Numerical analysis for enhancing transferred heat in porous counter flow heat exchanger

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Abstract. The numerical analysis is accomplished to simulate the heat transfer in double tube heat exchanger with and without porous media by utilizing ANSYS FLUENT 14.1. A geometry system is used to predict the heat transfers and pressure drop characteristics of the flow. Alumina (2.5mm diameter) as a porous media were added in to: inner tube (IP) , outer tube (OP) and both tube of heat exchanger (IOP) were tested with the variation of hot and cold fluid inlet temperature, mass flow rate ratio (mr) to evaluate their influence on effectiveness of heat exchanger, Nusselt number, number of heat transfer unit. The evaluating has been performed in the steady-state. Water was used as a working fluid in a double tube heat exchanger. The study was conducted at the hot and cold water mass flow rates between (0.0166 - 0.0833kg / s), (0.05- 0.116 kg/s) respectively. The inlet temperatures of cold and hot water were (20, 25 °C), (47, 55 °C) respectively. Noted from results that adding pad of porous increases effectiveness of the heat exchanger and the increasing rate of transferring heat as mass flow rate ratio increases and the highest value are obtained when alumina is in double pipe. Effectiveness decreases with increase in mass flow rate ratio but increases when using alumina as a porous media and the better value is obtained when alumina is in IOP, IP, OP and NP respectively. Effectiveness increased as the NTU increases.

Keyword: Numerical study- heat exchanger- porous media- forced convection-

1. Introduction

The methods of augmentation transferring heat are classified as effective or ineffectual processes. Those that demand outward potential to preserve an augmentation are called effective processes, such as surface or fluid vibration. On the other hand, the passive methods those which do not require external power, such as extended or rough surfaces. In addition, a crossbred mechanism which takes in two ways or more of all the effective and ineffectual processes are needed. Utilization of pored media for enhancement transmission of heat gives maximal implementation of heat transfer than the other mechanism. Pored pad with a maximal thermal conductivity has to protrude as an active way of heat transfer enhancements due to the high surface area to a volume proportion and intensive flow mingling. One example of industrial application involving porous media is heat exchanger . Narasimhan, et al [1] investigated numerically by using finite volume method to the impact of changeful permeability of a pored media (aluminum metal foam) pad in tube -to- tube heat exchanger in a cross flow (surface to volume ratio, \( \alpha = 100–300 \) m²/m³). The cooled fluid having (0.7 Prandtl number) is owning a laminar stream (100 > Re > 10) for (\( K_i = 10^{-5} \) and \( 10^{-10} \) m² respectively). The results showed that increased 20% on the Nu as a function of Re number.
of plate of twain parallel-sheet ducts to enhance transmission of heat as distinctive parameter to flow in the study. Darcy-Brinkmans-Forchheaimer pattern was utilized for modelling the current fluid during the extended surface at Reynolds number (200-2000). The results show that higher Nusselt number obtained when porous channel completely full and caused high pressure drop while utilizing Also, Moraga et al [2] examined numerically a fluid flow and heat transmission for concentrated heat exchanger which employed exhaust gases of a normal gas in pored media(alumina) lying in a cylindrical pipe (length=50cm) to rising the temperature of air passing during a void between two tubes. Numerical emulation are built for that amounts of Reynolds number guttural air passing, for (Re = 20000 - 1000-500-100) with consideration constant of excess air (ϕ = 4.88 & porosity Ɛ = 0.4) and exhaust gas inlet speed (U0 = 0.43 m/sec). Result gained with the two-dimensional pattern are donated, a linear velocity of the combustion chamber isn't influenced with air Reynolds number that passing in the outer circular passage. While Reynolds number augments from Re = 100 to 2000, average temperature rises in exit of a heat exchanger about (40 K) on the intermediate at a studied time. Hooman [3] numerically examined a performance of the air-cooled condenser (finned pipe bundle). Range of Reynolds number was (600<Re<700). In the condenser, the finned pipe bundles are explained by a porous matrix. The dimensions of the fin are (thickness 0.6mm, and the tip being obviously by a 60mm space). For the porosity that is equal 0.769, the excess of pressure drop is 0.22 whereas enhancements in heat transfer are 2.26 time. Hamdan [4] numerically investigated, fully developed laminar forced convection enhancement by means of porous fins in duct which parallel-plates are isothermal. A higher conductance pored fins had connected to the internal surface pored fins strike minimal pressure drop with similar increment weighing versus completely padded porous conduit. The rate of increasing in a Nusselt number is (50%) when compared with no porous.

