Heat Transfer of Nano-Fluids as Working Fluids of Swimming Pool Heat Exchangers

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Abstract: The present experimental study reports on enhancement of heat transfer by addition of nanoparticles to the working fluid of commercial swimming pool heat exchangers under laminar flow condition. Three different concentrations of Titanium dioxide nanoparticles were added to the water as working fluid of a typical forced convective heat exchanger used to transfer heat to public swimming pools. The experimental setup is capable of measuring velocity, heat transfer rate, and temperature at different points. TiO_2 nanoparticles with mean diameter of 20 nm were used. The effects of suspended nanoparticles concentration and that of Peclet number on forced convective heat transfer were investigated. It is observed that at 0.1%, 0.5% and 1% weight concentration of suspended TiO_2 nanoparticles, the average convective heat transfer coefficient improved by 1.1%, 15.9% and 31.6% respectively. The coefficient is further increased at higher Peclet numbers. The efficiency of heat exchanger is evaluated for different scenarios.

Keywords: Convective Heat Transfer, Nano-Fluids, Nanoparticles, Pool Heat Exchanger,Swimming Titanium Dioxide

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1 INTRODUCTION

Nowadays working fluids play a paramount role in heating and cooling industry. Keeping the problem of energy shortage and the necessity of cutting the corners in energy consumption in perspective, finding useful solutions and using economical equipment are the most important issues in the science of heat transfer. Naturally it is desired to minimize the heat exchangers in size while increasing their capacity to the maximum quantity available in order to deliver the required thermal load. The low efficiency of working fluids increases the energy consumption and as a result enlarges the equipment and increases other ancillary expenses.

Due to low conductivity of ordinary working fluids such as water and oil in comparison with solid metals or metal oxide particles, addition of solid substances with better thermal properties may improve the working fluids efficiency. The addition of micrometer- or millimeter- sized of these solid particles to the base fluids shows an increase in the thermal conductivity of resultant fluids but the presence of milli- or micro sized particles to the base fluids poses a number of problems. The particles do not form a stable solution and tend to settle down [1]. Agglomeration of particles is one of the main obstacles encountered in micro fluid experiments.

Although Nano-sized particles greatly reduce the problem of agglomerated particles, however it still occurs and can hinder the thermal capacity of the Nano-fluid, especially at concentrations over 5%. Agglomeration is more apparent when using oxide nanoparticles because they require a higher volume concentration compared to metallic nanoparticles in order to achieve the same heat transfer enhancement [2]. Furthermore, with respect to the sedimentation of solid particle suspensions, the application of particles in the Nano-scale inside the base fluid reduces the amount of the sediment and as a result decreasing the blockage and erosion of the pipe passages considerably.

Heat exchangers, different in size, design and capacity, are used in a variety of industries. One of the applications of heat exchangers is to heat (or to cool) the water in swimming pools. A swimming pool heat exchanger, either the shell and tube or the plate type, makes it possible to control the temperature of the swimming pool and extends the swimming season. It makes use of a simple method of heating water indirectly from a boiler or a solar panel. It can also be used to cool pool water in hot climates. In this case, water from a chiller passes over the heat exchanger tubes instead of boiler or solar panel. So far, the forced convective heat transfer of water and Al_2O_3Nano-fluid in both shell-and-tube and brazed-plate heat exchangers has been experimentally measured [3]. It is also reported that addition of TiO_2 and Al_2O_3 nanoparticles to the working fluid causes significant enhancement in heat transfer characteristics in a shell-and-tube heat exchanger under turbulent flow [4]. In a horizontal double-tube counter-flow heat exchanger the convective heat transfer coefficient of Nano-fluids is measured to be slightly higher than that of the base liquid by about 6-11% [5].

