Numerical investigation on natural convection in horizontal channel partially filled with aluminium foam and heated from above

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Abstract. Natural convection gets a great attention for its importance in many thermal engineering applications, such as cooling of electronic components and devices, chemical vapor deposition systems and solar energy systems. In this work, a numerical investigation on steady state natural convection in a horizontal channel partially filled with a porous medium and heated at uniform heat flux from above is carried out. A three-dimensional model is realized and solved by means of the ANSYS-FLUENT code. The computational domain is made up of the principal channel and two lateral extended reservoirs at the open vertical sections. Furthermore, a porous plate is considered near the upper heated plate and the aluminium foam has different values of PPI. The numerical simulations are performed with working fluid air. Different values of assigned wall heat flux at top surface are considered and the configuration of the channel partially filled with metal foam is compared to the configuration without foam. Results are presented in terms of velocity and temperature fields, and both temperature and velocity profiles at different significant sections are shown. Results show that the use of metal foams, with low values of PPI, promotes the cooling of the heated wall and it causes a reduction of Nusselt Number values with high values of PPI.

1. Introduction

Natural convection has a great importance in several practical applications, such as cooling of electronic components and devices, chemical vapor deposition systems and solar energy systems, solar energy system, drying process, fire safety research and safety in nuclear reactors. In this work, a numerical investigation on steady state natural convection in a horizontal channel partially filled with a porous medium and heated at uniform heat flux from above is carried out. The flow motion is strongly affected by the location of the heated surface, positioned on the upper or the lower wall of the horizontal channel [1-8]. Secondary flows are induced by buoyancy force due to the heating of the cavity walls, hence the local heat transfer increases [2-5, 7]. The onset point of the secondary flows delineates the region after which the two-dimensional laminar flow becomes three-dimensional and a transition motion of the flow from laminar to turbulent is observed [6-7]. The main flow shows a “C” loop behaviour very close to the flow inside open-ended channels and open cavities [6, 9]. The main flaw in natural convection is caused by the low heat exchanged between flow and
walls, and one of the technique to enhance the heat transfer is expanding the exchange surfaces. Porous media provided with high thermal conductivity, like metal foams, are an adequate method of heat transfer enhancement due to their large surface area to volume ratio and to intense mixing of the flow [10]. In [8, 11, 12], natural convection in high porosity metal foams heated from below is studied numerically and experimentally. Aluminum foam samples of different pore sizes and porosities were used to show the effects of metal foam geometry on heat transfer. The use of metal foams entails significant enhancements in heat transfer and metal foam disk is considered in [3, 12] to evaluate the effect of natural convection on the effective thermal conductivity. In [13], the design of aluminum foam was experimentally built and the calibration was done comparing the results of a flat plate with literature data and the agreement resulted excellent. The investigated foams had a pore density of 10 and 20 PPI and the bonding of the foam was performed employing a single epoxy or via brazing. A metal foam heat sink with rectangular section was placed in a large Plexiglas housing and a two dimensional model was considered for the experimental tests and for the numerical simulations [14]. A numerical investigation on natural convection for the idealized regular metal blocks is realized in [15] to reveal the detailed flow and the thermal behaviors at the pore level using the lattice Boltzmann method. Results show that the heat transfer enhances increasing the pore density and it weakens increasing the porosity. An experimental investigation on natural convection heat transfer in superposed metal foams with internal heat sources is studied in [16-18]. Numerical results of laminar fully developed natural convection in an inclined channel partially filled with metal foam are studied in [19]. An experimental investigation of air natural convection in horizontally-positioned copper metallic foams with open cells is performed in [20]: results show that the porosity influence on the heat transfer performance is more remarkable when the pore density is higher, and natural convection in the copper foam weakens its thermal resistance and enhances its heat transfer performance. The natural convection on metallic foam-sintered plate at different inclination angles is experimentally studied in [21]. The sintered foam surface with lower porosity and pore density results more suitable for heat transfer enhancement. Studies on partially opened cavities filled with porous media are reported in [17-20] and investigations on partially filled horizontal channels with porous media are reported in [22, 25].

