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Thermal Performance Assessment of a Double-Pipe Heat Exchanger Utilizing Metal Foam and Nanofluids Oxide

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ABSTRACT

The current numerical study aims to discover the outcome of the metal foam and nanofluids on the performance of a heat exchanger in contrast with smooth heat exchanger. Water was used as a working fluid at flow rate 2,3,4,5 Lpm. The outer pipe has diameter 60 mm and length 609.6mm, and an inner pipe with an diameter 20 mm. The numerical simulation was conducted by using ANSYS FLUENT 2022 R1 software. Metal foam with the porosity of 0.9 and 40 ppi was used. The cases considered in the analysis including a heat exchanger with a full copper foam baffle, metal foam with three shapes of holes triangle, square and circle and nano fluid of 1% CuO. Results showed the highest cold-water temperature is obtained when using full metal foam baffles with an increasing in the percentage 10% compared with smooth pipe, full metal foam baffles result in a substantial improvement Nu of up to 184 % while, CuO nonfluids enhances Nu by 50%, metal foam with circle holes provide PEC of 1.92 while, metal foam with circle holes combined with 1% CuO provide the maximum value PEC of 2.14.

Keywords: performance evaluate criteria; metal foam baffles, copper foam; heat transfer enhancement, nanofluids

1. Introduction

The heat exchanger is device that transmissions thermal energy from hot fluid to cold fluid. Heat exchangers are in public use in many engineering applications, such as air conditioning and space heating, Petro chemical plants, natural gas processing, boilers, condensers, chemical processes and power generation [1]. Several techniques have been used to improve heat transfer in the heat exchanger; these methods can be advantageous from a practical point of view and their application could lead to increased thermal evaluations, energy savings, time savings, and equipment longevity. One of the most significant of these varieties is the metal foam, which can improve heat transfer as a result of the flow region's many vortices. The metal used in foam formation can very commonly aluminum, but other metals like copper or nickel can also be employed. The metal foam may be classified as closed porosity (close cell) and effective porosity (open cell) depending on the method of production. Another way to improve thermal performance is to use nanofluids that can be applied to an engineering problem such as cooling of electronic equipment, chemical process, and heat exchangers. There are two ways for simulating the Nanofluids: single and two phase. In the first methods, the Nanofluids is assumed by the researchers to treated as common pure fluids and conventional equations of energy, momentum and mass are used, with an effect on viscosity and conductivity which can be obtained from the experimental data or theoretical models. There are several semi analytical and numerical methods which were used by several authors so as to simulate the Nanofluids flow and the heat transfer. Many numerical and experimental studies that have dealt with the effect of meatal foam and nanoparticles on the performance and effectiveness of the heat exchanger, including: Jenan et al., 2019 [2] carried out a numerical study of the counter flow double pipe heat exchanger with air (cold fluid) and water (hot fluid) flowing in the inner pipe. Adding (10) fins from metal foam with porosity (0.93) in the space between the two pips and distributed regularly by the axial space. The results displayed that the heat transfer means coefficient improved with (129%). Yang et al.,2019 [3] explored numerically the performance with natural convection, and a two-dimensional axisymmetric simulation model was created for the shell and tube by using open-cell metal foam that had a 0.94 porosity and a 15 PPI pore density and results indicated that metal foam can significantly increase the heat transfer rate. The whole melting time was reduced by 88.548% compared to tubes without metal foam, while temperature response rate and heat flux were both raised by 834.27% and 774.90%, respectively. Chen et al., 2020 [4] investigated experimentally the performance of a novel shell-and-tube heat exchanger with a baffle of copper metal foam of 5,10,20 and 40PPI with porosity (0.9118- 0.9520), the metal foam baffle heat exchangers with (40PPI) and porosity (0.9132) showed the better performance. The study also investigated the influence of the baffle thickness. It was found that the overall performance of heat exchangers with metal foam increased as the thickness of the metal foam increased. Mohammadi et al., 2020 [5] investigated numerically the total heat transfer rate and pressure drop along a shell and tube heat exchanger with six porous baffles, three values for the baffle cut (25%, 35%, and 50%), The results of the study indicated that the low percentage baffle cut (25%) improves the heat transfer rate with high produced pressure drop. Although using baffles with 20% porosity and cuts less than 40% increased the heat transfer, it can cause a large pressure drop. Naqvi et al.,2021 [6] numerically examined the performance of three different types of shell and tube heat exchangers with aluminum foam including helical, segmental, and clamping under the variations of the porosity as (ε = 0.6, 0.7, 0.8, 0.9 and 0.95) and radius of porous media as (Rp = 0.6, 0.7, 0.8, 0.9 and 1.0). For the shell side, Nu decreased with the increase in the porosity from 0.6 to 0.95. To obtain a better thermal performance, the shell must be partially filled with porous media rather than filling it. Tamkhade et al., 2023 [7] numerically evaluated the performance of a double pipe heat exchanger using nickel metal foam had pore densities (10 to 50PPI) and 0.9 porosity. The results showed that when the pore density changed from 10 PPI to 50 PPI, the heat transfer rate increased. Changda Nie et al., 2024 [8] investigated numerically performance of heat exchanger with metal foam baffles were compared with that with solid baffles. The results showed that the pressure drop of water with metal foam baffles increases with the increase of pore density and outlet water temperature with foam baffles first increases and then decreases with the increase of pore density, and it reaches the peak value at an optimum pore density. The optimum one increases with the increase of porosity and foam baffle thickness. The foam baffles with optimum pore parameters reduced the pressure drop by 12.9% and increase the outlet and inlet temperature difference by 31.0% compared to the solid baffles. Tijani et al., 2018 [9] Focused on the effect of adding nanoparticles (Al_2O_3 and CuO) (0.05, 0.15, and 0.3) % concentration to the radiator coolant (mixture of water and Ethylene glycol with (50:50) concentration to both fluids, the simulation was done using ANSYS fluent. The performance of heat transfer was estimated according to the heat transfer coefficient, thermal conductivity, Nusselt number, and rate of heat transfer of the nanofluids. The study showed that (0.3) % CuO consecration nanofluid appeared high heat rate against (0.3) % Al_2O_3 consecration. Arya et al., 2018 [10] investigated a double pipe heat exchanger made from copper with inner and outer diameters of (6.35 and 12.7) mm respectively and the tube length of 230 cm. Method of two step is used with different weight concentrations of (0.1, 0.2 and 0.3%). The results indicate that at a weight concentration of (0.3%), the heat transfer coefficient improved by (27%) in contrast with ethylene glycol. Ahmad et al., 2019 [11], investigated the influence of using multiwalled carbon water nanofluids in a double pipe counter-flow heat exchanger with porous aluminum media with a porosity of 67%. The outcomes showed that the utilization of the aluminum porous media plates improved the heat transfer and increased the coefficient of the heat transfer to the highest of (35%) at the smallest amount of concentration (0.04%) with three plates of the porous media in the test range. Kiani and et al., 2019 [12] studied the using of (CuO) nanofluid with (0.5 and 1) % concentrations with water, water ethylene-glycol (80:20) as base fluids in the automotive radiator to show the influence on the heat transfer enhancement using the cooling system of a four-cylinder Peugeot engine (405 XU7). The results demonstrated that heat transfer at a high speed of fan was (33.7) % which was larger than that at low speed for volume fraction of (1). Azhar Hussain Shah et al., 2024 [13] studied experimentally the performance on a double-pipe heat exchanger using water, iron oxide, and zinc oxide at concentrations extending from 0.10% to 0.175%. The results displayed that the heat transfer coefficient, friction factor, and pumping power of (Fe_2O_3) and (ZnO) are higher than those of baseline water at concentrations of 0.1, 0.125, 0.150, and 0.175%. Most researchers have studied the effect of adding metal foam or nano particles to the heat exchanger, but research that deals with adding both materials together is limited. Therefore, we tried to cover in our study the effect of adding metal foam with holes and nano particles on the thermal performance of the heat exchanger.

