hot liquid inlet temperature
±0.1	l/min	13	Cold liquid flow rate
±0.1	°C	6	Cold liquid inlet temperature
0.83%	-	18,000-40,000	Reynolds number
-	vol.%	0-0.15	Nanoparticle concentration

It was calculated that maximum uncertainty in the measurement of the overall heat transfer coefficient was

9.7% and most of this uncertainty related to the temperatures measurement. Also, uncertainty of experimental friction factor was calculated to be about 4.65%. It must be noted that the estimated uncertainties were based on the manufactures specification and not on a calibration of the instruments.

Results and discussion

Validation of the results

Before conducting systematic experiments on the application of nanofluids in the double tube heat exchanger, the reliability and accuracy of the experimental system were tested using de-ionized water as the working fluid. The obtained experimental data were then compared with the theoretical prediction of the heat transfer relations.

The overall heat transfer coefficient was calculated using heat transfer resistances theory as follows:

$$U_i = \frac{Q_{ave}}{A_i\,\Delta T_{LMTD}} = \frac{1}{\dfrac{1}{h_i} + \dfrac{A_i\,ln\left(\frac{r_o}{r_i}\right)}{2\pi Lk} + \dfrac{A_i}{A_o h_o}} \tag{11}$$

Where U_i is the overall heat transfer coefficient. For double tube heat exchangers without fins, ignoring the terms related to heat exchanger fouling, U_i is defined based on the inner surface of the tube. r_i and r_o are the inner and outer radii, respectively, and k is the thermal conductivity of the tube.

The tube side heat transfer coefficient (h_i) can be calculated by Gnielinski [41] correlation. Also, the annuli (shell) side heat transfer coefficient can be estimated by the correlation of Foust&Christian [42] to yield the value of h_o .Predictions of equation 11 in conjunction with the aforementioned correlations for the calculation of overall heat transfer coefficient U_i are compared with the experimental overall heat transfer coefficients of pure water in Figure 5.

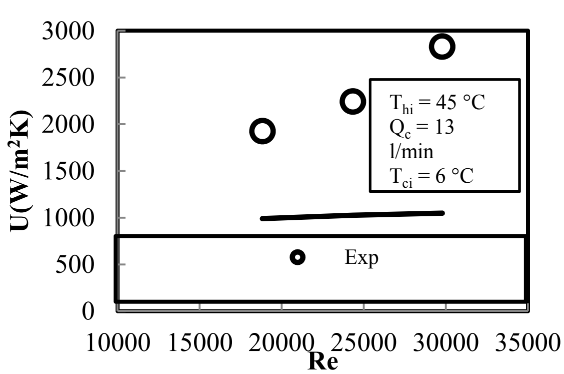

Fig.5. Effect of Reynolds numbers on the experimental overall heat transfer coefficient and the predictions of equation 11

As can be seen, poor agreement exists between the experimental data and the prediction of equation 11.Increasing the Reynolds number enhances the overall heat transfer coefficient. This effect is weakly predicted by equation 11.The results for other water inlet temperature including 55 and 65 °C, which are not reported here, are similar to those of 45 °C. The absolute average errors of the equation 11 in comparison with the experimental data for pure water was about 55%. Mehrabian et al. [43] have studied overall heat transfer characteristics of a double tube heat exchanger. They compared experimental data with predictions of standard correlations, which concluded experimental heat transfer coefficients for straight and smooth tube are significantly higher than the predicted values by standard correlations. The result of current research is compatible with the mentioned achievements. However, it should be noted that basically, the mentioned correlations have developed for a tube with constant heat flux or constant wall temperature boundary conditions, but double tube heat exchanger.

Figure 6 shows the calculated friction factor of distilled water obtained from experiment and the prediction of Colebrook [44] equation at various Reynolds numbers. An absolute average error of the prediction is 8%.Thus; comparison between the results showed a good agreement with Colebrook equation.

Having validated the accuracy of the experimental system, the experimental finding of nanofluids are presented and discussed in subsequent chapter.

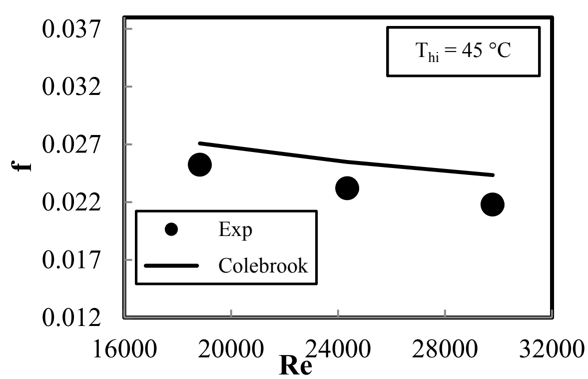

Fig.6. Comparison of the experimental friction factor of distilled water with the prediction of Colebrook equation

Heat transfer to nanofluid

In this study, the overall heat transfer coefficient and heat transfer rates of γ-Al_2O_3 /water nanofluids measured in the double tube heat exchanger. However, nanofluids flow rate, nanoparticle concentration and nanofluid inlet temperature were varied in order to determine the overall heat transfer coefficient and the heat transfer rate.

