Mixed convection in horizontal channels partially filled with aluminium foam heated from below and with external heat losses on upper plate

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Abstract. 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 both experimentally and numerically. The experimental test section is made of a horizontal wall and a parallel wall with heat losses toward the external ambient. A simplified two dimensional numerical model is considered to evaluate the dynamic and thermal fields inside the heated channel. The investigation allows to evaluate the effect of the aluminium foam on the mixed convection in the heated channel by wall temperature measurements and flow visualization. Results are given for heated channel without and with foam in terms of wall temperature profiles for different Reynolds number value, from 100 to 300, wall heat flux and for aluminium foam with 10 and 20 pore per inch. Experimental and numerical results are given in terms of wall temperature profiles and both allow to show that the foam determines lower wall temperature profiles along the fluid flow than the ones in the clean channel cases. Moreover, the foams with different pores per inch present different local thermal behaviours. The simplified numerical model allows to estimate the possible heat losses toward the external ambient.

1. Introduction
In recent years mixed convection in horizontal channels has been studied diffusively for its presence in many applications in different fields. Solar components, cooling of electronic equipments and chemical vapor deposition are just some of the fields concerned, as indicated in [1-5]. Low velocity forced flow in horizontal parallel plates channels, heated from below, can be affected by secondary flow which are induced by buoyancy and a three dimensional mixed convection flows is determined. Moreover, the local heat transfer increases due to the transition from a two-dimensional laminar flow to a three-dimensional one [1-3,6,7]. The onset point of the secondary flows is important 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. The unstable and stable vortex flow associated to the secondary flow is advantageous for some application such as cooling of electronic equipments [4]. On the contrary in other technological applications such as in the chemical vapor deposition processes used to grow thin crystal films on silicon substrates, the presence of the vortex flow determines a non-uniform deposition on the substrates [5]. The secondary flow and the enhancement of the heat transfer, in the so-called Poiseuille-Rayleigh-Benard flow, depends strongly
on the Rayleigh number, the Reynolds number, the Prandtl number of the channel flow, the boundary conditions, and the geometry of the channel, as well explained in [1]. The relevant literature on mixed convection in horizontal channels heated from below has been reviewed in [1-3,6-11].

Porous media with high thermal conductivity and porosity, such as metal foams, have emerged as an effective method of heat transfer enhancement due to their large surface area to volume ratio and to intense mixing of the flow. In the last two decades metal foams have been applied in heat transfer and their use in thermal engineering is growing quite quickly. Several investigations have been accomplished on this topic, as reviewed in [12]. The possibility to employ a metal foam to improve heat transfer in a horizontal channel has been investigated in several studies as reported in [12,13].

Investigations on mixed convection in horizontal channels filled or partially filled with a metal foam heated from below seems very few as indicated in [14]. Several investigations have been focused on instability in a fluid saturated porous medium in a heated channel, as recently reported in [15-17]. Moreover, mixed convection in horizontal channels partially filled with porous blocks intermittently inserted in transverse to the channel axis on the channel bottom wall have been extensively investigated, as reported for example in [18-20]. Some experimental investigations on mixed convection in vertical channel partially filled with metal foams have been carried out in [21,22]. Two porous plates are set along the channel walls and fill partially the channel. More recently, an experimental investigation on mixed convection in horizontal channel partially filled with an aluminum foam has been accomplished in [23].

It seems that mixed convection with porous medium inserted in horizontal channels heated from below has not been adequately investigated experimentally or numerically, though it has a lot of potential engineering applications like electronic equipment cooling, solar collectors, thermal energy storage, chemical vapour deposition (CVD) and thermal hydraulics of nuclear reactors. 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 experimentally and numerically studied. Results in terms of wall temperature profiles as a function of the pores per inch (PPI), Reynolds number values and wall heat flux are presented. The experimental results are compared with the ones carried out with a simplified two-dimensional numerical model. The comparison is accomplished in terms of wall temperature profiles.

