The effect of PPI on thermal parameters in compact heat exchangers with aluminum foam

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Abstract. A numerical analysis on a compact heat exchanger in aluminum foam is accomplished. The governing equations in two-dimensional steady state regime are written in local thermal non-equilibrium (LTNE). The physical domain under investigation is made up of a plate in aluminum foam with inside a single array of five circular tubes. The presence of the open-celled metal foam is modelled as a porous media by means of the Darcy-Brinkman-Forchheimer law. The metal foam is characterized by a porosity of 0.93 and different pores per inch (PPI), equal to 5, 10, 20 and 40 are analyzed. The LTNE assumption is considered to simulate the heat transfer between the fluid phase and the solid matrix of the foam. The compact heat exchanger at different air flow rates is studied with an assigned surface tube temperature. The results are shown in terms of heat and mechanical power ratio. Global parameters such as effectiveness and NTU, typical in compact heat exchangers, are showed. Furthermore, local air temperature and velocity profiles in the smaller cross section, between two consecutive tubes, as a function of Reynolds number are presented.

1. Introduction

Metal foams represent a relatively new class of materials applied in fuel cells [1], compact heat sinks [2] and solar thermal plants [3]. Metal foam heat exchangers are efficient, compact and lightweight for the open porosity, low relative density and high thermal conductivity of the cell edges and the large accessible surface area per unit volume. Aluminum foams combine low weight with good rigidity, strength, damping of vibrations and noise, shock resistance and low thermal conductivity [4]. The possible applications are dependent on many parameters such as the type of metal, the porosity (open or closed), the process by which the foam is obtained (temperature, pressure, residence time, melt flow etc.), density and overall costs. The material has a high ratio between surface area and pressure drop, and with uniform lower density. The pressure drop is lower than in ceramic structures when considering the unit of volume [5]. Hui Wang and Liejin Guo [6] carried out an experimental study on heat transfer and pressure drop of three stainless steel foam filled tubes under convection boundary condition. The stainless steel foam filled tubes are of different pore densities (10, 30 and 70 PPI) but have the same porosity of 0.93. It was demonstrated that the inertial drag is the dominant part of the pressure drop at higher velocity. A new correlation for the pressure drop through metal foams under high velocity was showed. The effect of the boundary condition on the heat transfer performance was investigated by comparing Nusselt number obtained in the study with that obtained under constant heat flux boundary condition. It was found that the Nusselt number obtained under convective boundary condition is much lower than that obtained
under constant heat flux boundary condition. Bayomy and Saghir [7] accomplished a study on heat transfer characteristics of aluminum metal foam subjected to a water flow both experimentally and numerically. The aluminum foam was subjected to a pulsating water flow. The pressure drop was measured across the aluminum foam heat sink and the effects of the dimensionless flow frequency and the flow amplitude on the heat transfer characteristics of the pulsating flow through the aluminum foam were analyzed. An empirical correlation of the average Nusselt number as a function of the dimensionless flow frequency and flow amplitude was developed. A comparison between the heat transfer characteristics of the steady and pulsating water flows was also conducted. Results also revealed an increase of 14% in the average Nusselt number for pulsating flow when compared to the steady flow. A numerical analysis on the heat transfer inside a open-cell aluminum metal foam was conducted by Ranut et al. [8]. For precisely recovering the microstructure of the foams, a 3D X-ray computed microtomography (μ-CT) is adopted. In this work, the results of μ-CT-based CFD simulations performed on different open-cell aluminum foams samples, for laminar flow regime, will be analyzed. The results demonstrate that open-cell aluminum foams are effective means for enhancing heat transfer. A convective heat transfer in metal foams was studied by Xu et al. [9] imposing the local thermal equilibrium (LTE) model and the local thermal non equilibrium (LTNE) model numerically. The results demonstrated that the heat transfer with LTE model is higher than that of LTNE model, and the LTNE influence becomes more obvious in these case: lower porosity, larger difference between the thermal conductivity of the fluid and solid phases, lower pore density and lower Reynolds values. Odabaee et al. [10] accomplished a numerical study in order to investigate the heat transfer from a metal foam wrapped solid cylinder in cross-flow. The system was compared with a finned-tube heat exchanger to discover much higher heat transfer rate with acceptable excess of pressure drop. Huisseune et al. [11] analyzed the performance of a metal foam heat exchanger correlated with the bare tube bundle and an existing louvered fin heat exchanger. An experimental investigation on forced convection in a compact heat exchanger made up with an aluminum foam plate was accomplished by Cicala et al. [12]. Colburn factor and Nusselt number were evaluated for different air mass flow rates and hot water inlet temperatures. A numerical investigation on a tubular heat exchanger in aluminum foam was carried out by Buonomo et al. [13] in Darcy–Forchheimer flow model and the thermal non-equilibrium energy assumption for an assigned porosity and pore density. The results were oriented to show the local behaviour of the system. The evaluation thermal and fluid-dynamic behaviours in a compact heat exchanger in aluminum foam was provided by Buonomo et al. [14] assuming the local thermal non-equilibrium hypothesis. The scope of the analysis was to identify the heat exchanger dimensions to realize a compromise between the heat transfer improvement and the increase in pressure drop. The present numerical study is an extension of the investigation [13] because the heat exchanger is studied considering different foams characterized by various pore density. The heat transfer rate, the viscous power dissipation, their ratio and the effectiveness of the system are shown. Furthermore, the air velocity and temperature profiles inside the metal foam are presented for different PPI values.

