Natural convection in a vertical channel with open-cell foams

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Abstract. Open-cell foams are promising to enhance heat transfer in many applications, such as electronics cooling. Though heat transfer coefficients are low, natural convection in open-cell foams is used in low-end applications, as consumer electronics or switching devices. In this paper air natural convection in a vertical channel equipped with an open-cell foam, with a symmetric uniform heat flux boundary condition at the vertical sides, is investigated numerically. The foam is modeled as a continuous porous medium under the assumption of Local Thermal Nonequilibrium (LTNE) between solid and fluid phases. Mass, momentum and energy equations are solved numerically under the appropriate boundary conditions by employing a commercial finite element code. The velocity of the fluid and the temperature of the solid as well as the global heat transfer coefficient, for different thermo-physical properties and morphologies of the foam, and for various channel aspect ratios, are predicted.

Nomenclature

| Symbol | Description |
|--------|-------------|
| $u$    | Velocity vector (m/s) |
| $u, v$ | Velocity components (m/s) |
| $x, y$ | Cartesian coordinates (m) |
| $C_p$  | Heat capacity (J/kg K) |
| $CV$   | Coefficient of variation |
| $d_c$  | Cell size (m) |
| $d_s$  | Strut size (m) |
| $f$    | Inertial factor |
| $g$    | Acceleration due to the gravity (m/s²) |
| $G$    | Geometric function (Tab. 2) |
| $H$    | Channel height (m) |
| $h$    | Global heat transfer coefficient (W/m² K) |
| $h_c$  | Interfacial heat transfer coefficient (W/m² K) |
| $h_v$  | Volumetric heat transfer coefficient (W/m³ K) |
| $k$    | Thermal conductivity (W/m K) |
| $K$    | Permeability (m²) |
| $L$    | Channel half width (m) |
| $Nu_s$ | Strut Nusselt number |
| $p$    | Pressure (Pa) |
| $Pr$   | Prandtl number |
| $q$    | Heat flux (W/m²) |
| $Ra_f$ | Strut Rayleigh number |
| $S_v$  | Specific surface area (1/m) |
| $T$    | Temperature (K) |

Greek symbols

| Symbol | Description |
|--------|-------------|
| $\beta$ | Isobaric compressibility (1/K) |
| $\epsilon$ | Porosity |
| $\mu, \sigma$ | Mean and standard deviation |
| $\mu_f$ | Dynamic viscosity (Pa s) |
| $\rho$ | Density (kg/m³) |

Subscripts

| Symbol | Description |
|--------|-------------|
| $0$    | Ambient |
| $d_c$  | Cell size |
| $eff$  | Effective |
| $f$    | Fluid |
| $s$    | Solid |

Superscripts

| Symbol | Description |
|--------|-------------|
| $*$    | Dimensionless |
1. Introduction
Metal foams are porous materials made up by cells which repeat through the space with a certain regularity. Depending on if the pores are open or not, it is possible to distinguish between open-cell foams and closed-cell foams. Open cell foams are used in many applications, such as bearing structures or acoustic shields [1]. Because of their tortuosity, high heat transfer area and effective thermal conductivity, open cell foams are also employed to enhance heat transfer in heat sinks for electronics [2-4], volumetric solar air receivers [5-6], porous burners [7-8], thermal energy storage [9-11].

Natural convection plays a primary role in many applications. A metal foam heat sink for natural convection in electronics was proposed by Bhattacharya and Mahajan [3]. They investigated both unfinned and finned metal foam heat sinks, with foam layers between fins. Experiments, run in both vertical and horizontal heat sinks, showed that finned metal foam provided better heat transfer performances compared to conventional heat sinks and to unfinned heat sinks; the larger the number of fins the higher the heat transfer enhancement. Better performances were exhibited by lower PPIs foams. Numerical and experimental results for a metal foam heated from below were presented by Phanikumar and Mahajan [12]. The numerical model was written under the assumption of Local Thermal Non-Equilibrium (LTNE), with local differences in the solid and fluid temperatures; governing equations are averaged over a Representative Elementary Volume (REV) [13]. Various pore sizes, porosities and metal/liquid combinations were investigated. Results showed that a LTNE model is necessary to investigate natural convection heat transfer in metal foams. Zhao et al. [14] investigated experimentally and numerically natural convection in steel alloy (FeCrAlY) open-cell foams. The effects of natural convection on the effective thermal conductivity of the foam were evaluated experimentally, under both vacuum and ambient conditions. A nearly 50% contribution of natural convection to the effective thermal conductivity was found. Qu et al. [15] carried out experiments on copper foams with various PPIs and porosities, for different inclinations. Results showed that the largest averaged Nusselt number was achieved with an about 60°-75° angle above the horizontal. Experiments carried out by De Schamphereleire et al. [16] showed that the performance of open-cell foams under natural convection was mostly affected by the height, pore density and bonding method of the foam. The combined numerical and experimental analysis presented by De Schamphereleire et. al [17] showed that radiation might also affect heat transfer. The numerical analysis of natural convection in a horizontal channel partially filled with a foam and heated from above, carried out both with LTE and LTNE by Buonomo et al. [18, 19], showed that in both cases low PPIs foams enhance the heat transfer. Two lateral reservoirs to stabilize flow driven by natural convection were also considered in [18].