2. Experimental apparatus
A schematic layout of the experimental set-up is shown in figure 1. The experimental apparatus consists of a convergent channel at the entrance of the main horizontal channel with the lower wall partially heated and a parallel unheated glass plate above, a diffuser, a flow control valve and an AC fan. The horizontal channel is 1.40 m long, 0.475 m width and the height can range from 10 mm to 100 mm. It is made up by a central part, long 0.400 m which represents the test section, with the lower wall uniformly heated, a parallel unheated glass plate above and two vertical unheated glass walls, as reported in figure 2. The upstream and downstream extension parts of the main channel have lower and vertical walls made of wood, figure 1. The blower attached to the channel through the diffuser provided a variable mass flow rate. The apparatus was located within a room, sealed to eliminate extraneous air currents.

The heated wall consists of two 400x530 mm² sandwiched phenolic fiberboard plates and is 4.8 mm thick. The upper glass plate is 500 x 600 mm² and 6 mm thick. The side walls of the heated part of the main channel are made of glass, which are machined to an accuracy of ±0.3 mm, in order to take pictures of the flow motion. The plate spacing is measured to an accuracy of ±0.25 mm by a dial-gauge equipped caliper. In some runs, the heated part of the channel is partially filled by a porous medium layer on the lower wall. The porous medium is an aluminum foam layer and it has a thickness equal to 20 mm whereas the length and the width were the same of the channel. The investigated aluminum foams can have 10 or 20 Pores Per Inches (PPI) with porosity equal to 0.97 and 0.95, respectively. The lower wall was made of two plates.
The heated plate facing the channel has the surface adjacent to the internal air coated with a 35 µm thick nickel plated copper layer. The low emissivity of nickel (0.05) minimized radiation effects on heat transfer. The rear plate is 2.6 mm thick. Its back surface is coated with a 17.5 µm thick copper layer, which is the heater. It is an electrical resistance obtained cutting the copper layer in a serpentine shape. Its runs are 19.6 mm wide with a gap of nearly 0.5 mm between each one, giving the heater a total length of 9.0 m. Its expected electrical resistance is 0.50 Ω. The lower wall is heated by passing a direct electrical current through the heater. In order to reduce conductive heat losses, a 150 mm Polystyrene block is affixed to the rear face of each principal wall. The narrow gaps between the runs, together with the relatively high thickness (4.8 mm) of the resulting low-conductive fiberglass are suitable to maintain a nearly uniform heat flux at the plate surface. Direct electrical current through the heaters is accomplished by using Agilent TarE3632Adc power supply.

The electrical power supplied is evaluated by measuring the voltage drop across the plate and the current passing through it. An HP-3465A digital multi-meter measures the voltage drop, while the current is calculated by the measured voltage drop across a reference resistance. To avoid electrical contact resistances, thick copper bars soldered both to the electric supply wire and to the ends of heater are bolted together. The dissipated heat flux is evaluated to an accuracy of ±2%.

Wall temperatures are measured by 0.50 mm OD ungrounded iron-constantan thermocouples embedded in each fiberboard plate and in contact with the outer layer. They are located at twelve longitudinal stations at three different z values. Fifteen thermocouples are affixed to the rear surface of the plates and embedded in the Polystyrene to enable the evaluation of conductive heat losses.

*Figure 1.* Layout of the experimental apparatus, dimensions in mm.

*Figure 2.* View of the test section.
ambient air temperature is measured by a shielded thermocouple placed near the leading edge of the channel. A Isotech TRU Model 938 ice point is used as a reference for thermocouples junctions. Their voltages are recorded by an Agilent 34980A multifunction measurement unit and a personal computer is used for the data collection and reduction. Calibration of the temperature measuring system shows an estimated precision of the thermocouple-readout system of ±0.1 °C.