2. Governing equations and Physical model
The 2D sketch of the tubular heat exchanger, under investigation, is showed in Figure 1. The physical model is a forced convection with a steady laminar flow in a metal foam heat exchanger. The heat exchange system is made up of an array of five tubes and they are enclosed inside the aluminum foam. The center-to-center distance between each tube, indicated as $2H$, is 0.038 m and the diameter $d$ of the tubes is 0.012 m. The heat exchanger is characterized by the height $H_{tot}$ of 0.19 m and the length $L_{mf}$ of 0.038 m. The local volume averaging process is used to obtain the microscopic flow equations as indicated in [15] in order to reproduce the thermal and fluiddynamic behaviors inside the domain. The variables of the equations are written evaluating the average of the local variables on an appropriate volume, called Representative Elementary Volume (REV), using the local volume averaging technique [16]. The aluminum foam is assumed homogeneous and isotropic and the thermophysical properties of the fluid and solid phases are considered constant. Furthermore, the viscous dissipation and buoyancy force are neglected, and the thermal contact resistances between the tube surface and foam is overlooked. The Darcy-Brinkman-Forchheimer model and LTNE condition are assumed to describe the behavior of the metal foam. Using the hypothesis above indicated, the governing equations are the following:
- Continuity equation

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0
\]  

(1)

- x- momentum equation

\[
\rho_f \left( u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = -\frac{\partial p}{\partial x} + \frac{\mu_f}{\phi} \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) - \frac{\mu_f}{K} u - \frac{C_F}{K^{1/2}} \rho_f \sqrt{u^2 + v^2} u
\]  

(2)

- y-momentum equation

\[
\rho_f \left( u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right) = -\frac{\partial p}{\partial y} + \frac{\mu_f}{\phi} \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) - \frac{\mu_f}{K} v - \frac{C_F}{K^{1/2}} \rho_f \sqrt{u^2 + v^2} v
\]  

(3)

where \( \phi \) is the porosity, \( \rho_f \) and \( \mu_f \) are fluid density and viscosity, \( u \) and \( v \) are correspondingly the velocity components in Cartesian coordinates, \( K \) and \( C_F \) are respectively the permeability and the inertial coefficient of metal foam.

