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RESEARCH PAPER

Measurement of Local Convective Heat Transfer Coefficient of Alumina-Water Nanofluids in a Double Tube Heat Exchanger

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Abstract

Heat transfer coefficient and thermal efficiency of γ-Al_2O_3/water nanofluids flowing through a double tube heat exchanger were experimentally investigated. The nanoparticles were well dispersed in distilled water at 0.05–0.15 %vol. A large number of experiments were performed at different fluid flow rates under turbulent flow regime (18,000<Re<40,000) and various nanofluid inlet temperatures ranging from 45 °C to 65 °C. The heat transfer coefficients were measured along the length of the heat exchanger. Results showed that the local heat transfer coefficients have an asymptotic behavior. Furthermore, the addition of these small amounts of nanoparticles to the base fluid augmented the heat transfer up to 16% at the best conditions. In the end, the thermal performance factor was calculated to find the optimum condition at which the nanofluid was used. It was shown that the thermal performance factor of this nanofluid could reach to 1.11. This value was obtained at the nanoparticle concentration of 0.15 vol.% and Reynolds number 18000.

Keywords:
Experimental,
Friction Factor,
Heat Transfer,
Nanofluid,
Reynolds Number

Introduction

It is believed that energy resources are limited and demands for energy usage are growing. This reality requires innovative technologies for energy saving. It was recently proved that nanofluids which are suspensions containing stable nanoparticles (10 nm to 100 nm size) in a base fluid can be one of the proper thermal energy transfer media. Nanofluids are currently attracting a great deal of experimental and theoretical study according to their properties and performance. A review of the recent literature shows that most of the papers in this field were mainly focused on flow and heat transfer in simple geometries, such as circular tubes. For example, Pirhayati et al [1] measured the convective heat transfer coefficient of nanofluid inside an inclined copper tube which its boundary was at the uniform heat flux. Their Results show that the heat transfer coefficient of nanofluid with different weight fractions increases with the increasing Reynolds number inside horizontal and inclined round tubes. Rostamzadeh et al [2] carried out an experimental investigation to analyze mixed convection heat transfer from Al_2O_3/water nanofluid inside a vertical, W-shaped, copper-tube with uniform wall temperature.

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The results showed that the rate of heat transfer coefficient improved with Reynolds number for average wall temperatures of 50 and 60 °C.

However, heat transfer and pressure drop experimental data for nanofluid flow in the complicated geometries such as double tube heat exchangers and compact heat exchangers are limited [3].It should be emphasized that in a double tube heat exchanger, heat transfer surface has not experienced constant temperature or constant heat flux boundary conditions. As a result, the heat transfer coefficient in such a geometry is not easy to calculate. As can be found, several correlations and empirical methods were developed for this purpose. All the previous methods can be classified into three groups as follows:

The first method includes the application of Logarithmic Mean Temperature Difference (LMTD) which was extensively used before [4-7]. In the second method, which can be named as "simplified LMTD", some simplifications are implemented on the first method, and in the third method, the heat transfer surface temperature is considered to be constant through the heat exchanger [8-11].

Zarringhalam et al [12] studied the rheological and thermal behavior of CuO/Water nanofluid in a double tube heat exchanger at different concentrations. The Results showed that the addition of nanoparticles increases the heat transfer coefficient compared to the base fluid. Moreover, it is concluded that increasing nanoparticle volume fraction and Reynolds number enhance the heat transfer coefficient and Nusselt number.

Raei et al [13] studied the turbulent convective heat transfer and flow characteristics of γ-Al_2O_3/water nanofluid in a double tube heat exchanger. Their results showed that nanofluids had a higher Nusselt number in comparison with pure water. They also reported that the greatest increase of the friction factor and the heat transfer coefficient were 23 and 25% respectively, which were observed when the nanoparticle concentration was 0.15 vol%.

As can be seen, several studies performed on the application of nanofluid in a circular smooth tube or other simple geometries where the heat transfer surfaces were subjected to constant surface temperature or constant heat flux boundary conditions in these studies. As can be seen, fewer works can be found on the thermal and fluid flow properties of different types of nanofluids in a double tube heat exchanger. Also, the average heat transfer coefficient was measured in almost all of the studies performed on the double tube heat exchangers, so far. Therefore, analysis of the variation of local convective heat transfer coefficient in double tube heat exchangers can be a novel subject for researchers.

