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Convective Mixed Heat Transfer in a Square Cavity with Heated Rectangular Blocks and Submitted to a Vertical Forced Flow

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Abstract: A numerical mixed convection investigation was carried out to study the enhancement of heat transfer in a square cavity with identical heated rectangular blocks adjacent to its upper wall, and submitted to a vertical jet of fresh air from below. The configuration so defined is an inverted "T"-shaped cavity presenting symmetry with respect to a vertical axis passing by the middle of the openings. The governing equations have been solved using the finite difference method. The parameters of this study are: Rayleigh number $10^4 \leq Ra \leq 10^6$, Reynolds number $1 \leq Re \leq 1000$, the opening width C=0.15, the height of blocks B=1/4, 1/2 and 3/4. The non-dimensional space between blocks and geometric parameter of the cavity are kept constant (D=0.5 and A=1). The results obtained, with Pr=0.72, show the validity of different solutions, when varying Re, Ra and B, on which the resulting heat transfer depends. Some useful correlations are also proposed.

Keywords: Heat transfer, Vertical jet, Rectangular blocks, component cooling, numerical simulation, inverted T form cavity.

Nomenclature

A	aspect ratio (l/h)
b	block height (m)
B	dimensionless block height (b/h)
c	opening diameter (m)
C	dimensionless opening diameter (c/h)
d	space between adjacent blocks (m)
D	dimensionless space between adjacent blocks (d/h)
h	channel height (m)
l	length of the calculation domain (m)
n	normal coordinate
Nu	mean and normalized Nusselt number (eq.5)

Pr	Prandtl number ($Pr = \nu/\alpha$)
Ra	Rayleigh number ($Ra = g\beta\Delta T h^3/(\alpha\nu)$)
Re	Reynolds number ($Re = v_0 h/\nu$)
T	temperature of fluid
t	time (s)
τ	dimensionless time ($t.\alpha/h^2$)
θ	dimensionless temperature of fluid [$= (T - T_c)/(T_H - T_C)$
u, v	velocities in x and y directions (m/s)
U, V	dimensionless velocities in x and y directions [$= (u, v)/v_0$]
v_0	average jet velocity at the entrance (m/s)
x, y	Cartesian coordinates (m)
X, Y	dimensionless Cartesian coordinates [$= (x, y)/h$]
α	thermal diffusivity (m^2/s)
β	volumetric coefficient of thermalexpansion (K^{-1})
λ	thermal conductivity of fluid (W/m.k)
ν	kinematic viscosity of fluid (m^2/s)
ρ	fluid density (kg/m^3)
ψ	stream function (m^2/s)
ϕ	dimensionless stream function (=ψ/α)
Ω'	vorticity (s^{-1})
Ω	dimensionless vorticity (=$\Omega' h^2/\alpha$)

Subscripts

C	cold wall
l	limit value
f	related to the forced convection
H	heated wall

1 Introduction

Numerous studies related to the phenomenon of mixed convection in channels have been made in order to investigate the heat transfer and fluid flow in such geometries [see, e.g., Islam et al. (2008); Accary et al. (2008); Ben-Arous and Busedram (2009) as relatively recent examples]. In most cases, such interest is dictated by its direct relation with the cooling of components in electronic industries.

Due to the progress in circuit integration, heat dissipation seems to be concentrated on fewer components, as the system volume shrinks. The development in the field of high-density and large scale integrated chips has led to numerous miniaturized electronic devices. Consequently, heat from such sophisticated electronic components is excessively generated. A rigorous control of their operating temperatures seems indispensable to prevent their getting damaged owing to an eventual overheat.

In mixed convection, important works have been reported on problems related to the cooling of electronic components [Bar-Cohen (1987)]. Generally, the components are disposed on horizontal cards [Kim and Anand (1994)] and [Leung and Chen (2000)] subjected to outside ventilation to evacuate the excess of generated heat.

A review of recent works shows the importance and the progress achieved in this field.

Furthermore, when the components are disposed on the internal surfaces of a horizontal channel, the space between the components remains not well ventilated even in the case when the cooling process is ensured by a forced flow [Herman and Kang (2001)]. In fact, the forced flow, being parallel to the surface containing the components, often engenders re-circulating movements between the blocks. Consequently, these regions are not well ventilated.