Effect of nanofluid flow rate

Figure 7 shows the measured overall heat transfer coefficient of the γ-Al_2O_3 /water nanofluid as a function of nanofluid flow rate at different concentration and at a fixed cold fluid flow rate (13 l/min) and cold fluid temperature (6°C).

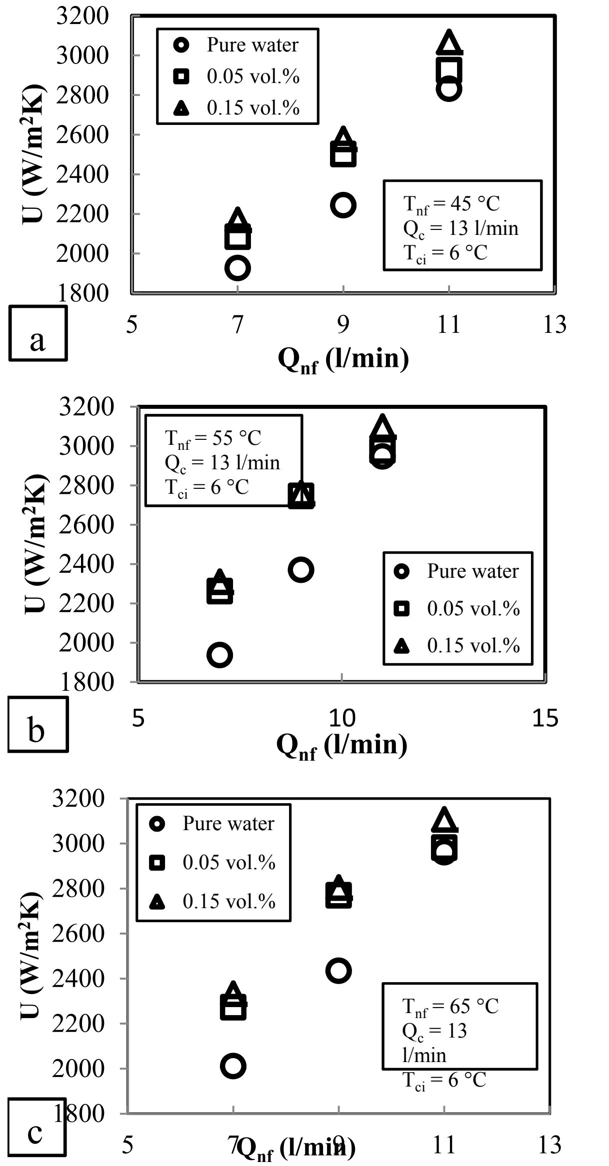

Fig.7. Overall heat transfer coefficient of γ-Al_2O_3/water nanofluid as a function of nanofluid flow rate at different nanofluid concentrations and inlet temperatures of (a) 45 °C, (b) 55 °C, and (C) 65 °C

The results showed that the overall heat transfer coefficient of nanofluids increased significantly with increasing of nanofluids flow rate. The overall heat transfer coefficient of nanofluids at a constant flow rate increased with increasing the nanoparticle concentration in comparison with pure water. Nevertheless, the rate of increasing the overall heat transfer coefficient with increasing concentration is less tangible.

Also, it is known that impact of nanoparticles on the overall heat transfer coefficient at low flow rate (low Reynolds numbers), is more obvious. The results are consistent with the results reported by Sajadi and Kazemi [16] that the rate of the heat transfer coefficient enhancement of nanofluid to that of pure water decreased with increasing the Reynolds number.

The same graph obtained for the heat transfer rate which is demonstrated in Figure 8. As can be seen in Figure 8, heat transfer rate increases when the flow rate and concentration of nanofluids increases. At the concentration of 0.15 vol%, the enhancement of about 10% in the heat transfer rate observed compared with the base fluid. Increasing the heat transfer rate is the most important reason for using nanofluids as heat transfer medium.

