Numerical Study of Convective Heat Transfer in a Horizontal Channel

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Abstract: This study is devoted to the investigation of natural convection in a two dimensional horizontal channel with rectangular heated blocks at the bottom. The aspect ratio of the channel is $A = L'/H' = 5$. The blocks are heated with a constant temperature while the upper plane of the channel is cold. The governing equations are solved using a finite volumes method and the SIMPLEC algorithm is used for the treatment of the pressure-velocity coupling. Special emphasis is given to detail the effect of the Rayleigh number and blocks height on the heat transfer and the mass flow rate generated by natural convection. Results are given for the following values of control parameters: Rayleigh number ($5\times10^3 \leq Ra \leq 7\times10^5$), Prandtl number Pr=0.71, opening width ($C = l'/H' = 0.15$), blocks gap ($D = d'/H' = 0.5$) and blocks height ($B = h'/H = 1/2$, 1/4 and 1/8).

Keywords: Heat transfer, horizontal channel, isothermal blocks, natural convection, chimney effect.

Nomenclature

A	aspect ratio of the channel $(= L'/H')$
B	dimensionless block height (=h'/H')
C	dimensionless opening diameter (=l'/H')
d'	space between adjacent blocks
D	dimensionless space between adjacent blocks (d'/H')
h'	block height

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H'	channel height
l'	opening diameter
L'	length of the calculation domain
M	mass flow rate (Case 1)
n	normal co-ordinate
Nu	mean Nusselt number (Case 1)
Pr	Prandtl number ($Pr = \nu/\alpha$)
Ra	Rayleigh number ($Ra = g\beta\Delta T H'^3/(\alpha\nu)$
T'	temperature of fluid
T'_H	imposed temperature on the blocks
T'_C	temperature of the cold surface
T	dimensionless temperature $[= ((T' - T'_C)/(T'_H - T'_C)]$
u', v'	velocities in x' and y' directions
$\vec{V}$	velocity vector
U, V	dimensionless velocities in x and y directions $[= (U', V') * H'/\alpha]$
x', y'	Cartesian co-ordinates
x, y	dimensionless Cartesian co-ordinates $[= (x', y')/H']$
α	thermal diffusivity
β	coefficient of volumetric thermal expansion
λ	thermal conductivity
ν	cinematic viscosity
ρ	fluid density
Ψ	dimensionless stream function

Subscripts

C	cold
d	double
eq	equivalent
h	horizontal
H	hot
max	maximum
s	single
v	vertical

1 Introduction

Heat dissipated in electronic systems is continuously increasing. It is known that a suitable thermal control of electronic components can guarantee that the operating temperature does not exceed a prescribed threshold above which system performance and reliability could be jeopardized. In many applications it is advantageous to employ natural convection, since it is cheap, maintenance and noise free and reliable (Peterson and Ortega 1990). The more frequently investigated configurations are the open cavities (Showole and Tarasuk 1993, Islam et *al.* 2008), horizontal channels (Campo et *al. 1999*), and vertical channels (Bilgen et *al.* 1995).

Packaging constraints and electronic considerations, as well as devices or system operating modes, lead to a wide variety of heat dissipation profiles along the channel walls. Many kinds of thermal wall conditions have been proposed in the literature to yield approximate conditions in prediction of thermal performance of such configurations. Along these lines, an interesting numerical study of natural convection was conducted by Penot et *al.* (2000). The authors considered a vertical channel, which simulates a chimney, placed in a closed and differentially heated cavity. Air natural convection in an asymmetrically heated channel with unheated extensions was investigated experimentally by Manca et *al.* (2002). Average Nusselt number and maximum dimensionless temperature were correlated to the Rayleigh number. A numerical investigation of free convection in a vertical isothermal channel was carried out by Desrayaud and Fichera (2002). Two rectangular blocks were symmetrically mounted on the channel surfaces.

An experimental and numerical investigation of the position effect of wall mounted rectangular blocks on the heat transfer (taking into account the angular displacement of the block) was carried out by Bilen et *al.* (2001). The experiments were conducted in a rectangular horizontal channel.