Experimental investigations and numerical simulations were performed by Leoni and Amon [Leoni and Amon (2000)] on an embedded electronics prototype system of a wearable computer.

Bilen et al. [Bilen et al (2001)], conducted an experimental and numerical investigation about the effect of the position of wall mounted rectangular blocks on the heat transfer from the surface, taking into account the angular displacement of the block as well as its spanwise and streamwise disposition. The experiments were conducted in a rectangular channel with variable parameters as: distance between adjacent blocks, block displacement angle and Reynolds number. The results showed that the most efficient parameters were Reynolds number and angular disposition. Distance between blocks has a slightly increasing effect on the heat transfer. Thanks to the use of orthogonal arrays and optimal sampling, an efficient exploration of the parameter space was performed to determine thermal conductivities, thermal contact resistances and heat transfer coefficients. Murakami and Mikic [Murakami and Mikic (2001)] presented an optimization study using a method of determining optimum values of the channel diameter, flow rate and number of channels for minimum pressure drop.

Various strategies have been explored to enhance the effectiveness of mixed convection cooling. These configurations include the implementation of an obstacle in the flow path of the coolant to destabilize the flow [Lehmann and Huang (1991)], using variable space length between two adjacent blocks [Chen and al. (2001)] or openings between blocks in re-circulating movement spaces [Najam et al. (2004); El Alami et al. (2009)].

These strategies certainly, enhanced, heat transfer between the blocks but, their vertical planes do not seem to be well ventilated when the jet is parallel to the block support and when the blocks are disposed on the bottom wall. The present work considers a new configuration in which the cooling jet is sent perpendicularly to block support and the blocks are placed on the upper plane of the channel. The effect of Reynolds number and the block height effects on mixed convective heat transfer are closely studied. Since this configuration presents a geometric periodicity, where the period is an inverted "T" shaped cavity, the studied domain is reduced to a geometric period in this work. Rigid adiabatic surfaces are introduced to eliminate any heat and mass transfer between the different cavities.

2 Physical problem and governing equations

The schematic representation of the studied configuration is depicted in figure 1 (a). As it's indicated above, the calculation domain is reduced to an inverted "T" shaped cavity which is submitted to a laminar convective vertical jet of fresh air from below, figure I (b). The blocks are isotherms at hot temperature T_H. The upper wall is adiabatic, while the lower plane is kept cold at temperature T_C.

The principal assumptions used in this work are:

- Physical properties of the fluid are assumed to be constant except for the density which is variable with temperature.
- Viscous dissipation of fluid energy is neglected.
- The terms of pressure are neglected in the energy balance performed using the enthalpy.
- The fluid is considered Newtonian and the flow is assumed to be laminar.

$$\frac{\partial \Omega}{\partial \tau}+\frac{\partial U\Omega}{\partial X}+\frac{\partial V\Omega}{\partial Y}=-\frac{Ra}{Re^2\,\mathrm{Pr}}\left(\frac{\partial \theta}{\partial X}\right)+\frac{1}{Re}\left(\frac{\partial^2 \Omega}{\partial X^2}+\frac{\partial^2 \Omega}{\partial X^2}\right) \tag{1}$$

$$\frac{\partial \theta}{\partial \tau}+\frac{\partial U\theta}{\partial X}+\frac{\partial V\theta}{\partial Y}=\frac{1}{Re\,\mathrm{Pr}}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \tag{2}$$

$$\frac{\partial^2 \Psi}{\partial X^2}+\frac{\partial^2 \Psi}{\partial X^2}=-\Omega \tag{3}$$

