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DESIGN AND CONSTRUCTION OF A HEAT EXCHANGER: USE OF NANOFLUIDS (GOLD NANOPARTICLES IN BASE FLUID)

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Abstract. This study aims to design and assess a shell and coiled tube heat exchanger that uses gold nanoparticles dispersed in a heated up base fluid. The theory about nanofluids, as well as the procedure to design and test a functional prototype of this heat exchanger are presented. 3D drawings showing the main dimensions of the device and other construction details are also included. Preliminary experimental tests at several operating conditions were carried out to verify the effect of the nanofluid in improving the heat exchanger efficiency. Results obtained from preliminary tests of the proposed coil heat exchanger, operating with distilled water and the nanofluid, show an increase in the overall heat transfer coefficient when the gold nanoparticles are used in the heated stream. This work provides an extra tool for the study on the unique characteristics of heat transfer using nanofluids.

Keywords: heat exchanger, nanofluids, gold nanoparticles, thermal-hydraulic design

1. INTRODUCTION

One of the most promising areas of nanotechnology is the use of nanofluids – fluids that contain metallic nanoparticles. In such fluids, some physical properties the carrier fluid, like thermal conductivity, can be greatly enhanced with only a small amount of nanoparticles. So, nanofluids gain relevance in Mechanical Engineering mainly in studies involving the design of heat exchangers. It has been shown that nanofluids are more efficient that the conventional known coolants, presenting two main benefits: heat transfer increment and lower surface area for heat exchangers (Khairul et al., 2013; Kannadasan et al., 2012). In order to prove the higher efficiency of nanofluids when compared to known coolants, it is necessary to build a heat exchanger that is suitable to this new technology. Even thought there are more simplified versions, such as double tube heat exchangers that accept smaller volumes, nanofluids require equipments adapted to its characteristics, which include: low usable volume (because it is a still in-development fluid) and the possibility of decantation when in big volumes/spaces, which can result in damages in the heat exchanger as well as the variation of the results obtained.

In the present paper is proposed the construction of a shell and coiled tube heat exchanger that will use nanofluids composed of gold nanoparticles previously dispersed in a base liquid in order to prove its heat transfer efficiency. This type of heat exchanger was chosen because it presents a secondary pattern of flow in normal plans in relation to its main flow, thus it causes an increase in the overall heat transfer coefficient (Genic et al., 2012).

2. DESIGN OF THE HEAT EXCHANGER

The design of the proposed heat exchanger was entirely based on the Effectiveness NTU-Method (Incropera, 2008). The effectiveness ($\mathcal{E}$) of a heat exchanger can be obtained by calculating the real (q) and the maximum (q_{max}) heat transfer rate, according to Eq. (1):

$$\varepsilon \equiv \frac{q}{q_{max}}, \tag{1}$$

with the real and the maximum heat transfer rates given from Eqs. (2) and (3), respectively:

$$q = C_h\left(T_{h,i}\text{-}T_{h,o}\right), \tag{2}$$

$$q_{max} = C_{min}\left(T_{h,i}\text{-}T_{c,i}\right), \tag{3}$$

where, $T_{h,i}$ and $T_{h,o}$ are, respectively, the inlet and outlet temperatures of the hot stream, and $T_{c,i}$ is the inlet temperature of the cold stream. C_h is the heat capacity of the hot fluid, and C_{min} is equal to C_h or C_c (heat capacity of the cold fluid), whichever is smaller. Taking into account that the global movement of both fluids occurs in the same direction, equations for a parallel flow heat exchanger were considered. So, the effectiveness can be also determined through Eq. (4) (Incropera, 2008):

$$\varepsilon = \frac{1\text{-}exp[\text{-}NUT(1+C_r)]}{1+C_r}, \tag{4}$$

with the number of transfer units (NTU) being defined by Eq. (5):

