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HEAT EXCHANGES INTENSIFICATION THROUGH A FLAT PLAT SOLAR COLLECTOR BY USING NANOFLUIDS AS WORKING FLUID

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ABSTRACT

This paper illustrates how practical application of nanofluids as working fluid to enhance solar flat plate collector efficiency. A numerical investigation of laminar convective heat transfer flow throw a solar collector is conducted, by using CuO-water nanofluids. The effectiveness of these nanofluids is compared to conventional working fluid (water), wherein Reynolds number and nanoparticle volume concentration in the ranges of 25–900 and 0–10 % respectively. The effects of Reynolds number and nanoparticles concentration on the skin-friction and heat transfer coefficients are presented and discussed later in this paper. Results show that the heat transfer increases with increasing both nanoparticles concentration and Reynolds number, where nanofluid CuO-water gives best improvement in terms of heat transfer.

Keywords: *flat plat solar collector, nanofluids, heat transfer, CuO nanoparticles.*

1. INTRODUCTION

Heat transfer is a very important phenomenon in energy systems, the field of renewable energies is one of these systems, it take a major interest around the world because it will be replaces the conventional energy sources (Hussein, 2015). Today**,** the most of the research is oriented towards this axis, where the improvement of the efficiency of these thermal systems based mainly on the energy efficiency of solar collectors (Chabane et al, 2013). In order to improve their efficiency and in addition to conventional methods such as geometric modification (Karwa and Chauhan, 2010; Bhushan and Singh, 2011; Lanjewar et al. 2011; Promvonge et al., 2012; Garcia et al., 2013; Azmi et al., 2014; Yadav et al.(2013; Nanan et al., 2014; Sandhu et al., 2014) and Solar selective coatings (Juang et al., 2010; Nuru et al., 2012; Khamlich et al., 2013; Kumar et al., 2014), a new technology was used based on the replacement of the classical heat transfer fluid by another class having higher thermal properties (Amoura et al., 2012) .

The idea to improve thermo-physical properties of conventional fluids is adding solid particles with a high thermal characteristics and nano-size, inside basic fluid (working fluid). This new generation of fluids named Nanofluids; this term was introduced first by Choi (1995), and usually stills use to characterize this colloidal suspension type. After these initiative studies of the ARGONNE laboratory group (Eastman et al., 1997; Choi, 2001; Keblinski et al., 2002), several experimental and numerical studies have been carried out in order to analyze and understand the hydrothermal behavior of this new class of working fluids.

First, an experimental studies are done on the thermo-physical properties of nanoluides (Eastman, 2001; Choi et al., 2001; Patel et al., 2003), which observe that the thermal conductivity of nanofluids are very high compared to that of conventional fluid and exceed those predicted by conventional approaches. After these results, Keblinski et al. (2002), Jang and Choi (2004) and then Evans (2006) developed thermal conductivity classic model of the nanofluid, where they introduced the effect of the Brownian motion and nanoparticles agglomeration effect, which explain the important improvement.

In recent years, a wide research work has been done in this research area to understand the thermo-hydraulic behavior of this new heat transfer fluid generation, they proved that nanoparticles suspension within the conventional heat transfer fluid increases its thermal conductivity and consequently improves the heat transfer (Rawi et al., 2017; Boulahia et al., 2017; Wakif et al., 2017; Boulahia et al., 2017; Ramesh and Gireesha, 2017).

Al-Rashed et al. (2017) studied the effect of magnetic field on natural convection inside a cubical cavity filled with CNT - water nanofluide; they found that for all the Rayleigh numbers the Bejan number increase by increasing nanoparticles volume concentration. Mixed convective incompressible flow of nanofluid through a vertical channel in presence of magnetite field has been investigated by Srinivasacharya and Shafeeurrahaman (2017), in the same context, Sravan Kumar and Rushi Kumar (2017), Govindaraju et al. (2017) and Sathish Kumar et al. (2017) carried out a numerical study on free convection heat transfer of nanofluids over a stretching sheet in the presence of a uniform magnetic field.