- Energy equation for the fluid phase

\[
(\rho c_p) \frac{\partial T_f}{\partial x} + \frac{v}{\partial y} = \phi k_f \left( \frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2} \right) + h_{sf} \alpha_{sf} (T_f - T_s)
\]  

(4)

- Energy equation for the solid phase

\[
(1 - \phi) k_s \left( \frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} \right) - h_{sf} \alpha_{sf} (T_f - T_s) = 0
\]  

(5)

where \( c_p \) is the specific heat, \( k_f \) and \( k_s \) are the fluid phase and solid matrix thermal conductivity, \( T_f \) and \( T_s \) are the temperature of fluid and solid phase of metal foam, respectively. In the last term in both energy equations, \( \alpha_{sf} \) and \( h_{sf} \) are respectively the specific surface area density and the interfacial heat transfer coefficient between the fluid and solid matrix, due to the LTNE hypothesis.

The relation of Calmidi are used to evaluate \( K \) and \( C_F \) [17]:

\[
K = 0.00073 (1 - \phi)^{0.0224} \left( \frac{d_f}{d_p} \right)^{-1.11} d_p^2
\]  

(6)
and

\[ C_F = 0.00212 (1 - \phi)^{-0.132} \left( \frac{d_f}{d_p} \right)^{-1.63} \]  

(7)

where \( d_f \) and \( d_p \) are respectively the ligament and pore diameter of the metal foam. These parameters are in relation with the characteristics of the aluminum foam by means of the following relations [18]:

\[ \frac{d_f}{d_p} = 1.18 \frac{1 - \phi}{3\pi} \left( \frac{1}{1 - e^{-(1 - \phi)/0.04}} \right) \]  

(8)

and

\[ d_p = \frac{0.0224}{\omega} \]  

(9)

\( \omega \) is the pores density of the metal foam, that is the number of pores across a linear inch. In this analysis it is variable and assumed values equal to 5, 10, 20, 40 numbers of pores per inch (PPI). The following correlations are adopted to evaluate \( \alpha_{sf} \) and \( h_{sf} \) [19]:

\[ \alpha_{sf} = \frac{3\pi d_f}{(0.59d_p)^2} \left( 1 - e^{-(1 - \phi)/0.04} \right) \]  

(10)

and

\[ h_{sf} = \begin{cases} 0.75Re_{d_f}^{0.4}Pr_{air}^{0.37} \left( \frac{k_f}{d_f} \right), & 1 \leq Re_{d_f} \leq 40 \\ 0.51Re_{d_f}^{0.5}Pr_{air}^{0.37} \left( \frac{k_f}{d_f} \right), & 40 \leq Re_{d_f} \leq 1000 \\ 0.26Re_{d_f}^{0.6}Pr_{air}^{0.37} \left( \frac{k_f}{d_f} \right), & 1000 \leq Re_{d_f} \leq 2 \times 10^5 \end{cases} \]  

(11)

where \( Re_{d_f} \) is the local Reynolds number referred to ligament diameter:

\[ Re_{d_f} = \frac{\rho_f u_0 d_f}{\mu_f} \]  

(12)

and \( Pr_{air} \) is the air Prandtl number that is evaluated as:

\[ Pr_{air} = \frac{\mu_f c_p}{k_f} \]  

(13)

The parameters of the metal foams, for different PPI, are listed in Table 1.

| PPI | \( \phi \) | \( d_p \) (m) | \( d_f \) (m) | \( K \) (m²) | \( C_F \) |
|-----|----------|-------------|-------------|-------------|---------|
| 5   | 0.93     | 4.48 e⁻³    | 5.51 e⁻⁴   | 2.72 e⁻²   | 0.092   |
| 10  | 0.93     | 2.24 e⁻³    | 2.79 e⁻⁴   | 6.79 e⁻⁸   | 0.092   |
| 20  | 0.93     | 1.12 e⁻³    | 1.37 e⁻⁴   | 1.69 e⁻⁸   | 0.092   |
| 40  | 0.93     | 5.59 e⁻⁴    | 6.89 e⁻⁵   | 4.25 e⁻⁹   | 0.092   |
3. Numerical model

The finite volume method is adopted to resolve the governing equations. Fluent is used to conduct the numerical simulations. Convergence criteria are imposed for the continuity and the velocity components equal to $10^{-5}$ while for the energy equal to $10^{-8}$. As the computational domain is utilized half of a single tube, as represented in Figure 2. The half tube is totally surrounded by the aluminum foam. The height of system is equal to $H$, the length $L_{mf}$ is the same of the physical domain and the radius of the tube is equal to $d/2$. In this configuration, the heat exchanger is placed in a parallel plates channel with a rectangular cross section of 0.019 m with $L_1$ and $L_2$ equal to 0.20 m and 0.80 m, respectively.

