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Analysis of a Double Pipe Heat Exchanger Performance by Use of Porous Baffles and Nanofluids

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Abstract**—**The present work is a numerical simulation of nanofluids flow in a double pipe heat exchanger provided with porous baffles. The hot nanofluid flows in the inner cylinder, whereas the cold nanofluid circulates in the annular gap. The Darcy-Brinkman-Forchheimer model is adopted to describe the flow in the porous regions, and the governing equations with the appropriate boundary conditions are solved by the finite volume method. The results reveal that the addition of metallic nanoparticles enhances the rate of heat transfer in comparison to conventional fluids but this augmentation is accompanied by an increase in pressure drop. The highest heat exchanger performances are obtained when nanoparticles are added only to the cold fluid.

Keywords**—**Double pipe heat exchanger, Nanofluids, Nanoparticles, Porous baffles.

I. Introduction

HEAT exchangers are the key components of any thermal system. They are found in thermal power plants, refrigeration and air conditioning systems, as well as automobile industry. Several techniques have been developed to improve the thermal-hydraulic performance of such devices which resulted on the reduction of their size and the cost of operation. These methods can be classified either as active methods which require an external power such as electrical or acoustic fields and surface vibrations, or as passive methods which employ, for the enhancement of heat transfer, surfaces of special geometries (fins and baffles) or additives for fluids.

In recent decades a growing interest has been focused on the technique using porous medium due to the fact that this kind of structure is encountered in multiple engineering applications such as cooling of electronic circuits, drying processes, filtration, thermal insulation, geothermal systems, ground water and oil flow, as well as heat exchangers. Several research studies have been conducted in this context. An analytical solution was obtained by Chikh et al. [1] for the problem of forced convection under fully developed conditions in an annular duct partially filled with a porous layer attached to the inner tube subjected to a constant heat flux. A similar problem was treated by the same authors [2] but with an isothermal boundary condition and by using the Darcy-Brinkman-Forchheimer model. They concluded that porous material can be used for insulation or enhancement of heat transfer according to its thermophysical properties. In another work, Chikh et al. [3] analyzed the effect of porous matrix addition on the hydrodynamic and thermal lengths. Guo et al. [4] found that a significant augmentation of heat transfer and a reduction of pressure drop may be achieved by use of porous medium and pulsating flow in an annulus. Alkam and Al-Nimr [5] inserted porous substrates at both sides of the inner cylinder. They found that the improvement in heat exchanger efficiency is important especially at high capacity ratios and that there is a critical value of the substrate thickness beyond which there is no substantial increase in the heat exchanger performance. A thermodynamic analysis was conducted by Allouache and Chikh [6] in a double pipe heat exchanger provided with a porous layer. They concluded that the case of an annular gap completely filled with a porous medium of high effective thermal conductivity leads to a substantial reduction of entropy generation rate. Kahalerras and Targui [7], and Targui and Kahalerras [8] showed that the use of porous fins or baffles may enhance substantially heat transfer at the expense of a reasonable pressure drop for optimal values of the porous medium properties. Using the Darcy-Brinhkman-Forchheimer model, Hashemi et al. [9] obtained an analytical solution for the problem of forced convection in micro-annulus filled with a porous medium. Two cases were considered; outer cylinder subjected to a constant heat flux and insulated inner cylinder and vice versa. Recently, Targui and Kahalerras [10] analyzed numerically the performance of a double pipe heat exchanger using porous baffles and pulsating flow. They found that the highest heat transfer rates are obtained when only the flow of the hot fluid is pulsating.

Another way to optimize the operation of thermal systems is the suspension of metallic nanoparticles in conventional fluids to improve their thermal properties such as thermal conductivity; these are the nanofluids. Many studies have been performed in this field, however, the works interested by the combined effects of nanofluids and porous medium are less numerous. Ghazvini and Shokouhmand [11] performed an analytical and numerical study on the use of water-CuO nanofluid as a coolant in a microchannel heat sink. Two approaches were used: fins model and porous medium approach. The problem of mixed convection boundary layer flow along a vertical plate embedded in a porous medium saturated by nanofluids was performed by Ahmad and Pop [12]. Different types of nanoparticles were used: Cu, Al_2O_3 and TiO_2. Cimpean and Pop [13] studied the fully developed mixed convection flow in an inclined porous channel saturated with different kind of nanofluids (water-Cu, water-Al_2O_3 or water-TiO_2). They concluded that nanofluids increase

