Numerical Study on the Effect of Geometrical Parameter on the Thermal-Hydraulic Performance of the Metal Foam Heat Exchanger

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Abstract. A numerical study was conducted to examine the heat transfer and pressure drop of the metal foam-wrapped array heat exchanger. Effects of geometrical parameters, including arrangement of metal foam heat exchanger (two arrangement: fully-filled of metal foam while other have a gap between metal foam and the chamber walls), porosity, and the pore density on the thermal hydraulic performance of copper metal foam heat exchanger are examined. Numerical modelling was done considering the thermal non-equilibrium energy model and the Darcy-Forchheimer flow model for metal foam heat exchangers in the Star-CCM+ program version 2019. Four types of metal foams at fixed dimensional thickness of 0.91 with variable properties were adopted, where the pores density was 5PPI & 20PPI and the porosity was 89\% and 95\%. The heat exchanger system is simulated over a range of Reynolds number, based on tube diameter, from 1333 to 6555 with tubes heated at a constant wall temperature. The layout and geometry of the foam heat exchangers were compared with bare tube heat exchanger. It was found that at low porosity 89\%, the configuration of metal foam heat exchanger that have pores density of 5PPI gives higher thermal performance (higher area goodness factor ratio) than 20 PPI. Also, decreasing porosity from 95\% to 89\% enhancement in the thermal performance was achieved. However, the metal foam heat exchanger that has a gap between metal foam and the chamber walls gave better thermal performance as compared with fully-filled foam heat exchanger.

Keywords. Copper metal foam, Area goodness factor, Foam layer thickness, High porosity, Staggered tube bundle, Configuration, Star CCM+.

1. Introduction
Metal foaming applications are distinguished by their complex geometry, which establishes good characteristics that cause a tortuous flow path with high turbulence Fourie and Du Plessis [1]. Many researchers, such as Bhattacharya and Mahajan [2], Boomsma et al. [3], and Lu et al. [4] indicated that employing a porous metal increase of heat transfer rate and pressure drop in the heat exchanger. However, the effect of pore density on the performance heat exchanger was investigated by Lu et al. [4]. It was found that an increase of pore density increases both pressure drop and the Nusselt number of the foam-filled tube, particularly when the conductivity ratio $k_f/k_s$ is very large. Kim et al. [5], studied the effect of changing the longitudinal distance between the staggered tubes on the thermal development region occurring in the sequent row. It was found that as the longitudinal distance increased, the heat transfer is decreased. Khan et al. [6], shows that an increase of longitudinal...
distance decreases significantly the heat transfer rate, while an increase of transverse distance decreases marginal the heat transfer rate. However, for compact tube-bundle, the heat transfer is higher than wide distance heat exchangers. In spite of the fact that the greater the density of the pores of the foam, the larger surface area will be, which leads to an increase in the resistance of fluid flow, i.e. a decrease in the amount of air and the forced thermal transfer, as demonstrated by researchers T’Joen et al. [7], Wang et al. [8] and Chen et al. [9]. Huisseune et al. [10] have confirmed that increasing the density of the pores increases the surface area of heat transfer and pressure drop. Furthermore, the tortuosity flow also leads to augment mixing of fluid flow inside the voids resulting in an increase in turbulence and an increase in the coefficient of heat transfer for fixed high porosity of open-cell metal foam. The phenomenon of thermal insulation described by Chen et al. [9] is that the low-velocity airflow rate decreases when the permeability is low, and therefore the heat transfer rates are greatly reduced. The permeability through the porous medium describes the smooth flow of liquid and the magnitude of the resistance flow of fluid through the solid-void matrix. In applications of a foam-covered heat exchanger, at high permeability and high Darcy number, the liquid flows faster so that heat transfer is greatly increased and thermal performance is improved through high thermal conductivity, as demonstrated by Rashidi et al. [11]. However, Xiao [12] experimentally demonstrated that the porous copper with high porosity and low pore density has high permeability. Also Rahmati et al. [13] noted that the low permeability means a low Darcy number, making the foam layer similar to the layer of the solid body so that it prevents flowing through it. An increasing in the surface area of foam leads to an increase amount of heat removal from the foam to the air flow, which was clarified and proven by Mehrdad [14]. Chumpia [15] confirmed that the increase of the transversal pitch of the pipes causes decrease both in the heat transfer and pressure drop, where the free space between the tube-bundle is extremely wider. Therefore, the fluid flow velocity is lower, and thus the heat transfer decreases because of the increase of the mass rate of air which flows away from the metal foam. As a result of both air velocity and pores density of the metallic foam, the mixing of airflow within the metallic foam increases, leading, as reported by Nawaz et al. [16], to an increase both in the rate of heat transfer and pressure drop. Fundamentally, Anuar et al [17] proved that the pressure drop is increased when the inlet velocity, pore density, and blockage ratio (gap between metal foam and the chamber walls) is increased. Metal foam bank tubes are a modern application and of great importance in heat exchangers in thermal transfer processes in industrial applications. It has good thermal characteristics, good flow resistance as well as lightweight, high durability, and long service life. Understanding the properties and requirements of airflow and heat transfer greatly influences the design of the heat exchanger apparatus of metal foam [7]. From the review of the literature it is clear that using metal foam on tube banks can result in benefits to heat exchanger. On the other hand, the tube that is covered with metal foam has enhancement in heat transfer but its application proved to be more challenging due to the large pressure drop on the air side. So, more itemized studies are required to investigate thermo-hydraulic performance of metal foam. Therefore in the present work, a numerical study of the turbulent forced convection were carried out in cross flow through staggered tube bundle that is wrapped by metal foam to study the impact of full filling of metal foam (there is no gap between the metal foam and the chamber walls) on thermo-hydraulic performance. Thus, two configurations of tube bundle wrapped with metal foam at fixed thickness were designed. Then, the transversal and longitudinal pitch distances were altered to get two designs: one of them is fully filled with metal foam, and the second design has a gap. For both configurations, two porosities have been taken 89% and 95% and for each porosity pores density of 5 PPI and 20 PPI were tested at the range of Reynolds number of (1311-6555).

