Numerical study of metal foam heat sinks under uniform impinging flow

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Abstract. The ever-increasing demand for performance improvement and miniaturization of electronics has led to a significant generation of waste heat that must be dissipated to ensure a reliable device operation. The miniaturization of the components complicates this task. In fact, reducing the heat transfer area, at the same required heat rate, it is necessary to increase the heat flux, so that the materials operate in a temperature range suitable to its proper functioning. Traditional heat sinks are no longer capable of dissipating the generated heat and innovative approaches are needed to address the emerging thermal management challenges. Recently, heat transfer in open-cell metal foams under an impinging jet has received attention due to the considerable heat transfer potential of combining two cooling technologies: impinging jet and porous medium. This paper presents a numerical study on Finned Metal Foam (FMF) and Metal Foam (MF) heat sinks under impinging air jet cooling. The analysis is carried out by means of the commercial software COMSOL Multiphysics®. The purpose is to analyze the thermal performance of the metal foam heat sink, finned or not, varying its geometric parameters. Results are presented in terms of predicted dissipated heat rate, convective heat transfer coefficient and pressure losses.

Nomenclature

| Symbol | Description |
|--------|-------------|
| A, b   | Coefficients |
| Cp     | Heat capacity at constant pressure (J/kg·K) |
| dc     | Cell size (m) |
| de     | Strut thickness (m) |
| D      | Inlet section diameter (m) |
| f      | Inertial factor |
| hc     | Heat transfer coefficient (W/m²·K) |
| h_v    | Volumetric heat transfer coefficient (W/m³·K) |
| H      | Heat sink height (m) |
| k      | Thermal conductivity (W/m·K) |
| kp     | Fin thermal conductivity (W/m·K) |
| K      | Permeability (m²) |
| n      | Normal vector |
| p      | Pressure (Pa) |
| PPI    | Pores per inch (1/in) |
| Pr     | Prandtl number |
| Q      | Heat rate (W) |

Greek symbols

| Symbol | Description |
|--------|-------------|
| ε      | Porosity |
| μ      | Dynamic viscosity (kg/m²·s) |
| ρ      | Density (kg/m³) |
| τ      | Shear stress (N/m²) |

Subscript

| Symbol | Description |
|--------|-------------|
| 0      | Ambient |
| eff    | Effective |
| f      | Fluid |
1. Introduction

With the advent of modern electronic computers, cooling electronic chips have become one of the principal subjects for heat transfer science and engineering. The technological progress in the field of microchips has long focused on miniaturization and on increasing the computational speed. The demand of high speed and miniaturization of electronic components results in increased power dissipation requirements for thermal management. Among various cooling techniques for electronic components, forced convective cooling with air features advantages of convenience and low cost. Limitation of the forced convective cooling with air, however, lies in the relatively low heat removal rates. In fact, traditional heat sinks are no longer capable of dissipating the generated heat and innovative approaches are needed to address the emerging thermal management challenges. The heat dissipation of a heat transfer medium is a complex function of many factors, e.g., available heat transfer area, heat conduction, heat transfer coefficient, and flow resistance characteristics. These factors are strongly dependent on the structure of the heat transfer medium; and they conflict with each other; for example, higher surface areas and stronger heat conduction capability usually lead to larger flow resistances.

Recently, heat transfer in open-cell metal foams under an impinging jet has received much attention due to the considerable heat transfer potential of combining two different cooling technologies: impinging jet and porous medium. Extensive studies have been carried out to enhance the forced convective cooling with air by utilizing heat sinks with different shapes, materials, flow patterns, etc... Aluminum metal-foam heat sinks have proven to be suitable in the thermal control of high-power electronic components, with excellent cooling performance under forced convective conditions [1]. Bhattacharya and Mahajan [2] proposed a heat sink made up by a finned metal foam heat exchanger. They inserted blocks of aluminium-foam between various parallel plate fins, and investigated the forced convection for foam samples with different porosities and PPI. The heat transfer coefficient turned out to be six times that in a conventional longitudinal finned heat sink. Kim et al. [3] investigated experimentally the heat transfer in a channel partially filled with an aluminium-foam heat sink, with an imposed uniform heat flux at its basis. Experiments were made for different PPI and Reynolds number, and results showed that the lowest thermal resistance was obtained with the lowest PPI foam. The comparison with a conventional parallel-plate heat sink showed the advantage to use an aluminium-foam based heat sink. Hsieh et al. [4] obtained Nusselt number correlations for different velocities, PPI and porosities, with an experimental apparatus similar to a processor cooling heat sink, with the heat source orthogonally invested by a stream of air. Shih et al. [5] showed that the performance of an aluminium-foam heat sink is affected by the height of a sink under impinging-jet flow conditions. The Nusselt number was not monotonically correlated with the height and an optimal height was found, as a function of PPI and porosity. Successively Shih et al. [6] demonstrated that the heat transfer performance of the same aluminium-foam heat sink improved by using an annular flow-restricting mask, that reduced the flow outlet section.