All the previous studies show that the improvement of the heat transfer coefficient is related to the presence of nanofluids. Despite the fact that there is no commercial solar collector available that uses these nanofluids, there are experimental studies that confirm its effectiveness by adding nanoparticle to improve the thermal efficiency, Natarajan et al. (2009) found that if the nanofluids are used as working fluids, the energy efficiency of solar water heaters increases significantly compared to conventional fluids. On the other hand, the dispersion of the nanoparticles gives an important increase in the heat transfer coefficient (Faizal et al., 2013; Sokhansefat et al., 2014; He et al., 2015; Menbari et al., 2016; Nasrin et al. 2016). Hence, they found that it is

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possible to reduce the exchange surface and improve the efficiency of these devices, which confirms the results of Natarajan et al. (2009). Maouassi et al. (2017) carried out a numerical study of forced convection heat transfer through a flat plate solar collector, the geometry is tested for 3D case, where water and SiO_2-H_2O nanofluid are used as a working fluid, the results obtained show that the increase of nanoparticles volume concentration gives an improvement in heat transfer rate reach 13.7% for 10% of nanoparticles volume concentration.

In this paper, we present a comparative study of heat transfer of nanofluid (CuO-water) flows through a solar flat plate collector. In the first part, we describe the problem and present the boundary conditions. After grid independence study, then we validated the results by comparing them with previous literature reports.

The key part of this work is to involve the simulation with the interpretation of numerical results obtained for this case, where the effect of nanoparticles are shown for various volume concentration and Renolds number, and studied systematically. At the end, this paper is footed by conclusion which summarizes the main results obtained.

2. PROBLEM DESCRIPTION AND MATHEMATICAL FORMULATION

The flow domain consists of an absorber plate and circular absorber tube. The absorber plate is covered with glass plate with an air gap. Design parameters and fixed geometric parameters have been taken similar to Maouassi et al. (2017), as indicated in Table. I.

Table 1 Design parameters of the solar collector.

	absorber plate (mm)	*Tube (mm)*	*Glass plate (mm)*	*air gap (mm)*
length	1 000	1010	1000	1000
wide	150	-	150	150
diameter	-	10	-	-
thickness	2	1	2	3

The heat transfer of nanofluide under laminar flow inside the absorber tube, as shown in Fig. 1, the flow is assumed steady and the fluid possesses uniform axial velocity V_0 and temperature T_0 profiles at the inlet of tube.

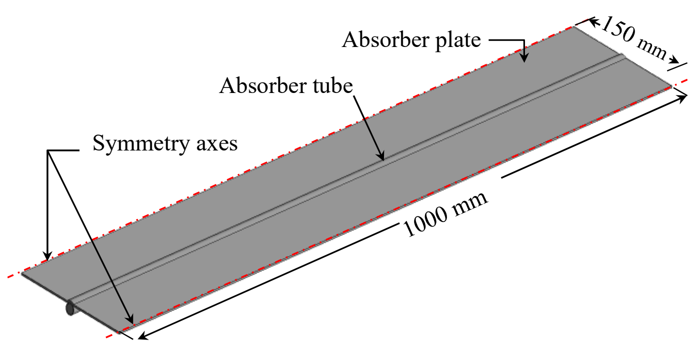

Fig. 1 Geometrical representation of the studied problem

The nanoparticles used in practical applications of heat exchange are very thins (≤ 50 nm), then the liquid-particle mixture may easily be fluidized and considered as a homogenous single-phase fluid (Xuan et al., 2000). By assuming negligible thermal equilibrium and slip between the phases, the apparent thermal properties of nanofluids can be estimated by using classical relationships of two-component mixture (Pak et al., 1998), and the mixture considered as a conventional single-phase fluid (Xuan et al., 2000) .