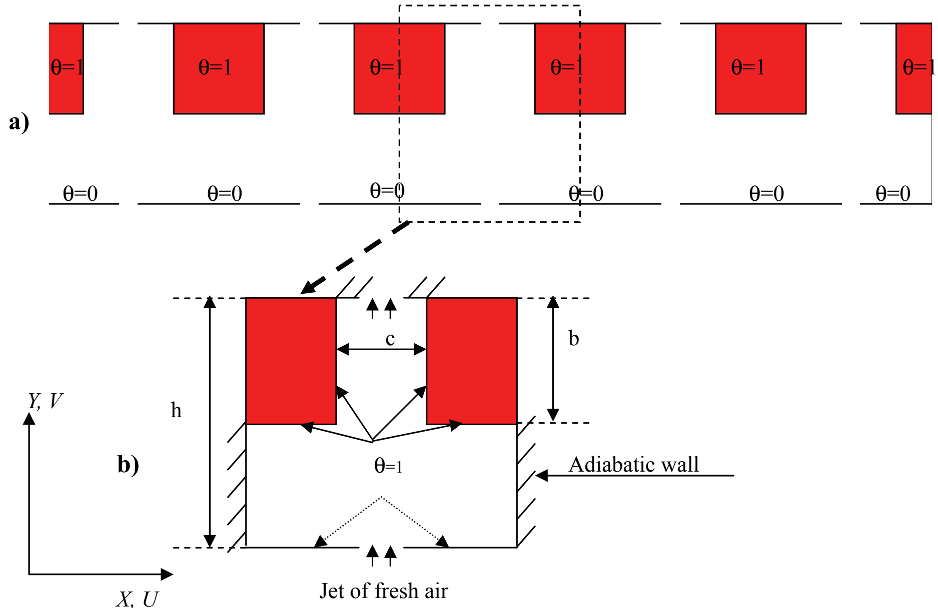

Figure 1: Studied configuration

$$U = -\frac{\partial \Psi}{\partial Y} \text{ et } V = \frac{\partial \Psi}{\partial X} \tag{4}$$

The dynamic and thermal boundary conditions are:
θ=0; Ω=0; U=0; V=1 and $\Psi = X - 0.425$ at the admittance opening. θ=0 on the lower cold wall
Ψ=0 on the rigid wall situated to the left of the axis passing through the middle of the openings and Ψ=C on those on the right. θ=1 on the heated blocks.

At the evacuation opening, U, V, θ Ω and Ψ, are extrapolated by adopting similar processes as shown in references [Tomimura and Fujii (1988); El Alami et al. (2005)] (their second spatial derivative terms in the vertical direction are equal to zero).

The mean and normalized Nusselt number over the active walls is:

$$Nu = \left[\frac{1}{1-B} \int_{1-B}^{1} \left. \frac{\partial \theta}{\partial n} \right|_{X=0.25} dY + \frac{1}{0.25} \int_{0}^{0.25} \left. \frac{\partial \theta}{\partial n} \right|_{Y=1-B} dX \right. \\ \left. + \frac{1}{1-B} \int_{1-B}^{1} \left. \frac{\partial \theta}{\partial n} \right|_{X=0.75} dY + \frac{1}{0.25} \int_{0.75}^{1} \left. \frac{\partial \theta}{\partial n} \right|_{Y=1-B} dX \right] \tag{5}$$

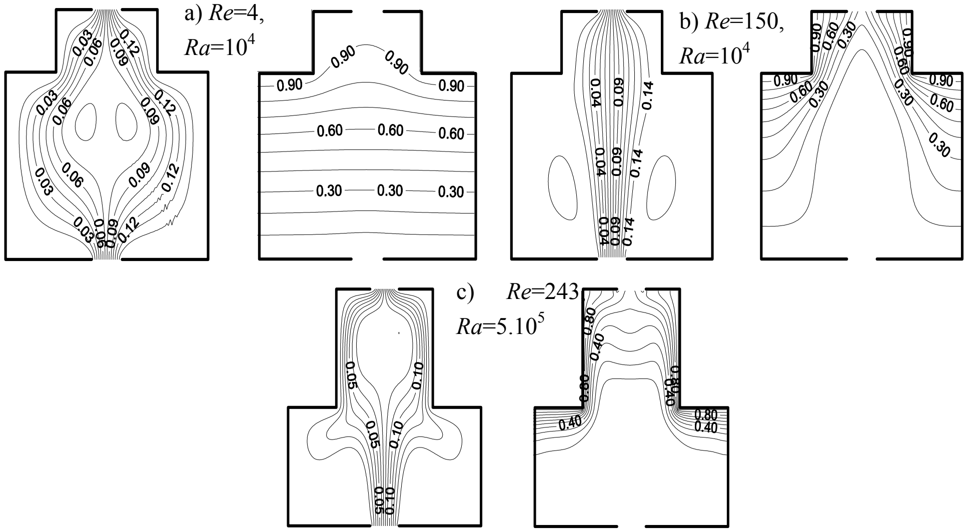

Figure 2: Different solutions obtained by varying *Ra*, *Re* and *B*.