$$NTU = \frac{UA}{C_{min}}, \tag{5}$$

where U is the overall heat transfer coefficient and A is the area of the heat transfer surface. For the design, it was assumed that the cold fluid is distilled water, entering the heat exchanger at room temperature (approximately 23ºC). The hot fluid is the nanofluid, entering the heat exchanger at 40ºC. The stainless steel coil tube has an internal diameter of 8 mm and wall thickness of 1 mm. The shell of the heat exchanger has an internal diameter of 70 mm, also built in stainless steel. The design procedure of the heat exchanger estimated a total length of the coil tube of 5.06 m, which was rolled up in 47 passes to be inserted in a shell with 320 mm of length. With the obtained dimensions, it was plotted a 3D model of the heat exchanger using the software SolidWorks®, which is shown in Fig. 1.
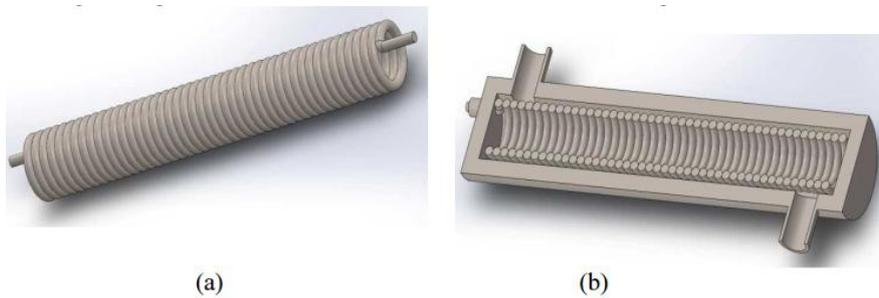

(a) (b)

Figure 1 – 3D views of the heat exchanger; a) coil used in the heat exchanger; b) internal section view of the heat exchanger.

3. EXPERIMENTAL SETUP AND PROCEDURES

3.1 Experimental setup

The shell and coiled tube heat exchanger is depicted in Fig. 2. In this figure, I shows (in red color): (1) nanofluid reservoir with capacity of 1 L, (2) heat exchanger, (3) temperature controller, (4) flow control valve, (A) nanofluid exit, (B) nanofluid entrance, (C) cold fluid entrance, (D) cold fluid exit. Figure 2-II stands out the temperature controller, where the number 1 to 5 in black color indicates the thermocouples placed in the system (Ferreira, 2015).
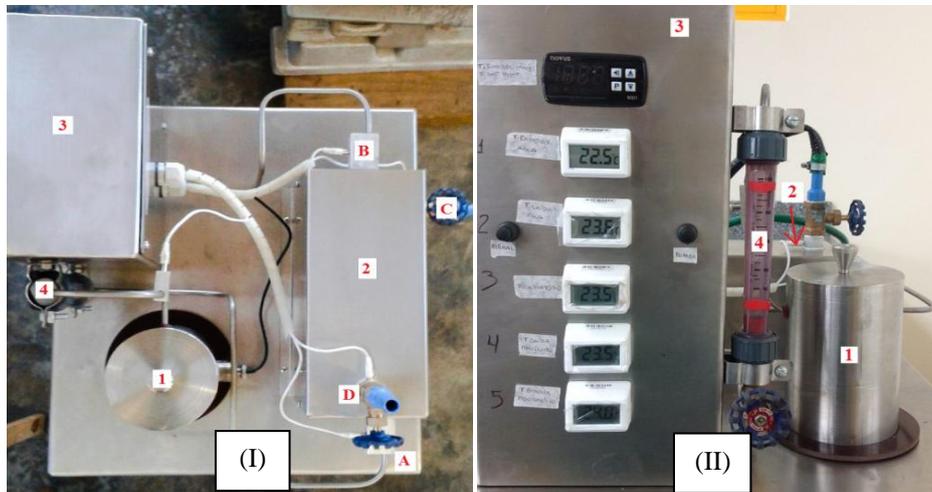

Figure 2 – (I) Superior view of the system; (II) Lateral view highlighting the central of temperature control.