It is reported, in the literature, that Pantzali et al. [6] carried out an experimental study on the efficacy of Nano-fluids as coolants in the plate heat exchangers while Zeinali et al. [7], [8] investigated thermal efficiency of water/Cu Nano-fluid with thermal boundary condition of constant temperature. They also showed that adding CuO and Al_2O_3 nanoparticles to the water, significantly increases the heat transfer coefficient and Nusselt number of laminar flow. Other study reports that the forced convective heat transfer coefficient of water/Al_2O_3 Nano-fluid with constant heat flux thermal boundary condition enhances with an increase in Reynolds number or the concentration of nanoparticles [9].

Improvement of heat transfer using water/Al_2O_3Nano-fluid is also reported by Wen and Ding [10] where it is postulated that addition of nanoparticles affects the development of thermal and hydrodynamic boundary layer. The thermal development length for Nano-fluid is more than base fluid and increases with increasing of nanoparticle concentration. In the turbulent flow, increasing the Reynolds number of Water/Cu Nano-fluid and its concentration helps the heat transfer [11]. Also, the addition of carbon nanotubes in the base fluid significantly enhances the heat transfer coefficient [12]. It is therefore of interest to examine other Nano-fluids in an attempt to observe their effect on the heat transfer in the typical heat exchangers used in the swimming pools. This is carried out in the present experimental study with TiO_2 nanoparticles added to the working fluid of swimming pool heat exchangers for the shell & tube type.

2 SAMPLE PREPARATION

Preparation of Nano-fluids which is carried out by adding nanoparticles to the base fluid is not a simple solid- liquid mixture. A proper Nano-fluid should bear four major factors: 1-uniformity of suspension, 2-stability of suspension, 3-low sedimentation of particles and 4-the particles remain chemically unchanged. In the present study, by adding cetyl trimethyl ammonium bromide (CTAB) as surfactant to the water/titanium dioxide Nano-fluid, the surface properties of the suspended particles were changed. This change causes the formation of clusters of particles due to the creation of a stable suspension. Here, CTAB was used due to its

chemical compatibility with the base fluid and TiO_2 nanoparticles. It also maintains the PH of the suspension and helps avoiding an erosive environment.
In this study TiO_2 nanoparticles with mean diameters of 20 nm and 99% purity were used. The two-step method for producing Nano-fluids was used: first CTAB was added to distilled water to raise solubility and stability of the Nano-fluid. The CTAB had been mixed in the ultrasonic vibrator for 15 minutes for better homogenization. Second, nanoparticles were added to this solution and finally to prevent clogging, sedimentation and adherence of nanoparticles in two-step method, the solution was homogenized in the ultrasonic vibrator for 60 minutes to disperse the nanoparticles. Breaking the clogged condition of particles and restoring them to their original condition is one of the essential steps of producing Nano-fluids, since the size and distribution of particles in the fluid play the most important role in determining the thermal and hydraulic behavior of the suspension. During the sonication the temperature of the solution was intensely increased which has a negative effect on solubility. Consequently, cold water bath was applied to control the temperature of the solution where the cycle of ultrasonic vibrator was 0.6.

3 EXPERIMENTAL SETUP

As it can be seen in Fig. 1, the experimental setup contains a closed loop of hot flow and an open loop of cold flow. Water/titanium dioxide Nano-fluid was used in the hot flow loop. A swimming pool heat exchanger, two pumps, a data logger, two flow meters, an electronic thermostat, electric heaters, thermo couples and two reservoir tanks are the equipment used in the experiment. The swimming pool heat exchanger is a shell-and-tube heat exchanger with 0.013 m^3 volume and 35.5 cm length which contains 39 tubes of 29 cm length, 11 mm outside diameter and 10 mm inside diameter, and a shell of stainless still with 0.253 m^2 exchanging area and 6.35 cm inside diameter. The hot fluid flows through the tubes and the cold fluid flows through the shell.
Valves and connections were insulted with an artificial silk to reduce heat dissipation. Four K–type thermocouples were used at the inlet and outlet of heat exchanger to measure the temperature. Calibrated thermocouples of accuracy of ±0.1°C were also used. The Rota-meters were calibrated by measuring the time needed to fill a 4-liter tank. Temperatures of inlet and outlet flows of the heat exchanger were recorded with a data logger. The hot fluid's temperature was maintained by an electronic thermostat of accuracy of ±0.1 °C. The thermostat temperature sensor was PT100 type thermocouple. The energy of the hot water tank was supplied by two electric heaters. Water was used as the fluid of the cold flow of the system. The temperature of the cold flow was kept constant during the experiment.