2. Model description

Geometry

1- Smooth heat exchanger.
2- Heat exchanger with metal foam baffles.
3- Heat exchanger with metal foam combined with nanofluids

2.1.1. Double pipe heat exchanger with metal foam model

The physical model geometry consists of two concentrical pipes as exposed in the detailed drawing. The inner pipe was made of copper with 711.2 mm in length and 20 mm diameter. The outer pipe has a 612.6 mm length and 60 mm diameter with 1.5 mm thickness and it was made from steel. The smooth annular gap of the heat exchanger was filled with open-cell copper foam with porosity (Ɛ) 0.9 and pore density 40 PPI. The copper foam baffle was made as a complete ring with a thickness (t) (10 mm) and it was distributed as nine baffles (Nb=9) as shown in figure (1)

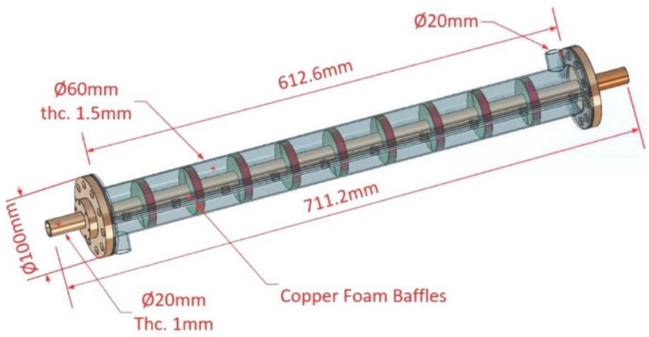

Figure 1. Geometry of Heat Exchanger with Metal Foam.

2.1.2. Heat exchanger with metal foam with holes

In this model internal holes are made in several shapes in metal foam baffle as shown in figures (2), (3) and (4) to study their effect on the thermal performance of the heat exchanger.