Under the previous assumptions, the governing equations including the two-dimensional transient equations of the momentum, energy and continuity for an incompressible flow are expressed in the following format:

$$\rho_{nf}\frac{\partial\left(u_i u_j\right)}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu_{nf}\frac{\partial u_i}{dx_j}\right) \tag{1}$$

$$\frac{\partial}{\partial x_j}\left(u_j T\right) = \frac{k_{nf}}{\rho_{nf}\, Cp_{nf}}\frac{\partial}{\partial x}\left(\frac{\partial^2 T_l}{dx_j^2}\right) \tag{2}$$

$$\frac{\partial}{\partial x_j}\left(\rho_{nf}\, u_j\right) = 0 \tag{3}$$

The physical properties of nanofluid, nanofluid density ρ_{nf} (Zhou et al., 2008), viscosity μ_{nf} (Brinkman et al., 1952) , Thermal conductivity k_{nf} (Maxwell et al., 1891) and Nanofluid specific heat Cp_{nf} (Xuan et al., 2000) are given with the below equations:

$$\rho_{nf} = \rho_s\phi + \rho_f\left(1-\phi\right) \tag{4}$$

$$\mu_{nf} = \mu_{nf}\left(1+2.5\phi\right)\mu_f \tag{5}$$

$$k_{nf} = \frac{k_s - 2k_f - 2\phi\left(k_f - k_s\right)}{k_s + 2k_f + \phi\left(k_f - k_s\right)}k_f \tag{6}$$

$$Cp_{nf} = \frac{\rho_s Cp_s\,\phi + \rho_f Cp_f\left(1-\phi\right)}{\rho_{nf}} \tag{7}$$

The thermo-physical properties of the nanofluid compositions used in this study are listed in Table 2.

For boundary conditions and operating parameters are taken similar to Maouassi et al., 2017). For inlet 'velocity inlet' boundary condition is specified and an 'outflow' condition is specified at the outlet for the water domain. Wall boundary conditions used to bound fluid and solid regions. The interface between water and absorber tube is defined as coupled wall condition. In the top of collector a constant solar heat flux equal to $800 W/m^2$ are imposed.

Table 2 Thermo-physical properties of nanofluide Amoura et al., 2012).

Thermo-physical properties	*Base fluid (Water)*	*Nanoparticles (CuO)*
ρ (kg/m^3)	1000	6350
k (W/ m K)	0.613	69
Cp (J/ kg.K)	4183	535
μ (kg m^{-1} s^{-1})	1.003 10^{-3}	-

The thermo-physical quantities of principal interest are the skin friction coefficient (C_f), and the local Nusselt number (Nu_x).

2.1 The Skin Friction Coefficient

The skin friction coefficient for the fully developed laminar flow in a circular tube is given by

$$C_f = \tau_s\frac{\rho u_m^2}{2} \tag{8}$$

Where τs is local shear stress, and um is the mean velocity given by

$$u_m = \frac{\int_0^R \rho u\, 2\pi\, rdr}{\int_0^R \rho\, 2\pi\, rdr} = \frac{2}{R^2}\int_0^R u\, rdr \tag{9}$$

2.2 Local Nusselt Number

The Nusselt number, which represents the dimensionless temperature gradient at the surface (Eq. 14) and provides a measure of the convection coefficient, is defined as

$$Nu_{x,nf} = \frac{h_{x,nf}\, D}{k_f} \tag{10}$$

Near the wall of tube absorber, the surface heat flux is equal to the convective flux the fluid (no-slip condition), which is expressed by Newton's law of cooling

$$q''_{sf} = q''_{conv} = h_{x,nf}\left(T_{x,w} - T_{x,m}\right) \tag{11}$$

Where $T_{x,\ w}$, $T_{x,\ m}$ are the local wall temperature and mean fluid temperature respectively, the mean temperature of a fluid flowing in a circular pipe of radius R is given by

$$T_m = \frac{\int_0^R \rho Cpu T\, 2\pi r dr}{\int_0^R \rho Cpu\; 2\pi r dr} = \frac{2}{u_m R^2}\int_0^R uT\, rdr \tag{12}$$

