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Research Article

Computational Fluid Dynamics Analysis for Heat Transfer Enhancement in Single Horizontal Pipe Heat Exchanger Full Filled Using Different Types of Porous Material

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Abstract:
As a result of significant scaling challenges and complex engineering, there has been considerable academic interest in recent years in the use of porous materials to improve forced convection heat transfer. The current work involves computational fluid dynamics (CFD) numerical simulation using three types of porous material (glass, steel and ceramics) with different diameters (0.004, 0.006 and 0.005 m), respectively to optimize heat transfer for a concentric single-tube heat exchanger with a length (1 m) and a diameter (0.03 m) exposed uniform heat flux on the outer wall (60kW/m^2). The governing equations of steady single-phase turbulent flow were solved using the commercial program (Ansys Fluent) for the Reynolds number range (10000-19000). Under the same operating conditions, the four cases of the heat exchanger tests were carried out, namely the three cases of the porous medium plus the exchanger without the porous medium. The numerical results showed the heat transfer rate (Nusselt number) improved by (92.563%) when using the porous material ceramic type compared to the empty pipe, while using the porous material (glass and steel) percentage increased (91.44 and 87.86%), respectively. Moreover, the friction factor may be affected by the inclusion of the porous material, and by increasing the Reynolds number gradually decreases. The current study proposes the inclusion of nanomaterials as a composite technology with porous material to improve the heat transfer properties and the flow of various fluids through the heat pipe.

Keywords: *Porous Media, Turbulent Flow, Heat Transfer, CFD, Single Pipe Heat Exchanger*

1. Introduction

Heat exchangers are pieces of equipment designed to make the process of transferring heat between two fluids that is, from one fluid with a high temperature to another without mixing the two fluids simpler and easier. Heat exchangers find extensive usage in a wide range of industrial engineering applications, including nuclear power plants, thermal power plants, chemical processing facilities, refrigeration and air conditioning systems, and dairy farms, among others. Conduction via the wall between the two fluids and convection inside the fluid itself carry out the heat transfer in the heat exchangers [1]. The use of barriers and fins for various technical applications and various methods has been proposed. To increase and improve the heat transfer properties in industrial applications there is a method in which porous materials are used that possess such mechanical properties as porosity, permeability and others [2].

