Mixed Convection Investigation in an Opened Partitioned Heated Cavity

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Abstract: Mixed convection in a rectangular partitioned cavity equipped with two heated partitions at a constant temperature, T_C, is investigated numerically. The right vertical wall is featured by two openings (C_1 and C_2) for admission of cooled air along the horizontal direction, while the lower wall has a single outlet opening (C_3) along the vertical axis. The left vertical wall is assumed to be isothermal at temperature T_C, while the other walls are cooled to a temperature $T_F < T_C$. The results show that the flow and heat transfer depend significantly on Reynolds number, Re, and block height, B. Correlation laws for the dimensionless heat evacuated trough the opening, Q_f, as a function of, Re, and, B, are expressly derived.

Keywords: mixed convection, rectangular partitioned cavity, heat transfer, numerical study, horizontal jet.

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
Pr	Prandtl number ($Pr=\nu/\alpha$)
Ra	Rayleigh number ($Ra=g\beta\Delta Th^3/(\alpha\nu)$)

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Re	Reynolds number ($Re = v_0 h/v$)
T	temperature of fluid
t	time (s)
τ	dimensionless time ($t\alpha/h^2$)
θ	dimensionless temperature of fluid
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 (m²/s)
β	coefficient of thermal expansion (K^{-1})
λ	thermal conductivity of fluid (W/mk)
v	kinematic viscosity of fluid (m²/s) fluid density (kg/m³)
ψ	stream function (m²/s) dimensionless stream function ($=\psi/\alpha$)
Ω'	vorticity
Ω	dimensionless vorticity ($=\Omega' h^2/\alpha$)

Subscripts

b	partition
C	cold wall
cr	critical value
f	related to the forced convection
H	heated wall
max	maximal value
min	minimal value
1, 2	partition position

1 Introduction

Mixed convection has been over recent years the subject of much interest due to its relevance to numerous and multivariate applications in the field of air conditioning, heating and refrigeration. In general, the shape of the cavity, the location and number of openings are important parameters determining the intensity of air motion and its efficiency in removing naturally heat from cavity.

Several studies have been conducted in this area using different types of cavities (cubical, parallelepiped, rectangular, cylindrical and other more or less complex shapes) and cavities with or without openings in order to determine the conditions

by which the efficiency of ventilation can be improved. Many authors have focused on natural convection and its multiple manifestations and instabilities [Bejan 2004, Choukairy et al 2006, Kaenton et al 2004, Lappa, 2004-2013]. Some analyses have been based on classical methods such as finite volumes and finite elements, others on new approaches such as the Lattice Boltzmann [Djebali et al 2009, Mohamad et al 2010].

A review of existing literature relevant to the present work is elaborated in the following.

Mahapatra et *al.* (2007) studied numerically mixed convection in a differentially heated and partitioned square cavity fitted by partitions; in their study the moving vertical walls are isothermal while the lower and upper walls and partitions are adiabatic. The Richardson number, *Ri*, was varied to simulate mixed convection flow (*Ri* = 1) and forced convection (*Ri* = 0.1). Comparison was also made with natural convection in order to understand the thermal behaviour of the studied configuration. The partitions' size and location were varied to study their effect on heat transfer. It was noted that for natural convection, higher heat transfer occurs in the enclosure when the partition at the base is near to the hot wall compared to the same when partition is located near the cold wall. This trend is reversed for *Ri* = 1 and 0.1. The effect of the partition placed at ceiling was found to be opposite in characteristics to the phenomenon in the enclosure having partition at the base. The effect of the location of partition on heat transfer is marginal for *Ri* =1 and more pronounced for *Ri = 0.1*. When partition height exceeds 0.3 H, the heat transfer in case of opposing mixed convection is more than that of natural convection in centrally portioned enclosure. Centrally located partition provides better insulating effect compared to offset partition for partition height exceeding 0.2 H and this effect is less in case of mixed convection.

Aydin and Yang (2000) studied numerically laminar mixed convection in a cavity where the flow is induced by movement of the edges of the cavity and heat sources are placed at the middle of the cavity. The phenomenon of heat transfer was investigated for a series of values of Grashof and Reynolds numbers (*Gr/Re^2*) and different sizes of the heat source, b. It was found that when Gr/Re^2 increases, three regimes appear forced, mixed and natural convection. For *b*= 1/5, the increase of *Gr/Re^2* from 0.1 to 10, is in favour of natural convection.