3 Numerical method

Various methods can be used for solving the coupled Navier-Stokes and energy equations [see, e.g., Djebali et al. (2009); El Alami et al. (2009) ; Achoubir et al. (2008); Bucchignani (2009); Mezrhab and Naji (2009)].

Here the governing equations are solved numerically using a finite difference technique. A central difference scheme is used for the discretization of all spatial derivative terms in the Poisson, energy and vorticity equations. The discretization of diffusive terms is performed using a second order central difference scheme (C.D.S). The convective terms are discretized using a second order upwind scheme [Thakur and Shyy (1993)]. The final discretized forms of equations (1) and (2) were solved by using the alternate direction implicit method (ADI). The stream functionΨ, the Poisson equation was discretized by a central difference scheme and solved by the Successive Over Relaxation (SOR) method. For more details see [Frankel (1950), Najam (2004)]. At each time step, variation by less than 10^{-4}, over all grid points for the stream function is considered as a convergence criterion. As a result of a grid independence study, a grid size of 121x121 was found to model accurately the flow fields described in the corresponding results. Time steps considered are ranging between 10^{-6} and 10^{-4}. The accuracy of the numerical model was verified by comparing results from the present study with those obtained by Amahmid et al. [Amahmid et al (1999)] for natural convection in repetitive geometries. Good agreement was obtained in Ψ_{max} and *Nu* terms. When a steady state is reached, all the energy furnished by the hot walls to the fluid must leave the cavity through cold surface (with the opening). This energy balance was verified by less than 3% in all cases considered here.

Table 1: Model validation

Ra	Our results (upwind)		Amahmid et al (1999) (CDS)	
10^3	Ψ_{ext}=0.024		Ψ_{ext}=0.024	
10^4	Ψ_{ext}=0.77		Ψ_{ext}=0.80	
10^5	Ψ_{min}=-12.12 Ψ_{max}=8.93	Ψ_{ext}=9.49	Ψ_{min}=-13.64 Ψ_{max}=9.89	Ψ_{ext}=11.00

4 Results and discussion

This study is aiming to analyse the interaction between an external imposed flow, characterised by *Re*, and the natural convection developed in the cavity. We note that, if the objective of this investigation is the *chimney effect*, the forced flow debit in the inlet will be a fundamental unknown of the problem and we must determine it as a function of the control parameters (without *Re*).

Heat transfer rates across the cold and the hot walls, flow and temperature fields are examined for ranges of Reynolds number ($1\leq Re \leq 1000$), distance between adjacent blocks ($0.25\leq B \leq 0.75$), Rayleigh number ($10^4 \leq Ra \leq 10^6$) and other parameters of the problem (*D*=0.5, *C*=0.15, *A*=1, *Pr*=0.72), in this section.

Essentially, the flow structure is composed of natural convection cells and forced convection open lines. Three kinds of solutions are highlighted. Their existence is due to the conflict between natural and forced convection, in the considered ranges of Reynolds number and block sizes. When two natural convection cells exist, the forced convection open lines go a round the cells, or pass between them. These two solutions are symmetrical, bicellular and, respectively, called *Extra-Cellular*, and *Intra-cellular flows* (called *ECF* and *ICF*) as shown in fig. 2a and 2b for $B = 1/4$ (streamlines on the left and isotherms on the right). When the natural convection cells disappear, by increasing Re, we have only the forced flow (*FF*) in the cavity. This situation is observed for *B*=1/2, fig.2c.

The discussion results paragraph is organized as follow: first, we detail the case of *B*=3/4 with varying *Re* and *Ra* and then, we present the case of *B*=1/2. For *B*=1/4, remarks are made.

4.1 Flow structure and thermal field

Case of B=3/4

In this section, typical flow structures are presented. For *Ra*=10^4, the problem solution is symmetric. *ECF* and *FF* flow structure are obtained as shown, respectively, in figs.3a, *Re*=4 and 3b, *Re*=100. Note that the *ICF* solution is not detected in this case. There is no sufficient space down the blocks for developing convective cells. The corresponding isotherms show that, for weak *Re* values, the vertical walls of the blocks are not well ventilated, contrarily to the horizontal ones.