Odabaee, et al [5] numerically investigated heat transferred of aluminum mineral foam covered rigid roller in a cross stream and use air as a working fluid and the Reynolds number that use is (Re < 2 ×105). Effects of free flow speed and mineral foam properties like porosity and permeability on fluid flow besides heat have studied. Results have been united with heat exchanger having finned-channel to notice a large amount of heat transmission with an acceptable increase pressure losses which lead to a big area quality parameter to mineral foam covered rigid roller. Dehghan, et al [6] numerically investigated forcible convection in a ported tube, two-dimensional laminar with entrance and exit slots. Fluent program is using to solve continuity. Momentum and energy equation using a control volume method. Heat exchanger parallel plate channel, aluminum sphere as a porous media and water as a working fluid in a heat exchanger was applied. One wall of the channel was exposed to uniform heat flux while another wall is isolated. The Brinkmen-Forcheimer-extended Darcy pattern had employed to analyze flowing in pored pad in which the boundary and inertia effects were accepted to in accounting and a thermic desperation influences were not taken when used equation of energy. Statistical analytics were conducted to find the influence of Reynnold number, the diameter of particle to friction factor and heat transmission at the range, of Re (100<Re<800). It showed that the increase in a Nusselt number is(30%) when a particle diameter decrease from 0.2 to 0.1.

Abouei Mehrizi, et al [7] numerically investigated the enhancement of heating transferred in a porous medium (sand) platelet exchange heat and using air and water as a working fluid. This heating exchanger has designed by quadruple bore with entrance and exit ports that insulated to prevent heat to transfer. Three warm fins placed to fix temperature. Range of Reynolds number and the porosity employed (10<Re<60) (0.3<ε<0.9) respectively. At various Prandtl and Reynld numbers, amount of heat transferred and the average exit temperature of the fluid increased with decreasing the porosity. The results illustrated that higher effect of porous media for Nusselt number at a highest Prandtl and Reynold number. As well the place of fin is a noticeable impact on a Nusselt number. The result showed when adding a porous media the mean Nusselt number gains (2.36,1.2, 0.7) times to a fin (3-1) at (Pr=0.7 and Re=60), Nu increase (1.7, 4.5, 1.5) time for fin (3, 2, 1) at (Pr=6.5) and (Re=40).
All the studies have been reviewed, the pad (porous media) was placed in a specific place and does not changed, so that the aims of the present work in to investigate the effect of addition porous media porous media( Alumina) at three position IP, OP, and IOP in a double tube heat exchanger on its thermal performance. Numerical study by using (FLUENT PACKAGE 14.1) to illustrate the effect of four configurations on its effectiveness, under study state conditions. Also, this study concentrated on the effect of mass flow rate at two values inlet temperature to double pipe counter flow heat exchanger on the effectiveness, Nu, NTU, LMTD and heat transfer rate.

2. Mathematical formulation

Geometrical model illustrated in the current study included of tube in tube owning entrance and exit parts as shown in Figure1a and Figure1b, this tube from aluminum and the dimension of it are (63, 31mm) outer diameter, (2.5, 1.5mm) thickness, (1110mm) length, of outer and inner tube respectively, but the length of test section of (50mm) is in middle of heat exchanger and its increment length is (25mm, 36mm) at entrance and exit respectively of test section, it is used in order to remove eddies and obtain more uniform velocity profile. The porous media consist of alumina sphere (diameter=2.5mm, density=3868.253 kg/m³ and thermal conductivity is (36.014 kJ/kg.K). This system that shown in Figure1b is sketched by using a software program called GAMBIT PREMIUM 2009.
Water is used as a working fluid and flowing properties are presumed to be stable case, Newtonian fluid, not able to be compressed and $(r, \theta, z)$ dimensionally flow. The conserving equations for energy, momentum, and continuity equations are formed as following:

Continuity equation

\[
\rho \left( \frac{1}{r} \frac{\partial}{\partial r} (rv_r) + \frac{1}{r} \frac{\partial v_\theta}{\partial \theta} + \frac{\partial v_z}{\partial z} \right) = 0
\]  

(1)

Momentum Equation in the $r$-component:

\[
\frac{\partial}{\partial t} \left( \frac{1}{r} r v_r \right) + \rho \left( r v_r \frac{\partial v_r}{\partial r} + v_\theta \frac{\partial v_r}{\partial \theta} - \frac{v_r}{r} \frac{\partial \rho}{\partial r} + v_z \frac{\partial v_r}{\partial z} \right) = \mu \left( \frac{1}{r} \frac{\partial}{\partial r} \left( \frac{1}{r} \frac{\partial v_r}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 v_r}{\partial \theta^2} - 2 \frac{\partial v_\theta}{\partial r} + \frac{\partial^2 v_r}{\partial z^2} \right) - \frac{\partial p}{\partial r} + \rho g_r
\]  

(2)

Momentum Equation in the $\theta$-component:

\[
\frac{\partial}{\partial t} \left( \frac{1}{r} r v_\theta \right) + \rho \left( r v_r \frac{\partial v_\theta}{\partial r} + v_\theta \frac{\partial v_\theta}{\partial \theta} - \frac{v_\theta}{r} \frac{\partial \rho}{\partial \theta} + v_z \frac{\partial v_\theta}{\partial z} \right) = \mu \left( \frac{1}{r} \frac{\partial}{\partial r} \left( \frac{1}{r} \frac{\partial v_\theta}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 v_\theta}{\partial \theta^2} - 2 \frac{\partial v_\theta}{\partial r} + \frac{\partial^2 v_\theta}{\partial z^2} \right) - \frac{\partial p}{\partial \theta} + \rho g_\theta
\]  

(3)

Momentum Equation in the $z$-component:

\[
\frac{\partial v_z}{\partial t} + \rho \left( r v_r \frac{\partial v_z}{\partial r} + v_\theta \frac{\partial v_z}{\partial \theta} + v_z \frac{\partial v_z}{\partial z} \right) = \mu \left( \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial v_z}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 v_z}{\partial \theta^2} + \frac{1}{r^2} \frac{\partial^2 v_z}{\partial z^2} \right) - \frac{\partial p}{\partial z} + \rho g_z
\]  

(4)

Porous media are modeled by the addition of momentum source(S) term to the standard fluid flow equation, and $S$ has been calculated by the following equation[8]:

\[
S = -\left[ \frac{\mu}{\alpha} \dot{v} + C_2 \frac{1}{\varepsilon} \rho |\dot{v}| \right]
\]  

(5)

\[
\alpha = \frac{dp^1 \varepsilon^2}{175(1-\varepsilon)^3}
\]  

(6)

Energy equation:

\[
\rho C_p \frac{\partial T}{\partial t} + \rho C_p \left( v_r \frac{\partial T}{\partial r} + v_\theta \frac{\partial T}{\partial \theta} + v_z \frac{\partial T}{\partial z} \right) = K \left( \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2} \right)
\]  

(7)

The only change in the energy equation is when porous media are used to change thermal conductivity (K) to the(K_{eff}) effective thermal conductivity that is calculated from this equation [9]:

\[
K_{eff} = k_\alpha + K_d
\]  

(8)

\[
k_\alpha = K \left( 1 - \sqrt{1 - \varepsilon} + \frac{2 \sqrt{1 - \varepsilon}}{1 - \gamma^2} \ln \left( \frac{1}{\gamma^2} \right) - \beta + 1 - \frac{\beta - 1}{1 - \gamma^2} \right)
\]  

(9)

\[
\beta = 1.25 \left( \frac{1 - \gamma}{\varepsilon} \right)^{\frac{10}{9}}
\]  

(10)

\[
\gamma = \frac{K_{eff}}{K_{st}}
\]  

