Numerical Investigation on Heat Transfer and Performance Number of Nanofluid Flow inside a Double Pipe Heat Exchanger Filled with Porous Media

Ehsan Aminian*, Babak Ahmadi
Department of Mechanical Engineering,
Iran University of Science and Technology, Iran
E-mail: e_aminian@mecheng.iust.ac.ir,
babak_ahmadi@mecheng.iust.ac.ir
*Corresponding author

Received: 8 June 2018, Revised: 20 September 2018, Accepted: 4 December 2018

Abstract: Two common methods to augment heat transfer are the application of nanofluids and porous inserts. In the present work, heat transfer inside a double tube heat exchanger filled with porous media is analyzed numerically using two phase mixture model for the nanofluid flow and the Darcy-Brinkman-Forchheimer model for the flow inside porous media. Basically, porous media improve heat transfer at the expense of increasing pressure drop. A new PN (Performance number) -defined as the ratio of heat transfer to pressure drop on the base state (without porous media and nanoparticles)- is introduced to better judge the first law's performance of configurations. Results indicated that by keeping $Da_i=Da_o=0.1$ and increasing Reynolds number from 500 to 2000, an increase of 56.09% was observed in the performance number. Furthermore, maintaining Reynolds number at Re=500 and changing $Da_i=Da_o$ from 0.0001 to 0.1, results in an increase of 138%. For pressure drop, by keeping $Da_i=Da_o=0.1$ and increasing Reynolds number from 500 to 2000, it is 40 times. Furthermore, maintaining Reynolds number at Re=500 and changing $Da_i=Da_o$ from 0. 1 to 0.0001, the pressure drop is 250 times. Besides, adding 3% nano particles to the base fluid enhances the performance number by about 50% and increase pressure drop by about 20%.

Keywords: CFD, Heat Exchanger, Performance Number, Porous Media

Reference: Aminian, E., and Ahmadi, B., "Numerical Investigation on Heat Transfer and Performance Number of Nanofluid Flow Inside a Double Pipe Heat Exchanger Filled with Porous Media", Int J of Advanced Design and Manufacturing Technology, Vol. 11/No. 4, 2018, pp. 31–46.

Biographical notes: Ehsan Aminian received his BSc and MSc in Mechanical Engineering from the Universities of SBU and IUST respectively. He is currently a PhD student. His current research interests include improving heat transfer by nanoparticles and porous media and optimization of inserting multilayer porous media in energy systems including the heat exchanger, heat sink and heat pipe and enhancement pool boiling. **Babak Ahmadi** is studying MSc in Mechanical Engineering at the University of IUST and received his BSc in Mechanical Engineering from the University of IAUSR. His current research interests include heat transfer enhancement methods, optimization of energy systems, vortical flows and computational fluid dynamics.

1 INTRODUCTION

Nowadays, the population growth and the corresponding increase in energy demand, have led researches toward designing more efficient thermal systems. Heat exchangers have many industrial applications including automobile industry, thermal power plants, air conditioning systems and refrigeration systems. Hence, the efficiency of the heat exchangers is an important subject. Various approaches have been adopted to increase the heat transfer rate of the heat exchangers, such as increasing the surface area in contact with the fluid (through various types of fins and baffles in the channels), varying channel's area using vortex generators, turbulent flow, magnetic control of ferrofluids, electronic and audio fields, as well as surface vibrations. One of the most effective techniques is increasing the heat transfer surface area. Currently, fins are produced in complex shapes, such as louvered fins and slit fins or surfaces protrusions for vortex generators.[1], [2].

Some applications of porous media cited in the literature include food industry, filtration, thermal insulation, ground water, oil flow and various heat exchangers. The significant increase in the temperature of the heat transfer and accidental displacement of the fluid using porous medium has made it more effective than baffles and fins. Particularly, metal foam, which has high porosity, high permeability, high strength structure, low density and large surface area per unit volume has attracted much attention. Numerous studies have also focused on the application of nanofluids. The use of solid particles as a suspended additive to the base fluid is a method of increasing heat transfer. Strengthening the thermal conductivity of conventional liquids by suspending solid particles with a millimeter or micrometer order of magnitude, is a method with the history of more than 100 years. However, this method was ignored at the time due to problems regarding sedimentation, erosion, fouling and the increase in pressure drop. The potential benefits of the nanofluids include long-term sustainability and high-conductivity compared to microscopic particles with the diameter of several millimeters (or even micrometers) dispersed in the liquid. Specifically, the pressure drop and erosion in micro-channels are greatly reduced. [3], [4].

2 LITERATURE REVIEW

Several studies in both fields have been done experimentally and numerically. Abbassi et al. [5] carried out an experimental investigation on heat transfer performance of a 10 nm TiO2/Water nanofluid (deionized water) in a vertical annulus with non-uniform heat flux applied at tube wall. They concluded that the heat transfer coefficient of the nanofluid is higher than that of the pure water. Additionally, the heat transfer coefficient increases by increasing the volume concentration. Javadpour and et al. [6] did an experimental investigation on steady forced convection heat transfer under laminar flow regime in a horizontal annular tube. They calculated local and averaged heat transfer coefficients. Their results showed that the heat transfer coefficient of the nanofluid is higher than that of the base fluid at various Reynolds numbers. Teamah et al. [7] numerically investigated the laminar forced convection inside a pipe flow, partially or completely filled with porous media. The study was conducted for three different cases. In the first case, the porous medium is placed in the form of a cylinder in the center of the tube. In the latter case, the porous medium has an annular shape, and in the third case, the porous medium has a cylindrical shape, but it is only placed in the initial part of the pipe in a short length. They used the Darcy-Brinkman-Forchimer model to describe the fluid transfer. In all cases, the placement of the porous medium resulted in an increase in the Nusselt number in the pipe compared to the case without using the porous medium. They also calculated the critical radius for each case. With the raduis that is above the critical radius, a negative effect on the thermal performance and the pumping power is made. Mahdavi et al. [8] carried out a numerical investigation entropy generation and convection heat transfer through a pipe that was partially filled with porous media for two different cases. In the first case, the porous medium in the pipe center was in the form of a cylinder, and in the second case, the porous medium adhered to the wall as an annular. They used the finite volume approach considering the flow is laminar, fully developed and incompressible. They used Darcy-Brinkman-Forchimer model and concluded that the location of the porous medium had a significant effect on the heat transfer performance of the pipe. In addition, by observing a performance number less than 1 in most of the configurations, it was concluded that the increased heat transfer rate cannot compensate for the increase in pumping power. However, if the only goal is to achieve a higher heat transfer rate, one of the configurations that is more appropriate is used. Momourian et al. [9] presented a two-dimensional analysis for the simulation of a double tube heat exchanger filled with porous media. They investigated the effect of turbulent flow on the heat transfer performance of the heat exchanger. They used Darcy-Brinkman-Forchimer model to analyze the flow in the porous medium and used k-ε model for turbulence flow modeling. They concluded that an increase in Reynolds number increases the averaged Nusselt number. Additionally, increasing the Darcy number, reduces the averaged Nusselt number.

Moreover, numerous experimental studies have also been carried out. Hosseini et al. [10] experimentally investigated the steady state laminar forced convection in a horizontal tube with a circular cross section, partially filled with porous medium, receiving constant thermal flux. The results showed that there was a significant increase in heat transfer due to the effects of the porous medium and the magnetic field up to 2.4 times in case of using them simultaneously. They also referred to the greater impact of the porous medium compared to the magnetic field in terms of increasing heat transfer coefficient and pressure drop inside the tube. In addition, some analytical studies were also conducted. Lu et al. [11] examined analytically the forced convection flow and heat transfer in a channel composed of two parallel plate that was partially filled with metal foams. For the thermal modeling of fluid flow in a porous medium, the modified Brinkman model was used for the Darcy equation. They also applied the non-equilibrium model for the energy model in the porous portion of a channel partially filled with porous media. The velocity and temperature profiles along the channel composed of two parallel plate was investigated using predictive analytical methods. They also evaluated the effects of the key parameters on heat transfer and flow resistance. Their results showed better heat transfer performance and higher pressure drop in the case of using metal foam compared to the case without it. However, the pressure drop in the partially filled channel with metal foam is lower than that of a pipe fully filled with metal foam. The relative height of the foam, porosity and permeability significantly affects the flow distribution in the partially filled channel.

