Investigation on Thermal and Fluid Dynamic Behaviors in Mixed Convection in Horizontal Channels with Aluminum Foam and Heated from Below

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Abstract. In this paper, mixed convection in a horizontal channel partially filled with a porous medium and the lower wall heated at uniform heat flux is studied experimentally and numerically. A simplified two-dimensional problem is modelled and solved numerically. The domain is made of a principal channel and two channels with adiabatic walls, one upstream and the other one downstream the principal channel. The heated wall temperature profiles as a function of the Ri values are presented. Average Nusselt numbers are evaluated. The experimental test section is made up of a horizontal wall and a parallel adiabatic wall. The distance between the horizontal walls is equal to 40 mm. The porous medium is an aluminium foam and it is placed over the heated lower wall. The porous plate has a thickness equal to 20 mm. The aluminium foam has 10, 20 and 40 PPI. The experiments are performed with working fluid air. The Reynolds numbers investigated are between 5.0 and 250, these being in the laminar regime. The Richardson number, Ri , holds values in the range 1 - 2000. Results in terms of wall temperature profiles, local and average Nusselt numbers are presented for different Reynolds and Rayleigh number values. Some comparison between experimental and numerical results are accomplished.

1 Introduction

Mixed convection has a great importance in practical applications in various modern systems, such as electronic cooling, chemical vapor deposition, fire research, solar and geothermal energy plants and flat plate heat exchangers, as indicated in [1,2]. The use of a porous medium in such applications gets a great importance for promoting the heat transfer with high efficacy and many investigations have been carried out on metal foams. For example, Barletta et al. [3], investigate the transition to absolute instability in a porous layer with horizontal throughflow. The importance of this analysis is due to the possible experimental failure to detect growing perturbations which are localized in space and which may be convected away by the throughflow. Ozgen et al. [4] examined by using the finite difference method for different values of the parameters observing that the Rayleigh

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and Peclet numbers have an effect on flow and temperature fields. In particular, in mixed convection in horizontal parallel plates channels, heated from below, secondary flows are induced by buoyancy forces and the local heat transfer increases; consequently, a two-dimensional laminar flow becomes three-dimensional [5,6]. The onset point of the secondary flows is relevant, because it delineates the region after which the two-dimensional laminar flow becomes three-dimensional and a transition motion from laminar to turbulent flow is observed.

Dixon et al. [7] show an experimental analysis considering mixed convection in a horizontal channel partially filled with a porous layer and an over lying fluid layer. The numerical prediction of a critical Péclet number at which the Nusselt number is minimum is confirmed experimentally, showing that a small cross flow can reduce heat transfer by disrupting natural convection currents. The transition to forced convection occurs at larger Péclet numbers than have been predicted numerically. Celli et al. [8] have investigated on the nonlinear stability of a rectangular porous channel saturated by a fluid. They consider the results of the linear stability analysis carried out by Barletta et al. [9] by studying the nonlinear governing equations for the perturbations instead of their linearized formulation. These investigation shows that the viscous dissipation is acting inside the channel as internal heat generator and this is the first nonlinear demonstration of viscous dissipation induced thermal instabilities in the literature. The same [10] carry out a stability analysis of a mixed convection problem in an inclined parallel-plate channel with uniform heating (or cooling) from the top and bottom. The channel is filled with a saturated homogeneous porous medium and the momentum equation is given by Darcy's model. The results indicate that the longitudinal rolls are always more unstable than oblique and transverse rolls.

Ould-Amer et al [11] show a numerical analysis on forced convection cooling of heat generating blocks mounted on a wall in a parallel plate channel in laminar regime. They study the features of heat transfer with a porous matrix between the blocks. Results show that the use of porous media enhances the heat transfer rate on the vertical sides of the blocks. An experimental study on forced convection in a rectangular channel is carried out by Kurtbas and Celik [12]. They study the heat transfer characteristics of the mixed convection flow through a horizontal rectangular channel with three different pore densities (10, 20 and 30 PPI). They apply three values of a uniform heat flux, and temperatures are measured on the entire surfaces of the walls. Their results are shown in terms of average and local Nusselt numbers as a function of Reynolds and Richardson numbers.