In this paper, air natural convection in a vertical channel, equipped with an open-cell foam, with a uniform heat flux boundary condition at the vertical sides, is investigated numerically. Governing equations are written under the Volume Averaging Technique (VAT) [13] and the LTNE between the two phases assumption. Two reservoirs are employed to model the flow entrance and exit conditions. The problem is numerically solved with a finite element code. The velocity of the air and the solid phase temperature as well as the global heat transfer coefficient, for different channel sizes and foam characteristics and morphologies, are predicted.

2. Mathematical model
2.1. Geometry and governing equations
The vertical foam channel, H high and 2L wide, is sketched in figure 1a. Governing mass, momentum and energy equations for both the fluid and solid phases, are averaged over a REV of the foam.

Under the steady state, laminar flow, no thermal dispersion and tortuosity, thermophysical properties independent of the temperature and direction, Boussinesq approximation for natural convection assumptions we can write

\[ \nabla \cdot \langle \mathbf{u} \rangle = 0 \] (1)

\[ \frac{\rho}{\varepsilon^2} \nabla \cdot \langle \mathbf{u} \rangle = -\nabla \langle p \rangle + \frac{\mu_f}{\varepsilon} \nabla^2 \langle \mathbf{u} \rangle - \frac{\mu_f}{K} \langle \mathbf{u} \rangle - \frac{\rho g \beta}{\sqrt{K}} \langle T_f \rangle \langle \mathbf{u} \rangle + \rho g \beta \left( \langle T_f \rangle - T_0 \right) \] (2)
Air is assumed to be an ideal gas and, therefore, its isobaric compressibility is \( \beta = 1/T_0 \). For the reservoirs, the same equations are employed except that for the solid phase, by assuming \( K \rightarrow \infty \), \( \varepsilon = 1 \) and \( h_v = 0 \).

The investigated computational domain, \( H \) high and \( L \) wide because of the symmetry, is sketched in figure 1b. The two reservoirs, \( H \) high and \( 4L \) wide, upstream and downstream of the channel allow to simulate the thermal and fluid dynamic behavior far away from the inflow and outflow regions [20].

### 2.2. Closing coefficients, boundary conditions and numerical modeling

Governing equations (1) - (4) require the so-called closing coefficients to be determined. For a porous material they are the permeability, \( K \), the inertial factor, \( f \), the thermal conductivities, \( k_{eff,f} \) and \( k_{eff,s} \), the volumetric heat transfer coefficient, \( h_v \). Most of them depend on the cell size, \( d_c \). According to Gibson and Ashby [21], the cell is the unit that periodically repeats through the space in a foam. In the following reference will be made to the cell size as the diameter of the largest sphere inscribed in the cell [22], which is quite similar [23] to the hydraulic diameter proposed by Schlegel et al. [24].

As to authors' knowledge, no correlations between the cell size, \( d_c \), and Pores Per Inch (PPI) are available in the literature. Cell sizes of foams with different PPIs and the same porosity at each PPI were measured by Bhattacharya and Mahajan [2]; their values are reported in the left side of table 1. They have been used in the present work to derive correlations between cell size and PPI. In the above derivation we assumed the cell size to be independent of the porosity. In order to validate this assumption the average, \( \mu_{d_c} \), the standard, \( \sigma_{d_c} \), deviation as well as the coefficient of variation, \( CV \), were derived and reported in the left side of table 1. The assumption is reliable since the maximum value of the coefficient of variation is about 5%. Starting from the average cell size for each PPI, correlations were derived, which are reported in the right side of table 1. In the following cell sizes and PPI are correlated by the function with the highest coefficient of determination, \( R^2 = 0.987 \),

\[
d_c = \left[- 0.921 \ln(\text{PPI}) + 5.3564 \right] 10^{-3}
\]