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significantly the heat transfer rate even at small additions of nanopaticles in the base fluid. Mixed convection of a nanofluid in a vertical partially porous channel was investigated by Hajipour and Dehkordi [14] taking into account viscous dissipation. The model used for the nanofluid incorporates the effects of Brownian motion and thermophoresis. The purpose of the work of Zafiri et al. [15] was to perform a thermal-hydraulic analysis of nanofluids (water-Al_2O_3 and water-TiO_2) as coolants in VVER-1000 reactor core using porous medium approach. Conjugate heat transfer, natural convection-conduction, in a porous square cavity filled with nanofluids (water-Cu, water-Al_2O_3 and water-TiO_2) and heated by a triangular thick wall was treated by Chamkha and Ismael [16]. They found that heat transfer is improved or reduced with the increase of volume fraction according to the values of Rayleigh number and triangular wall thickness for which there is an optimum value leading to a maximum average Nusselt number.

The present work is a contribution to previous studies on the improvement techniques of heat exchangers performances. For this purpose, we undertook a numerical analysis of thermal-hydraulic performance of a double pipe heat exchanger provided with porous baffles and in which circulate one or two nanofluids.

II. Mathematical Formulation

The physical model (Fig. 1) is a parallel-flow double pipe heat exchanger of length ℓ and inner and outer radius r_i and r_o respectively. Porous baffles of porosity ε, width w, height h_p and spacing s is inserted in the annular gap. The outer cylinder is thermally insulated and the length behind the last baffle is chosen high enough so that fully developed conditions at the heat exchanger exit can be assumed. The hot and cold nanofluids (water – nanoparticles) enter the inner and annular gap respectively at uniform velocities U_{ih} and U_{ic} with constant temperatures T_{ih} and T_{ic}. Nanoparticles are added to both base fluids (NF1) or only to one of them (NF2: hot fluid and NF3: cold fluid).

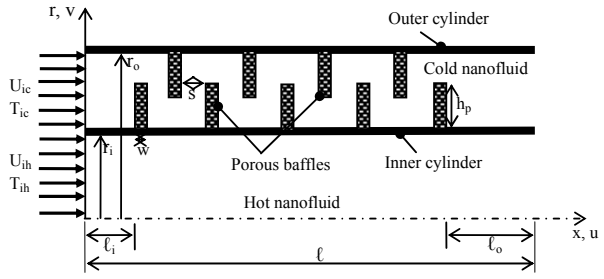


Fig. 1 Physical domain

In order to simplify the problem some assumptions are made: the flow is two-dimensional, laminar and in steady state with no internal heat generation and neglecting viscous dissipation. Both the base fluid (water) and nanoparticles are in thermal equilibrium and there is no slip between them. The thermophysical properties of the two nanofluids (hot and cold) are assumed to be the same and constant. The nanofluid is Newtonian and incompressible, and the porous medium considered as homogeneous and isotropic is in local thermal equilibrium with the nanofluid. It is also assumed that the thickness of the inner cylinder, of good conducting material, is very thin and its thermal resistance is neglected.

Under the above assumptions, and adopting the nanofluid model proposed by Tiwari and Das [17] with the Brinkman-Forchheimer extended Darcy model [18], the governing equations can be written as follows:

Continuity:

$$\frac{\partial U}{\partial X}+\frac{1}{R}\frac{\partial (RV)}{\partial R}=0 \tag{1}$$

Momentum:

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial R}=-\frac{\partial P}{\partial X}+\frac{R_\mu}{Re}\frac{R_{\mu nf}}{R_{\rho nf}}\left[\frac{\partial^2 U}{\partial X^2}+\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial U}{\partial R}\right)\right]-\frac{\varepsilon}{ReDa}\frac{R_{\mu nf}}{R_{\rho nf}}U-\frac{C\varepsilon^2}{ReDa}|\vec{V}|U \tag{2}$$

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial R}=-\frac{\partial P}{\partial R}+\frac{R_\mu}{Re}\frac{R_{\mu nf}}{R_{\rho nf}}\left[\frac{\partial^2 V}{\partial X^2}+\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial V}{\partial R}\right)-\frac{V}{R^2}\right]-\frac{\varepsilon}{ReDa}\frac{R_{\mu nf}}{R_{\rho nf}}V-\frac{C\varepsilon^2}{ReDa}|\vec{V}|V \tag{3}$$

Energy:

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial R}=\frac{R_k}{\varepsilon RePr}\frac{R_{knf}}{R_{(\rho c_p)nf}}\left[\frac{\partial^2 \theta}{\partial X^2}+\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial \theta}{\partial R}\right)\right] \tag{4}$$

where U and V are the dimensionless velocity components in the axial (X) and radial (R) directions, $|\vec{V}|=\sqrt{U^2+V^2}$, P is the dimensionless pressure, C the inertial coefficient and θ the dimensionless temperature. In the inner cylinder and the non porous regions of the annular gap: $\varepsilon = 1$, $R_k = 1$ and $Da \to \infty$.