The grid is made up of rectangular cells in the air channel and around the tube and triangular cells in the region of metal. Three different types of grids were investigated to find an independent solution from the mesh. They are constituted by 5786 cells, 23272 cells and 92702 cells, respectively. In corresponding of an inlet air velocity equal 0.12 m/s and of a metal foam characterized by 20 PPI, the evaluation of average $\dot{h}$ on the tube surface, as showed in Table 2, points out that the grid with 5786 cells had 0.08% error than the mesh with 92702 cells. The grid adopted for the computational was the one with 5786 elements because with it a compromise between solution accuracy and convergence time was obtained.

In the analysis, the Reynolds number $Re_d$, evaluated as:

$$Re_d = \frac{\rho u_0 d}{\mu}$$

range from 100 to 1000. The inlet air velocities $u_0$ is calculated from the definition of $Re_d$. The boundary conditions of the problem are the following: the upper and bottom edges are symmetric, the tube wall is an isothermal edge at $T_{tube}=323.16$ K, the inlet air velocity $u_0$ assumes the values above indicated, the temperature at the inlet, indicated as $T_{in,air}$, is 288.16 K and to the exit is imposed the outflow condition.

4. Results

The thermal and fluid-dynamic studies are carried out for different inlet air velocities and for a fixed temperature on the external surface of tube of 323.16 K. The state regime is laminar and the LTNE model is considered to write the energy equations. In the Figure 3.a and 3.b the heat transfer rate $\dot{Q}$ and the viscous power dissipation $\dot{W}$, required to circulate the air inside the heat exchanger are reported for different PPI values:
Table 2. Grid Independence and Numerical Results

| Cells Number | $\bar{h}_{\text{tot}}$ (W/m$^2$K) | % error |
|--------------|-------------------|---------|
| 5786         | 146.88            | 0.08%   |
| 23272        | 146.79            | 0.02%   |
| 92702        | 146.76            | ------- |

Figure 3. (a) Heat Transfer Rate $\dot{Q}$ and (b) Viscous Power Dissipation $\dot{W}$ for different PPI values.

It can be seen that the heat transfer rate increases with increasing pores density, but this improvement in terms of heat exchange is more visible for higher values of air mass flow rate. In addition, it can be observed as also the mechanical power increases with increasing of PPI number and of the mass flow rate. The ratio between the heat transfer rate and the mechanical power is showed, in the Figure 4, in order to know which PPI number maximizes the thermal power but at the same time does not require high mechanical power. The ratio between thermal and mechanical power decreases both with the increase of PPI and $Re_d$. This means that for high PPI values there is a high heat transfer rate but there is also a higher mechanical power required to pump the working fluid and therefore it is more convenient to use foams with a smaller number of PPI. In the Figure 5.a, the temperature profile along the median cross section into metal foam is displayed for $Re_d$ equal to 1000. One can observe two temperature profiles, one referred to the fluid phase and another associated to the solid matrix, due to the LTNE model assumption, for all PPI values.

Figure 4. Ratio $\frac{\dot{Q}}{\dot{W}}$ for different PPI values.
Figure 5. Temperature profile (a) and velocity profile (b) along the median section for different PPI.

The difference of temperature between fluid and solid phases decreases with increasing pores density and, for this motivation, in corresponding to higher values of PPI it is possible to assume the LTE model. Furthermore, in the Figure 5.b, the velocity profiles along the median cross section in the foam are reported for $Re_{f}=1000$, showed a partial overlap of the profiles for the different PPI values.

