Electronic components cooling by natural convection in horizontal channel with slots

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Abstract

A numerical study of natural convection from a two dimensional horizontal channel with rectangular heated blocks is performed. Because of the problem periodicity, the studied domain is reduced to a “T” form cavity that presents a symmetry with respect to a vertical axis passing through the middle of the openings. The governing equations are solved using a control volume method, and the SIMPLEC algorithm is used for treatment of the pressure-velocity coupling. Special emphasis is given to detail the effect of the blocks spacing (gap) on the heat transfer and the mass flow rate generated by the natural convection. The results are given for the parameters of control as, Rayleigh number \( (10^4 \leq Ra \leq 8 \times 10^5) \), Prandtl number \( (Pr = 0.72) \), opening width \( (C = l'/H' = 0.15) \), blocks gap \( (0.15 \leq D \leq 1.0) \) and the blocks height \( (B = 0.5) \). The results show that the flow structure and the heat transfer depend significantly on the control parameters. Two principal kinds of problem solution are raised.

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Nomenclature

\( B \) dimensionless block height \((h'/H')\)
\( C \) dimensionless opening diameter \((l'/H')\)
\( d' \) space between adjacent blocks
\( D \) dimensionless space between adjacent blocks \((d'/H')\)
\( g \) gravitational acceleration
\( h' \) block height
\( H' \) channel height
\( H \) dimensionless channel height
\( l' \) opening diameter
\( L' \) length of calculation domain (cavity)
\( M \) mass flow rate (Eq. (5))
\( n \) normal coordinate
\( Nu \) mean Nusselt number (Eq. (6))
\( Nu_h \) mean Nusselt number along horizontal planes of blocks
\( Nu_v \) mean Nusselt number along vertical planes of blocks
\( P \) pressure of fluid
\( P' \) dimensionless pressure
\( Pr \) Prandtl number \((Pr = v/\alpha)\)
\( Ra \) Rayleigh number \((Ra = g\beta\Delta T H^3/\nu \nu)\)
\( T \) temperature of fluid
\( T_H \) imposed temperature on blocks
\( T_C \) temperature of cold surface
\( T' \) dimensionless temperature of fluid \(\left[=\left(\left(T' - T_C\right)/(T_H - T_C)\right)\right]\)
\( U', V' \) velocities in \(x'\) and \(y'\) directions
\( U, V \) dimensionless velocities in \(x\) and \(y\) directions \(\left[=(U', V')*H'/\nu\right]\)
\( x', y' \) Cartesian coordinates
\( x, y \) dimensionless Cartesian coordinates \(\left[=(x', y')/H'\right]\)
\( \alpha \) thermal diffusivity
\( \beta \) volumetric coefficient of thermal expansion
\( \lambda \) thermal conductivity of fluid
\( \nu \) kinematic viscosity of fluid
\( \rho \) fluid density
\( \Psi \) dimensionless stream function

Subscripts
\( C \) cold
\( H \) hot
\( \text{max} \) maximum
1. Introduction

The passive character of cooling by natural convection makes it very attractive for application in electronic devices. Ever increasing cooling requirements necessitate, however, the search for methods of intensification of natural convection, especially in geometries found in electronic devices. Cooling of computer boards can be studied by idealizing them as forming horizontal or vertical channels. Evaluation of the heat transfer increase induced by natural convection in a horizontal channel is the objective of the present study.

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 are proposed to yield approximate conditions for prediction of the thermal performance of such configurations [1]. Along the same lines, a numerical study of natural convection was conducted by Penot et al. [2]. The authors considered a vertical channel, which simulates a chimney, placed in a closed and differentially heated cavity. The chimney walls were capped isotherms. Air natural convection in an asymmetrically heated channel with unheated extensions has been investigated experimentally by Manca et al. [3]. Average Nusselt number and maximum dimensionless temperature are correlated to the Rayleigh number. A numerical investigation of free convection in a vertical isothermal channel was conducted by Desrayaud and Fichera [4]. Two rectangular blocks are symmetrically mounted on the channel surfaces. An experimental and numerical investigation about the effect of the position of wall mounted rectangular blocks on the heat transfer taking into account the angular displacement of the block was conducted by Bilen et al. [5]. The experiments were conducted in a rectangular horizontal channel. Murakami and Mikic [6] presented an optimization study using a method of determining the optimum values of channel diameter, flow rate and number of channels for minimum pressure drop. Various strategies were developed to enhance the heat transfer, including placement of an obstacle in the flow path of the coolant to destabilize the flow [7], using openings between blocks in recirculating movement spaces [8] or variable space length between adjacent blocks [9].