In this research, an experimental study has been performed to measure local convective heat transfer coefficient and friction factor of stabilized γ-Al_2O_3/water nanofluid in a fully-developed turbulent flow regime in a double tube heat exchanger. Experiments performed at different nanofluid concentrations, operating temperatures, and nanofluid flow rates.

Experimental

Setup and Procedure

Schematic view of the experimental setup is shown in Fig. 1. This apparatus has already been used in similar researches [13,14-16] and also details of the experimental setup have been described in the mentioned references.

Uncertainty Analysis

In this work, an uncertainty analysis was performed. The range of operating conditions and their corresponding measurement uncertainties are summarized in Table 1.

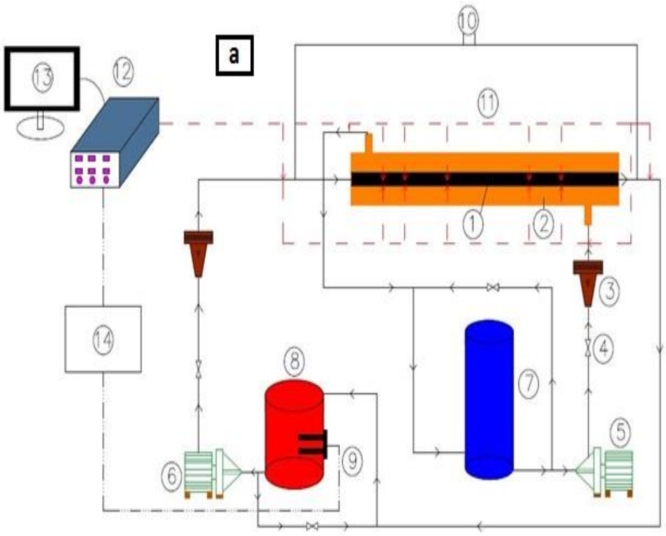

Fig. 1. Schematics of the experimental setup:

Table 1. The range of operating conditions and their measurement uncertainty

Condition	Range	Unit	Uncertainty
Hot liquid flowrate	7-11	l/min	±0.1
Hot liquid inlet temperature	45-65	°C	±0.1
Cold liquid flowrate	13	l/min	±0.1
Cold liquid inlet temperature	6	°C	±0.1
Heat transfer coefficient	8,100-16,400	W/m^2 K	9.7%
Friction factor	0.02-0.034	-	4.65%
Reynolds number	18,000-40,000	-	0.83%
Nanoparticle concentration	0-0.15	vol.%	0.1%

Nanofluids Preparation and Properties

In this work, aluminum oxide nanoparticle with 99 % purity and 20 nm average particle size was purchased from US Research Nanomaterials, Inc., USA. The almost spherical nanoparticles were dispersed mechanically in distilled water as the base fluid. To provide stable nanofluid, no chemical was added in order to prevent any probable complication.

Different concentrations of nanofluid including 0.05 and 0.15 of vol% were prepared. For this purpose, an accurate three decimal places balance was used to weigh a certain amount of γ-alumina nanoparticle. This nanoparticle was then added to distilled water as the base fluid. Mixing with a magnetic stirrer for half an hour, and ultrasonication for 3 h guarantees the stability of the nanofluid up to 28 h. The ultrasonic vibrator (BANDELIN Co.) had a frequency of 35 kHz and power of 240 kW.

In the design of the experiment, some experiments were repeated to check the repeatability of the data. Furthermore, some tests were repeated with a longer period of time in order to check the stability of the nanofluids. All of these repeated data showed an acceptable accuracy and proved the nanofluid stability.

Density and specific heat capacity of γ-Al_2O_3/water nanofluid were calculated at the average of the fluid inlet and outlet temperatures as follows [17,18]:

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

$$C_{p_{nf}} = \frac{\varphi\rho_p C_{p_p} + (1-\varphi)\rho_{bf} C_{p_{nf}}}{\rho_{nf}} \tag{2}$$

The dynamic viscosity and the thermal conductivity of the nanofluid have been calculated using the correlations proposed by Williams et al. [19]. Their correlations were based on experimental data specifically for γ-Al_2O_3-water nanofluid as shown in Eqs. (3) and (4):

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

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

Data Reduction

As previously stated, three different methods were used by the researchers to calculate the average heat transfer coefficient in a double tube heat exchanger. A detailed explanation of these three methods was presented elsewhere [15]. In this research, a novel method is used for the calculation of the average heat transfer coefficient. In this method, data of local heat transfer coefficient along the heat exchanger are implemented.

