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MIXED CONVECTION IN HORIZONTAL CHANNELS HEATED BELOW WITH EXTERNAL HEAT LOSSES ON UPPER PLATE AND PARTIALLY FILLED WITH ALUMINIUM FOAM

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ABSTRACT
In this paper an experimental investigation on mixed convection in air in a heated channel partially filled with an aluminum foam is carried out. The aluminum foam layer is sets on the lower heated wall of the channel. The channel upper plate has a heat transfer toward the external ambient. The investigation allows to evaluate the effect of the aluminum 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, form 10 to 300, wall heat flux and for aluminum foam with 10 and 20 pore per inch. Lower wall temperature values are detected for the channel with foam with respect to the channel clean without foam. Foams with higher pore per inch shows lower average wall temperature values whereas the local wall temperatures showed different behaviors for the different pore per inch values. The presence of foam in heated channel determined weaker secondary motions with respect to the clean cases. The effect of the foams seems more significant for the high Reynolds number values and average Nusselt number increases with the foam presence in the heated channel.

INTRODUCTION
In recent years the practical applications in many technological fields of mixed convection in horizontal channels has determined a diffuse interest in thermal engineering research on this topic. The research activities have been increased to realize thermal performance improvements in solar components, in cooling of electronic equipments and in chemical vapor deposition are just some of the fields concerned, as indicated in [1-7]. In forced convection flow with low velocity in horizontal parallel plates channels, heated from below, secondary flow can be arisen induced by buoyancy. It affects significantly the fluid motion and a three dimensional mixed convection flows is detected [3-5]. It is called Poiseuille-Rayleigh-Benard flow [2,3].

The transition from two-dimensional laminar flow to three-dimensional one determines a local heat transfer increases [3-6,9,10]. Then the onset point of the secondary flows is an important information because it outlines the zone where the two-dimensional laminar flow becomes three-dimensional with a transition motion from laminar to turbulent flow. It should be underlined that unstable and stable vortex flow allow to enhance the convective heat transfer and which can be very useful in application such as cooling of electronic equipments [1,6-8]. In other side, in different technological applications such as chemical vapor deposition processes employed in the growth of thin crystal films on silicon substrates, the vortex flow determines a non-uniform deposition on the substrates [2-5].

In PRB flow, the secondary flow and the heat transfer enhancement depend strongly on the Rayleigh number, the Reynolds number, the Prandtl number, the boundary conditions and the geometry of the channel, as well explained in [3-5]. The main relevant literature on mixed convection in horizontal channels heated from below has been reviewed in [4-6,7-13].

In order to enhance the heat transfer employing or reducing the available volume, metal foams can be a possible solution due to their high thermal conductivity, 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 [14,15].

It seems that few investigations on mixed convection in horizontal channels filled or partially filled with metal foams heated from below have been accomplished, as also indicated in [16]. Some studies on instability in horizontal channel heated from below and filled with fluid saturated porous medium have been focused in [17-19]. Mixed convection in horizontal channels partially filled with porous blocks intermittently inserted in the channel, on the bottom wall, have been investigated in [20-22]. A numerical investigation on PRB flow in a horizontal channel partially filled by a porous medium was performed in [23]. Experimental investigations on mixed convection in vertical channel partially filled with metal foams have been carried out in [24]. 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 presented in [25].

As showed in the previous review, it seems that mixed convection in horizontal channel heated from below and partially filled with porous media has received few attention 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. There is a lack of information on thermal and fluidynamic behaviors on this important configuration. In this paper an experimental and numerical investigation on mixed convection in a horizontal channel partially filled with a porous medium with the lower wall heated at uniform heat flux is presented to extend the results given in [29,30]. 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.

NOMENCLATURE

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

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

Subscripts

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

EXPERIMENTAL APPARATUS AND TEST SECTION

A sketch of the experimental apparatus is reported in Fig. 1. It consists in a mail channel 1.40 m long with a rectangular cross section wide 0.475 m and with a variable height from 0.010 m to 0.100 m. The main channel present unheated entrance and exit sections, channel extensions, both 0.500 m long, the test section, 0.400 m long and placed in the central part of the channel, with the heated bottom wall and the unheated upper wall. The walls of the channel extensions are made of wood. Two fans attached to the channel through a nozzle provided a variable mass flow rate. The apparatus was located within a room, sealed to eliminate extraneous air currents.

