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Local Thermal Non-Equilibrium Investigation on Natural Convection in Horizontal Channel Heated from Above and Partially Filled with Aluminum Foam

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Abstract

The configuration of two horizontal parallel walls, with heated upper plate and open cavities, gets considerable attention in many thermal engineering applications. 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 local thermal non-equilibrium (LTNE) hypothesis is invoked. A three-dimensional model is realized and solved by means of the ANSYS-FLUENT code. Results are presented in terms of velocity and temperature fields and profiles, and they show that the use of porous medium improves the heat transfer in the channel due to the aluminum foam high conductivity.

Keywords: Metal foam; heat transfer; natural convection; LTNE; horizontal channel.

1. Introduction

In natural convection applications, the configuration of two horizontal parallel walls, with heated upper plate and open cavities, gets considerable attention in many thermal engineering applications, such as cooling of electronic components and devices, chemical vapor deposition systems and solar energy systems.

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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 in LTNE conditions. 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 [1-7]. Secondary flows are induced by buoyancy force due to the heating of the cavity walls, hence the local heat transfer increases [2-4, 6]. 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 [5-6]. The main flow shows a "C" loop behavior very close to the flow inside open-ended channels and open cavities [5, 8]. 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 [9]. In [7, 10], natural convection in high porosity metal foams heated from below is studied numerically and experimentally. In [11], 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 numerical simulation investigates steady laminar incompressible non-Darcian natural convection heat transfer in an enclosed cavity that is filled with a fluid-saturated porous medium and the two-equation model is used to separately account for the local fluid and solid temperature [11, 12]. Numerical results of laminar fully developed natural convection in an inclined channel partially filled with metal foam are studied in [13]. An experimental investigation of air natural convection in horizontally-positioned copper metallic foams with open cells is performed in [14]: 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 [15]. A numerical investigation of the natural convection heat transfer in a rectangular cavity filled with a heat-generating porous medium by adopting the local thermal non-equilibrium (LTNE) model is reported in [16].

Studies on partially opened cavities filled with porous media and investigations on partially filled horizontal channels with porous media are reported in [17-21].

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, in LTNE conditions. 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. Channel partially filled with metal foam configuration is compared to the configuration without foam.

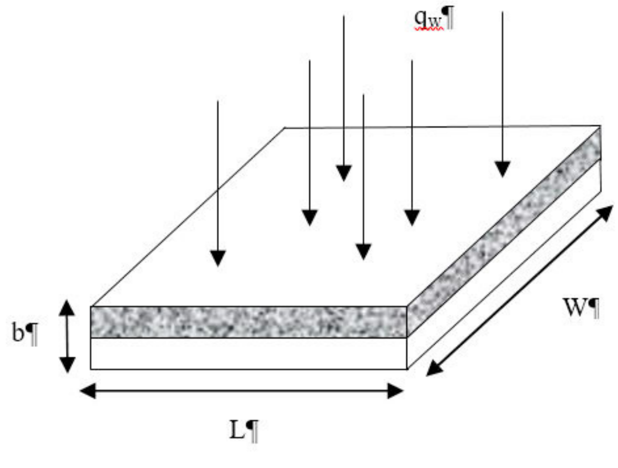

Fig. 1. Physical Domain.

Nomenclature

a	thermal diffusivity, m^2/s	p	pressure, Pa
a_{sf}	specific surface area, m^{-1}	P	dimensionless pressure
b	horizontal channel gap, m	Pr	Prandtl number
g	gravitational acceleration, m/s^2	$\dot{q}$	heat flux, W/m^2
Gr	Grashof number	Ra	Rayleigh number
h(x)	local heat transfer coefficient, W/m^2K	Ra*	modified Rayleigh number
h_{sf}	interfacial heat transfer coefficient, W/m^2K	T	temperature, K
k	thermal conductivity, W/mK	u, v, w	velocity components along x, y and z, m/s
L	plate length, m	U, V, W	dimensionless velocity components
L_x	length of the reservoir, m	x, y, z	Cartesian coordinates, m
L_y	height of the reservoir, m	X, Y, Z	dimensionless Cartesian coordinates
L_z	width of the reservoir, m	B	volumetric coefficient of expansion, 1/K
Nu(x)	local Nusselt number	ɵ	dimensionless temperature
Nu	average Nusselt number	υ	kinematic viscosity, m^2/s
		ρ	density, kg/m^3

2. Physical model description and numerical procedure

The analyzed system is tridimensional and it 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 Fig. 1, distance between the two horizontal plates is *b*, equal to 40 mm, the plate length is *L*, equal to 400 mm, as the dimension of external reservoir, and the plate width is *W*, equal to 475. 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, so two reservoirs are considered at the extremities of the channel, as suggested in reference [1, 3].