Numerous studies on the subject of nanofluid have been addressed. Chamka et al. [12] depicted the natural heat transfer in a trapezoidal cavity partially filled with the porous layer and nanofluid and partially with non-Newtonian fluid layer, by thermal lines. Nanofluids with water as the base fluid and silver, copper, alumina or titanium as nanoparticles were tested. They investigated the effects of the Rayleigh numbers, Darcy number, the volume fraction of the nanoparticles and porosity layer thickness using the finite volume method. The effects of different parameters on the overall heat transfer, the thermal conductivity of nanoparticles and porous media were also explored. The results showed that the use of water and silver nanofluid significantly affected the increased heat transfer relative to the gravitational angle of the cavity. They also stated that the relative volume of nanoparticles has led to an increase in the Nusselt number, but its effect was weaker than the effect of the Darcy number on a wall with a sloping surface. In addition, Lotfi et al. [3] numerically analyzed the forced convection with nanofluid water and Al_2O_3 in a horizontal tube. They were pioneers to study such flows using the two-phase Eulerian. By comparison with the experimental values, it was noticed that the results of the mixed model are in better agreement with the experimental values. Their results showed that the rate of heat transfer decreases by increasing the size of nanoparticles. Akbarinia et al. [13] simulated two-phase mixture model in laminar flow nanofluid of Al_2O_3 and water in an elliptic channel with a constant heat flux applied as the boundary condition. They solved three-dimensional Navier-Stokes and energy equations of the base fluid and also volume fraction equation of the nanofluid through a finite volume approach. They also investigated the Brownian motion of the nanoparticles (that intensively depend on temperature) to determine the thermal conductivity and dynamic viscosity of the nanofluid. The results point out that increasing the volume fraction of nanoparticles causes an increase in Nusselt number and also decreases the friction factor, if other conditions are kept constant. Furthermore, the larger the cross section area of the duct, the lower the erosion is due to friction of the duct. Moreover, Manca et al. [14] carried out a numerical analysis of turbulent flow along with forced convection of nanofluid water with Al_2O_3 in a circular tube at which the constant heat flux is applied. They also used two different methods in their study: single-phase model and the two-phase model. They concluded that the heat transfer coefficient can be augmented by increasing the concentration of nanoparticles or the Reynolds number.

Numerous studies have been conducted using both nanofluid and porous media simultaneously. Siavashi et al. [1] considered using nanoparticles and porous media simultaneously to improve heat transfer inside an annulus. They used the two-phase mixture model and the corresponding Darcy-Brinkman-Forchimer model to simulate the flow of the nanofluid in a porous medium. They carried out a parametric study on the effects of the thickness of the porous layer and Reynolds number, as well as the concentration of nanoparticles and the position of the porous medium (that was placed on the inner wall or the external wall). Considering the first law and second law of thermodynamics, they introduced a new parameter which is called the performance number that indicates the ratio between the strength of heat transfer and pressure drop. Their results showed that the concentration of nanoparticles and the fluid velocity have significant effects on the performance of the heat exchanger and entropy generation. In addition to the permeability, the thickness of the porous medium can also affect the performance of the heat exchanger. Furthermore, their results indicated that there is an optimum thickness for the porous medium for each nanofluid with a specific velocity and the addition of nanoparticles to the base fluid would improve the performance of the system in all cases. Additionally, Dehkordi et al. [15] performed numerical and experimental analyses on the flow of nanofluid in a vertical rectangular channel, partially filled with open cell metal foam applying constant heat flux to its boundary. Nanofluid water-Al_2O_3 was used with different concentrations. To obtain the velocity and temperature profiles of the nanofluid, they used a finite difference method. Brownian motion and thermophysical effects of nanoparticles were also examined. The experimental and numerical results obtained were in good agreement. The use of nanofluids in the distillation of water, increased heat transfer, by about 20%. Also, increasing the concentration of nanoparticles did not show a significant effect on the pressure drop. Adding nanoparticles with a concentration of 5% to the base fluid increases the pressure drop across the channel by about 3%. In addition, Pop et al. [16] reviewed the steady natural convection in two-dimensional form in a cavity partially filled with porous media that uses nanofluid, as well as internal heat generation. They used the Darcy equation to model the motion. Furthermore, they used a new model for energy equation and nanoparticle concentration. They used the finite difference method and introduced a non-dimensional parameter that incorporates the temperature and concentration of nanoparticles. The effect of Rayleigh number and Luis number were also investigated. They found that the addition of nanoparticles and the use of porous media reduced the temperature and increased the heat transfer. They stated that the thermal conductivity of nanofluid is much higher than the thermal conductivity of the porous medium. When thermal conductivity of the nanoparticles and the porous medium is close to each other, the addition of nanoparticles has no effect on heat transfer.

This paper covers a numerical modeling on heat transfer of the nanofluid (water and Al_2O_3) in conjunction with porous media in a double pipe heat exchanger. The novelty of this paper is that the addition of porous media on the performance of the heat exchanger (considering both heat transfer and pressure drop simultaneously) in a new geometry (double pipe heat exchanger) is investigated. Then, the effects of different parameters on the performance of the heat exchanger are evaluated.

3 MATHEMATICAL FORMULATION

A schematic representation of the problem is shown in "Fig. 1". In the case of equal volumetric flow rates for the cold and hot fluids, various conditions (such as permeability, flow velocity and thicknesses of the porous medium) were studied. For this purpose, the flow is analyzed in the fully developed region. In this study, there are two concentric pipes with a length of 1.5 m, while the radius of the inner tube and outer tube are 4 cm and 10 cm respectively. The results are expressed in terms of the amount of the heat transfer rate and the pressure drop in the heat exchanger as well as the performance number of the heat exchanger. It is noteworthy that heat generation inside the system is not considered and the outer part of the heat exchanger is insulated.

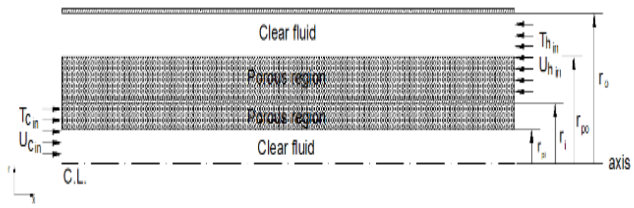


Fig. 1 Schematic geometry of the problem.

The following assumptions were made to simplify the modeling of the heat exchanger:

- The nanofluid flow was assumed stable, symmetric, laminar and incompressible.
- The nanofluid was assumed homogeneous and isotropic.
- The thermophysical properties of nanoparticles and fluids such as density, viscosity and heat transfer coefficient were assumed constant.
- The chemical properties of the metal foam used in this study (aluminum) are assumed stable and in thermal equilibrium with the environment.
- The physical properties used for the nanoparticles with the diameter of 38 nm and water that were employed in this study such as density, thermal conductivity and dynamic viscosity of the water fluid and Al2O3 nanoparticles were calculated at 20 ℃ as shown in "Table 1".
-

Table 1 The thermal properties of Al_2O_3 and Water

Property	Unit	Water	Al_2O_3
Density	$kg.m^{-3}$	998.2	3880
Special heat	$J.kg^{-1}.K^{-1}$	4182	773
Thermal conductivity	$W.m^{-1}.K^{-1}$	0.6	40
Dynamic viscosity	$Kg.m^{-1}.s^{-1}$	$1003*10^{-6}$	-
Nanoparticles diameter	nm	-	38

According to the Darcy's law, the fluid velocity is a function of the pressure gradient, which is used for low-Reynolds number flows. For the current study with much higher Reynolds number, the Darcy's law must be modified to calculate the inertia and viscous effects more accurately. Forchimer correction terms refer to the effects of inertia and Brinkman terms account for the correction of viscosity effects. [17] Besides, the no-slip and impermeable wall boundary conditions are considered. The Two-phase mixture model is used for the nanofluid solution composed of water and nanoparticles. The continuity, momentum and energy equations are solved with the aid of the two-phase mixture model. For the second phase, volume fraction equations are solved. By using these two sets of equations, the problem is solved. The continuity equation for mixtures is given by “Eq. (4)”.

$$\nabla.(\rho_m v_m) = 0 \tag{1}$$
q

Where v_m and ρ_m are the mass-averaged velocity and the mixture density, respectively. These parameters are defined as “Eqs. (2) and (3)”:

$$v_m = \frac{\sum_k^n \emptyset_k \rho_k v_k}{\rho_m} \tag{2}$$

$$\rho_m = \sum_k \emptyset_k \rho_k \tag{3}$$

In which ϕ_k is the volume fraction of the k phase. The Momentum equation can be written as “Eq. (4)” based on the two-phase mixture model [14]:

$$\nabla.(\rho_m v_m v_m) = -\nabla p + \nabla.[\mu_m(\nabla.v_m v_m^T)] + \nabla.(\sum_{k=1} \varphi_k \rho_k v_{dr,k} v_{dr,k}) \tag{4}$$

where μ_m is the mixture viscosity that is defined as “Eq. (4)”:

$$\mu_m = \sum_k \phi_k \mu_k \tag{5}$$

The effective viscosity of solids and nanoparticles is calculated using the curve reported by Miller and Gideaspoe [18], where the viscosity of the nanoparticles is defined as “Eq. (4)”:

$$\mu_{np} = -0.188 + 537.42\phi_{np} \tag{6}$$

$v_{dr,k}$ is the drift velocity of the phase k that is defined as “Eq. (4)”:

$$v_{dr,k} = v_k - v_m \tag{7}$$

The energy equation for a mixture without thermal sources is obtained from “Eqs (8) and (9)”, [2], [4]:

$$\nabla.\sum_k (\emptyset_k v_k \rho_k C_{P_k} T) = \nabla.(k_m \nabla T) \tag{8}$$

$$k_m = \sum_k \phi_k k_k \tag{9}$$

Where k_m is known as the effective thermal conductivity of the mixture. $v_{np,bf}$ is the relative velocity that is defined as the ratio of the velocity of the second phase v_{np} to the velocity of the first phase and is expressed as “Eq. (4)”:

$$v_{np,bf} = v_{np} - v_{bf} \tag{10}$$

The drift velocity is related to the relative velocity according to “Eq. (4)”:

$$v_{dr,k} = v_{np,bf} - \sum_k^n \frac{\emptyset_k \rho_k}{\rho_m} v_k \tag{11}$$