These equations are associated to the following boundary conditions:

At the inlet:

$$Inner\ cylinder: U=\frac{(R_i^2-1)}{R_m}, V=0, \theta=1 \tag{5a}$$

$$Annular\ gap: U=1, V=0, \theta=0 \tag{5b}$$

At the exit:

$$\frac{\partial U}{\partial X}=0, V=0, \frac{\partial \theta}{\partial X}=0 \tag{6}$$

On the axis:

$$\frac{\partial U}{\partial R}=0, V=0, \frac{\partial \theta}{\partial R}=0 \tag{7}$$

At the inner cylinder wall:

$$\begin{cases} U=0, V=0 \\ \left.\frac{\partial \theta}{\partial R}\right|_h=\left.\frac{\partial \theta}{\partial R}\right|_c \quad (nanofluid-nanofluid) \\ \left.\frac{\partial \theta}{\partial R}\right|_h=R_k\left.\frac{\partial \theta}{\partial R}\right|_p \quad (nanofluid-porous) \end{cases} \tag{8}$$

At the outer cylinder wall:

$$U=0, V=0, \frac{\partial \theta}{\partial R}=0 \tag{9}$$

At the porous-nanofluid interfaces: Continuity of

pressure, velocity components, temperature and heat fluxes.

The following variables were used to convert the governing equations and boundary conditions to dimensionless form:

$$X=\frac{x}{D_h}, R=\frac{r}{D_h}, U=\frac{u}{\varepsilon U_{ic}}, V=\frac{v}{\varepsilon U_{ic}}, P=\frac{p}{\rho_{nf}U_{ic}^2}, \theta=\frac{T-T_{ic}}{T_{ih}-T_{ic}}$$

where $D_h = 2(r_o - r_i)$ is the hydraulic diameter and T the temperature.,

The dimensionless parameters appearing in the above equations are defined as:

$$Re=\frac{\rho U_{ic} D_h}{\mu}, Da=\frac{K}{D_h^2}, Pr=\frac{\mu C_P}{k}, R_\mu=\frac{\mu_e}{\mu_{nf}}, R_{\mu nf}=\frac{\mu_{nf}}{\mu}$$
$$R_\rho=\frac{\rho_{nf}}{\rho}, R_k=\frac{k_e}{k_{nf}}, R_{knf}=\frac{k_{nf}}{k}, R_{(\rho C_P)nf}=\frac{(\rho C_P)_{nf}}{\rho C_P}$$
$$R_r=\frac{r_o}{r_i}, R_m=\frac{\dot{m}_c}{\dot{m}_h}$$

where ρ is the density, μ is the dynamic viscosity, K is the permeability of the porous medium, C_P is the specific heat at constant pressure, k is the thermal conductivity, Re is the Reynolds number, Da is the Darcy number, Pr is the Prandtl number and ṁ is the mass flow rate. The subscripts h, c, e and nf indicate, respectively, the parameters relative to hot and cold fluids, to porous medium (effective) and to nanofluid.

Nanofluid properties

The density of nanofluid is defined as:

$$\rho_{nf} = (1-\varphi)\rho_f + \varphi\rho_s \tag{10}$$

The viscosity of the nanofluid is calculated using the Brinkman model [19]:

$$\mu_{nf} = \frac{\mu_f}{(1-\varphi)^{2.5}} \tag{11}$$

The heat capacitance of the nanofluid is given by:

$$(\rho C_P)_{nf} = (1-\varphi)(\rho C_P)\rho_f + \varphi(\rho C_P)_s \tag{12}$$

The thermal conductivity of nanofluid is obtained according to Maxwell model [20]:

$$k_{nf} = k_f \frac{k_s+2k_f - 2\varphi(k_f-k_s)}{k_s+2k_f + \varphi(k_f-k_s)} \tag{13}$$

where φ is the solid volume fraction. The subscripts f and s indicate the parameters relative to base fluid and nanoparticles.