Metal foams with high thermal conductivity are an effective method of heat transfer enhancement due to their low weight, good strength, rigidity, damping of vibrations and noise, shock resistance [13]. In literature, metal foams are used in heat exchangers because of their high transfer capabilities. Boomsma et al. [14] show an experimental analysis on open-cell metal foam used as compact exchangers. Their results reveal that thermal resistance of metal foam heat exchangers is almost onethird of the available conventional heat exchanger. Metal foam have a high ratio between surface area and pressure drop, and with uniform lower density, and considering the unit of volume, the pressure drop is lower than in ceramic structures [15]. Leong and Jin [16,17] carry out an experimental investigation on oscillating flow through a rectangular channel filled with opencell metal foam. They analyze aluminum foam of 10, 20 and 40 PPI as the porous media shown data on pressure drops and velocities. The experiments show that the oscillating flow features in an open-cell metal foam are governed by the kinetic Reynolds number Re based on a hydraulic diameter and the dimensionless flow displacement amplitude A_{ph} . Subsequently, they use the porous media to obtain the heat transfer performance of metal foam heat sinks subjected to oscillating flows of various frequencies. Results show that higher heat transfer

rates can be obtained when metal foams are subjected to oscillating flow, considering length averaged Nusselt number for both oscillating and steady flows.

A one-dimensional numerical analysis of heat transfer of metal foams is carried out by Dukhan et al. [18], considering only one pore density, 10 PPI. The analysis shows that the temperature decreases exponentially along the foam in the flow direction. Tzeng and Jeng [19] analyze the convective heat transfer and pressure drop in porous channels in which the flow enters the channel through a 90-deg turn. Metal foam with a porosity of 0.93 are used in their study. Results show that the wall temperature has the maximum value where the entering flow impinges on the duct wall. Buonomo et al. [20,21] numerically and experimentally study mixed convection in a horizontal channel filled with metal foam partially heated at uniform heat flux. They consider an aluminum foam with 10 PPI and $\varepsilon=0.909$, the fluid is air. Their results are given in terms of solid and fluid temperatures, velocity profile along the channel and local and average Nusselt numbers, showing that diffusive effect causes lower temperature values inside the solid and fluid temperature is higher in all considered cases. On the other hand, for heated channel with smaller aspect ratios, an average Nusselt number increases for solid and fluid phases.

Cimpean et al. [22] show the steady fully developed mixed convection flow of a nanofluid in an inclined channel filled with a porous medium, and extend the problem [23] to the case when the porous medium is saturated by a nanofluid instead of a Newtonian fluid. The effects of the mixed convection parameter λ , the Péclet number, the inclination angle of the channel to the horizontal γ and the nanoparticle volume fraction ϕ on the fluid and heat transfer characteristics are discussed for three different nanofluids as Cu–water, Al_2O_3 –water and TiO_2 –water.

Chen and Lue [24] study the fully developed mixed convection in a vertical porous channel with a uniform heat flux imposed at the plates, using the Brinkman-Forchheimer extended Darcy model. Kamath et al. [25] studied on flow assisted mixed convection in aluminum metal foams of different pores per inch (PPI), with high porosity in the range of 0.9 to 0.95. They develop a correlation for Nusselt number as a function of Richardson number, Peclet number and porosity. Their results indicate that metal foams have good transfer enhancement capability. A recent study of Carpenter and Da Silva [26] on aluminum foam experimentally investigate metal foams at three different values of PPI (10, 20 and 40 PPI), uniformly heated from below. Their results show that the number of sections does not affect the overall pressure drop across the test section if all segments have the same PPI. Besides, they observe a higher transfer coefficient when air flows through the foam with smaller PPI, compared to the test section in the reversed orientation. Mixed convection in horizontal channels partially filled with porous media inserted transversally to the channel axis on the channel bottom wall are extensively investigated, as reported in [27-29] In these works, mixed convection in a horizontal channel partially filled with a porous medium and the lower wall heated at uniform heat flux is experimentally and numerically studied. The investigation is carried out by means of a visualization technique and heated wall temperature measurements.

In this work, mixed convection in a horizontal channel partially filled with a porous medium and the lower wall heated at uniform heat flux is studied experimentally and numerically. The experimental test section is made up of a horizontal wall and a parallel adiabatic wall. The bottom wall consists of a fiberboard plate and it is the heater. The upper and the side walls of the channel are made of glass rectangular plates. The porous medium is an aluminum foam and it is placed over the heated lower wall. The experiments are performed with working fluid air, the regime is laminar. The surface temperature, in the experimental apparatus, is measured by a set of thermocouples embedded in the fiberboard plate in the very proximity of the backside of the copper. Mineral oil vapor is employed for flow visualization. The visualization is made possible by means of a laser

sheet, generated by a He-Ne laser source. Flow visualization is performed to detect the flow patterns into the cavity.