The local heat flux at the surface is obtained by applying Fourier's law to the fluid at r = R

$$q''_{sf} = k_{nf}\left(\left.\frac{\partial T}{\partial r}\right|_{r=R}\right)_{sf} \tag{13}$$

By combining the foregoing equations (Eq. 11 and 13), we obtain the local convection coefficient

$$h_{x,nf} = -\frac{k_{nf}}{T_{x,w} - T_{x,m}}\left(\left.\frac{\partial T}{\partial r}\right|_{r=R}\right)_{sf} \tag{14}$$

And the Nusselt number of nanofluid are given by

$$Nu_{x,nf} = -\frac{2k_{nf}}{k_f}\frac{R}{T_{x,w} - T_{x,m}}\left(\left.\frac{\partial T}{\partial r}\right|_{r=R}\right)_{sf} \tag{15}$$

The rate of the improvement in the heat transfer coefficient is calculated as follows

$$Nu(\%) = -\frac{Nu_{nf} - Nu_{water}}{Nu_{water}} \times 100 \tag{16}$$

3. NUMERICAL METHOD AND CODE VALIDATION

To conduct numerical simulation, the computational domain was meshed using finite volume method, by using a preprocessor of FLUENT called GAMBIT (version 2.4.6). This simulation based on the solver of steady state implicit pressure integrated in FLUENT software (version 6.3). And the Governing partial differential equations, for mass and momentum, are solved for the steady incompressible flow. The velocity-pressure coupling is affected through SIMPLE algorithm developed by Patankar (1980). Second order upwind schemes were chosen for the solution schemes, and laminar flow condition was used.

3.1 Meshing and Grid Independence Study

The computational domain was meshed and done by using appropriate grid cells, with suitable size. Three dimensional computational domains are built and used, as shown in Figure 2. To solve the previous system of governing equations (1-3), the computational domain was meshed with the finite control volume method which has been successfully used by several authors (Beyers et al., 2001) and Maïga et al. (2004).

Several non-uniform grids have been thoroughly tested to ensure the accuracy and the consistency of numerical results, which has shown that the 40 x 60x 500 non-uniform grid (32, 24 and 155 nodes, respectively, along the radial, tangential and axial directions) appears satisfactory for the absorber tube (Figure 3); with refining mesh near the tube wall and the entrance region.

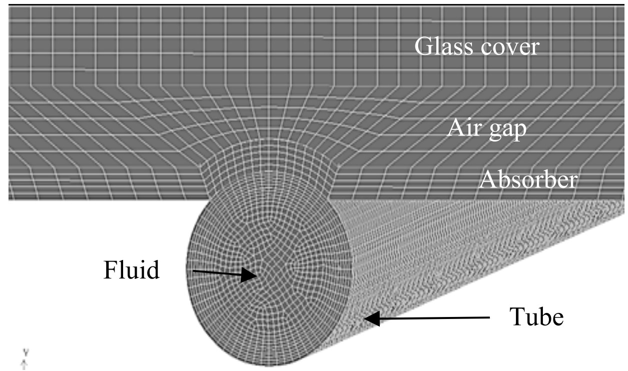

Fig. 2 Mesh validation curves for the outlet temperature evolution.