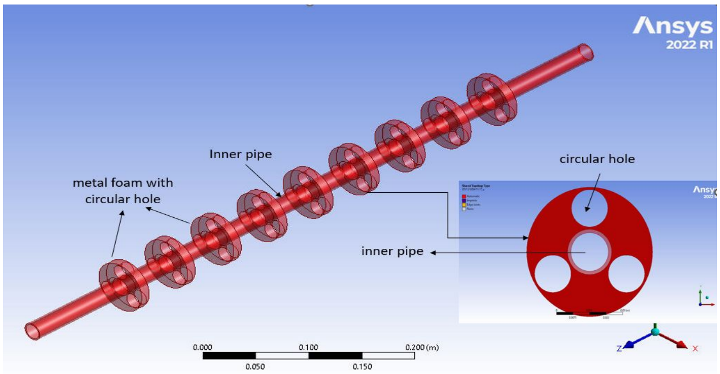

Figure 2. Metal foam with three circle holes

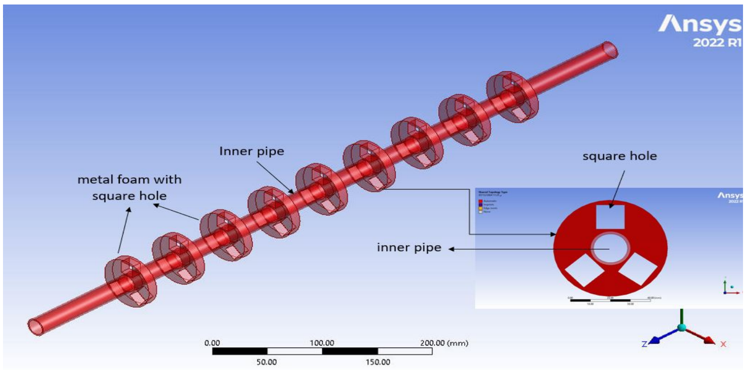

Figure 3. Metal foam with three square Holes.

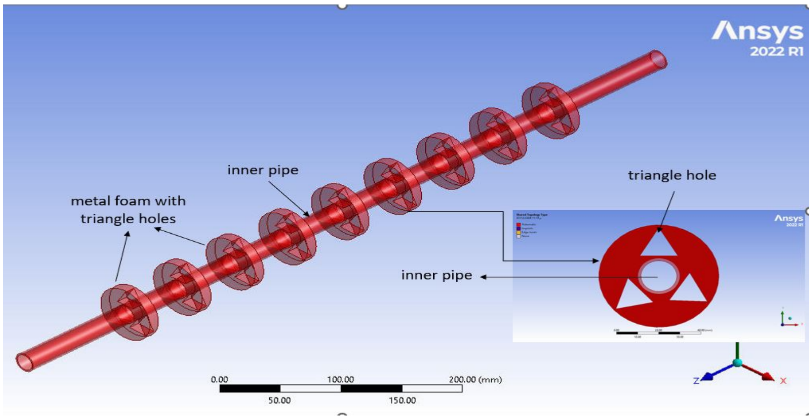

Figure 4. Metal Foam with Three Triangle Holes.

2.2. Mesh generation

Using ANSYS FLUENT 2022 R1 software the mesh was created. The utilization of structured hexahedral cells is carefully considered. It aims to minimize numerical dispersion as much as probable by carefully constructing the mesh, particularly near to the wall region. In the present work, a mixed of Hexa -Tetra, Wedges mesh was used as shown in figures (5) and (6).

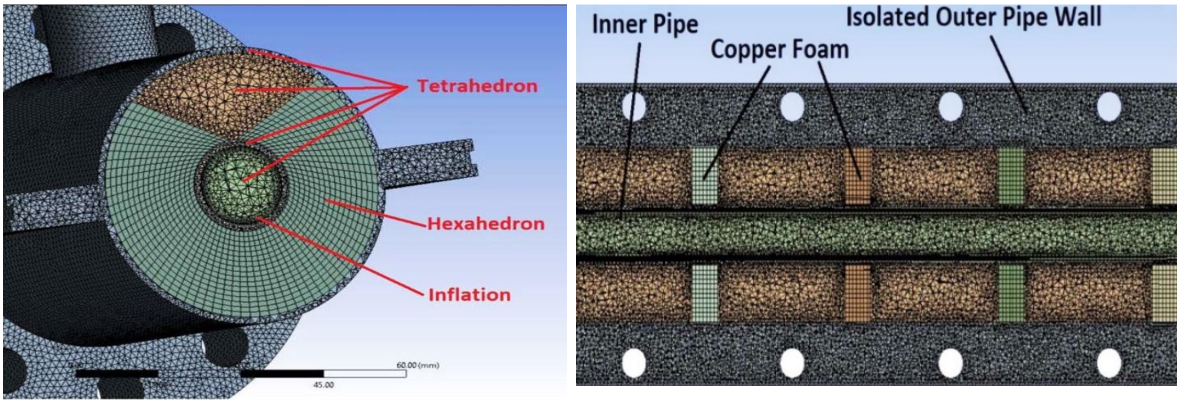

Figure 5. Mesh of Heat Exchanger with Metal Foam.

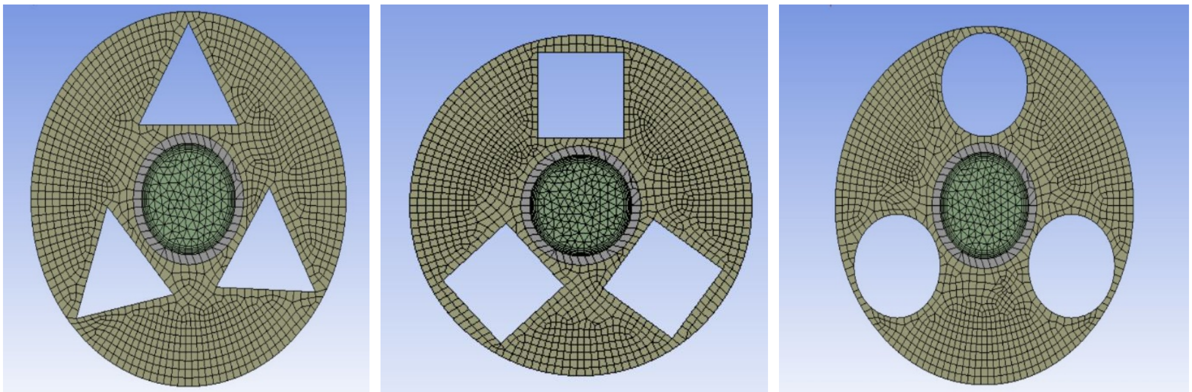

Figure 6. Mesh of Metal Foam with Holes.