Bahlaoui et *al.* (2009) studied numerically mixed convection and surface radiation within a horizontal ventilated rectangular cavity heated from below and with an adiabatic vertical thin partition on the heated surface. The cavity was submitted to an imposed flow of ambient air (considered as a cooling fluid) through an opening located on the low part of the left vertical wall, with the forced flow leaving the cavity through an outlet opening placed on the upper part of the right vertical

wall. The fluid properties were evaluated at a mean temperature and the airflow was assumed to be laminar, and obeying the Boussinesq approximation. The effect of the Reynolds number ($200 \leq Re \leq 5000$) and the baffle position from the inlet ($0.25 \leq L_b \leq 1.75$), on fluid flow and heat transfer characteristics was studied in detail. Results of this study show that the radiation effect leads to a better homogenization of the temperature inside the cavity by reducing the cold zone space in the entrance region. It was found that the radiation effect reduces the convective Nusselt number component and the latter is favoured by the Reynolds number, Re, and the displacement of the partition away from the inlet,L_b. All the parameters Re, L_b and ε were found to have a positive effect on the total heat transfer. In addition, instabilities of physical origin were observed for $L_b = 1.5$. The best cooling condition for the cavity, represented by the decrease of mean and maximum temperatures of the fluid, was obtained by increasing the parameters Re, L_b and ε. Also, the contribution of radiation to the overall heat transfer was found to be generally not negligible even for the weaker value of ε considered ($\varepsilon=0.15$).Finally, it was found that the majority of the heat released by the heat wall, is evacuated through the exit while varying the controlling parameters in their respective ranges (the heat evacuated through the left cold wall was observed to be promoted by the increase of L_b and ε and by the decrease of Re).

Fenglei et al. (2007) investigated mixed convection and heat transfer augmentation in a large rectangular cavity with a vertical cooling surface. The experiment was conducted by varying different parameters such as jet diameter, jet injection orientation and the enclosure aspect ratio. Cooling water was circulated through copper tubes, welded to the backside of the copper plate, to generate a nearly isothermal surface. All the other walls were insulated. The wall opposite the vertical cooling surface could be moved to change the enclosure aspect ratio. The hot air entered the enclosure from the jet and left through an opening at the bottom of the movable wall. Thermocouples and heat flux sensors were embedded in the copper plate to measure the temperature and heat flux of the cooling surface. The results of this study show that the heat transfer augmentation by forced jets is controlled by the jet Archimedes number, fluid properties, injection orientation, and enclosure aspect ratio.

The case of a square cavity with heated rectangular blocks and submitted to vertical forced flow was studied by Meskini et *al (2011)* . The configuration so defined is an inverted "T"-shaped cavity presenting symmetry with respect to a vertical axis. 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. Their 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 have been also proposed.

The steady mixed convection and related heat transfer from a heated semi-circular cylinder immersed in power-law fluids has been considered by Chandra and Chhabra (2012). The momentum and thermal energy equations have been solved numerically over the following ranges: 0≤Ri≤2; 1≤Re≤30 and 1≤Pr≤100. The combined effects of the forced and free convection on the flow and thermal fields have been visualized in terms of the streamline and isotherm contours. Their results show that average Nusselt number increases with an increase in the value of the Reynolds number, Prandtl number and Richardson number. Broadly, shear-thinning viscosity facilitates heat transfer whereas shear-thickening has an adverse effect on it. Other interesting analyses are due to Amirouche and Bessaïh (2012), Abbassi, Halouani, Chesneau and Zeghmati, (2011), Dihmani, Amraqui, Mezrhab and Laraqi, (2012), Ferahta, Bougoul, Médale and Abid (2012), Kriaa, El Alami, Najam and Semma, (2011), etc.

Further studies along these lines, in open and closed cavities, in variable geometries with heating partitions, will help to find the right combination that promotes heat transfer and thus better cooling. In the present paper, in particular, we study mixed convection in a rectangular cavity with two heating partitions, two inlet openings on the right vertical wall and one outlet opening on the lower horizontal wall. Our main objective is to improve cavity ventilation (a subject with extensive background application to the field of commercial refrigeration). Accordingly, we develop a systematic study based on the principal problem control parameters.

2 Physical Model and Mathematical Formulation

2.1 The mathematical model

A rectangular enclosure equipped with two heating partitions is subjected to a cooled air jet at temperature T_F ($T_F < T_C$) through two inlet openings arranged on the right vertical wall (Figure 1). An outlet opening is placed on the lower horizontal wall of the cavity. The partitions are heated at constant temperature T_C. It is assumed that flow and heat transfer are two-dimensional; fluid properties are constant and the Boussinesq approximation is valid.