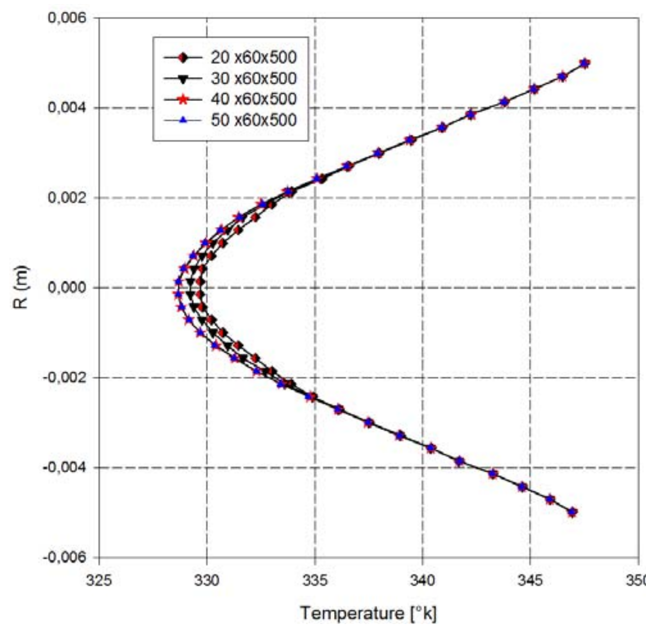

Fig. 3 Mesh validation curves for the outlet temperature evolution.

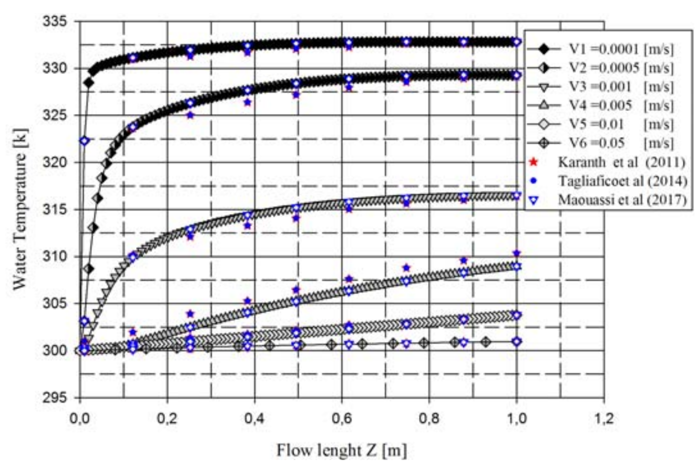

Fig. 4 Comparison of fluid temperature distribution between the present results and those reported in the literature.