The effective thermal conductivity of the fluid phase, \( k_{eff,f} \), is assumed as the porosity-weighted thermal conductivity of the fluid phase, \( k_{eff,f} = k_f \varepsilon \), as to the effective thermal conductivity of the solid phase, \( k_{eff,s} \), one can use the correlation \( k_{eff,s} = 0.313(1 - \varepsilon) k_s \) proposed by Iasiello et al. [25], where thermo-physical properties were assumed to be independent of the direction. The other porous media
coefficients needed to close equations (1) – (4) are resumed in table 2. Correlations for $K, f, d/d_c$, $G$ and $S_c$ are taken from Calmidi [26]. To the authors’ knowledge, no direct correlations for the interfacial heat transfer coefficient, $h_s$, are available for open-cell foams. However, if foam struts are approximated as a bank of staggered cylinders, reference can be made to a strut Nusselt number, $N_u_s$, given by the correlation for horizontal cylinders with a uniform boundary heat flux proposed by Churchill and Chu [27]. Boundary conditions of the 2D problem are presented in table 3. The imposed heat flux is $|q|=1000 \text{ W/m}^2$.

The problem is solved by employing 2D rectangular coordinates and accounting for symmetry conditions in both fluid-dynamics and heat transfer. Open boundary conditions with a uniform ambient temperature are employed at the ends of the reservoirs and a uniform heat flux boundary condition is employed at the wall. Mass, temperature and heat flux continuity is imposed at the free fluid/foam interfaces. The viscous shear stress continuity is assumed in the momentum equation. Different boundary conditions were imposed by Alazmi and Vafai [28], which provide similar results for small Darcy numbers (less than $10^{-4}$) as those accounted for in the present study.

Equations were solved by means of the finite element commercial code COMSOL Multiphysics, with the fully-coupled direct solver PARDISO. A boundary layer triangular mesh with about 150,000 elements was built up. Solution grid independence was verified with larger number of elements which showed negligible variations of the temperature distribution through the foam domain. Second-order polynomials were employed for the finite element solution; streamline and crosswind diffusion stabilization was used in fluid flow equations. A $10^{-4}$ RMS relative tolerance of criteria was set up.

### 3. Results and comments

The dimensionless components of the fluid velocity vector, $u^*$ and $v^*$, the dimensionless solid temperature, $\Delta T_s^*$, the global heat transfer coefficient, $h$, reported in table 4 were predicted; results are presented in this section.

The line averaged temperatures of the solid, $\langle T_s \rangle$, and the fluid, $\langle T_f \rangle$, phases are defined as reported in equations (6a) and (6b), respectively.

### Table 1. Correlations between cell size, $d_c$ (mm), and PPI.

| PPI | 5 | 10 | 20 | 40 | Correlations | $R^2$ |
|-----|---|----|----|----|-------------|------|
| $d_c$ | 4.02 | 3.13 | 2.70 | 1.90 | $d_c 10^3 = -0.921 \ln(\text{PPI}) + 5.3564$ | 0.987 |
|      | 3.80 | 3.28 | 2.90 | 2.02 | $d_c 10^3 = 4.0324 e^{-0.019 \text{ PPI}}$ | 0.969 |
|      | 4.00 | 3.20 | 2.58 | 1.80 | $d_c 10^3 = 0.0013 \text{ PPI}^2 - 0.1101 \text{ PPI} + 3.4083$ | 0.969 |
|      | 3.90 | 3.10 | 2.80 | 1.80 | $d_c 10^3 = -0.0515 \text{ PPI} + 3.8817$ | 0.922 |
|      | 3.80 | 2.96 | 2.70 | 1.98 | $d_c 10^3 = 0.00212 |d_c| 0.00073$ | 0.987 |
|      |      |      | 2.60 | 2.00 | $d_c 10^3 = 0.0013 |d_c| 0.00073$ | 0.987 |

### Table 2. Porous media closing coefficients.

| Equation (2) | Equations (3) and (4) |
|--------------|-----------------------|
| $K = 0.00073 \left(1 - \varepsilon\right)^{-0.224} \left(d_s/d_c\right)^{-1.11}$ | $h_s = h_c S_c = \frac{N_u_s k_f}{d_s}$ |
| $f = 0.00212 \left(1 - \varepsilon\right)^{-0.132} \left(d_s/d_c\right)^{-1.63}$ | $N_u_s = 0.36 + 0.521 \left(Ra_s\right)^{0.25} \left[1 + 0.442/Pr\right]^{9/16} \left[1 - 0.442/Pr\right]^{1/4}$ |
| $d_s/d_c = 1.18 \left(1/G\right)^{1/2}$ | $Ra_s = g \beta |q| d_f \rho^2 C_p h C_s \nu^2$ |
| $G = 1 - \varepsilon^{-0.04}$ | $S_c = G (\frac{\pi}{5.9 d_c}) (d_s/d_c)$ |
Table 3. Boundary conditions of the problem.