The heat exchanger efficiency is calculated as

$$E=\frac{(\dot{m}C_P)_h[\theta_{ih}-\theta_{oh}]}{C_{min}[\theta_{ih}-\theta_{oc}]}=\frac{(\dot{m}C_P)_c[\theta_{oc}-\theta_{ic}]}{C_{min}[\theta_{ih}-\theta_{oc}]} \tag{14}$$

where $C_{min}=min[(\dot{m}C_P)_h,(\dot{m}C_P)_c]$.

The local friction factors in the inner cylinder and the annular gap are defined as:

$$f_h = -\frac{2R_{\rho nf}\frac{dP_{mh}}{dX}}{U_{mh}^2} \tag{15a}$$

$$f_c = \begin{cases} -\frac{2R_{\rho nf}\frac{dP_{mc}}{dX}}{\varepsilon^2 U_{mc}^2} & porous \\ -\frac{2R_{\rho nf}\frac{dP_{mc}}{dX}}{U_{mc}^2} & elsewhere \end{cases} \tag{15b}$$

where P_m and U_m are the dimensionless mean pressure and dimensionless mean velocity over a section.

The average friction factors are obtained by integration of the local values over the heat exchanger length:

$$f_{mh}=\frac{1}{L}\int_0^L f_h dX \quad , \quad f_{mc}=\frac{1}{L}\int_0^L f_c dX \tag{16}$$

To make a comparison with the case of fluids without nanoparticles, the following ratios are introduced:

$$\eta_E=\frac{E}{E_{\varphi=0}} \quad , \quad \eta_{fh}=\frac{f_{mh}}{f_{mh\varphi=0}} \quad , \quad \eta_{fc}=\frac{f_{mc}}{f_{mc\varphi=0}} \tag{17}$$

III. Numerical Procedure

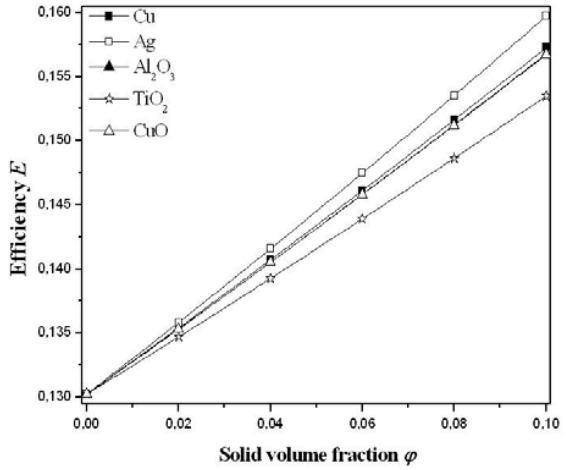
The governing equations with the associated boundary conditions are solved numerically using the finite volume method [21]. The velocity and pressure are linked by the SIMPLE algorithm, and the power law scheme is employed in the discretizing procedure to treat the diffusion and the convective terms. The obtained algebraic equations are solved using a line by line technique, combining between the tridiagonal matrix algorithm and the Gauss-Seidel method. A uniform zonal grid with different step sizes in each region (porous and fluid) is employed in the two directions by locating the finer meshes near the axis and solid walls in the transverse direction. To analyze the effect of the grid size on the numerical solution, various grid systems are tested and a grid system of 400×100 (in X and R directions respectively) is chosen in view of saving computation time. For convergence criteria of the iterative process, the relative variations of velocity components and temperature between two successive iterations are required to be smaller than the value of 10^{-6}.

IV. Results

Due to large number of control parameters, some of them are kept constant. All computations are conducted for water as base fluid (Pr = 7), a porosity ε = 0.95, an inertial coefficient C = 0.35, a Darcy number $Da = 10^{-3}$, a Reynolds number Re = 300, a mass flow rate $R_m = 1$, a viscosity ratio $R_\mu = 1$ and a thermal conductivity ratio $R_k = 1$. The radius ratio and the heat exchanger length are set equal to 2 and 60. The dimensions of the porous baffles are W = 0.5, S = 2 and $H_p = 0.6$. The parametric study is focused on the effects of the volume fraction ($0 \le \varphi \le 0.1$) and types (Cu, Ag, Al_2O_3, CuO, TiO_2) of nanoparticles. Table I summarizes the thermophysical properties of the base fluid and different nanoparticles considered.