Results and Discussions

Validation of The Experimental System

Before evaluating the heat transfer performance of the nanofluids, the pure water is used as the working fluid for estimating the reliability and accuracy of the experimental system. The results of the experimental Nusselt number are compared with those obtained from the Gnielinski equation [20], which is defined as follows:

$$Nu_D = \frac{\left(\frac{f}{8}\right)(\mathrm{Re}_D-1000)\,\mathrm{Pr}}{1+12.7\left(\frac{f}{8}\right)^{\frac{1}{2}}\left(\mathrm{Pr}^{\frac{2}{3}}-1\right)} \tag{5a}$$

$$f = (0.79\ln \mathrm{Re}_D - 1.64)^{-2} \tag{5b}$$

The friction factor can be calculated from the Colebrook equation [23] as follows:

$$\frac{1}{\sqrt{f}} = -2\times\log\left(\frac{2.51}{\mathrm{Re}\sqrt{f}}+\frac{\varepsilon/D}{3.7}\right) \tag{6}$$

where ε is roughness of stainless steel 316 tube and was considered 0.002 mm in this study [14].

Fig. 2 compares pure water experimental Nusselt numbers with the prediction of Gnielinski correlation. Relative average error compared to Gnielinski correlation is 8%. The average relative error between the experimental friction factor of distilled water and the prediction of Colebrook [21] relation is 8%. It is noted that the Nusselt number and friction factor results of the experiment have a good agreement to empirical equations.

Local Heat Transfer Coefficient

Fig. 3 shows the local convective heat transfer coefficient versus the axial distance of the annulus at three different volumetric flow rate. Table 2 summarizes the measurements and calculations of the local heat transfer coefficient at different temperatures for different concentrations of alumina nanofluids under a turbulent flow regime at Q_{nf}=11 l/min. The results clearly show that using γ-Al_2O_3 nanofluid, moderately increase the convective heat transfer, especially at the entrance region and at the higher volumetric flow rate.

It can be seen that the maximum enhancement rate is observed at the inlet of the test section and it decreases as approaches to the end of the test section. It can be attributed to the variation of the thermal boundary layer thickness in this distance. At the inlet of the test section, the thickness of the thermal boundary layer and the resultant thermal resistance is small. It causes the heat transfer coefficient to increase. On the other hand, at the end of the test section, the thermal boundary layer and the thermal resistance increase which result in a reduction of the heat transfer coefficient.

In addition, in comparison to pure water, when the nanoparticle concentration increases, the heat transfer coefficient rises. It can be seen that increasing the fluid flow rate from 7 to 11 l/min causes the heat transfer coefficient to be increased. Increasing fluid flow rate increases Re and consequently increases the turbulence and better dispersion of nanoparticles in the liquid bulk. As a result, the temperature distribution is flattened and the temperature gradient between the heat transfer surface and the working fluid is sharpened.

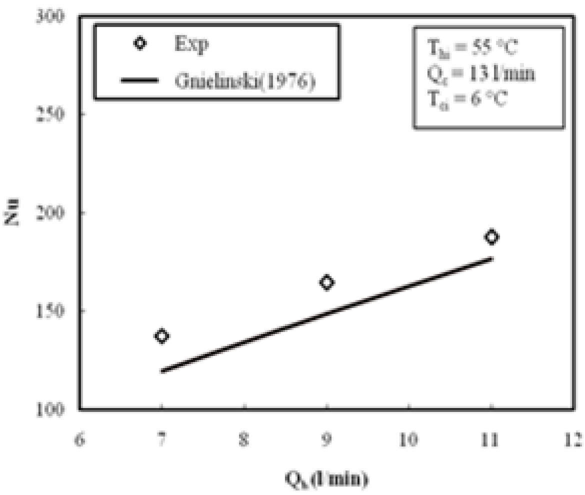

Fig. 2. Nusselt number results for experiments with distilled water and Gnielinski equation

Table 2. Experimental data of local heat transfer coefficient (kW/m^{2}K) of alumina-water nanofluids at Q_{nf} =11l/min