Where the relative velocity is calculated from “Eq. (4)”:

$$v_{np,bf} = \frac{\rho_{np} d_{np}^2}{18\mu_{bf} f_{drag}} \frac{(\rho_{np} - \rho_{bf})}{\rho_{np}} \alpha \tag{12}$$

The particle acceleration in the second phase is defined as “Eq. (4)”:

$$\alpha = g - (v_m.\nabla)v_m \tag{13}$$

Where g is the acceleration vector of gravity. The drag coefficient f_{drag} can also be calculated from the Schiller and Naumann curves according to “Eq. (4)”:

$$f_{drag} = \begin{cases} 1 + 0.15\,\mathrm{Re}_{np}^{0.687} & , \quad \mathrm{Re}_{np} \le 1000 \\ 0.0183\,\mathrm{Re}_{np} & , \quad \mathrm{Re}_{np} \ge 1000 \end{cases} \tag{14}$$

In the two-phase mixture model, the relative volume of the second phase is calculated according to “Eq. (4)”, [2], [4-5]:

$$\nabla.(\phi_{np} \rho_{np} v_{np}) = -\nabla.(\phi_{np} \rho_{np} v_{dr,np}) \tag{15}$$

Considering the two-phase mixture model, Darcy-Brinkman-Forchimer equation can be expressed as “Eq. (4)”, [17]:

$$\frac{\rho_m}{\varepsilon^2}[(v_m.\nabla)v_m] = -\nabla p + \frac{\mu_m}{\varepsilon}\nabla^2 v_m - \frac{\mu_m v_m}{K} - \frac{\rho_m \varepsilon C_d}{\sqrt{K}} v_m |v_m| + \nabla.(\sum_{k=1}^n \alpha_k \rho_k v_{dr,k} v_{dr,k}) \tag{16}$$

Where C_d is the inertial coefficient in the porous medium and can be calculated from "Eq. (4)", [1]:

$$C_d = \frac{1.75}{\sqrt{150}\varepsilon^{3/2}} \tag{17}$$

The energy equation can be expressed as "Eq. (4)", [6-7]:

$$\nabla.\sum_{k=1}^{n}\left(\alpha_k v_k \rho_k C_{P_k} T\right) = \nabla.\left(k_{eff}\nabla T\right) \tag{18}$$

Where k_{eff} is the effective thermal conductivity in a porous medium and nanofluid mixture, defined as "Eq. (4)":

$$k_{eff} = (1-\varepsilon)k_p + \varepsilon k_m \tag{19}$$

In cylindrical coordinates, boundary conditions are represented as "Eqs. (4) to (23)":

$$0 \rightarrow r_i \quad \begin{array}{l} x=0 \ : \ U_{C_{in}} = U_{in} \ , \ v_r = 0 \ , \ T = T_{C_{in}} \ , \\ x = L \ : \ \dfrac{\partial v_x}{\partial x} = \dfrac{\partial v_r}{\partial x} = \dfrac{\partial T}{\partial x} = 0 \ , \end{array} \tag{20}$$

$$r_i \rightarrow r_o \quad \begin{array}{l} x=0 \ : \ U_{C_{in}} = U_{in} \ , \ v_r = 0 \ , \ T = T_{C_{in}} \ , \\ x = L \ : \ \dfrac{\partial v_x}{\partial x} = \dfrac{\partial v_r}{\partial x} = \dfrac{\partial T}{\partial x} = 0 \ , \end{array} \tag{21}$$

$$\begin{cases} r = r_i \ : \ v_x = v_r = 0 \\ r = r_o \ : \ v_x = v_r = 0 \end{cases} \tag{22}$$

$$-k_{eff}\left.\frac{\partial T}{\partial r}\right|_{r=r_i^-} = k_{eff}\left.\frac{\partial T}{\partial r}\right|_{r=r_i^+} = q'' \ , \ \left.\frac{\partial T}{\partial r}\right|_{r=0} = 0 \ , \ k_{eff}\left.\frac{\partial T}{\partial r}\right|_{r=r_o} = 0 \tag{23}$$

In addition, the boundary conditions (24) are applied at the interface of the porous media and the clear region to ensure the continuity of the velocity, pressure, stress, heat flux and temperature:

$$\begin{array}{l} v|_c = v|_p \\ T|_c = T|_p \qquad k_{eff}\dfrac{\partial T}{\partial r}|_c = k_{eff}\dfrac{\partial T}{\partial r}|_p \\ [\dfrac{\partial v_x}{\partial r} + \dfrac{\partial v_r}{\partial x}]_c = \dfrac{1}{\varepsilon}[\dfrac{\partial v_x}{\partial r} + \dfrac{\partial v_r}{\partial x}]_p \\ \dfrac{\partial v_r}{\partial r}|_c = \dfrac{1}{\varepsilon}\dfrac{\partial v_r}{\partial r}|_p \end{array} \tag{24}$$

The subscript c and p indicate the clear region and porous medium, respectively. The local heat transfer coefficient at the wall can be calculated from "Eq. (4)":

$$q'' = hA(T_w - T_b) = k_{eff}A\left(\frac{\partial T}{\partial r}\right)_w \tag{25}$$

In which w refers to the wall and T_b is the fluid temperature that can be calculated from "Eq. (4)":

$$T_b = \frac{\int_{r_i}^{r_o} T|v_m|rdr}{\int_{r_i}^{r_o}|v_m|rdr} \tag{26}$$

The hydraulic diameter of the annulus can be defined as "Eq. (4)":

$$D_h = 2(r_o - r_i) \tag{27}$$

In addition, two useful dimensionless numbers, the Darcy number and the Reynolds number are defined by "Eqs (28) and (29)":

$$\mathrm{Re} = \frac{\rho_m U_{in} D_h}{\mu_m} \tag{28}$$

$$Da = \frac{K}{D_h^2} \tag{29}$$

Volumetric flow rate is calculated from "Eq. (4)". The equality of the flow rates can be expressed as "Eq. (4)":

$$Q_m = U_m A_m \tag{30}$$

$$Q_{C_{in}} = Q_{h_{in}} \tag{31}$$

4 NUMERICAL SOLUTION

By the discretization of the equations expressed in the previous section, the problem is solved using the finite volume approach. The SIMPLE method is used for the pressure-velocity coupling. Second-order upwind and central difference approximation has been used for discretization. The convergence criterion is set to 10^{-6} for the residual of the continuity equation. The solution procedure should be continued until the convergence criteria is satisfied. For the mesh independency study, various mesh sizes were used in order to determine a suitable size for the mesh in which there is a balance between the accuracy and the available computational sources.

The independency of the results from the mesh size is confirmed by the calculated value of the total heat transfer of the flow as in "Fig. 2". Finally, a mesh resolution of 1500 * 100 cells along x and r respectively, is the most suitable number of nodes for the presented calculations.

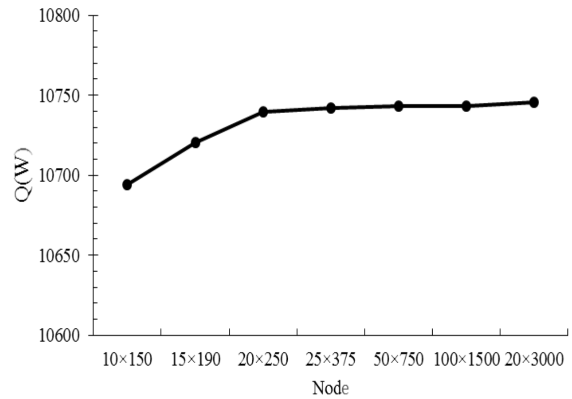


Fig. 2 Grid-independency study in a double pipe heat exchanger (for the case in which both sides of the heat exchanger are completely filled with porous media, Rei=1000, Dai=Dao=0.001).

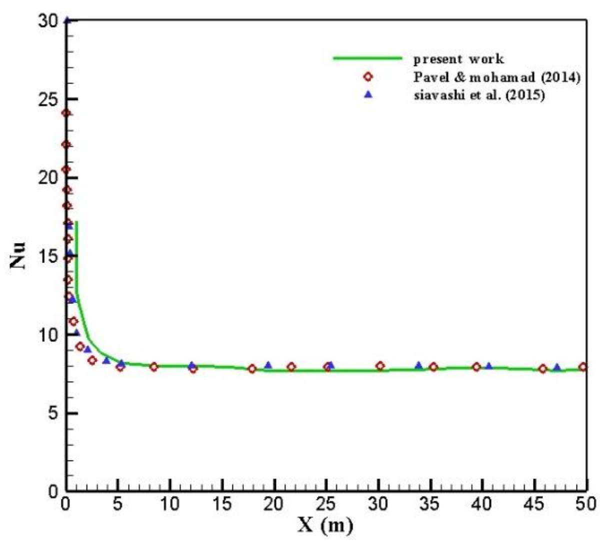


Fig. 3 The comparison of the results of present work with the results of Siavashi et al. [1] and the experimental results of Pavel and Mohamed [19] (Da = 10-4, rp = 0.04).