The effectiveness $\varepsilon$ and $NTU$, typical global parameters in a compact heat exchanger, are showed in the Figure 6:

It can be seen that the effectiveness of the system increases with increasing of PPI values and this occurs for all Reynolds numbers. These parameters are evaluated as [20]:

$$\varepsilon = \frac{\dot{Q}}{C_{\text{min}} \cdot (T_{\text{tube}} - T_{0})}$$  \hspace{1cm} (15)

$$NTU = -\ln(1 - \varepsilon)$$  \hspace{1cm} (16)

with $C_{\text{min}}$ the air-side heat capacity $\dot{m}c_p$ and $T_{\text{tube}}$ and $T_{0}$ the free stream tube temperature and the air inlet temperature applied in the 2D simulations, respectively.

Figure 6. The effectiveness for different PPI.
5. Conclusions
A numerical investigation on a heat exchanger with aluminum foams characterized by 5, 10, 20, 40 PPI has been accomplished in order to evaluate the thermal and fluiddynamic behaviour of the system. The results in terms of heat transfer rate show that the foam with higher pores density value improves the heat transfer than aluminum foam with a lower PPI. In addition, the results in terms of ratio between heat and mechanical power show that the use of metal foam with a lower PPI values is more efficient than one with a higher PPI value. Furthermore, the different temperature profiles in the median section into the metal foam for fluid and solid phases are showed, demonstrating that the difference temperature between two phases decreases with increasing pores density. As a final observation, the efficiency has been evaluated and it increases with increasing PPI values.

6. Nomenclature

| Symbol | Description                                      | Unit       |
|--------|--------------------------------------------------|------------|
| A      | Cross section                                    | m²         |
| C_f    | Drag factor coefficient                          |            |
| c_p    | Specific heat at constant pressure               | J/kg K     |
| d      | Tube diameter                                    | m          |
| d_f    | Fiber diameter                                   | m          |
| d_p    | Pore diameter                                    | m          |
| f      | Friction factor                                  |            |
| h      | Heat transfer coefficient                        | W/m² K     |
| h_sf   | Interfacial heat transfer coefficient            | W/m² K     |
| H      | Half pitch                                       | m          |
| H_ext  | Heat exchanger height                            | m          |
| j      | Colburn factor                                   |            |
| k      | Thermal conductivity                            | W/m K      |
| K      | Porous permeability                              | m²         |
| L      | Thickness of porous media                        | m          |
| m      | Mass flow rate                                   | kg/s       |
| Nu     | Nusselt number                                   |            |
| NTU    | Number transfer of units                         |            |
| p      | Static pressure                                  | Pa         |
| PPI    | Number of pores per inch                         |            |
| Pr     | Prandtl number                                   |            |
| Q      | heat transfer rate                               | W          |
| r      | Radius tube                                      | m          |
| Re     | Reynolds number                                  |            |
| s      | Curvilinear abscissa                             | m          |
| T      | Temperature                                      | K          |
| u      | x-velocity                                       | m/s        |
| u_Lo  | inlet air velocity                              | m/s        |
| v      | y-velocity                                       | m/s        |
| W      | Mechanical power                                 | W          |
| x      | Cartesian axis direction                         | m          |
| y      | Cartesian axis direction                         | m          |
| \(\alpha_{sf}\) | specific surface area density                   | m⁻¹        |
| \(\Delta\) | Difference                               |            |
| e      | Effectiveness                                    |            |
| \(\mu\) | Viscosity                                       | kg/m s     |
| \(\rho\) | Density                                        | kg/m³      |
| \(\phi\) | Porosity                                      |            |
| \(\omega\) | Number of pores per inch                     | m⁻¹        |
Subscripts

- **0**: Inlet condition
- **clean**: System without foam
- **d**: Tube diameter
- **df**: Fiber diameter
- **f**: Fluid phase
- **mf**: Metal foam
- **s**: Solid phase of metal foam

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