2. Physical model and governing equations
Figure 1 illustrates two configurations of a staggered three-row tubes bundle both characterized by tube with diameter D, the transversal pitch ST and longitudinal pitch SL. The tubes bundles are wrapped with metal foam of a dimensional thickness of (0.5(DF-D)/D) is fixed 0.91. The first design has gap between the metal foam and the chamber walls, while the other design is fully filled with metal foam. The geometrical details of the two configurations are listed in table 1. Four samples are used in each design, which includes two porosities of 89% and 95%, where both tested for value of
pore density 5 PPI and 20 PPI, as listed in table 2. It is modeled as a homogenous and isotropic porous media.

Figure 1. The dimension of Physical Model, A: $1^{\text{st}}$ configuration, B: $2^{\text{nd}}$ configuration.

Table 1. Geometrical properties of applied heat exchangers.

| Item                                      | Heat exchangers Description | $1^{\text{st}}$ configuration | $2^{\text{nd}}$ configuration |
|-------------------------------------------|------------------------------|--------------------------------|--------------------------------|
| Tube diameter, D (mm)                     |                              | 20                             | 20                             |
| Dimensionless foam thickness, $0.5(D_F-D)/D$ |                              | 0.91                           | 0.91                           |
| Foam Thickness, t (mm)                    |                              | 18.2                           | 18.2                           |
| Transversal pitch, (mm)                   |                              | 69.2                           | 82.1                           |
| Longitudinal pitch, (mm)                  |                              | 44.5                           | 38.6                           |
| Tube numbers (N)                          |                              | 5                              | 5                              |
| Row number ($N_{row}$)                    |                              | 3                              | 3                              |
| H.X. width: L2 (mm)                       |                              | 200                            | 200                            |
| H.X. height: L3 (mm)                      |                              | 138.4                          | 138.4                          |
Table 2. Four samples of metal foam.

