Heat Transfer Performance Analysis in Porous Heat Exchanger

Assistant professor PhD. eng. **Petre OPRIŢOIU***

*Technical University of Cluj-Napoca, Department of MTC, str. Observatorului, nr.72-74, 400363, Cluj-Napoca, Romania, E-mail: petre.opritoiu@mtc.utcluj.ro

Abstract: *In the case of electronic components cooling, out of many ways and variants of existing heat transfers, the fluid single-phase forced convection heat is an effective and widely practiced method, especially due to its simplicity and economical character. In order to allow the evacuation of increasing heat fluxes, the method was optimized by making special heat exchangers, more and more compact, as meso-channel, micro channels [1] and more recently the porous metal heat exchangers [2,3].*
The objective of the research is to obtain a functional relationship between the cellular structure of open-cell metallic foam and the heat transfer characteristics for forced convective flows. Ideally foams would simultaneously maximize the amount of heat transfer and minimize the pump power needed to force the fluid passing through.
The working method used, is the numerical simulation using FLUENT software in which the actual characteristics of porous metal indicated by the manufacturing company, Erg Materials and Aerospace, were introduced [4].
A simple simulation model is developed in FLUENT to capture the most important behavioral trends of energy flow due to forced convection and conduction through cell ligaments of the cellular foam [5, 6, 7]. The overall heat transfer coefficient of the heat exchanging system and the pressure drop experienced by the fluid flow are obtained.
Trends of heat transfer predicted by the model are expected to be valid for a wide range of open-cell foams and are in reasonable agreement with available experimental data on aluminum foams [8]. The model is applied to evaluate the heat transfer efficiency of open-celled foams as heat sinks for power electronic components. Heat dissipation in high power electronics poses serious challenges for the integration of materials selection with thermal design, circuit design and manufacturing technology.

Keywords: *simulation, cellular structure, flow characteristics, heat transfer performance.*

1. Introduction

Metal foams find applications in ultra light structures where stiffness, strength and toughness are emphasized. Open-celled metal foams can be used to enhance heat transfer in applications such as cryogenic heat exchangers, heat exchangers for airborne equipment, coal combustors, compact heat sinks for high power electronic devices, heat shielding for aircraft exhaust, compact heat exchangers, liquid heat exchangers, air-cooled condenser-cooling towers and regenerators for thermal engines [9]. More uses of these relatively new materials are expected in the future, particularly because low density foams with remarkably uniform and regular cell morphologies are currently being developed using affordable processing methods.
One of the primary reasons to study forced convection in metal foams is to provide information necessary for the possible applications of these materials in electronic cooling and other thermal systems. The fibers of these materials could be thought of as a network of complex extended surfaces giving the advantage of increasing the interfacial area. In addition to the increased interfacial area the formation of eddies or fluid mixing promotes the heat transfer enhancement (fig.1).
Through the phenomenon of turbulence, an intensification of the process of heat transfer and faster exhaustion of heat accumulated when using this type of heat exchanger can be noticed. The overall heat transfer coefficient of the heat exchanging system is calculated, and the pressure drop experienced by the fluid flow obtained. These results are used to analyze and guide the design of optimum foam structures that would maximize heat transfer per unit pumping power. The objective is to select materials that enable the aluminum foam to operate with high power density (up to 100 W/m^2), while maintaining its temperature below that needed to ensure acceptable reliability: usually 90°C.
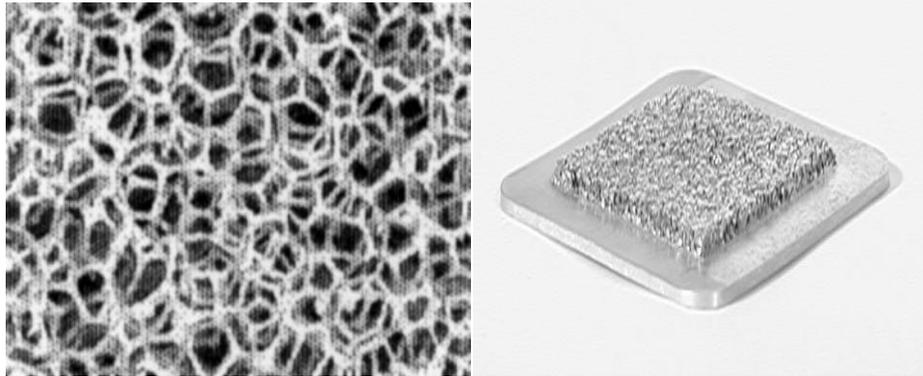
Fig. 1. Micro-computed-tomography of open-celled aluminum foam and aluminum foam heat exchanger [3].

2. Specification of the model

The present work provides a simulation model for the heat transfer in open-cell aluminum foam, when they are used in forced convective mode with a low conductive fluid. The simulation model has been idealized in several ways to limit the complexity of forced convective flow across open-celled foam. Some of the simplifying assumptions made in order to analyze heat transfer in the porous material likely lead to overestimate the actual level of heat transfer but nevertheless should capture the approximate functional dependence of the different control variables such as foam density, cell size and fluid velocity [9]. Assumptions:

1. The porous medium is isotropic and homogeneous with constant porosity, or the porosity variations can be accurately calculated,
2. Natural convection and radiation heat transfer effects inside the porous medium can be neglected,
3. The physical properties of the solid and fluid phase remain constant throughout the temperature range,
4. The solid and the fluid phases are in the thermal equilibrium,
5. Flow is steady and fully developed.