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

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{1}$$

$$\rho\frac{Du}{Dt}=\rho f_x-\frac{\partial p}{\partial x}-\frac{2}{3}\frac{\partial\left(\mu\overline{\nabla}\cdot\overline{V}\right)}{\partial x}+2\frac{\partial}{\partial x}\left(\mu\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left[\mu\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)\right]+\frac{\partial}{\partial z}\left[\mu\left(\frac{\partial u}{\partial z}+\frac{\partial w}{\partial x}\right)\right]$$

$$\rho\frac{Dv}{Dt}=\rho f_y-\frac{\partial p}{\partial y}-\frac{2}{3}\frac{\partial\left(\mu\overline{\nabla}\cdot\overline{V}\right)}{\partial y}+2\frac{\partial}{\partial y}\left(\mu\frac{\partial v}{\partial y}\right)+\frac{\partial}{\partial x}\left[\mu\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)\right]+\frac{\partial}{\partial z}\left[\mu\left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)\right] \tag{2}$$

$$\rho\frac{Dw}{Dt}=\rho f_z-\frac{\partial p}{\partial z}-\frac{2}{3}\frac{\partial\left(\mu\overline{\nabla}\cdot\overline{V}\right)}{\partial z}+2\frac{\partial}{\partial z}\left(\mu\frac{\partial w}{\partial z}\right)+\frac{\partial}{\partial x}\left[\mu\left(\frac{\partial u}{\partial z}+\frac{\partial w}{\partial x}\right)\right]+\frac{\partial}{\partial y}\left[\mu\left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)\right]$$

$$\varphi\rho_f c_{p,f}\frac{\partial T_f}{\partial t}+\rho_f c_{p,f}\left(u\frac{\partial T_f}{\partial x}+v\frac{\partial T_f}{\partial y}+w\frac{\partial T_f}{\partial z}\right)=\varphi k_f\left(\frac{\partial^2 T_f}{\partial x^2}+\frac{\partial^2 T_f}{\partial y^2}+\frac{\partial^2 T_f}{\partial z^2}\right)+h_{sf}a_{sf}\left(T_s-T_f\right) \tag{3}$$

$$\left(1-\varphi\right)\rho_s c\frac{\partial T_s}{\partial t}=\left(1-\varphi\right)k_s\left(\frac{\partial^2 T_s}{\partial x^2}+\frac{\partial^2 T_s}{\partial y^2}+\frac{\partial^2 T_s}{\partial z^2}\right)-h_{sf}a_{sf}\left(T_s-T_f\right) \quad (4)$$

The local convective heat transfer over the heated plate is described by the local Nusselt number, defined as:

$$Nu\left(x\right)=\frac{h\left(x\right)b}{k}=\frac{1}{\theta_{w,upper}\left(x\right)} \quad (5)$$

Numerical solutions of governing equations are carried out using the commercial code Ansys-Fluent 12.2 [22] reminding that they represent a set of coupled, non-linear, partial differential equations. Operating conditions are set as follow: operating temperature is equal to 300 K and 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/m^2. 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 grid-independence analysis, paying attention, in particular, to regions characterized to unexpected temperature and velocity gradients. The analysis is carried out in steady-state regime and considering metal foam with 10 PPI. Evaluated parameters are Nusselt number and 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. Percentage error is evaluated and it is chosen to compare results. In Table 1, values are reported. The choice of the mesh, used for subsequent simulations, was performed looking for the right compromise between accuracy of simulation and calculation time. Based on these two parameters, Mesh 2 is chosen. It presents a slightly higher error than Mesh 3 but, with respect to the latter, presents a lower calculation times.

Table 1. Percentage error.