| Porosity 89% | Porosity 95% |
|--------------|--------------|
| Pore density | Pore density |
| Sample 1     | Sample 2     | Sample 3     | Sample 4     |
| 5PPI         | 20PPI        | 5PPI         | 20PPI        |

It is possible to represent the computational geometry for two configurations by a symmetrical shape in flow direction, [18], [19], as shown in Figure 2. Moreover, the boundary conditions are presented in Table 3 and Figure 2. However, the upstream distance from entry air to the center of the first-row is about five times of tube diameter D, while downstream distance is ten times of tube diameter D [20]. The tube length can be assumed long sufficient so that the end impacts can be neglected and as a consequence the airflow domain can be regarded two-dimensional [21]. The heat exchange system is characterized by the height $2S_{T1}$ [22].

A: 1st configuration.

B: 2nd configuration.

**Figure 2.** The computational field for two configurations, A: 1st configuration, B: 2nd configuration.

| Boundary conditions | input data |
|---------------------|------------|
| At $x = 0$          | $u = 0$    | Inlet velocity $u_{in} = 1, 2, 3, 4 & 5$ |
| $0 \leq y \leq S_f$ | $T = T_{in}$ | |
| At $x = L$          | $v = \frac{\partial u}{\partial x} = 0$ | |
| $0 \leq y \leq S_f$ | $\frac{\partial T_f}{\partial x} = 0$ | Inlet temperature $T_{in} = 25^\circ C$ |
| At the wall boundary: | $u = v = 0$ | Pipe temperature $T_{pipe} = 100^\circ C$ |
| $0 < x < L$  | $\frac{\partial T_f}{\partial y} = 0$ | |
| $y = S_f$  | $\frac{\partial T_f}{\partial y} = 0$ |
| $0 < x < L$  | $y = 0$ | Symmetric Plane |
| $y = 0$  | $u = v = 0$ |
| At the pipe boundary: | $T_f = T_c = T_{pipe}$ | |
The pipe wall temperature \((T_{pipe})\) is assumed to be uniform and constant, (wall temperature will be higher than the ambient temperature). This tube will be simulate tube in an air cooled condenser where the tube side fluid undergoes phase change. However, the flow of air is represented as the cross-flow at a uniform, fixed velocity with a constant temperature of 25°C, at the entrance of the duct. The airflow inside the rectangular duct can be incompressible, viscous, and fully-developed turbulent flow. Odabaee and Hooman [20], and [23] assumed the local thermal equilibrium (LTE) condition exit between the solid and fluid phases to investigate the effect of porous materials on heat transfer in their study. Alvandifar et al. [24] also simulated the forced convection through metal foam heat exchanger with the assumption of LTE. This assumption gives an inaccurate perception because fluid temperature differs from solid temperature, especially when using air and copper because the thermal conductivity ratio is high. Therefore, local thermal non-equilibrium (LTNE) is assumed in our work with two energy equations for the fluid and solid phases. Moreover, the Forchheimer–Brinkman Darcy model is adopted assuming turbulent boundary layer flow with no internal heat generation and neglecting viscous dissipation. Then the mass, momentum and energy conservation equations can be written for solid and fluid as follows [25]:

- **Mass conservation equation**

\[
\left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} \right) = 0
\]  

(1)

- **X- Momentum conservation**

\[
\frac{\rho_f}{\varepsilon^2} \left( u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = -\frac{\partial p}{\partial x} - \frac{\mu}{K} - \frac{P_f C_f}{\sqrt{K}} \left[ \frac{\partial}{\partial x} \left( \rho \frac{\partial u}{\partial x} + \rho \frac{\partial u}{\partial y} \right) \right]
\]  

(2)

- **Y- Momentum conservation**

\[
\frac{\rho_f}{\varepsilon^2} \left( u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right) = -\frac{\partial p}{\partial y} - \frac{\mu}{K} - \frac{P_f C_f}{\sqrt{K}} \left[ \frac{\partial}{\partial y} \left( \rho \frac{\partial v}{\partial x} + \rho \frac{\partial v}{\partial y} \right) \right]
\]  

