NUMERICAL STUDY OF THE EFFECT OF THE INSULATION OF THE WALLS ON NATURAL CONVECTION IN A DIFFERENTIALLY HEATED CAVITY

A. Raji*, M. Hasnaoui, S. Goujon-Durand***, P. Vasseur******

*Faculty of Sciences and Technologies, Department of Physics, University Sultan Moulay Sliman, Team of Flows and Transfers Modeling (EMET), Laboratory of Physics and Mechanics of Materials, BP 523, Béni-Mellal, Morocco
Tel.: 00212 23 48 51 12; fax: 00212 23 48 52 01, E-mail: abderaji@fstbm.ac.ma; abderaji@yahoo.fr

**Faculty of Sciences Semlalia, Department of Physics, University Cadi Ayyad, UFR TMF, BP 2390 Marrakesh, Morocco
Tel.: 00212 44 43 46 49; fax: 00212 44 43 74 10, E-mail: hasnaoui@ucam.ac.ma

***Université Paris XII, France

****Ecole Polytechnique de Montréal, Canada

Received: 27 Sept 2007; accepted: 31 Oct 2007

In this work, we study numerically the natural convection in a square cavity using the Navier-Stokes equations with the Boussinesq approximation. A finite difference method based on the control volume notion is used to resolve, in nondimensional forms, the natural convection equations. The results are obtained for various combinations of the parameters of base, such as the Rayleigh number ($10^3 \leq Ra \leq 5\cdot10^6$), the relative thermal conductivity ($0.1 \leq K \leq$ ≤ 100), the relative width of the walls ($0 \leq e \leq 0.2$) and the relative spacing ($0.05 \leq e_{spa} \leq 0.3$). The numerical simulations of natural convection in a cavity having walls of finite width have shown the possibility to reduce significantly the heat transfer by using appropriate isolation techniques.

Keywords: natural convection, numerical study, effect of the insulation

Abdelghani Raji

Organization: Teacher at the Faculty of Sciences and Technics in Beni Mellal (Morocco).
Education: Doctorates in heat transfer (1994 and 2000) after attending Cadi Ayyad University. The title of the first doctorate was: "Etude numérique des écoulements et des transferts de chaleur par convection dans des cavités en interaction et dans un canal de longueur finie". The title of the second thesis was: "Etude numérique du phénomène de la convection mixte dans des cavités ventilées avec et sans effet de rayonnement".
Experience: Head of the physics department from 2000 to 2003.
Main range of scientific interests: heat transfer by conduction, convection, radiation, porous media…
Publications: from 2004:

− Coupling between mixed convection and radiation in an inclined channel locally heated. *Journal of Mechanical Engineering, 2004.*

− Interaction between natural convection and radiation in a square cavity heated from below. *Numerical Heat Transfer, Part – A: Applications, 2004*.

− Multiple steady state solutions resulting from coupling between mixed convection and radiation in an inclined. *Heat and Mass Transfer, August 2005.*

− Combined effect of radiation and natural convection in a rectangular enclosure discreetly heated from one side. *International Journal of Numerical Methods for Heat & Fluid Flow, 2006.*

− Multiplicité de solutions en convection naturelle couplée au rayonnement dans une cavité horizontale. *Physical & Chemical News, 2006.*

− Mixed convection in a horizontal channel with emissive walls and partially heated from below. *Numerical Heat Transfer, Part – A: Applications, 2007.*

− Multiple steady state solutions for natural convection in a tilted rectangular slot containing non-Newtonian power-law fluids and subject to a transverse thermal gradient. *Numerical Heat Transfer, Part – A: Applications, 2007.*

− Parallel flow convection in a shallow horizontal cavity filled with non-newtonian power-law fluids and subject to horizontal and vertical uniform heat fluxes. In Press in *Numerical Heat Transfer, Part – A: Applications, 2007.*

− Coupled natural convection and radiation in a horizontal rectangular enclosure discretely heated from below. *Accepted in Numerical Heat Transfer, Part – A: Applications, 2007.*

Nomenclature

A – aspect ration of the cavity, H'/L'
e – relative width of the wall, l'/L'
e_{ins} – relative width of the insulating layer, l'_{ins}/L'
e_{spa} – spacing between the walls, l'_{spa}/L'
g – acceleration due to gravity, m/s^2
h – mean convection heat transfer
K – relative conductivity, λ/λ_f
l' – dimensional width of the wall, m
l'_{ins} – dimensional width of the insulating layer, m
l'_{spa} – dimensional spacing between the walls, m
P – dimensionless pressure, $(P' + \rho_0 g y')/\rho_0(\alpha/L')^2$
Pr – Prandtl number, ν/α
Ra – Rayleigh number, $g\beta\Delta T' L'^3/(\nu\alpha)$
r_w – relative conductivity of the wall, λ_w/λ_f
r_{ins} – relative conductivity of the insulator, λ_{ins}/λ_f
$\Delta T'$ – temperature difference, $(T'_H - T'_C)$
(u, v) – dimensionless horizontal and vertical velocities, $(u', v')/(a/L')$
(x, y) – dimensionless Cartesian coordinates, $(x', y')/L'$

Greek letters

α – thermal diffusivity (m^2·s^{-1})
β – thermal expansion coefficient of the fluid (K^{-1})
Γ – diffusion coefficient
λ – thermal conductivity (W·m^{-1}·K^{-1})
ν – kinematic viscosity of fluid (m^2·s^{-1})
ψ – dimensional stream function (m^2·s^{-1})
Ψ – dimensionless stream function, ψ/α
θ – dimensionless temperature, $\theta = (T' - T'_C)/(T'_H - T'_C)$
ρ_0 – fluid density at the temperature T_0 (kg·m^{-3})