Recently, using 3-D printing and investment casting techniques, Krishnan et al. [7] fabricated and studied FMF heat sinks with regular foam structures, and concluded that on an equal pumping power basis, FMF heat sinks outperform conventional plate-fin heat sinks. Feng et al. [8] carried out a combined experimental and numerical study on finned metal foam (FMF) and metal foam (MF) heat sinks under impinging air jet cooling. Comparisons of experimental and numerical results reveal that using the laminar Darcy’s extended model can predict fairly both the heat transfer and pressure drop of MF and FMF heat sinks under high Reynolds numbers. A numerical investigation was carried out to characterize the thermal performance of finned metal foam heat sinks subject to an impinging air flow by Feng et al [9]. The main objective of the study was to quantify the effects of all relevant configurational parameters (channel length, channel width, fin thickness, and fin height) of the heat sink upon the thermal performance. Various simulation cases for different combinations of channel
parameters were carried out to obtain the Nusselt number correlation. Based on the inviscid impinging flow, a pressure drop correlation was derived for impinging flow in finned metal foam heat sinks. By using the above referred correlations, the thermal performance of finned metal foam heat sinks was compared with that of conventional plate-fin heat sinks. It was demonstrated that the finned metal foam heat sinks outperformed the plate-fin heat sinks on the basis of given weight or given pumping power.

This paper presents a numerical study on finned metal foam (FMF) and metal foam (MF) heat sinks under impinging air jet cooling. The numerical analysis is carried out by means of the commercial software COMSOL Multiphysics®. The purpose is to analyze the thermal performance of the metal foam heat sink, finned or not, varying its geometric parameters. Results are presented in terms of dissipated heat rate, convective heat transfer coefficient and pressure losses.

2. Mathematical model

The Finned Metal Foam (FMF) heat sink is represented in Fig.1. The heated plate is located at the bottom of the heat sink. The air flow comes from a circular section located on the top, takes heat from the plate and exits from the heat sink lateral sides. The Metal Foam (MF) heat sink has the same configuration but without fins. The heated plate is a 68 x 68 mm² square. The ratio of the impinging flow section diameter, D, to the heated plate side, W, is 0.25, 0.50, 0.75, 1.0. The height of the computational domain, H, is 10, 20, 30, 40 mm. The fins thickness, t, is 2 and 4 mm. The number of fins used in the computations is 1, 2, 4, 6, 8, and 10. The porosity and the PPI of the aluminum foam are 0.88, 0.91, 0.94, 0.97 and 5, 10, 20, 40, respectively.

2.1. Governing equations

The porous medium is treated as an equivalent homogeneous medium, and governing equations are written by using the Volume Averaging Technique (VAT). Due to the physics of the problem, a Local Thermal non-Equilibrium (LTNE) model is employed. The flow is assumed to be steady and laminar. Effects of buoyancy, thermal dispersion and thermal tortuosity are neglected. Thermophysical properties are considered to be independent of the temperature. The governing equations for the porous domain are:

![Figure 1. Finned Metal Foam (FMF) heat sink.](Image)
\[ \nabla \cdot \langle \mathbf{u} \rangle = 0 \]  
(1)

\[ \frac{\rho}{\varepsilon^2} \langle \mathbf{u} \rangle \cdot \nabla \langle \mathbf{u} \rangle = -\nabla \langle p \rangle + \frac{\mu}{\varepsilon} \nabla^2 \langle \mathbf{u} \rangle - \frac{\mu_f}{K} \langle \mathbf{u} \rangle - \frac{\rho f}{\sqrt{K}} [\langle \mathbf{u} \rangle] [\langle \mathbf{u} \rangle] \]  
(2)

\[ (\rho C_p) \langle \mathbf{u} \rangle \cdot \nabla \langle T_f \rangle = \nabla \cdot \left( k_{\text{eff},f} \nabla \langle T_f \rangle \right) + h_v \left( \langle T_f \rangle - \langle T_f \rangle \right) \]  
(3)

\[ 0 = \nabla \cdot \left( k_{\text{eff},s} \nabla \langle T_v \rangle \right) - h_v \left( \langle T_v \rangle - \langle T_f \rangle \right) \]  
(4)

while the Laplace equation is used for the aluminium fins.