T_{nf} (°c)	x(cm)	φ = 0	φ = 0.05%	φ = 0.15%
45	5	43.18	56.19	42.31
	10	29.62	36.69	30.33
	20	7.81	9.13	9.71
	40	4.61	7.61	6.41
	h_{av}	11.93	12.94	13.39
55	5	48.81	44.97	44.59
	10	34.55	33.76	33.29
	20	9.35	11.48	11.21
	40	5.71	6.29	7.81
	h_{av}	14.01	14.47	15.06
65	5	51.46	49.13	46.66
	10	36.24	36.61	36.41
	20	9.44	11.01	13.36
	40	7.23	7.81	8.05
	h_{av}	15.22	15.78	16.35

Table 2 shows that the enhancement of the heat transfer coefficient increases with increasing the temperature. Several mechanisms may cause these improvements. These could be due to the increase in the Brownian motion of particles by increasing the temperature [22].

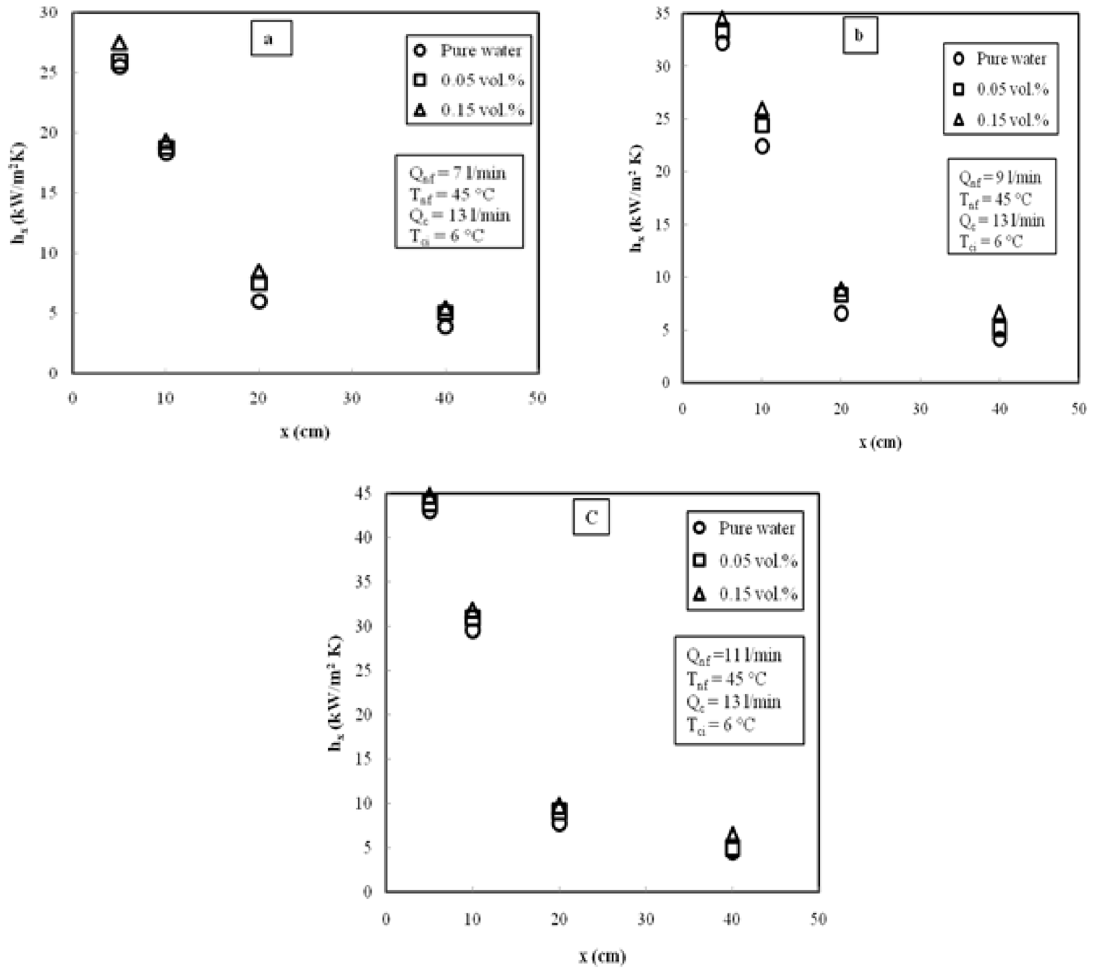

Fig. 3. Local heat transfer coefficient variation of γ-Al2O3/water nanofluid along axial distance at Tnf = 45 °C, a) Qnf = 7 l/min, b) Qnf = 9 l/min, c) Qnf = 11 l/min