2 Physical model and governing equations

A simplified two-dimensional horizontal parallel walls domain is considered in the numerical investigation in steady state regime. The physical domain presents a principal heated channel partially filled with an aluminium foam plate on the bottom heated wall, and two adiabatic channels, one upstream and the other one downstream the heated channel, as indicated in Fig. 1. The geometrical parameters are the distance between the horizontal walls, the thickness of the aluminium foam plate and the heated plate length. The bottom wall of the principal channel is heated at uniform heat flux and the parallel upper wall is a glass plate which allows heat transfer with the external ambient. An upper external reservoir over the glass plate, four times the channel height, simulates the natural convection toward the external ambient. All thermo-physical fluid properties are assumed temperature independent, except for the density in the buoyancy force term that can be adequately modelled by the Boussinesq approximation. The thermo-physical properties of the air are evaluated at the ambient temperature, T_0 , which is assumed to be 300 K in all cases. The compression work and viscous dissipation are assumed negligible. Thus, the governing equations for laminar, steady state regime together with the above assumptions can be written as:

-Continuity equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

-x- momentum equation

$$\rho_f \left(\frac{u}{\varepsilon^2} \frac{\partial u}{\partial x} + \frac{v}{\varepsilon^2} \frac{\partial u}{\partial y} \right) = -\frac{\partial p}{\partial x} + \frac{\mu_f}{\varepsilon} \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) - \delta \left(\frac{\mu_f}{K} u + \frac{C_F}{K^{1/2}} \rho_f \sqrt{u^2 + v^2} u \right) + \rho_f g \beta_f (T_f - T_0) \quad (2)$$

Fig. 1. Geometry of the numerical model.

-y-momentum equation

$$\rho_f \left(\frac{u}{\varepsilon^2} \frac{\partial v}{\partial x} + \frac{v}{\varepsilon^2} \frac{\partial v}{\partial y} \right) = -\frac{\partial p}{\partial y} + \frac{\mu_f}{\varepsilon} \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) - \delta \left(\frac{\mu_f}{K} v + \frac{C_F}{K^{1/2}} \rho_f \sqrt{u^2 + v^2} v \right) \quad (3)$$

where ε is the porosity, ρ_f and μ_f are density and viscosity of the air, u and v are the velocity components in Cartesian coordinates, K and C_F are the permeability and inertial coefficient of the Aluminum foam, respectively. The correlations of Bhattacharya et al. [30] are adopted for the evaluation of permeability K and the Forchheimer coefficient

-Fluid phase energy equation

$$(\rho c_p)_f \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) = (\varepsilon k_f + (1-\varepsilon)k_s) \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (4)$$

The governing equations (1)-(4) for porous zone are also valid for the clean region with $\delta=0$ and porosity $\varepsilon=1$.

The governing equations (1)-(4) are solved with following boundary conditions: the upper wall is adiabatic; the bottom wall is heated with constant heat flux. Non slip velocity is applied on all walls. The free surface condition is imposed to the exit sections. The numerical simulations are carried out by means of Ansys Fluent code [31]

3 Experimental study

The experimental test section is made up of a horizontal heated lower wall and a parallel adiabatic upper wall partially filled with a plate in aluminum foam placed on the heated wall, in Fig. 2a. The heated lower wall consists of a fibreboard plate, 3.2 mm thick, coated with a 17.5 mm thick copper-layer, which is the heater. To reduce the heat losses from the heated wall toward the external ambient, a 150 mm polystyrene block is affixed to rear face of heated plate. The upper and the side walls of the channel are made of glass rectangular plates, in order to take pictures of the flow motion during the flow visualization. The distance between the horizontal walls is equal to 40 mm and the distance between the plate in aluminum foam and the upper wall is 20 mm. The plate in aluminum foam has a thickness equal to 20 mm. The length of the test section in the flow direction is 400 mm, the same of the porous plate, and the width, W , is 450 mm. Three plates in aluminum foam are employed in the experiments with a pore density equal to 10, 20 and 40 pores per inches (PPI). The plate in aluminum foam were provided by M-Pore [32].