Although these strategies enhanced the heat transfer between the blocks, the vertical planes of the blocks do not appear well ventilated in the case of a horizontal channel submitted to a convective horizontal jet. Hence, Najam et al. [10,11] send the jet perpendicularly to the horizontal channel with rectangular blocks on its lower plane. Because of the problem periodicity, the calculation domain is reduced to a “T” form cavity. The mass flow rate generated by the natural convection in such a configuration was not considered in these studies. Their objective was to study the interaction between the forced flow and the convective cells (mixed convection) and its effect on heat transfer for a fixed mass flow rate (debit) at low values of Reynolds number ($1 \leq Re \leq 100$). However, for this range of $Re$, we consider that the debit of air aspired by natural convection can not be neglected. It can be greater than the fixed rate (in mixed convection) for a same value of $Ra$. The aim of this work is to examine the natural convection contribution to heat transfer and mass flow rate in the same configuration. The mass flow rate is considered as a fundamental unknown of the problem and is not imposed at the inlet opening.
2. Physical problem and governing equations

The geometry of the problem investigated herein is depicted in Fig. 1(a). The system is made of a horizontal parallel plate channel with rectangular heated blocks placed on the lower plate. Openings are provided on the plates as shown in this figure. Because of the geometric periodicity of the channel, the studied domain is reduced to a “T” form cavity, Fig. 1(b). The blocks are heated at a constant temperature $T_H$. The upper plate is cold at a temperature $T_C$, while the other sides of the cavity are insulated.

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

$$\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} = 0$$  \hspace{1cm} (1)

$$\frac{\partial U}{\partial t} + U \frac{\partial U}{\partial x} + V \frac{\partial U}{\partial y} = -\frac{\partial P}{\partial x} + Pr \left( \frac{\partial^2 U}{\partial x^2} + \frac{\partial^2 U}{\partial y^2} \right)$$  \hspace{1cm} (2)

$$\frac{\partial V}{\partial t} + U \frac{\partial V}{\partial x} + V \frac{\partial V}{\partial y} = -\frac{\partial P}{\partial y} + Pr \left( \frac{\partial^2 V}{\partial x^2} + \frac{\partial^2 V}{\partial y^2} \right) + Pr Ra T$$  \hspace{1cm} (3)

$$\frac{\partial T}{\partial t} + U \frac{\partial T}{\partial x} + V \frac{\partial T}{\partial y} = \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}$$  \hspace{1cm} (4)

![Diagram of channel with openings and calculation domain](image)

Fig. 1. (a) Model of channel with openings and (b) calculation domain.
Referring to Fig. 1, the dimensionless variables are: 
\[ x = \frac{x}{H_0}, \quad y = \frac{y}{H_0}, \quad U = \frac{U}{U_0}, \quad V = \frac{V}{U_0}, \quad T = \frac{T - T_C}{T_H - T_C}, \]
\[ P = \left( p + \rho g y H \right) \frac{H}{\rho^2} \quad Ra = \frac{g \beta \Delta H}{\nu^2}, \quad Pr = \frac{\nu}{\alpha} \text{ with } (\Delta T = T_H - T_C). \]

The imposed boundary conditions in terms of pressure and velocity are similar to those of natural convection flow in a vertical channel [4,12]:

- \( T = 1 \) on the block surfaces, and \( T = 0 \) on the upper horizontal wall of the cavity;
- \( U = V = 0 \) on all the rigid walls;
- \( T = U = \frac{\partial T}{\partial x} = 0 \) at the inlet opening;
- \( \frac{\partial T}{\partial x} = 0 \) for \( 0.5 \leq y \leq 1 \); \( x = 0 \) and \( 1 \);
- \( \frac{\partial T}{\partial y} = 0 \) for \( y = 0 \); \( 0.25 \leq x \leq x_0 \) and \( x_1 \leq x \leq x_2 \);
- \( P = 0 \) for \( y = 1 \); \( x_0 \leq x \leq x_1 \) (outlet opening);

where \( x_0 = \frac{D + 0.35}{2}; \ x_1 = \frac{D + 0.65}{2} \) and \( x_2 = D + 0.25 \); \( M \) is the induced mass flow rate. It is calculated as:
\[ M = \int_{x_0}^{x_1} V \big|_{y=1} \, dx \quad (5) \]

At the evacuation opening, \( T, U \) and \( V \) are extrapolated by adopting similar processes to those shown in Ref. [13] (their second spatial derivative terms in the vertical direction are equal to zero).