Average Heat Transfer Coefficient

Fig. 4 shows that heat transfer in nanofluids is higher than distilled water and this will increase in a fixed volumetric flow rate with slight increases in nanoparticles concentration. In addition, heat transfer increases with an increase in the volumetric flow rate and inlet temperatures.

As Fig. 4 shows, at higher concentration levels, no sensible increase is obtained in the heat transfer of nanofluid. For example, at a volumetric flow rate of 7 l/min and inlet temperature of 45 °C, the value of h_{nf}/h_{bf} was 1.12 and 1.16 for 0.05% and 0.15% concentration, respectively, which showed merely 4% improved for 0.10% increase in nanofluid concentration. Therefore, increasing the concentration of nanoparticles in a small range studied in this work had no significant influence on the heat transfer enhancement.

This observation is not in accordance with the observations of Xuan and Li [23] and also Heris et al. [24] who reported considerable enhancement of heat transfer coefficient of nanofluids with increasing the nanoparticles concentration. However, some other researches like Fotukian and Nasr [25] and also Sajadi and Kazemi [26] obtained similar results like those obtained in this study.

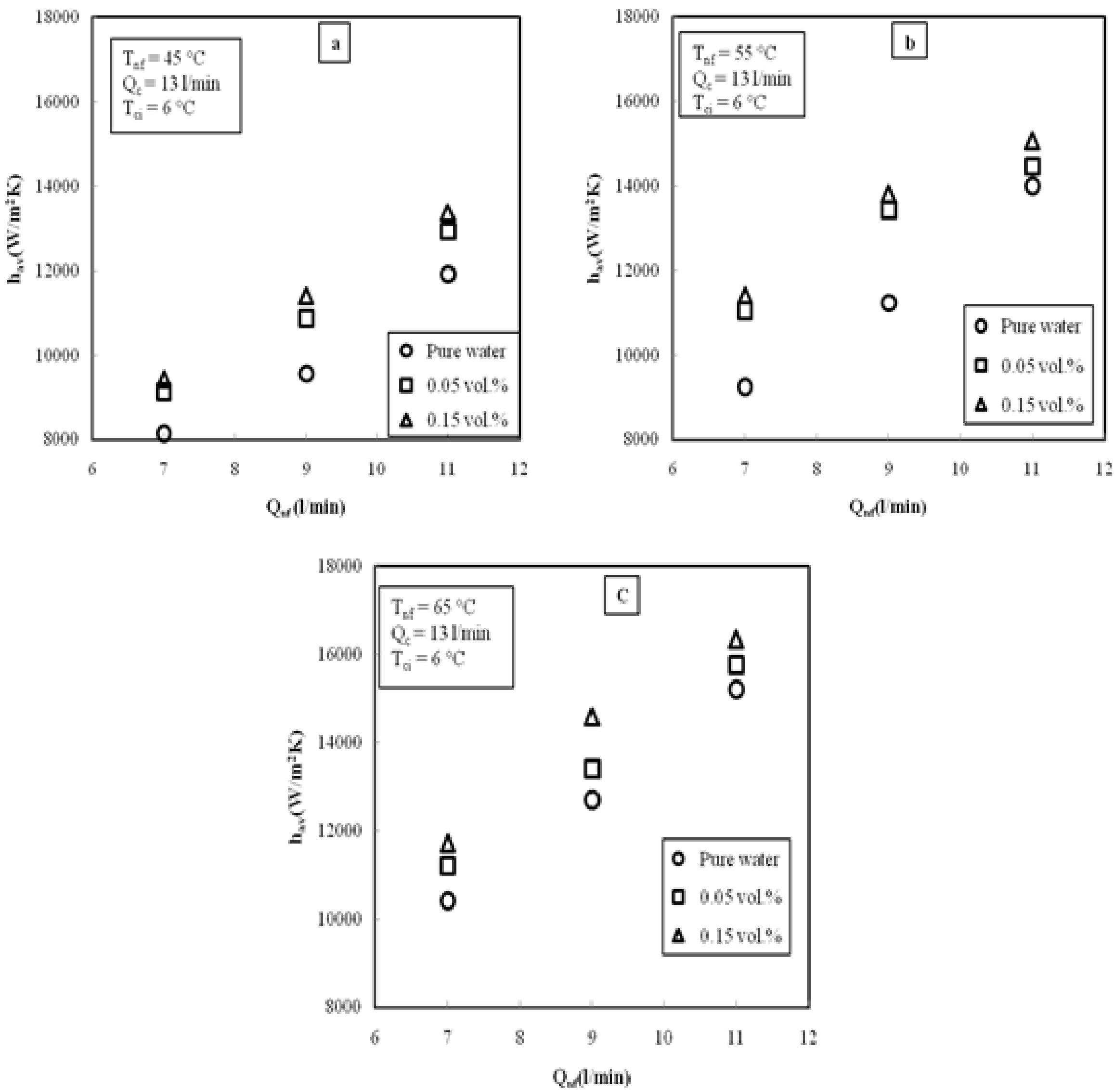

Fig. 4. The average heat transfer coefficient of γ-Al_2O_3/water nanofluid as a function of the nanofluid flow rate at different nanofluid concentrations and inlet temperatures of a**)** 45 °C, b) 55 °C, c) 65 °C

It is shown that the addition of nanoparticles improves the heat transfer coefficient of nanofluids compare with pure water. One of the reasons which was previously proved to be efficient in nanofluid heat transfer mechanism is the improvement of thermal conductivity of the working fluid with the addition of nanoparticles [27-30]. It was also shown that a decrease of boundary layer thickness due to the Brownian motion of nanoparticles may also be another reason for nanofluid heat transfer improvement. The random displacement of nanoparticles in the base fluid agitates the thermal boundary layer, and as a result, higher heat transfer coefficients may be achievable. When a concentration of nanoparticles was added, effects of both mechanisms were magnified and consequently, higher heat transfer coefficient may be obtained at higher concentrations. Furthermore, increasing the fluid inlet temperature causes the enhancement in the heat transfer coefficient. It is found that maximum improvement in heat transfer coefficient compared with water was 23% which was obtained at 0.15 vol.% of γ-Al_2O_3/water nanofluid, the temperature of 65 °C, and flow rate of 11 l/min.