(3)

- **Energy conservation**

For the fluid:

\[
(\rho C_p) \left( u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) = \frac{\partial}{\partial x} \left( k_x \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( k_y \frac{\partial T}{\partial y} \right) + h_s a_{sf} (T_s - T_f)
\]  

(4)

For the solid:

\[
0 = \frac{\partial}{\partial x} (k_x \frac{\partial T}{\partial x}) + \frac{\partial}{\partial y} (k_y \frac{\partial T}{\partial y}) + h_s a_{sf} (T_f - T_s)
\]  

(5)

Where; \(I\) in the momentum equations is equal to 1 for foam region, and equal to 0 for fluid region. While, \(\varepsilon\) is the porosity, \(\mu\) is the dynamic viscosity, and \(C_p\) is specific heat whereas \(k_x\) & \(k_y\) are the effective thermal conductivity of fluid and solid, models Boomsma and Poulikakos [26] was used to calculate them. The drag coefficient is defined as \(C_{fe}\) and \(K\) permeability, where their values are computed using the model proposed by Fouri and Du Plessis [1] as:

\[
K = \frac{\varepsilon^2 * d^2}{36(\chi - 1) * \chi}
\]  

(6)

\[
C_f = (3 - \chi)(\chi - 1) \frac{C_{D, sf} \chi^{1.5}}{24 * \varepsilon^2}
\]  

(7)

Where; \(d\) is characteristic dimension defines as:
\[
d = \frac{2 * d_p}{(3 - \chi)} \tag{8}
\]

And the pore diameter \(d_p\) can be estimated as functions of pore density \(\omega\), Calmidi [27]:
\[
d_p = \frac{0.0254}{\omega} \tag{9}
\]

In equations (6 to 8), the tortuosity \(\chi\) can be estimated as functions of pore density \(\omega\), Smit and Du Plessis [1]
\[
\chi = 2 + 2 \cos \left( \frac{4\pi}{3} + \frac{1}{3} \cos^{-1}(2\varepsilon - 1) \right) \tag{10}
\]

Also, Fourier and Du Plessis submitted mathematical model to calculate \(a_{sf}\) [1] as:
\[
a_{sf} = \frac{3}{d_c} (3 - \chi)(\chi - 1) \tag{11}
\]

Finally, \(h_{sf}\) is interfacial heat transfer coefficient of solid/fluid field of metal foam, Lu et al. submitted mathematical model to calculate [4] as:
\[
Nu_{sf} = \frac{h_{sf} d}{k_f} = \begin{cases} 
0.76 Re_d^{0.4} Pr^{0.37}, & (1 \leq Re_d \leq 40) \\
0.52 Re_d^{0.3} Pr^{0.37}, & (40 \leq Re_d \leq 10^3) \\
0.26 Re_d^{0.6} Pr^{0.37}, & (10^3 \leq Re_d \leq 2 \times 10^4)
\end{cases} \tag{12}
\]

Where; Reynolds number \(Re_d\) is based on the shape factor \(d\), which is a function of fiber diameter \(d_f\) as:
\[
d = \left(1 - e^{-\left(1 - \varepsilon\right)/0.04}\right) d_f \tag{13}
\]
\[
d_f = 1.18 d_p \sqrt[3]{\frac{(1 - \varepsilon)}{3\pi} \frac{1}{1 - e^{-\left(1 - \varepsilon\right)/0.04}}} \tag{14}
\]

In order to analyze the thermal--hydraulic characteristics of the enhanced tube, some parameters are required such as the heat transfer coefficient per unit length (h), Nusselt number (Nu), and friction factor (f) which defined as follows:
\[
h = \frac{Q}{\pi * D * (T_{pipe} - T_{fb})} \tag{15}
\]
\[
Nu = \frac{h * D}{k_f} \tag{16}
\]