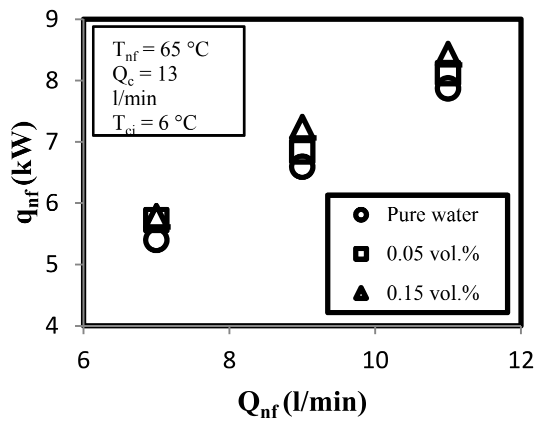

Fig.8. Heat transfer rate of γ-Al_2O_3/water nanofluid as a function of nanofluid flowrate at different concentrations

Effect of inlet temperature of nanofluids

In order to find out the effect of temperature on the overall heat transfer coefficient and the heat transfer rate, nanofluids tested at three different inlet temperatures including 45, 55, and 65 °C. The variation of the overall heat transfer coefficient with nanofluid inlet temperature at different nanofluid flow rates and at a fixed cold fluid flow rate (13 l/min) and cold fluid temperature (6°C) and nanoparticle concentration (0.15 vol. %) is shown in Figure 9.

As shown in Figure 9, it is visible that with increase in nanofluid temperature, the overall heat transfer coefficient increases. This increase is more tangible at lower flow rate. These improvements in the overall heat transfer coefficient of nanofluids with increased fluid temperature can be created due to two factors. First: enhancement of thermal conductivity of nanofluid with temperature. Second: decrease of viscosity of the base fluid with temperature increase. As a result, Brownian motion of nanoparticles inside the fluid increases, consequently convection-like effects are remarkably increased which lead to increase in overall heat transfer coefficient. "In this study, all tests have been done at constant cold fluid temperature and cold flow rate.

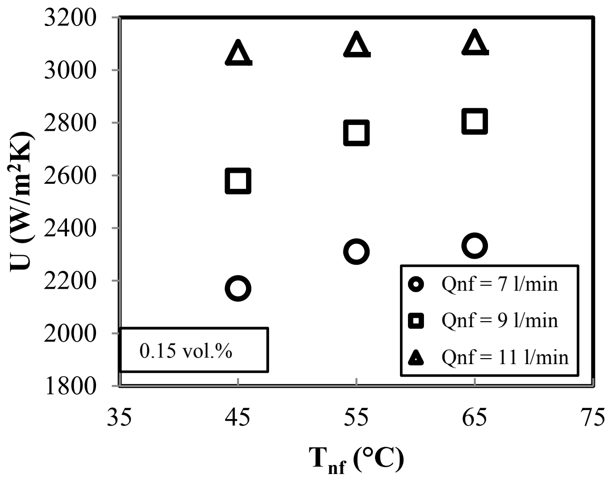

Fig. 9. Overall heat transfer coefficient of γ-Al_2O_3/water nano fluid as a function of inlet temperatures at different nanofluid flowrates (0.15 vol. %)

Therefore, increase in hot fluid input temperature the temperature difference between fluids along the heat exchanger will increase, this will cause increase convective heat coefficient in both fluids.

Therefore, overall heat transfer coefficient as a function of heat transfer coefficient of both sides of heat exchanger will increase. On the other hand, increase in volume flow rate in hot fluid side, which causes increase in convective heat transfer coefficient in tube side as well as effective heat transfer to the wall.

This phenomenon causes increase in wall temperature; therefore, significant increase in cold fluid temperature will be observed.

In summary, overall heat transfer coefficient increases by increase in hot fluid flow rate." The results are consistent with the results of Aghayari et al. [45]. They also reported that the overall heat transfer coefficient of Al_2O_3/water nanofluid in double tube heat exchanger increases with increase in fluid inlet temperature. Although Varmahmoodi et al. [46] studied the effect of temperature on the overall heat transfer coefficient of Fe_2O_3/water nanofluid in an air-finned heat exchanger and reported that an increase in temperature reduces the overall heat transfer coefficient.

In addition, Zamzamian et al. [23] have reported that with increase in temperature, heat transfer coefficient increases in double tube heat exchanger. But, Duangthongsuk & Wongwises [22] vice versa have reported that with increase in inlet temperature,heat transfer coefficient decreases in double tube heat exchanger.

Figure 10 shows the effect of temperature on heat transfer rate at 0.15 vol. % of nanofluids.

The results show that the heat transfer rate significantly increases with increasing temperature of nanofluids. When the temperature of nanofluid increases from 45 to 65 °C, the heat transfer rate raises an average of about 53%.