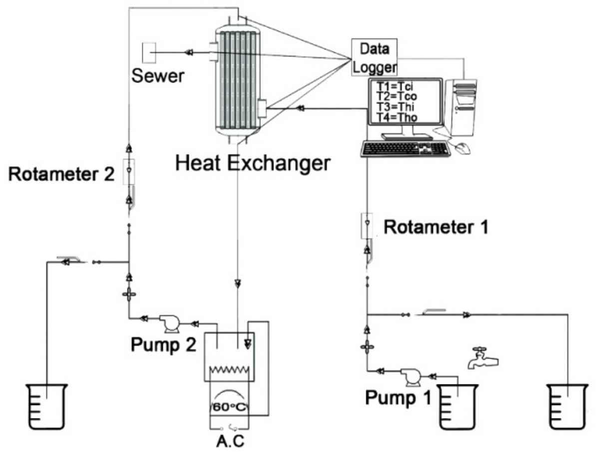

Fig. 1 Experimental setup

4 ANALYSIS

The procedure to obtain forced convective heat transfer coefficient in the present study is as follows:
The overall heat transfer rate was obtained by implementing the energy balance equation for hot and cold fluids in the swimming pool heat exchanger. If the heat exchange between the heat exchanger and surrounding, potential and kinetic energy changes are negligible, the specific heat capacities are constant, and due to the unchanged phase of fluids, the equation is:

$$\dot{q} = \dot{m}_h C_{Ph}(T_{hi} - T_{ho}) = \dot{m}_c C_{Pc}(T_{co} - T_{ci}) \qquad (1)$$

Here, *h*, *c*, *i,* and *o* pertain to the hot and cold fluids, inlet and outlet respectively. More specifically, the rate of transferred heat is given as follows:

$$\dot{q} = U_i A_i F \frac{(T_{ho}-T_{ci})-(T_{hi}-T_{co})}{ln\frac{T_{ho}-T_{ci}}{T_{hi}-T_{co}}} = \rho \dot{V} C_{Ph}(T_{hi} - T_{ho}) \qquad (2)$$

Where $'\dot{V}'$is the volumetric flow rate of the hot fluid, 'U_i' is the overall heat transfer coefficient, 'F' is the correction factor and $\Delta T_{lm,CF} = \frac{(T_{ho}-T_{ci})-(T_{hi}-T_{co})}{\ln\frac{T_{ho}-T_{ci}}{T_{hi}-T_{co}}}$is the logarithmic mean temperature difference and $'A_i'$ is the hot flow side heat transfer area of the heat exchanger. In the present study, the evaluation of overall heat transfer coefficient is based on constant wall temperature.
For the swimming pool heat exchanger in which the formation of sedimentary layers on the exchanger's

surfaces is refrained, the overall heat transfer coefficient is given by the equation below:

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{D_i \ln \frac{D_o}{D_i}}{2k_w} + \frac{D_i}{D_o}\frac{1}{h_o} \tag{3}$$

Where D_i and D_o are the inner and outer diameters of tubes respectively. Generally, the Nusselt number for a flow passing through a solid surface is defined as $Nu = f(Re, Pr)$ where:

$$Nu = C_1 Re^{\alpha} Pr^{\beta} \tag{4}$$

Since the heat exchanger and the geometry of it are fixed, the tube wall resistance ($\frac{D_i \ln \frac{D_o}{D_i}}{2k_w}$) becomes constant. Also, if the cold flow rate is constant and its mean temperature does not change for different hot flow rates, the cold flow resistance will remain approximately constant. Therefore, the heat transfer coefficient will only depend on the heat transfer coefficient of hot flow.

