A Numerical Analysis on a Compact Heat Exchanger in Aluminum Foam

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Abstract. A numerical investigation on a compact heat exchanger in aluminum foam is carried out. The governing equations in two-dimensional steady state regime are written in local thermal non-equilibrium (LTNE). The geometrical 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 modeled as a porous media by means of the Darcy-Forchheimer law. The foam has a porosity of 0.93 with 20 pores per inch and the LTNE assumption is used to simulate the heat transfer between metal foam and air. The compact heat exchanger at different air flow rates is studied with an assigned surface tube temperature. The results in terms of local heat transfer coefficient and Nusselt number on the external surface of the tubes are given. Moreover, local air temperature and velocity profiles in the smaller cross section, between two consecutive tubes, as a function of Reynolds number are showed. The performance evaluation criteria (PEC) is assessed in order to evaluate the effectiveness of the metal foam.

1. Introduction

The metallic foams represent an efficient improvement for the heat transfer in general, thanks to their huge value of surface area/volume ratio. They are similar to the heat sinks where the heat flows along the fins, while in this case the ligaments of the foam are responsible to the propagation of heat. Moreover they have low weight, good rigidity and strength, damping of noise and high thermal conductivity [1]. Owing to these advantages, the metal foams can be used in heat exchangers [2], fuel cells[3], heat sinks[4] or solar thermal plants[5]. Various type of metallic foams have been investigated for their effectiveness in enhancement of the thermal behavior of the system, principally aluminum, copper, nickel. In fact many parameters can be changed in base of the possible application for the metal foam, such as the type of metal, the porosity (open cells or closed cells), the density surface area and the overall costs. The metal foams were born around thirty years ago, in particular they are used for military applications, but now they are expanding in other sectors for commercial uses like heat exchanger, noise barrier, insulator or shock absorber. The effectiveness of the open-cell metal foam to build an efficient compact the heat exchanger has been proved for a lot of cases [5-9] because it reduced the thermal inertia of the system, increase the surface heat exchange area and thus reduces the whole size of the heat exchanger. Zafari et al. [10] numerically investigated with a 3D simulation the heat transfer inside a porous metal foam using a real geometry for the computational mesh obtained through a micro-tomography images. The foam is not isotropic and the non-equilibrium thermal condition is assumed. The results showed that the increment of the porosity brings about a...
decrease of the pressure drop inside the foam due to lower resistance along the foam. Moreover the thermal equilibrium between the air and metal happens only on a short length of the foam. Xu et al. [11] analyzed the LTE and LTNE models in metal foam for heat exchangers using the volume-averaging method. This method uses a representative elementary volume (REV) as a reference where the physical quantities are assumed homogeneous. Obviously the REV size is larger than the pores of foam and smaller of a reference size of the system. The results showed that the LTE model overestimates the heat transfer results respect to the LTNE model. Lin et al [12] numerically investigated the thermal performance of a porous graphite foam heat exchangers for vehicle cooling application. Four different geometric configurations are analyzed in order to obtain the best performance with lower flow resistance. The CFD analysis showed that the wavy corrugated configuration present the best thermal and dynamic performance. Kim et al [13] experimentally investigated on the heat transfer of an aluminum foam brazed between two flat tubes of an fin-water-air heat exchanger for different type of the porous foam at various porosity and PPI. The results showed that the thermal performance are better respect to the conventional fin although the foam brings about a higher pressure drop. Nevertheless with the foam the compactness of the heat exchanger is improved. A small compact heat exchanger with open-cell aluminum foam is experimentally investigated in Boomsma et al. [14]. The aluminum foam is brazed onto a heat spreader plate and it is crossed by the water as working fluid. The data showed that the compression rate in the foam increases the heat transfer rate but even the pressure drop. Odabaee et al. [15] numerically investigated the metal foam performance inside a cylinder with the local thermal equilibrium model. The results were compared to those of a finned-tube heat exchanger and it was proved that the heat transfer rate is higher even though the pressure drop is more evident. Moreover an optimal foam porosity is obtained with a numerical study in [16] in order to achieve the best compromise between the heat transfer rate and the pressure drop inside a heat exchanger. A numerical comparison between the performance of a metal foam heat exchanger and a conventional louvered fin heat exchangers was made in Huisseune et al. [17]. For the same fan power, the metal foam heat exchanger has an heat transfer rate higher than the conventional heat exchanger up to 6 times. For the same size the finned heat exchanger has better performance respect to the metal foam heat exchanger.

After this short review, it seems that the heat exchanger with metal foam is not been fully understood, therefore in this paper the thermal performances of a compact heat exchanger with foam is numerically analyzed. The results in terms of air velocity and temperature profiles inside the foam, local heat transfer coefficient along the tubes, average Nusselt number and PEC coefficient are presented.

2. Physical Model
A schematic 2D model of the compact heat exchanger is shown in Figure 1. Five tubes are arranged in linear way along the height at midway of the aluminum foam spaced apart between them of 26 mm. An assigned constant temperature is assigned on the external surface of the tubes and assumed equal to 323.2 K (50.0 °C). The compact heat exchanger is placed in a parallel plates channel with a transversal section height of 200 mm. The thickness of the metal foam is 40 mm and the diameter of tubes is 12 mm.