The experimental apparatus includes the test section, instruments and air facility, in Fig. 2b. Air is set into the test section from the laboratory room by means of a fan system positioned downstream. The air enters between two sheets of galvanized steel in order to obtain a fully developed laminar flow at the inlet section.

Temperatures are measured by thermocouples (type T) deployed along the flow direction. An Isotech instrument mod. 938 ice point, with an accuracy of ± 0.04 °C and 50 channels, is used as thermocouple reference junctions. The thermocouples are calibrated before their installation. All told, 12 are fixed in the center line of test section. An AGILENT 34980A multifunction measurement unit and a computer are used for the data acquisition. For the measurement of the mass flow rate, a hot wire anemometer is used. It is located at 60 tube diameters from the inlet of the exit tube. In this section the flow is fully developed and laminar. The anemometer is calibrated with four rotameter Krohne type GA24/RDN 15 [33]. The range of the Reynolds number varies from 5.0 to 250.

Fig. 2. Sketch of: (a) test section and (b) experimental set up.

4 Data reduction

In this work, the local heat flux is determined as:

$$q_c(x) = q_{\Omega}(x) - q_k(x) - q_r(x) \quad (4)$$

where $q_{\Omega}(x)$ is the local heat flux due to Ohmic dissipation, assumed uniform along x , $q_k(x)$ is the local conduction heat losses from the plate and $q_r(x)$ is the local radiative heat flux from the plate. For each run, the terms $q_k(x)$ is calculated by means of a numerical procedure, a three-dimensional distribution of the temperature being assumed in the Polystyrene. Therefore, $q_k(x)$ on the wall is a function of both x and z coordinates, and its values are averaged along z . The predicted temperatures for some configurations of the system are previously compared with those measured by thermocouples embedded in the Polystyrene insulation and the relationship is particularly good, the maximum deviation being 3%. A two-dimensional radiative cavity is made of the two plates, considered as diffuse-grey surfaces and two black edge sections at room temperature. In all cases, the radiative heat losses are not greater than 2% of the Ohmic dissipated power. The $q_r(x)$ terms are calculated for each temperature distribution of the wall, ambient temperature and plate spacing, by dividing each plate into sixteen strips along its length. Each strip is assumed at the spanwise average temperature. The average and local Nusselt number are

determined as functions of parametric values of the Reynolds and Richardson numbers (or the Grashof number).

The local Nusselt number is defined as:

$$Nu_i = \frac{q_c b}{(T_i - T_o)k_{air}} \quad (5)$$

the subscripts i indicates the axial location at which the temperature is measured. T_o is a reference temperature to form the temperature difference. It is the temperature upstream of the inlet of the heated test section. This reference temperature is a known quantity. It is convenient to use it in the evaluation of the present Nusselt number results in order to obtain the heat transfer coefficient. The average Nusselt number is evaluated as:

$$\overline{Nu} = \frac{\int_0^L Nu dx}{\int_0^L dx} \cong \frac{\Delta x}{L} \sum_{i=1}^n Nu_i \quad (6)$$

Where L is length of the test section, n is number of the temperature measurements points on the surface, Δx is the spacing the two thermocouples.

The Nusselt number results will be plotted as a function of Reynolds, Grashof and Richardson numbers, defined:

$$Re = \frac{\rho V b}{\mu} ; \quad Gr = \frac{g \beta q_c b^4}{\nu^2 k_{air}} ; \quad Ri = \frac{Gr}{Re} \quad (7)$$

The uncertainty of Nusselt, Reynolds and Grashof numbers were evaluated with the Kline and McClintock [34] methodology. It is estimated that the uncertainty on the Nusselt number is 4.97%, the Reynolds number is 2.98%, and the Grashof numbers is 5.64%.

5 Results

5.1 Numerical results

The analysis was carried out by varying the heat flux of the test section and keeping constant the Reynolds number equal to 150, 200 and 250.

Figures 3-4 show the fields of velocity and temperature at heat flux equal to 100 W/m². Fig.3 shows that velocity values are higher without the metal foam and they are smaller only in proximity of the walls. The presence of the foam reduces considerably velocity values and they are zero in the inlet of the porous zone. The comparison of temperature fields in Fig. 4 shows that values are lower in presence of the foam, so gradients are more gradual. When the channel is clean, cold air hits the heat flux and the thermal gradient is higher. This phenomenon leads to a slowing down of the flow and the creation of a zone of recirculation in the terminal part of the channel.