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Numerous studies have demonstrated that using porous materials to enhance heat transfer outperforms other methods in this regard, and porous media with high thermal conductivity are well-known for their ability to do so. Porous media find widespread applications in various fields and industrial settings. Examples include solar energy storage systems, thermal insulation systems, large electrical coil cooling in high-power machinery, heat pipe, and heating equipment in power plants, and the pipeline network in heat exchangers [3]. Kurhade Anant Sidhappa et al. [4] carried out an investigation using twisted tape inserts with circular holes in the tube to enhance forced convection heat transfer. The results obtained within the Reynolds number (2000 to 12000) showed that the number of offspring increases with an increase in Reynolds number, with the highest number of offspring obtained when inserting a twisted tape with a torsion ratio of 5.5. Additionally, the coefficient of friction increases when the torsion ratio is (5.5). Used three different types of twisted tape with different torsion ratios (5.5, 6.5 and 8.5). Ehsan Fadhil Abbas et al. [5] described experimental and computational research that used spiral wire gaskets inserted into the pipe bundle to enhance heat transfer in a tubular casing exchanger. Utilize four distinct measures (4.8, 6.4, 8 and 9.6 mm) for the steps separating the spiral rings. The comparison findings demonstrated that employing a spiral wire filler with a step of 9.6 mm enhances the heat transfer coefficient on the pipe side and that utilizing the same filler yielded the best efficiency. S. K. Kadhim [6] analysis of the process of heat transfer by free convection in a longitudinally finned cylinder featuring a rectangular section and a triangular section. The researcher arranged eight fins at an angle of 45, placing them at various inclination angles between 0 and 90 degrees. The experiments were carried out using a thermal overflow range of (2279.24 - 4418.84 W/m^2) and for the limitations of the number of Rayleigh ranging from (16800000- 346000000). The findings demonstrated that convectional heat transfer increases with increasing Rayleigh number; the maximum amount was observed in the case of a rectangular fin at an angle of inclination of 30°, and the lowest amount occurred at an angle of inclination of 60° (inside the airway). M. E. Nakhchi et al. [7] carried out an experimental investigation utilizing four internal tube models (smooth tube, semi-annular finned tube, annular finned tube, and annular spiral finned tube) to enhance the performance of the double-tube heat exchanger. Using hot and cold water as a heat transfer medium, the tests were conducted in two modes of parallel and opposite flow, within the range of Reynolds numbers of (500 to 2250) for the other fluid's flow in the outer tube and (1000 to 6000) for the fluid's flow in the inner tube. The data collected demonstrated that the spiral annular finned tube model produced the maximum coefficient of convective heat transfer. Adnan M. Hussein [8] examines the combined load via a porous media between two inclined plates that are heated from below by a heat source and chilled from above using both computational and experimental methods. employed glass balls with a diameter of 12 mm as a porous medium. In the range of $100 < Ra < 650.5 < Pe < 100.0^{o} < ø < 90^{o}$, the experimental and numerical findings showed that the number of offspring grows with the ratio Ra/pe and the angle's inclination from (0^{o} to 90^{o}). Tahseen Ahmad Tahseen [9] provided a useful study to enhance heat transmission using stratified forced convection in a horizontally placed cylindrical tube utilizing two different types of porous media, including glass balls (10 mm in diameter) and gravel grains (10 mm in diameter), as well as continual thermal overflow. The obtained results demonstrated that, in both types of medium porosity, the number of localized offspring increases with increasing pellet number and that the distribution of dimensionless temperatures decreases with increasing length of the dimensionless tube and is less valuable for the yield. Kifah H. Hilal and Noor Samir Lafta [10] investigated the effects of adding a porous medium -type alumina ball with a diameter of (2.5 mm) on heat transmission in a double-tube heat exchanger through experimentation. Three cases (within a tube, outside a tube, and in both tubes) were utilized in the testing to examine the porous medium. The outcomes of the reverse flow scenario demonstrated that while the efficiency of the heat exchanger is better when utilizing a porous medium than when employing a conventional exchanger, an increase in the flow ratio (hot water flow rate /cold water flow rate) resulted in a drop in the heat exchanger's efficiency for each instance. K. H Hilal [11] presented an experimental investigation of increasing the rate of heat transfer by forced convection of a duct with dimensions of length, width, and height (1, 0.125, 0.125 m) respectively with a square cross-section. Coiled metal wire was used as a porous medium to enhance heat transfer. The experimental results indicated that as the fluid velocity increases, the number of nuclides gradually increases and the friction factor decreases. Xingzhen Zhu et al. [12] presented a numerical investigation to improve the forced convection heat transfer of a two-dimensional cooling channel by inserting a porous medium with a working fluid to enhance the average number of Nusselt. The Ansys program was used in numerical simulations, specifically the finite element volume method. The numerical results showed that with the addition of porous material, the thermal layers in the cooling channel also decrease, and the total heat sink of the fuel increases with a decrease in porosity. Mohammad Hadi Mohammadi et al. [13] a numerical analysis of the heat transfer rate and pressure drop along a tube of a shell and tube type heat exchanger containing six barriers filled with porous medium to increase heat transfer was reviewed with three different values of permeability (10^{-9}, 10^{-12} and 10^{-15} m^2) and porosity range (0.2-0.9).

Porosity contributed just 5% to both heat transmission and pressure loss, whereas baffle cut had the greatest influence on both. The ideal values for permeability, porosity, and baffle cut are then determined using a genetic algorithm to maximize heat transmission and minimize pressure loss in the system.

The goal of the current numerical analysis is to determine how the sort of porous materials employed in the annular gap of a single-tube heat exchanger exposed to heat flux of (60 kW/m^2) on external walls surface with an outer diameter of (0.03 m) and a length of (1 m)would affect the exchanger's thermal performance under parallel flow conditions. To evaluate the performance of porous media with that of the typical exchanger, three distinct types of spherical porous materials balls made of ceramic, steel, and glass with varying diameters (0.005, 0.006 and 0.004 m) with porosity (0.07, 0.84 and 0.525) respectively were chosen for this purpose.

2. Methodology of Numerical Solution

The finite volume method for numerical simulation of a three-dimensional tube was used in the current study. To estimate the governing equations of the flow of water inside the mathematical model represented by the equation of momentum, mass, and energy, a second-order method called the reverse of the wind direction was used. In the case of using a porous medium, the PRESTO compression method is used because it is also an interpolation method. This approach corresponds to steep pressure gradient problems. The SIMPLE scheme was used to pair velocity and pressure for the numerical solution [14-18]. The general methodology flow chart is shown in Figure 1. The first phase in the CFD simulation of a porous material channel was the preliminary investigation, which involved gathering data for the horizontal porous media channel modeling data input. Figure 2 shows the case of reaching numerical convergence in a mathematical solution where the temperature at the outlet of the tube must also be constant, and the natural residuals of the fluid temperature and other working variables are less than (10^{-6} and 10^{-8}), respectively.