The transient dimensionless equations in terms of temperature, θ, rotational Ω, and stream function ψ are :

$$\frac{\partial \Omega}{\partial t} + \frac{\partial (u\Omega)}{\partial X} + \frac{\partial (v\Omega)}{\partial Y} = -\frac{Ra}{PrRe^2}\left(\frac{\partial \theta}{\partial X}\right) + \frac{1}{Re}\left(\frac{\partial^2 \Omega}{\partial X^2} + \frac{\partial^2 \Omega}{\partial Y^2}\right) \tag{1}$$
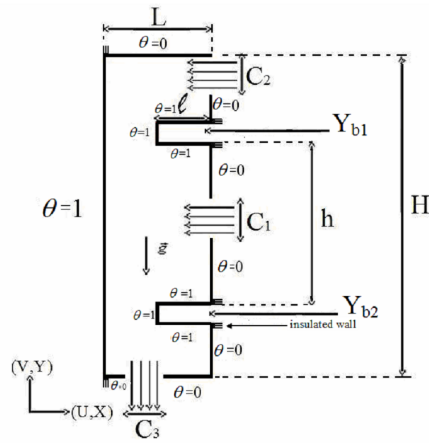
Figure 1: Studied configuration

$$\frac{\partial \theta}{\partial t} + \frac{\partial (u\theta)}{\partial X} + \frac{\partial (v\theta)}{\partial Y} = \frac{1}{Re\,Pr} \left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} \right) \tag{2}$$

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

$$u = -\frac{\partial \Psi}{\partial Y} \; et \; v = \frac{\partial \Psi}{\partial X} \tag{4}$$

2.2   The Boundary conditions

Position of partitions for D=1/2: $\begin{cases} \Upsilon_{b1} = 0.562 \\ \Upsilon_{b2} = 2.343 \end{cases}$

The stream function is linear at the openings $C1$ and $C2$ such that $\psi = aY+b$, its value is given by:

Opening C_1 : ($X = 1$, $1.312 \le Y \le 1.687$) ; $\psi = 0.2Y-0.262$

Opening C_2 : ($X = 1$, $2.625 \le Y \le 3$) ; $\psi = 0.2Y-0.45$

Opening C_3 : ($0.312 \le X \le 0.687$, $Y = 0$) ; ψ, θ, Ω, U and V are extrapolated by adopting similar processes as shown in references [Tomimura and Fujii (1988); El Alami et *al.* (2005), (2008); Rouijaa et *al.* (2011)].

The initial condition for *u, v, ψ* and *θ* is equal to 0.

On solid walls, the stream function is given for:

- *(0.6875 ≤ X ≤ 1, Y =0)* et *(X=1, 0< Y < 1.3125,* including the partitions) : *ψ = 0*

- *(X=1, 1.687 ≤ Y ≤ 2.625,* including the partitions) : *ψ = 0.075*

- *(0≤ X ≤1, Y =1) : ψ = 0.15*

- *(X=0, 0≤ Y ≤1) : ψ =0.15*

- *(0≤ X ≤0.3125) : ψ =0.15*

The thermal boundary conditions on solid walls:

- *θ* = 1 on the left vertical wall of the cavity and on the vertical and horizontal sides of partitions.

- *θ* = 0 on both upper and lower horizontal walls of the enclosure and on the right vertical wall, partitions being excluded (Figure 1)

- $\frac{\partial \theta}{\partial n} =$ 0 on an insulation portion inserted between each heated wall in contact with a cold wall, with n denoting the dimensionless normal coordinate to the portion of the insulated wall.

- On all solid walls : *U = V = 0*

The dimensionless heat evacuated trough the outlet opening C_3 is given as:

$$Q_{fo} = \int_{X_1}^{X_2} (-\frac{\partial T}{\partial Y} + v.Pe.T)dX \tag{5}$$

where X$_1$= 0.6875 and X$_2$= 0.3125

The heat exchanged in terms of active surfaces is defined by:

Vertical wall:

$$Q_{fv} = \frac{1}{(Y_2 - Y_1)} \int_{Y_1}^{Y_2} \left(\frac{\partial \theta}{\partial X}\right)_X dY \tag{6}$$

Horizontal wall:

$$Q_{fh} = \frac{1}{(X_3 - X_4)} \int_{X_4}^{X_3} \left(\frac{\partial \theta}{\partial Y}\right)_Y dX \tag{7}$$

The heat value exchanged by the cavity is the sum of these three quantities:

$$Q_f = Q_{fo} + Q_{fv} + Q_{fh} \tag{8}$$

2.3   The numerical method

The governing equations were solved numerically by using a finite difference technique. A central difference formulation was used for the discretization of all spatial derivative terms in the Poisson, energy and vorticity equations [Roache (1982)]. Diffusive terms have been discretized by using an upwind scheme. The final discretized forms of equations (1) and (2) were solved by using the Alternate Direction Implicit method (ADI). Values of stream function were obtained with equation (3) using a Successive Over – Relaxation method (SOR). At each time step, variation by less than 10^{-4}, over all grid points for the stream function has been 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 and thermal fields for 10^{-6} time step (Table 1). The accuracy of the numerical model was verified by comparing results from the present study with those obtained for natural convection in a trapezoidal cavity [Kalache (1987)].