Mass flow rate is evaluated by measuring the velocity with a hot wire anemometer Dantec Mini CTA 54T30 with a 55P11 probe. The sensor is located at 2500 mm from the inlet section of a circular duct, with a diameter of 30 mm, in order to have a fully developed laminar flow, figure 1. Furthermore, the hot wire probe is placed at about 6.0 m away from the outlet rectangular section of the test section. The hot wire probe is calibrated at 15°C, 20°C and 25°C in the velocity range from 0.010 m/s to 1.10 m/s. The maximum uncertainty in this range is about ±4%, the uncertainty on the measurement of the duct, diameter is ±1% and the uncertainty of the location of the sensor is ±2%.

It was observed that the steady-state condition was reached after 4–6 h.

3. Data reduction

The governing parameters in the problem are the buoyancy parameter Grashof, $Gr$, Rayleigh, $Ra$, and Richardson number, $Ri$, the Reynolds number, $Re$, and the Prandtl number, $Pr$. The Grashof and the Reynolds numbers are defined as:

$$Gr = \frac{g\beta q_c b^4}{v^2 k} ; \quad Ra = Gr Pr ; \quad Re = \frac{u b}{v} ; \quad Ri = \frac{Gr}{Re^2} ; \quad Pr = \frac{\nu}{a}$$

where $q_c$ is the average convective heat flux:

$$q_c = \frac{1}{L_0} \int q_c(x)dx$$

 thermo-physical properties were evaluated at the reference temperature $T_r = (T_w + T_b) / 2$ with:

$$T_w = \frac{1}{L_0} \int T_w(x)dx$$

where $T_w$ is the average wall temperature along the heated lower plate.

Local convective heat flux, $q_c(x)$, was not uniform because of radiation and conduction heat losses. Experimental data were reduced by first introducing, in the equations presented above, the local convective heat flux.

$$q_c(x) = q_{c1}(x) - q_{k}(x) - q_r(x)$$

where $q_{c1}(x)$ 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 $q_k(x)$ were calculated by means of a numerical procedure, a three-dimensional distribution of the temperature being assumed in the Polystyrene.

Therefore, $q_k$ on the wall was a function of both $x$ and $z$ coordinates, and its values were averaged along $z$. The predicted temperatures for some configurations of the system were previously compared with those measured by thermocouples embedded in the Polystyrene insulation and the relationship was very good, the maximum deviation being 3%.

A two-dimensional radiative cavity was 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 were not greater than 2% of the Ohmic dissipated power. The $q_r(x)$ terms were 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 was assumed at the span wise average temperature.
The uncertainty in the calculated quantities was determined according to the standard single sample analysis recommended by Moffat [24]. The uncertainty of the Rayleigh and Reynolds numbers were 6% and 7%, respectively.

4. Model description and numerical procedure
A preliminary simplified numerical simulation is accomplished in a two-dimensional, steady state and incompressible flow. The domain is made of a principal channel, partially filled with an aluminum foam plate, and two channels with adiabatic walls, one upstream the principal channel and the other downstream, figure 3. In this problem the geometrical parameters are the distance between the horizontal walls, the thickness of the aluminum foam plate and the heated plate length. The principal channel is made up of a uniformly heated horizontal wall at uniform heat flux and a parallel glass plate located above 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. Viscous dissipation, heat generation and pressure work are all assumed to have negligible effect on the velocity and temperature fields, therefore they are neglected. All the thermophysical properties of the fluid and the solid matrix of the porous medium are assumed constant except for the variation in density of the air with temperature (Boussinesq approximation) giving rise to the buoyancy forces. The thermo-physical properties of the fluid and the solid matrix of the porous medium are evaluated at the ambient temperature, \( T_0 \), which is equal to 300 K in all cases. In the porous medium region, the generalized flow model, known as the Brinkman-Forchheimer-extended Darcy model, is used in the governing equations. It is assumed that fluid and solid phase of porous medium are in local thermal equilibrium. Governing equations for the considered problem are not given in the present paper but the reader can refer to [25, 26].