Most of the previous studies on heat transfer in porous medium are based on Darcy's law which is only valid for Reynolds numbers less than about 10 [10,11]. Under such conditions, the fluid and the solid matrix may be assumed to be in local thermal equilibrium so that the fluid saturated porous medium can be treated as a continuum. The model assumes and justifies that there is local thermal equilibrium between the solid and the fluid. This approach eliminates the need for rigorous numerical modeling of three-dimensional flow and heat transfer in and around the pores. Another advantage is that the current simulation model is easily verified by experiments [8].
Consider a rectangular block of porous foam, having a constant cross-sectional area. The analysis uses the typical parameters reported by the foam manufacturers such as the porosity (ε) and the area density (σ), defined as the ratio of the surface area of the foam to the volume (table 1). The hydraulic diameter is determined based on the size of the compressed porous cell (d_P), of the metal filament's diameter (d_l) and the porous density (PPI).

TABLE 1: Metal foam properties [4].

No.	PPI	ε [-]	d_l [mm]	d_P [mm]	Size LxWxt [mm]	σ [m^2/m^3]
1	10	0.914	0.406	5.08	250.8 x 101.6 x 50.8	809.1
2	10	0.704	0.406	1.93	250.8 x 101.6 x 50.8	2053.1
3	10	0.682	0.406	1.24	250.8 x 101.6 x 50.8	3169.3
4	20	0.924	0.203	2.90	250.8 x 101.6 x 50.8	1240.2
5	20	0.774	0.203	0.89	250.8 x 101.6 x 50.8	3593.7
6	20	0.679	0.203	0.63	250.8 x 101.6 x 50.8	5104.3
7	40	0.923	0.102	1.70	250.8 x 101.6 x 50.8	1800.8
8	40	0.918	0.102	1.70	250.8 x 101.6 x 101.6	1800.8

The mean cell size (a) for these foams, according to the ERG catalogue, are 2.0, 1.0 and 0.5 mm, respectively, correspond to 10, 20 and 40 PPI. Thermo-physical properties of aluminum used in the simulation are: density 1.6(10PPI), 2.7(20PPI) and 7.9(40PPI) kg/m^3, specific heat 895J/kgK, thermal conductivity 218W/mK.

The final overall dimensions of the compressed foam blocks used in pressure-drop and heat transfer simulations were 250mm(H) × 100mm(L) × 50mm(W), with the cross-sectional area normal to the flow direction measuring 250mm×100mm. The small thickness is suitable for small foam cooling design such those used in electronic cooling. To make them functional heat exchanger, each foam was brazed in a central position to a 12.7 mm thick heat spreader plate made by solid aluminum.

A typical flow and heat transfer configuration is shown in fig. 2. Cooling fluid having velocity u_0, temperature T_∞ and pressure p_0, is forced into the foam at x =0 (the inlet) and exits at x = L (the outlet) with temperature T_e and pressure p_e. A heat source ($q^"$) is bonded or joined to a thin conductive substrate on which a block of open-cell aluminum foam of length L and thickness W is attached. The foam is then placed in a channel, and cooling fluid of velocity u_0 at a temperature T_∞ is pumped through the open celled material, thereby affecting heat transfer from the hot source to the cooling fluid.
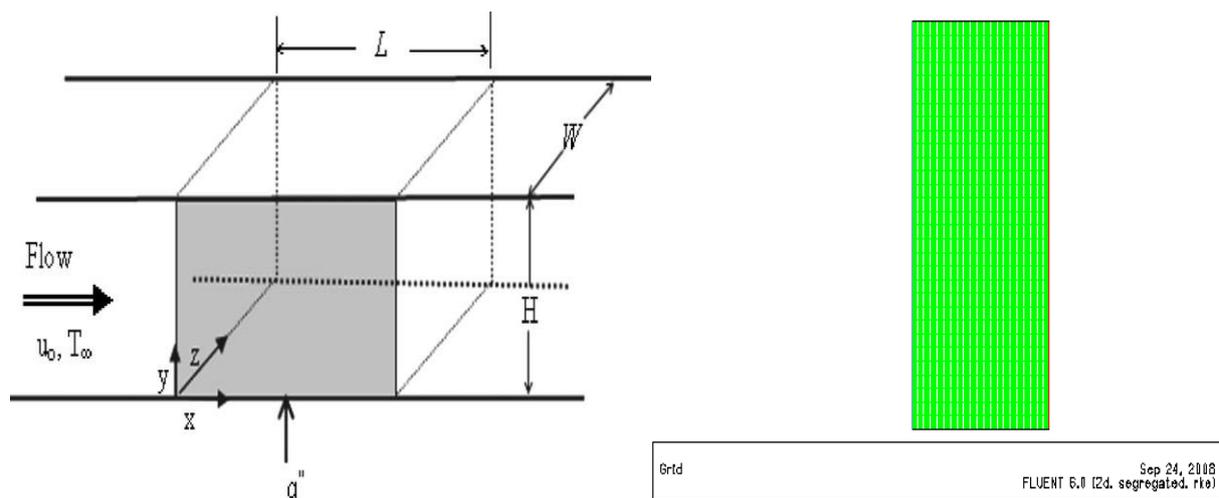
Fig. 2. Porous heat exchanger geometrical model [13] and the equal-sized cubic cells grid [14].