The mean Nusselt number over the active walls of the blocks is:
\[ Nu = \int_0^{0.25} \frac{\partial T}{\partial y} \big|_{y=0.5} \, dx + \int_0^{0.5} \frac{\partial T}{\partial x} \big|_{x=0.25} \, dy + \int_0^{0.5} \frac{\partial T}{\partial x} \big|_{x=x_2} \, dy + \int_{x_2}^{1} \frac{\partial T}{\partial y} \big|_{y=0.5} \, dx \quad (6) \]

(i) and (jj) represent the mean Nusselt number along the horizontal active planes of the blocks \( Nu_h \). The terms (ii) and (j) represent that along the vertical planes \( Nu_v \).

3. Numerical method

The governing equations of the problem were solved numerically using a control volume method [14]. The QUICK scheme [15] was adopted for discretization of all convective terms of the advective transport equations (Eqs. (2)–(4)). The final discretized forms of Eqs. (1)–(4) were solved by using the SIMPLC (SIMPLE consistent) algorithm [16]. As a result of a grid independence study, a grid size of 81 \( \times \) 81 was found to model accurately the flow fields described in the corresponding results. The time steps considered range between \( 10^{-5} \) and \( 10^{-4} \). The accuracy of the numerical model was verified by comparing the results from the present study with those obtained by De Vahl Davis [17] for natural convection in a differential heated cavity, Table 1,

| \( Ra = 10^4 \) | \( \Psi_{\text{max}} = 5.098 \) | \( \Psi_{\text{max}} = 5.035 \) | 1.2 |
| \( Ra = 10^6 \) | \( \Psi_{\text{max}} = 17.113 \) | \( \Psi_{\text{max}} = 17.152 \) | 0.2 |
and then with the results obtained by Desrayaud and Fichera [4] in a vertical channel with two ribs symmetrically placed on the channel walls, Table 2. We note that good agreement was obtained in the \( W_{\text{max}} \) and \( M \) terms. When a steady state is reached, all the energy furnished by the hot walls to the fluid must leave the cavity through the cold surface (with the opening). This energy balance was verified within less than 3% in all cases considered here.

4. Results and discussion

In this section, the heat transfer rate across the cold wall and the hot walls and the flow and temperature fields are examined for the blocks gap range of \( 0.15 \leq D \leq 1.0 \), Rayleigh number \( Ra = 10^5 \) and other parameters of the problem \( (B = 0.5; C = 0.15, Pr = 0.72) \). We note that the Rayleigh number value is chosen in favour of natural convection.

The particularity of this problem is the appearance of different solutions when varying the parameter \( D \). The flow structure 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 recirculating movement inside the jet of fresh air or to the Rayleigh–Bénard convection.

4.1. Flow structure and isotherms

The flow structure and the thermal field are, respectively, presented by the streamlines and isotherms in Fig. 2(a–d). These figures show that the cells appear just up to 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 planes of the blocks (called horizontal active walls) and the cold wall of the cavity. In the second case, the cells appear inside the jet because of the recirculating flow. Hence, three kinds of problem solution are obtained in the range \( 0.15 \leq D \leq 1.0 \): when \( D \) is less than 0.4 (\( 0.15 \leq D \leq 0.3 \)), the open lines pass between the closed cells. This solution is called intra-cellular flow (ICF), Fig. 2(a). The corresponding isotherms show that the major part of the heat exits through the upper opening. In the range \( 0.4 \leq D \leq 0.8 \), the recirculating cells appear inside the jet as shown in Fig. 2(b) and (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 cavity. So, the isotherms are too tight near this wall, and heat leaves the cavity through the rigid plane and through the opening, homogeneously. For \( D \geq 0.9 \), three principal cells constitute the problem solution: two cells inside the jet and the other one outside it. This latter appears above the block in the right Fig. 2(d), for \( D = 1.0 \), or in the left (figure not presented). Two secondary recirculating cells exist near the inlet opening. This solution is called asymmetrical solution (AS). We note that the horizontal hot wall of the block on the right is more ventilated than that of the other block on the left as shown by the corresponding isotherms. That is because of the existence of this third cell.