The experimental test section was the heated channel made of two horizontal walls with the bottom wall uniformly heated and the upper parallel wall unheated and made in glass, as showed in Fig.2. The upper glass plate was 400 x 475 mm^2 and 3 mm thick. The heated bottom wall was made up of two 400x530 mm^2 sandwiched phenolic fiberboard plates and was 4.8 mm thick. The plate facing the channel was 3.2 mm thick and its surface adjacent to the internal air was 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 was 1.6 mm thick. Its back surface was coated with a 17.5 μm thick copper layer, which was the heater. It was an electrical resistance obtained cutting the copper layer in a serpentine shape. Its runs were 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 was 0.50 Ω. The lower wall was heated by passing a direct electrical current through the heater. The narrow gaps between the runs, together with the relatively high thickness (4.8 mm) of the resulting low-conductive fiberglass were suitable to maintain a nearly uniform heat flux at the plate surface. In order to allow pictures of the flow motion inside the channel, the side vertical walls of the channel were made of glass plates and machined to an accuracy of ±0.3 mm. Moreover, the plate spacing was measured to an accuracy of ±0.25 mm by a dial-gauge equipped caliper. The heated part of the channel was partially filled by a aluminum foam layer placed on the heated lower wall. The aluminum foam layer had a thickness equal to 20 mm whereas the length and the width were the same of the channel. The investigated aluminum foam had 10, and 20 Pores Per Inches (PPI) with porosity equal to 0.97 and 0.95, respectively.

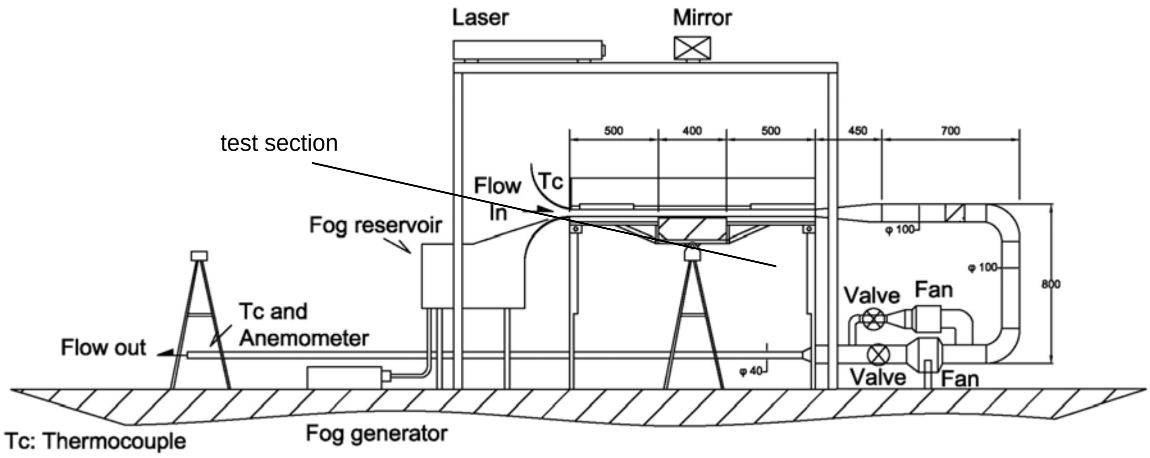

Figure 1: LAYOUT OF THE EXPERIMENTAL APPARATUS, DIMENSIONS IN MM

In order to reduce conductive heat losses, a 150 mm Polystyrene block was affixed to the rear face of each principal wall. Direct electrical current through the heaters was accomplished by using a Hewlett-Packard 6260B dc power supply. The electrical power supplied was evaluated by measuring the voltage drop across the plate and the current passing through it. An HP-3465A digital multi-meter measured the voltage drop, while the current was 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 were bolted together. The dissipated heat flux was evaluated to an accuracy of ±2%.