| Cartesian coordinates | Momentum | Energy |
|-----------------------|----------|--------|
| \( x = -3L \) \( -H \leq y \leq 0 \) \( H \leq y \leq 2H \) | \( p - 2 \mu \frac{\partial u}{\partial x} = 0 \) | \( T = T_0 = 20^\circ C \) |
| \(-3L < x < L \) \( y = -H \) \( y = 2H \) | \( p - 2 \mu \frac{\partial v}{\partial y} = 0 \) | \( T = T_0 = 20^\circ C \) |
| \(-3L < x < 0 \) \( y = 0 \) \( y = H \) | \( u = 0, \ v = 0 \) | \( \partial T / \partial y = 0 \) |
| \( x = 0 \) \( 0 \leq y \leq H \) | \( u = 0, \ v = 0 \) | \( |\mathbf{q}| = -k_{eff,f} \partial \langle T_f \rangle / \partial x + k_{eff,f} \partial \langle T_f \rangle / \partial x = 1000 \text{ W/m}^2 \) |
| \( x = L \) \( 0 \leq y \leq H \) | \( u = 0, \ \partial v / \partial x = 0 \) | \( \partial \langle T_f \rangle / \partial x = \partial \langle T_f \rangle / \partial x = 0 \) |
| \( x = L \) \( -H \leq y \leq 0 \) \( H \leq y \leq 2H \) | \( u = 0, \ \partial v / \partial x = 0 \) | \( \partial T_f / \partial x = 0 \) |
| \( 0 < x < L \) \( y = 0 \) \( y = H \) | \( u|_j = u|_{j \rightarrow f}, \ v|_j = v|_{j \rightarrow f} \) | \( \langle T_f \rangle = T_f \) |

\[ a) \ \langle T_f \rangle_{H,L} = \frac{1}{H} \int_0^H \langle T_f \rangle \ dy \quad b) \ \langle T_f \rangle_{H,L} = \frac{1}{H} \int_0^H \langle T_f \rangle \ dx \quad (6) \]

In order to underline the usefulness of the LTNE model, differences between the temperatures of the solid and fluid phases, for \( \text{PPI} = 5, 10, 20, 40, \varepsilon = 0.90, H/L = 1.50, k_f/k_s = 380 \), are presented in figure 2, which shows that the larger the PPI the lower the temperature differences. This implies that the volumetric heat transfer coefficient, \( h_v \), increases with PPI; thus the heat transfer between the two phases is enhanced and a local thermal equilibrium for high PPIs may be assumed.

Dimensionless velocity \( v^* \) as a function of \( x^* \), for \( H/L = 3.0, \text{PPI} = 5, \varepsilon = 0.85 \) and \( 0.95, y^* = 0.0, 0.50, 1.0, k_f/k_s = 3800 \), is presented in figure 3. The figure shows that the velocity profiles are practically independent of the \( y^* \) coordinate. One can also notice that there is no channeling effect in the foam. The figure also exhibits lower velocities at the smaller porosity, since the smaller the porosity the larger the resistance to the fluid flow through the foam. The afore said buoyancy effect is far larger at the larger porosity where higher temperatures are attained since the fraction of metal in the porous domain is smaller.

Dimensionless velocity \( v^* \) as a function of \( x^* \), for \( H/L = 3.0, \text{PPI} = 5, \varepsilon = 0.90, y^* = 0.0, 0.50, 1.0, k_f/k_s = 38, 3800 \), is presented in figure 4. The figure highlights a fair dependence of the velocity on the \( y^* \) coordinate. It also shows the marked channeling effect described in Szekely and Poveromo [29] and in Vafai [30], which implies a flow maldistribution in the proximity of the confining walls of the

Table 4. Process parameters.