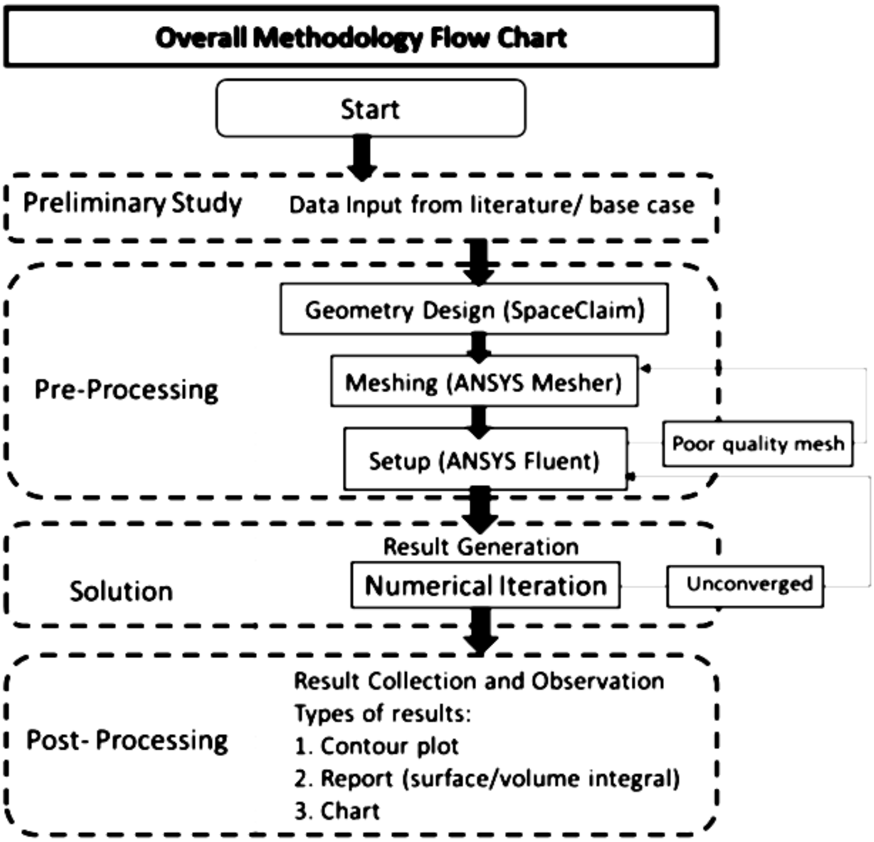

Fig. 1. Flow chart of the numerical simulation process.

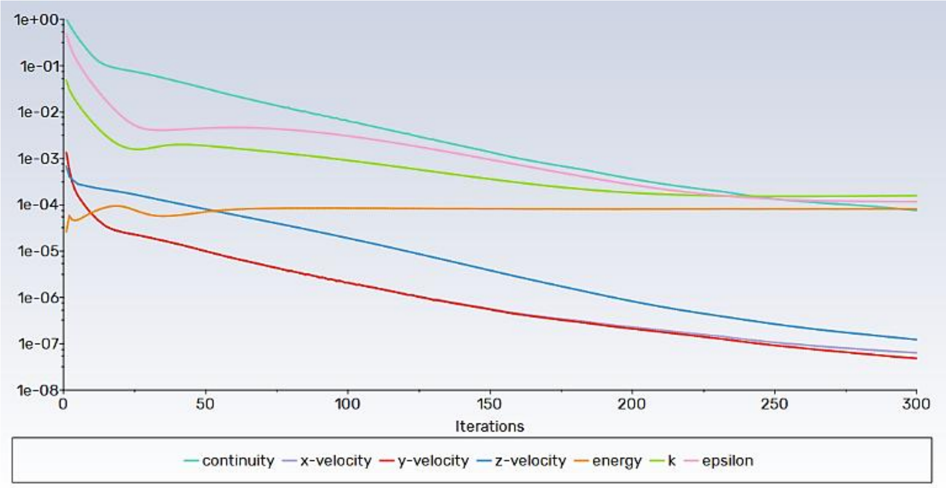

Fig. 2. Converging numerical simulation of the computational model.