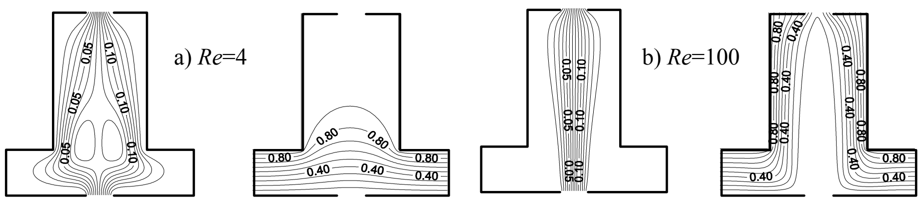

Figure 3: Streamlines and isotherms for *Ra*=10^4 and *B*=3/4

When increasing *Ra* (*Ra*=10^5), the cell sizes increase for low values of *Re* as shown in fig.4a, *Re*=10. The cells are located between the blocks. The horizontal planes of the blocks are refreshed but along the vertical ones, there is, practically, no heat exchange as presented by the corresponding isotherms. For relatively high values of *Re* (*Re*=100) the main remarks are the disappearance of the connective cells and the development of boundary layer flow, as shown by the thermal field in fig.4b. Hence, all the active block walls are well ventilated.

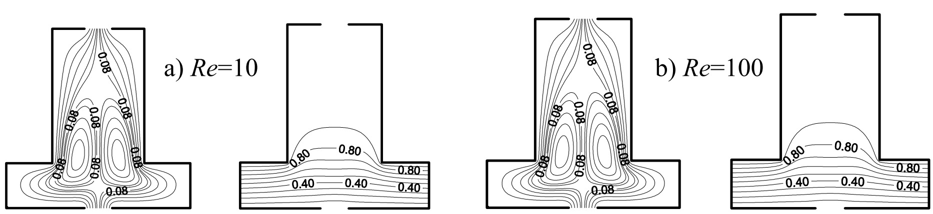

Figure 4: Streamlines and isotherms for *Ra*=10^5 and *B*=3/4

For *Ra*=10^6, the major result is that the problem solutions are not symmetric at low Reynolds number (*Re* $\leq$50) as shown in fig.5a for *Re*=10. Natural convection cell sizes increase significantly and occupied all the space between blocks and strangle the jet of fresh air (there are no open lines in the cavity, for *Re*=10). We have, practically, natural convection mode. The corresponding thermal field shows that heat transfer is sensitively, enhanced along the vertical planes. For *Re* up to 50, fig.5b (*Re*=100) the problem solution becomes symmetric again: the powerful jet promotes the existence of the symmetry. The flow structure is of *ECF* kind, and convective cells are elongated inside the forced jet and in the entire area between the blocks. Consequently, fresh air is constrained to pass close to the vertical hot walls. Isotherms show that heat exchange is more important than in the case of *Re*=10, along the major part of vertical heated planes. A stratification phenomenon is observed in the middle of the cavity. When *Re* increases, advantageously, the cell sizes and the stratified zone are, significantly, reduced under the influence of the powerful forced jet. This *Re* value is very close to critical *Re* (Re_{cr}) for which convective cells disappear and the forced convection mode is installed. All heated walls are well ventilated, fig.5c.