Several studies have also been performed to analyse the influence of the third dimension or other types of boundary conditions on the flow structure and heat transfer in cavities (Koo et *al.* 2005, Lappa 2005, Melnikov and Shevtsova 2005, Punjabi et *al.* 2006, Mosaad et *al.* 2005). Murakami and Mikic (2001) presented an optimisation study to determine optimum values of the channel diameter, flow rate and number of channels for minimum pressure drop. Various other strategies have been developed in the literature to enhance heat transfer, including placement of an obstacle to destabilise the flow (Lehmann and Huang 1991), using openings between blocks (Kim and Anand 2000), or variable space length between adjacent blocks (Chen et *al.* 2001).

Although these strategies can enhance the heat transfer between the blocks, the vertical planes of the blocks do not appear well ventilated in the case of an horizontal channel submitted to a convective horizontal jet. Hence, Najam et *al.* (2002, 2003a) orientated the jet perpendicularly to the horizontal channel with rectangular blocks on its lower plane. Because of the problem periodicity, the calculation domain was reduced to a cavity having the form of a “T”. The mass flow rate, generated by natural convection in such configuration, was neglected. In a previous study dealing with natural convection in the same geometry (without openings), Amahmid et *al.* (1999) demonstrated that there exist situations in which the effect of the domain must be taken into account and there is no periodicity in the problem solution.

In this work, we extend the calculation domain to a finite horizontal channel formed by five T – cavities. We evaluate heat transfer and mass flow rate induced by natural convection in such cavity.

2 Physical problem and governing equations

The system under consideration is a two dimensional horizontal channel containing heating blocks regularly distributed on its lower adiabatic wall and maintained at a constant hot temperature T'_H (fig. 1). The upper wall of the channel is maintained at the cold temperature T'_C, while the other sides of the channel are insulated.

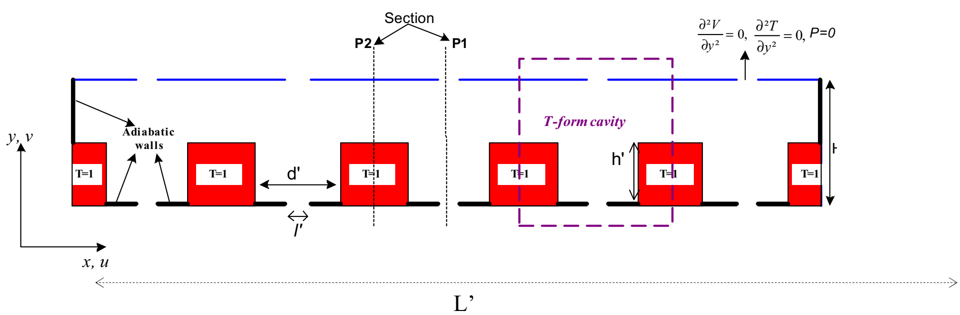


Figure 1: Model of the horizontal channel with blocks and openings

The flow is considered steady, laminar, and incompressible with the Boussinesq approximation. The dimensionless governing equations can be written as follows:

Continuity equation :

$$\vec{\nabla}\cdot\vec{V}=0 \tag{1}$$

Momentum equation:

$$\frac{\partial\vec{V}}{\partial t}+(\vec{V}\cdot\vec{\nabla})\vec{V}=-\vec{\nabla}P+\mathrm{Pr}\,\nabla^2\vec{V}+Ra\,\mathrm{Pr}\frac{\vec{g}}{g}T \tag{2}$$

Energy equation:

$$\frac{\partial T}{\partial t} + \vec{V} \cdot \vec{\nabla} T = \Delta T \tag{3}$$

Referring to fig. 1, the dimensionless variables are:

$$\begin{aligned} &x = \frac{x'}{H'}, \quad y = \frac{y'}{H'}, \quad u = u'\frac{H'}{\alpha}, \quad v = v'\frac{H'}{\alpha}, \\ &T = \frac{T' - T'_C}{T'_H - T'_C}, \quad P = \frac{(P' + \rho g y')H'^2}{\rho \alpha^2} \end{aligned} \tag{4}$$

$Ra = \frac{g\beta\delta T H'^3}{\alpha\nu}$, $Pr = \frac{\nu}{\alpha}$ with $(\delta T = T'_H - T'_C)$.

