24th ABCM International Congress of Mechanical Engineering
December 3-8, 2017, Curitiba, PR, Brazil

COBEM-2017-1245
NUMERICAL SIMULATION OF CONVECTION IN A LID-DRIVEN CAVITY FILLED WITH HETEROGENEOUS POROUS MEDIUM

Allan P. de Camargo
Felipe S. Santos
Vinícius G. Poletto
Fernando C. De Lai
Silvio L. M. Junqueira
Research Center for Rheology and Non-Newtonian Fluids (CERNN), Postgraduate Program in Mechanical and Materials Engineering (PPGEM), Federal University of Technology – Paraná (UTFPR), Curitiba, Brazil.
felipe_ssantos91@hotmail.com, allan_pais@terra.com.br, vinsptt19@gmail.com,
fernandodelai@utfpr.edu.br, silvio@utfpr.edu.br

Abstract. *The convection within a lid-driven square porous cavity subjected to a gravitational stable condition is numerically simulated. The porous media is modeled in the pore-scale with the solid constituent as square, solid, identical and equally spaced heat conductive blocks. To evaluate the influence of the number of blocks over the heat transfer, the balance equations are considered separately for the fluid and the solid domains with coupling boundary conditions applied at the block surfaces. The modeling equations are solved numerically via the Finite Volume method using the QUICK and SIMPLE schemes. The fluid volume fraction and the Prandtl number are kept constant as* $\varphi=0.36$ *and* $Pr=1$*. The effect of the variation of the Reynolds number for* $Re=(1000, 2500)$*, the Grashof number* $Gr=(10^5, 10^6, 10^7)$ *and the number of blocks* $N=(1, 4, 9, 16, 25, 36, 49, 64)$ *is investigated. Results demonstrate that buoyancy restrains the flow and eventually stagnate the fluid near the cavity base establishing a conduction-dominant heat transfer regime. In general, the introduction of blocks imposes more resistance to the flow reducing the fluid circulation and, consequently, the surface-averaged Nusselt number. Nevertheless, the blockage effect leads to the heat transfer augmentation for a number of cavity configurations.*

Keywords: *heterogeneous porous media; lid-driven cavity; pore-scale flow; gravitational stable condition.*

1. INTRODUCTION

Studies concerning the convection within lid-driven cavities are useful in modeling complex engineering process dealing with lubrication, metal coating and polymer processing (Ismael, *et al.*, 2014; Cheng, *et al.*, 2010; Ramakrishna, *et al.*, 2012).

Usually, the fluid confined in the enclosure is also under the effects of buoyancy that arise from the action of a specific mass gradient and the gravity. The alignment of the gradient with the buoyant forces may result in a number of flow patterns that may promote or hinder the heat transfer (Cheng, 2010). For instance, if a negative thermal gradient aligned with gravity happens (which is also in the negative direction) a gravitational unstable condition occurs and mixed convection is observed with buoyant-induced flow (free convection) coexisting with forced convection (related with the lid movement). The numerical results of Cheng (2011) for the heat transfer in a lid-driven cavity whose bottom is kept at a temperature higher than the one for the sliding-lid elucidate the dependence of the surface-averaged Nusselt (Nu_{av}) with the Reynolds number (Re). The variation of the Grashof number, Gr, on the other hand, leads to a stratified flow pattern creating a clock-wise circulation on the cavity upper half and a counter clock-wise circulation on the lower half. On the other hand, there is the gravitational stable condition, which is achieved aligning a positive thermal gradient (negative specific mass gradient) with the gravity. In such situation, the fluid is stable-stratified and the free convection does not occur spontaneously. Unless the convection is promoted by external means, like for instance the movement of the lid, the fluid remains quiescent and the only mean to transfer heat is by diffusion. The lid-driven cavity subjected to a gravitational stable condition is an interesting topic because there is a strong competitive effect between the fluid inertia, which tends to penetrate the flow toward the base of the enclosure, and the buoyancy that hinders the fluid flow. Such condition is studied numerically by Iwatsu *et al*. (1993) and Poletto *et al*. (2015) in a square cavity whose lid is at a temperature higher than the one for the base. In their results two heat transfer patterns are observed, namely, a convection-dominant and conduction-dominant. The latter is related with the increase in Gr, or decrease in Re, which leads to the stagnation of the fluid near the enclosure base and the restraining of the flow to the cavity upper half. The heat transfer is mainly conductive and, therefore, the Nu_{av} has unitary scale. The convection-dominant situation occurs as the fluid circulation spreads all across the enclosure. Poletto *et al*. (2016) proposed a scale to characterize both regimes based on the inverse of the Richardson number, i.e., as $O(Re^2/Gr)>1$ represents a convective regime and $O(Re^2/Gr)<1$ a conductive one. Additionally, Iwatsu *et al*. (1992) considered a cavity whose lid moves periodically according to a sinusoidal function. In this case resonance is observed whether the lid moves with the natural frequency of the fluid, amplifying the heat transfer. The three-dimensional aspects of such flow are also debated based on numerical results by Iwatsu *et al*. (1995).