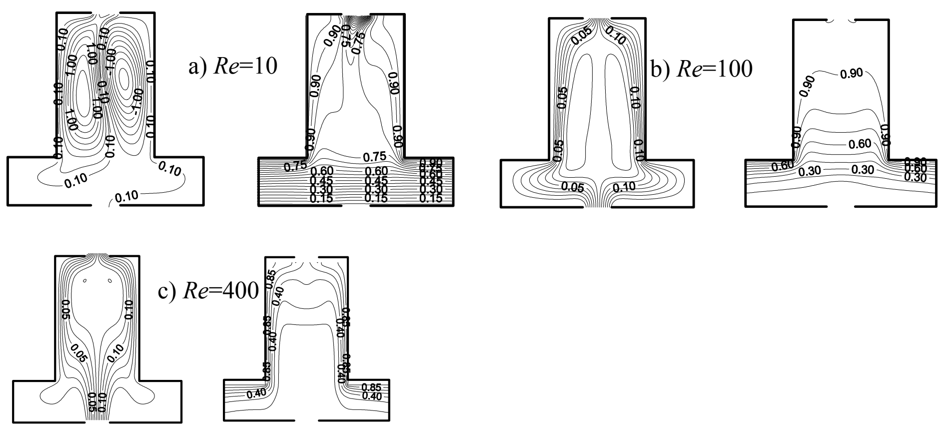

Figure 5: Streamlines and isotherms for $Ra=10^6$ and B=3/4

Case of B=1/2 and B=1/4

For B=1/2, $Ra=10^4$ (figures not presented in this paper), the flow structure and thermal field are in the same manner of those obtained at $Ra=10^5$. The only particularity is that the convective cells are weak (their sizes are very small compared to those of $Ra=10^5$), disappear early when varying *Re* ($Re_{cr} \cong 6$) and yield place to forced convection mode. For relatively high values of *Ra*, the conflict between natural convective cells and forced flow discussed in the last case appears again, fig.6a, 6b. Natural convection dominates at low *Re* and forced regime is installed at high values of *Re*.

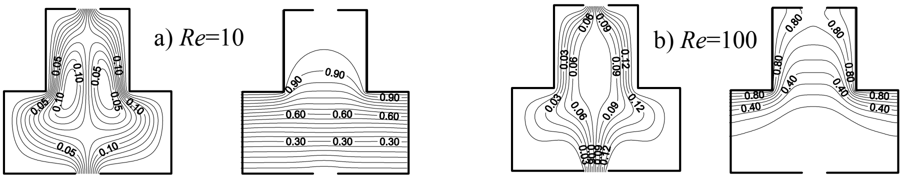

Figure 6: Streamlines and isotherms for Ra=10^5 and B=1/2

Streamlines and isotherms, for B=1/4, are presented in figure 2, for Re=4, 150 and Ra=10^4. Only in this case, we have detected the three kinds of the problem solution: *ECF*, fig.2a; *ICF* fig. 2b and FF (figure not presented). Note that the *ICF* solution is absent in the cases of B=1/2 and 3/4. As of B=3/4, dissymmetric solutions are detected for these cases always at high Ra and low values of Re. In this later case (B=1/4) the space given to convective cells is important. Hence, they must be stronger and must resist, more than in the first cases, to the forced jet. Contrarily, Results show that the cells are weaker than those of B=1/2 or 3/4: the thermal gradient zone is reduced and located in the upper part of the cavity. The rest of the cavity is, practically, isotherm. Consequently, air in the lower zone of the cavity is not well heated to develop stronger convective cells.

In fig.7, we present the critical Reynolds number (Re_{cr}) variation with Rayleigh number for B=3/4, 1/2 and 1/4 (Re_{cr} is a value of Re for which the convective cells disappear). For each value of B, the curve delimits two zones: that of the bottom corresponds to the mixed convection mode: the convective cells, always, exist in the cavity. The zone of the top of Re_{cr} curve is related to the forced convection mode. We note that Re_{cr} decreases when the block height is reduced. As discussed above, the convective cells are much stronger when B increases, because the increasing thermal gradient zone. Note that the, gap between Re_{cr} profiles for different values of B, becomes more and more important by increasing Ra.

4.2 Heat Transfer

Heat transfer, through the cavity, has been calculated in normalized Nusselt number term, by using eq. (5) for the different values of B (B=3/4, 1/2 and 1/4). We present its variation with Reynolds number, for fixed value of Ra ($Ra = 5 \times 10^5$), fig.8. Generally, Nu is an increasing function of Re and B. This result is expected because the more we increase the speed of the fresh air blast, the more we evacuate heat from the cavity towards the outside. we notice that, the gap between Nu curves for different B increases with Re values. This is because of the reduction of heated surfaces when reducing B. The other reason is, as indicated above: when Re