A CFD code for solving equations of flow, two-phase mixture modeling for nanofluid and the Darcy-Brinkman-Forchimer model for porous media has been used. In the first step, a simple tube partially filled with porous media and a constant heat flux applied to its outer side was considered. In order to validate the code for the fluid flow in a porous medium, a comparison was made against the experimental data available in the paper by Pavel and Mohammed [19] for a laminar flow in a tube, partially filled with porous medium, along with a constant heat flux at the external boundary. Furthermore, the local Nusselt number along with the length of the pipe was compared well with results given in the paper by Siavashi et al. [1]. The results of this study differ from those of Powell and Mohammed by about 2.1% that shows the validity of the modeling. The results are shown in “Fig. 3”.

For the purpose of validation, a nanofluid of water and Al_2O_3 with a concentration of 1.6% with constant heat flux applied to the wall of the tube is considered. Local heat transfer coefficient along the pipe is calculated using the two-phase mixture model. The code has been validated against the results of the two-phase mixture model presented by Göktepe et al. [20] for the nanofluid flow using a two-phase mixture method. The discrepancy between the results of the present work and the paper by Göktepe et al. is approximately 2.6%. The results are shown in “Fig. 4”. The two-phase mixture model is not capable of taking into account phenomena such as pipe clogging and nanofluid instability, due to its uncertainty.

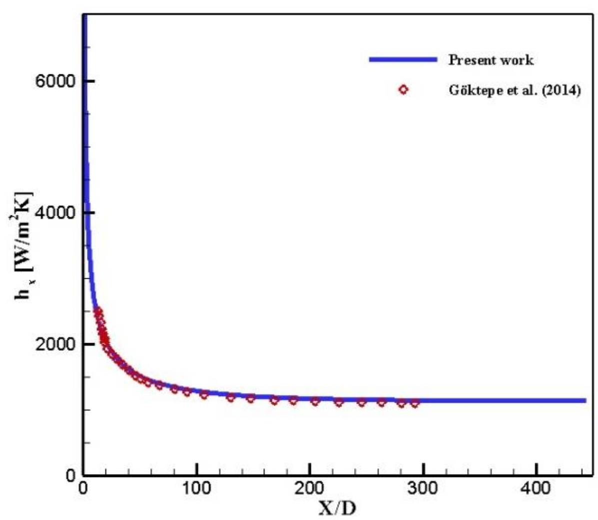


Fig. 4 The comparison of the local heat transfer coefficient of the present work with results of Göktepe et al. [20] (for water and Al2O3 as the nanofluid with Reynolds number of 1050 in a simple tube).

5 RESULTS AND DISCUSSION

In this paper, the effect of using porous metal foam and nanofluid on heat transfer augmentation of a double tube heat exchanger (with a reasonable penalty in terms of pressure drop) was studied. The nanofluid is composed of water as the base fluid and Al2O3 as the nanoparticle. The porous medium used in this study has a porosity of 0.8 and different permeability in terms of the Darcy number ranging from 0.1 to 0.0001. The ultimate goal of this study is to increase heat transfer while maintaining the lowest pressure drop possible in various conditions.

This can be done by considering a parameter that take into account both heat transfer and pressure drop. This parameter is called the performance number that is defined as "Eq. (32)".

$$PN = \frac{\dfrac{Q}{Q_b}}{\left(\dfrac{\Delta P_i + \Delta P_o}{(\Delta P_i + \Delta P_o)_b}\right)^{\frac{1}{3}}} \tag{32}$$

In order to increase heat transfer in a double tube heat exchanger, porous medium with any thickness can be used either in the inner tube or the outer annulus. On the other hand, the addition of a porous medium leads to an increase in the pressure drop in nanofluid due to the existence of a solid matrix.

Nanoparticles with a volume fraction of 3%, Reynolds number of 500-2000 (Laminar flow regime) for the nanofluid flowing in the inner tube and equal flow rates of the hot and cold fluids are considered. Temperature contour and velocity vector for the nanofluid flow at the length $1<x<0.5$ and Reynolds number of 500 (for the flow in the inner tube) are shown in "Fig. 5", in order to give better insight into the performance of the heat exchanger compared to the effect of the porous medium and the flow velocity.

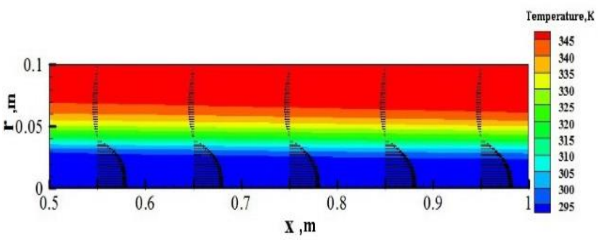


Fig. 5 Temperature contour and velocity vector for the heat exchanger without porous media (Rei=500).

Temperature contour and velocity vector for a heat exchanger filled with porous media on both sides at the Darcy number of 0.0001 and at the Reynolds number of 500 (for the flow in the inner tube) are shown in "Fig. 6". Additionally, the application of the porous medium with the Darcy number of 1000 fold greater than the previous case, yields to the results shown in "Fig. 7". By the comparison of "Fig. 6 and Fig. 7", the effect of the 1000 fold increase in the permeability of the porous medium can be seen. Moreover, by keeping the Darcy number at 0.0001 and changing the inner-tube Reynolds number into 2000, the effect of the flow velocity on heat transfer and velocity profile can be seen as in "Fig. 8 ". The velocity in the neighborhood of the wall of the tube is increased which can affect the heat transfer near the wall. Additionally, due to the permeability of the porous medium, the heat transfer is augmented that results in the decrease of the temperature of the nanofluid near the wall and a reduction in the thickness of the thermal boundary layer.

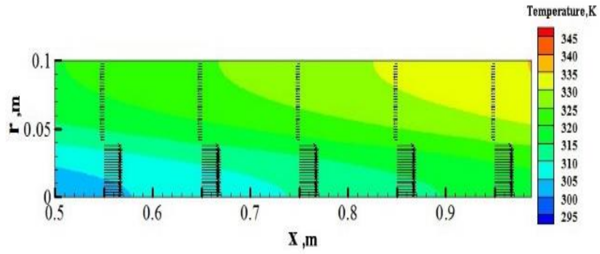


Fig. 6 Temperature contour and velocity vector for the heat exchanger filled with porous media (Re_i=500, Da_i=0.0001, Da_o=0.0001).

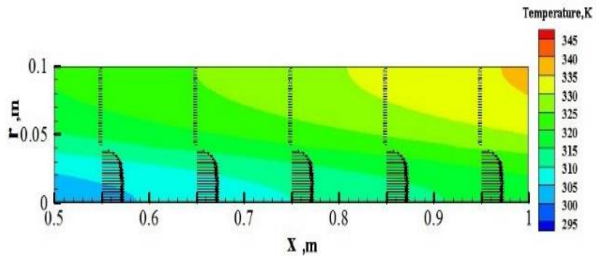


Fig. 7 Temperature contour and velocity vector for the heat exchanger filled with porous media (Re_i=500, Da_i=0.1, Da_o=0.1).

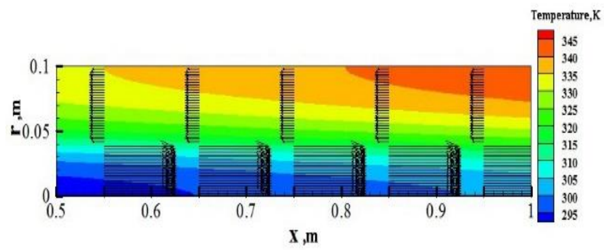


Fig. 8 Temperature contour and velocity vector for heat exchanger filled with porous media (Re_i=2000, Da_i=0.0001, Da_o=0.0001).

Based on the definition of the performance number, the ratio of the increase in heat transfer to the increase in pressure drop is a suitable criterion for evaluating the relative effectiveness of each case considered. It should be noted that the performance number always has a positive value (PN>0). The higher it is than 1, the more efficient is the heat exchanger i.e. the increase in heat transfer is worth the corresponding pressure drop.

In this section, results are divided into two main categories as follows:

- Effect of the Reynolds number on the performance number
- Effect of the Darcy number of the porous medium on the performance number
- Effect of the porous media thickness on the performance number

5.1. Effect of the Reynolds Number on the Performance Number

In this subsection, the range of the Reynolds number in which the flow is in the developing condition of the laminar regime is considered. So, laminar flow is investigated in terms of the performance number. The Reynolds number always has a direct relationship with the performance number (i.e. increasing the Reynolds number leads to an increase in the performance number), such that for Re=2000, the highest achievable performance number would be 4.61 and 6.4 using porous medium with Da_i=0.01 and Da_i=0.1 respectively. However, for the case of Re=500, the highest achievable performance number is 3.61 and 6 using porous media with Da_i=0.01 and Da_i=0.1 respectively. Consequently, increasing the velocity of the nanofluid can increase the heat transfer. The aforementioned relationship is also confirmed by Siavashi, et al. [1].