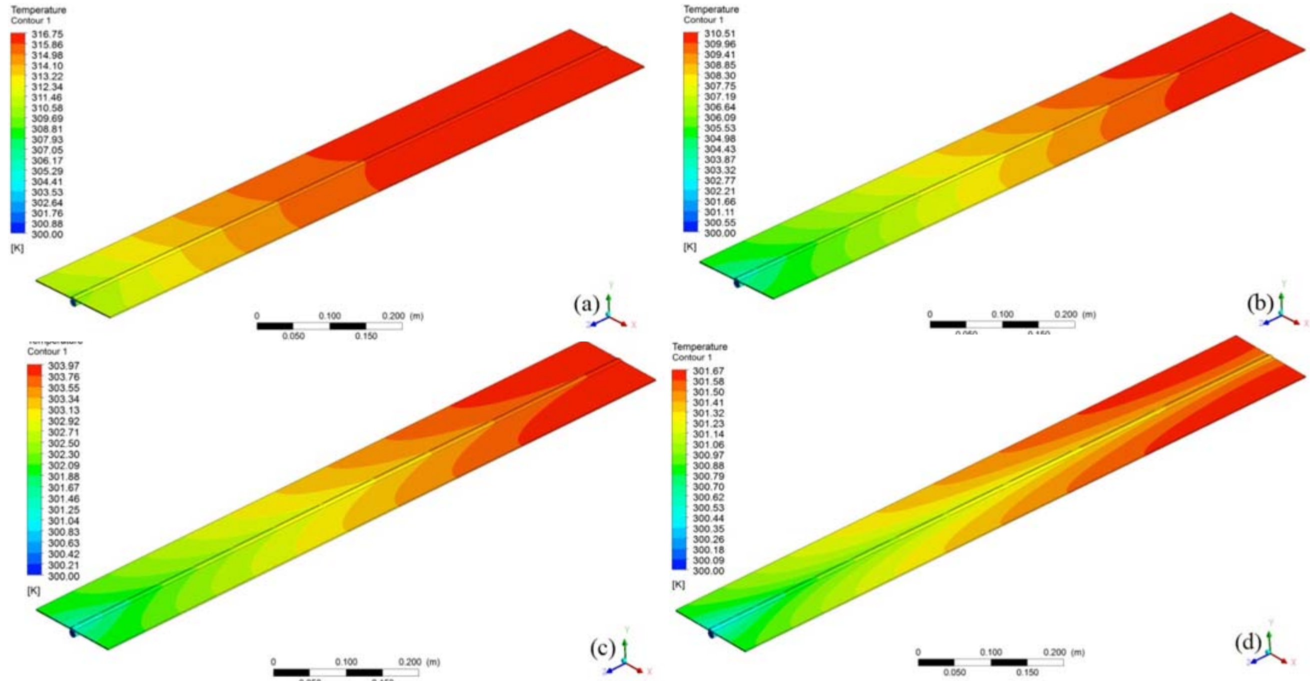

Fig. 5 Temperature distribution on the absorber for (a) V= 0.001 m/s, (b) V= 0.005 m/s, (c) V= 0.01 m/s, (d) V= 0.05 m/s.

3.2 Code Validation

In order to validate the present results, the axial temperatures evolution of the working fluid along the absorber tube presented by Karanth et al. (2011), Tagliafico et al. (2014) and Maouassi et al. (2017).

The comparison results are exhibited in Fig. 4. It can be seen from this results that, a very good matching between our results and those of the authors; which gives confidence in the numerical method employed Fig. 5 illustrate the temperature evolution distribution on the absorber plate for different values of inlet velocity (V= 0.001 to 0.05 m/s) presented by Maouassi et al. (2017).

4. NUMERICAL METHOD AND CODE VALIDATION

We study the nanofluids heat transfer flow inside an absorbent tube, where he heated by the absorbent plaque. We consider the flow regime as laminar, and varying the number of Reynolds between 25 and 900, for the different nanoparticles concentrations (1%, 3%, 5%, and 10%). Then we plot the results, pressure drop coefficient and Nusselt number.

4.1 Pressure Drop Coefficient

Key parameter of the present study is the volume concentration of nanoparticles. Fig. 6 exhibits the effect of the nanoparticles volume percentage "ϕ" on the skin friction coefficient, for each Reynolds number.

It is clearly noted that there is a slight increase in skin friction coefficient (Cf) when the Reynolds number is low (Re ≤ 100), this slight increase in Cf, it is due to the presence of the nanoparticles and lower fluid flow velocities, but when the Reynolds number exceeds 100 this problem doesn't occur.

Fig. 7 represent the skin friction coefficient variations versus Reynolds numbers, it will be noted that for the same Reynolds number, and despite the variation in the nanoparticles concentration volume; the (C_f) value is almost identical, this confirm and supports the previous result in Fig. 5. These results, therefore, reflect the insignificant influence of the nanoparticles volume concentration on the skin friction coefficient (C_f).

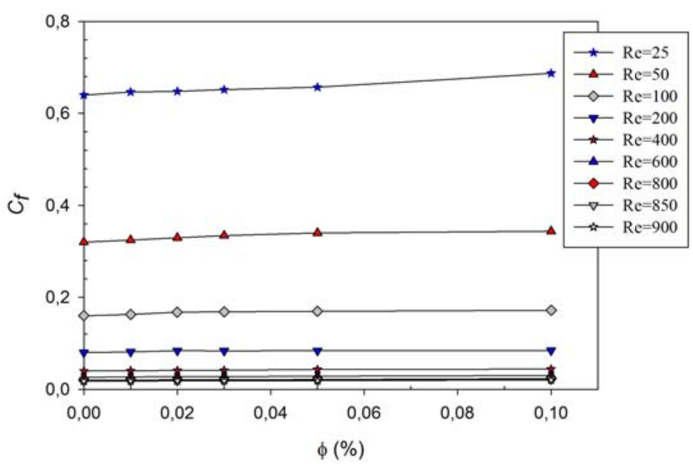

Fig. 6 Skin friction coefficient Variations versus volume fraction of CuO nanoparticles.