Thermo-physical properties of the upper glass plate are: \( \rho = 2800 \text{ kg m}^{-3}, k = 0.81 \text{ W m}^{-1} \text{ K}^{-1}, c_p = 800 \text{ J kg}^{-1} \text{ K}^{-1} \). The lower heated plate has \( \rho = 1206 \text{ kg m}^{-3}, k = 2.0 \text{ W m}^{-1} \text{ K}^{-1}, c_p = 361 \text{ J kg}^{-1} \text{ K}^{-1} \). The geometrical parameter in the simulations are the same of the experimental section along its longitudinal section. For the aluminum foam it is assumed a permeability equal to \( 1.17 \times 10^{-7} \text{ m}^2 \) and \( 1.12 \times 10^{-7} \text{ m}^2 \) for 10 and 20 PPI, respectively.

The commercial CFD code Ansys-Fluent [26] is employed to solve the governing equations. The SIMPLE scheme is chosen to couple pressure and velocity. The porous medium model is active in the porous region and the LTE is employed. The convergence criteria of \( 10^{-6} \) for the residual of continuity equation and velocity components and \( 10^{-8} \) for the residuals of energy are assumed.

A grid dependence test is accomplished to realize the most convenient grid size by monitoring local wall temperature for wall heat flux equal to 100 W/m\(^2\) and a Reynolds number equal to 200. Three...
different uniform grids are tested with 112x1120, 160x1600, 226x2260 nodes in the channel. The 160x1600 node grid is the best compromise between accuracy and computational time. This mesh size was employed for all considered cases.

It should be underlined that the two-dimensional geometrical model has been employed just to have some indication on the main flow and the effect of the aluminum foam. It is not able to detect the transversal secondary motions that are present in the channel [23].

5. Results and discussion

The experiments were carried out for $100 \leq Re \leq 300$ and $1.0 \times 10^5 \leq Ra \leq 1.8 \times 10^6$ with a corresponding Richardson number values from 18 to 220 and for two wall heat flux equal to 100 and 200 Wm$^{-2}$. The longitudinal aspect ratio was 10 both in the clean cases, without porous medium, and in the cases with the foam. The transversal aspect ratio was 12.45 in all cases.

Thermal behaviours in all cases are evaluated by wall temperature profiles. In figure 4, experimental wall temperature profiles along the axis for all considered cases, with and without foam (clean case) and two different wall heat flux values, 100 W m$^{-2}$ and 200 W m$^{-2}$ and $Re = 300$, show that temperature increases along the flow up to $x$ about 300 mm, where a maximum temperature value is attained, and after it decreases. The comparison between the channel without foam or with foam allows to observe that the presence of aluminum foam determines a wall temperatures reduction. For the cases with the aluminum foam with 10 PPI lower temperature values than the ones for 20 PPI are detected in the inlet region whereas the opposite is observed in the outlet zone also in accordance with the results given in [23]. In fact, in the inlet zone the fluid penetrates in 10 PPI foams better than in the 20 PPI and the surface heat transfer on the heated wall is favorite for 10 PPI. The inversion of the values along the axial direction is due to a better contact between the 20 PPI foams layer and the heated wall than the one between the foam with 10 PPI and the bottom heated wall. Consequently, a better heat transfer is obtained in the case of the 20 PPI foam which presents lower temperature values than the ones for 10 PPI foams for $x > 100$. The effect is more evident for the highest considered wall heat flux, as can be noted in figure 4b.

![Figure 4](a) ![Figure 4](b)

**Figure 4.** Wall temperature profiles along the heated plate for channel with (10 and 20 PPI) and without (clean 4) aluminum foam plate, for $Re = 300$ and: a) $q=100$ W m$^{-2}$ and b) $q=200$ W m$^{-2}$. 
In figures 5-7 the comparisons between experimental and numerical results are reported for Reynolds number values equal to 100, 200 and 300 and two wall heat flux values, 100 W m$^{-2}$ and $q=200$ W m$^{-2}$.

For the all $Re$ values considered, it is noted that the numerical temperature values are higher than the experimental ones but the trends are very similar on a quality level. In fact for the two different

![Graph](attachment:image1.png)

**Figure 5.** Comparison between experimental and numerical wall temperature profiles for $Re=100$ and: a) $q=100$ W m$^{-2}$ and b) $q=200$ W m$^{-2}$.