The cubic unit cell is chosen for its simplicity, allowing for approximate closed-form solutions for important heat transfer parameters. It is elegant and efficient to create mesh in one process. This also is more robust when topologically complex problems such as the open-cell metallic foam are being investigated. This means that the mesh generated required no intervention or correction so that solution is, as in flow/heat transfer presented here straightforward.

Uniform temperatures T_1=85°C (358K) on the bottom (y=0) and T_2=22°C (295K) on the top (y=H) are prescribed. The sandwich structure is thermally insulated at both ends and in lateral sides. The temperature of the cooling fluid increases continuously as it flows along the heat exchanger and so the driving force for heat transfer is continually diminished (fig.3).

We therefore wish to establish the fluid temperature as a function of downstream position x, the plate temperature T_1 and the properties of the foam. Several simplifications are made to arrive at an approximate solution for the steady-state temperature distribution of the fluid inside the heat exchanger.
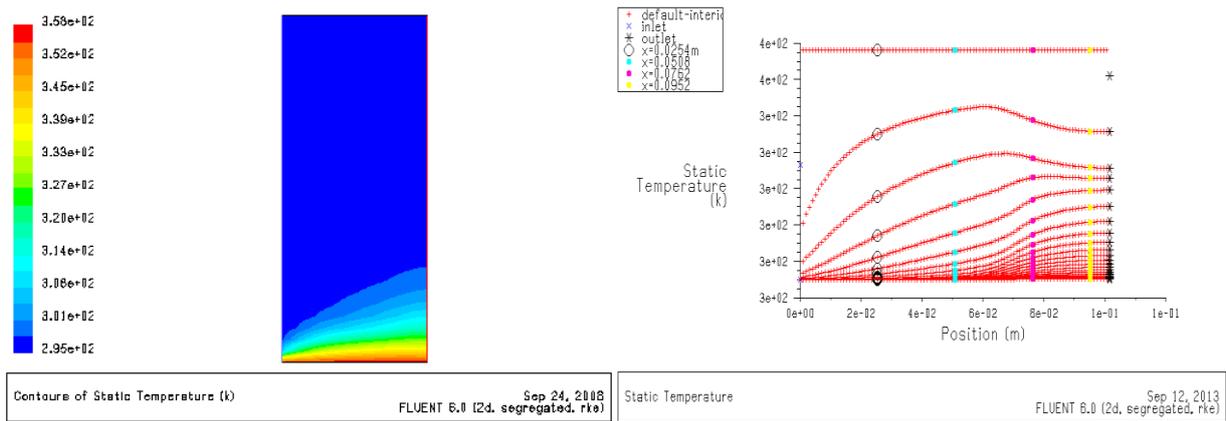
Fig. 3. Contour of static temperature and corresponding xy plot [8].

Turbulence intensity values and resistance coefficients of viscous and inertial type, required for running the FLUENT software for a porous medium were obtained using MATHCAD program, starting from equations specific to Brinkman's porous environment [8]. The overall heat flux dissipation, heat transfer coefficient and pressure drop caused as the fluid flows across the foam are calculated using the model and the optimum foam properties for the best heat transfer performance are suggested.

3. Application: porous heat exchanger for power electronic devices

The model presented in previous section can be used to analyze a variety of heat exchanger applications where the large surface density of open-celled metal foams is required. The maximum amount of heat that the fluid can extract from the system was determined according to equation [9]:

$$Q = \dot{m} c_p (T_1 - T_0) \tag{1}$$

where $\dot{m}$ is the flow rate, c_p is the specific heat, T_0 and T_1 are the inlet and the base temperatures of the fluid measured by thermocouples. Heat dissipation plots were obtained for different porosities and porous densities. The model and its underlying assumption were verified by simulation for several aluminum foam samples using air as the coolant [8]. All simulations were performed under steady state conditions and were carried out in the range of average air velocity 0.8 < v < 8 m/s.

For both water and air cooling, dissipative heat flux removed by a porous heat exchanger of 50 mm thickness as a function of relative density (ρ) is plotted on fig. 4, for 10, 20 and 40 PPI porous densities, porosity ε= 0.927, $λ_{s,Al}$ = 200W/mK, ΔT= T_1-T_0= 63°C (336K).
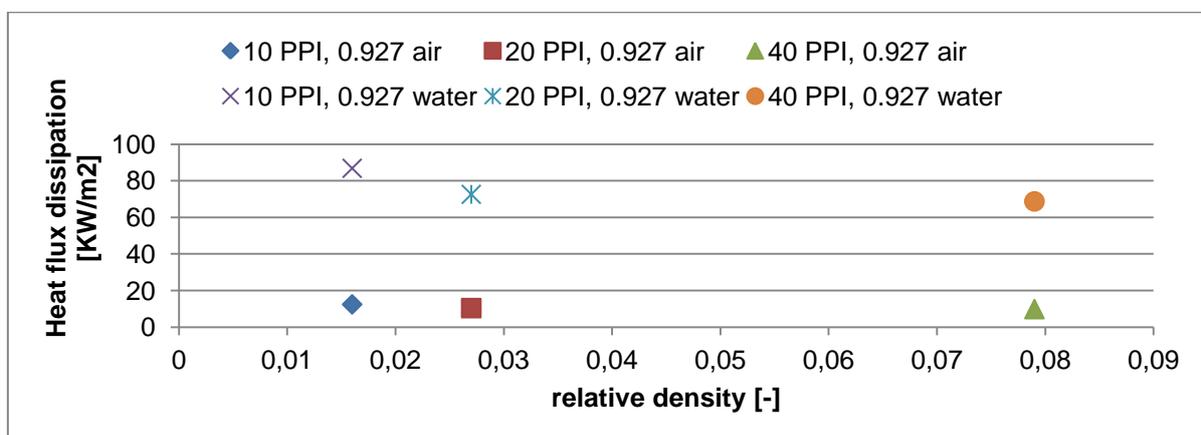
Fig. 4. Heat flux dissipation as a function of relative density.