Wall temperatures were measured by 0.50 mm OD ungrounded iron-constantan (J-type) thermocouples embedded in each fiberboard plate and in contact with the outer layer. They were located at twelve longitudinal stations at three different z values. Fifteen thermocouples were affixed to the rear surface of the plates and embedded in the Polystyrene to enable the evaluation of conductive heat losses. Their voltages were recorded by an Agilent 34980A multifunction measurement unit and a personal computer was used for the data collection and reduction. Calibration of the temperature measuring system showed an estimated precision of the thermocouple-readout system of ±0.1 °C.

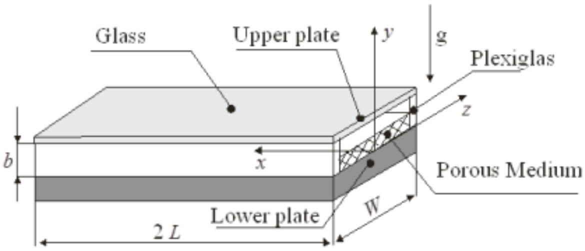

Figure 2: VIEW OF THE TEST SECTION

The ambient air temperature was measured by a shielded thermocouple placed near the leading edge of the channel. An Isotech instrument mod. 938 ice point, with 50 channels and an accuracy of ±0.03 °C, was used as a reference for thermocouple junctions. Their voltages were recorded to 1μV by an Agilent 34980A data acquisition system. The acquired data were processed and elaborated by an Excel spreadsheet. Calibration of the temperature measuring system showed an estimated uncertainty of the thermocouple-readout system of ±0.1 °C. Mass flow rate was calculated by measuring the velocity with a hot wire anemometer Dantec Mini CTA 54T30 with a 55P11 probe The sensor was 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, Fig. 1. The hot wire probe was calibrated at 15°C, 20°C and 25°C in the velocity range from 0.02 m/s to 1.0 m/s. The maximum uncertainty in this range was about 4%, the uncertainty on the measurement of the duct diameter was 1% and the uncertainty of the location of the sensor was 2%.

DATA REDUCTION AND UNCERTAINTY ANALYSIS

The governing parameters in the problem are the buoyancy parameter Richardson number, $Ri=Gr/Re^2$, the Reynolds number, Re, and the Prandtl number, Pr. The Grashof and the Reynolds numbers are defined as:

$$Gr = \frac{g\beta q_c b^4}{\nu^2 k}, \quad Re = \frac{u_i b}{\nu}, \quad Ri = \frac{Gr}{Re^2} \tag{1}$$

where q_c is the average convective heat flux:

$$q_c = \frac{1}{L}\int_0^L q_c(x)dx \tag{2}$$

the thermophysical properties were evaluated at the reference temperature $T_r = (T_w + T_0)/2$ with:

$$T_w = \frac{1}{L}\int_0^L T_w(x)dx \tag{3}$$

where T_w is the average wall temperature along the heated lower plate.
The local and average Nusselt numbers are defined, respectively, as

$$Nu(x) = \frac{h(x)b}{k} \tag{4}$$

$$Nu = \frac{hb}{k} = \frac{q_c}{T_w - T_0}\frac{b}{k} \tag{5}$$

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

$$q_c(x) = q_\Omega(x) - q_k(x) - q_r(x) \tag{6}$$

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 $q_k(x)$ 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 spanwise average temperature. The uncertainty in the calculated quantities was determined according to the standard single sample analysis recommended by Moffat [26]. The uncertainties of Grashof, Reynolds and local and average Nusselt numbers were 7%, 5%, 5% and 6%.