The imposed boundary conditions, in terms of pressure and velocity, are similar to those traditionally used for natural convection flow in a vertical channel (thermosiphon), (Penot and Dalbert 1983, Chappidi and Eno 1990): $T = u$=0, $v = \frac{M}{C}$, $P = -\frac{M^2}{2H^2}$ at the inlet openings. T =1 on the block surfaces and T=0 on the upper horizontal solid walls of the cavity. The other thermal boundary conditions are shown in figure 1.

At the evacuation openings, T, u and v are extrapolated by adopting a process similar to that used in references (Tomimura and Fujii 1988, Najam et *al.* 2003b), (their second spatial derivative terms in the vertical direction are equal to zero, $\frac{\partial^2 T}{\partial y^2}$=0, $\frac{\partial^2 u}{\partial y^2}$=0, $\frac{\partial^2 v}{\partial y^2}$=0).

The mean Nusselt number over the active walls of the blocks is calculated as:

$$Nu = \int_{hotwalls} \frac{\partial T}{\partial n} d\tau$$

(n and τ are the normal and tangential co-ordinates, respectively)

The mass flow rate is calculated as:

$$M = \int_{outletopenings} v dx$$

3 Numerical method

Due to the periodic nature of the geometry under consideration, fast calculations were often restricted to an elementary domain (Bilgen et *al.* 1995, Najam et *al.* 2003a). Hence, we will consider the extended calculation domain shown in fig. 1. The other important objective is to illustrate the effect of the principal parameters on heat transfer and mass flow rate generated by natural convection.

Table 1: comparison of our results and those of De Val Davis (1983) and Le Quéré (1985)

	De Val Davis 1983	**Le Quéré 1985**	**Present study**	**Maximum deviation**
$Ra = 10^4$	$\Psi_{\max} = 5.098$	$-------$	$\Psi_{\max} = 5.035$	1.2%
$Ra = 10^5$	$\Psi_{\max} = 9.667$	$-------$	$\Psi_{\max} = 9.725$	0.6%
$Ra = 10^6$	$\Psi_{\max} = 17.113$	$\Psi_{\max} = 16.811$	$\Psi_{\max} = 17.152$	0.2%
$Ra = 10^7$	$-------$	$\Psi_{\max} = 30.170$	$\Psi_{\max} = 30.077$	0.3%

Table 2: Comparison of our results and those of Desrayaud and Fichera (2002)

$Ra = 10^5$ $(A = 5)$	**Desrayaud and Fichera (2002)**	**Present study**	**Maximum deviation**
$\Psi_{\max}$	151.51	152.85	0.9%
M	148.27	151.72	2.2%

The governing equations of the problem were solved, numerically, using a finite volumes method. The QUICK scheme developed by Leonard (1979) was used for the discretization of all convective terms of the advective transport equations. The coupling pressure-velocity was solved using the SIMPLEC (SIMPLE Consistent) algorithm (Van Doormaal and Raithby 1984). Time steps considered were ranging between 10^{-5} and 10^{-4}. The accuracy of the numerical model was verified by comparing our results with those obtained by De Vahl Devis (1983) and Le Quéré et *al.* (1985), for natural convection in a differential heated cavity (table 1), and then with the results obtained by Desrayaud and Fichera (2002), in a vertical channel with two ribs, symmetrically, placed on the channel walls (table 2). Good agreement in Ψ_{max} and M values was obtained.

When the steady state is reached, all the energy furnished by the hot walls to the fluid must leave the cavity through the cold surface (with openings). This energy balance was verified by less than 3% in all cases considered here.

A grid refinement study was conducted to ensure that the solutions are grid independent. Six uniform grid sizes were evaluated as shown in table 3. Differences between the maximum of the stream function $\Psi_{\max}$ and the average Nusselt number Nu are, respectively, 0.6% and 0.4% for the 350×70 and 400×80 grid sizes. For the 400×80 and 450×90 grid sizes they are reduced to 0.1%. According to such study, the 400×80 grid size has been used for the problem simulation.