Friction Factor

The friction factor of the working fluid is calculated at various concentrations of nanoparticles (Table 3). Increase of the friction factor at low Reynolds numbers is more significant at a higher volume fraction of nanoparticles. When the flow velocity is low, the viscous forces is greater than the inertia forces and, as a result, increasing the concentration of nanoparticles causes more shear stress among the fluid layers, and hence, the greater friction factor may be expected. In addition, the results show that the friction factor of the nanofluids slightly increases with increasing the nanoparticle concentration.

Table 3. The experimental data for the friction factor

Re	f		
	φ=0	φ=0.05	φ=0.15
18500	0.0252	0.0282	0.0313
24100	0.0231	0.027	0.0288
29400	0.0217	0.0242	0.0254

Thermal Performance Factor

As shown in the previous sections, aqueous γ-alumina nanofluid enhances the heat transfer coefficient when using inside the heat exchanger instead of water. However, this advantage is accompanied by a penalty of increasing pressure drop which is a great disadvantage from an industrial point of view.

In order to simultaneously consider the effect of using nanofluid on the heat transfer coefficient, besides the pressure drop, the thermal performance factor is a suitable definition which was previously used in different studies.

It was shown that nanoparticles simultaneously enhance the thermal conductivity and the viscosity of the base fluid. Increase of these two physical properties has contradicting effects on the heat transfer while increasing of the viscosity increases the pressure drop. The thermal performance factor, η, can be defined as [31]:

$$\eta = \left(\frac{Nu_{nf}}{Nu_f}\right)\left(\frac{f_f}{f_{nf}}\right)^{\frac{1}{3}} \tag{7}$$

Thermal performance factor evaluates the heat transfer enhancement technique regarding the required pumping power. It was shown that when the thermal performance factor was greater than unity, the heat transfer improvement technique would be a good choice for implementing in practical and industrial applications. Fig. 5 demonstrates the variation of thermal performance factor as a function of Reynolds number for γ-Al_2O_3/water nanofluids at different nanofluid concentrations.