In the present problem, the lid-driven cavity filled with heterogenous porous media is considered. Since the modeling of transport phenomena in porous media is strongly related to the visual definition, the heterogeneous porous media here investigated is conceived in the pore-scale as the solid to fluid interface is visually distinguished (Merrikh and Lage, 2005a). Remarkably, the task of mapping geometrically the pore nuance is very complex in natural porous media as outlined by Nield and Bejan (1998) since the distribution of the pore size and pore morphology is not uniform. Rather than modeling complex geometries, it is possible to consider the morphology of the solid constituent within the cavity a as periodical array of solid, square and heat conductive blocks (House *et al*., 1990; Merrikh and Mohamad, 2001; Lee and Ha, 2006; Merrikh and Lage, 2005b; Junqueira *et al*., 2013; Lage *et al*., 2016).

Numerical studies of the lid-driven cavities filled with a heat conductive solid block, subjected to a gravitational stable condition were developed by Poletto *et al*. (2015) and Poletto *et al*. (2016). For the convection-dominant situation, the block introduction is able to increase the heat transfer up to a limiting block size. For larger blocks, the flow hindrance is so severe that the Nu_{av} drops. For the conduction-dominant situation, in which the fluid stagnation near the cavity base occurs, the presence of the solid obstacle usually decreases the heat transfer. Indeed, the presence of the obstacle restrains the flow to a very small channel between its upper surface and the top lid, extending the stagnated fluid region. However, for a large block, the flow is not able to accommodate on such tiny channel and, therefore, envelops all the obstruction re-establishing the convection process. Although the aspects of a single block introduction have been investigated regarding the variation of the flow parameters and the block dimension, an analysis focusing on the variation of the number of blocks has yet to be presented.

In the present work, the influence of the number of blocks, N, on the heat transfer process in a square cavity subjected to a gravitational stable condition is numerically analyzed. The fluid volumetric fraction and the Prandtl number are kept constant, φ =0.36 and Pr=1. It is considered cavities filled with N = 1, 4, 9, 16, 25, 36, 49 and 64 blocks for Re=(1000, 2500) and Gr=(10^5, 10^6, 10^7).

2. PROBLEM FORMULATION

The square cavity with dimension L, shown in Figure 1, has adiabatic vertical side walls while the top and the bottom surfaces are at temperature T_h and T_c, respectively, with $T_h > T_c$. The gravity acts aligned with the positive temperature gradient that implies a gravitational stable condition in which natural convection does not occur spontaneously. The flow is promoted by prescribing a horizontal and constant velocity U_H on the cavity lid. The velocity components u and v are defined on the directions x and y, respectively.

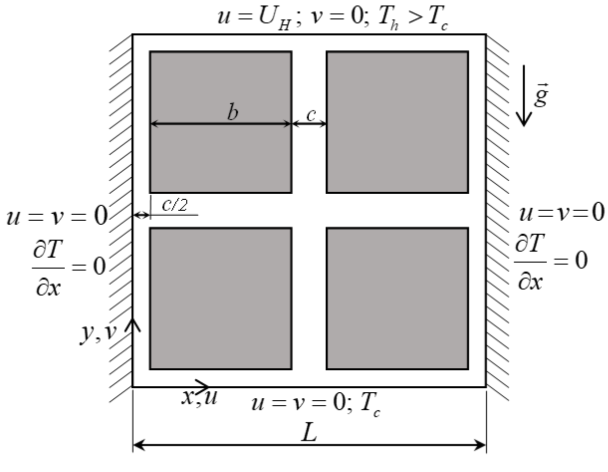

Figure 1. Problem geometry and boundary conditions.

The porous medium is modelled on the pore scale (heterogeneous approach) in which the solid matrix is devised as an array of identical, solid, disconnected, heat conductive blocks of dimension b, uniformly distributed across the cavity. The distance between the surfaces of two adjacent blocks is c while between the cavity wall and the first row of obstacles is $c/2$, as depicted in Figure 1. Porosity, φ, is defined as a ratio of the pore volume and the porous medium total volume (Nield and Bejan, 2006), being possible to write φ as a function of blocks properties $\varphi=1-N(b/L)^2$ (Merrikh and Lage, 2005b).