Table 3: Grid sensitivity tests, $Ra = 10^5$

Grid size	$\Psi^*_{\max}$	Nu^*
250×50	15.3038607	8.84438324
300×60	13.6698637	8.68144894
350×70	12.2157311	8.43402702
400×80	**12.1455757**	**8.40323635**
450×90	12.1291549	8.39295971
500×100	12.1034937	8.38846588

$\Psi^*_{\max}$ and Nu^* are related to an individual T-form cavity

4 Results and discussion

The effect of Rayleigh number on the flow structure and heat transfer has been analyzed for different blocks height. In this section, the control parameters considered are C=0.15, D=0.5, B=1/2, 1/4, 1/8 and $5\times10^3 \leq Ra \leq 7\times10^5$.

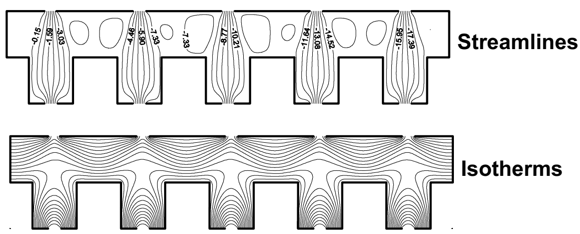
The flow structures is, essentially, composed of open lines, which represent the aspired air by vertical natural convection (thermal drawing), and closed cells which are due to the Rayleigh-Bénard mechanism or to the recirculating movement inside the jet of fresh air.

4.1 Streamlines and isotherms

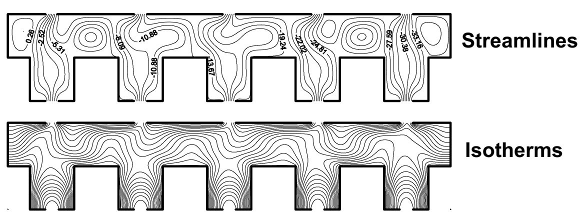
4.1.1 Case of B=1/2

When the Rayleigh number value is less than 5×10^3, the heat transfer is controlled, essentially, by conduction (results not shown here). For $Ra \geq 5\times10^3$, the flow structure and the thermal fields are presented by streamlines and isotherms in figs. 2-a,b,c. These figures show that the cells appear just above the blocks or inside the jet of aspired air. In the first case, the cells existence is due to the Rayleigh-Bénard convection, developed between the horizontal sides of the blocks (called: horizontal active walls) and the cold wall of the channel. In the second case the cells appear inside the jet because of the recirculating flow. Hence, two kinds of solutions are obtained in the range $10^4 \leq Ra \leq 7\times10^5$: when Ra is less than 3×10^4($10^4 \leq Ra \leq 2\times10^4$), the open lines (fresh air) pass between closed cells. This solution is called *Intra-Cellular Flow* (ICF), fig. 2-a. It is symmetric with respect to the vertical section P_1. The corresponding isotherms show that the major part of heat exchange occurs through the upper openings. They are not very distorted and present extrema at the planes P_2. We remark, in this case, that the active surfaces of the blocks are not well ventilated. For $Ra = 3 \times 10^4$, the flow structure is no longer periodic with breaking of the solution symmetry (fig. 2-b). This solution is called *Mixed-Solution* (MS) because it presents convective cells inside and outside the open lines.

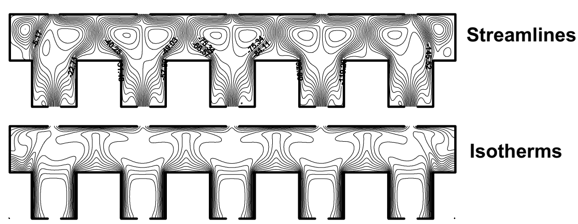
In the range $4 \times 10^4 \leq Ra < 6 \times 10^5$, the recirculation cells appear inside the jet as shown in figs. 2-c. This solution is called *Extra-Cellular Flow* (ECF). In this case, the fresh air, aspired by natural convection, is in direct contact with the cold plane of the channel. So the isotherms are very tight near this wall, and this means that heat leaves the channel through the rigid plane and through the upper openings, homogeneously. Remarkably, the isotherms exhibit the appearance of the *chimney effect* as it was defined by Bade (1994): the flow structure resembles the separated boundary layer flow that exists in the case of the vertical plane problem. The isotherms are very tight near the vertical walls of the blocks and they are, practically, horizontal in the jet zone (air is stratified in this zone). The blocks are well