When a steady state is reached, (after 100 iterations) all the energy released by the hot walls to the fluid must leave the cavity through the cold surface and the openings. This energy balance was verified by less than 3 % in all cases considered here (Table 2 and 3).

Table 1: Values of Ψ and Nu for different meshes

Mesh	Ψ_{max}	Nu
81x81	0,3520	26,8859
101x101	0,3520	27,5333
121x121	0,3338	27,9154
141x141	0,3356	28,1476

Table 2: Comparing Ψ_{MAX} obtained with the present code with those of [Kalache (1987)]

Gr	Ψ_{MAX} (Present code)	Ψ_{MAX} [Kalache (1987)]	Deviation
2500	5.42	5.37	1 %
5000	9.74	9.77	3 %
10000	15.43	15.41	.13 %

Table 3: Comparing of our results in the present code with those of [Le Queré et De Roquefort (1985)]

	[De Val Devis et al. [(1983)]	[Le Queré et De Roquefort (1985)]	Our results	Maximal relative error
$Ra=10^4$	Ψ_{max}=5.098	——-	Ψ_{max}=5.035	1.2%
$Ra=10^5$	Ψ_{max}=9.667	——-	Ψ_{max}=9.725	0.6%
$Ra=10^6$	Ψ_{max}=17.113	Ψ_{max} =16.811	Ψ_{max}=17.152	2%
$Ra=10^7$	——-	Ψ_{max} =30.170	Ψ_{max} =30.077	0.3%

3 Results and Discussions

3.1 Flow structure

In this section, the results presented were obtained for $Ra = 10^5$, $(10 \leq Re \leq 200)$, D= 1/2, A= 3 and B = 1/2. Pictures of the flow patterns in the channel, stream function and temperature fields for different control parameters are reported below. The flow structure is generally composed of open lines representing the forced flow (the ventilation jet) and natural convection closed-cells. The size and the number of these cells depend significantly on control parameters. The existence of different solutions is principally due to the conflict between natural convection and forced flow as explained bellow.

3.2 Effect of Reynolds number, Re

For Re = 10 (Figure 2a), there's apparition of three convective cells located respectively between the partitions, in the lower and in the upper part of the cavity. Isotherms distortion shows that the top cell has greater intensity than the medium one which has a slight deformation due to the inertia effect. The forced flow which is characterized by a bit of open lines leading from the top opening and added to that of the bottom opening, is constrained to pass close the heated faces of partitions, and so the isotherms around the obstruction are getting tight and show a good heat exchange along all active obstruction faces. The bottom of the cavity is occupied by a cell of low intensity circulating over the outlet opening. This part of the cavity is the seat of a thermal stratification, as illustrated by the isotherms at this location. The isotherms show that, in this case, only the underside of the top obstruction is not well ventilated. This is because there is a separation of fresh air

jet close this plate. The heat is evacuated from the cavity to the outside through the cold vertical wall and through the horizontal opening at the bottom. However, poor heat exchange zones appear on the cold vertical plate, especially, in front of partitions. For, *Re* = 50, Figure 2b, the jet becomes relatively strong (number of open lines is important by comparing to the case of *Re* = 10, while the cells size is reduced, especially concerning the one in the lower part of the cavity. The size of the recirculating cells in front of the partitions is increased with Re and then the open lines of the jet are much distorted. The isotherms show that there is good heat exchange in the areas where the convective cells are in direct contact with the cold vertical wall. This shows also that the active upper faces of the two partitions are more ventilated than the lower faces. The thermal stratification in the lower part of the cavity is less pronounced in this case. At *Re* = 100, Figure 2c, increasing the intensity of the jet further reduced the size of convective cells. Mixed convection is well established, there is no convective cell in the bottom of the cavity. The extent of thermal stratification is reduced. Also we can remark clearly an increase in heat transfer through the major part of the cold vertical wall. Note the opening evacuation contributes to heat exchange more and more when Re increases. When *Re* = 150, the open lines become more numerous. Therefore the convective cells size is reduced and while the recirculating cells size increases considerably and push the fresh air jet to pass close the vertical cold plane; in front of vertical faces of the partitions, we have a better contact between cooled air and the vertical surface and so, heat transfer trough this plane is enhanced. The thermal stratification is destroyed leading to decrease heat exchange trough the horizontal face of the cavity. Isotherms are much distorted along the lower face of the upper partition because of the detachment of the forced flow in the vicinity of this face. At *Re* = 200, open lines have greatly reduced the size of convective cells. There is a clear predominance of the forced flow. Detachment of forced flow at the partitions level increases and the heat transfer becomes weak between the partitions and the cool air. This situation is unfavourable to the cooling of the partitions. The competition between natural convection (convective cells) and forced flow (open lines) is practically equitable for $Ra = 10^5$ and $Re = 100$.