In this work, a numerical investigation on steady state natural convection in a horizontal channel partially filled with a porous medium and heated from above is carried out. A three-dimensional model is realized and solved by means of the ANSYS-FLUENT code. The computational domain is made up of the principal channel and two lateral extended reservoirs at the open vertical sections. Furthermore, a porous plate is considered near the upper heated plate and it fills the channel partially. The numerical simulations are performed with working fluid air. Different values of assigned wall heat flux at top surface are considered and the configuration of the channel partially filled with metal foam is compared to the configuration without foam.

![Figure 1. Physical Domain.](image)
2. Mathematical Model

The analyzed system is composed by two horizontal parallel plates, with the upper plate heated at uniform heat flux, $\dot{q}$, and the lower insulated plate. As shown in Figure 1, the distance between the two horizontal plates is $b$, the plate length is $L$, as the dimension of the external reservoir, and the plate width is $W$. The dimension values of the model are reported in Table 1.

| Table 1. Geometric values of the model. |
|-----------------|-----------------|
| $b$             | 40 mm           |
| $b/2$           | 20 mm           |
| $L$             | 400 mm          |
| $W$             | 475 mm          |

The natural convective flow between the two horizontal plates is considered to be laminar, incompressible, and two-dimensional. Under steady-state condition, the governing equations, with constant thermophysical properties and with the Boussinesq approximation, are in conservative form [22].

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}
\]

\[
\frac{\partial (u^2)}{\partial x} + \frac{\partial (uv)}{\partial y} = - \frac{1}{\rho} \frac{\partial p}{\partial x} + v \nabla^2 u \tag{2}
\]

\[
\frac{\partial (uv)}{\partial x} + \frac{\partial (v^2)}{\partial y} = - \frac{1}{\rho} \frac{\partial p}{\partial y} + u \nabla^2 v + g \beta (T - T_\infty) \tag{3}
\]

\[
\frac{\partial (uT)}{\partial x} + \frac{\partial (vT)}{\partial y} = a \nabla^2 T \tag{4}
\]

In Equation (4), the terms of dissipation and the total derivative of pressure are neglected, since this is usually done in natural convection phenomena when temperature gradients are relatively weak [22]. These equations are made nondimensional by introducing the following variables.

\[
X = \frac{x}{b}, \quad Y = \frac{y}{b}, \quad U = \frac{ub}{v}, \quad V = \frac{vb}{v}, \quad P = \frac{(p - p_\infty)b^2}{\rho v^2} \tag{5}
\]

\[
\theta = \frac{(T - T_\infty)}{\dot{q}b/k}, \quad Gr = \frac{g \beta q b^4}{k v^2}, \quad Pr = \frac{v}{a}, \quad Ra = Gr Pr
\]

An enlarged computational domain of finite extension has been employed in this investigation to simulate the free-stream conditions of the flow far away from the region of the thermal disturbance induced by the heated plate, as suggested in reference [1, 4, 13]. The local convective heat transfer over the heated plate is described by the local Nusselt number, defined as:

\[
Nu(x) = \frac{h(x)b}{k} = \frac{1}{\theta_{w,upper}(x)} \tag{6}
\]

The corresponding average value can be written as
3. The numerical solution

The numerical solutions of the governing equations, reminding that they represent a set of coupled, non-linear, partial differential equations, are carried out using the commercial code Ansys-Fluent 12.2 [26].

The operating conditions are set as follow: the operating temperature is equal to 300 K and the pressure is equal to 1 bar. In the analysis a uniform heat flux is applied on the top wall of the channel and it is equal to 220 W/m² and to 350 W/m². Furthermore, a porous plate is considered near the upper heated plate and it has a thickness equal to 20 mm whereas the length and the width are the same of the channel. The aluminum foam has 10, 20 and 30 PPI and its porosity is equal to 93%.

Different structured mesh distributions were tested to accomplish the grid-independence analysis, paying attention, in particular, to regions characterized to unexpected temperature and velocity gradients. The analysis is carried out in steady regime and considering the metal foam with 10 PPI. The evaluated parameters are the Nusselt number and the average temperature on the heated plate. Three simulation are performed to analyze the sensitivity of the meshes employed. Richardson's extrapolation equation allows to estimate a reference value of a generic quantity, comparing two successive configurations related to the considered meshes. The percentage error is evaluated and it is chosen to compare the results. In Table 2, the values are reported.