If the bulk mean temperature does not differ for different flow rates, all the physical properties will remain almost the same and the combination of Eq. (3) and Eq. (4) could be written as:

$$\frac{1}{U_i} = \frac{1}{h_i} + C = \frac{m}{u^{\alpha}} + C \tag{5}$$

Where 'm' and 'C' are constant. 'h_i', can therefore be evaluated from the intercept of the diagram of $\frac{1}{U_i}$ vs. $\frac{1}{u^{\alpha}}$. Since the value of 'α' is not known, it has to be estimated first. A plot of $log\frac{d(\frac{1}{U})}{du}$ vs. $log\, u$ will eliminate the constant 'C' and the slope will give ($-\alpha-1$). The constant 'm' can also be evaluated with this intercept. Then a plot of $\frac{1}{U}$ vs. $\frac{1}{u^{\alpha}}$ will provide the intercept value 'C', which is then used to calculate the heat transfer coefficient from Eq. (5). The Nusselt number correlation can then be found. The value of 'α' is found to be around 1/3 for developed laminar flow. This can be verified if the plot of $\frac{1}{U}$ vs. $\frac{1}{u^{1/3}}$is a straight line for a large range in the small u limit. The equations below were used to evaluate Nano-fluid properties [4], [13]:

$$\rho_{nf} = (1-\phi_V)\rho_f + \phi_V \rho_P \tag{6}$$

$$(\rho C_P)_{nf} = (1-\phi_V)(\rho C_P)_f + \phi_V (\rho C_P)_p \tag{7}$$

$$\mu_{nf} = \mu_f(1+2.5\phi_V) \quad for\ \phi_V < 0.05 \tag{8}$$

$$k_{nf} = k_f \frac{k_p + 2k_f - 2\phi_V(k_f - k_p)}{k_p + 2k_f + \phi_V(k_f - k_p)} \tag{9}$$

Where *nf*, *f* and *p* subscripts are related to the Nano-fluid, base fluid and nanoparticles respectively and ϕ_V is the volume fraction of suspended nanoparticles [13]. The fluid flow velocity in tubes was obtained by volumetric flow rate divided by inside area of tubes ($u = \frac{\dot{V}}{A_i}$).

5 RESULTS AND DISCUSSION

To verify the accuracy of the experiments the results obtained using distilled water were compared with known relations. The tube side Nusselt number of distilled water flow based on constant wall temperature can be obtained by:

$$Nu = \frac{h_i D_i}{k_{nf}} = 3.66 + \frac{0.0668\left({D_i}/{L}\right)RePr}{1+0.04\left[\left(\frac{D_i}{L}\right)RePr\right]^{\frac{2}{3}}} \tag{10}$$

Where h_i is the heat transfer coefficient of hot flow passing through the tubes and D_i is the inner diameter of tubes. Considering the inlet and outlet temperatures of the heat exchanger, Table 1 shows the Prandtl number of water-TiO_2 Nano-fluid at different concentrations of suspended nanoparticles.

Table 1 Prandtl number of water-TiO2 Nano-fluid under test conditions

Weight concentration	Prandtl number
Distilled water	4.34
0.1%	4.34
0.5%	4.33
1%	4.307

As shown in Fig. 2, a fine match exists between the data obtained from Eq. 10 and the experimental data for h_i. Figs. 3, 4 and 5 show the overall heat transfer coefficient (U_i), the convective heat transfer coefficient (h_i) and the Nusselt number of hot flow respectively versus Peclet number for different weight fractions of suspended nanoparticles. The results indicate that with an increase in Peclet number or weight concentration of suspended nanoparticles, convective heat transfer coefficient will also increase. For the shell-and-tube swimming pool heat exchanger at 0.1%, 0.5% and 1% weight concentration of suspended TiO_2 nanoparticles, the average augmentations of convective heat transfer coefficient are 1.1%, 15.9% and 31.6% respectively.