It is seen that, even with forced air convection in aluminum foam heat exchanger, with relative density of 0.079(40 PPI), heat is efficiently dissipated (q= 10 kW/m^2, if air cooling and q= 69kW/m^2, if water cooling), indeed making it a very attractive device due to its compactness and light weight. Under these conditions the difference between the temperature of the wall and the air does not exceed 63°C. The heat transfer from the foam to the fluid will increase as either the porosity decreased (thus increasing surface area for heat transfer) or as the relative density (ρ) increases (thus increasing heat conduction through the ligaments) or as the velocity of the fluid increases (fig.5).
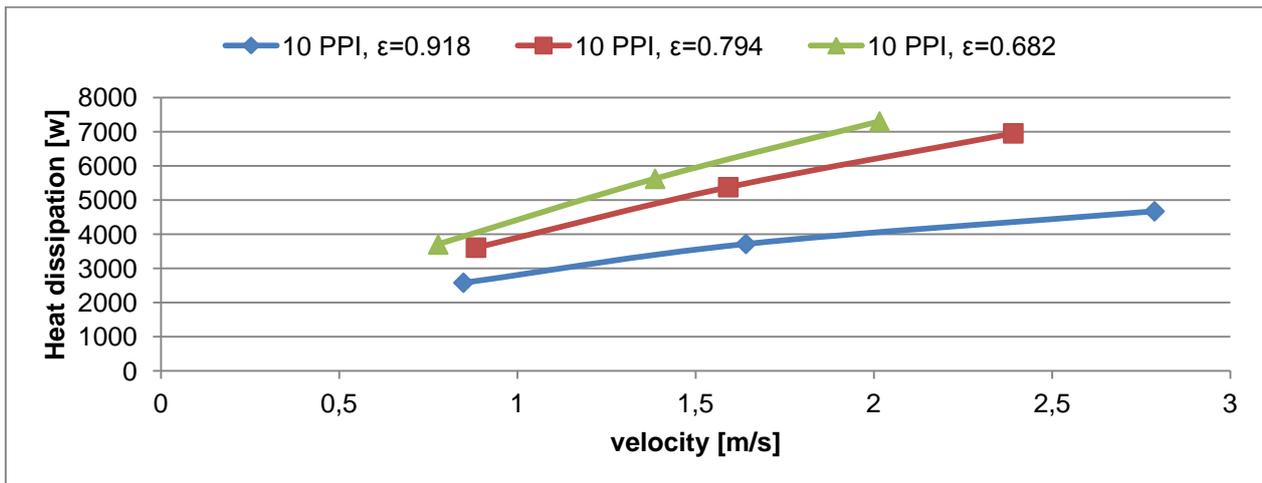
Fig. 5. Heat flux dissipation plotted against flow velocity, for 10 PPI porous density and porosities ε=0.918, 0.794 and 0.682.

Bigger heat dissipative was obtained for higher velocities and for foams with higher surface area densities or lower porosity. With σ>1000 m^2/m^3 even at very low densities (ρ=0.02), the conclusion drawn from these results is that open-celled metal foams well qualify as compact heat exchangers. Compact heat exchanger generally require σ>700 m^2/m^3 and are essential in applications where the size and weight of the heat exchanger is constrained due to design considerations. The average heat transfer coefficient is defined as [9]:

$$\alpha = \frac{q}{T_w - T_f} \tag{2}$$

where T_w is the average wall temperature, measured by the thermocouple, T_f is the average value of the fluid temperature, calculated as the mean of the inlet T_0 and the outlet T_2 , temperature of the air in the test section. For metal foams α depends on the fundamental variables: foam density, channel width, fluid velocity, etc. the simulated values of α is much larger than that measured by experiments. The overestimates may be attributed to the simplifying assumptions of the model in order to obtain simple solutions.