2.1 Physical Model Description

Observe the movement of fluids and the transfer of heat in a pipe that has water introduced into a porous media layer covering its wall. The length of the pipe (1 m), which is completely coated with a different kind of porous material (glass, steel, and ceramics) with porosity (0.525, 0.84 and 0.07) respectively, allows for the full development of the temperature and velocity fields, as seen in Figure 3. The porous material is homogeneously inserted into the horizontal tube along the fluid flow. With the presence of porous material, the heat transfer processes are divided into heat conduction from the pipe wall to the surface of the balls, while heat transfer by convection from the surface of the balls to the working fluid as well as from the wall to the working fluid as shown in Figure 4. A constant uniform heat flux of $(60\ \mathrm{kW/m^2})$ is applied to the external upper and lower wall pipe. The flow is turbulent, single phase, steady state, incompressible, and three-dimensional. This study ignores the influence of radiation and natural convection. In this model, water with constant thermophysical characteristics is used. To analyze the mathematical model of heat transfer in a heat exchanger, a set of hypotheses is imposed so that the resulting mathematical model is simple enough for analysis. The following hypotheses have been presented to find formulas for the equations of thermal equilibrium, heat transfer rate, and boundary conditions:

- The exchanger operates under stable conditions (flow rate, fluid inlet temperature).
- The exchanger is hermetically insulated with the perimeter (perfect thermal insulation).
- There is no heat source inside the exchanger.
- The fluid in the flow section has a consistent velocity and temperature distribution.
- The wall resistance is uniformly distributed inside the exchanger.
- The resistance of longitudinal conduction in the fluid and the wall is inappreciable.
- There is no phase change of the fluid (condensation or evaporation).
- The surface area is uniformly distributed to the fluid.

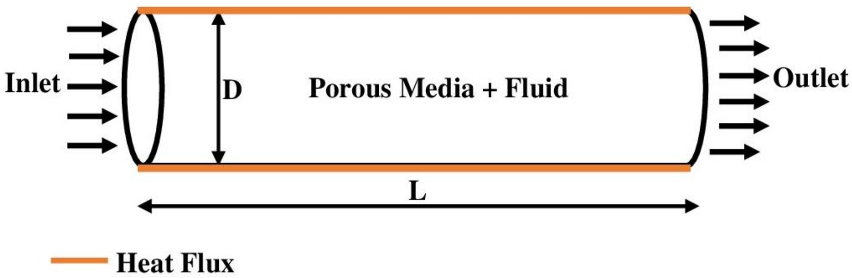

Fig. 3. Schematic diagram of computational model with boundary condition.

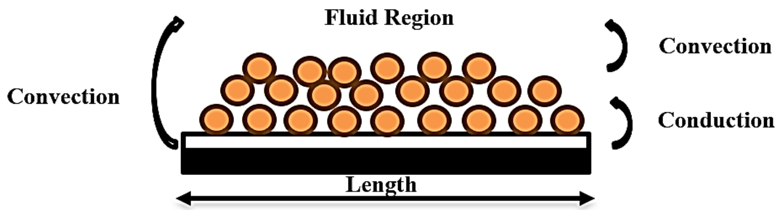

Fig. 4. Types of heat transfer process for fluid flow through porous material.

2.2 Fluid Governing Equation

In a single-phase pipe heat exchanger full of porous media, the governing equations for steady-state turbulent flow and heat transfer of working fluid water must be solved concurrently. The two sets of governing equations for the clear fluid and porous medium are as follows. The flow in the clear fluid areas and the porous medium are expressed using the Navier-Stokes equation and the Brinkman-Forchheimer-extended Darcy model, respectively [19 and 20]. In the field filled with the porous medium for the simulation of momentum transfer the extended Brinkmann-Forchheimer model of the Darcy model was used. To investigate the effect of orientation on the velocity field, a numerical scheme of the transient solution was included in the governing equations of the flow as a result of porosity changing in the porous matrix [21]. The following is a representation of the equations for the conservation of mass, momentum, and thermal energy, where subscripts 1 and 2 stand for the parameters in the clear fluid regions and the porous media, respectively [22-25].

Equation of continuity of porous zone:

$$\frac{\partial u_1}{\partial x}+\frac{\partial v_1}{\partial y}+\frac{\partial w_1}{\partial z}=0 \tag{1}$$

Equation of continuity of fluid zone:

$$\frac{\partial u_2}{\partial x}+\frac{\partial v_2}{\partial y}+\frac{\partial w_2}{\partial z}=0 \tag{2}$$