The choice of the mesh, used for the subsequent simulations, was performed looking for the right compromise between the accuracy of the simulation and calculation time. Based on these two parameters, the Mesh 2 is chosen. It presents a slightly higher error than the Mesh 3 but, with respect to the latter, presents a lower calculation times.

4. Results and discussions

The objective of this study is to analyze natural convection in a horizontal channel partially filled with a porous medium and heated at uniform heat flux from above. Different values of assigned wall heat flux at top surface are considered and the configuration of the channel partially filled with metal foam is compared to the configuration without foam.

![Diagram](image)

(a) Section 1, in the plane z = 0 mm. (b) Section 2, in the plane y = 20 mm.

**Figure 2. Analyzed sections.**
Results are presented in terms of velocity and temperature fields, referring to the planes shown in Figure 2. Furthermore, both temperature and velocity profiles are shown at different three significant sections (Figure 3), to obtain a description of the natural convection inside the open-ended cavity.

![Figure 3](image1.jpg)  
(a) Line 1.  (b) Line 2.  (c) Line 3.  

**Figure 3.** Analyzed sections.

Figures 4 and 5 show temperature fields in sections 1 and 2 with $\dot{q} = 220 \text{ W/m}^2$, comparing the configurations with and without foam. In Figure 1, temperature values are higher in the metal foam, where the convective heat transfer is more difficult. On the heated plate, temperature values are higher in the central zone and they decrease in proximity of the edges in all configurations, as shown in Figure 5. Temperature values increase for higher PPI values.

![Figure 4](image2.jpg)  
(a) Clean configuration.  (b) With foam, 10 PPI.  
(c) With foam, 20 PPI.  (d) With foam, 30 PPI.  

**Figure 4.** Temperature field, Section 1, $\dot{q} = 220 \text{ W/m}^2$. 
Figures 6, 7 and 8 show temperature fields in Sections 1 and 2, considering different values of PPI foam, and varying the heat flux value.

Figure 6 shows the effect of the foam with 10 PPI: temperature values are lower at the exterior sections of the foam, then they rise in the centre part of it. In the lower part of the channel, far from the heated plate and without foam, temperature values are close to ambient temperature value.

As shown in Figures 7 and 8, the qualitative trend of temperature values is the same with a 20 PPI or 30 PPI foam, but values are higher, thanks to the foam effect. In each configuration, values are higher when the heat flux applied on the top wall of the channel $\dot{q}$ is equal to 350 W/m$^2$.

The maximum temperature value is observed in Figure 8, when heat flux and PPI foam are higher.
(a) Section 1, $\dot{q} = 220 \text{ W/m}^2$.

(b) Section 2, $\dot{q} = 220 \text{ W/m}^2$.

(c) Section 1, $\dot{q} = 350 \text{ W/m}^2$.

(d) Section 2, $\dot{q} = 350 \text{ W/m}^2$.

Figure 6. Temperature field, Section 1 and 2, with foam 10 PPI.

(a) Section 1, $\dot{q} = 220 \text{ W/m}^2$.

(b) Section 2, $\dot{q} = 220 \text{ W/m}^2$. 
Section 1, $q = 350$ W/m$^2$.

Section 2, $q = 350$ W/m$^2$.

Figure 7. Temperature field, Section 1 and 2, with foam 20 PPI.

Section 1, $q = 220$ W/m$^2$.

Section 2, $q = 220$ W/m$^2$.

Figure 8. Temperature field, Section 1 and 2, with foam 30 PPI.
Figure 9. Velocity field, Section 1, $\dot{q} = 220 \text{ W/m}^2$.

Figure 9 shows velocity fields in sections 1 and 2 with $\dot{q} = 220 \text{ W/m}^2$, comparing the configurations with and without foam. In clean configuration, convective motions are not obstructed. Due to the foam, air gets slower and velocity values decrease.

Figure 10. Temperature profile, $\dot{q} = 220 \text{ W/m}^2$.