| Geometry | Velocity | Temperature | Convection |
|----------|----------|-------------|------------|
| \( x^* = \frac{x}{L} \) | \( u^* = \frac{u}{u} \) | \( \Delta T_x^* = \frac{\langle T_f \rangle^* - T_0}{|\mathbf{q}| / k_{eff,f}} \) | \( h = \frac{\langle T_f \rangle^*}{|\mathbf{q}| / k_{eff,f}} \) |
| \( y^* = \frac{y}{H} \) | \( v^* = \frac{v}{v} \) | \( \Delta T_f^* = \frac{\langle T_f \rangle^* - T_0}{|\mathbf{q}| / k_{eff,f}} \) | \( \langle T_f \rangle^*_{H,L} - \langle T_f \rangle^*_{H,L} \) |
Figure 2. Differences between the temperature of the solid and fluid phases, for $\varepsilon = 0.90$, $H/L = 1.50$, $k_s/k_f = 380$ and different PPIs.

A dimensionless velocity at $x^* = 0.10$ about 3 times larger than that at $x^* = 1.0$ is reported in the figure.

Dimensionless solid phase temperature $\Delta T^*_s$ as a function of $y^*$, for $H/L = 2.0$, PPI = 5, $\varepsilon = 0.85$, 0.90, 0.95, $k_s/k_f = 3800$, is presented in figure 5. The figure shows that the lower the porosity the higher the dimensionless temperature of the solid phase, likely because lower velocity at lower porosity decreases heat transfer between the two phases.

Dimensionless solid phase temperature $\Delta T^*_s$ as a function of $y^*$, for $H/L = 3.0$, PPI = 5, 10, 20, 40, $\varepsilon = 0.90$, $k_s/k_f = 3800$, is presented in figure 6. The figure shows that increasing PPI increases the solid phase temperature, because of the increasing resistance to the fluid flow.

The global heat transfer coefficient $h$ as a function of either $\varepsilon$ and $k_s/k_f$, for $H/L = 2.0$ and PPI = 5, or $H/L$ and PPI, for $\varepsilon = 0.90$ and $k_s/k_f = 3800$, is presented in figures 7a and 7b, respectively. Figure 7a points out the marked dependence of heat transfer on the $k_s/k_f$ ratio, which ranges between that of a SiC-air and that of a metal-air foam. The figure also shows that increasing porosity decreases the global heat transfer coefficient, because of the decrease in the specific surface area of the foam.
The figure 7b exhibits a strong dependence of the global heat transfer coefficient on both the channel aspect ratio and the PPIs. This because at equal height, the wider the channel the thicker is the boundary layer, making heat transfer coefficients smaller as for channels; on the other hand, higher PPIs means higher specific surface area and then higher heat transfer coefficients, as already observed for porosity dependence.

4. Conclusions

Air natural convection in a vertical channel, equipped with an open-cell foam, with a symmetric uniform heat flux boundary condition at the vertical sides, has been investigated numerically with a commercial finite element code. The effects of the porosity and Pores Per Inch, the ratio of the solid to the fluid thermal conductivity and channel aspect ratio of the foam on the velocity of the air, on the temperature of the solid phase and on the global heat transfer coefficient were investigated.

In all investigated cases the dimensionless vertical component of the air velocity in the foam was fairly independent of the dimensionless vertical coordinate. A marked channelling effect in the proximity of the confining walls of the porous medium was exhibited at the lower values of the solid to fluid phases thermal conductivity ratio. The dimensionless temperature of the solid phase exhibits a fair increase both at decreasing porosities, since lower velocity at lower porosity decreases heat transfer between the two phases, and at larger PPIs, because of the increasing resistance to the fluid flow.

Figure 5. Dimensionless solid phase temperature $\Delta T_s^*$ vs. $y^*$, for $H/L = 2.0$, PPI = 5, $k_s/k_f = 3800$, $\varepsilon = 0.85, 0.90, 0.95$.

Figure 6. Dimensionless solid phase temperature $\Delta T_s^*$ vs. $y^*$, for $H/L = 3.0$, PPI = 5, 10, 20, 40, $k_s/k_f = 3800$, $\varepsilon = 0.90$.

Figure 7. Global heat transfer coefficient vs.: a) $\varepsilon$ (with $k_s/k_f = 100$) and $k_s/k_f$ (with $\varepsilon = 0.85$), for $H/L = 2.0$ and PPI = 5; b) $H/L$ (with 20 PPI) and PPI (with $H/L = 3.0$), for $\varepsilon = 0.90$ and $k_s/k_f = 3800$. 
The global heat transfer coefficient strongly increases at decreasing porosities, because of the decrease in the specific surface area, and at increasing PPI for the same reason; it also rapidly decreases with the channel aspect ratio because of the different boundary layer developing.

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