(11)

\[
K_d = 0.1 K_f \rho \varepsilon
\]  

(12)
\[ p_{e_i} = \frac{\nu_i dp}{ae} \]  
\[ V_i = \frac{4m}{3.14 \cdot D^2 \cdot \rho} \]

3. Numerical procedure

A control-volume used as a basic technique that forms the next steps is used for solution, [10]:

1- A grid is created on the domain, Figure 2 shows mesh of tube in tube heat exchanger. (3.1914) million cells will be applied in the current cases.
2- Algebraically groups of equation of velocities, pressure, and conserving scalars have organized through integrated the governed equations on every controls volume.
3- The discrete equations have literalize then resolved repeatedly.

Bounds limit have given to every section of computational field as following:

The inlet mass flow rate to the internal tube is fixed at a range of (0.0166, 0.033, 0.833) kg/sec The mass flow rate inlet to the external tube is specified over a range of (0.05, 0.066, 0.1166) kg/sec. Constant temperatures inlet to the internal and external tubes, are (47, 25) °C respectively. The outlet pressure is given at the access domain, is supposed to be atmospheric pressure. No sliding bounds limit has setting at the tube surface. These limits are employed to restrict fluid and solid zones.

![Figure 2. Mesh of tube in tube heat exchanger](image)

4. Results and Discussions

Heat is transferred from the hot to cold water flowing inside the tube in tube heat exchanger is analyzed by using ANSYS FLUENT. Figures. 3 to 6 show temperature contours of tube in tube heat exchanger in three dimensions at different temperatures and mass flow rates that enter heat exchanger with and without porous media temperature distribution along the sections, it has been noted declivity of turns and at the center of tube, the temperature at maximum value is occurred. Part (a) from Figures 3 to 6 shown heat transfer between hot and cold fluid for four cases NP, IP, OP and IOP respectively for fixed inlet conditions (mr=0.332, Tci=25°C, Thi=47°C). These figures show the temperature is reduced in the cases that use porous media when compared with NP case, but the minimum temperature is obtained when porous media filled double pipes. The drop in temperature when using alumina occurs because mechanization for transferred heat at porous media are convection heat transferred of tube surface to the water, transferred heat by convection from alumina beads to water also conduction for alumina to tube
surface that has high thermal conductivity, and the other cause is drop in surface temperature due to the use of porous media which increase surface area and mixing of fluid due to the presence of particles[11].

Part (b) from these Figures 3 to 6 shows the heat transfer between hot and cold water in double pipe heat exchanger for four cases NP, IP, OP and IOP as a respectively fixed inlet condition \((m_r=0.5, T_{ci}=25°C, T_{hi}=47°C)\). And part (c) from these figures also show the heat transfer inside heat exchanger for four cases NP, IP, OP and IOP respectively for fixed condition \((m_r=0.718, T_{ci}=25°C, T_{hi}=47°C)\). These figures show the minimum \(m_r\) is in part (a) so that in this case high heat transfer but in parts (b, c) the increase in mass flow rate leads to decrease in heat transfer between hot and cold water and increase in surface temperature when compared with part (a) for four cases NP, IP, OP, IOP because increasing inlet mass flow rate to tube in tube heat exchanger leads to change of flow pattern inside heat exchanger destroying thermal boundary layer. At part (a) low mass flow rate takes more time to make gradual change in temperature in transverse section.

**Figure 3a.** Temperature distribution at NP case and \((0.05,0.0166 \text{ kg/s})\) cold and hot mass flow rate respectively.

**Figure 3b.** Temperature distribution at NP case and \((0.066,0.033 \text{ kg/s})\) cold and hot mass flow rate respectively.

**Figure 3c.** Temperature distribution at NP case and \((0.116,0.0833 \text{ kg/s})\) cold and hot mass flow rate respectively.
Figure 4a. Temperature distribution at IP case and (0.05,0.0166 kg/s) cold and hot mass flow rate respectively.

Figure 4b. Temperature distribution at IP case and (0.066,0.033 kg/s) cold and hot mass flow rate respectively.