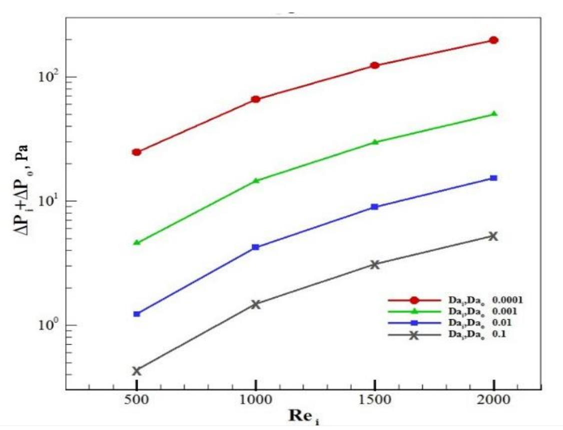


Fig. 9 Total pressure drop of two parts of the heat exchanger due to increased Reynolds number in the inner tube.

The heat transfer increase outweighs the added pressure drop. An example of these results is shown in “Fig. 9”. The increase in the heat transfer coefficient (h) is due to the increase of the nanofluid velocity near the wall. Hence, the heat transfer is increased significantly such that the pressure drop increases due to the use of porous media can be compensated.

5.2. Effect of the Darcy Number of the Porous Medium on the Performance Number

This subsection can be divided into two parts. The first case is when both sides of the heat exchanger use the porous media in terms of permeability. The second case is related to the application of the porous medium with different for each side of the heat exchanger.

5.2.1. Using porous medium with the same permeability on the two sides of the heat exchanger

Two cases can be considered as follows:

- **Low permeability**

As shown in “Fig. 10”, if porous media with low permeability porous media (Da=0.001) is used, the highest performance number is achieved either with both sides of the heat exchanger which are filled completely with porous media or when the outer annulus is clear while the inner tube contains porous media more than a half.

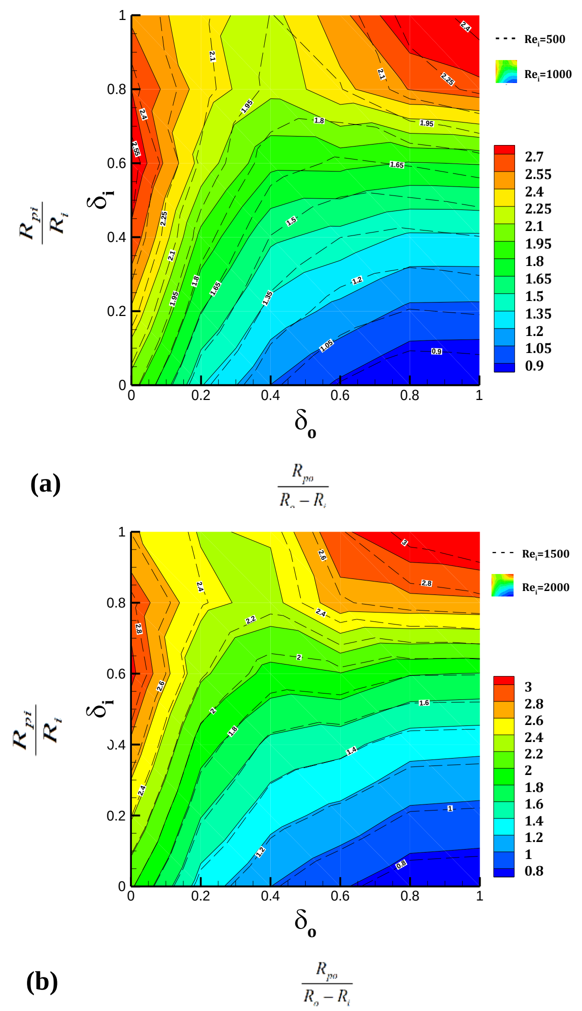


Fig. 10 Performance number for different porous metal foam thicknesses (Da_i=0.001, Da_o=0.001): (a): Re_i=500, 1000 and (b): Re_i=1500, 2000.

- **High permeability**

If high permeability porous media is used, the effect of porous media thickness will be the same in both cases of Da=0.1 and Da=0.01. As can be seen from “Fig. 11 and Fig. 12”, an increase in the porous media thickness in the inner tube always results in an increase in the performance number. However, the outer annulus shows different characteristics, because its effect depends on the thickness of the porous medium in the inner tube.

If the inner tube does not contain porous media, the performance number increases by the decrease in the thickness of the porous medium in the outer annulus. If porous media is added to the inner tube, the performance number increases with increasing the porous media thickness in the outer annulus. The highest performance number is obtained when the heat exchanger is filled with porous medium in both inner tube and the outer annulus. On contrary, the lowest performance number is achieved when only the outer annulus is filled with the porous medium.

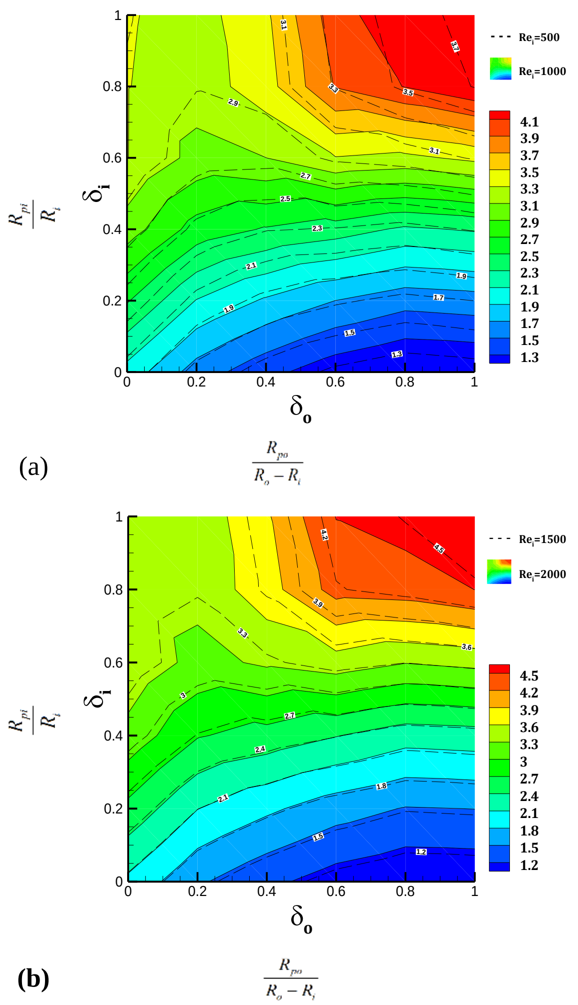


Fig. 11 Performance number for different porous metal foam thicknesses (Da_i=0.01, Da_o=0.01): (a): Re_i=500,1000 and (b): Re_i=1500, 2000.

Generally, the results indicate that the Darcy number of the porous medium has a direct relationship with the performance number of the heat exchanger. As the Darcy number increases, the heat transfer increase outweighs the added pressure drop. Hence, the performance number increases.

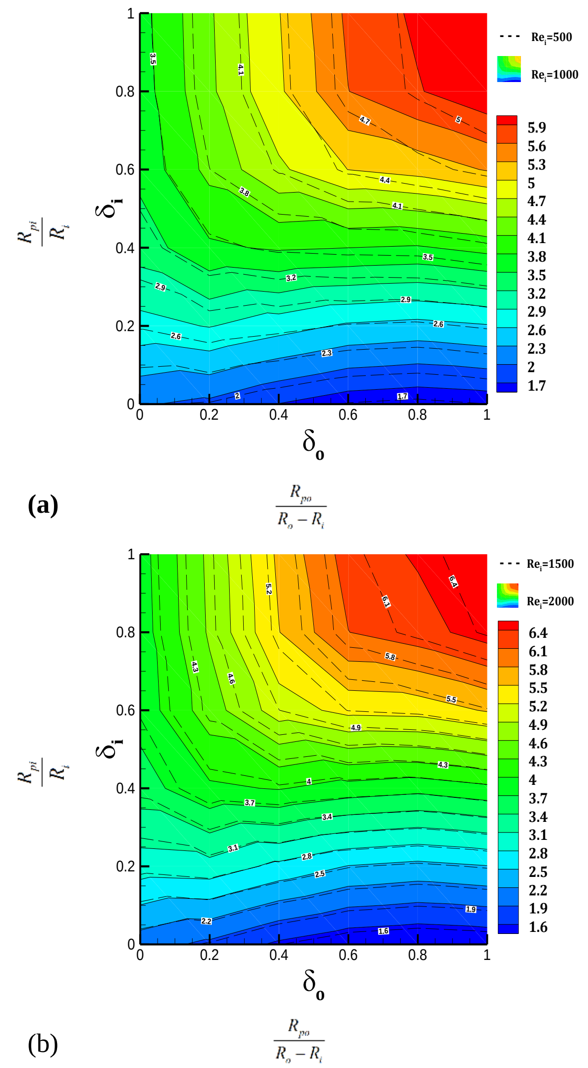
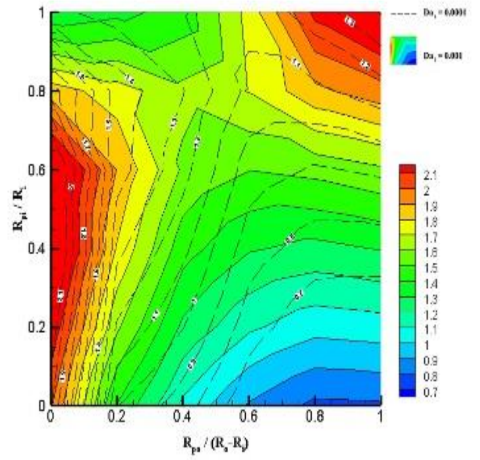


Fig. 12 Performance number for different porous metal foam thicknesses (Da_i=0.1, Da_o=0.1): (a) : Re_i=500,1000 and (b): Re_i=1500, 2000.