3. Numerical assumptions

The evocative investigation is based on the following assumptions:
1- The annular is simulated under steady-state conditions.
2- Three-dimensional simulation
3- All thermophysical properties of the solid and fluid are constant.
4- The porous metal foam is uniform, rigid, isotropic, and homogeneous.
5- The flow is turbulent ($k-\varepsilon$) model.

6- The working fluid is Newtonian and incompressible

4. Governing equations

The presentation of the governing differential equations supports the analysis of heat transfer and fluid flow movements. The Brinkman-Forchheimer extended Darcy model and a volume-averaged generalized momentum equation, is provided to characterize the flow field within as an isotropic porous medium, while the Navier-Stokes equation represents the flow field in the fluid area. The Forchheimer-Brinkman extended Darcy equation with local thermal non-equilibrium for heat transfer and fluid flow in the copper foam was used as follows:

4.1. Continuity equation

The continuity equation for the three-dimensional averaged flow conservation of mass through a porous material is as follows:

$$\varepsilon \frac{\partial \rho}{\partial t} + \nabla.\left(\rho_f \vec{V}\right) = 0 \tag{1}$$

The porosity (ε) can be calculated as the ratio of connected void to total volume:

$$\varepsilon = 1 - \left(\frac{\rho_{m.f}}{\rho_s}\right) * 100\% \tag{2}$$

Since the porosity is independent of time and the fluid density is constant under the steady-state assumption, the conservation of mass equation for the fluid (1) will be the same as it: $\vec{\nabla}.\vec{V} = 0$ [14]

4.2. Momentum equation

The Forchheimer-Brinkman extended Darcy equation with the Boussinesq approximation will be utilized to simulate fluid flow through the copper foam:

$$\frac{\rho_f}{\varepsilon^2}\left(\vec{V}.\nabla\right)\vec{V} = -\nabla P + \rho_f \vec{g}\left[1 - \Phi(T - T_o)\right] + \frac{\mu_f}{\varepsilon}\nabla^2.V - \frac{\mu_f}{K}V - \frac{\rho_f C_1}{K^{1l2}}|V|.V \tag{3}$$

The porous media permeability (K) can be represented as below:

$$K = 0.00073\, dp^2 (1\text{-}\varepsilon)^{-0.224} \left[\frac{d_f}{d_p}\right]^{-1.11} \tag{4}$$

The pore diameter was evaluated by using the following equation:

$$d_p = 0.0245/PPI \tag{5}$$

$$\frac{d_f}{d_p} = 1.18 \sqrt[2]{\frac{(1\text{-}\varepsilon)}{3\pi}}\, \frac{1}{1\text{-}e^{-(1\text{-}\varepsilon)/0.04}} \tag{6}$$

The inertial loss coefficient (C1) was found by the equation[15] :

$$C_1 = 0.00212\, (1\text{-}\varepsilon)^{-0.132} \left(\frac{d_f}{d_p}\right)^{-1.36} \tag{7}$$

4.3 The energy equation

Two energy equations will be required for the local thermal non-equilibrium model in order to reflect the conservation of energy in the saturated fluid and the porous metal foam matrix, as described by [16]:

$$\varepsilon\left(k_f \nabla^2 T\right) + a_{sf} h_{sf}\left(T_s - T_f\right) = (\rho C_P)_f \vec{V}.\nabla T \tag{8}$$

kf: fluid thermal conductivity,
asf: interfacial area density
hsf: local heat transfer coefficient at the contacting surface between solid and fluid.

The interfacial area density and local heat transfer coefficient were found by the following equations

$$a_{sf} = \frac{3\pi d_f}{\left(0.59 d_p\right)^2}\left(1 - e^{\frac{-(1-\varepsilon)}{0.04}}\right) \tag{9}$$

$$h_{sf} = 0.76 Re_d^{0.4} Pr^{0.37} \frac{k_f}{d}, \qquad 1 \le Re_d \le 40 \tag{10}$$

$$h_{sf} = 0.52 Re_d^{0.5} Pr^{0.37} \frac{k_f}{d}, \qquad 40 \le Re_d \le 10^3 \tag{11}$$

$$h_{sf} = 0.26 Re_d^{0.6} Pr^{0.37} \frac{k_f}{d}, \qquad 10^3 \le Re_d \le 2 * 10^5 \tag{12}$$

Dispersion thermal conductivity was found as follows:

$$d = \left(1 - e^{\frac{-(1-\varepsilon)}{0.04}}\right).d_f \tag{13}$$

$$Re_d = \frac{\rho_f.u.d}{\mu_f} \tag{14}$$

Solid matrix:

$$(1-\varepsilon)k_s\nabla^2 T_s + a_{sf}h_{sf}\left(T_f - T_s\right) = 0 \tag{15}$$

According to [18], the local convective heat transfer at the porous-fluid interface between the fluid in the fluid area and the edge of foam ligaments may be represented as follows:

$$k_s\left(\frac{\partial T_s}{\partial r}\right) = h_{sf}\left(T_s - T_f\right) \tag{16}$$

Where: (k_s) is solid thermal conductivity.