The flow diagram of the system is illustrated in Fig. 3. Nanofluid is kept in the reservoir (2) that has a thermocouple inside. Beneath this reservoir there is a thermal resistance that heats up the nanofluid until 40°C. The temperature is shown at the central of temperature control (A) that is put in action by a relay (B) when the nanofluid starts to cool down, thus maintaining the temperature approximately constant. After the heated, nanofluid goes to the pump (D), that has a by-pass (E) to adjust the flow. At the pump outlet a thermocouple measures the inlet temperature of the hot fluid ($T_{h,i}$); then the nanofluid enters the heat exchanger (1) in order to transfer heat to the cold fluid that passes through the shell; after that the nanofluid passes through a flow control valve (F) and then a thermocouple measures the outlet temperature of the hot fluid ($T_{h,o}$). Finally, the nanofluid returns to the reservoir completing the cycle. All the thermocouples are represented by the letter C.
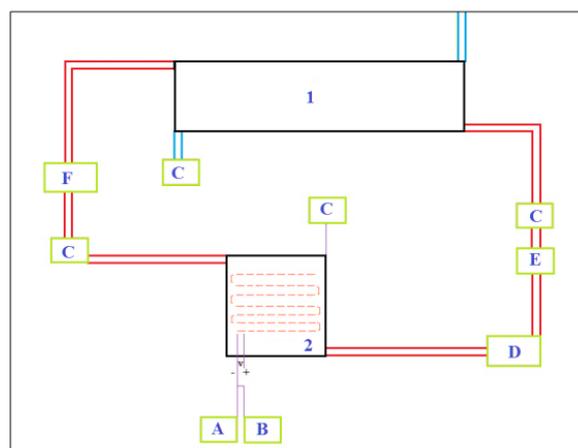

Figure 3 – Scheme of the heat exchanger system.

The cold fluid (water) comes from a common faucet and its inlet temperature ($T_{c,i}$) is measured by a conventional thermometer before entering the shell; after leaving that one, its outlet temperature ($T_{c,o}$) is measured by a thermocouple and then is discarded directly into a drain.

3.2 Preparation of gold nanofluid

Gold nanoparticles were obtained by the standard Turkevich´s method (Turkevich *et al*., 1951) using 20 mL of 1.0 mM of chloroauric acid ($HAuCl_4.3H_2O$ – Vetec) as a precursor, and 2 mL of aqueous solution of sodium citrate 1%

($Na_3C_6H_5O_7.2H_2O$ – Synth) as a reducing agent. Initially, the precursor agent was heated under magnetic stirring until its boiling then the reducing agent was added. The nanoparticles solution was gradually formed until a ruby red color, characteristic of colloidal gold. Spherical nanoparticles with average diameter of about 12 nm were obtained. The two-step method was used in the synthesis of gold nanofluid based on distilled water, with final volume 900 mL and volumetric fraction of nanoparticles ($\emptyset$) equal to 0.01222.

3.3 Properties of nanofluid

The thermal conductivity of nanofluids can be calculated by Eq. (6) (Ranga Babu *et al.*, 2017). The first part of equation considers the static conductivity (Maxwell, 1881) and the second one considers the conductivity associated to Brownian motion of nanoparticles (Koo and Kleinstreuer, 2004). The expression for the thermal conductivity of the nanofluid, k_{nf} is given by:

$$k_{nf} = k_{bf}\left[\frac{k_{np}+2k_{bf}+2(k_{np}\text{-}k_{bf})\emptyset}{k_{np}+2k_{bf}-(k_{np}\text{-}k_{bf})\emptyset}\right]+5*10^4\beta\emptyset\rho_{bf}c_{p,bf}\sqrt{\frac{\kappa_BT}{\rho_{np}d_{np}}}f, \tag{6}$$

where k is the thermal conductivity, the subscript np and bf refers to nanoparticles and the base fluid respectively, β is the thermal expansion coefficient, ρ is the density, c_p is the specific heat at constant pressure, κ_B is the Boltzmann's constant, T is the hot fluid film temperature, and f is the friction factor, which is considered equal to 1, according to Kakaç et al. (2012). The thermal expansion coefficient is given by:

$$\beta = 0.0137(100\phi)^{-0.8229} \tag{7}$$

The density and the specific heat of nanofluids were estimated by Pak and Cho (1998) and can be calculated by Eq. (8) and Eq. (9), given respectively by:

$$\rho_{nf} = (1\text{-}\phi)\rho_{bf}+\phi\rho_{np}, \tag{8}$$

$$c_{p,nf} = \frac{(1\text{-}\phi)\rho_{bf}c_{p,bf}+\phi\rho_{np}c_{p,np}}{\rho_{nf}} \tag{9}$$

The dynamic viscosity (μ) of nanofluids can be calculated by:

$$\mu_{nf} = \frac{\mu_{bf}}{1\text{-}34.87\phi^{1.03}\left(\frac{d_{np}}{d_{bf}}\right)^{-0.3}}, \tag{10}$$

which considers that an increase in volumetric fraction of nanoparticles increases the ratio between dynamic viscosity of nanofluid and dynamic viscosity of base fluid (Corcione, 2011); d_{bf} is the diameter of base fluid and it is determined by:

$$d_{bf} = \sqrt[3]{\frac{6M}{N\pi\rho_{bf,0}}}, \tag{11}$$

where M is molar mass of base fluid, N is Avogadro's constant and $\rho_{bf,0}$ is the density of base fluid at 293K.