Equation of momentum of the porous zone:

$$\frac{1}{\beta^2}\left[u_1\frac{\partial u_1}{\partial x}+v_1\frac{\partial u_1}{\partial y}+w_1\frac{\partial u_1}{\partial z}\right]=\frac{-1}{\rho}\frac{\partial p_1}{\partial x}+\frac{\mu}{\rho}\left[\frac{\partial^2 u_1}{\partial x^2}+\frac{\partial^2 u_1}{\partial y^2}+\frac{\partial^2 u_1}{\partial z^2}\right]-\frac{\mu/\rho}{K}-\frac{C_F}{\sqrt{K}}\left|u_1\right|u_1 \tag{3}$$

$$\frac{1}{\beta^2}\left[u_1\frac{\partial v_1}{\partial x}+v_1\frac{\partial v_1}{\partial y}+w_1\frac{\partial v_1}{\partial z}\right]=\frac{-1}{\rho}\frac{\partial p_1}{\partial y}+\frac{\mu}{\rho}\left[\frac{\partial^2 v_1}{\partial x^2}+\frac{\partial^2 v_1}{\partial y^2}+\frac{\partial^2 v_1}{\partial z^2}\right]-\frac{\mu/\rho}{K}-\frac{C_F}{\sqrt{K}}\left|v_1\right|v_1 \tag{4}$$

$$\frac{1}{\beta^2}\left[u_1\frac{\partial w_1}{\partial x}+v_1\frac{\partial w_1}{\partial y}+w_1\frac{\partial w_1}{\partial z}\right]=\frac{-1}{\rho}\frac{\partial p_1}{\partial z}+\frac{\mu}{\rho}\left[\frac{\partial^2 w_1}{\partial x^2}+\frac{\partial^2 w_1}{\partial y^2}+\frac{\partial^2 w_1}{\partial z^2}\right]-\frac{\mu/\rho}{K}-\frac{C_F}{\sqrt{K}}\left|w_1\right|w_1 \tag{5}$$

Equation of momentum of the fluid zone:

$$\left[u_2\frac{\partial u_2}{\partial x}+v_2\frac{\partial u_2}{\partial y}+w_2\frac{\partial u_2}{\partial z}\right]=\frac{-1}{\rho}\frac{\partial p_2}{\partial x}+\frac{\mu}{\rho}\left[\frac{\partial^2 u_2}{\partial x^2}+\frac{\partial^2 u_2}{\partial y^2}+\frac{\partial^2 u_2}{\partial z^2}\right] \tag{6}$$

$$\left[u_2\frac{\partial v_2}{\partial x}+v_2\frac{\partial v_2}{\partial y}+w_2\frac{\partial v_2}{\partial z}\right]=\frac{-1}{\rho}\frac{\partial p_2}{\partial y}+\frac{\mu}{\rho}\left[\frac{\partial^2 v_2}{\partial x^2}+\frac{\partial^2 v_2}{\partial y^2}+\frac{\partial^2 v_2}{\partial z^2}\right] \tag{7}$$

$$\left[u_2 \frac{\partial w_2}{\partial x} + v_2 \frac{\partial w_2}{\partial y} + w_2 \frac{\partial w_2}{\partial z} \right] = \frac{-1}{\rho} \frac{\partial p_2}{\partial z} + \frac{\mu}{\rho} \left[\frac{\partial^2 w_2}{\partial x^2} + \frac{\partial^2 w_2}{\partial y^2} + \frac{\partial^2 w_2}{\partial z^2} \right] \tag{8}$$

Equation of thermal energy of porous zone:

$$(\rho C_p) \left[u_1 \frac{\partial T_1}{\partial x} + v_1 \frac{\partial T_1}{\partial y} + w_1 \frac{\partial T_1}{\partial z} \right] = k \left[\frac{\partial^2 T_1}{\partial x^2} + \frac{\partial^2 T_1}{\partial y^2} + \frac{\partial^2 T_1}{\partial z^2} \right] + \lambda_1 \tag{9}$$

Equation of thermal energy of fluid zone:

$$(\rho C_p) \left[u_2 \frac{\partial T_2}{\partial x} + v_2 \frac{\partial T_2}{\partial y} + w_2 \frac{\partial T_2}{\partial z} \right] = k \left[\frac{\partial^2 T_2}{\partial x^2} + \frac{\partial^2 T_2}{\partial y^2} + \frac{\partial^2 T_2}{\partial z^2} \right] + \lambda_2 \tag{10}$$

2.3 Grid of Numerical Study

To designate the optimization of numerical results in simulations, network independence is the appropriate term used using cells of small size in the computational domain. The term is called the independence of the network so that the network is divided into small parts to reach a correct result reached by the numerical model. The technique of ordinary CFDs consists of starting from the coarse grid and gradually improving it until the detected changes in numerical values are smaller than the previously specified acceptable error. There are this kind of two issues. First off, there are a few issues that arise when using other CFD software since it might be challenging to obtain even a single coarse mesh. Second, it may take longer to refine a mesh by a factor of two or greater. For software meant to be used as an engineering tool and built to tight production constraints, this is objectionable. Furthermore, the additional problems have contributed greatly to the reputation of CFD as a very complex, time-consuming, and expensive technology. To attain grid independence, the Nusselt number was ultimately calculated and organized in each case as shown in Figure 5