Where \(k_f\) fluid thermal conductivities coefficient, \(T_{pipe}\) is the temperature of the pipe (constant wall temperature), \(T_{fb}\) is the bulk temperature of the inlet and outlet fluid, \(D\), Pipe diameter. The factor of friction of a staggered heat exchanger can be described in terms of Reynolds number as follows:
\[
f = \frac{2 * \Delta P}{N_{row} * \rho * U_{in}^2} \tag{17}
\]

Where \(\Delta P\) is the difference of pressure upstream and downstream, and \(N_{row}\), Number of pipe rows, while the Reynolds number base on pipe diameter is:
\[
Re_d = \frac{\rho * U_{in} * D}{\mu} \tag{18}
\]
3. Numerical solution

In this study, the numerical analysis was performed using commercial Computational Fluid Dynamics (CFD) software (Simcenter STAR-CCM+, Version 2019.1). The governing equations were solved by the control volume, finite-volume method in staggered grid arrangements. The second upwind scheme is used to discretize the convective terms in momentum and energy equations and the SIMPLE algorithm of Patankar and Spalding [28] is used to ensure the coupling between velocity and pressure. The coupled set of governing equations is solved using pressure correction for divergent velocity field. The pressure correction equation is solved using the successive-under relaxation technique with the under-relaxation factor chosen with range \((0.8)\). With the corrected pressure, velocity and temperature fields are updated. Moreover, relative variations between two iterations are set to be smaller than \(10^{-5}\). The iterations continue until the relative change between iterations is equal to or less than \(10^{-5}\), at which point the simulation is considered to have converged. Additionally, the thermo-physical properties of air under the temperature of 298 K are selected. The grid system is applied in representing the computational model [29]. A polyhedral mesh was created with irregular size of 1 mm. At the wall boundary, the mesh is refined with prism mesh layers, and also at the interface boundaries for foam/fluid in two fields. The schematic of mesh is presented in figure 3.

![Figure 3. The generated mesh of the problem.](image)

For each configuration and porosity of foam, grid independence calculations were performed within four sets of grid sizes at \(Re_D=5666\) and pore density 5PPI, as shown in table 4. It is clear from the table that the difference in the value of Nusselt number between set grid No.3 and No.2 was within the range of 0.9\%-2.1\%, while the difference between set grid No.4 and No.3 was within range of 0.22\%. From this, it is found that the value of Nusselt number does not vary between grid No.3 and No.4, therefore set grid No.3 was adopted.

| Table 4. Nusselt number shifting depending on the grid size at different porosity and configuration. |
|---|---|---|---|---|
| First Configuration | Grid No. | 1 | 2 | 3 | 4 |
| porosity | No. of Cells | 67593 | 84525 | 105238 | 112642 |
| 89% | Nu | 367.141 | 332.739 | 339.494 | 340.231 |
| porosity | No. of Cells | 63416 | 82249 | 102159 | 110753 |
| 95% | Nu | 367.141 | 189.425 | 193.252 | 193.763 |
| Second Configuration | Grid No. | 1 | 2 | 3 | 4 |
| porosity | No. of Cells | 64375 | 80540 | 100238 | 105901 |
| 89% | Nu | 358.423 | 369.627 | 377.020 | 378.228 |
| porosity | No. of Cells | 61796 | 78986 | 98277 | 104125 |
| 95% | Nu | 195.944 | 202.638 | 206.875 | 207.315 |
To examine the validity of the numerical method, the predicted total heat transfer for a staggered three-row tube bundle coated with open-cell aluminum foam at range of air velocity are compared with experimental work of Chumpia [15] which have the same shape, arrangement, and foam specifications as shown in figure 4. This figure shows that the model predictions are in reasonable agreement with experimental data of Chumpia [15].