Figure 5 shows velocity and temperature profiles along a vertical line in the center of the plate, along the channel section in which the fluid flows. Fig.5 shows that, when the channel is clean, the velocity profile is always parabolic. In presence of the metal foam, the velocity profile is vertical and the value is equal to zero. The temperature is constant in the metal foam as shown in the diagrams on the right in Fig. 5.

Fig. 3. Velocity fields at different Reynolds number; $q=100\text{W/m}^2$, 20 PPI: (a) $Re = 150$, (b) $Re = 200$, (c) $Re = 250$.

Fig. 6 shows the profiles of the local Nusselt number along a horizontal section in the middle of the channel, in proximity of the heated plate. The Nusselt number trend is parabolic and values are lower in the configuration without the foam.

Fig. 4. Temperature fields at different Reynolds number; $q=100\text{W/m}^2$, 20 PPI: (a) $Re = 150$, (b) $Re = 200$, (c) $Re = 250$.

Fig. 5. Velocity and temperature profiles for $q=100\text{ W/m}^2$, with 20 PPI at different Reynolds number, (●) clean, (o) aluminum foam: (a) $Re=150$, (b) $Re=200$ and (c) $Re=250$.

5.2 Experimental results

Experimental investigations are performed in order to validate the numerical model. Results are presented in terms of Nusselt number, as a function of Reynolds and Richardson numbers. Four conditions of the channel are considered: the first is clean plate, without porous medium; the others with porous medium, with 10, 20 and 40 PPI.

Fig. 6. Local Nusselt number profiles for $q=100 \text{ W/m}^2$, with 20 PPI at different Reynolds number values, (●) clean, (o) aluminum foam: (a) $Re=150$ and (b) $Re=250$.

Figure 7 shows the trend of average Nusselt number as a function of Reynolds number for eight configurations. Nusselt number depends on the convective flow carried out locally; increasing the Reynolds number, the Nusselt number grows. For the metal foam of 10 PPI at low Reynolds numbers, the Nusselt number increases gradually increasing the Reynolds number; the same behaviour can be seen for the metal foam of 20 PPI. It is observed the Nusselt number for the metal foam of 20 PPI increases more gradually.

Fig. 7. Average Nusselt number as a function of Reynolds number.

In Figs. 8(a) and 8(b), the ratio Nu_{avg}/Re ratio increases increasing the Richardson number. When Richardson number increases, the buoyancy driving force is predominant with respect to inertial force. Furthermore, the Nu/Re ratio increases because Nusselt number increases or Reynolds number decreases. But if Reynolds number decreases, Nusselt number does not increase, thus decreasing both, with a greater decrease of Reynolds number. The prevalence of the buoyancy shows that the natural convection is more developed than forced convection, in accordance with numerical analysis.

6 Conclusions

A numerical and experimental investigation on mixed convection in air with metal foam is carried out. Results of numerical investigation are in good agreement with results from experiments.

Velocity and temperature fields show that increasing heat flux, the natural convection it is predominant on the forced one. The numerical results highlight, for the considered Reynolds number range, that the motion of the fluid in the case without porous medium is not significantly affected by the thermal gradient due to heat flux generated by the heated plate and the velocity profile is parabolic, with the maximum at the center of the transversal section. The presence of the porous medium determines a restriction of the section. The velocity profiles are uniform inside the porous medium with low velocity

Fig. 8. Nu/Re ratio trend as a function of Richardson number for eight different configurations.

values and the parabolic profiles are developed inside the clean part of the channel. The experimental analysis has confirmed that the use of the porous medium increases the heat transfer promoting the heat dissipation of the surface with high efficacy.

7 Nomenclature

b	channel spacing, m
g	acceleration due to the gravity, ms^{-2}
Gr	Grashof number
h	heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$
k	thermal conductivity, $\text{Wm}^{-1}\text{K}^{-1}$
L	length of the plate, m
Pr	Prandtl number
q	heat flux, Wm^{-2}
Ra	Rayleigh number, = Gr Pr
Re	Reynolds number
Ri	Richardson number
T	temperature, K
u_i	average velocity at inlet section of the channel, ms^{-1}
x, y, z	Cartesian coordinate, m
W	width of the plate, m

Greek symbols

β	volumetric coefficient of expansion, K^{-1}
ν	kinematic viscosity, m^2s^{-1}

Subscript

c	convective
k	conductive
0	ambient air
r	radiative
Ω	Ohmic dissipation

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