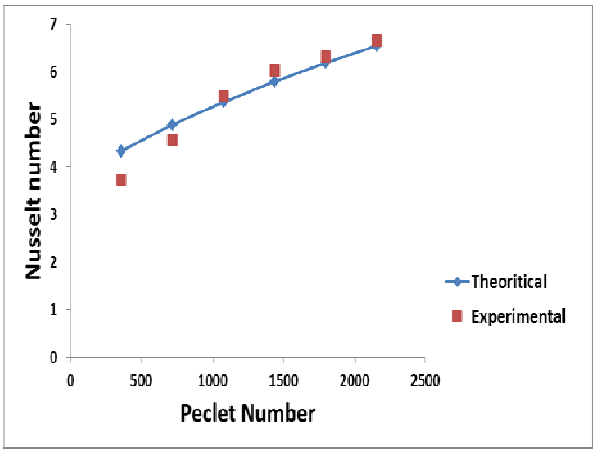

Fig. 2 Comparison of the experimental and predicted values for distilled water

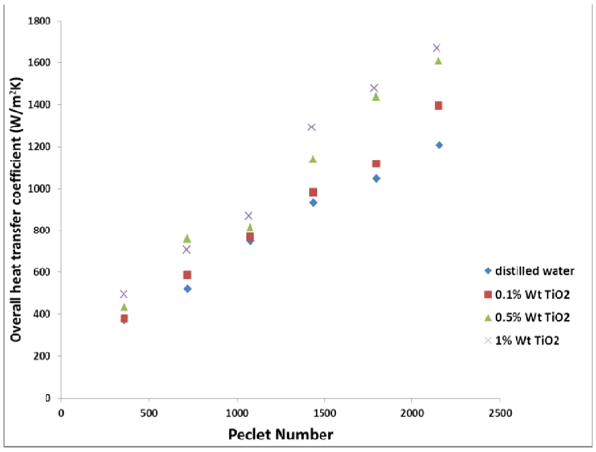

Fig. 3 The overall heat transfer coefficient of water-TiO_2Nano-fluid versus Peclet numbers for different weight concentrations of suspended nanoparticles

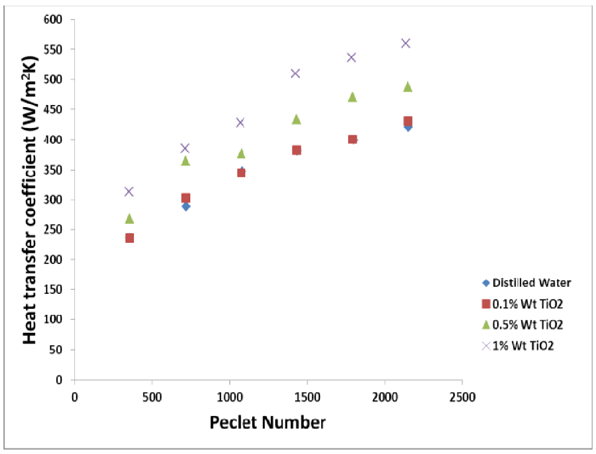

Fig. 4 The heat transfer coefficient of water-TiO_2Nano-fluid versus Peclet numbers for different weight concentrations of suspended nanoparticles

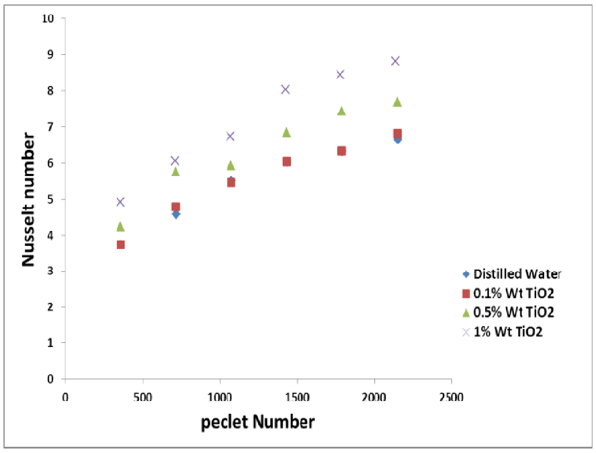

Fig. 5 The Nusselt number of water-TiO_2Nano-fluid versus Peclet numbers for different weight concentrations of suspended nanoparticles