Mesh	n_x	n_y	$n_{y\,foam}$	n_z	$\bar{T}_w$(plate) [K]	Nu_{avg}(plate)	$\%\Delta\bar{T}_w$(plate)	$\%\Delta Nu_{avg}$(plate)
Mesh1	30	12	16	30	462.55	1.90	0.56	2.78
Mesh2	40	16	24	40	460.62	1.91	0.14	2.00
Mesh3	60	24	32	60	459.97	1.95	0	0

3. Results and discussion

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 in LTNE condition. Different values of temperature gap between solid foam and air in the channel are assigned: 10, 20 and 30 K. For brevity, results show the effect of temperature gap equal to 20 K 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, referring to two planes: Section 1, in plane z = 0 mm, and Section 2, in plane y = 20 mm. Furthermore, both temperature and velocity profiles are shown at two different significant sections, to obtain a description of natural convection inside the open-ended cavity: Line 1, at x = 200 mm, and Line 2, at z = 0 mm.

Figures 2 and 3 show temperature fields in sections 1 and 2 with ΔT = 20 K, comparing configurations with and without foam. In Fig. 2, temperature values are similar to environmental temperature values in the inlet section of air and metal foam reduces in the channel, as shown in Fig. 3.

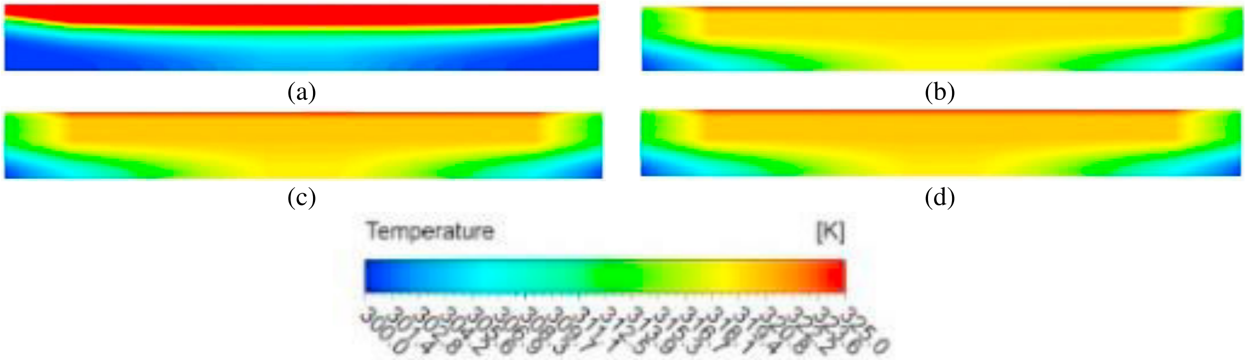

Fig. 2. Temperature field, Section 1: (a) without foam; (b) with foam, 10 PPI; (c) with foam, 20 PPI; (d) with foam, 30 PPI.

Fig. 4 shows temperature profiles along Lines 1 and 2 for different values of ΔT and when the channel is partially filled with metal foam: temperature values are lower in inlet sections of the channel, due to the contact of cold external air, then they reach the maximum in the center of heated plate.

Fig. 5 shows temperature profiles along Lines 1 and 2 for different values of PPI, comparing configuration with and without foam: temperature values change suddenly between clean zone and porous medium. It is caused by the upper heated plate, because convective effects are lowered.

Fig. 6 shows heat transfer coefficient and Nusselt number profiles: values are higher in fluid phase.