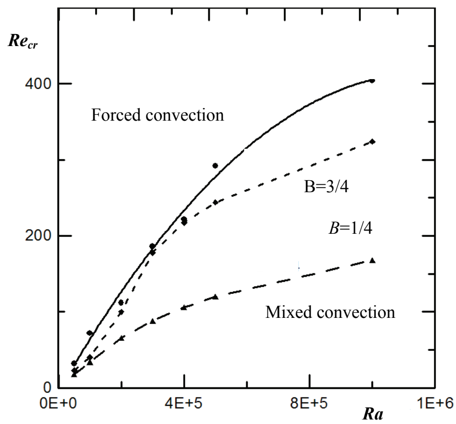

Figure 7: Re_{cr} Variation with Ra, for different values of B

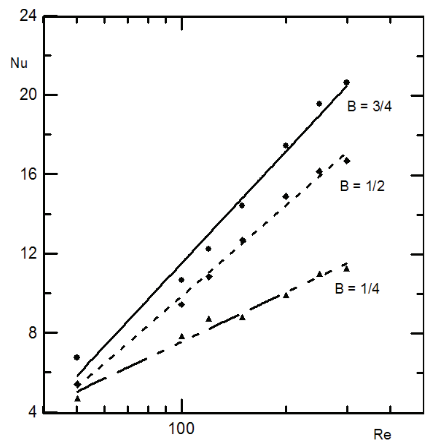

Figure 8: Nusselt number variation with Re for different values of B, $Ra = 5 \times 10^5$

increases, for Ra up to 10^5 and for a given value of B, we have a boundary layer flow between blocks and the forced jet is constrained to pass close to the vertical walls and then, to enhance heat exchange trough the vertical hot planes. The Nu variation with Re is correlated for different values of B as below:

$$Nu = 3.60 \times ln(Re) - 9.06 B = 1/4$$

$$Nu = 6.61 \times ln(Re) - 20.61 B = 1/2$$

$$Nu = 8.15 \times ln(Re) - 26.08 B = 3/4$$

It should be noted that this work complements a series of studies, realized in our laboratory, for which the main purpose is to seek a better way of ventilation to cool electronic components. In this sense, we compare the heat transfer obtained in this simulation with the configuration studied by Najam et al. [Najam et al. (2003)]. In the latter study, the blocks are placed on the lower wall and the studied configuration is a "T" shaped cavity. This comparison is done by comparing the Nusselt numbers for B=1/2, fig.9. We can deduce that, in the two cases, we have obtained the same form of Nusselt number variation with Reynolds (logarithmic function). The Nusseult numbers comparison shows, clearly, that heat exchange is more important in de case of "T" shaped cavity than the one obtained in our study. Hence, to place electronic components on the upper plane of the cavity can't be a good possibility to evacuate heat. So this configuration is not favourable to a rigorous control of integrated circuit operating temperatures.

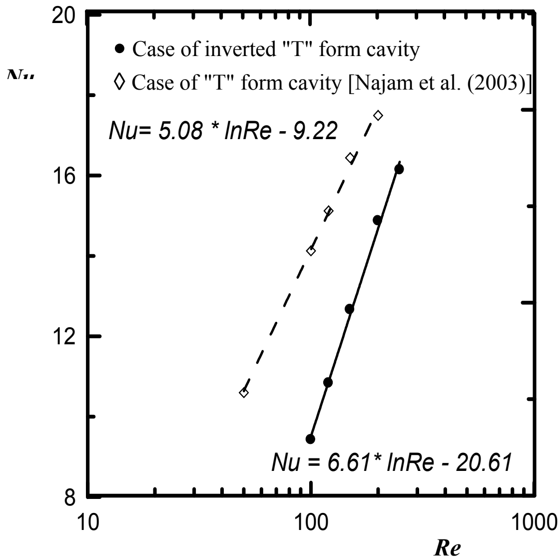

Figure 9: Comparaison of Nusselt numbers, cases of our word and the "T" form cavity [Najam et al. (2003)], $Ra = 5 \times 10^5$, B=1/2

5 Conclusion

Mixed convection in a square cavity containing rectangular heated blocks on its upper wall (an inverted "T" shaped cavity) has been studied numerically. Three solutions have been obtained when varying *Re* and *Ra*, *ECF*, *ICF* and *FF*. It is demonstrated that there is a critical *Re* (Re_{cr}) above which the natural convection cells disappear and the flow in the cavity is governed by forced convection (*FF* solution). Re_{cr} increases with *Ra* for each value of *B*. Non symmetric solutions have been found at low *Re*, for all choose values of *B*, while the boundary conditions and the geometry of the system are symmetric.

Heat Transfer has been calculated in term of Nusselt number and its variation with *Ra* has been presented. Correlations of *Nu* with *Re* have been proposed for different values of *B*. We note that *Nu* increases with *Re* and *B*, and that the *Nu* curve in the case of *B*=1/4 has a weak slope compared to those of the other cases.

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