3.4 Assessment of the overall heat transfer coefficient

A mathematical model applied to this type of heat exchanger was developed by Salimpour (2009). Geometric parameters considered in this mathematical model are showed in Fig. 4, where $2R_c$ is the coil diameter (50 mm), b is the coil pitch (5 mm), d is the tube diameter (5 mm) and D is the shell diameter (70 mm).
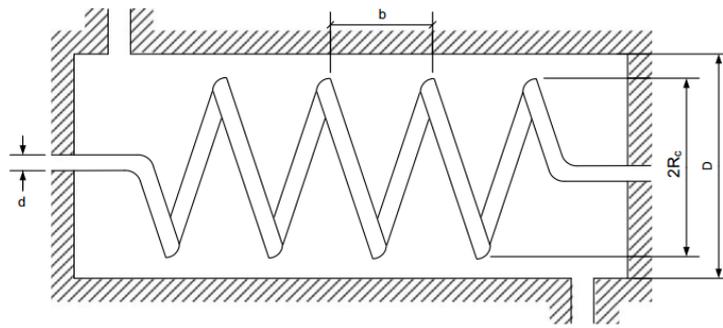
Figure 4 – Illustration of a shell and coiled tube heat exchanger

The Reynolds number (*Re*) inside the tube is calculated by Eq. (12) and outside by Eq. (13), where the subscripts *i*, *o*, *h*, *c* and *ṁ* refers to inside, outside, hot fluid, cold fluid and mass flow, respectively.

$$Re_i = \frac{4\dot{m}_h}{\pi d_{i,t}\mu_h} \qquad (12)$$

$$Re_o = \frac{4\dot{m}_c}{\pi D_{hid}\mu_c} \qquad (13)$$

D_{hid} is the hydraulic diameter, which is given by Eq. (14):

$$D_{hid} = \frac{4(V_s\text{-}V_t)/(L_s\text{-}L_t)}{\pi(D_s+d_{o,t})}, \qquad (14)$$

where *V* is the volume and *L* is the length. The subscripts *s* and *t* refers to shell and tube, respectively (Jamshidi *et al.*, 2013).

The Prantl numbers (*Pr*) inside and outside the tube are given by (Salimpour, 2009):

$$Pr_i = \frac{c_{p,h}\mu_h}{k_h} \qquad (15)$$

$$Pr_o = \frac{c_{p,c}\mu_c}{k_c} \qquad (16)$$

Other important parameter is the Nusselt number (*Nu*). For the fluid inside the tube (nanofluid) it is given by (Jamshidi et al., 2013):

$$Nu_i = 0.152De^{0.431}\gamma^{-0.277}Pr_i^{1.06}, \qquad (17)$$

where *De* is the Dean number given by:

$$De = Re_i\sqrt{\frac{d_{i,t}}{2R_c}} \qquad (18)$$

and *γ* is a dimensionless pitch given by (Salimpour, 2009):

$$\gamma = \frac{b}{\pi 2R_c} \qquad (19)$$

The Nusselt number for the fluid outside the tube is calculated by Eq. (20) (Salimpour, 2009):

$$Nu_o = 19.64Re_o^{0.513}\gamma^{0.938}Pr_o^{0.129} \qquad (20)$$

The heat transfer coefficient is given by (Salimpour, 2009):

$$h_i = \frac{Nu_i k_h}{d_{i,t}}, \qquad (21)$$

$$h_o = \frac{Nu_o k_c}{D_{hid}}.$$ (22)

Finally, the overall heat transfer coefficient can be calculated by:

$$\frac{1}{U_o} = \frac{A_o}{A_i h_i} + \frac{A_o ln(d_o/d_i)}{2\pi k_t L} + \frac{1}{h_o},$$ (23)

where A is the surface area of heat transfer and d is the tube diameter (Salimpour, 2009).

4. PRELIMINARY RESULTS AND DISCUSSION

Tests of heat exchange were made using water as cold fluid at constant flow rate 34.55 L/h and average inlet temperature of 26°C.