For the swimming pool heat exchanger, Fig. 6 shows the convective heat transfer coefficient of hot flow with different Reynolds numbers versus weight fraction of suspended nanoparticles. This diagram shows a better picture of the impact of concentration of suspended nanoparticles on convective heat transfer coefficient. In addition, as it can be seen, changes in the convective heat transfer coefficient are approximately regular in different Reynolds numbers.

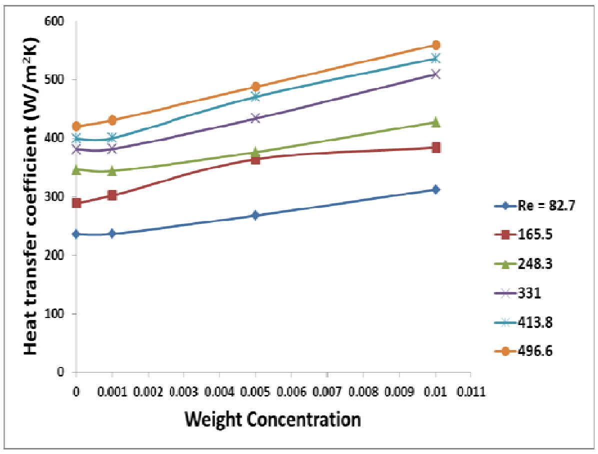

Fig. 6 The convective heat transfer coefficient of water-TiO2 Nano-fluid with different Reynolds numbers versus weight concentration of nanoparticles

6 CONCLUSION

In this experimental study the heat transfer coefficient of water-TiO_2Nano-fluid under laminar flow condition were measured in a shell and tube swimming pool heat exchanger. The effects of Peclet number and weight fraction of suspended nanoparticles on convective heat transfer were investigated. The results indicate that the convective heat transfer coefficient of Nano-fluids enhances obviously compared with that of the base fluid. The convective heat transfer coefficient increases with augmentation of Peclet number. Also, with an increase in concentration of suspended nanoparticles, the convective heat transfer coefficient improves at a constant Reynolds number. These results indicate that Nano-fluids are effective fluids capable of improving heat transfer, and as a result they can be used to minimize heat exchangers size and lower energy consumption.

7 NOMENCLATURE

h Convective heat transfer coefficient, [W/m^2K]

U Overall heat transfer coefficient, [W/m^2K]

q Heat transfer rate, [W]

u Fluid flow velocity, [m/s]

$\dot{m}$ Mass flow rate, [Kg/s]

$\dot{V}$ Volumetric flow rate, [m^3/s]

C_P Specific heat, [J/KgK]

K Thermal conductivity, [W/Mk]

D Diameter of the tube, [m]

T Fluid temperature, [^oC]

A Heat transfer area, [m^2]

Greek symbols

μ Dynamic viscosity, [pa.s]

ρ Mass density, [Kg/m^3]

ΔT_{lm} Logarithmic mean temperature difference, [^oC]

Φ_V Volume concentration of suspended nanoparticles

Subscripts

o outlet, inner

i inlet, outer

c Cold water

h Hot water

w Wall

nf Nano-fluid

p Nanoparticle

f Base fluid

Non-dimensional Numbers

Re Reynolds number, [ρud/μ]

Nu Nusselt number, [hD/K]

Pr Prandtl number, [$C_p\mu/K$]

Pe Peclet number, [Re×Pr]

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