TABLE I
THERMOPHYSICAL PROPERTIES OF FLUID AND NANOPARTICLES

	ρ (kg/m^3)	C_p (J/kg K)	k (W/m K)
Water	997	4179	0.6
Copper Cu	8933	385	401
Silver Ag	10500	235	429
Alumina Al_2O_3	3970	765	40
Copper oxide CuO	6320	532	77
Titanium dioxide TiO_2	4250	686	9



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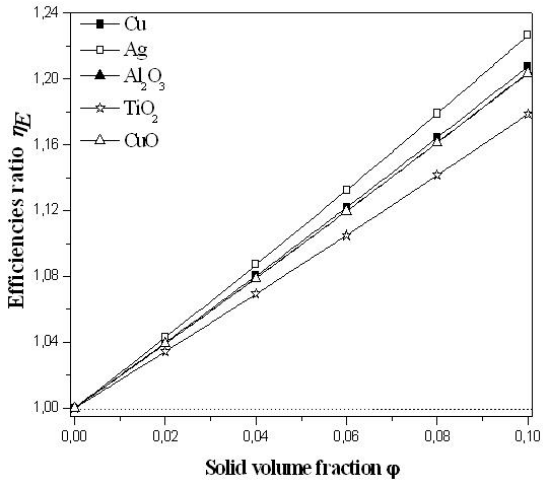


(b)

Fig. 2 E (a) and η_E (b) versus φ for various types of nanoparticles

The evolution of heat exchanger efficiency with the volume fraction for different types of nanoparticles is illustrated in Fig. 2 (a). E increases with φ due to the rise of the mixture thermal conductivity by adding metallic nanoparticles more heat conductive than the base fluid as it appears in Table I. It should also be noted that the efficiency is more sensitive to the volume fraction than to the type of nanoparticles. The highest thermal performance is obtained with the nanoparticles of Ag, whereas with the lowest rate is found for the water-TiO_2 nanofluid which has the lowest thermal conductivity.

To highlight the interest of suspension of nanoparticles in conventional fluids, we introduced the efficiencies ratio η_E normalized by the respective value at φ = 0. By examining Fig. 2 (b), we can say that the presence of nanoparticles is beneficial since this ratio is always greater than unity and increases with φ. As an example for φ = 0.1, the highest rate of enhancement, obtained with silver nanoparticles, is around 23%, whereas the lowest rate of improvement, found with nanoparticles of titanium dioxide, is about 18%.

In order to reveal the contribution of simultaneous use of nanofluids and porous medium, we illustrated on Fig. 3 the evolution of the ratio R_E normalized by the value of the heat exchanger efficiency without porous baffles and nanoparticles. For this porous medium permeability (Da = 10^{-3}) and low effective thermal conductivity (R_k=1), the insertion of baffles contributes moderately to the improvement of heat exchanger performance but at the expense of a quite large increase in pressure drop as it will be seen subsequently. As an example and for nanoparticules of alumina at φ = 0.1, the enhancement passes from 20% if we consider only the effect of nanoparticles to 23% if we also take into account the influence of inserting porous baffles.

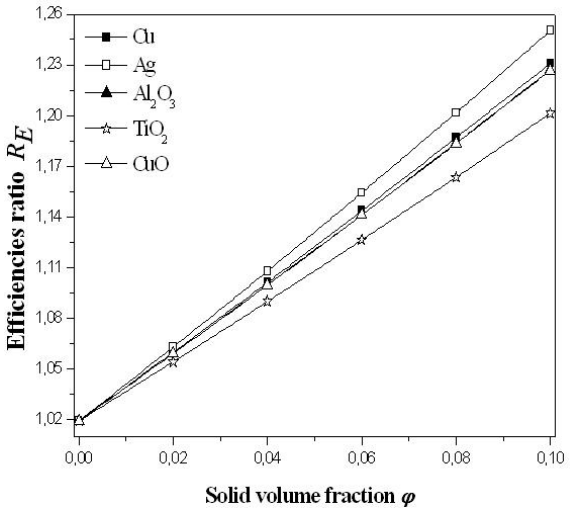
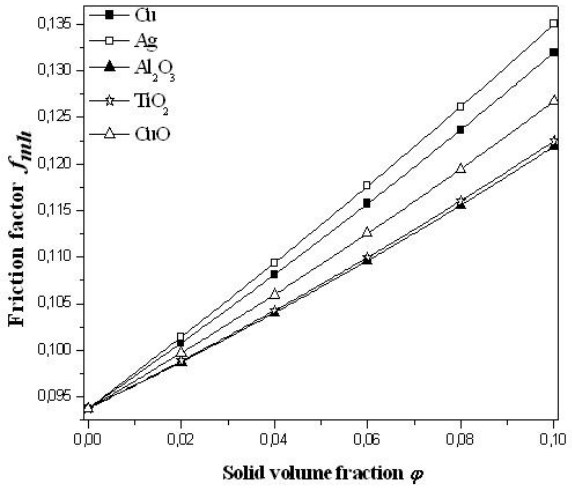