![Figure 1: Schematic Model](image-url)
Thermal contact resistances between the metal foam and all the solid surface, the parallel plates and circular tubes, are considered negligible. The bottom and top surfaces of the channel are assumed adiabatic. Air enters in the channel with an uniform velocity and temperature and flows through the foam. The temperature and velocity distributions are very difficult to calculate locally inside the metal foam because it is a porous media. The foam structure is not regular and thus a continuous approach is not suitable to simulate the thermal behavior inside the foam. Therefore, it is necessary to use the local volume averaging method to model this irregular porous media as a continuous [18]. By using the local volume averaging method the variables of the equations are written estimating the average of the local variables over an appropriate volume. In base of these aspects, the metal foam is modeled with the Darcy-Forchheimer law:

\[
\frac{\Delta p}{L} = \frac{\mu}{K} V + \rho C_f V \cdot V
\]  

(1)

\(\Delta p\) is the pressure drop, \(V\) is the vector velocity of fluid phase (air), \(L\) is the thickness of the porous media, \(\mu\) and \(\rho\) are, respectively, the dynamic viscosity and the density of the air, \(K\) is the permeability of the porous medium and \(C_f\) is the drag factor coefficient. To evaluate \(K\) and \(C_f\) the relation of Calmidi and Mahajan are used [19]:

\[
K = 0.00073 (1 - \varepsilon)^{0.224} \left( \frac{d_f}{d_p} \right)^{-1.11} d_p^2
\]

(2)

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

(3)

where \(\varepsilon\) is the porosity of the metal foam, \(d_f\) and \(d_p\) are respectively the ligament and pore diameter of the metal foam. These diameters are in relation with the parameters of the metal foam by means of the following relation [19]:

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

(4)

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

(5)

\(\omega\) is the pore density of the metal foam, that is the number of pores across a linear inch. It is fixed at 20 pore per inch (PPI). Local Thermal Non-Equilibrium model is employed. The fluid phase and the solid phase are not in thermal equilibrium and thus two energy equations are necessary.

Therefore, the hypothesis in this study are:
1) The physical domain is two-dimensional
2) The aluminum foam is homogeneous and isotropic.
3) The air flow is laminar and incompressible
4) Viscous dissipation and work of pressure variation are negligible.
5) The thermophysical properties of the both phases are constant.
6) The thermal contact resistances are neglected.

The governing equations [20] are:
\[ \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \] (6)

\[ \frac{\rho}{\varepsilon^2} \left( u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = \frac{\partial p}{\partial x} - \frac{\mu u}{K} - \frac{\rho C_f}{K^{1/2}} V u + \mu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \] (7)

\[ \frac{\rho}{\varepsilon^2} \left( u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right) = \frac{\partial p}{\partial y} - \frac{\mu v}{K} - \frac{\rho C_f}{K^{1/2}} V v + \mu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) \] (8)

u and v are the component velocity. It is important to see the Darcy term and the Forchheimer extension term in the momentum equations.

Energy equation for the liquid phase:

\[ \left( \rho C_p \right)_f \left( u \frac{\partial T_f}{\partial x} + v \frac{\partial T_f}{\partial y} \right) = \frac{\partial}{\partial x} \left( k_{f,\text{eff}} \frac{\partial T_f}{\partial x} \right) + \frac{\partial}{\partial y} \left( k_{f,\text{eff}} \frac{\partial T_f}{\partial y} \right) + h_{sf} \alpha_{sf} (T_f - T_s) \] (9)

Energy equation for the solid phase:

\[ 0 = \frac{\partial}{\partial x} \left( k_{s,\text{eff}} \frac{\partial T_s}{\partial x} \right) + \frac{\partial}{\partial y} \left( k_{s,\text{eff}} \frac{\partial T_s}{\partial y} \right) + h_{sf} \alpha_{sf} (T_f - T_s) \] (10)

where \( c_p \) is the specific heat, T is the local temperature, \( k_{\text{eff}} \) is the effective thermal conductivity, \( h_{sf} \) are the local heat transfer coefficient and \( \alpha_{sf} \) is the surface area density that indicates the whole contact area between the both phases. The subscript f and s indicate respectively the fluid phase and the solid phase. The calculation of \( k_{\text{eff}} \) is not reported here for brevity but in Boomsma et al. [21] is described while for both \( \alpha_{sf} \) and \( h_{sf} \) are adopted the following correlation [19]:

\[
h_{sf} = \begin{cases} 
0.76 \left( Re_d \right)^{0.4} \left( Pr_{pcm} \right)^{0.57} \frac{k_f}{d_f} & , \quad 1 \leq Re_d \leq 40 \\
0.52 \left( Re_d \right)^{0.5} \left( Pr_{pcm} \right)^{0.57} \frac{k_f}{d_f} & , \quad 40 \leq Re_d \leq 1000 \\
0.26 \left( Re_d \right)^{0.6} \left( Pr_{pcm} \right)^{0.57} \frac{k_f}{d_f} & , \quad 1000 \leq Re_d \leq 2 \times 10^5 
\end{cases} \] (11)

\[
\alpha_{sf} = \frac{3\pi d_f}{(0.59 d_p) \left(1 - e^{\frac{d_p - \varepsilon}{0.04}} \right)} 
\] (12)