RESULTS AND DISCUSSION

The experiments were carried out for $25 \le Re \le 300$ and $1.0 \times 10^5 \le Ra \le 1.8 \times 10^6$, for two wall heat flux, 100 and 200 Wm^{-2}, with a corresponding Richardson number values from 18 to 220. The longitudinal aspect ratio was

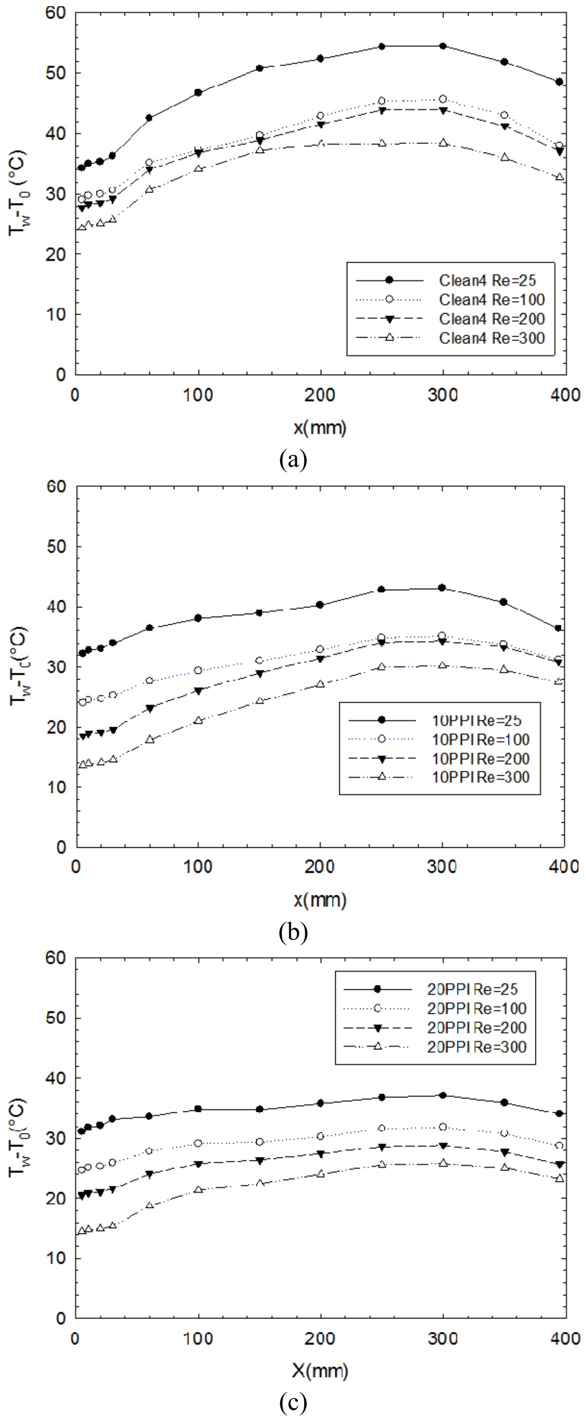

Figure 3: WALL TEMPERATURE PROFILES FOR WALL HEAT FLUX EQUAL TO 200 W/M^2, DIFFERENT REYNOLDS NUMBERS AND B=40 MM, FOR: a) CLEAN, b) 10 PPI AND c) 20 PPI

10 and the transversal aspect ratio was 12.5 in all cases. Wall temperature profiles along x for z=0 and a wall heat flux equal to 200 W/m^2 are shown in Fig. 3 for different Reynolds number values and for clean case, in Fig. 3a, and the cases with aluminum foam plate, 10 and 20 PPI, in Figs. 3b and 3c. Wall temperature decreases increasing

the Reynolds numbers, as expected. The profiles in the clean cases, in Fig. 3a, present the highest temperature values. Maximum wall temperature in almost all cases are attained at about x=300 mm. The effect of aluminum foam layer can be inferred by wall temperature profiles presented in Figs. 3b and 3c for two pore per inch values, 10 and 20. It is interesting to observe, at same Re values, that wall temperature profiles have lower values than the ones for clean cases. Moreover, maximum wall temperatures also in these cases are reached at about x=300 mm. Slight different trends are noted between profiles for 10 PPI and 20 PPI. For 10 PPI, in Fig. 3b, profiles in the outlet section zone present very close values with Re≤200 due to the secondary motions inside the channel. For cases in Fig. 3c with 20 PPI foam, wall temperature profiles present some differences also in the outlet region with a forced convection prevalence in the heat transfer. In these cases, secondary motions are reduced due to smaller passages and greater tortuosity inside the foam.