(a) $Ra = 10^4$



(b) $Ra = 3 \times 10^4$



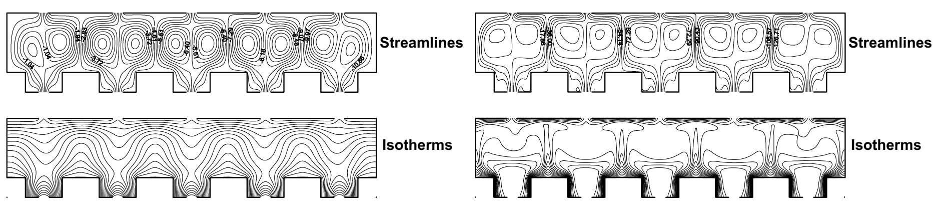
(c) $Ra = 5 \times 10^5$

Figure 2: Streamlines and isotherms for $Ra = 10^4$ 3×10^4 and 5×10^5 ; B=1/2

ventilated and this solution seems to be the best for heat exchange. When *Ra* exceeds 7×10^5, the flow becomes unsteady with the development of oscillations of the thermal and flow fields.

4.1.2 Case of B=1/4 and 1/8

The only particularity, in these cases, is the absence of the ICF structure. We note that the cell sizes increase when reducing the blocks height *B*. For *B*=1/4, the flow structure is of the ECF kind as presented in figs. 3-a and 3-b, respectively, for $Ra = 10^4$ and 5×10^5. The problem solution is nearly symmetric. The isotherm lines show that the heat exchange is weak near the horizontal faces of the blocks and that is important near the vertical ones. The zones around the middle of the T – cavities tend to be more stratified than in the case *B*=1/2. The symmetry of the solution is conserved. For $Ra = 5\times10^5$, the isotherms show that all the active walls of the blocks are well ventilated because the development of the aforementioned *chimney effect*. For *B*=1/8, the major remark is that the cell sizes are large in this case (fig. 4-a and 4-b for $Ra = 10^4$ and 4×10^5). Consequently, the fresh air aspired by natural convection is obliged to flow along the active walls of the blocks and heat exchange is very important through the block surfaces as shown by the corresponding isotherms. The solution given in fig. 4-b for $Ra = 4\times10^5$, is the limit of the steady states.



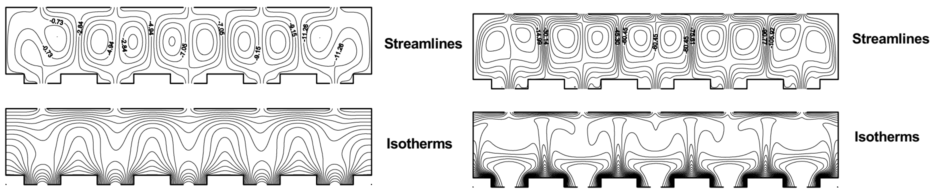
(a) $Ra = 10^4$ (b) $Ra = 5\times10^5$

Figure 3: Streamlines and isotherms (a) $Ra = 10^4$, (b) $Ra = 5\times10^5$, *B*=1/4

4.2 Heat transfer and mass flow rate

The average Nusselt number variation according to the Rayleigh number *Ra* is presented in fig. 5 for different values of B. Generally, the Nusselt number increases (with practically linear tendency in logarithm scales) with Ra and decreases when B is reduced. This decrease is not due solely to the reduction of the hot surface, but also to the lower thermal drawing between blocks.

It should be noted that the gap between the *Nu* curves increases gradually with Ra.