For turbulent flows which are expected to occur in metal foams, the convective heat transfer coefficient calculated from equation (2) is typically on the order of 10^2~10^4 W/mK. It is noticed that at the same porosity, the thermal convection coefficient is higher if the porosity density is higher, the difference between them increases with decreasing speed (fig.6).
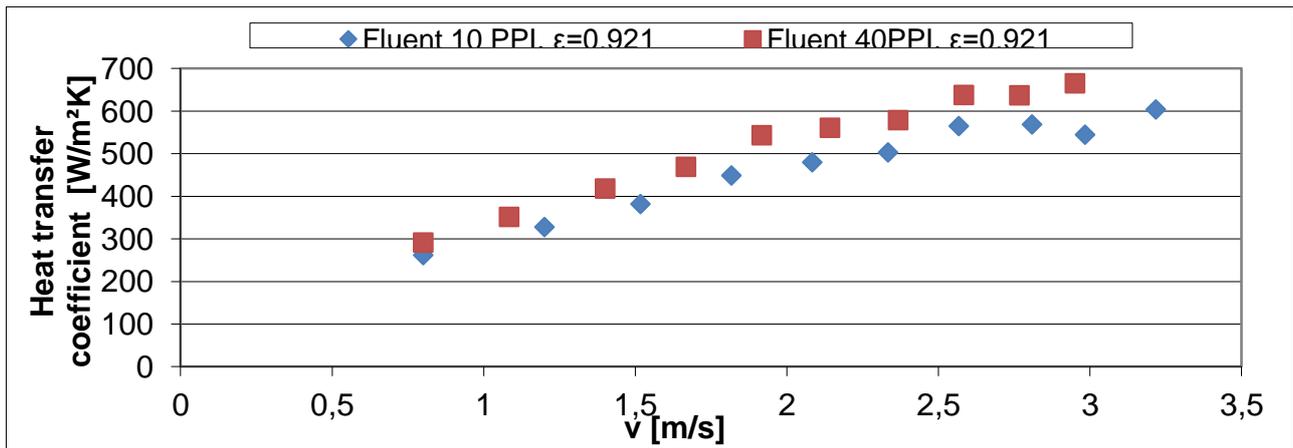
Fig. 6. Heat transfer coefficient plotted against flow velocity, for 10 and 40PPI porous densities and porosity ε=0.921.

For air at atmospheric pressure and room temperature with $v_{max.}$=3.5 m/s, one calculates from equation (2) that $α_{max.}$=665 W/(m2K). At the same time, it can be observed that at the same speed and porosity density, the thermal convection coefficient grows with decreasing porosity and increasing relative density (fig.7, 8).
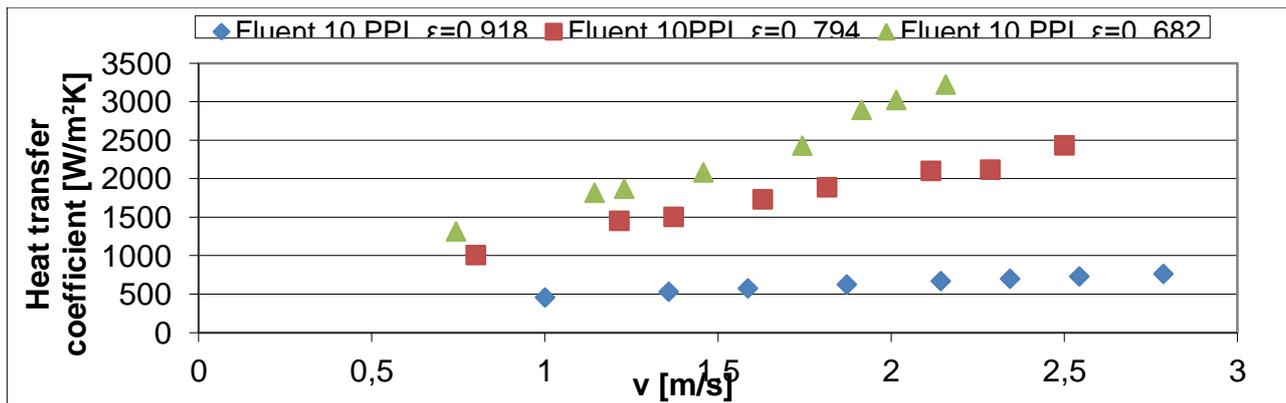
Fig. 7. Heat transfer coefficient plotted against flow velocity, for 10 PPI porous density and porosities ε=0.918, 0.794 and 0.682.
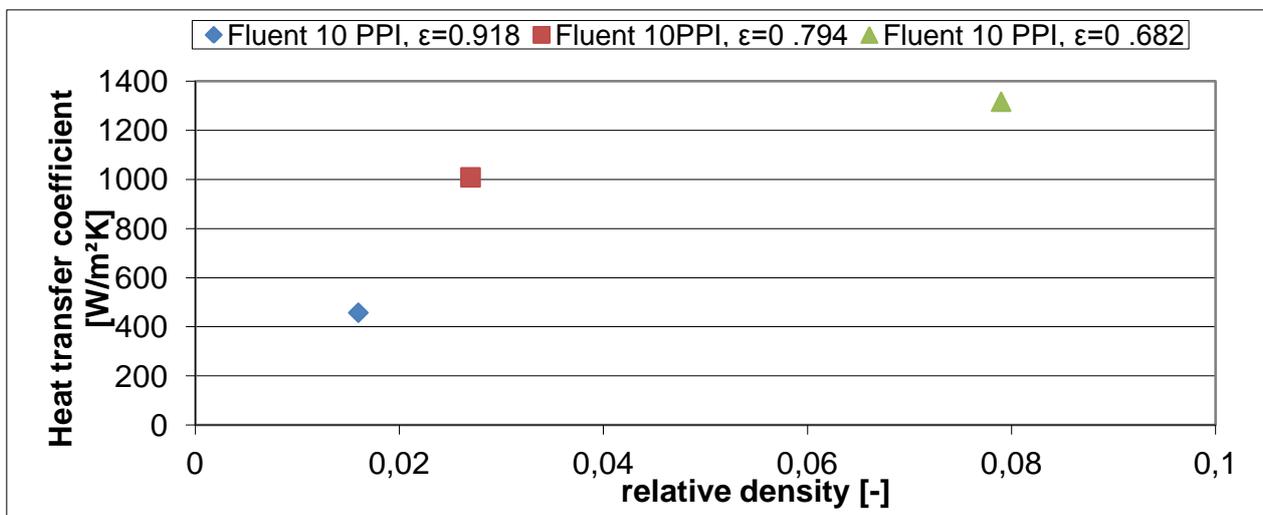
Fig. 8. Heat transfer coefficient as a function of relative density, for 10 PPI porous density and porosities ε=0.918, ε=0.794 and ε=0.682.