5.2.2. Using porous medium with different permeability's on the two sides of the heat exchanger

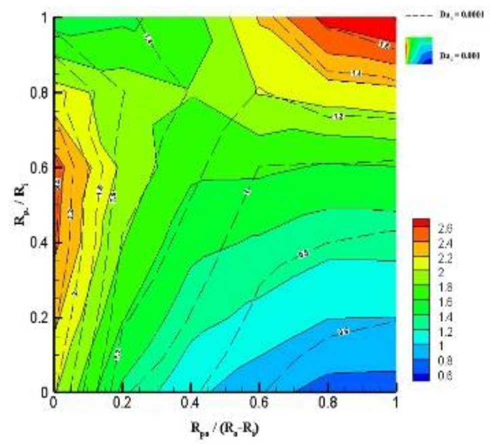
If the permeability of the porous medium used in the inner tube is low, the best performance of the heat exchanger occurs in two configurations. The first configuration is that both sides of the heat exchanger are filled with porous media and the second configuration is that porous media is only used in the inner tube of the heat exchanger. The aforementioned configurations are shown in “Fig. 13 and Fig. 14”. If the permeability of the porous media in the inner tube is medium, two configurations can be considered. The first configuration is that both sections of the heat exchanger are filled with porous media and the second configuration is that the outer annulus is empty of the porous medium and the inner tube is almost fully filled with porous medium.

$\frac{R_{pi}}{R_i}$



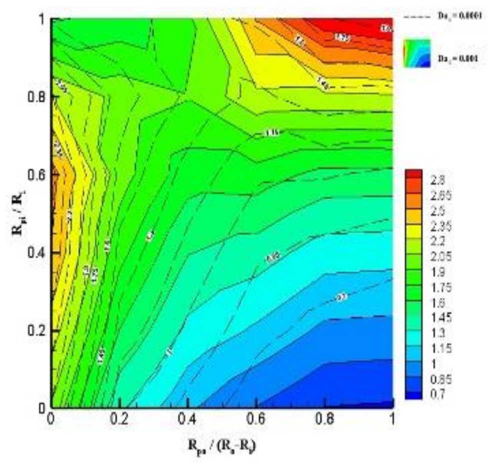
(a) $\frac{R_{po}}{R_o - R_i}$

$\frac{R_{pi}}{R_i}$



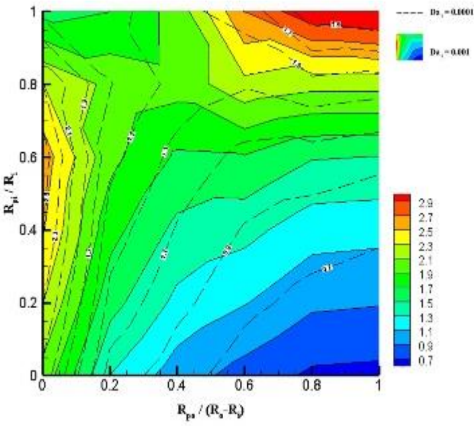
(b) $\frac{R_{po}}{R_o - R_i}$

$\frac{R_{pi}}{R_i}$



(c) $\frac{R_{po}}{R_o - R_i}$

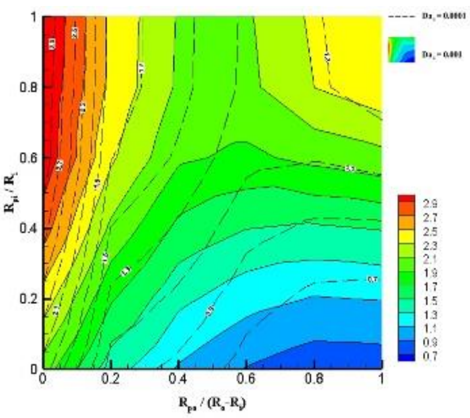
$\frac{R_{pi}}{R_i}$



(d) $\frac{R_{po}}{R_o - R_i}$

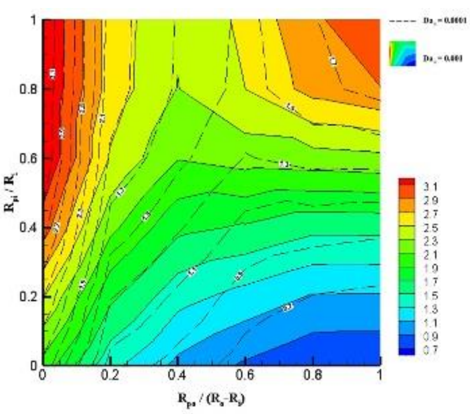
Fig. 13 Performance number for different porous metal foam thicknesses (Dai=0.0001, 0.001, Da_o=0.0001): (a): Re_i=500, (b): Re_i=1000, (c): Re_i=1500 and (d): Re_i=2000.

$\frac{R_{pi}}{R_i}$

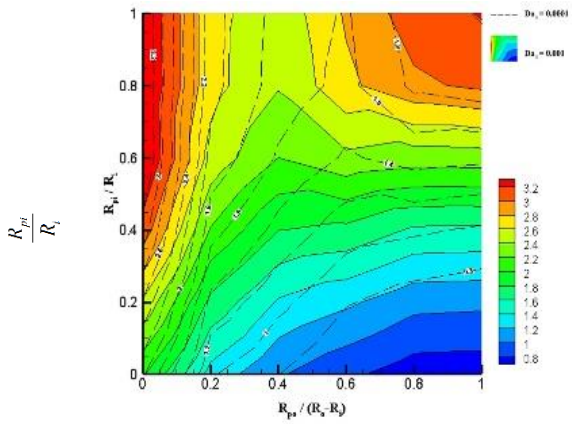


(a) $\frac{R_{po}}{R_o - R_i}$

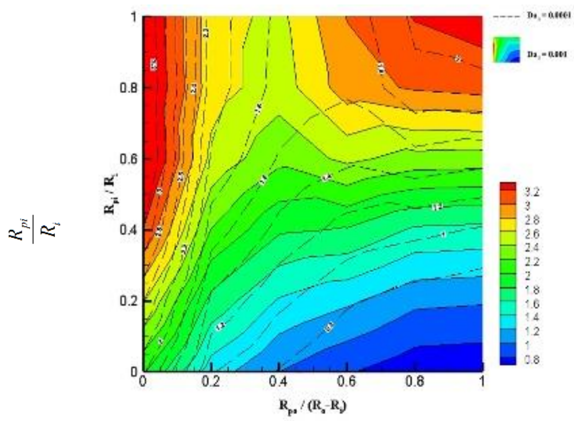
$\frac{R_{pi}}{R_i}$



(b) $\frac{R_{po}}{R_o - R_i}$

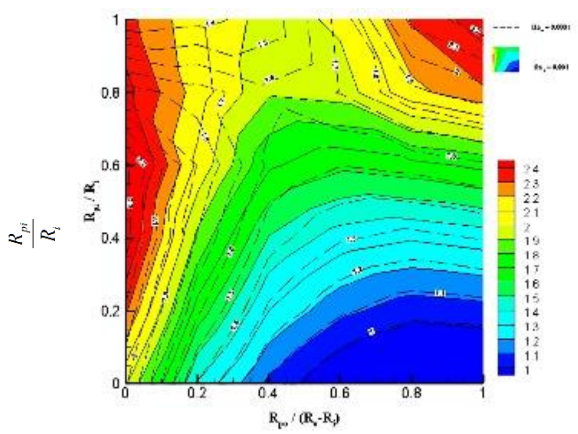


(c) $\frac{R_{po}}{R_o - R_i}$

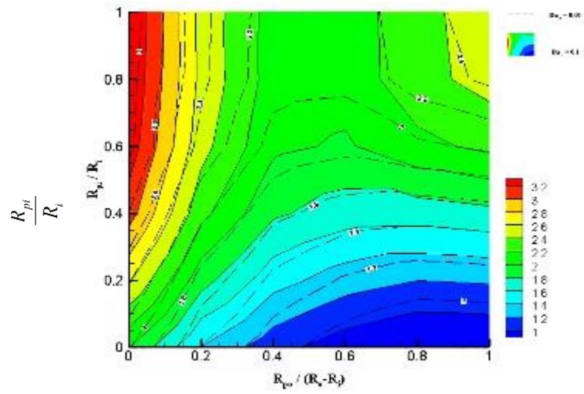


(d) $\frac{R_{po}}{R_o - R_i}$

Fig. 14 Performance number for different porous metal foam thicknesses (Dai=0.0001, 0.001, Da_o=0.01): (a): Re_i=500, (b): Re_i=1000, (c): Re_i=1500 and (d): Re_i=2000.