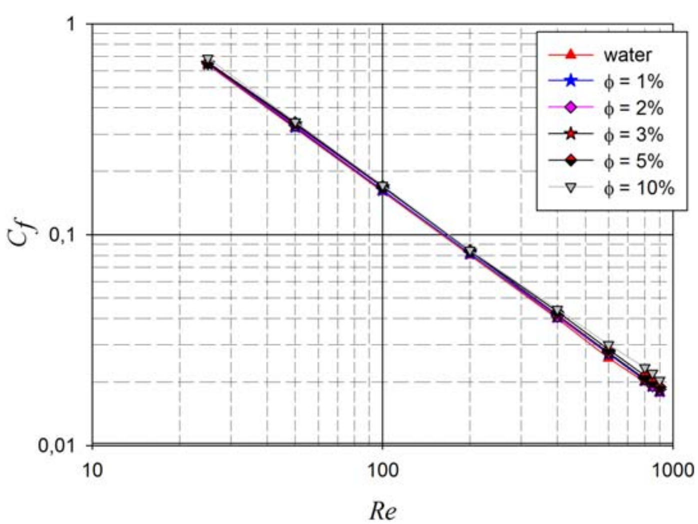

Fig. 7 Skin friction coefficient Variations versus Reynolds numbers.

4.2 Nusselt Number

In this section, the numerical results of forced convection heat transfer using CuO-water nanofluid inside a flat plate solar collector are discussed. Fig. 8, represents Nusselt number evolution of "Nu" as a function of the nanoparticles volume fraction "ϕ". We note a significant increase of Nusselt number with the increase in CuO nanoparticle volume concentration for all its value.

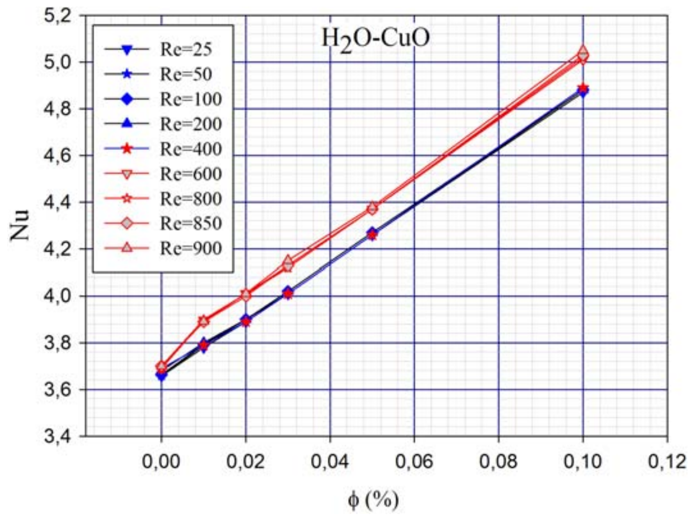

Fig. 8 Heat transfer coefficient Variations versus CuO nanoparticles volume concentrations.

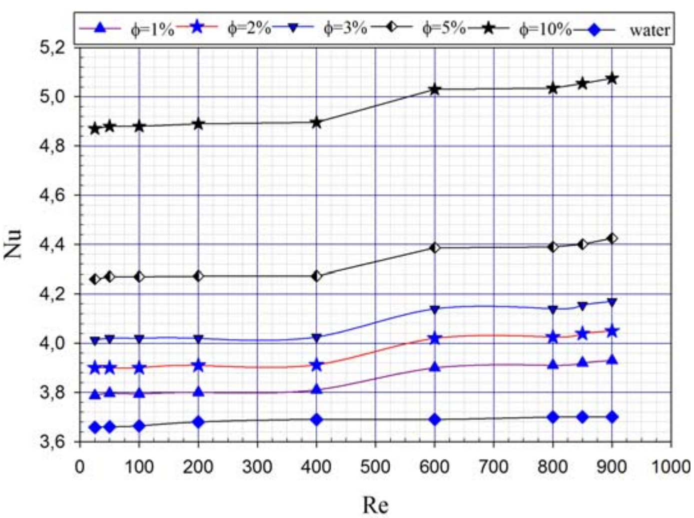

Fig. 9 Heat transfer coefficient Variations versus CuO nanoparticles volume concentrations.