![Figure 4: Comparison of the total heat transfer computed from simulation numerically with the corresponding data measured experimentally by Chumpia. [15]](image)

4. Results

4.1. Pressure drop

In this section, the effects of design parameters are studied such as: the gap between the metal foam and the chamber walls (either fully filling of metal foam or not), porosity and pore density, on the pressure drop and heat transfer for a staggered tube bundle with covering tube a layer of metal foam at fixed dimensional thickness 0.91. Figure 5 displays the pressure drop plotted against air velocity for two configurations and bare tube. As seen in the figure, generally speaking for metal foam heat exchanger (i.e. two design of heat exchanger) and bare tube heat exchanger, the pressure drop reveals a similar behaviour with respect to the air velocity, that the pressure drop is increased with increasing of the air velocity. The pressure drop in the metal foam heat exchangers is higher as compared to the pressure drop in the bare tube heat exchanger. Considering the two configurations with the same porosity as seen in figure 5, for a given air velocity, the pressure drop increases as the number of pores per inch increases passing from 5 to 20 PPI. The pores density of 20PPI and 5 PPI shows the highest and the lowest pressure drop, respectively. Comparing the geometric data at fixed porosity for two configurations of the various foams, it is shown that the fully-foamed design (second configuration), results in significantly higher pressure drop compared to the design with a gap between metal foam and the chamber walls (first configuration). In fact, the total pressure drop at fixed porosity (89%) for the second configuration is over (1.4 and 2.2) times that of the first configuration for pore density 5 PPI and 20 PPI, respectively. However, the impact of the porosity on the pressure drop for two designs of the metal foam heat exchanger can be seen in figure 6. It shows that the first configuration of foam heat exchanger at the highest porosity (0.95) with pores density of (5 and 20) PPI can cause up to (4 and 16) times, respectively, higher resistance compared to the bare tube heat exchanger. Meanwhile, the foam with lower porosity (0.89) with pores density (5 and 20) PPI can increase the pressure drop to a maximum of (6 to 41) times, respectively. The results indicate that foams with small pores diameter and low porosity will result in higher pressure drop; moreover, the heat exchanger that is fully filled with metal foam gives higher pressure drop.
Figure 5. Pressure drop variation with Reynolds number for difference pore density and two configuration: Porosity: A: 89%, B: 95%.

Figure 6. Pressure drop variation with Reynolds number for difference pore density and porosity: two configuration: A = 1st configuration, B = 2nd configuration.

To analyse the performance difference of two configurations of metal foam heat exchanger, figures 7 and 8 represent the velocity contours for low Reynolds number (1311) and high Reynolds number (6555) of the flow over tube bundle for low and high porosity (89% 95%), respectively, at pores density 5 and 20 PPI for each porosity. One can observe that the velocity contours for the two configurations of foam heat exchanger follow the same trend as the Reynolds number, pores density and porosity changes. It is clear that the pressure drop is the pressure difference between the inlet and outlet of the computational domain. So; it was due to the existence of a solid obstacle as well as a foam layer. The solid obstacles results in a hydraulic resistance by blocking apart of flow passage leading to local increase in the velocity of the fluid passing around the body as shown in figure 7 and 8. As the fluid passes through the domain, a part of fluid that passes through the foam layer results in a form drag and viscous drag. Hence, there is reduction of the local velocity at the rear of cylinders due to the blockage effect and the stagnation point at this region. The blockage effect is weakened as porosity increased. Hence, the permeability decreases at low porosity, so the flow passage in foams become smaller, which results in smaller fluid flow into the foams and the velocity through the foam becomes lower. Moreover, for fixed porosity, when pores density is high, the distributed velocity contours into the metal foam is low because of the flow resistance to penetrate through the foam. Furthermore, microstructure causes the flow disturbance which would become more severe with the increase of the Reynolds number. So, the recirculation zone behind tubes grows by increasing the Reynolds number especially at low both porosity and the pores density. However, the central region between upper and lower rows presents the highest velocity at high Reynolds number and low Reynolds number within the whole computational domain, which is nearly double times the velocity of the inlet free stream. This should be taken into account in heat exchanger designs.
Figure 7. Velocity contour at low and high Reynolds numbers for Porosity=89%: A=1st, 5PPI, B=1st, 20PPI, C=2nd, 5PPI, D=2nd, 20PPI.

Figure 8. Velocity contour at low and high Reynolds numbers for Porosity=95%: A=1st, 5PPI, B=1st, 20PPI, C=2nd, 5PPI, D=2nd, 20PPI.