Fig. 3 R_E versus φ for various types of nanoparticles

The heat transfer improvement observed previously is unfortunately accompanied by an increase of the friction factors values as it appears in Figs. 4 and 5. This is due firstly, to the rise of the flow resistance by addition of nanoparticles and secondly, to the modification of nanofluid viscosity which becomes larger than that of the base fluid, leading to a slowdown of its motion. The highest values of f_{mh} and f_{mc} are obtained with the Ag nanoparticles, whereas the lowest values are found with the TiO_2 and Al_2O_3 nanoparticles. These figures also show that the increase of pressure drop in comparison to a conventional fluid varies between 30% and 45% by changing the value of volume fraction and nature of nanoparticles.



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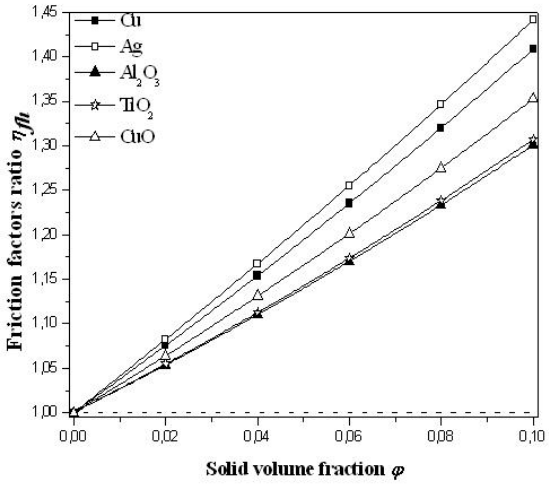
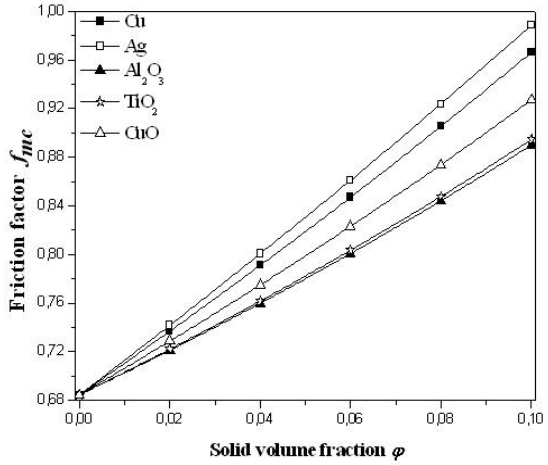
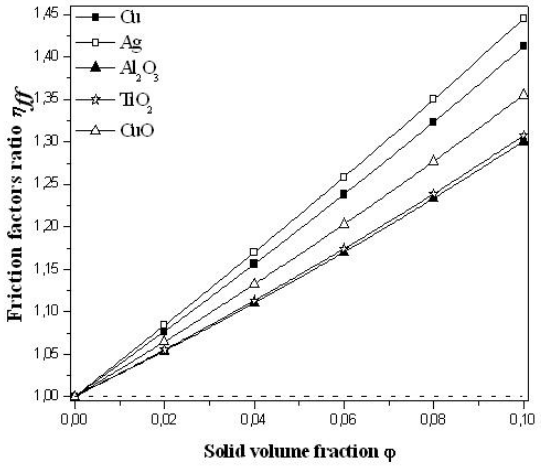


(b)

Fig. 4 f_{mh} (a) and η_{fh} (b) versus φ for various types of nanoparticles



(a)



(b)

Fig. 5 f_{mc} (a) and η_{fc} (b) versus φ for various types of nanoparticles

Fig. 6 shows that friction factor in the annular gap rises significantly compared to the non porous heat exchanger with conventional fluids. For alumina nanoparticles at φ = 0.1, the pressure drop increases by 30% if we consider only the effect of nanoparticles and by 166% if we also take into account the influence of inserting porous baffles.