(a) $Ra = 10^4$ (b) $Ra = 4 \times 10^5$

Figure 4: Streamlines and isotherms (a) $Ra = 10^4$, (b) $Ra = 5 \times 10^5$, B=1/8

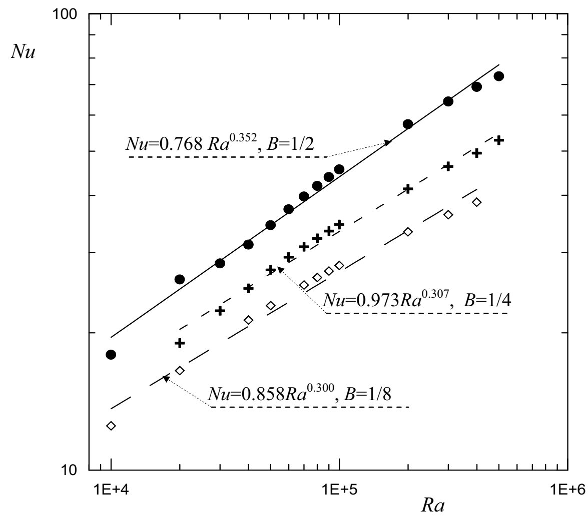


Figure 5: Variation of mean Nusselt number according to *Ra* for *B*=1/2, 1/4 et 1/8

In the range of low values of *Ra* ($Ra \leq 2 \times 10^4$) the numerical values of *Nu* are less than those given by the correlation. The *Nu* correlations with *Ra* are:

$$Nu = 0.768Ra^{0.352}, \quad B = 1/2 \tag{5}$$

$$Nu = 0.973Ra^{0.307}, \quad B = 1/4 \tag{6}$$

$$Nu = 0.858Ra^{0.300}, \quad B = 1/8 \tag{7}$$

They have been obtained with maximum deviation less than 6%. There exist some similar relations in the literature in the cases of vertical channel without blocks ($Nu = 0.62 \times Ra^{1/4}$, Sparrow and Azevedo 1985), and with blocks ($Nu = 0.45 \times Ra^{0.29}$, Kwak and Song 1998).

The other outcome of the problem is the rate of induced mass flow. In fig. 6, we present *M* variation with *Ra* for different values of *B*. The mass flow rate increases with *Ra*. The mass flow rate correlations with *Ra* are shown below:

$$M = 0.027Ra^{0.541}, \quad B = 1/2 \tag{8}$$
$$M = 0.022Ra^{0.550}, \quad B = 1/4 \tag{9}$$
$$M = 0.012Ra^{0.596}, \quad B = 1/8 \tag{10}$$

The maximum deviation is less than 3% for all these cases.

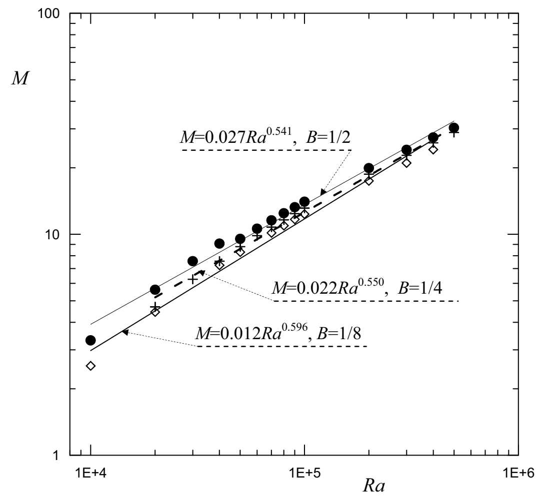


Figure 6: Variation of Mass flow rate according to *Ra* for *B*=1/2, 1/4 et 1/8

5 Conclusion

A numerical investigation has been conducted to evaluate heat and mass flow rate induced by natural convection in a finite channel with heated rectangular blocks at the bottom. The study has considered parametric variations of the Rayleigh number and the blocks height.

The results show the existence of different solutions of the problem (ICF, ECF and MS) on which the resulting heat transfer and mass flow rate depend significantly. ICF and ECF solutions are symmetric and present a periodicity for the Three T – cavities inside the channel. The mixed solution (MS) is no longer periodic with breaking of the solution symmetry. In this case, the results are similar to those of Amahmid et *al.* (1999) for the same problem without openings. Remarkably, the ICF solution indicates the development of the *chimney effect*. It allows a good ventilation of the active planes of the blocks.

The mean Nusselt number increases with Ra and decreases when B is reduced. Nu correlations with Rayleigh number have been obtained. They are similar to those found in the case of a vertical smooth or ribbed channel.

Like the Nusselt number, the Mass flow rate increases with Ra. Correlations with Ra of this fundamental unknoun of the problem have been also given.

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