3.3   *Effect of partition size, B*

Reynolds number is fixed at $Re = 100$; $Ra = 10^5$. Results are presented for three values of the partition height: $B = 1/4$, $1/2$ and $3/4$. In general, increasing the block size reduces the number of cells in the cavity and leads to promote natural convection on the forced flow. This result is trivial because when we increase the partition height, B, the heated surface enlarges. Concerning the thermal field, we note that when B is low, the walls of the blocks are practically well ventilated,
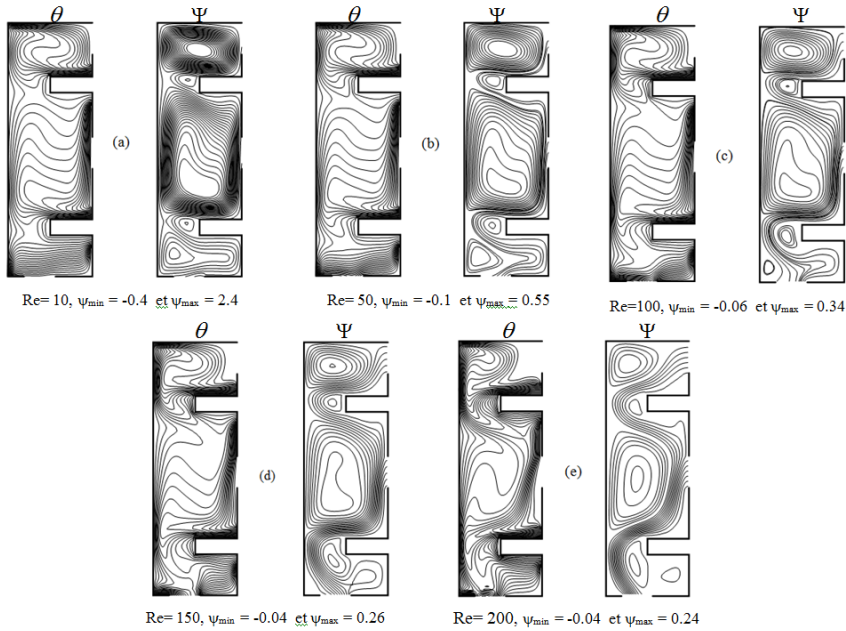
Figure 2: Streamlines and isotherms structure for different Reynolds numbers Re, (Ra=10^5, B=1/2)

and the heat removal from the cavity to the outside occurs through all the solid cold walls, except for two parts on the left vertical wall which are in front of the blocks. Further increase of *B* leads to a poor heat exchange along the lower face of the upper partition. Note that the recirculating cells appear as earlier as the block height is small. The streamlines obtained for $B = 1/4$, (Figure 3a) show that the flow structure is multicellular. Reciculating flow, in front of vertical surfaces of the partitions, disadvantages heat exchange along these walls. The corresponding thermal field shows that all the horizontal planes of the blocks are well ventilated compared to the previous case. For, $B = 1/2$, as described above, the cells size is reduced, especially those situated in front of the blocks, Figure 3b. Horizontal faces of the two blocks are not well ventilated and there is a poor heat exchange along them as indicated by the corresponding isotherms. By further increasing the size of the two heating partitions to, $B = 3/4$, the main point is the disappearance of driving cells in front of the blocks. So, the number of the closed cells is reduced to only two. The isotherms show that the horizontal walls of the blocks do not practically contribute to the heat exchange between those above and the ventilation jet. They show also that there is a thermal stratification in the zone between the partitions.