Two types of hot fluid were used in order to compare their properties such as density (ρ), thermal conductivity (k), specific heat (c_p) and dynamic viscosity (μ) as well as their overall heat transfer coefficient (U): gold nanofluid and its base fluid (distilled water). The flow rate varied from 25 to 50 L/h with increment rate of 5 L/h. The average inlet temperature was 40°C.

The software Engineering Equation Solver - EES® was used to obtain the base and the cold fluid properties, and to calculate all parameters previously described.

Figure (5) shows U as a function of Reynolds number; it clearly states that gold nanoparticles when suspended in the water enhanced the heat transfer properties of the nanofluid. The condition of flow was turbulent regime in both cases. Reynolds number of water varied from 3147 to 6368; for nanofluid ranged from 3456 to 6956. The overall heat transfer coefficient obtained by nanofluid varied from 2868 to 3449 W/m²K, and obtained by distilled water ranged from 2749 to 2849 W/m²K.
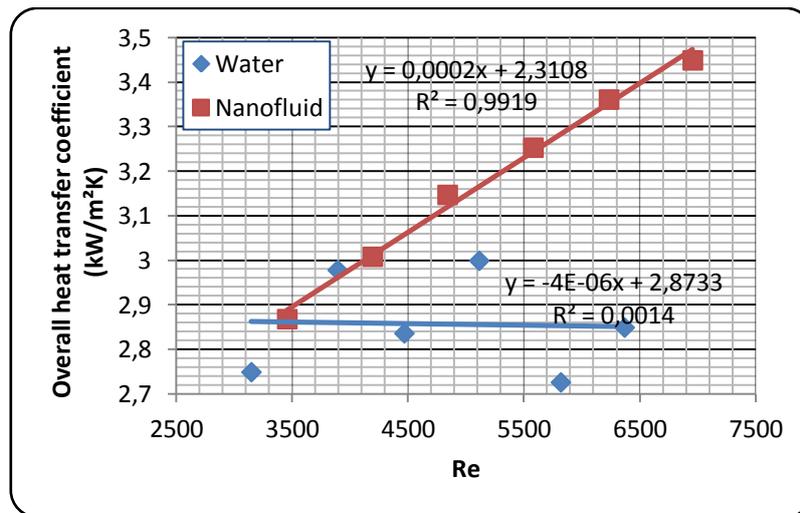

Figure 5 – Overall heat transfer coefficient as a function of Reynolds number

It was observed a growth in the overall heat transfer coefficient with the increase of flow rate for the nanofluid, and its average value was 11% greater than the value from water. Kumar et al. (2014) realized an enhancement of 18% of the overall heat transfer coefficient of Al_2O_3/water nanofluid, at nanoparticles volumetric fraction of 0.1%, in comparison with its base fluid, in turbulent flow in helically coiled tube; and Kumar et al. (2013), observed an enhancement of 7% at similar conditions, except the flow regime, that was laminar.

In the present work, the water's behavior was non-linear; as a consequence, the differences between specific heat of the nanofluid and of the base fluid were non-linear. Generally, a decrease in specific heat is expected from aqueous nanofluids compared to the base fluid, due to the high specific heat of the water (Riazi et al., 2016). Future tests should be done in order to investigate it better.

The results of dynamic viscosity of nanofluid and distilled water were shown in Fig. 6. Nanofluid viscosity was in average 13% greater than that of the base fluid.
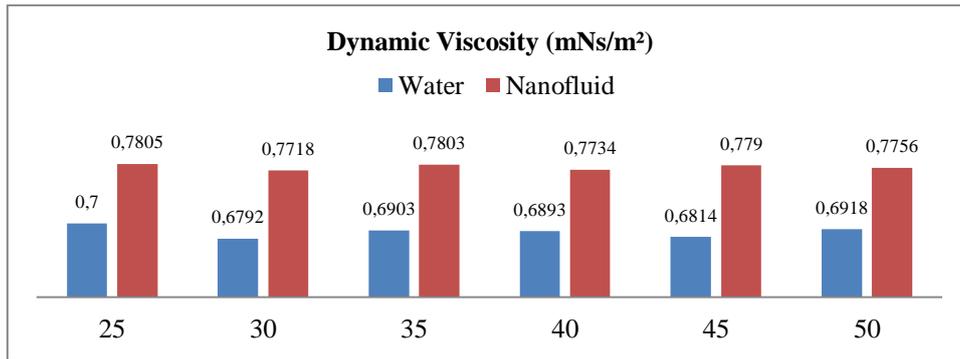
Figure 6 – Dynamic viscosity as a function of flow rate (L/h)