Pressure drop per unit length for porous media may be expressed using Forchheimer equation:

$$\frac{\Delta P}{L} = \frac{\mu_f}{K} \cdot v_f + \frac{c_F}{\sqrt{K}} \rho_f \cdot v_f^2 \tag{3}$$

where $\Delta p/L$ - pressure drop per unit length, μ_f - fluid viscosity, K - permeability, v_f - velocity, c_F - inertial coefficient, ρ_f - fluid density.

The pressure drop is also influenced by the porosity, the density of porosity and the fluid velocity and thickness. At the same porosity density the pressure drop is influenced by the flow rate of working agent and thickness. At the same porosity and flow rate, the pressure drop increases with the density of porosity (fig.9).
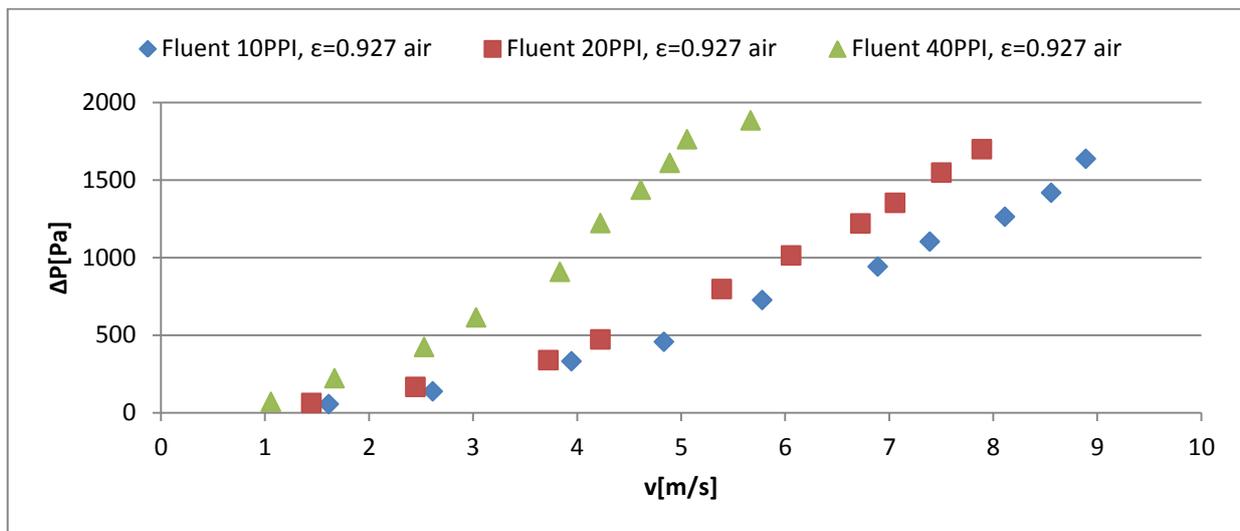
Fig. 9. Comparison between simulated pressure drop data at different velocities, for 10, 20 and 40 porous densities and porosity ε=0.927.