4.2. Heat transfer

Figure 9 & 10 describe the temperature contour for low porosity 89% and high porosity 95% respectively, on the metal foam tubes of the two designs of heat exchanger. One is fully filled with metal foam (first configuration) while the other (second configuration) has a gap between metal foam and the chamber walls at fixed dimensionless thickness of metal 0.91 for two pores density 5 and 20 PPI at low and high Reynolds number. The mechanism of the heat removal from the tubes can be by both conduction through the foam ligaments and convection through the foams. According to the figure, generally for low and high Reynolds number, temperature distribution changes by increasing the Reynolds number. As the Reynolds number increases, the thermal boundary layer thickness on the tubes decreases and the convective heat transfer rate from tube rows increases. However, thickness decrease of the thermal boundary layer of second configuration (fully filling of metal foam) was highly clearer than the first configuration. At low Reynolds numbers, the fluid core warms up due to higher heat conduction or diffusion with respect to convection. Moreover, at fixed porosity and a
given Reynolds number, when the diameter of the cell borders is smaller (increasing pores density), thermal conductivity is increased. This could be due to the internal surface area depending inversely on the diameter and the transversal section of the heat transference increasing when the relative density increases. This results in a thicker thermal boundary layer. For low porosity the contour, temperature is higher than that for high porosity.

Figure 9. The temperature contour for low and high Reynolds number for Porosity=89%: A=1st, 5PPI, B=1st, 20PPI, C=2nd, 5PPI, D=2nd, 20PPI.

Figure 10. The temperature contour for low and high Reynolds number for Porosity=95%: A=1st, 5PPI, B=1st, 20PPI, C=2nd, 5PPI, D=2nd, 20PPI.

Figure 11 displays the effect of pore density and geometric design (first and second configuration) upon the Nusselt number within range of Reynolds numbers. The behavior trend of Nusselt number will be similar with respect to Reynolds number, which is increased as the Reynolds number is increased. According to this figure, at fixed porosity of pores density has a great influence on the heat transfer coefficient; a given Reynolds number heat transfer coefficient increases with increases in PPI.
This is because, with the increase in PPI, specific surface area increases. Hence more surface area will be available for heat transfer and also with increase in PPI, pore density increases. Thus fluid has to pass through more tortuous path which makes better mixing and enhances the heat transfer rate. However, the impact of pores density at low porosity 89% is very clear as compared to high porosity 95%. Though, for porosity 89%, the Nu values of metal foam tube banks at pores density (5PPI and 20 PPI) can be higher than that of the bare tube bank about (7.6-8.8) times at 1st configuration, and about (7.6-9) times at 2nd configuration. On the other hand, at porosity of 95%, the Nusselt values of metal foam at pores density of 5PPI and 20 PPI become higher than that of the bare tube bank about (3.65-3.85) times at 1st configuration, and about (3.66-4 times) at 2nd configuration. From this, one can get that the improvement percentage in the Nusselt number by using fully filling metal foam tubes bank (second configuration) as compared to the first configuration, is about 5% and 4% for the porosities 89% and 95%, respectively at pores density of 20PPI. The results indicate that using fully filling foam heat exchanger gives slight improvement in heat transfer rate as compared with first configuration. However, heat transfer rate at porosity of 89% is about (2.2) times higher than porosity of 95%, as shown in figure 12. This is due to higher heat transfer rate by conduction (through the solid) when the porosity is lower.

![Figure 11. Nusselt number variation with Reynolds number for difference pore density and two configuration: Porosity: A 89%, B=95%](image)

![Figure 12. Nusselt number variation with Reynolds number for difference pore density and Porosity: A= 1st configuration, B=2nd configuration.](image)

4.3. Thermal performance

To evaluate the performance of the copper foaming coated heat exchangers relative to the conventional bare tube design, from both heat transfer and pumping power points of view, the area goodness factor A.G.F. was used for metal foam heat exchanger and normalized by that of a bare-tube heat exchanger, (the area goodness factor ratio (A.G.F.R.) for all samples here), as follows:
\[ A.G.F. = \frac{j}{f^{1/3}} = \frac{Nu}{Re_D^* (Pr^* f)^{1/3}} \]  
\[ \text{(19)} \]