2.2. Closing coefficients
In order to solve porous medium governing equations, closing coefficients are required for
permeability, \( K \), inertial coefficient, \( f \), thermal conductivities, \( k_{\text{eff},f} \) and \( k_{\text{eff},s} \), and volumetric heat transfer coefficient, \( h_v \). Permeability and inertial coefficient are obtained from the following equations [10]:

\[ K = \frac{d_c^2}{1039 - 1002 \varepsilon} \]  
(5)

\[ f = \frac{0.5138 \varepsilon^{2.39}}{d_c} \sqrt{K} \]  
(6)

where \( d_c \) is the cell size. For the energy equation, the effective thermal conductivities, \( k_{\text{eff},f} \), \( k_{\text{eff},s} \), and the volumetric heat transfer coefficient, \( h_v \), are required. The effective thermal conductivity, \( k_{\text{eff}} \) is [11]:

\[ k_{\text{eff}} = k_{\text{eff},s} + k_{\text{eff},f} = \frac{k_s (1 - \varepsilon)}{3} + k_f \varepsilon \]  
(7)

The volumetric heat transfer coefficient, \( h_v \), is obtained from the following correlation [12]:

\[ h_v = \left( 32.50 \varepsilon^{0.38} - 109.94 \varepsilon^{1.38} + 166.65 \varepsilon^{2.38} - 86.98 \varepsilon^{3.38} \right) \text{Re}_c^{0.438} \varepsilon^{0.438} \frac{k_f}{d_c^2} \]  
(8)

where \( \text{Re}_c = |\mathbf{u}| \rho d_c/\mu \) is the cell Reynolds number.

2.3. Boundary conditions
Boundary conditions of the problem are resumed in Table 1. A rectangular coordinate system with the origin located at the bottom-left corner is employed for the three-dimensional model. For the heated plate, a 70 °C uniform temperature is imposed. The thermal conductivity of the aluminium fins is set equal to 170 W/m K. In the inlet flow section, a Robin boundary condition is used for both solid phase of the foam energy equation and aluminium fin equation. In particular, for the solid phase foam equation, a correlation for staggered cylinder in crossflow is used [13]:

\[ h_v = A \text{Re}^b \text{Pr}^{0.37} \frac{k_f}{d_s} \]  
(9)
where the Reynolds number is defined as \( Re = \rho |u| d_s / \mu \), with \( d_s \) the strut thickness, and the constants \( A \) and \( b \) depending on the Reynolds number [13].

### Table 1. Boundary conditions of the problem.

| Momentum (fluid) | Energy (fluid) | Energy (solid) | Fin |
|------------------|----------------|----------------|-----|
| Inlet \( u = -|u| n \) \( T_o = 20^\circ C \) | \( -k_{off,s} \frac{\partial \langle T_s \rangle^y}{\partial n} = h_e \left( \langle T_s \rangle^y - T_o \right) \) | \( -k_p \frac{\partial T_s}{\partial n} = h_e \left( T_s - T_o \right) \) |
| Other | \( |u| = 0 \) | \( \frac{\partial \langle T_s \rangle^y}{\partial n} = \frac{\partial \langle T_s \rangle^y}{\partial n} = \frac{\partial T_s}{\partial n} = 0 \) |
| Heat sink sides | \( p = 101 \text{ kPa} \) \( \tau_{ij} = 0 \) | \( \frac{\partial \langle T_s \rangle^y}{\partial n} = \frac{\partial \langle T_s \rangle^y}{\partial n} = \frac{\partial T_s}{\partial n} = 0 \) |
| Foam/fin Interface | \( |u| = 0 \) | \( k_{off,f} \frac{\partial \langle T_s \rangle^f}{\partial n} + k_s \left( 1 - \varepsilon \right) \frac{\partial \langle T_s \rangle^y}{\partial n} = k_p \frac{\partial T_f}{\partial n} \) |
| Bottom | \( |u| = 0 \) | \( \langle T_f \rangle^f = \langle T_s \rangle^y = T_e = 70^\circ C \) |

For the aluminium fin, the following equation is used [14]:

\[
h_e = 0.6 \Re^{0.5} \frac{k_f}{t} \tag{10}
\]

where the Reynolds number is defined by using the fin thickness as the characteristic length.

### 3. Results

The solid-phase dimensionless temperature on the heated plate, for \( D/W = 1 \) and \( D/W = 0.25 \), are reported in Fig.2. It is shown that at higher \( D/W \) values the air reaches the whole heat sink, while, at lower \( D/W \) values not the whole heat sink is employed, since most of the heated plate remains at \( \langle T_s \rangle^y/\langle T_s \rangle^y_{y=0} = 1 \). This means that, when the impinging section is larger, the efficiency of the heat sink is improved since all the heated plate contributes to heat transfer. However, it should be pointed out that reducing the impinging sections makes the central region of the heat sink colder than that of the larger impinging section case, since the air velocity locally increases.