As seen compression has profound effect on the pressure-drop behavior of compressed foam. As expected, those foams which possess the highest solid fraction (highest porous density and lowest ε), generated the largest pressure drop (fig.10).
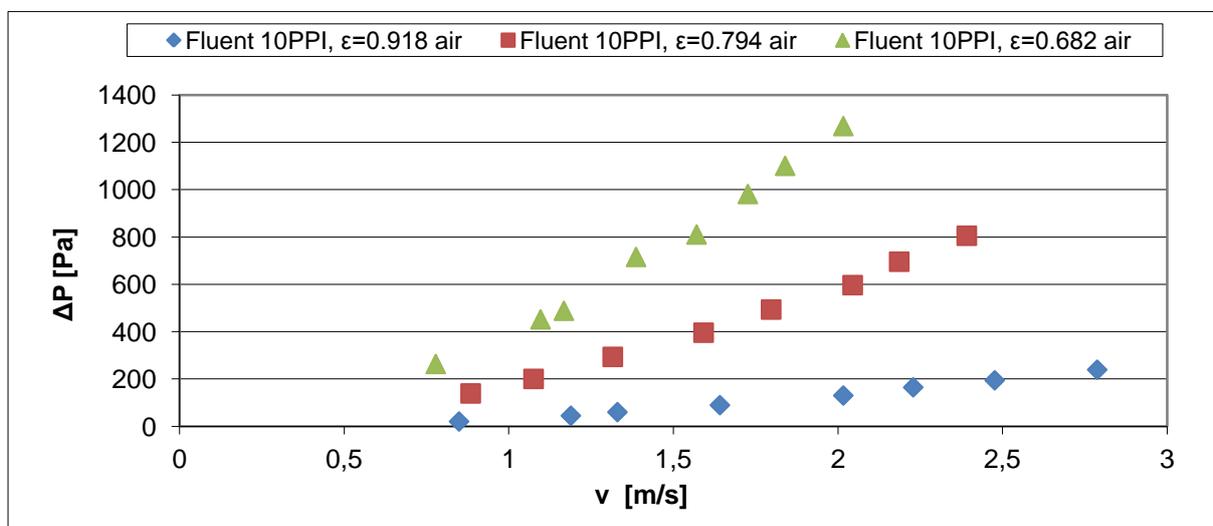
Fig. 10. Comparison between simulated pressure drop data at different velocities, for 10 PPI porous density and different porosities ε=0.918, ε=0.794 and ε=0.682.

The predicted pressure drop are plotted in fig. 11, as functions of relative foam density ρ, for air at T_2 =22°C flowing across the duct filled with aluminum foam. These results are generally in agreement with the experimental measurements by [8].

Whilst a foam having the smallest cell size a=0.5mm (40PPI), and highest relative density ρ=0.079 maximizes heat transfer it unfortunately also maximize pressure drop. The predicted minimum pressure drop occurs at a relative density equal to 0.016 (10 PPI) for aluminum ERG foams.

A high pressure drop not only increases the operational cost but often also leads to intolerable noises. One of the advantages of designing exchangers with foams is that they still have excellent heat transfer coefficient even if the mean flow velocity falls below the critical value required to maintain turbulent flow. Also, the foam itself may act as a noise suppressor due to its sound absorption capabilities.
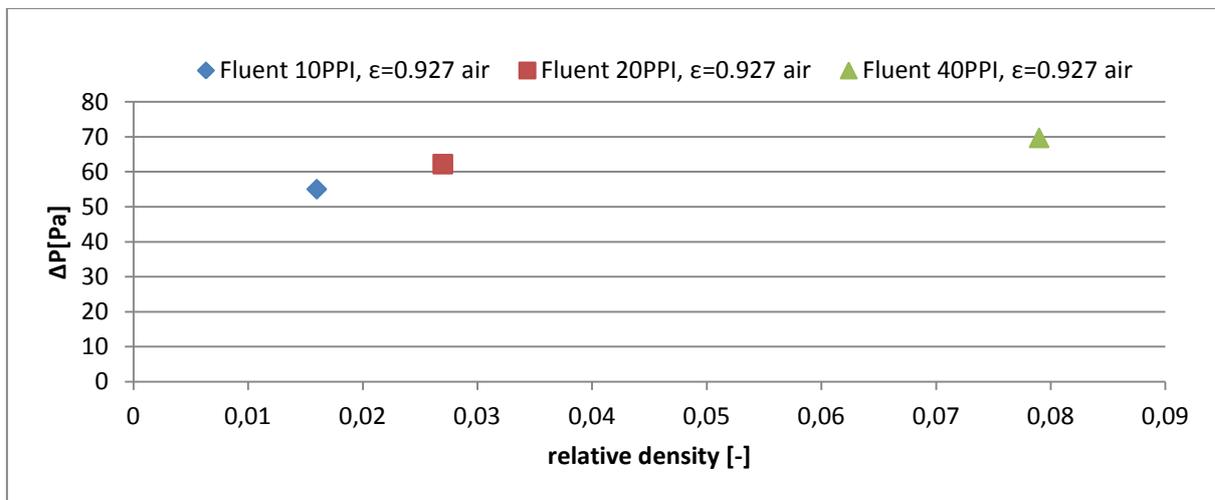
Fig. 11. Pressure drop as a function of relative density ρ=0.016(10 PPI), 0.027(20PPI) and 0.079(40 PPI).

The heat transfer performance of the foam may be best characterized by a non-dimensional index I_1 combining both α and Δp [9]:

$$I_1 \equiv \frac{\alpha}{C_1} \frac{1}{\Delta p}\dagger$$
(4)
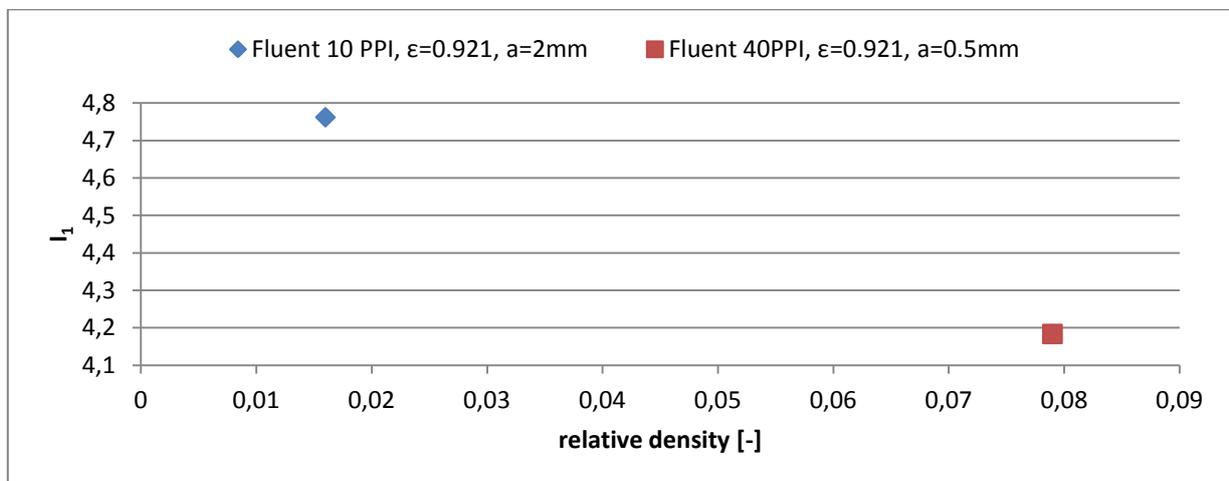
Fig. 12. The index of the foam performance as a function of relative density ρ=0.016(10 PPI) and 0.079(40 PPI).

Fig. 12 plots the index of the foam performance I_1, as a function of foam relative density ρ. Clearly there exists an optimum foam density which would maximize I_1. The cell size (a), does not affect ρ_{optim} but at a given density foam with larger cell size perform better than those having smaller cell size. If the weight of an heat exchanger is cause for concern in applications such as airborne equipments, the appropriate index for performance scaling may be taken as $I_2=I_1/\rho$. Here, the best performance is achieved by a foam with $\rho \rightarrow 0$ having the largest cell size [9].

In any heat exchanger design, the heat convection performance of the heat exchanger must be weighed against the energy required to operate the system, which is the pumping power in this configuration. In fig. 13 the optimal design is that which minimize the distance from the point to the origin of the plot [15].
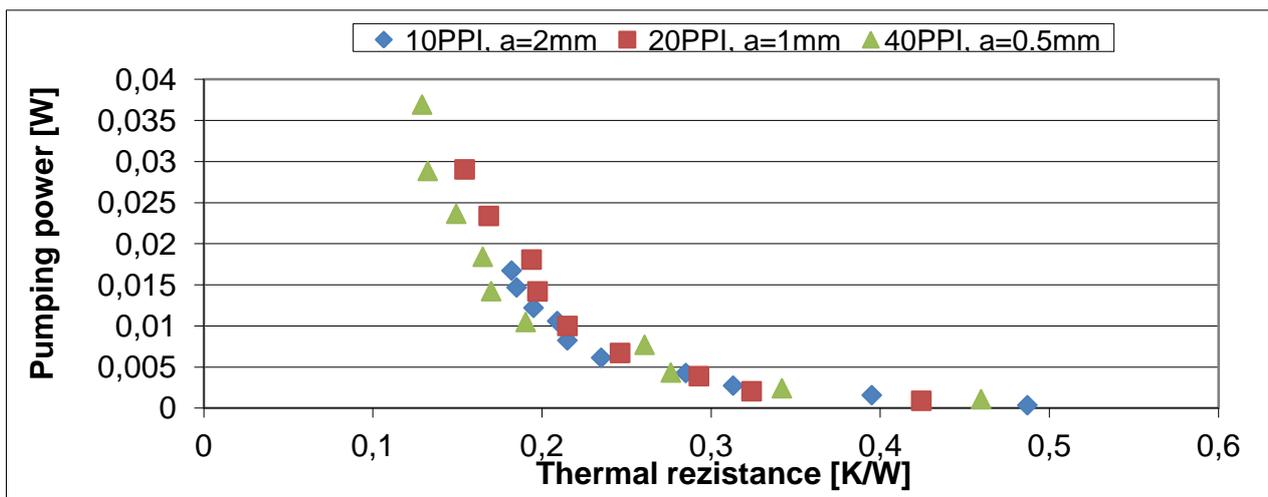
Fig. 13. Plot of the required pumping power against the corresponding thermal resistance.

This point was obtained by foam with 40PPI porous density and for a thermal resistance of 0.129 K/W. The metal foam heat exchangers decreased thermal resistance by nearly half when compared to currently used heat exchangers designed for the same application.

4. Conclusions

The simulated model is applied to evaluate the heat transfer efficiency of open-celled foams as heat sinks for power electronic components. The investigations were carried out for different types of open-cell porous aluminum having different geometrical characteristics: porosity, pore density and different flow rates of coolant. Through numerical simulation were identified optimal geometric characteristics necessary for this type of exchanger to achieve the most efficient cooling of electronic equipment that it serves, in terms of pressure drop as low as possible.

An interpretation of results was carried out, in order to apply the conclusions that result from the present research on the existing types of open cell porous aluminum heat exchangers and the design of new ones. Solutions for the overall heat transfer coefficient and pressure drop as functions of cell morphologies and other relevant heat transfer parameters are obtained, which can be used to guide the design of optimum cellular structures that maximize the heat transfer rate per unit pumping power (the energy expended driving the convective flow).

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