Subscripts and Superscripts

board – boarding
C – cooled
cond – conduction
f – fluid
H – heated
ins – insulator
max – maximum
r – relative
s – solid
spa – spacing
w – wall
′ – dimensional variables

Introduction

During the last decades, natural convection in closed cavities has received great efforts of research. This special attention was dictated by the importance of this phenomenon in the design of the physical models related to the field of solar engineering, the cooling of the printed circuit boards and the design of buildings. Hence in several practical applications, the use of materials with high thermal conductivity will lead to an undesirable increase of the temperature of the fluid inside the cavity. The maintenance of a desirable temperature can be achieved if an adequate insulation of the walls of the cavity is taken into account. Heat exchange between the walls of the cavity and the fluid can be controlled by recourse to thermally insulating materials. A problem of base well documented in the literature specialized in heat transfer is that of a rectangular enclosure without partition. An exhaustive review on this subject is presented in references [1-4]. Compared to the studies relating to rectangular cavities, those concerning partially or completely partitioned cavities are relatively less documented. It is obvious that the addition of one or more partitions has a considerable effect on the heat transfer and the fluid flow structure inside the cavity. Thereafter, the subject needs to be further explored and other works are necessary to improve our knowledge in this field by taking account of the influence of partitions; especially if it is known that their dimensions, their conductivities, their number and the way in which they are placed are among the parameters which control the importance of this influence in various technological aspects [5-10].
The main goal of this study consists in numerically simulating the two-dimensional natural convection in a differentially heated cavity whose vertical walls have a finite conductivity and a thickness. The effect of the insertion of an insulating material on the internal surfaces of the walls will be also investigated in parallel with the technique of the thermal boarding (use of a double wall separated by an insulator). The effects of these various techniques of insulation on the heat transfer and the fluid flow will be studied for various values of the Rayleigh number, Ra, the relative thermal conductivities r_w and r_{ins} and for various combinations of the geometrical parameters.

Problem formulation

The configurations under study with the system of coordinates are sketched in Fig. 1. They consist of a differentially heated cavity whose vertical walls have a finite thickness without insulation (Fig. 1, *a*), with an internal insulation (Fig. 1, *b*) and with an insulation using the thermal boarding technique (Fig. 1, *c*). For these three types of configurations, the horizontal walls are adiabatic. The fluid flow is assumed to be laminar and two-dimensional and the Boussinesq approximation is adopted for the variations of the density in the buoyancy term.

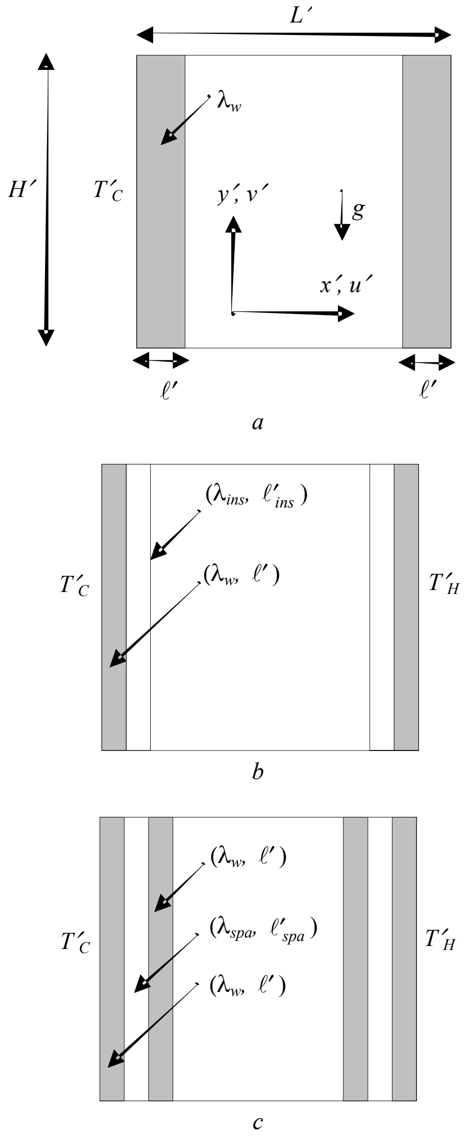


Fig. 1. *Studied configurations*

The equations governing the dynamic and thermal fields are those of Navier-Stokes coupled with the equation of energy transport. In stationary and non-dimensional form, they are written as:

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{\partial P}{\partial x}+\Gamma\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right), \qquad (1)$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{\partial P}{\partial y}+\Gamma\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+RaPr\theta, \qquad (2)$$

$$u\frac{\partial \theta}{\partial x}+v\frac{\partial \theta}{\partial y}=K\left(\frac{\partial^2 \theta}{\partial x^2}+\frac{\partial^2 \theta}{\partial y^2}\right). \qquad (3)$$

The parameters Γ and K are respectively the diffusion coefficient and the relative thermal conductivity. A value of unity is allotted to Γ in the fluid area while an "infinite" value (1015) is affected for it in the solid area. The relative thermal conductivity K is equal to unity in the fluid medium and its value is k_r for the solid walls. The parameters Γ and K are easily implanted in the numerical code and their use satisfies automatically the boundary conditions at the solid-fluid interfaces.