As the pore resolution model allows the distinguishing the fluid and solid constituent's interface, the balance equations can be applied separately in each domain and coupling boundary conditions can be prescribed across the interface.

Notably, the solid and fluid domains are assumed with constant properties. The buoyancy effect is, alternatively, modeled using the Boussinesq-Oberbeck approximation (Bejan, 2013), represented in Equation (1). The subscript c denotes that the value of the specific mass ρ [kg/m^3] at temperature T_c. The isobaric volume expansion coefficient is β [1/K].

$$\rho_c - \rho \approx \rho_c \beta \left(T - T_c \right) \tag{1}$$

The equations are cast in dimensionless form for convenience using the variables as follow:

$$(X,Y) = \frac{(x,y)}{L};\ (U,V) = \frac{(u,v)}{U_H};\ \theta = \frac{T - T_c}{T_h - T_c};\ P = \frac{p}{\rho U_H^2} \tag{2}$$

The dimensionless variables are then defined as the velocity components U and Y, along the space vector X and Y, respectively. The non-dimensional temperature θ and the pressure P are also considered. The set of Equations (3 – 6) for the fluid domain is presented in terms of the mass, X-momentum, Y-momentum and energy balance equations. The flow of a Newtonian like fluid is hypothetically laminar, two-dimensional with the effects of viscous dissipation and thermal radiation not considered.

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \tag{3}$$

$$U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{1}{Re}\left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right) \tag{4}$$

$$U\frac{\partial V}{\partial X} + V\frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{1}{Re}\left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right) + Ri\theta \tag{5}$$

$$U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y} = \frac{1}{Re\,Pr}\left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} \right) \tag{6}$$

In Equations (4) - (6) the dimensionless groups of Reynolds number, Re, Richardson number, Ri, Grashof number, Gr, and Prandtl number, Pr, respectively, are defined as follow:

$$Re = \frac{U_H L}{\upsilon};\ Ri = \frac{Gr}{Re^2};\ Gr = \frac{g\beta(T_h - T_c)L^3}{\upsilon^2};\ Pr = \frac{\upsilon}{\alpha} \tag{7}$$

The fluid kinematic viscosity is υ [m^2/s] and α [m^2/s] is the thermal diffusivity. The Richardson number Ri is a relative measure of the buoyancy effects over the inertial ones.

By assuming constant properties and steady state regime one can write the solid domain energy balance equation as the two-dimensional heat conduction equation:

$$\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} = 0 \tag{8}$$

Furthermore, the boundary conditions written in dimensionless form results in the following set of equations:

$$X=0 \text{ and } X=1:\ U=V=0;\quad \frac{\partial \theta}{\partial X} = 0 \tag{9}$$

$$Y=0: \ U=V=0; \quad \theta=0 \tag{10}$$

$$Y=1: \quad U=1; \quad V=0; \quad \theta=1 \tag{11}$$

The coupling between the solid and the fluid domains (Merrikh and Lage, 2005b) is accomplished by satisfying the equality between the temperature and the heat flux computed for each domain on the block surfaces, as demonstrated by Equation (12) where the subscripts *f* and *s* stand for the fluid and solid constituents, respectively. The velocity of the fluid on the solid surface is also set to be zero.

$$\theta_f=\theta_s; \ \left.\frac{\partial\theta}{\partial\eta}\right|_f=K\left.\frac{\partial\theta}{\partial\eta}\right|_s; \ U=V=0 \tag{12}$$

The unit vector defined in the normal direction of the surfaces of the blocks is η. The parameter $K= k_s/k_f$ is the solid-fluid thermal conductivity ratio.

In the present work, the results are presented in terms of the stream function Ψ, defined in Equation (13) according to (Kimura and Bejan, 1983) for a discrete domain comprised of control volumes.

$$\Psi=\Psi_{i,j}=\Psi_{i,j-1}+\int_0^1 UdY=\Psi_{i-1,j}+\int_0^1 -VdX \tag{13}$$

The value of $\Psi_{i,j}$ is computed at the center of the control volume *i,j*. Thus, $\Psi_{i-1,j}$ e $\Psi_{i,j-1}$ are the values of the stream function to the left and underneath of the control volume center, *ij*, respectively.