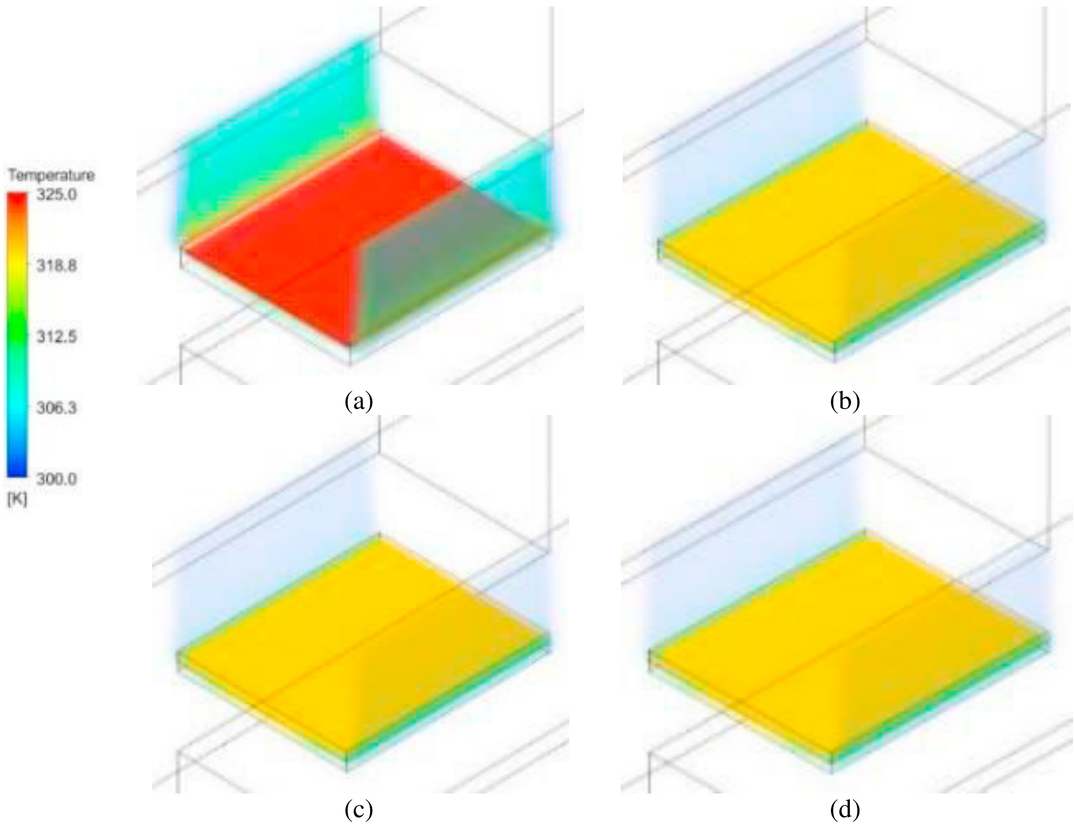

Fig. 3. Temperature field, Section 2: (a) without foam; (b) with foam, 10 PPI; (c) with foam, 20 PPI; (d) with foam, 30 PPI.

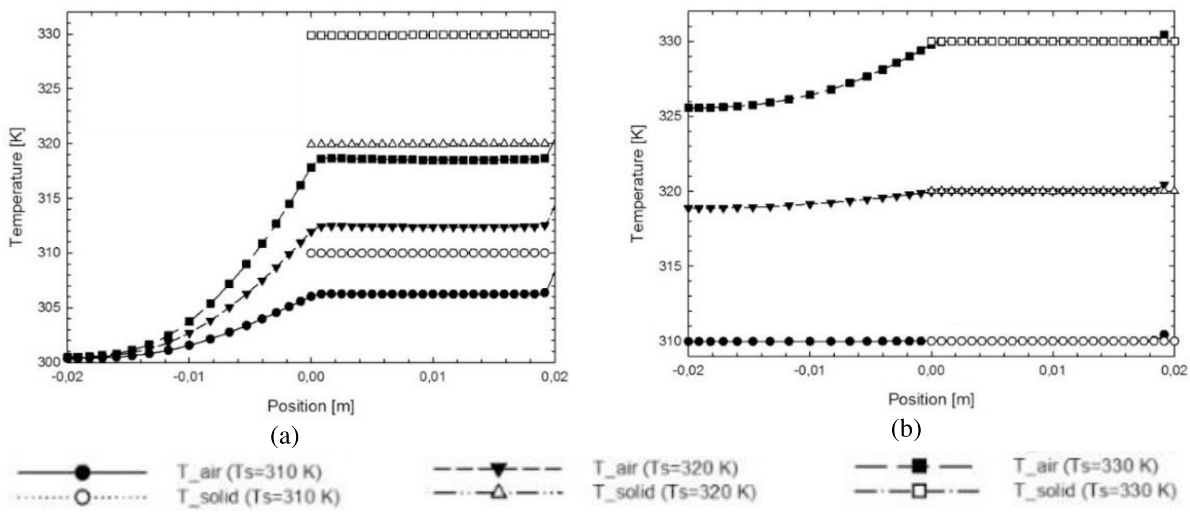

Fig.4. Temperature profile for different ΔT: (a) line 1, x = 200 mm; (b) line 2, z = 0 mm.