The thermal and dynamic boundary conditions associated to this problem are:

$$\begin{aligned}
&\theta = 0 && \text{for } x = 0;\\
&\theta = 1 && \text{for } x = 1;\\
&\frac{\partial\theta}{\partial y} = 0 && \text{for } y = 0 \text{ and } y = 1;\\
&u = v = \Psi = 0 && \text{on the solid walls.}
\end{aligned} \qquad (4)$$

The average heat quantity transferred through the active walls of the cavity is defined by mean of the following expression:

$$Q=\int_0^1 \frac{\partial\theta}{\partial x}\,dy \qquad (5)$$

$$Nu=\frac{Q}{Q_{cond}}, \qquad (6)$$

where Q_{cond} is the heat flux by conduction evaluated by equation (5) for $Ra = 0$.

Numerics

The discretization of the governing equations, written in primitive variables, is carried out by using a finite difference method and the numerical integration is based on the control-volume approach with the SIMPLER algorithm and the hybrid scheme for interpolation [11]. The technique of the finite volume method is based on the principle of conservation of the various physical quantities (mass, momentum and energy). In order to prevent that physical incompatibilities of the pressure do not generate a convergence of the total solution towards a nonphysical solution, a uniform shifted grid was used for the velocity. The numerical code was validated by comparing the results obtained in the case of a differentially heated cavity with those of the Bench-Mark solution of De Vahl Davis [3]. As it can be seen in Table 1, the agreement observed is excellent and the maximum relative deviation remains less than 1.67 and 1.34 % respectively in terms of Ψ_{max} and Q. The numerical code was also validated by comparing our results with those of the references [12-14] in the case of a cavity containing an obstacle placed at its centre. The comparative results, presented in Table 2, show an excellent agreement with a maximum deviation less than 1 %. Also for all the computations, it was carefully verified that the heat provided by the heated wall to the fluid is equal to that leaving the cavity through the cold wall with 0.2 % as a maximum difference. It is interesting to announce also that in the pure conduction mode, the heat flux through a cavity containing various solid layers, having each one a thermal conductivity, coincides perfectly with that calculated analytically by using the concept of thermal resistance. The agreement is excellent and the maximum deviation remains lower than 0.5 %. The sensitivity of the results with respect to grid size is summarized in Table 3 in terms of Ψ_{max} and *Nu*. It can be retained that a uniform grid of 80×80 can describe correctly the heat transfer and fluid flow. In fact, a refinement of the mesh to 120×120 involves relatively weak maximum deviations but induces consequently higher computing times. The parameter of relaxation varies between 0.4 for low values of *Ra* and 0.1 for high values of this parameter. Finally, the convergence criterion, based on the correction of the pressure, is checked when the corrected terms are sufficiently weak and the existing difference between the fields of pressure before and after each correction is nonsignificant.

Table 1

Validation of the numerical code in terms of the maximum stream function and the average heat quantity

	Present study		Bench Mark [3]	
Ra	Ψ_{max}	*Q*	Ψ_{max}	*Q*
10^3	1.176	1.102	1.174	1.117
10^4	5.078	2.221	5.071	2.238
10^5	9.642	4.496	9.612	4.509
10^6	17.031	8.880	16.750	8.817

Table 2

Comparative results of the average heat quantity for a square enclosure containing a conducting square solid body at the centre

Ra	k_r	Present study	House et al. [12]	Merrikh and Lage [13]	Lee and Ha [14]
10^5	0.2	4.623	4.624	4.605	4.631
10^5	5	4.317	4.324	4.280	4.324

Table 3

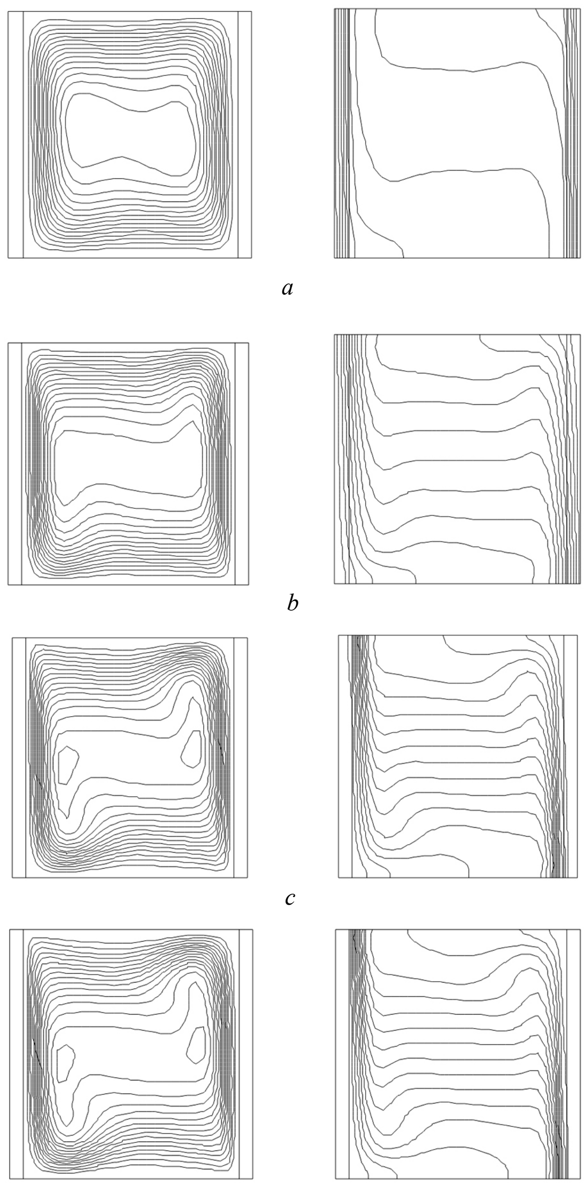
Variations of the maximum stream function, Ψ_{max}, and the average heat quantity, *Q*, according to the grid size for $Ra = 10^6$