The isotherms are employed to interpret the temperature field. The heat transfer is quantified by the local (Nu) and the surface-averaged (Nu_{av}) Nusselt numbers, both computed on the top sliding-lid:

$$Nu=-\left.\frac{\partial\theta}{\partial Y}\right|_{Y=1}; \ Nu_{av}=\int_0^1 -\left.\frac{\partial\theta}{\partial Y}\right|_{Y=1} dX \tag{14}$$

A numerical solution is pursuit for Equations (3) - (6) and Equation (8) via the Finite Volume Method (Patankar, 1980). The SIMPLE pressure-velocity scheme (Patankar and Spalding 1972) is employed; the PRESTO is applied to interpolate the pressure in a staggered-grid and the QUICK scheme models the advection terms (Leonard, 1979).

3. RESULTS

A summary of the problem variables is presented in Table 1. A unitary Prandtl number, Pr=1, the solid-fluid thermal conductivity ratio, K=1, and the volume fraction φ=0.36 are considered. Noteworthy, the Reynolds and the Grashof numbers regard typical values for the laminar flow regime (Ghia *et al*., 1982). The number of blocks are allowed to vary between the perfect square numbers from 0 (clear cavity) to N=64.

Table 1. Summary of problem parameters.

Re	1000, 2500
Gr	10^5, 10^6, 10^7
Pr	1
K	1
N	0, 1, 4, 9, 16, 25, 36, 49, 64
φ	0.36

Verification results are presented in Table 2, where results for the convection in a lid-driven cavity subjected to a gravitational stable condition filled with a single solid and the heat conductive block are provided by Poletto *et al.* (2015). In addition, results of the natural convection in a horizontally heated square cavity filled with N=64 blocks (De Lai *et al.*, 2011) are also displayed. The accuracy of present work results evaluated through the error function in Equation (15) gives credence to the results obtained here.

$$Error=1-\frac{Nu_{av}}{\left.Nu_{av}\right|_{\text{ref}}} \tag{15}$$

Table 2. Verification results: Nu_{av} for φ=0.36.

N	*Pr*	*Re*	*Gr*	Poletto *et al.* (2015)	De Lai *et al.* (2011)	Present	*Error* [%]
1	1	1000	10^5	9.482		9.499	0.179
1			10^6	1.194		1.193	0.084
1			10^7	1.144		1.1391	0.428
64	0.71	-	10^7	-	2.426	2.391	1.443

Figure 2 brings the grid accuracy test in terms of the local Nusselt number, *Nu*, over the sliding-lid. As discussed by Poletto *et al*. (2015), the most difficult case simulated is observed for *Ri*=1, where a strong competitive effect between the fluid inertia and buoyancy is verified. Furthermore, the higher the *Re*, the thinner the boundary layers (Bejan, 2013), requiring an elevation on the number of control volumes right next to the wall to assure the accurate calculation of the velocity and thermal gradient. Therefore, the critical case is defined for *Re*=2500, *Gr*=6.25x10^6 and *N*=64 blocks. Still regarding Figure 2, only the interval 0<*X*<0.3 is displayed, since for *X*>0.3 the *Nu* tends to unity. Notice that the curve of the uniform mesh of 72x72 control volumes does not converge with the others meshes.

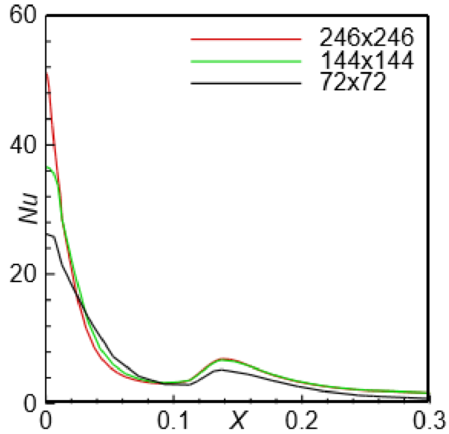

Figure 2. Comparison of Nu results calculated from different meshes.

Although the *Nu* profiles for the meshes 144x144 and 246x246 differs significantly in the region close to the edge of the lid, the values of Nu_{av}, respectively, 1.669 and 1.676, show an Error of less than 0.4%. As the main analysis of the present work focus on the Nu_{av}, the 144x144 uniform mesh is chosen for the simulations.

The results are portrayed in Figure 3 for a convection-dominant regime with *Gr*=10^5. For the clear cavity, *N*=0, increasing *Re*=1000 to 2500 causes an intensification in the fluid circulation Ψ and, consequently, in the heat transferred, which is reflected in the Nu_{av} increase. In addition, the streamlines for both *Re* spread across the enclosure indicating that the flow reaches the base, where the isotherms are concentrated.