5. Numerical setup

As a first step in the ANSYS FLUENT setting, double precision with four processes parallel solver was selected. The pressure was atmospheric, the time option was selected as a steady-state, and the formulation of the water velocity as absolute is considered by the solving method. By defining gravity (-g) in the (Y) direction. The type of solving and flow governing models (energy, viscous,) are selected for setting up the model. By activating the energy equation, the main settings of the numerical simulation are displayed. Viscous k-epsilon($k-\varepsilon$) , Realizable model was selected because it can enhance the accuracy of the fast flows and improve the accuracy of the eddy's flows using two equations of($k-\varepsilon$).

5.1.Boundary conditions setup

The boundary conditions which are applied in this study are listed in the following Table (1)

Table 1. Boundary Condition.

Inlet boundary condition for cold water		
1	Temperature	30 ºC
2	Volume flow rate	2 Lpm
Inlet boundary condition for hot water		
1	Temperature	75 ºC
2	Volume flow rate	2,3,4 and 5 Lpm
Pipe wall boundary condition		
Inner pipe	Thermal condition	via system coupling
	Material	steel, copper
Adiabatic wall	Heat Flux	0 W/m^2

6. Thermophysical properties of nanofluids

The particles are supposed to move at the same speed as the fluid due to the small size and low concentration of the nanoparticles. Furthermore, by taking into account the local thermal equilibrium, the nanofluids combination may be approximated to behave as a typical single-phase fluid with characteristics to be assessed as functions of those of the components [20]. The nanofluids properties depend on the physical properties of both the base-fluid and the nano particles. Table (4) shows the thermophysical properties.

The density and the heat capacity of the nanofluid can be calculated from the equation of [17]

$$\rho nf = \varphi p\rho p + (1 - \varphi p)\,\rho w \tag{17}$$

Where

ρnf : density of the nanofluids.

ρp : density of the nanoparticles samples.

ρw : density of the water (base fluid).

φp : nanoparticles' volume fraction.

The thermal conductivity and the viscosity proposed as follows:-

$$knf/kw = (1 + 7.47\emptyset) \tag{18}$$

Where

knf : thermal conductivity of the nanofluids samples.

kw : thermal conductivity of water (base fluid).

$$\mu nf = \left[\frac{1}{1-34.87\left(\frac{\mathrm{dp}}{\mathrm{dw}}\right)^{-0.3}(\varphi \mathrm{p})^{1.03}}\right] \tag{19}$$

Where

$$dw = 0.1\ \left(\frac{6M}{N\pi\ \rho w}\right)^{0.33} \tag{20}$$

M= Molecular weight of water = $1.8 * (10)^{-0.2}$kg/mol.
N= Avogadro number =$6.02 * * (10)^{23}$
μnf : Dynamic viscosity of the nanofluids.
μw : Dynamic viscosity of water (base fluid).

7. Performance evaluate criteria (PEC)

The performance evaluates criteria of the heat exchanger for the cold-water side is calculated as below [18]:

$$PEC= \frac{\frac{Nu\,mf}{Nu\,s}}{\left(\frac{f\,mf}{f\,s}\right)^{0.3}} \tag{21}$$

8. Results and discussions

The numerical results of double pipe heat exchanger are investigated under various boundary conditions, including alterations in the water flow rate at values of 2, 3, 4, and 5 lpm, with intake temperatures for hot and cold water set at 75°C and 30°C, respectively. Each volumetric flow rate value corresponded to a Reynolds number ranging from 7422 to 18556.

8.1. Effect of metal foam and nanofluids on water outlet temperature

The effect of varying the hot water flow rate, or (Re), on the output temperature of cold water utilising various metal foam baffles is displayed in Figure (7). The simulation results illustrated that the outlet temperature of cold water increases with an increase in the flow rate or the Re for all baffle models. This occurs as a result of the augmented flow velocity of hot water, which facilitates an improved heat transfer rate between the cold and hot water.The general heat transfer equation indicates that fluid flow rate and temperature differential influence the heat transfer rate. The heat transfer rate (Q) escalates with an increase in the hot water flow rate, while the cold water flow rate remains unchanged. This leads to an augmentation in the temperature differential (ΔT) of the cold water, hence elevating its temperature. In general numerous factors contribute to the elevation of cold water temperatures in heat exchangers. The primary metal foam enhances thermal mixing and dispersion within the shell, hence augmenting heat transfer efficiency in the system. Secondly, the research employed copper metal foams, characterised by their elevated thermal conductivity, hence enhancing heat transfer. Finally, the results indicate that the maximum temperature of cold water was achieved while employing full metal foam baffles compared to other scenarios. As a result, the augmentation in heat exchange between the hot and cold fluids resulted in an 10 % increase in the temperature of the cold water while, CuO nonfluids enhances cold temperature by 2% compared to the smooth pipe.