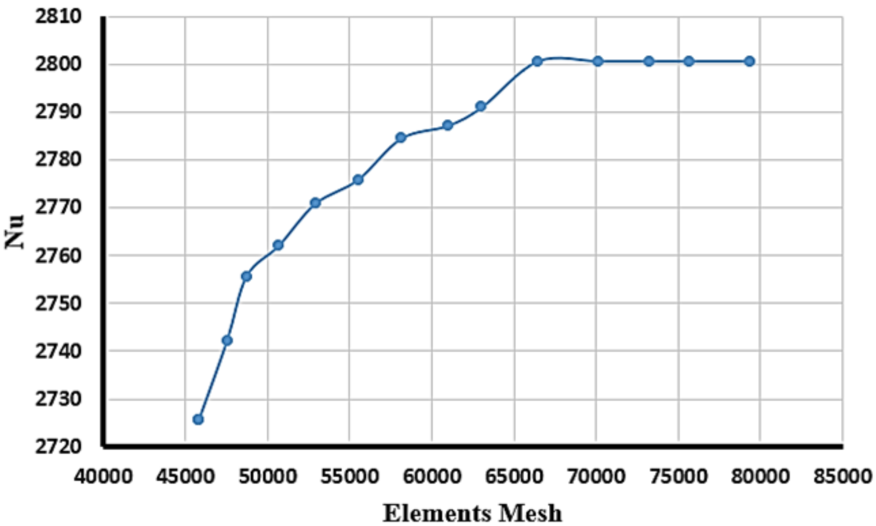

Fig. 5. Analysis of grid independence for the numerical model.

3. Results and Discussions

Figure 6 shows the change of the Nusselt number with the Reynolds number for an empty pipe filled with porous medium of the three types. Can gradually increase of the Nusselt number by increasing the fluid velocity inside the pipe and adding the porous material the Nusselt number increases gradually by observing the ceramic material gave the highest improvement in the percentage of heat transfer (92.563 %) compared to the materials (glass and steel) by a percentage (91.44 and 87.86 %), respectively, compared to the pipe without porosity. In addition, the porosity significantly affects, as the porosity decreases, the heat transfer rate increases, so the order of heat transfer of porous materials (ceramics, glass, steel) with porosity (0.07, 0.525, 0.84), respectively.

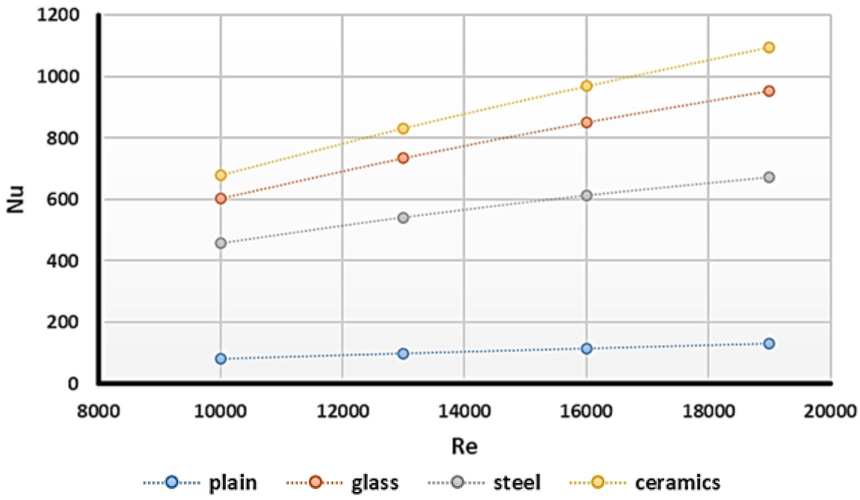

Fig. 6. Change of Nu against Re of pipe with/without porous media.

Figure 7 shows the change of the friction factor against the Reynolds number in two different cases, the first is the single pipe of the heat exchanger with the working fluid only, and the second case is the porous material pipe of different materials inserted with water. The friction factor can be observed gradually decreasing as the fluid velocity increases, the inverse relationship also decreases with the inclusion of the porous material, but with higher values compared to the empty pipe, this means the porous material obstructed the flow inside the heat exchanger pipe.