For the *Re* considered, the *Gr* is increased with the results shown in Figure 4. The hindrance due to the buoyancy intensification is so severe that the fluid near the base stagnates. The flow is restricted to the upper half of the cavity, as can be observed by the concentration of streamlines. The isotherms in the stagnated fluid are vertically stratified indicating a conduction regime that is corroborated by the unitary Nu_{av}. Curiously, Ψ for *Re*=2500 and *Gr*=10^7 is higher than *Re*=1000 and *Gr*=10^6 even if the opposite behavior for the Nu_{av} is observed.

The variation of the number of blocks, *N*, affects differently each one of the flow regimes. For the convection-dominant situation, incrementing *N* leads to a decrease in Ψ as the flow resistance is pronounced. Interestingly, Ψ for *N*=4 is higher than *N*=1 because the fluid finds a less resistant path to flow. For *N*>4, the Ψ keeps dropping.

Considering the convection-dominant situation, the introduction of a single block, *N*=1, in Figure 3 causes an intensification in Nu_{av} compared with the clear fluid case. When *N* is increased, such effect vanishes as the Nu_{av} keeps decreasing eventually reaching a unitary value.

On the results for the predominance of conduction, shown in Figure 4, the increment of *N* also reduces Ψ, and, consequently, Nu_{av}. Notably, the *N*=4 configuration is not so restrictive to the flow and shows a Ψ higher than that for *N*=1. However, due to the restrictive action of buoyancy, the increase in Nu_{av} is mild. The resistance imposed over the flow seems to increase with *N*, causing a drop in Ψ, making Nu_{av} tend to unity.