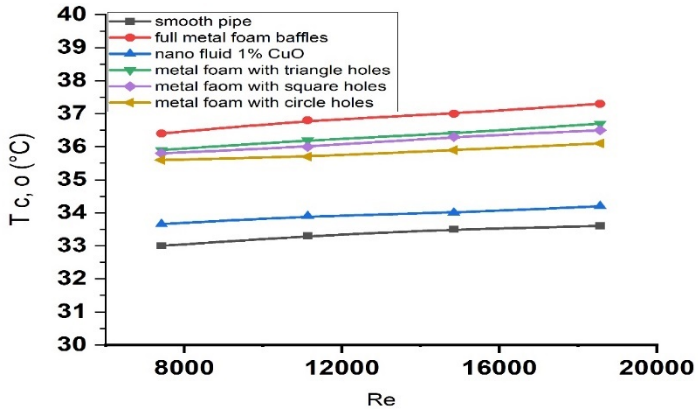

Figure 7. Effect of metal foam and nanofluid on cold water temperature

8.2. Effect of metal foam and nanofluids on Nu

Figure (8) illustrates the impact of altering the hot water flow rate, or Re, on the Nu of cold waters using different metal foam baffle models. The results indicated that the Nu increases with a rise in the Re for all baffle types employed in double-pipe heat exchangers. Augmenting the hot water flow rate, while maintaining a steady cold flow, contributes to a higher Re on the hot side, hence improving convective mixing and diminishing thermal resistance within the turbulent regime. Therefore, the higher inertia force relative to the viscous shear forces results in a substantial improvement in the overall heat transfer coefficient (U), facilitated by enhanced boundary layer renewal and increased turbulence. Moreover, the use of porous metal foam as a partition significantly enhances the effective surface area and promotes mixing. And the high thermal conductivity of copper foam baffles supports heat transfer through the wall, while the porous architecture interrupts the thermal boundary layer, resulting in enhanced U values. Accordingly, (Nu) increases by approximately 83% as (Re) ascends from 7422 to 18556, aligning with previous correlations that demonstrate a direct link between Nu and Re in turbulent flow. Also, the application of full metal foam baffles results in a substantial improvement of up to 184 % while, CuO nonfluids enhances Nu by 50% relative to a smooth pipe, attributed to the increased disturbance due to the sharp edge of this shape, augmented hydraulic interaction, and the expanded surface area for heat transmission. Overall, the observed improvements in Nusselt number (or heat transfer coefficient) are ascribed to the synergistic effects of elevated turbulence intensity, augmented surface area, and boundary layer disruption, all of which boost exchanger performance.

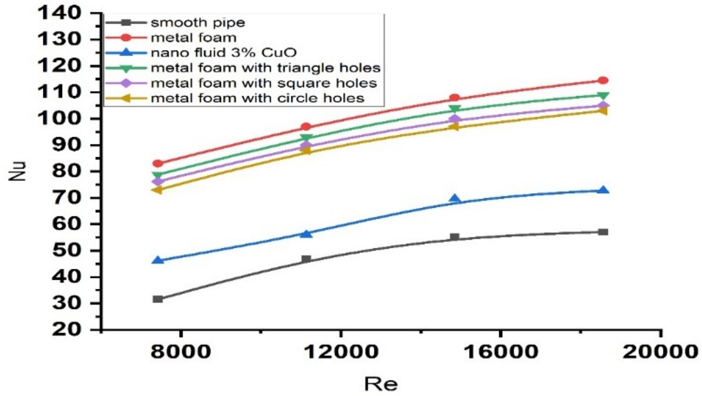

Figure 8. Effect of Metal Foam and Nanofluid on Nu.

8.3. Effect of metal foam and nanofluids on pressure drop

Figure (9) illustrates the relationship between pressure drop (Δp) and (Re) for diverse configurations of metal foam baffles and nanofluids.Therefore, the results demonstrate that porous foam structures significantly increase pressure drop relative to a smooth pipe due to increased flow resistance and intricate internal flow pathways. The complete usage of full metal foam baffles leads to a 4-times increase in pressure drop relative to a smooth pipe. So, the pressure drop is the most critical factor in minimizing the size and expense of a heat exchanger.As a result, the hole baffle configurations achieve a compromise between the improvement of heat transmission and hydraulic performance. Consequently, the pressure drop is reduced by 1.6 times for metal foam with circular perforations, compared to the whole baffle foam model.The simulation data also show that the added geometrical modifications (holes) significantly influence flow dynamics and friction characteristics. The augmented pressure drop associated with incorporating metal foam baffles can be attributed to several factors. Primarily, there is significant contact between the fluid and the walls, leading to an entirely stationary boundary layer, which is regarded as a stagnant region that exacerbates the pressure drop.

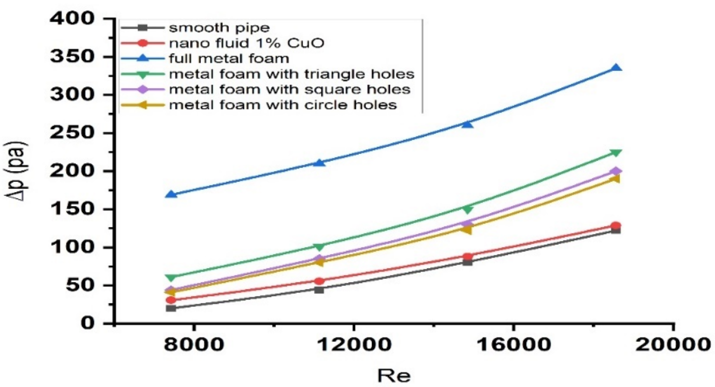

Figure 9. Effect of Metal Foam and Nanofluid on Pressure Drops.