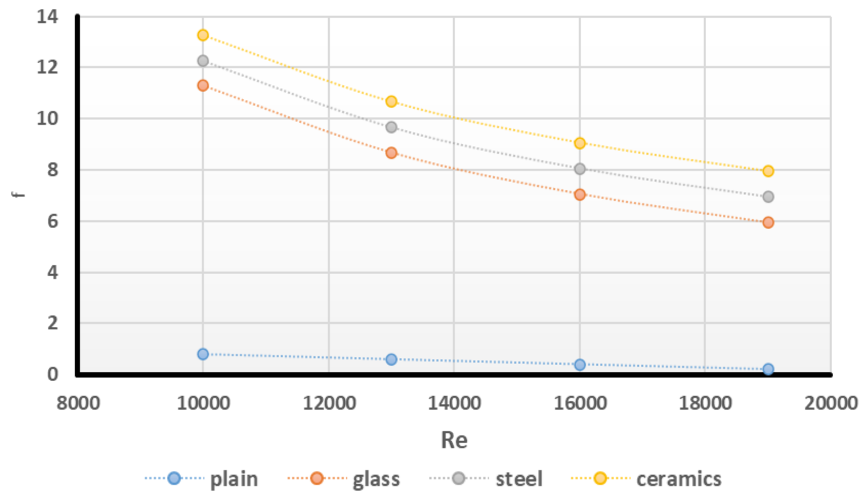

Fig. 7. Change of Re against f of pipe with/without porous media.

In Figure 8, the temperature distribution of the water liquid within the smooth heat exchanger's single tube and the tube containing a filled porous material are displayed. This break of the temperature cloud scheme occurs when the liquid penetrates the porous medium due to the speed fluctuation. Compared to the smooth tube, the thermal stratification in the porous media tube has considerably softened, and the cloud picture shows that the temperature near the axis of the porous media zone rises more quickly. Moreover, it can be observed that the temperature increases by increasing the fluid flow along the horizontal axis of the heat exchanger tube, where with the inclusion of the porous material, the fluid gains heat higher compared to the empty tube, where the heat gradient of the smooth tube is almost equal to the high value on the outer surface and low inside, while the tube filled with porous material, where the temperature gradient changes in the middle of the tube, the temperature rate becomes higher compared to the smooth tube.