Mesh	Ψ_{max}	*Q*	CPU(s)
40×40	17.55	9.14	191
80×80	17.03	8.86	1031
120×120	16.92	8.82	6346

Results and discussion

1st configuration: walls without insulation

Typical results of the streamlines and the isotherms illustrating the effect of the relative conductivity of the walls are presented in Fig. 2 for $Ra = 10^6$, $e = 0.05$ and r_w ranging between 0.1 and 50. For walls with a low thermal conductivity ($r_w = 0.1$), Fig. 2, *a* shows that the fluid circulation is relatively weak. The corresponding isotherms indicate that the horizontal heat gradients in the solid walls are important indicating a great resistance to the heat flux between the extreme faces of each wall.



a

b

c

d

Fig. 2. *Streamlines and isotherms for $Ra = 10^6$, $e = 0.05$ and different values of r_w:*
a) $r_w = 0.1$ ($\Psi_{max} = 6.62$, $Nu = 1.46$),
b) $r_w = 1$ ($\Psi_{max} = 11.42$, $Nu = 3.92$),
c) $r_w = 10$ ($\Psi_{max} = 16.01$, $Nu = 7.13$)
d) $r_w = 50$ ($\Psi_{max} = 16.85$, $Nu = 7.76$)

This will lead to a limited interaction with the fluid inside the cavity. For $r_w = 1$, Fig. 2, *b* shows an improvement of the fluid circulation characterized by higher values of Ψ_{max} and a spacing of the isotherms at the level of the walls as a consequence of the reduction in their thermal resistance. By increasing r_w to 10, Fig. 2, *c* shows that the fluid circulation continues to be intensified. The corresponding isotherms show an uniformisation of the temperatures of the two walls and a better interaction of the latters with the fluid. For high values of relative conductivity ($r_w = 50$), Fig. 2, *d* shows that the flow and temperature fields are qualitatively similar and quantitatively comparable to those described in the preceding figure corresponding to $r_w = 10$.

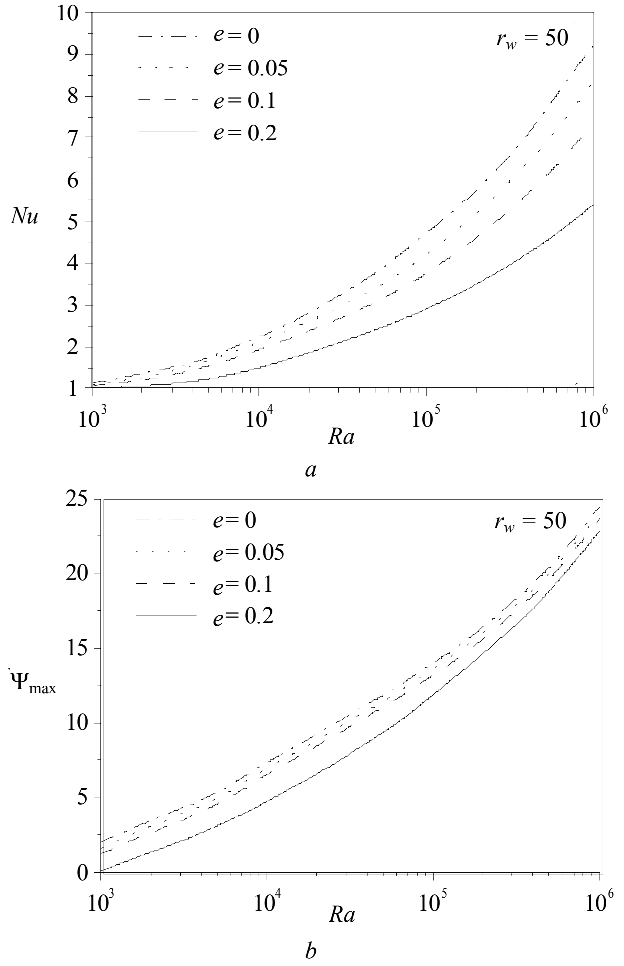


Fig. 3. *Variations of Nu and Ψ_{max} with Ra for e = 0.05, r_w = 50 and different values of the relative width of the walls*

The thickness of the walls was varied in the range $0 \le e \le 0.2$ in order to analyze its effect on the fluid flow and heat transfer. The results obtained are presented in Fig. 3 in terms of variations of *Nu* and Ψ_{max} with *Ra* for $r_w = 50$ and various values of *e*. The results of base obtained for $e = 0$ (absence of conduction in the walls) are also presented for a comparison purpose. For values of $Ra \le 1500$, the heat transfer is done mainly by conduction and is slightly affected by the thickness of the walls. For higher values of *Ra*, the effect of the thickness becomes visible and the Nusselt number decreases when this last parameter increases. For *Ra* = 106 and *e* = 0.2, the reduction of *Nu* reaches approximately 48 % when compared to the reference case (*e* = 0). This tendency is a consequence of the increase in the conductive thermal resistance of the walls with the thickness.