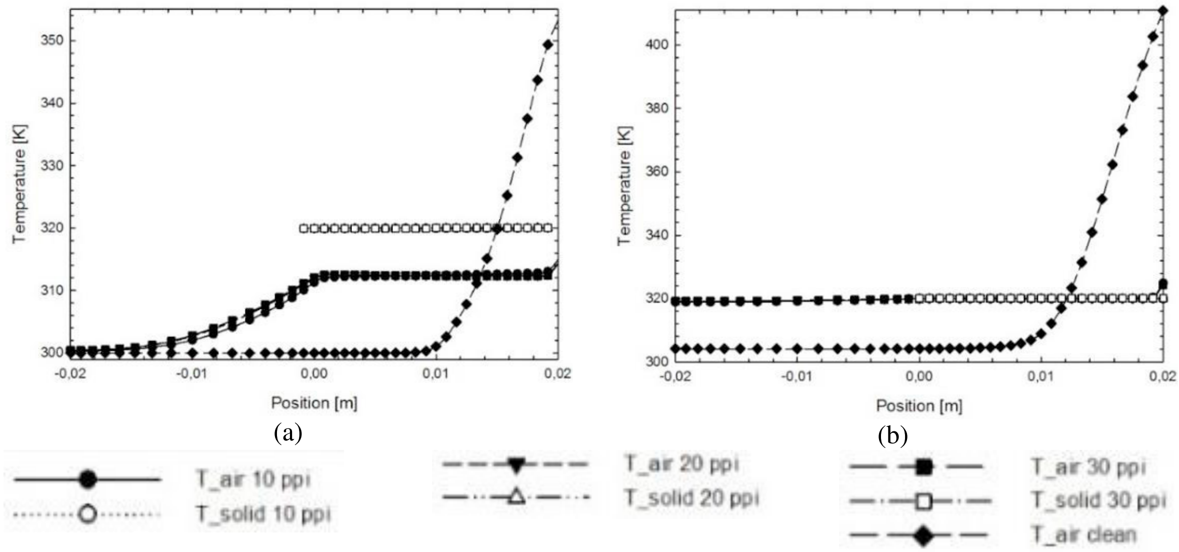

Fig.5. Temperature profile for different PPI: (a) line 1, x = 200 mm; (b) line 2, z = 0 mm.

4. 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, considering a LTNE model. A three-dimensional model is realized and a porous plate is considered near the upper heated plate. Different values of assigned temperature gap between fluid and solid phases are considered and the configuration of 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, heat transfer coefficient and average Nusselt number values are evaluated. Results show that the use of metal foams improves heat transfer, promotes the cooling of the heated wall and reduces volumes in which convective motions can evolve. 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

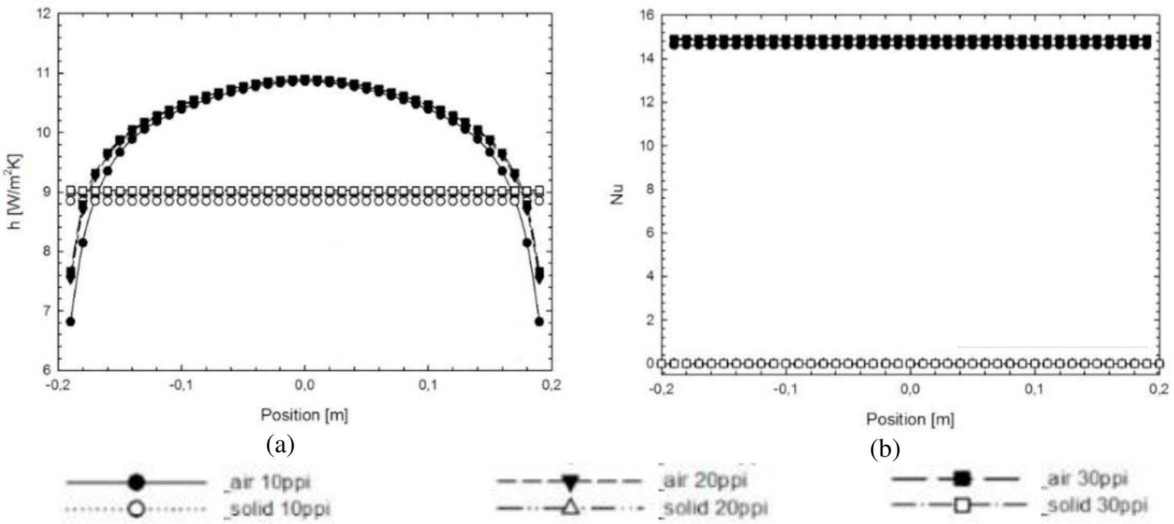

Fig.6.: (a) Heat transfer coefficient profile; (b) Nusselt number profile;

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.

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