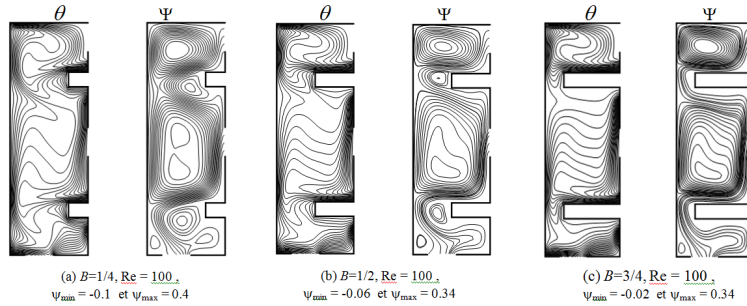We can note that there is an absence of cells in the bottom of the cavity.

(a) *B*=1/4, Re = 100 ,
ψ_{min} = -0.1 et ψ_{max} = 0.4

(b) *B*=1/2, Re = 100 ,
ψ_{min} = -0.06 et ψ_{max} = 0.34

(c) *B*=3/4, Re = 100 ,
ψ_{min} = -0.02 et ψ_{max} = 0.34

Figure 3: Streamlines and isotherms for different height of partitions

3.4   *Heat transfer*

The heat transfer through the cold faces with the opening is given in term of Q_f (Eq. 8), Figure 4. It presents the variation of the quantity of an average heat Q_f according to *Re* , for, *Ra* = 10^5 (calculated per area unit as indicated in equations (5,6 and7)). Generally, Q_f increases with *Re* for the three configurations (*B*=1/4, 1/2 and 3/4). This result is expected because more we increase the speed of the fresh air blast, more we evacuate heat from the cavity towards outside. Note that Q_f increases also with the block height for all the range of Reynolds number: for example, the Q_f curve related to the case of *B*=3/4 is up to the one for B=1/2. It should be noted that the increase in Q_f with, *B*, is not due to the increase of heated surface (since it was divided by it) but the fact that the closed cells in the bottom of the cavity and which are unfavorable to the heat exchange through the outlet opening gradually disappear as *B* increases, as explained above. For each value of *B*, the variation of Q_f with *Re* has two aspects: for *Re* <100, we see that Q_f increases slightly with *Re*, the curves for different values of *B* are almost horizontal. When Re exceeds 100, there is significant increase of Q_f with *Re*. This is because in the range of *Re* <100, there is always the presence of a convective cell near the outlet opening which deforms the forced jet and so, disadvantages the heat exchange through this opening. As soon as this cell disappears (for Re > 100), the forced flow easily go out carrying with him the major part of the heat exchanged by the cavity. The Q_f variations with *Re* may be correlated as follow:

• **Case of *B* = 3/4**: Q_{f1} = 6.036$Re^{0.061}$ for 10≤*Re*≤100 ; Q_{f2} = 0.966$Re^{0.462}$ for 100<*Re*≤200

• **Case of B = 1/2**: $Q_{f1} = 6.025Re^{0.050}$ for $10 \leq Re \leq 100$; $Q_{f2} = 0.872Re^{0.472}$ for $100 < Re \leq 200$

• **Case of B = 1/4** : $Q_{f1} = 6.025Re^{0.035}$ for $10 \leq Re \leq 100$; $Q_{f2} = 1.593Re^{0.329}$ for $100 < Re \leq 200$

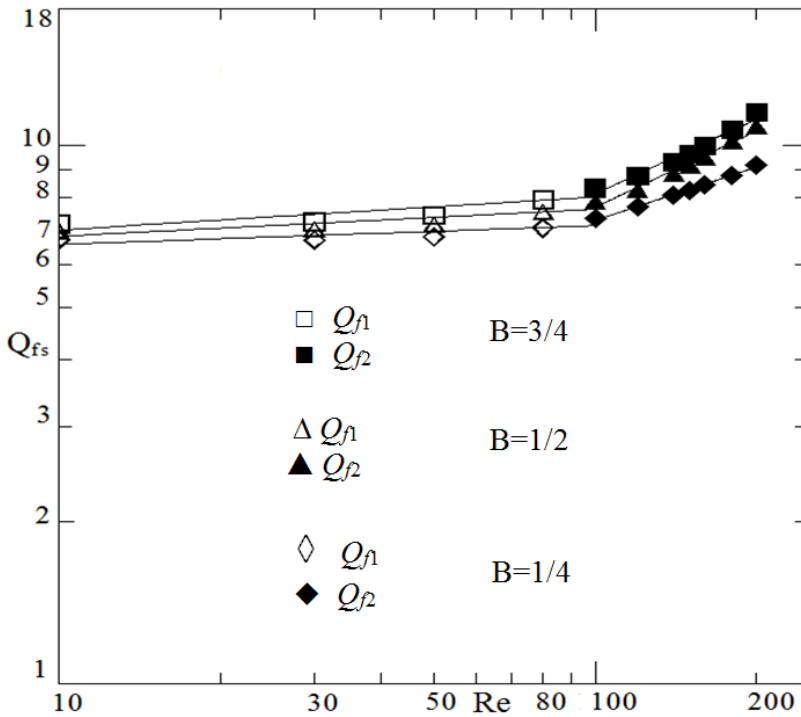Figure 4: Q_f as a function of Re for B=1/4, 1/2, 3/4 ($Ra= 10^5$)