Globally, silver nanoparticles who led to the highest heat transfer rates also gave the largest values of friction factors and vice versa for TiO_2 nanoparticles; it is then necessary to find a compromise to choose the best nanofluid. From the results of heat exchanger efficiency and friction factors, the water-Al_2O_3 nanofluid seems to be the most interesting in this context.

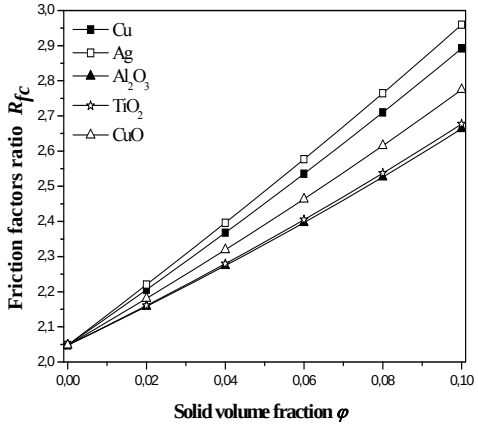
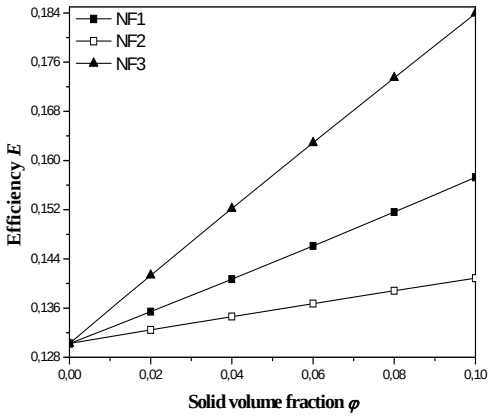


Fig. 6 R_{fc} versus φ for various types of nanoparticles

Up to now the nanoparticles have been dispersed in both hot and cold fluids (NF1), we will now consider two other configurations; NF2 where pure water flows in the annular gap and a water-copper nanofluid circulates in the inner cylinder, and vice versa for NF3 case. The effect of volume fraction on the heat exchanger efficiency for the three situations is

illustrated in Fig. 7. Whatever the considered configuration, there is an improvement of the thermal heat exchanger performance by addition of metallic nanoparticles. The highest performances are found when nanoparticles are added only to the base fluid of the annular gap (NF3), whereas the lowest values of E are obtained for the case where nanofluid flows only in the inner cylinder (NF2). The highest rates of enhancement are around 41% for case NF3 against 21% and 8% for NF1 and NF2 respectively.



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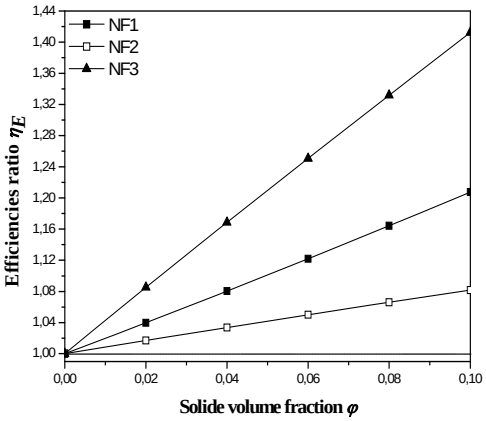

(b)

Fig. 7 E (a) and η_E (b) versus φ for various configurations: water-Cu

V. Conclusion

The present work is a numerical simulation of a double pipe heat exchanger performance. It illustrates the interest of simultaneous use of porous baffles and nanofluids. The results have been exploited by highlighting the influence of some relevant parameters such as the volume fraction and type of nanoparticles. It is found that the gain in heat transfer is more related to the solid volume fraction than to the nature of nanoparticles. The highest rate of improvement is obtained with Ag nanoparticles, whereas the lowest rates of heat transfer are obtained with TiO_2 nanoparticles. Unfortunately, this increase in heat transfer is accompanied by a slowing of the flow in the heat exchanger and an increase of friction factors with the highest values obtained for water-Ag nanofluid. The comparison between different configurations allowed us to conclude that it is preferable to use a nonofluid in the annular gap where circulates the cold fluid and are attached the porous baffles in order to achieve the best thermal performances.

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