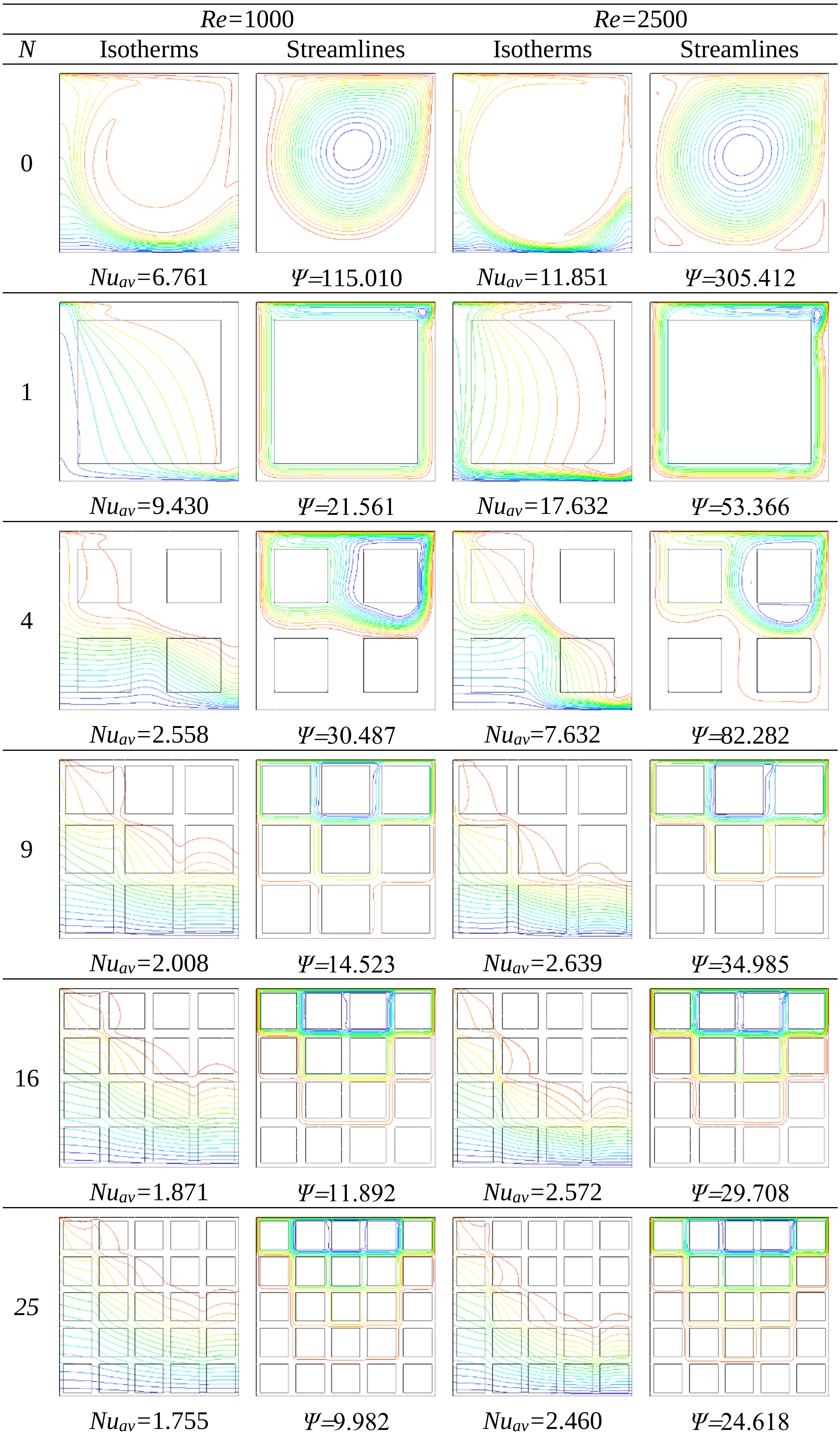

Figure 3. Results for convection-dominant heat transfer for $Gr=10^5$.

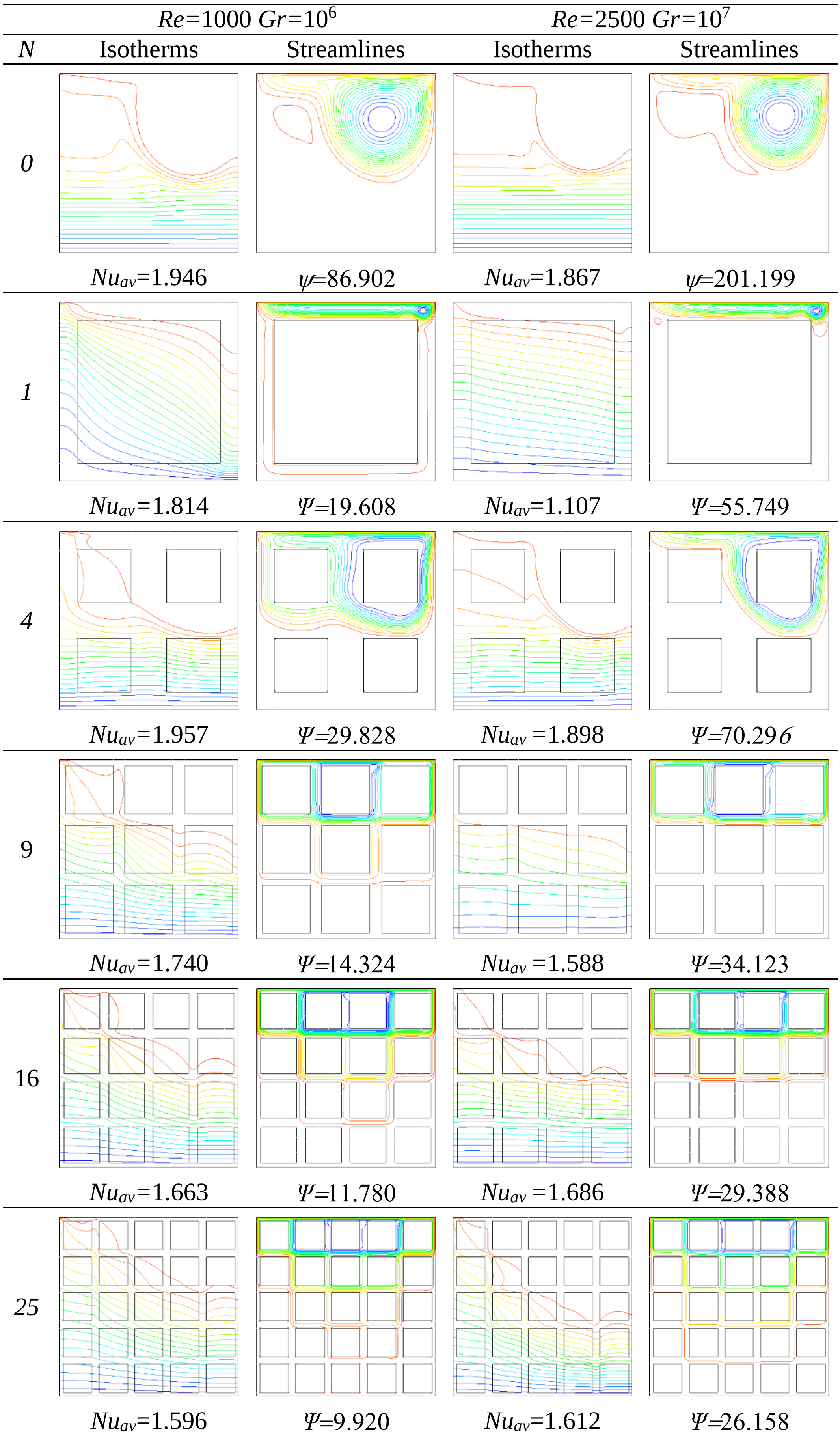

Figure 4. Results for conduction-dominant heat transfer.