4   Conclusions

Mixed convection in a rectangular partitioned cavity with two heated blocks on the right vertical cold wall has been studied numerically. The results show that competition is established between the forced jet of fresh air and natural convection. In the range of weak Reynolds number ($10 \leq Re \leq 100$), convective cells are produced and the resulting natural convective regime is dominant. The heat exchange is essentially due to these convective cells which by turning bring cooled air from the cold wall to the horizontal faces of the partitions. This mode of ventilation seems to be appropriate to improve the cooling of the horizontal faces of the partitions. Beyond $Re = 100$, however, the forced flow takes over on natural convection. When

such a regime of mixed convection is established, the heat exchange increases with increasing *Re*. The amount of heat Q_{FS} has been correlated to the Reynolds number *Re*. Heat transfer has been found to increase further with an increase in the partitions size.

5   References:

Abbassi M.A., Halouani K., Chesneau X. and Zeghmati B., (2011), Combined Thermal Radiation and Laminar Mixed Convection in a Square Open Enclosure with Inlet and Outlet Ports, *Fluid Dyn. Mater. Process.*, Vol. 7, No. 1, pp. 71-96.

Amirouche Y. and Bessaïh R., (2012), Three-Dimensional Numerical Simulation of Air Cooling of Electronic Components in a Vertical Channel, *Fluid Dyn. Mater. Process*, Vol. 8, No. 3, pp. 295-310

Aydin O., Yang W. (2000): Mixed convection in cavities with a locally heated lower wall and moving sidewalls, *Numerical Heat Transfert*, Part A, Vol. 37, pp. 695-710.

Bahlaoui A., Raji A., Hasnaoui M., Naimi M. (2009): Mixed convection cooling combined with surface radiation in a partitioned rectangular cavity, *Energy Conversion and Management* Vol. 50, pp. 626-635.

Bejan A. (2004): Convection Heat Transfer, 3rd ed., Wiley, Hoboken, NJU,.

ChandraA., Chhabra R.P. (2012): Mixed convection from a heated semi-circular cylinder to power-law fluids in the steady flow regime. *Int. J. Heat and Mass Transfer.* Vol 55, pp. 214–234

Choukairy Kh., Bennacer R, El Ganaoui M. (2006): Transient behaviours inside a vertical cylindrical enclosure heated from the sidewalls, *Num. Heat Transfer (NHT). Part A: Applications.* Vol. 50, No 8, pp. 773 – 785.

De Vahl Davis G. (1983): NATURAL CONVECTION OF AIR IN A SQUARE CAVITY: A BENCHMARK NUMERICAL SOLUTION. INT J NUMER METH FLUID, Vol. 3, PP. 249–264.

De Vahl Davis G. and Jones I.P. (1983) ; Naturel convection in a square cavity : a comparision exercise, *Internalional Journal of Numerical Methods of fluids*, Vol 3, pp: 227-248

Dihmani N., Amraqui S., Mezrhab A. and Laraqi N., (2012), Numerical Modelling of Rib Width and Surface Radiation Effect on Natural Convection in a Vertical Vented and Divided Channel, *Fluid Dyn. Mater. Process.*,.Vol. 8, No. 3, pp. 311-322

Djebali, El Ganaoui M., Sammouda H., Bennacer R. (2009): Some benchmark solutions of a side wall heated cavity using Lattice Boltzmann approach. *Fluid*

Dyn. Mater. Process., Vol.164, no.1, pp.1-21,.

El Alami M.; Najam M.; Semma E.; Oubarra A.; Penot F. (2005) **:** Electronic components cooling by natural convection in an horizontal channel with slots. *Energy conversion and Management,* Vol. 46, pp. 2762-2772.

El Alami M.; Semma E. A.; Najam M.; Boutarfa R. (2008): Convective heat transfer in a horizontal channel with openings and isothermal rectangular blocks. *Fluid Dyn. Mater. Process.*, Vol. 5, No1, pp. 1-9.