As Fig. 5 shows, the thermal performance that investigated in ranges of Reynolds number and nanofluid concentration is always higher than unity. It reveals that the augmentation of the heat transfer by using the nanoparticles is higher than the pressure drop penalty. The decrease in the Reynolds number and increase in nanoparticle concentration causes the thermal performance to generally grow.

Over the studied range, the maximum thermal performance factor of 1.11 is found with the use of nanofluid of 0.15 by volume at Reynolds number of 18,000. As shown, at all the operating conditions, increasing the heat transfer coefficient by using nanofluids compensated for the disadvantage of increasing pressure drop. Similar results were reported in a large number of studies [32-34,29,16]. In addition, some few studies concluded that the thermal performance factor may decrease at higher concentrations of the nanoparticle. They related this observation to particle agglomeration and sedimentation.

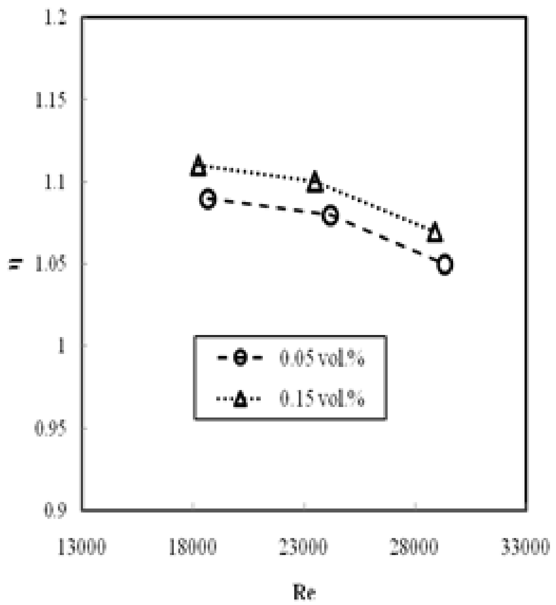

Fig. 5. Evolutions of thermal performance factor of the test section.

In this study, the results are in contradiction to the results of other researchers [35,9]. It is not easy to explain these contradictions. These could be attributed to several experimental parameters such as type of base fluid, particle geometry (size, shape or even type), nanofluid stabilization and preparation techniques (utilizing sonication or surfactant or variation in pH), and also a method of synthesis of nanoparticle [36-39]. Therefore, more experimental and theoretical studies must be carried out to deeply explore the application of nanofluids in the heat transfer apparatus. The experimental data are summarized in Table 3.

Conclusions

In this study, the local convective heat transfer coefficient and the friction factor of γ-Al_2O_3/water nanofluid in a double tubular heat exchanger has been measured experimentally. The experiments were conducted at wide ranges of nanofluid flow rate, nanoparticle volume concentrations and at different nanofluid inlet temperatures. The experimental results are summarized in the following conclusion:

- By adding a small amount of γ-Al_2O_3 nanoparticles to distilled water, the heat transfer coefficient of nanofluids increases. Also, increasing the concentration of the nanoparticles in the range of this research didn't have a substantial impact on the heat transfer coefficient.
- The thermal performance factor for the given nanofluids is greater than unity, and maximum thermal performance factor was about 1.11 at 0.15% particle concentration and Reynolds number 18,000. So, nanofluids can be used as working fluid in heat transfer and it helps engineers to design more efficient heat exchangers.

Nomenclature

Cp	specific heat (J/kg °C)
D	Diameter (m)
Exp	experimental
f	friction factor
h	hour
ID	Inner diameter (mm)
k	thermal conductivity (W/m.K)
l	liter
L	length (m)
Nu	Nusselt number
P	pressure (Pa)
PC	personal computer
Pr	Prandtl number
Re	Reynolds number
T	temperature (°C)
vol	volume
x	Axial distance

Greek Symbols

φ	volume fraction
ε	roughness (m)
μ	viscosity (Pa.s)
ρ	density (kg/m3)
Δ	difference
η	Thermal Performance Factor

Subscripts

av	average
b	bulk
bf	base fluid
c	cold
h	hot
nf	nanofluid
p	particle
x	local

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