The influence of varying *N* over Nu_{av} is depicted in Figure 5 (a), as for *N*>25 the Nu_{av} follows the tendency to reach a unitary value. Analogously Figure 5 (b) shows that as *N* is incremented, Ψ continuously drops.

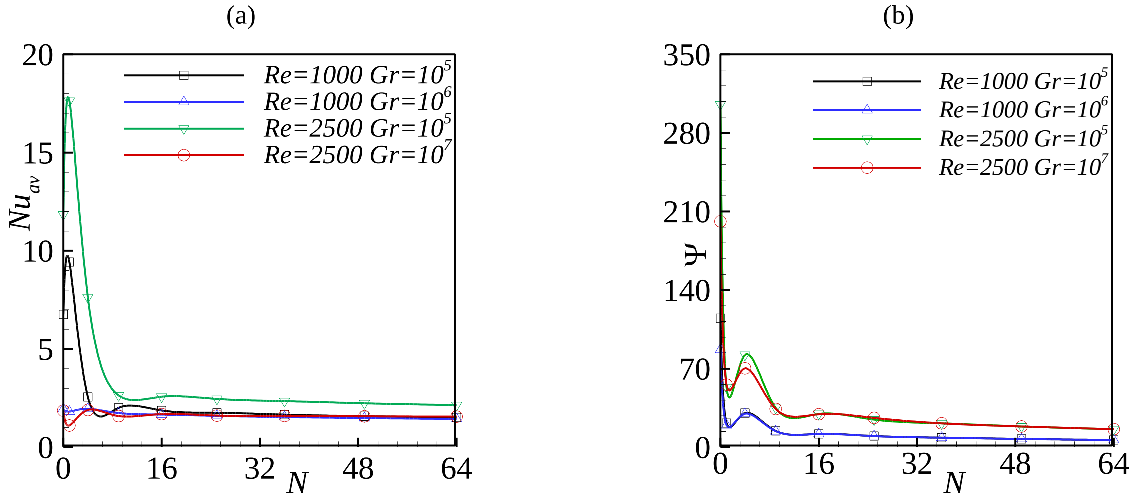

Figure 5. Results for the number of blocks influence over the convection. (a) Nu_{av}. (b) Ψ.

4. CONCLUSIONS

In the present work, the influence of the placement of solid blocks on the convection process inside a square lid-driven cavity subjected to a stable gravitational condition is numerically simulated. The heat transfer is characterized by two dominant patterns: convective and conductive. In the latter, the buoyancy effect suppresses the momentum and the fluid remains stagnated in the vicinity of the base of the enclosure, while in the first, the flow spreads throughout the domain. The introduction of blocks imposes more resistance to the flow in both standards, causing the fluid circulation to decrease. Usually, the Nu_{av} follows the decreasing trend with the increment in the number of blocks. However, the introduction of a single block in a convection-dominant configuration and *N*=4 in a pure conductive case influences the temperature field in such a way the temperature gradients are intensified and consequently, Nu_{av} increases.

5. ACKNOWLEDGEMENTS

The authors would like to acknowledge the support of IRF/CENPES/PETROBRAS, the PRH-ANP/MCT program (PRH10-UTFPR) and the National Council for Scientific and Technological Development (CNPq).

6. REFERENCES

Bejan, A., 2013. *Convection heat transfer*. John Wiley & Sons Inc, Hoboken, NJ, U.S.A, 4th Ed.

Cheng, T. S. 2010, "Effect of temperature gradient orientation on the characteristics of mixed convection flow in a lid-driven square cavity". Comput. Fluids, vol. 29, pp. 965-978.

Cheng, T. S. 2011, "Characteristics of mixed convection heat transfer in a lid-driven square cavity with various Richardson and Prandtl numbers". International Journal of Thermal Sciences, vol. 50, pp. 197-205

De Lai, Fernando C.; Franco, Admilson T.; Junqueira, Silvio L. M., 2011. "The effects of solid termal conductivity and volume fraction in the natural convection inside a heterogeneous enclosure". In: *8th Thermal Engineering Joint Conference – AJTEC2011, 2011*; Honolulu, Hawaii.

Ghia, U., Ghia, K. N., Shin, C. T., 1982. "High-Re solution for incompressible flow using the Navier-Stokes equations and Multigrid Method". *Journal of Computational Physics*, Vol. 48, pp 387 – 411.