An augmentation in viscosity can be inconvenient when it requires a higher pumping power, but this fact can be compensated because nanofluids exhibits other thermal properties significantly enhanced, such as thermal conductivity and heat transfer coefficient. Figure 7 displays the values of thermal conductivity of distilled water and nanofluid. The thermal conductivity of the nanofluid was in average 25% greater than that of the base fluid (distilled water); Liu et al. (2006) observed an enhancement up to 23.8% for copper/water nanofluid at a nanoparticles volumetric fraction of 1%, compared to the base fluid.
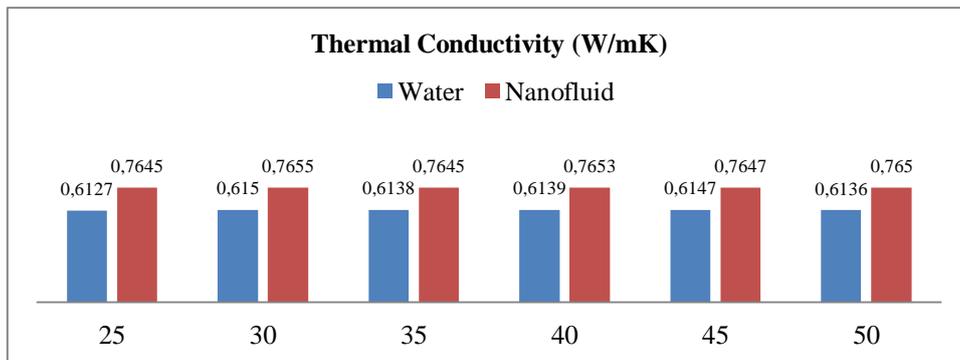

Figure 7 – Thermal conductivity as a function of flow rate (L/h)

There are few studies on metallic nanofluids in the literature, possibly due to their higher price and difficult dispersion in confrontation with non-metallic nanofluids. Moreover, dissimilar conditions such as preparation method, size and purity of nanoparticles, measurement equipments, scale of agglomeration, among others, generate a lack of consistency on results from different researchers about viscosity of nanofluids, and it is difficult to establish an analysis based on them (Murshed and Estellé, 2017). For this reason, the comparison between performances of nanofluid and its base fluid, under turbulent regime, can be estimated through Mouromtseff number (M_o), expressed by Eq. (24). If the ratio of Mouromtseff numbers of nanofluid and its base fluid is higher than 1, the nanofluid is considered efficient as coolant, and the higher the ratio, the better the efficiency of the nanofluid (Halelfadl et al., 2014; Timofeeva et al., 2011; Timofeeva et al., 2009).

$$M_o = \frac{\rho^{0.8} k^{0.67} c_p^{0.33}}{\mu^{0.47}} \ , \tag{24}$$

The results of Mouromtseff numbers of nanofluid and its base fluid, as well as ratio of them, are shown in Tab. 1.

Table 1. Results of Mouromtseff numbers

Flow rate (L/h)	Mouromtseff Number (M_o)		$M_{o,nf}/M_{o,bf}$
	Nanofluid	Base fluid	
25	103.7	82.27	1.26
30	104.3	85.19	1.22
35	103.7	80.58	1.29
40	104.2	81.36	1.28
45	103.8	77.18	1.34
50	104.0	76.83	1.35

For all flow rates Mouromtseff number ratio of nanofluid and distilled water was higher than 1, thus it indicates the possibility of apply the gold nanofluid as coolant.

5. CONCLUSIONS

The analysis of the experimental results obtained in this work allow to conclude that nanofluid showed an increase in the overall heat transfer coefficient of about 11% when compared to the value obtained with the base fluid.

The presence of nanoparticles increased the viscosity of the distilled water; nanofluid viscosity was in average 13% greater than that of the base fluid. However, this drawback was compensated by the average thermal conductivity of the gold nanofluid, 25% greater than that of the base fluid (distilled water). Besides, the ratio of Mouromtseff numbers of nanofluid and its base fluid is higher than 1 for all flow rates, and estimates the potential of the gold nanofluid to be applied as fluid in a heat exchanger system.

Preliminary results indicate that nanofluids can be more efficient than distilled water when used in a shell and coiled tube heat exchanger.

6. ACKNOWLEDGEMENTS

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