Where \( Re_d \) is the local Reynolds number referred to ligament diameter:

\[
Re_d = \frac{\rho V d_f}{\mu} \] (13)

The parameter of the metal foam are listed in table 1.
Table 1. Parameters of the aluminum metal foam.

| Parameters: | PPI | ε   | d_f (m) | d_p (m) | K (m²) | c_f |
|-------------|-----|-----|---------|---------|--------|-----|
| Values      | 20  | 0.9353 | 3.321e⁻⁴ | 2.723e⁻³ | 1.172e⁻⁷ | 0.1 |

3. Physical Model

The finite volume approach to solve the governing equations is used [22]. For the pressure-velocity coupling the SIMPLE algorithm is employed; the gradient evaluation for the spatial discretization is based on least square cell; the PRESTO algorithm is used for the pressure calculation is based on PRESTO and for energy and momentum equation the second order upwind scheme is utilized.

A comparison between the model with five tubes and another model with half tube is accomplished. The results are the same in term of heat transfer coefficient and Nusselt number along the tube as it can be seen in figure 2. Therefore, in order to reduce the computational cost, the computational domain is a half tube with the symmetry surface condition as it showed in figure 3.

![Figure 2](image1.png)

**Figure 2**: Comparison between the array of five tubes model (model A) and the half tube model (model B) in term of heat transfer coefficient along the tube.

The grid consists in rectangular cells in the air channel and triangular meshes in the foam region.

![Figure 3](image2.png)

**Figure 3**: Computational domain.

A study to obtain a solution independent from the mesh is made with three different meshes, 11828 cells, 47165 cells and 189855 cells. The first mesh is chosen because it gives good results and the computational cost is reduced. Moreover, the model is then compared with the work of Odabaee et al. [15] for the validation. The same boundary conditions, properties and mesh are used and the results present a good agreement.
4. Results and Discussion
The results are presented at different inlet velocity (0.12 ms\(^{-1}\) to 0.18 ms\(^{-1}\)) and at assigned constant temperature to the tubes for simulating the HFT equal to 323.15 K. The flow is laminar and the LTNE model is assumed for the heat exchange between the air and the foam. For the dimensionless number, the reference length is the diameter of the tube (d=0.012 m).

![Figure 4: Temperature (a) and velocity (b) profile at the centerline of the domain.](image1)

The figure 4 showed the temperature and velocity profile for an inlet velocity of 0.12 m/s along the centerline of the foam. It can be seen that there are two temperature profiles, one for the air and another for the foam because the LTNE model is assumed and so there is no thermal equilibrium between the two phases. About the velocity, the presence of the metal foam uniformizes the velocity profile at exit of the foam region and it becomes nearly equal to the inlet velocity profile.

![Figure 5: Nusselt number along the centreline at varying inlet velocity.](image2)

A comparison between the different inlet velocity is made in figure 5 in term of Nusselt Number. Obviously higher inlet velocity implies higher number of Nusselt number and therefore the heat transfer is improved. To evaluate the effectiveness of the metal foam inside a heat exchanger a comparison is made with the CLEAN case without the metal foam. The figure 6 showed the ratio
between the heat transfer coefficient and pressure drop with and without the metal foam as function of Reynolds number.

![Graphs showing ratio $h/h_{clean}$ and $\Delta p/\Delta p_{clean}$ vs Reynolds number.]

Figure 6: ratio $h/h_{clean}$ (a) and $\Delta p/\Delta p_{clean}$ (b) vs Reynolds number.

It is important to see that the heat transfer is improved because the ratio between the heat transfer coefficients is more than ten times but the pressure drop is not negligible with the metal foam. Finally, the PEC is evaluated to understand the thermal performance of the heat exchanger with the metal foam. The PEC utilized is the following [23]:

$$PEC = \frac{\left( Nu \cdot f^{1/3}\right)_m}{\left( Nu \cdot f^{1/3}\right)_{clean}}$$

(14)

Where $Nu$ is the Nusselt number and $f$ is the friction factor. The subscripts $m$ and $clean$ are referred respectively to the metal foam and clean case. The figure 7 showed the PEC at varying Reynolds number.

![Graph showing PEC vs Reynolds number.]

Figure 7: The PEC number vs Reynolds number.
5. Conclusion
In this paper a metal foam compact heat exchanger was numerically studied. The heat transfer rate from a tube in cross-flow can be increased by adding metal foam. Nevertheless there is an increase of the pressure drop. The thermal physical quantity presented the same paths of Odabaee et al [15] and Solmus [20]. Moreover, the PEC number is also evaluated in order to compare this system with other system that using another enhancement application. Further investigations are needed to study the thermal contact resistance between the surface wall and metal foam and to overcome the large pressure drop inside the foam.

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