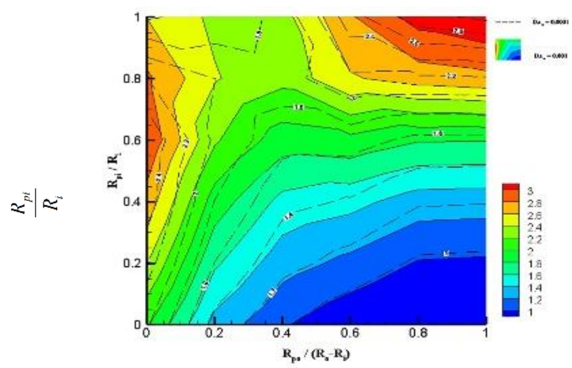


(a) $\frac{R_{po}}{R_o - R_i}$

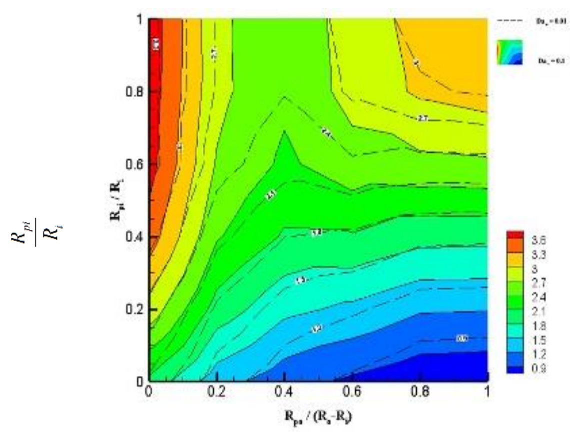


(b) $\frac{R_{po}}{R_o - R_i}$

Fig. 15 Performance number for different porous metal foam thicknesses (Dai=0.001, Re_i=500): (a): Da_o=0.0001, 0.001 and (b): Da_o=0.01, 0.1.



(a) $\frac{R_{po}}{R_o - R_i}$

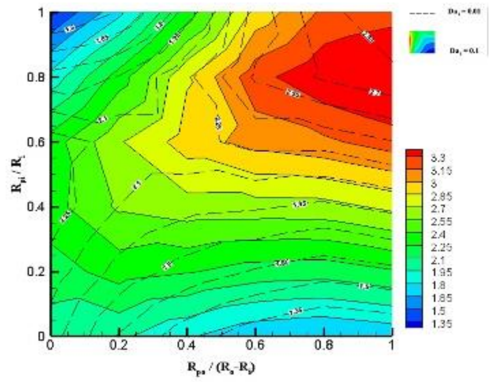


(b) $\frac{R_{po}}{R_o - R_i}$

Fig. 16 Performance number for different porous metal foam thicknesses (Dai=0.001, Re_i=1500): (a): Da_o=0.0001, 0.001 and (b): Da_o=0.01, 0.1.

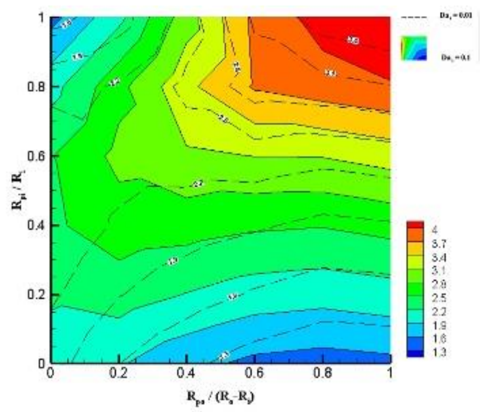
It can be seen in “Fig. 15 and Fig. 16”, by the increase of the permeability of the porous medium in the outer annulus, the highest performance number is only achieved when the outer annulus and the inner tube are completely filled with porous media.

$\frac{R_{pi}}{R_i}$



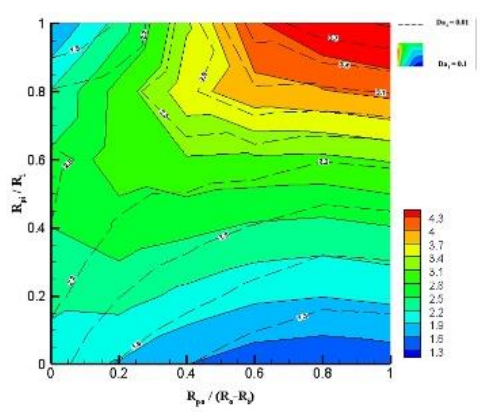
(a) $\frac{R_{po}}{R_o - R_i}$

$\frac{R_{pi}}{R_i}$



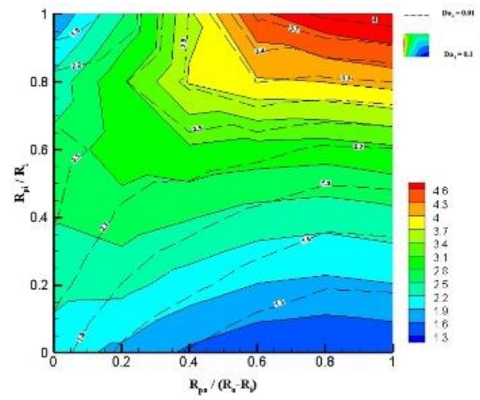
(b) $\frac{R_{po}}{R_o - R_i}$

$\frac{R_{pi}}{R_i}$



(c) $\frac{R_{po}}{R_o - R_i}$

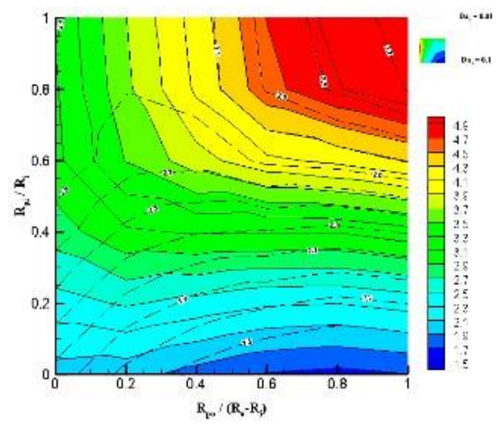
$\frac{R_{pi}}{R_i}$



(d) $\frac{R_{po}}{R_o - R_i}$

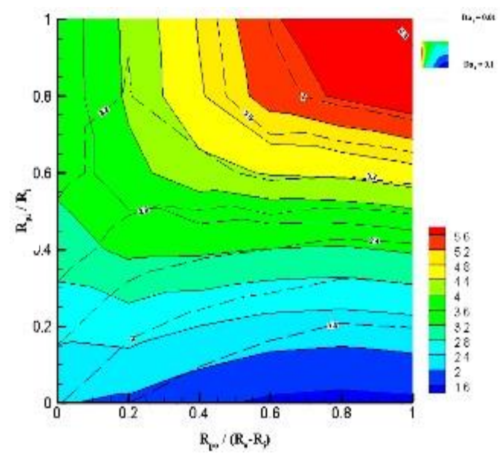
Fig. 17 Performance number for different porous metal foam thicknesses (Dai=0.01, 0.1, Da_o=0.0001): (a): Re_i=500, (b): Re_i=1000, (c): Re_i=1500 and (d): Re_i=2000.

$\frac{R_{pi}}{R_i}$



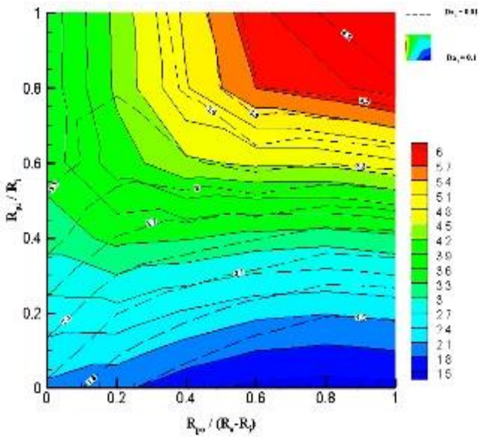
(a) $\frac{R_{po}}{R_o - R_i}$

$\frac{R_{pi}}{R_i}$



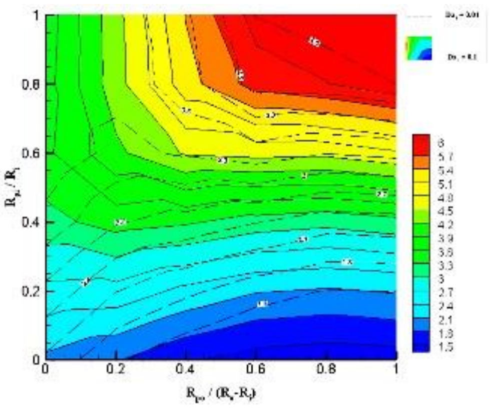
(b) $\frac{R_{po}}{R_o - R_i}$

$\frac{R_{pi}}{R_i}$



(c) $\frac{R_{po}}{R_o - R_i}$

$\frac{R_{pi}}{R_i}$



(d) $\frac{R_{po}}{R_o - R_i}$

Fig. 18 Performance number for different porous metal foam thicknesses (Dai=0.01, 0.1, Da_o=0.01): (a): Re_i=500, (b): Re_i=1000, (c): Re_i=1500 and (d): Re_i=2000.