Figure 4c. Temperature distribution at IP case and (0.116,0.0833 kg/s) cold and hot mass flow rate respectively.

Figure 5a. Temperature distribution at OP case and (0.05,0.0166 kg/s) cold and hot mass flow rate respectively.
Figure 5b. Temperature distribution at OP case and (0.066, 0.033 kg/s) cold and hot mass flow rate respectively.

Figure 5c. Temperature distribution at OP case and (0.116, 0.0833 kg/s) cold and hot mass flow rate respectively.

Figure 6a. Temperature distribution at IOP case and (0.05, 0.0166 kg/s) cold and hot mass flow rate respectively.

Figure 6b. Temperature distribution at IOP case and (0.066, 0.033 kg/s) cold and hot mass flow rate respectively.
Figure 6c. Temperature distribution at IOP case and (0.116, 0.0833 kg/s) cold and hot mass flow rate respectively.

Figure 7 shows the theoretical results of mass flow rate and porous media on the effectiveness of the heat exchanger. It is observed from that the effectiveness decreases with an increase in mass flow rate ratio, but increases when porous media is used.

Figure 7. The theoretical effect of mr and porous media on effectiveness at (mr=0.332, 0.5, 0.718, Tci=20 °C, Thi=55 °C)

Figure 8 shows the theoretical relation between the effectiveness and NTU with the adding of porous media. This figure shows that effectiveness increases with increasing NTU and this increment will be increased when alumina is used as a porous media because of increasing surface area between water and pad through the tubes of a heat transfer.

Figure 9 depicts the relation between NTU and Cm/Cmax for four cases NP, IP, OP and IOP. This figure shows that NTU decreases with increase in heat capacity rate and the maximum NTU is obtained when the heat capacity rate equals zero but the NTU depends on overall heat transfer coefficient as a result to increase so that the case of IOP is higher than that of no porous and is less different that of IP, OP cases[12].
Figure 10 describes the relation between the number of transfer units (NTU) with Cmin/Cmax at the IOP case for experimental results obtained from ref. [13] and numerical results. It shows a good agreement between them with a maximum deviation (17%) percent.

**Figure 8.** Theoretical effectiveness- NTU relation and effect of porous media at (Tci=25°C, Thi=47°C)

**Figure 9.** Theoretical NTU- Cmin/Cmax relation and effect of porous media at (Tci=25°C, Thi=47°C)
5. Conclusion
In this study a porous heat exchanger was investigated numerically. The heat transfer rate increases as the mass flow rate ratio increases also the heat load is effected by using alumina and the highest value is obtained when alumina is in double pipe. Effectiveness decreases with increase in mass flow rate ratio but increases when using alumina as a porous media and the better value is obtained when alumina is in IOP, IP, OP and NP respectively. Effectiveness Increased as the NTU increases. Adding porous media increases effectiveness of heat exchanger

**Nomenclature**

| Symbol | Description                        | Unit                 |
|--------|------------------------------------|----------------------|
| $C_p$  | Specific heat at constant pressure. | (kJ/kg°C)            |
| $D_p$  | Diameter of particle               | (m)                  |
| $K_f$  | Thermal conductivity of fluid      | (W/m°C)              |
| $K_{eff}$ | Effective thermal conductivity | (W/m°C)              |
| $K_a$  | Alumina thermal conductivity       | (W/m°C)              |
| $g$    | Acceleration due to gravity        | (m/s²)               |
| IP     | Inner porous                       |                      |
| IOP    | Inner-outer porous                 |                      |
| NP     | Non porous                         |                      |
| NTU    | Number of heat transfer units      |                      |
| OP     | Outer porous                       |                      |
| $P$    | Pressure                           | (N/m²)               |
| $Pr$   | Prandtl number, $Pr = \frac{\nu}{\alpha}$ |                      |
| $Re$   | Reynolds number, $Re = \frac{VD}{\nu}$ |                      |
T  Temperature (°C)
υ  Velocity component in the (r,θ,z) (m/s)
μ  Fluid viscosity (kg/m.s)
ρ  Density (kg/m³)
ε  Porosity
θ  Kinematic viscosity of air (m²/s)

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