Figure 10 shows temperature profiles along Lines 1, 2 and 3, considering the heat flux equal to 220 W/m$^2$, and comparing the configurations with and without foam. In Figure 10(a), the maximum value is reached in the middle of Line 1 and temperature values are lower at the far end, due to the contact with a colder air. Considering foams with 20 and 30 PPI, temperature values are higher; the heat
transfer improves with foam 10 PPI. In Figure 10(b), when the channel is clean, temperature values are close to

(a) Nusselt number.  
(b) Heat transfer coefficient.

Figure 11. Profiles of Nusselt number and heat transfer coefficient along Line 1, $q = 220 \text{ W/m}^2$.

(a) Line 1.  
(b) Line 2.  
(c) Line 3.

Figure 12. Temperature profile, foam 10 PPI.

(a) Nusselt number.  
(b) Heat transfer coefficient.

Figure 13. Profiles of Nusselt number and heat transfer coefficient along Line 1, foam 10 PPI.
ambient temperature, then they increase in proximity of the heated plate. Temperature values are higher in presence of the foam. In Figure 10(c), in the clean configuration, temperature values are constant at the first, then they grow rapidly. When the foam partially fill the channel, temperature values grows rapidly at the fist, then they become quite constant.

Profiles of Nusselt number and heat transfer coefficient present the same qualitative trend, as shown in Figure 11. Values are lower in the middle of the Line 1 in all configurations.

Figure 12 shows temperature profile along Lines 1, 2 and 3 varying the heat flux and considering the foam with 10 PPI. In Figure 12(a), temperature values grow with heat flux and are higher in the middle of the Line 1. In Figures 12(b) and 12(c), temperature values increase in the lower part of the channel, then they are quite constant in the metal foam.

In Figure 13, the values of Nusselt number and heat transfer coefficient are higher when $\dot{q}$ is equal to 350 W/m² and they decrease in the middle of Line 1.

5. Conclusions

In this work, a numerical investigation on steady state natural convection in a horizontal channel partially filled with a porous medium and heated at uniform heat flux from above is carried out. A three-dimensional model is realized and a porous plate is considered near the upper heated plate. Different values of assigned wall heat flux at top surface are considered and the configuration of the channel partially filled with metal foam is compared to the configuration without foam. Results are presented in terms of velocity and temperature fields, and both temperature and velocity profiles at different significant sections are shown, to obtain a description of the natural convection inside the open-ended cavity. Finally, Average Nusselt number values are evaluated. Results show that the use of metal foams with low values of PPI promotes the cooling of the heated wall. The presence of the metal foam causes a reduction of Nusselt Number values and it is more evident with high values of PPI. The use of metal foam deteriorates the heat transfer for higher values of PPI, because of the reduction of the volume in which convective motion develops. A foam with 10 PPI increases the heat transfer and temperature values on the heated wall are lower than in clean configuration.

6. Nomenclature

- $a$ thermal diffusivity, m²/s
- $b$ horizontal channel gap, m
- $g$ acceleration due to the gravity, m/s²
- $Gr$ Grashof number
- $h(x)$ local convective coefficient, W/m²K
- $k$ thermal conductivity, W/mK
- $L$ plate length, m
- $L_x$ length of the reservoir, m
- $L_y$ height of the reservoir, m
- $L_z$ width of the reservoir, m
- $Nu(x)$ local Nusselt number
- $Nu$ average Nusselt number
- $P$ pressure, Pa
- $P$ dimensionless pressure
- $Pr$ Prandtl number
- $\dot{q}$ heat flux, W/m²
Ra Rayleigh number  
$Ra^*$ modified Rayleigh number, $Ra(b=L)$  
$T$ temperature, K  
$u, v, w$ velocity components along x, y and z, m/s  
$U, V, W$ dimensionless velocity components  
$x, y, z$ Cartesian coordinates, m  
$X, Y, Z$ dimensionless Cartesian coordinates,  
$B$ volumetric coefficient of expansion, 1/K  
$\vartheta$ dimensionless temperature  
$\nu$ kinematic viscosity, m$^2$/s  
$\rho$ density, kg/m$^3$  

Subscripts  
$\infty$ free stream condition  
lower lower plate  
max maximum value  
upper upper plate  
$w$ wall

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