If a porous medium with high permeability is used in the inner tube, the best performance number is achieved when both sides of the heat exchanger are fully filled with porous medium. As it is shown in "Fig. 17 and Fig. 18", even the lowest performance number is more than 1, which indicates the impressive effect of the porous media on the heat exchanger performance.

6 CONCLUSION

In this paper, heat transfer in a double pipe heat exchanger including nanofluid (water and Al_2O_3) and porous media was modeled numerically. Two phase mixture model and Darcy-Brinkman-Forchheimer model were used accordingly for the nanofluid flow and flow inside the porous media. Then, the effects of different parameters on the performance number were investigated.

1. In most of the cases, adding porous media can improve the performance number. For instance, keeping the Reynolds number at Re=500 and changing Da_i=Da_o from 0.0001 to 0.1, enhances the performance number by 138%. Although, in some cases, adding porous media may result in a negative effect and leads to a performance number lower than 1.
2. In all cases, increasing the permeability of the porous medium (either in the inner tube or in the outer annulus) increases the performance number of a double pipe heat exchanger. However, the effect of the porous medium permeability of the inner tube is more than that of the outer annulus.
3. As the Reynolds number increases, the performance number of the heat exchanger also increases. For example, by using porous media with Da_i=Da_o=0.1 in both sides of a double pipe heat exchanger and increasing the Reynolds number from 500 to 2000, the performance number was increased by 56.09%.
4. Adding nanoparticles to the base fluid (water) can increase the performance number. Adding nano particles with the concentration of 3% to the base fluid enhances the performance number by about 50%.

7 NOMENCLATURE

a	Acceleration (m s^{-2})
C_d	Drag coefficient
C_p	Specific heat J/(kg·K)
Da	Darcy number (m^2)
D_h	Hydraulic diameter (m)
f_{drag}	Drag coefficient
g	Aacceleration due to gravity (m/s^2)
h	Heat transfer coefficient ($Wm^{-2}K^{-1}$)
K	Permeability of porous medium (m^2)
k	Thermal conductivity (W m^{-1} K^{-1})
L	Length (m)
P	Pressure (Pa)
PN	Performance number
Δp	Pressure drop (Pa)
Q	Heat transfer rate (W)
r	Radius (m)
Re	Reynolds number

R_I	Internal radius of annulus (m)
R_o	External radius of annulus (m)
T_{in}	Inlet temperature (K)

Greek symbols

ε	Porosity
μ	Viscosity (kg m-1 s-1)
φ	Volume fraction
ρ	Density(kg m-3)

Subscripts

b	Base condition
bf	Base fluid
c	Cold
dr	Drift
eff	Effective
h	Hot
in	Inlet
i	Inner
m	Mixture
np	Nanoparticle
o	Outer
p	Porous medium

REFERENCES

[1] Siavashi, M., Bahrami, H. R. T., and Saffari, H., Numerical Investigation of Flow Characteristics, Heat Transfer and Entropy Generation of Nanofluid Flow Inside an Annular Pipe Partially or Completely Filled with Porous Media Using Two-Phase Mixture Model, Energy, Vol. 93, No. 2, 2015, pp. 2451 – 2466.

[2] Huisseune, H., Schampheleire, S. D., Ameel, B., and Paepe, M. D., Comparison of Metal Foam Heat Exchangers to a Finned Heat Exchanger for Low Reynolds Number Applications, Int. J. Heat Mass Transf., Vol. 89, 2015, pp. 1 – 9.

[3] Lotfi, R., Saboohi, Y., and Rashidi, A. M., Numerical Study of Forced Convective Heat Transfer of Nanofluids: Comparison of Different Approaches, Int. Commun. Heat Mass Transf., Vol. 37, No. 1, 2010, pp. 74–78.

[4] Mahdi, R. A., Mohammed, H. A., Munisamy, K. M., and Saeid, N. H., Review of Convection Heat Transfer and Fluid Flow in Porous Media with Nanofluid, Renew. Sustain. Energy Rev., Vol. 41, 2015, pp. 715–734.

[5] Abbassi, Y., Talebi, M., Shirani, A. S., and Khorsandi, J., Experimental Investigation of TiO2/Water Nanofluid Effects on Heat Transfer Characteristics of a Vertical Annulus with Non-Uniform Heat Flux in Non-Radiation Environment, Ann. Nucl. Energy, Vol. 69, 2014, pp. 7–13.

[6] Javadpour, A., Najafi, M. and Javaherdeh, K., Experimental Study of Steady State Laminar Forced Heat Transfer of Horizontal Annulus Tube with Non-Newtonian Nanofluid, J. Mech. Sci. Technol., Vol. 31, No. 11, 2017, pp. 5539–5544.

[7] Teamah, M. A., El-Maghlany, W. M., and Khairat Dawood, M. M., Numerical Simulation of Laminar Forced Convection in Horizontal Pipe Partially or Completely Filled with Porous Material, Int. J. Therm. Sci., Vol. 50, No. 8, 2011, pp. 1512–1522.

[8] Mahdavi, M., Saffar-Avval, M., Tiari, S., and Mansoori, Z., Entropy generation and heat transfer numerical analysis in pipes partially filled with porous medium, Int. J. Heat Mass Transf., vol. 79, Dec. 2014, pp. 496–506.

[9] Milani Shirvan, K., Ellahi, R., Mirzakhanlari, S., and Mamourian, M., Enhancement of Heat Transfer and Heat Exchanger Effectiveness in a Double Pipe Heat Exchanger Filled with Porous Media: Numerical Simulation and Sensitivity Analysis of Turbulent Fluid Flow, Appl. Therm. Eng., Vol. 109, 2016, Part A, pp. 761–774.

[10] Sheikhnejad, Y., Hosseini, R., and Avval, M. S., Experimental Study on Heat Transfer Enhancement of Laminar Ferrofluid Flow in Horizontal Tube Partially Filled Porous Media Under Fixed Parallel Magnet Bars, J. Magn. Magn. Mater., Vol. 424, 2017, pp. 16 – 25.

[11] Lu, W., Zhang, T., and Yang, M., Analytical solution of forced convective heat transfer in parallel-plate channel partially filled with metallic foams, Int. J. Heat Mass Transf., Vol. 100, 2016, pp. 718 – 727.

[12] Alsabery, A. I., Chamkha, A. J., Hussain, S. H., Saleh, H., and Hashim, I., Heatline Visualization of Natural Convection in a Trapezoidal Cavity Partly Filled with Nanofluid Porous Layer and Partly with Non-Newtonian Fluid Layer, Adv. Powder Technol., Vol. 26, No. 4, 2015, pp. 1230–1244.

[13] Shariat, M., Akbarinia, A., Nezhad, A. H., Behzadmehr, A., and Laur, R., Numerical Study of Two Phase Laminar Mixed Convection Nanofluid in Elliptic Ducts, Appl. Therm. Eng., Vol. 31, No. 14–15, 2011, pp. 2348–2359.

[14] Bianco, V., Manca, O., and Nardini, S., Numerical Investigation on Nanofluids Turbulent Convection Heat Transfer Inside a Circular Tube, Int. J. Therm. Sci., Vol. 50, No. 3, 2011, pp. 341 – 349.

[15] Hajipour, M., Molaei Dehkordi, A., Mixed-Convection Flow of Al2O3–H2O Nanofluid in a Channel Partially Filled with Porous Metal Foam: Experimental and Numerical Study, Exp. Therm. Fluid Sci., Vol. 5, 2014, pp. 49–56.

[16] Groșan, T., Revnic, C., Pop, I., and Ingham, D. B., Free Convection Heat Transfer in a Square Cavity Filled with a Porous Medium Saturated by a Nanofluid, Int. J. Heat Mass Transf., Vol. 87, 2015, pp. 36–41.

[17] Nield, D. A., Bejan, A., Convection in Porous Media. Springer New York, USA, 2006, pp. 154 –176.

[18] Miller, A., Gidaspow, D., Dense, Vertical Gas-Solid Flow in a Pipe, AIChE J., Vol. 38, No. 11, 1992, pp. 1801–1815.

[19] Pavel, B. I., Mohamad, A. A., An Experimental and Numerical Study on Heat Transfer Enhancement for Gas Heat Exchangers Fitted with Porous Media, Int. J. Heat Mass Transf., Vol. 47, No. 23, 2004, pp. 4939–4952.

[20] Göktepe, S., Atalık, K., and Ertürk, H., Comparison of Single and Two-Phase Models for Nanofluid Convection at the Entrance of a Uniformly Heated Tube, Int. J. Therm. Sci., Vol. 80, 2014, pp. 83–92.