The effect of the Rayleigh number on the Nusselt number and the maximum stream function is presented in Fig. 4 for $e = 0.05$ and various values of relative conductivity r_w of the walls. For the lower considered value of r_w ($r_w = 0.1$), the Nusselt numbers remain generally close to unity, showing thus the resistant effect of the walls to the heat transfer and this even for high values of the Rayleigh number. For a given value of *Ra*, we can note an increase in the heat quantity restored to the fluid by the active surfaces. This increase of *Nu* with r_w cease as soon as this last parameter becomes higher than 50 as it can be seen in Fig. 5 where the Nusselt number and the maximum stream function tend asymptotically to limiting values. For $Ra = 10^6$, it is interesting to note that the use of walls with a conductivity such $r_w = 10$ would lead to a reduction of 9.8 % of *Nu* when compared to walls of relative conductivity $r_w = 50$. This reduction reaches 16.6 % when the conductivity of the walls is $r_w = 5$.

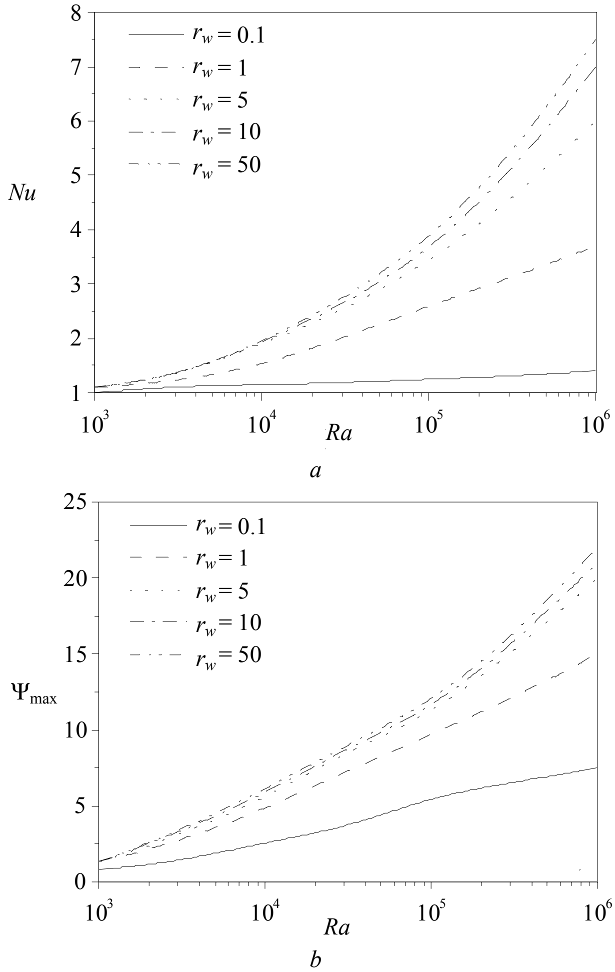
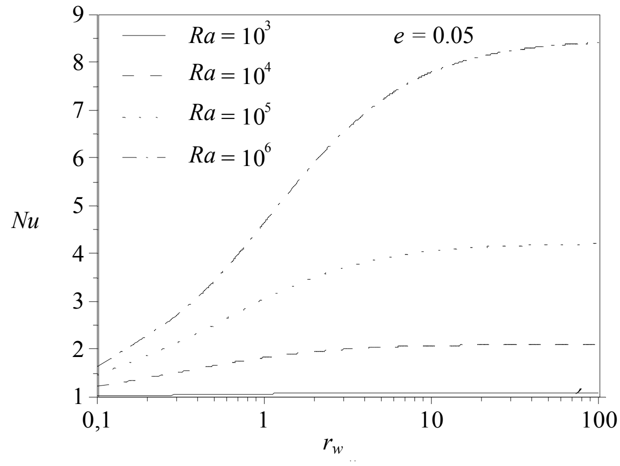
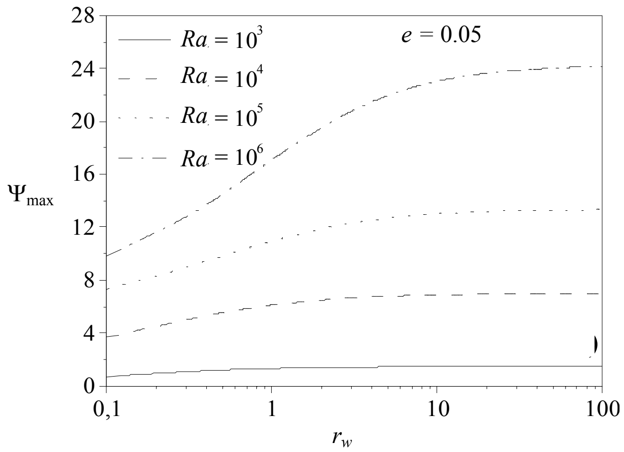


Fig. 4. *Variations of Nu and Ψ_{max} with Ra for e = 0.05 and different values of the relative conductivity r_w of the walls*



a



b

Fig. 5. *Effect of the relative thermal conductivity r_w of the walls on Nu and Ψ_{max} for e = 0.05 and different values of the Rayleigh number*