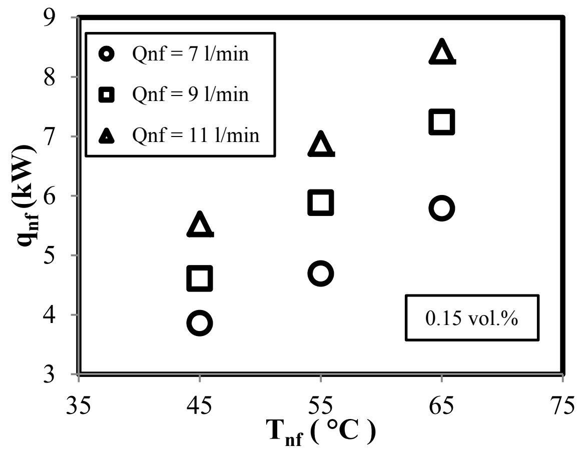

Fig.10. Heat transfer rate of γ-Al_2O_3/water nanofluid as a function of inlet temperatures at different nanofluid flow rate (vol. 0.15%)

Flow properties of nanofluids

Variation of isothermal friction factor versus Reynolds number at various concentrations of nanoparticles is illustrated in Figure 11.

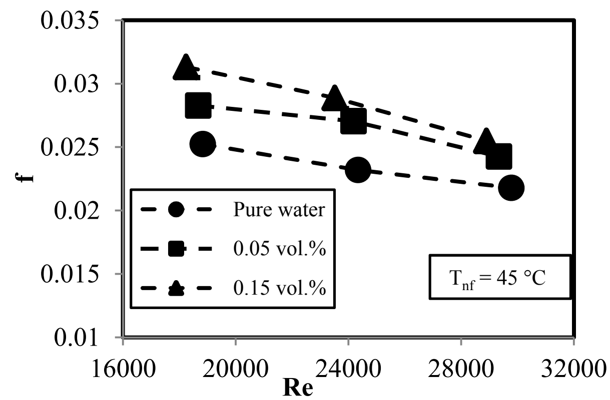

Fig.11 .Variations of friction factor with Reynolds number

It is shown that the friction factor reduces as the Reynolds number increases. The effects of particle loading on the friction factor are more visible at low Reynolds numbers. At low Reynolds numbers, the friction factor increases with an increase in the volume fraction of nanoparticles. This is due to the fact that at low flow velocities, the ratio of the viscous forces to the inertia forces is greater; therefore, adding the nanoparticles to pure water leads to more increase of shear stress among the fluid layers and hence, the friction factor increase is more considerable. Maximum enhancement of the friction factor is obtained about 25% at the nanoparticle concentration of 0.15 vol. % and at Reynolds number of about 18,000.

Generally the Nusselt number is a function of Reynolds number, Prandtl number and nanofluids volume concentrations. Based on curve fitting of the experimental

Nusselt number data, a new correlation has been derived to predict the nanofluids Nusselt number:

$$Nu_{Reg} = 0.208 Re^{0.69} Pr^{0.097} \varphi^{0.039} \tag{12}$$

It should be noted that in this equation the Reynolds number is between 18000 to 40000, the volume concentration is less than 0.15% and the Prandtl number is between 2.8 to 4.This equation predicts the nanofluids Nusselt number within +6% and - 4%.

In a same way with the Nusselt number, a new friction factor correlation as a function of Reynolds number and nanofluids volume concentration is proposed as follow:

$$f_{Reg} = 1.44 Re^{-0.37} \varphi^{0.069} \tag{13}$$

This equation predicts the nanofluids friction factor within +3% and - 3%.Table 4 shows a summary of experimental results.

Conclusion

In this paper, the overall heat transfer coefficient and the flow characteristic of γ-Al_2O_3/water nanofluid in a double tube counter flow heat exchanger has been measured experimentally under turbulent flow regime. The experiments were conducted at wide ranges of flow rate for nanofluid, nanoparticle volume concentrations and at different nanofluid inlet temperatures. The following conclusions can be achieved from this study:

- By suspending a small amount of γ-Al_2O_3 nanoparticles, overall heat transfer coefficient of nanofluids increases. Maximum enhancement of the overall heat transfer coefficient is 19.3% in comparison with pure water which was obtained at 0.15 vol. % of γ-Al_2O_3.
- Increasing the nanoparticles concentration had not considerable effect on the overall heat transfer enhancement in the range of concentration studied in this work.Also, the rate of overall heat transfer coefficient enhancement of nanofluid to that of pure water decreased with increasing the nanofluid flow rate.
- Increasing the nanofluid inlet temperature from 45 to 65 °C increase the overall heat transfer coefficient which is due to the large increase in the nanofluid temperature difference comparing with the less increase in the logarithmic mean temperature difference. Also, it is found that the increase in the nanofluid inlet temperature creates the average heat transfer rate enhancement of about 53% at the highest volume fraction of nanofluid (0.15 vol.%).
- The experimental results indicate that the overall heat transfer coefficient and the heat transfer rate of nanofluid have been improved with the enhancement in the nanofluid flow rate.
- The friction factor of nanofluid increased with increasing the volume fraction of nanoparticles. The maximum friction factor was about 25% greater than that of pure water which was occurred at the highest volume fraction of nanofluid (0.15 vol.%).

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