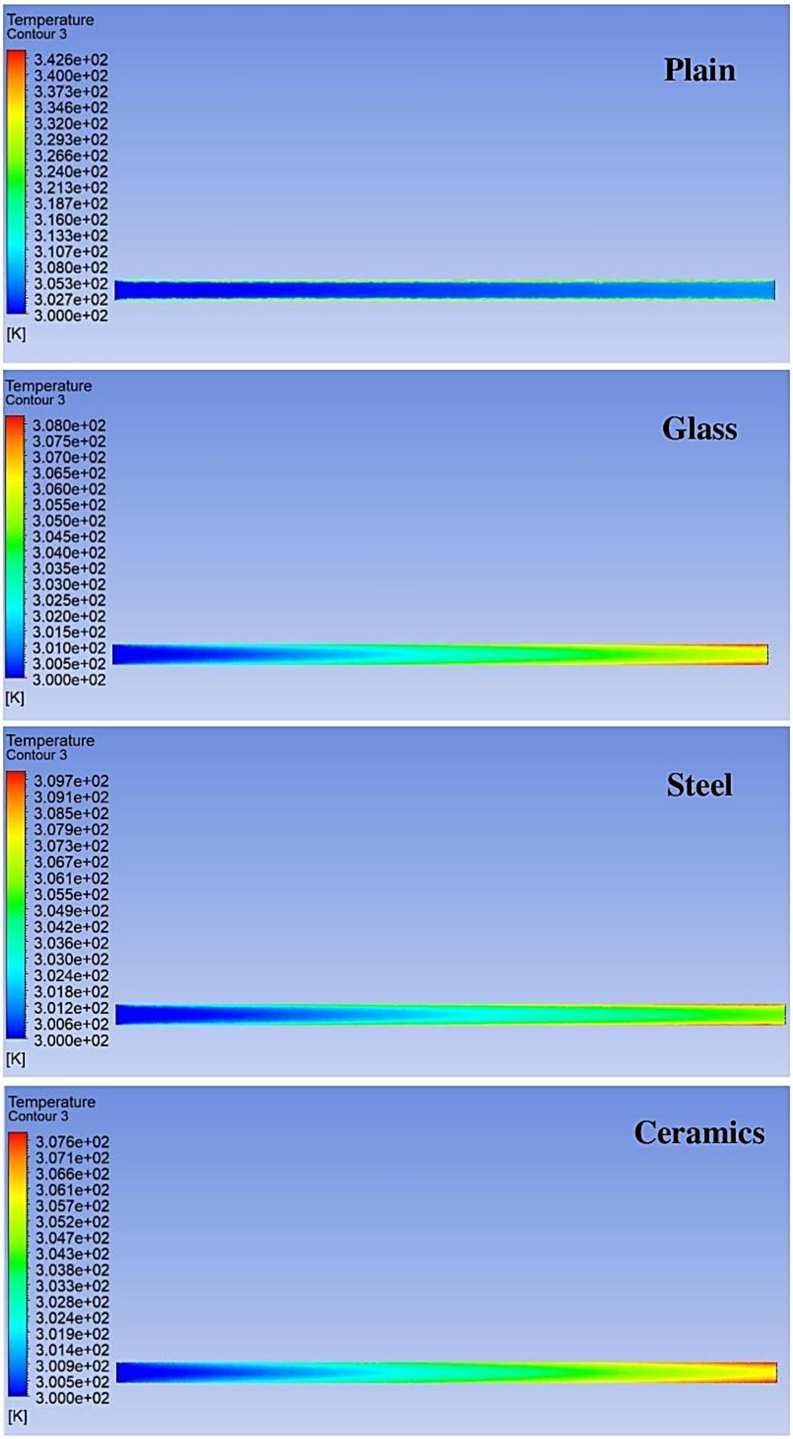

Fig. 8. Contour temperature of plain pipe and various types of porous media at Re of (10000).

The velocity distribution of the working fluid in the filled porous media pipe is depicted in Figure 9. Since the resistance of the porous medium is added, the position entering it will abruptly encounter resistance, causing its velocity to diminish and ultimately causing a stoppage. It is discovered that the velocity in the full-filled porous media pipe of the wall is lower than the velocity in the empty one and that the radial velocities of the unfilled porous media zone in the full-filled porous media pipe of the wall are significantly higher than those of the other zones. This suggests that more water flows into the unfilled porous media zone, increasing the turbulent kinetic energy of the water.

Fig. 9. Contour velocity of plain pipe and various types of porous media at Re of (10000).

4. Conclusions

The heat exchanger was subjected to numerical simulation tests with three different types of porous media in a parallel flow system and water was used as the heat transfer medium. The following conclusions were drawn from the comparison of the pressure drop and thermal conductivity of the porous media to those obtained for the exchanger when tested in its normal state, that is, without the porous medium:

- In general, the rate of heat transfer increases as the Reynolds number increases for all states. The effect of increasing the water flow on increasing the coefficient of heat transfer by forced convection was obvious.
- Because porous media provide more friction to the flow in addition to the friction already there from the fluid's viscosity and the roughness of the pipe surface, the inclusion of a porous medium in the exchanger increased the pressure drop.
- When employing porous material of the ceramic type as opposed to an empty tube, the heat transfer rate (Nusselt number) increased by 92.563%, while the percentage increase for porous material (glass and steel) was 91.44 and 87.86 %, respectively.
- Prevent the occurrence of the phenomena of Eddy generation and separation due to the presence of porous materials.
- Using a porous material causes an increased pressure drop through the pipe.
- The coefficient of heat transfer by localized forced convection gradually decreases with an increase in the length of the axis of the flow of the working fluid and increases with an increase in the Reynolds number.
- The friction factor gradually decreases by increasing the Reynolds number.
- The temperature distribution increases towards the axis of fluid flow inside the pipe. It can be observed when the porous material is inserted higher compared to the empty tube.

To enhance the heat transfer characteristics and the flow of different fluids via the heat pipe, the current study suggests using nanoparticles as a composite technology with porous material.

Nomenclature

$C_{1\varepsilon}$ and $C_{2\varepsilon}$	Equation constant
C_F	Inertial coefficient
C_p	Specific heat at constant pressure of working fluid ($J.kg^{-1}. K^{-1}$)
f	Friction factor
K	Permeability of porous media
k	Thermal conductivity of the working fluid ($W.m^{-1}. K^{-1}$)
Nu	Nusselt number
P	The pressure of working fluid ($N.m^{-2}$)
Re	Reynolds number
T	Temperature of working fluid (K)
u	Working fluid velocity in the direction of x ($m.s^{-1}$)
v	Working fluid velocity in the direction of y ($m.s^{-1}$)
w	Working fluid velocity in the direction of z ($m.s^{-1}$)
μ	Dynamic viscosity of working fluid ($N.s.m^{-2}$)
λ	Viscous of heating
ρ	Density of working fluid ($kg.m^{-3}$)

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