2nd configuration: walls with insulation

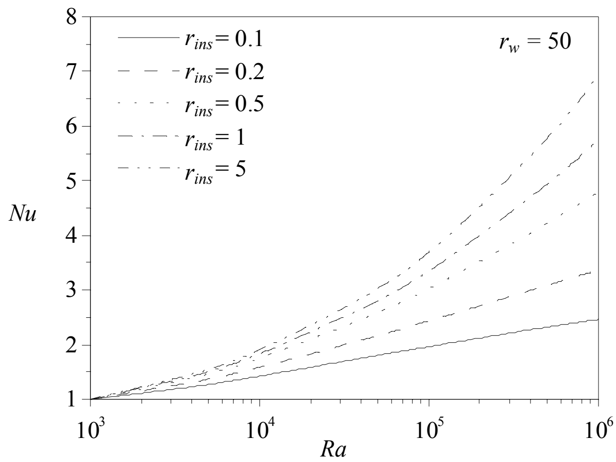
In this section, we will examine the effect of the increase of the thermal resistance of the walls by adding an insulating layer on the interior surfaces. The conductivity of this layer will be varied in order to obtain a rate of heat transfer comparable or lower than that obtained by using materials of relative conductivities of r_w = 5 or 10. The thicknesses of the insulating layer and that of the wall are fixed respectively at e_{ins} = 0.0125 and e = 0.0375, in such a way that the total thickness remains equal to that of the already studied configuration ($e_{ins} + e = 0.05$).

The effect of the addition of an insulating layer on internal surfaces of the walls will be examined for a relative conductivity of the walls fixed at r_w = 50 and a thickness e = 0.0375. Hence, variations of *Nu* and Ψ_{max} with *Ra* are shown in Fig. 6 for e_{ins} = 0.0125 and various values of the relative conductivity of the insulator. It can be seen that for the low values of *Ra*, the Nusselt number is practically independent of the relative conductivity of the insulator. It is a foreseeable result since these values of *Ra* correspond to a mode for which conduction contributes largely to the heat transfer. The increase of *Ra* supports the fluid circulation within the cavity and generates consequently an increase of *Nu*. Also, for a given value of *Ra*, the Nusselt number decreases when relative conductivity of the insulator decreases. A comparison with the results of the already studied configuration shows an appreciable reduction of the rate of heat transfer. The percentages of this reduction compared to walls of different relative conductivities are presented in Table 4 for $Ra = 10^6$ and for various values of the relative conductivity of the insulator.

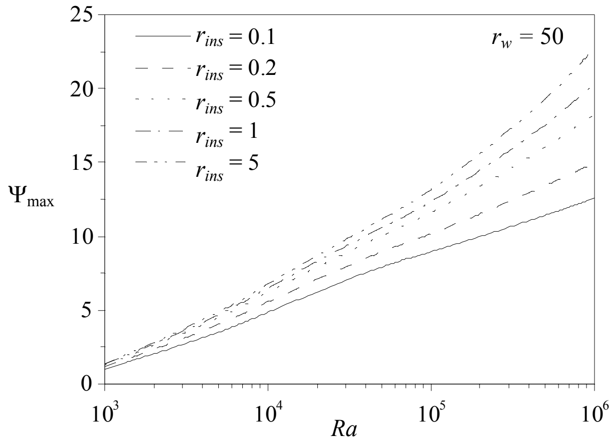
Table 4

Percentage of reduction of *Nu* due to the addition of an insulation for $Ra = 10^6$ and various values of the conductivity of the insulator

$\frac{Nu - Nu_{ins}}{Nu} \times 100$	r_{ins}				
	0.1	**0.2**	**0.5**	**1.0**	**5.0**
r_w = 50	68.3	56.6	38.4	26.2	11.2
r_w = 10	64.8	52.0	31.6	18.1	1.6
r_w = 5	62.0	48.1	26.1	11.4	---



a



b

Fig. 6. *Effect of Ra on Nu and Ψ_{max} for rw = 50 and various values of the relative thermal conductivity r_{ins} of the insulating layer*

3^{rd} configuration: case of thermal boarding

Another technique of insulation consists in using two walls separated by a fluid or solid layer playing the role of an insulator. The thickness of the space between the two walls is taken equal to that of the insulator (e_{spa} = = 0.0125). The results obtained by using this technique of insulation are found identical to those of the second configuration. For the same values of r_{ins}, the maximum deviation compared to this situation remains lower than 0.45 %. Hence, the air was considered as insulator and its thickness was varied to study its impact on the heat transfer.

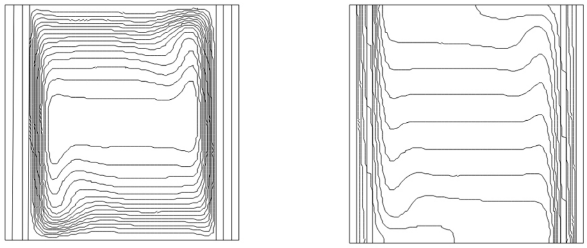
The effect of the spacing between the walls is examined in this section by producing streamlines and isotherms for $Ra = 5{\cdot}10^6$, $r_w = 50$ and different values of e_{spa}. The walls have each one a thickness equal to $e = 0.025$ so that the total thickness is equal to that of the wall of Fig. 1, *a*. For an air layer of thickness $e_{spa} = 0.05$, Fig. 7, *a* shows a structure close to that obtained for a differentially heated cavity. The corresponding isotherms show a clear stratification in the central part of the cavity. An increase of e_{spa} to 0.1 supports the formation of convective cells of low intensities in the insulating layers (Fig. 7, *b*). The examination of the isotherms between the walls shows that conduction is the heat transfer dominating mode in these areas. From Fig. 7, *c*, it can be seen that an increase of e_{spa} to 0.15 is accompanied by an improvement of the fluid circulation in the insulating fluid layers and of a reduction of the general circulation of the fluid within the cavity which will lead to a reduction of the Nusselt number. The numerical tests carried out showed that $e_{spa} = 0.15$ is the optimum value which tolerates a weaker thermal interaction between the active walls of the enclosure for the considered value of *Ra*. A spacing beyond this critical value supports the role of natural convection in the insulating layers and generates a better interaction between the active walls of the cavity and the fluid. This results in a noticeable improvement of the general circulation of the fluid and an increase of *Nu*. The Fig. 7, *d* corresponding to $e_{spa} = 0.2$, shows a distortion of the isotherms between the walls resulting from the intensification of the flow in the insulating layers and a stratification of the temperature better than that observed in the case of Fig. 7, *c*.