The effect of Reynolds number on heat transfer coefficient (Nu) is illustrated in Fig. 8 for both pure water and the CuO-water nanofluide, a slight increase in Nusselt when the Reynolds number is low ($Re \leq 400$), but when the Reynolds number exceeds this value, each increase in Reynolds number is accompanied with an increase brusque in Nusselt number, this increase in Nu is due to the presence of nanoparticles Brownian-motion for higher fluid flow velocities, consequently the heat transfer intense through the working fluid increase.

Fig. 9 represents the comparison in term of rate improvement of heat transfer coefficient for various CuO nanoparticles volume concentrations. We notice then, a considerable increase of the improvement rate compared to pure water which can reach an average value of 34.7 % for ϕ= 10 %.

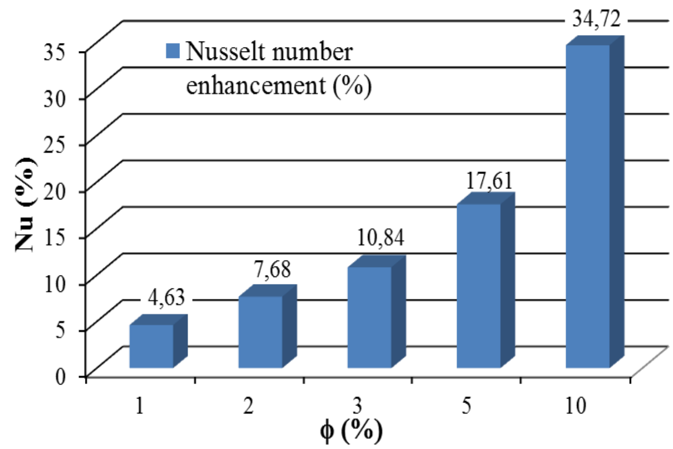

Fig. 10 The improvement rate of heat transfer coefficient versus CuO nanoparticles volume concentrations.

5. CONCLUSIONS

In this paper, the flow and heat transfer characteristics of CuO-water nanofluid for arrange of the Reynolds number (25to 900) with a wide range of volume concentration (0 to 10%) are studied numerically.

We can confirm from analyze of thermal-hydraulic properties of nanofluids:

- The presence of nanoparticles in the base fluid (pure water) increases significantly Nusselt number.
- Nusselt number is growing with the increase of volume fraction, the increase in heat coefficient results improve the heat transfer through the energy systems compared to the base fluid (pure water) case.
- The gain in heat is accompanied by a slight increase in skin friction coefficient for the low Reynolds numbers due to the presence of the nanoparticles and lower flow velocities, but when the Reynolds number exceeds 100 this problem doesn't occur.

The results show that, Nusselt number and heat transfer coefficient, of nanofluid are strongly dependent on nanoparticle and increase by increasing of the volume concentration of nanoparticles, and the insignificant influence of the nanoparticles volume concentration on the skin friction coefficient.

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NOMENCLATURE

c_f	skin friction coefficient
c_p	specific heat (J/kg·K)
h	Heat transfer coefficient (W/m^2·K)
k	thermal conductivity (W/m·K)
Nu	Nusselt number
q''	heat flux (W/m^2)
Re	Reynolds number
R	tube radius (m)
T	temperature (K)
u	axial velocity (m/s)
Greek Symbols	
ϕ	Volume concentration (%)
μ	Viscosity (kg m^{-1} s^{-1})
ρ	density (kg/m^3)
Subscripts	
f	fluid

nf nanofluid
sf surface
s solid

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