Results are reported in terms of the heat transfer coefficient, evaluated with the following equation:

\[
h_e = \frac{\dot{Q}}{W^2 \left( \langle T_s \rangle^y - T_o \right)} \tag{11}
\]

where \( \dot{Q} \) is the heat rate and \( W^2 \) the area of the heated plate. The heat transfer coefficient is calculated on the heated plate.

Dissipated heat rate as a function of the velocity, for the FMF and MF heat sinks, together with data by Feng et al. [8], are reported in Fig. 3. The figure exhibits differences between predictions by the present model and Feng et al.’s model less than 8% for the FM model and less than 18% for the FMF.
Figure 2. Solid-phase dimensionless temperature on the heated plate: a) $D/W = 1$; b) $D/W = 0.25$.

Convection heat transfer coefficient and pressure drop are evaluated for $\varepsilon = 0.92$, PPI = 10, $|u| = 0.5$ m/s, $t = 2$ mm, $H = 20$ mm and $D/W = 1$.

The heat transfer coefficient and the pressure drop as a function of the number of fins, for $t = 2$ mm and 4 mm, are presented in Fig. 4. The figure points out that, both for 2 and 4 fins, the heat transfer coefficient increases, attains a maximum and then decreases. This behaviour is due to a competition between two factors: the increase in the number of fins, which enhances the dissipated heat rate, and the reduction in the foam inserts volume. In the first part of the curves, the increase in the fins number improves the heat sink performance, but it reduces the foam volume, thus reducing the dissipated heat. As to the pressure drop, more fins means less foam volume, thus less pressure drop due to the reduction of foam inserts volume.

The heat transfer coefficient and pressure drop as a function of the sink height, for FMF and MF heat sinks with four fins, are reported in Fig.5. The heat transfer coefficient reduces with height because the impinging jet reaches the hot plate more difficultly, and also pressure drop reduces since a larger fraction of the fluid exits the heat sink from the side walls of the foam. The comparison of the MF heat sink with the FMF heat sink shows that the latter exhibits a higher heat transfer coefficient but also a higher pressure drop.

Figure 3. Dissipated heat rate vs. the velocity, for the FMF and MF heat sinks.
The heat transfer coefficient and pressure drop as a function of the ratio of the impinging jet diameter to the side length of the heated plate, for FMF and MF sinks with four fins, are reported in Fig.6. We notice that the larger the section the larger the heat transfer coefficient, since increasing the plate size decreases the velocity of the fluid but allows the fluid to reach the plate in an easier way. The MF configuration performs better than the FMF one at low values of $D/W$ because it is finned and has a smaller free area; thus the contribution of cold air is drastically reduced. Pressure drop in the MF configuration is unaffected by the size of the impinging section since the mass flow rate is unchanged, while pressure drop in FMF is affected by the size of impinging section because of the fins, that induce a concentrated pressure drop.

The heat transfer coefficient and pressure drop as a function of the porosity, for FMF and MF sinks with four fins, are reported in Fig.7. The figure shows that increasing the porosity reduces the heat transfer coefficient, because it is inversely proportional to the porosity, as well it reduces also the pressure drop, since the foam contains a decreasing fraction of solid.

The convection heat transfer coefficient and pressure drop as a function of PPI, for FMF and MF sinks with four fins, are reported in Fig.8. The figure exhibits a slight increase in heat transfer coefficient and a marked increase in pressure drop at increasing PPI. Heat transfer in MF configuration is almost independent of PPI, since at high values of the volumetric heat transfer coefficient a LTE condition is attained and enhancing PPI is useless.
Figure 6. Heat transfer coefficient and pressure drop vs. the ratio of the impinging jet diameter to the side length of the heated plate, for FMF and MF sinks with four fins.

Figure 7. Heat transfer coefficient and pressure drop vs. the porosity, for FMF and MF sinks with four fins.

Figure 8. Heat transfer coefficient and pressure drop vs. PPI, for FMF and MF sinks with four fins.
4. Conclusions
In this paper a numerical analysis of heat transfer and pressure drop in a heat sink under impinging air jet cooling is carried out. Two different heat sink configurations have been investigated: a metal foam heat sink and a finned metal foam heat sink. The air-saturated foam is treated as an equivalent homogenous medium, by employing the Local Thermal non-Equilibrium model. The problem is numerically solved by means of the COMSOL Multiphysics® code. The effects of the ratio of the impinging jet diameter to the heated plate side, the height of the heat sink, the fins thickness, the number of fins, the foam porosity and PPI are analyzed. Results show that when the impinging jet diameter is equal to the heated plate side, both the heat transfer coefficient and pressure drop in the finned metal foam heat sink are larger than those in the metal foam heat sink. The metal foam heat sink performs better than the finned metal foam heat sink when the impinging jet diameter is smaller than the heated plate side.

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