The effect *Ra* on the Nusselt number and the maximum stream function is presented in Fig. 8 for $r_w = 50$ and various values of e_{spa}. It can be seen that, for a fixed value of *Ra*, the Nusselt number decreases with the increase of the thickness of separation. Compared to the case of a wall without insulation, the technique of the boarding appears effective to reduce the rate of heat transfer through the cavity. This technique leads to performances better than those of materials having conductivities of $r_w = 5$ and 10. Quantitatively, the percentage of this reduction is presented in Table 5.

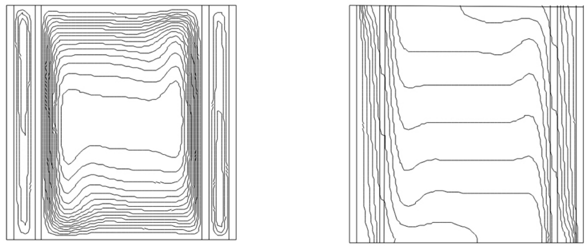
Table 5

Percentage of reduction of *Nu* in comparison with a cavity without insulation for $r_w = 50$, $Ra = 10^6$ and different values of the spacing

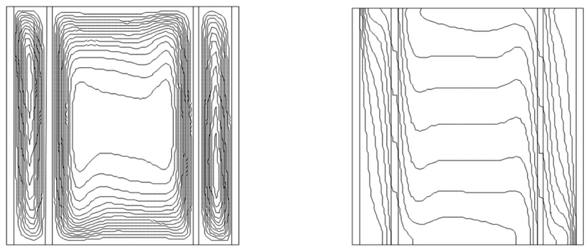
$\frac{Nu - Nu_{board}}{Nu} \times 100$	e_{spa}			
	0.05	**0.1**	**0.2**	**0.3**
$r_w = 50$	39.88	62.03	74.22	70.36



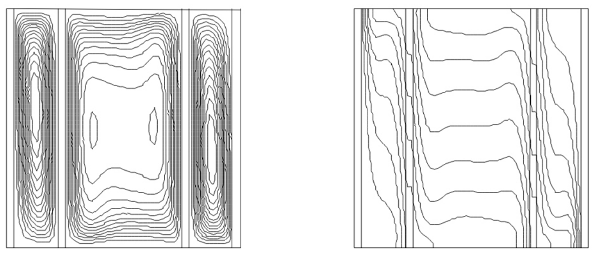
a



b

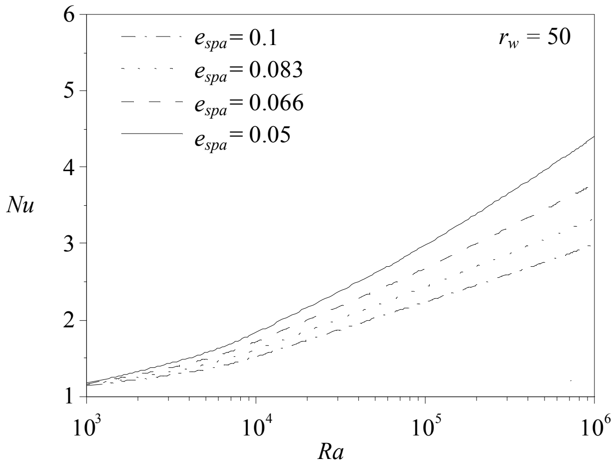


c

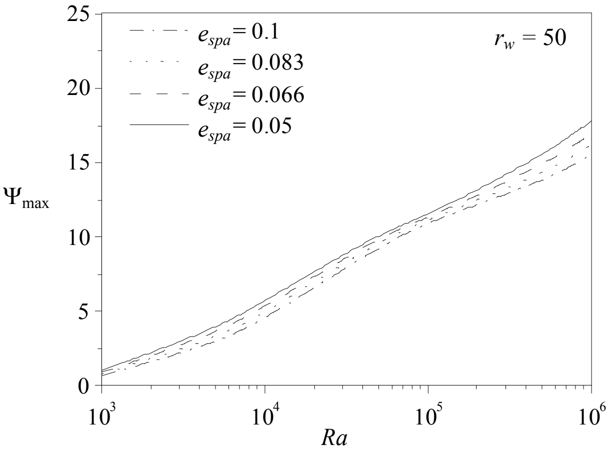


d

Fig. 7. *Streamlines and isotherms for* $Ra = 5{\cdot}10^6$, $r_w = 50$, $e = 0.025$ *and different values of the spacing between the walls: a)* $e_{spa} = 0.05$ ($\Psi_{max} = 18.78$, $Nu = 6.82$), *b)* $e_{spa} = 0.1$ ($\Psi_{max} = 14.96$, $Nu = 4.17$), *c)* $e_{spa} = 0.15$ ($\Psi_{max} = 12.82$, $Nu = 3.90$) *d)* $e_{spa} = 0.2$ ($\Psi_{max} = 15.28$, $Nu = 4.13$)



a



b

Fig. 8. Effect of the Rayleigh number on Nu and Ψ_{max} for r_w = = 50 and various values of the relative spacing between the walls