In Figs 3, for Re=100 the comparison among the profiles shows, clearly, that the presence of aluminum foam causes a wall temperatures reduction. Moreover, along the channel, wall temperature profiles for 10 PPI has lower value than the ones for 20 PPI in the inlet section region and the opposite is in the outlet section zone. In the inlet section 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.

Average Nusselt number values referred to the Reynolds number are reported as a function of Richardson number, in Fig. 4. It is interesting to observe that the Nu for 20 PPI and 10 PPI are higher than the ones related to the clean cases. At the increasing of the Ri values the average $Nu/Re^{0.68}$ values increase and tend to asymptotic values related to the natural convection. For low Ri values the trends are opposite and the curves tends to an and asymptotic value related to a fully developed flow in forced convection.

The heat transfer enhancement due to the presence of metal foams is highlighted in Fig. 5 where the ratio between Nusselt numbers for the cases with aluminium foam and the corresponding clean case is showed. It is noted that the use of the metal foam improve the heat transfer and the foams with 20 PPI seems more convenient.

CONCLUSIONS

An experimental investigation on mixed convection in horizontal parallel plate channels partially filled with aluminum foam was accomplished. The porous layer was set on the lower heated wall and wall temperature measurements were carried out at for configuration without, clean cases, and with the foam. The clean cases were investigated for two height of the channel and the cases with the foam for one channel gap. The experimental investigation was carried out for Reynolds number values from 100 to 300 and two wall heat flux values.

Results given in terms of wall temperature profiles and average Nusselt numbers allow to detect that the foam determined lower temperature values along the wall than the ones in clean channel cases. Foams with higher PPI determined higher average Nusselt number values and whereas the local wall temperatures showed different behaviors for the different pore per inch values. Foams with lower PPI presented a better thermal behavior at the inlet zone of the heated part of the channel whereas in the other part a better behaviors were given by the foam with higher PPI.

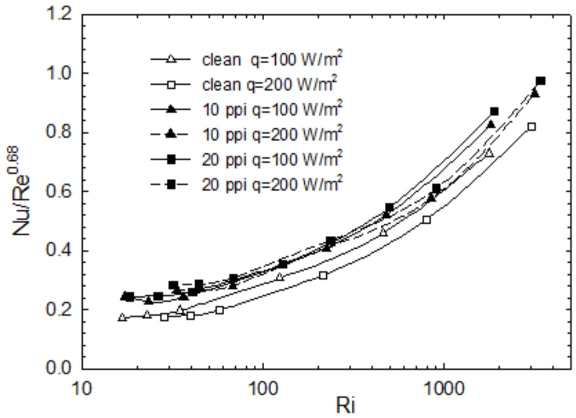

Figure 4: $Nu/Re^{0.68}$ AS A FUNCTION OF RICHARDSON NUMBER FOR CLEAN CHANNEL AND ALUMINUM FOAM PLATE WITH 10 AND 20 PPI, FOR WALL HEAT FLUX EQUAL 100 W/m^{-2} and 200 W/m^{-2}.

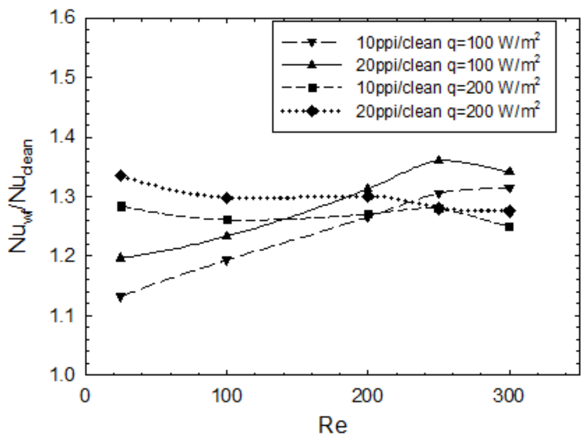

Figure 5: RATIO BETWEEN AVERAGE NUSSELT NUMBER FOR CHANNEL WITH ALUMINUM FOAM AND CLEAN CHANNEL

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