8.4. Effect of metal foam and nanofluids on performance evaluates criteria (PEC)

This section presents a balanced comparison between the enhancement in heat transfer rate and Nusselt number relative to the increase in pressure drop and friction factor. Since the performance evaluation criteria (PEC) is a critical metric that reconciles the trade-off between thermal enhancement and pressure drop; it serves as a useful indication of overall heat exchanger performance. A poor thermal design for the heat exchanger happens when the PEC is less than one, meaning that the increase in the friction factor is larger than the increase in the Nusselt number. Figure (10) illustrates a comparative analysis of the Performance Evaluation Criterion (PEC) for several baffle topologies, both with and without the integration of metal foam. Unmodified plain metal foam exhibits the lowest pressure drop coefficient (1.44), signifying that although the foam enhances thermal conductivity, it fails to significantly reduce pressure drop without geometric optimization. The results show that for all the configurations studied, the metal foam circular hole apertures demonstrate the highest PEC value of 1.92

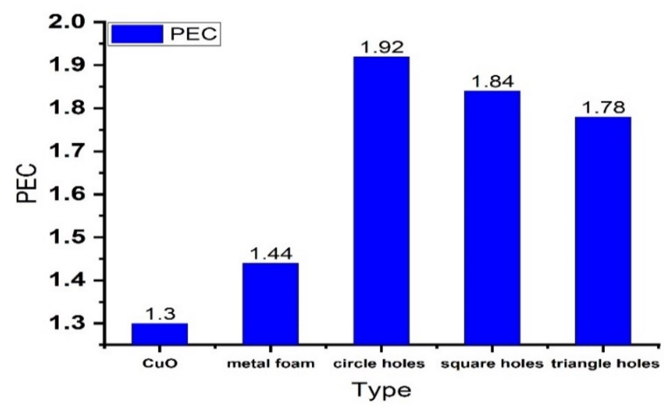

Figure 10. Effect of Metal Foam and Nanofluid on PEC.

8.5. Effect of metal foam combined with nanofluids on performance evaluates criteria (PEC)

Figure (11) analyses the simulation data pertaining to the Performance Evaluation Criteria (PEC) of two heat exchanger models utilising circular-holed metal foam and metal foam with circle holes combined with CuO nanofluids at 1% concentrations. . The highest PEC value of 2.14 is achieved at metal foam with circle holes combined with 1% CuO . The thermal advantages from improved conduction and turbulence dominate, resulting in superior PEC values.

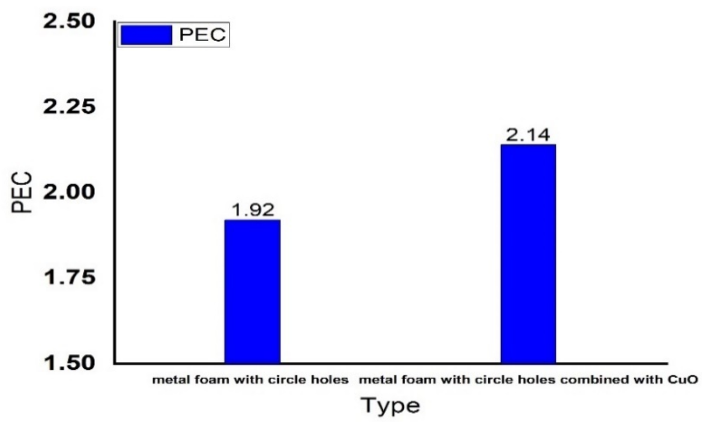

Figure 11. Effect of Metal Foam Combined with Nanofluid on PEC.

8.6. Comparison of temperature contours

In this section, a comparison was made between contours of temperature fordifferent models of heat exchangers.This comparison is made at the same properties of the copper foam (40PPI and Ɛ=0.9) with a flow rate (2 lpm). A comparison of contour shapes are shown in figure (12) at mid (yz) plane. The temperature is depicted

through a color gradient, with red representing the highest temperature (approximately 75°C) and blue indicating the lowest temperature (around 30°C).
(a): Smooth Pipe: The temperature distribution is moderately gradual, with a visible decrease in temperature from the inlet to the outlet. The heat transfer is limited due to the nonappearance of any turbulence in the flow.
(b): Full metal foam baffles: The outline significantly improves heat transfer by making additional turbulence in the flow. This results in a more uniform temperature distribution
(c–e): Metal foam with triangle, square, and circle holes: The addition of holes more improves heat transfer. These holes progress the mixing of the fluid inside the pipe, thus growing convective heat transfer.
(I) Metal foam with circle holes combined with nanofluids: It likely delivers the greatest balance of heat transfer presentation, due to surface area development and industrialized fluid conductivity.