Fenglei N.; Haihua Z.; Peterson P. F.; Woodcock J.; Henry R. E. (2007) : Investigation of Mixed Convection in a Rectangular Enclosure, *Nuclear Engineering and Design* Vol. 237, pp. 1025-1032

Kaenton J, Semma E., Timchenko V. El Ganaoui M., Leonardi E., De Vahl Davis G. (2004): Effects of anisotropy and solid/liquid thermal conductivity ratio on flow instabilities during inverted Bridgman growth, *Int. J. Heat Mass Transfer*, Vol. 47, pp. 3403-3413.

Kalache D. (1987): Contribution à l'étude de la convection naturelle en cavités trapézoïdales chauffées par dessous, *Thèse de l'Université de Poitiers*

Ferahta F.Z., Bougoul S., Médale M. and Abid C., (2012), Influence of the Air Gap Layer Thickness on Heat Transfer Between the Glass Cover and the Absorber of a Solar Collector, *Fluid Dyn. Mater. Process.*,.Vol. 8, No. 3, pp. 323-338

Kriaa M., El Alami M., Najam M. and Semma E., (2011)**,** Improving the Efficiency of Wind Power System by Using Natural Convection Flows**,** *Fluid Dyn. Mater. Process*, Vol. 7, No. 2, pp. 125-140

Lappa M., (2004), Combined effect of volume and gravity on the three-dimensional flow instability in non-cylindrical floating zones heated by an equatorial ring, Physics of Fluids, Vol. 16, No. 2, pp. 331-343

Lappa M., Piccolo C., Carotenuto L., (2004), Mixed buoyant-Marangoni convection due to dissolution of a droplet in a liquid-liquid system with miscibility gap, European Journal of Mechanics/B Fluids, Vol. 23, No.5, pp 781-794

Lappa M. (2005) Thermal convection and related instabilities in models of crystal growth from the melt on earth and in microgravity: Past history and current status, *Cryst. Res. Technol.*, Vol. 40, No.6, pp. 531-549

Lappa M., (2007a), Secondary and oscillatory gravitational instabilities in canonical three-dimensional models of crystal growth from the melt, Part2: Lateral heating and the Hadley circulation, *Comptes Rendus Mécanique*, Vol. 335, No (5-6) , pp. 261-268.

Lappa M. (2007b), Secondary and oscillatory gravitational instabilities in canonical three-dimensional models of crystal growth from the melt, Part1: Rayleigh-

Bènard systems, *Comptes Rendus Mécanique*, Vol. 335, No. (5-6) , pp. 253-260

Lappa M., (2011), Some considerations about the symmetry and evolution of chaotic Rayleigh–Bénard convection: The flywheel mechanism and the "wind" of turbulence, *Comptes Rendus Mécanique*, Vol. 339, pp. 563–572

Lappa M., (2013), On the Existence and Multiplicity of One-dimensional Solid Particle Attractors in Time-dependent Rayleigh-Bénard Convection, *Chaos*, Vol. 23, No. 1, 013105

Le Queré and T. Alziary De Roquefort (1985) ; Comptution of natural convection in two-dimensional cavities with chebyshev Polynomials. *J. Computational Physics* Vol. 57, pp. 210-228

Mahapatra S.K.; Sarkar A.; Sarkar A. (2007): Numerical simulation of opposing mixed convection in differentially heated square enclosure with partition, *International Journal of Thermal Sciences* Vol. 46, pp. 970-979.

Meskini A., Najam M., El Alami M. (2011): Convective Mixed Heat Transfer in a Square Cavity with Heated Rectangular Blocks and Submitted to a Vertical Forced Flow. *Fluid Dyn. Mater. Process.*, Vol.7, pp. 97-110.

Mohamad A.A., Bennacer R,. El Ganaoui M. (2010): Double dispersion, natural convection in an open end cavity simulation via Lattice Boltzmann Method, *International Journal of Thermal Sciences*, Vol. 49, No. 10, , Pages 1944-1953

Najam M.; El Alami M.; Hasnaoui M.; Amahmid A. (2002): Etude numérique de la convection mixte dans une cavité en forme de T soumise à un flux de chaleur constant et ventilée par le bas à l'aide d'un jet d'air vertical, *Compte Rendu de l'Académie des Sciences*, C. R. Mécanique Vol. 330 , pp.461-467, (France).

Roache P.J. (1982): Computational fluid dynamics, *Hermosa Publishers*

Rouijaa H., El Alami M., Semma E., Najam M. (2011)*:* Natural Convection in an Inclined T-Shaped Cavity**.** *Fluid Dyn. Mater. Process***,** Vol.7, No. 1 pp. 57-70

Tomimura T.; Fujii M. (1988): Laminar mixed heat transfer between parallel plates with localised heat sources. *In: Proc. Int. Symp. on Cooling Technology for Electronic Equipment*, Honululu, pp. 233–247.