Where \( j \) is Colburn factor:

\[ j = \frac{Nu}{Re_D^* Pr^{1/3}} \]  
\[ \text{(20)} \]

\[ A.G.F.R. = \frac{\text{Area Goodness factor of foam tube}}{\text{Area Goodness factor of bare tube}} = \frac{(j / f^{1/3})_{\text{foam}}}{(j_o / f_o^{1/3})_{\text{bare}}} \]  
\[ \text{(21)} \]

The ratio between the heat transfer rate and the mechanical power is shown in figure 13, to determine which key parameter such as: configuration, porosity and pores density maximizes the thermal power yet at the same time does not require a high mechanical power. It is clear from the figure, in general for fixed porosity, the area goodness factor ratio of the foam heat exchanger that has a gap between the metal foam and the chamber wall (first configuration) is higher than fully filled metal foam heat exchanger (second configuration). This means that for the second configuration, there is a high heat transfer rate but there is also a mechanical power required to pump the working fluid, which was higher than the first configuration, therefore it was more convenient to use first configuration. Also, at low porosity of 89%, the configuration of metal foam heat exchanger that has pores density of 5PPI gives higher area goodness factor ratio (i.e. low pumping mechanical power) than 20 PPI. As the porosity of the metal foam heat exchanger is increased to 95% the area goodness factor ratio is decreased. This cause is from a reduction heat transfer by the conduction due to high porosity in spite of the pumping power required is lower.

![Figure 13. Area Goodness factor ratio variation with Reynolds number for difference porosity, pore density and two configuration.](image)

5. Conclusion

Numerical investigation on the thermal performance of 20 mm outer diameter staggered tube bundles covered with fixed dimensional thickness 0.91 of metallic foam are performed for two arrangements of heat exchanger. One is with a gap between the metal foam and the chamber walls while the second is fully filled with metal foam. The effects of porosity on the thermal performance are different from variations in the entire metallic foam. At low porosity of 89%, the configuration of metal foam heat exchanger that has pores density of 5PPI gives higher thermal performance (higher area goodness factor ratio) than 20 PPI. This means that the pumping power required for the pores density 5PPI is
lower. Though, increasing the porosity of the metal foam heat exchanger to 95% results in reduction in the thermal performance. However, the thermal performance of the foam heat exchanger that has a gap between the metal foam and the chamber wall (first configuration) is higher than the fully filled metal foam heat exchanger (second configuration). This means that for the second configuration, there is a high heat transfer rate but there is also a mechanical power required to pump the working fluid, which was higher than the first configuration, causing reduction in thermal performance; therefore, the first configuration is more convenient to use.

### Nomenclature

| Symbol | Description |
|--------|-------------|
| a_s | The solid-fluid specific surface area, m² |
| A.G.F | Area Goodness Factor |
| A_f | Frontal area of the core, A_f=b/a |
| b | Height of the rectangular channel, m |
| c_p | Specific heat of fluid-phase, J/kg·K |
| C_f | Inertia drag coefficient |
| d | Characteristic dimension, m |
| d_p | Pore diameter, m |
| D_f | Diameter of the Foam, m |
| g | Gravity acceleration, m/sec² |
| h_s | Solid-to-fluid interfacial specific heat transfer coefficient, W/m²·K⁻¹ |
| J | The Colbrun Factor |
| K | Permeability of the porous medium, m² |
| m | Mass flow rate, kg/sec |
| Nu | Nusselt number |
| PPI | Pore per inch |
| Re | Reynolds number, Re= u_in a/ν_f |
| S_t | Transversal distance |
| t | Time, sec |
| u_v | Dimensional velocity components, m/s |
| χ | Foams tortuosity |
| ν_f | Kinematic viscosity, m²/sec |
| f | Fluid phase of metal foam |
| F | Foam material |
| P | Pipe/pore |
| se | Solid effective |

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