From Table 5, it can be seen that the percentage of reduction of the heat transfer increases with e_{spa} as long as this parameter is lower or equal to 0.2. Beyond this value a decrease is observed for e_{spa} = 0.3. This testifies of the existence of a critical value of the spacing between the walls which would engender the maximum reduction of the heat transfer. In Fig. 9, the variations of the Nusselt number, with e_{spa}, are presented for r_w = 50, e = 0.025 and different values of Ra. The examination of this curve confirms well that a spacing of about 0.2 induces a minimum heat transfer for $Ra = 10^6$. Beyond this critical value, any increase of e_{spa} would lead to an increase in the intensity of the flows in the space between the walls and consequently to an increase in the resulting heat transfer. For $Ra = 5{\cdot}10^6$, it can be seen that the value of the critical spacing inducing a minimum heat transfer is of about 0.15. This critical value, lower than that corresponding to $Ra = 10^6$, is due to the early development of the effect of the natural convection in the fluid layers intended to play an insulating role. It appears thus obvious that the technique of the thermal boarding is effective to ensure a good insulation. But, it is important to underline that the performances of this technique are achieved to the detriment of the space allowed to the fluid inside the cavity.

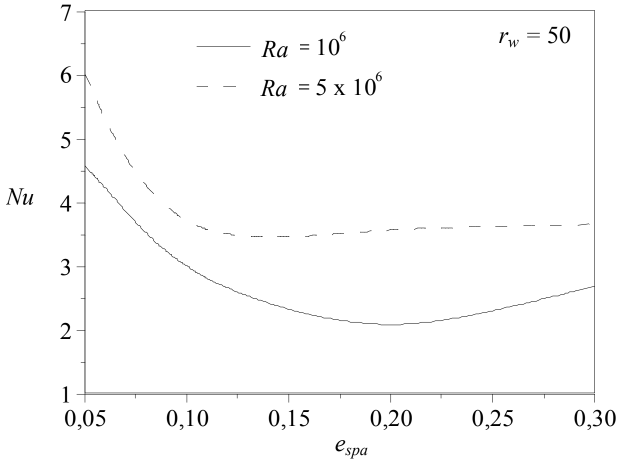


Fig. 9. *Variations of the Nusselt number with the spacing between the walls for various Ra*

Conclusion

In this work, a numerical study was carried of on the natural convection in a square cavity with conducting vertical walls by using two techniques of heat insulation. The obtained results show that the thermal performances of walls with high relative conductivities can be improved considerably by using an insulating layer. The technique of the thermal boarding was found interesting to reduce in an appreciable way the heat transfer in comparison to the cases of the walls with or without insulation. In this technique, the existence of a range of the spacing between the walls which generates weak interactions with fluid medium was observed.

References

1. Bejan A. Convective heat transfer. John Wiley & Sons, 1984.
2. Yang K.T. Natural convection in enclosures. In Hand Book of Single Phase Convective Heat Transfer. New York, 1987.
3. De Vahl Davis G. Natural convection of air in a square cavity: a bench-mark numerical solution // Int. J. for Numerical Methods in Fluids. 1983. Vol. 3. P. 249-264.
4. Rasoul I., Prinos P. Natural convection in an inclined enclosure // Int. J. for Numerical Methods for Heat & Fluid Flow. 1997. Vol. 7, No. 5. P. 438-478.
5. Acharya S. and Jetli R. Heat transfer due to buoyancy in a partially divided square box // Int. J. Heat Mass Transfer. 1990. Vol. 33, No. 5. P. 931-942.
6. Kelkar K.M., Patankar S.V. Numerical prediction of natural convection in square partitioned enclosures // Num. Heat Transfer. 1990. Part A. Vol. 17. P. 269-285.

7. Ntibarufata E., Hasnaoui M., Bilgen E., Vasseur P. Natural convection in partitioned enclosures with localized heating // Int. J. for Numerical Methods for Heat & Fluid Flow. 1993. Vol. 3. P. 133-143.
8. Vasseur P., Hasnaoui M., Bilgen E. Analytical and Numerical study of natural convection heat transfer in inclined composite enclosures // Applied Scientific Research. 1994. Vol. 52. P. 187-207.
9. Mamou M., Hasnaoui M., Bilgen E., Vasseur P. Natural convection heat transfer in inclined enclosures with multiple conducting solid partitions // Num. Heat Transfer. Part A. 1994. Vol. 25. P. 295-315.
10. Raji A., Hasnaoui M., Zrikem Z. Natural convection in interacting cavities heated from below // Int. J. for Numerical Methods for Heat & Fluid Flow. 1997. Vol. 7, No. 6. P. 580-597.
11. Patankar S.V. Numerical heat transfer and fluid flow. New York: Hemisphere, 1980.
12. House J.M., Beckermann C., Smith T.F. Effect of a centered conducting body on natural convection heat transfer in an enclosure // Numer. Heat Transfer. Part A. 1990. Vol. 18. P. 213–225.
13. Merrikh A.A., Lage J.L. Natural convection in an enclosure with disconnected and conducting solid blocks // Int. J. Heat Mass Transfer. 20005. Vol. 48, No. 7. P. 1361–1372.
14. Lee J.R., Ha M.Y. Numerical simulation of natural convection in a horizontal enclosure with a heat-generating conducting body // Int. J. Heat Mass Transfer. 2006. Vol. 49. P. 2684–2702.