![Graph](attachment:image2.png)

**Figure 6.** Comparison between experimental and numerical wall temperature profiles for $Re=200$ and: a) $q=100$ W m$^{-2}$ and b) $q=200$ W m$^{-2}$.
PPI values, in all cases the profiles present an intersection between the curves and the curve slopes are very similar both in the experimental and in numerical results. In figure 5, for $Re=100$ the differences between the experimental and numerical profiles are the greatest. These differences decrease increasing the $Re$ value, as can be noted in figures 6 and 7 for $Re=200$ and 300, respectively. Moreover, for higher wall heat flux cases the differences between the wall temperature profiles with different PPI value increase. These two last observations suggest that the secondary motions become weaker increasing the Reynolds numbers as well as decreasing the wall heat flux. A two dimensional numerical simulation can give a good estimation of the main fluid flow inside the heated channel.

In figure 8 air temperature and stream function fields are presented for the cases clean and with aluminum foam, 10 PPI and 20 PPI, for $Re=100$ and 300 with $q=100$ W m$^{-2}$. For $Re=100$ the buoyancy pushes the fluid flow toward the upper wall in the outflow zone of the heated channel. This determines a heat loss toward the external ambient. For $Re=300$ the heat losses seem significantly decreased because of the higher velocity of the forced flow which reduces the buoyancy effect. The flow does not move adjacent the upper plate in the outlet zone. For the heated channel partially filled with aluminum foam and $Re=100$, the heated fluid reaches closer the upper plate before the middle zone of the heated channel and the heat losses are higher than in the clean case. Moreover, in the case with 10 PPI the external plume is stronger. For $Re=300$, the forced flow with higher velocity sweeps in advance the heated fluid and it does not allow to the hottest fluid to reach completely the upper plate. The heat losses toward the external ambient are significantly reduced.

6. Conclusions

Experimental and numerical results, given in terms of wall temperature profiles, showed that the foam determined lower temperature values along the wall than the ones in clean channel cases in accordance also with [23]. Different behaviours of local wall temperatures was detected for the foam with different PPI, for lower PPI at the inlet zone of the heated plate lower temperature were attained whereas the opposite was noted in the other part of the heated channel. Similar trends on a quality level were observed between numerical and experimental wall temperature profiles. A two dimensional numerical model can give a good evaluation of main flow inside a heated channel from below in mixed convection.

![Figure 7. Comparison between experimental and numerical wall temperature profiles for $Re=300$ and: a) $q=100$ W m$^{-2}$ and b) $q=200$ W m$^{-2}$.](image-url)
Figure 8. Numerical results, fields for $q = 100 \text{ W m}^{-2}$ and $Re=100$ and 300: a) air temperature fields and b) stream function fields

Nomenclature

- $b$ channel spacing, m
- $c_p$ specific heat
- $g$ acceleration due to the gravity, m s$^{-2}$
- $Gr$ Grashof number, Eq. (1)
- $h$ heat transfer coefficient, W m$^{-2}$ K$^{-1}$
- $k$ thermal conductivity, W m$^{-1}$ K$^{-1}$
- $L$ length of the plate, m
- $Pr$ Prandtl number, Eq. (1)
- $q$ heat flux, W m$^{-2}$
- $Ra$ Rayleigh number, $= Gr Pr$, Eq. (1)
- $Re$ Reynolds number, Eq. (1)
- $Ri$ Richardson number, Eq. (1)
- $T$ temperature, K
- $u_i$ average velocity at inlet section of the channel, m s$^{-1}$
- $x$ horizontal coordinate distance, m
- $y$ vertical coordinate distance, m
- $z$ coordinate along the width of the plates, m
- $W$ width of the plate, m

Greek symbols

- $\beta$ volumetric coefficient of expansion, K$^{-1}$
- $\nu$ kinematic viscosity, m$^2$ s$^{-1}$

Subscript

- $c$ convective
- $k$ conductive
- $i$ inflow
0 ambient air
r radiative
w wall
Ω Ohmic dissipation

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