<table>
<thead>
<tr>
<th>( Ra = 10^5 ) (( A = 5 ))</th>
<th>Desrayaud and Fichera [4]</th>
<th>Present study</th>
<th>Maximum deviation (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>( W_{\text{max}} )</td>
<td>151.51</td>
<td>152.85</td>
<td>0.9</td>
</tr>
<tr>
<td>( M )</td>
<td>148.27</td>
<td>151.72</td>
<td>2.2</td>
</tr>
</tbody>
</table>
The existence of these different solutions is principally due to conflict between vertical natural convection, which developed near the vertical planes of the blocks, Rayleigh–Bénard convection, which is due to the vertical thermal gradient, and the solid–fluid friction along the adiabatic vertical walls. When $D$ is less than 0.9, all the solutions are symmetric with respect to the vertical axis passing through the middle of the openings. For $D \geq 0.9$, the solution symmetry is destroyed. For relatively high values of $D$ ($D \geq 0.5$), the isotherms show the appearance of the chimney effect as defined in Ref. [18], and the flow structure is analogous to the separated boundary layer flow, which exists in the case of the vertical plane problem. The isotherms are too tight near the vertical walls of the blocks, and they are practically horizontal in the cell zones (air is stratified in this area). Secondary cells appear just near the jet inlet: the flow coming in the cavity vertically
through the inlet opening is more and more relaxed by increasing $D$. So, to return to the boundary layer, near the vertical walls where the pressure drop is weak, it creates a recirculation movement.

4.2. Heat transfer and mass flow rate

The mean Nusselt number variation with $D$ is presented in Fig. 3. Generally, $Nu$ is weak in the range of low values of $D$ ($D \lesssim 0.4$) and increases linearly with $D$ when $D$ exceeds 0.4. This increase (in the range $0.4 \leq D \leq 1.0$) is due to the appearance of the boundary layer flow along the vertical planes of the blocks. Consequently, the vertical active planes are well ventilated. Note that they represent $\frac{2}{3}$ of all the active heated surfaces. The Nusselt number presents a minimum at $D \simeq 0.4$. This value of $D$ is related to the limit of the ICF existence domain. This solution seems to be unfavourable to the heat transfer. The average Nusselt numbers along the vertical planes, $Nu_v$ ((ii) and (j) Eq. (6)), and along the horizontal planes of the blocks, $Nu_h$ (terms (i) and (jj)), are calculated. They are graphed versus $D$ in Fig. 4. The most distinguishing characteristic of Fig. 4 is that $Nu_v$ increases with $D$ and, contrarily, $Nu_h$ decreases. In the case of the ICF solution, the vertical active planes are not well ventilated at low values of $D$, since there is not good contact between these planes and the major part of the fresh air (which leaves the cavity rapidly), while the horizontal planes are relatively well ventilated because of their contact with the Rayleigh–Bénard cells. When $D$ increases, the chimney effect appears, and the fresh air passes close the vertical active walls. In this case, the flow separates just up from the blocks, and so, the horizontal active walls are not well ventilated.

The mean Nusselt number is correlated to the block spacing $D$ as follows:

$$Nu = 21.60 \times D - 1.41 \quad \text{for} \quad D \geq 0.4$$

(7)
The maximum deviation is less than 4%.

Another outcome of the problem is the rate of the induced mass flow, $M$. In Fig. 5, we present the $M$ variation with $D$. Contrarily to the $Nu$ curve, the mass flow rate is important at lower val-

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**Fig. 4.** $Nu_h$ and $Nu_v$ variations with $D$.

**Fig. 5.** Mass flow rate variation with $D$. 

$M = -5.85D + 20.77$
ues of $D$ and presents a maximum for $D \approx 0.3$. In the range $0.15 \leq D \leq 0.3$, the thermal drawing is important because the vertical active planes are closed. The following relation is linking $M$ and $D$:

$$M = -5.65 \times D + 20.78 \quad \text{for } D \geq 0.4$$

with a maximum deviation less than 6%.

5. Conclusion

A numerical investigation was conducted to study the enhancement of heat transfer in a cavity with heated rectangular blocks. The results show the existence of different solutions of the problem (ICF, ECF and AS) on which the resulting heat transfer and mass flow rate depend significantly. The mean Nusselt number presents a minimum at $D \approx 0.4$, the value of $D$ for which the ICF solution disappears and the ECF one appears. For $D$ greater than 0.4, the $Nu$ increases linearly with $D$, and a correlation is proposed (Eq. (7)). The horizontal planes of the blocks are ventilated at low values of $D$, and the vertical ones are well ventilated at high $D$.

The mass flow rate variation with $D$, contrarily to the $Nu$ variation, presents a maximum at $D \approx 0.3$ and then decreases as correlated by Eq. (8). This relation shows that the debit of air aspirated by natural convection is not negligible in the considered range of Rayleigh number ($15 \leq M \leq 25$).

The ECF solution provides good ventilation of the vertical active planes of the blocks because of the appearance of the chimney effect. This solution is useful for heat exchange. It is noteworthy that the heat transfer is enhanced by increasing the space between the blocks.

References