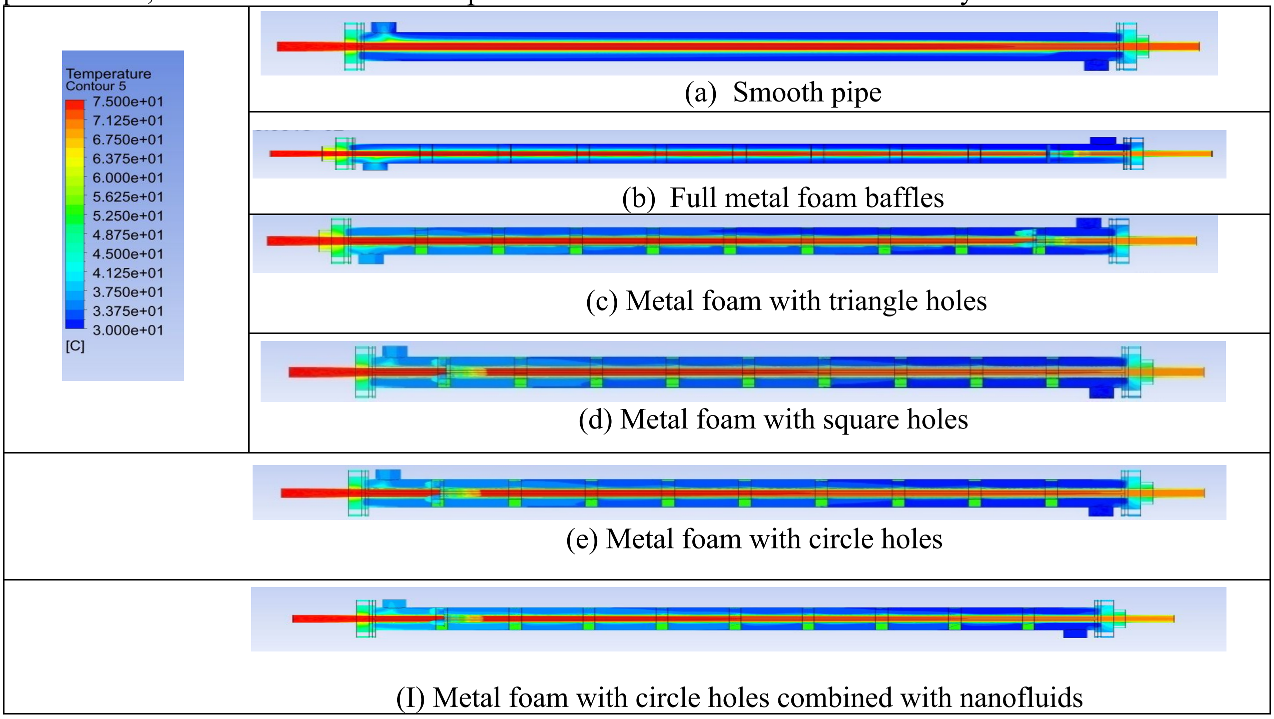

Figure 12. Temperature Contour.

9. Conclusion

After offering and discussing the attained results, the following conclusions are made:
1.The highest cold-water temperature is obtained when using full metal foam baffles with an increasing in the percentage 10% compared with smooth pipe.
2. Full metal foam baffles results in a substantial improvement Nu of up to 184 % while, CuO nonfluids enhances Nu by 50% relative to a smooth pipe.
3.The complete usage of full metal foam baffles leads to a 4-times increase in pressure drop relative to a smooth pipe.Consequently, the pressure drop is reduced by 1.6 times for metal foam with circle holes.
4. Metal foam with circle holes provide PEC of 1.92.
5. Metal foam with circle holes combined with 1% CuO provide the maximum value PEC of 2.14.

Declaration of Competing Interest

There are no conflicts of interest regarding the publication of this manuscript.

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Author Contributions

The main work was carried out by Mohammed B. Nati as the primary researcher, including the preparation, writing, and execution of the study. Dr. Abbas J. Jubear Al-Jassani participated as a principal researcher, contributed to the main idea of the work, and supervised the practical and experimental aspects. Dr. Mohammed

Ghalib Al-Azawy and Dr. Adel G. Nasser contributed to developing and supervising the theoretical aspects of the work. All the authors discussed the results and the final version of this paper.

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Nomenclature

Symbol	Title	Units
A_i	Internal surface area of inner smooth pipe	m^2
A_o	External surface area of inner smooth pipe	m^2
C_{pc}	Specific heat of cold fluid	J/kg. °C
C_{ph}	Specific heat of hot fluid	J/kg. °C
d_i	Inner diameter of inner pipe	m
d_i	Outer diameter of inner pipe	m
D_o	Outer diameter of outer pipe	m
D_i	Inner diameter of outer pipe	m
D_h	Hydraulic diameter	m
d_f	Fiber diameter	m
d_p	Pore diameter	m
f_{mf}	Friction factor with metal foam	-
f_s	Friction factor without metal foam (smooth)	-
g	Acceleration of gravity	m/s^2
k	Thermal conductivity	W/m. °C
K	Permeability of the porous medium	m^2
L	length	m
$\dot{m}_c$	Mass flow rate of cold fluid	kg/sec
$\dot{m}_h$	Mass flow rate of hot fluid	kg/sec
Nu	Nusselt number	-
Q	Heat transfer rate	W
Re	Reynolds number	-
T	Temperature	°C
$T_{c,i}$	Inlet temperature of cold fluid	°C
$T_{c,o}$	Outlet temperature of cold fluid	°C
$T_{h,i}$	Inlet temperature of hot fluid	°C
T_{ho}	Outlet temperature of hot fluid	°C
t	Thickness	mm

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