House, J. M.; Beckermann, C.; Smith, T. F., 1990. "Effect of a centered conducting body on natural convection heat transfer in an enclosure". *Numerical Heat Transfer*, A, Vol. 18, pp. 213-225.

Ismael, M. A.; Pop, I.; Chamka, A. J., 2014. "Mixed convection in a lid-driven square cavity with partial slip". *Int.J.Therm.Sci*, Vol. 82, pp. 47-61.

Iwatsu, R.; Hyun, J. M.; Kuwahara, K., 1992. "Convection in a differentially-heated square cavity with a torsionally oscillating lid". *International Journal of Heat and Mass Transfer*, Vol. 35, pp. 1069 – 1076.

Iwatsu, R.; Hyun, J. M.; Kuwahara, K.,1993. "Mixed convection in a driven cavity with a stable vertical temperature gradient". *International Journal of Heat and Mass Transfer*, Vol. 36, pp. 1601 – 1608.

Iwatsu, R.; Hyun, J. M.; Kuwahara, K.,1993. "Three-dimensional driven-cavity flows with a vertical temperature gradient". *International Journal of Heat and Mass Transfer*, Vol. 36, pp. 1601 – 1608.

Junqueira, Silvio L. M.; De Lai, Fernando C.; Franco, Admilson T.; Lage, José L., 2013. "Numerical investigation of natural convection in heterogeneous rectangular enclosures". *Heat Transfer Engineering*, Vol. 34, pp; 1-10, 2013.

Kimura, S. and Bejan, A., 1983, "The 'heatline' visualization of convective heat transfer". *ASME Journal Heat Transfer*, Vol.105, pp. 916-919.

Lage, J. L., Junqueira, S. L. M., De Lai, F. C.; Franco, A. T., 2016. "Aspect ratio effect on the prediction of boundary layer interference in steady natural convection inside heterogeneous enclosures". *International Journal of Heat and Mass Transfer*, Vol. 92, pp. 940–947.

Lee, J. R.; Há M. Y., 2006. "A numerical study of natural convection in a horizontal enclosure with a conducting body". *International Journal of Heat and Mass Transfer*, Vol. 48, pp. 3308-3318, 2006.

Leonard, B. P., 1979. "A stable and accurate convective modelling procedure based on quadratic upstream interpolation". *Computer Methods in Applied Mechanics and Engineering*, Vol. 19, n. 1, pp. 59–98.

Merrikh, A. A.; Mohamad, A. A., 2001. "Blockage effects in natural convection in differentially heated enclosuers". *Enhanced haet transfer*, Vol. 9, pp. 55 – 72.

Merrikh, A. A.; Lage, J. L., 2005a. "From continuum to porous-continuum: *The visual resolutoin impact on modeling natural convection in heterogeneous porous media*. In: Transport Phenomena in Porous Media III. Edited by Ingham, D B.; Pop, I, Elsevier, Oxford, UK.

Merrikh, A. A; Lage, J. L., 2005b. "Natural convection in an enclosure with disconnected and conducting solid blocks". *International Journal of Heat and Mass Transfer*, Vol. 48, pp. 1361 – 1372.

Nield. D. A.; Bejan, A., 2006. *Convection in porous media*. Springer-Verlag, New York, U.S.A., 2nd edition,.

Patankar, S.V., 1980. *Numerical Heat Transfer and Fluid Flow*, Hemisphere Publishing, New York.

Patankar, S. V.; Spalding, D. B., 1972. "A calculation procedura for heat, mass and momentum transfer in three-dimensional parabolic flows". *International Journal of Heat and Mass Transfer*, Vol. 15, pp. 1787-1806.

Poletto,V.G.; De LaiI, F. C.; Franco, A .T.; Junqueira, S. L. M., 2015. *Convection analysis in a lid-driven square cavity partially filled with a solid heat-conducting block: blockage effects*; In 23rd ABCM International Congress of Mechanical Engineering, RJ.

Poletto,V. G.; De Lai, F. C.; Franco, A. T.; Junqueira, S. L. M., 2016 *Numerical simulation of the convection in a non-homogeneous lid-driven square cavity subjected to a gravitational stable condition*. In: 2016ASME HT/FE/ICNMM, Washington-D.C.

Ramakrishna, D.; Basak, T.; Roy, S. 2012. *A complete heatline analysis on mixed convection within a square cavity: effects of thermal boundary conditions via thermal aspect ratio*. Int. J. Therm. Sci., Vol. 57, pp. 98-111.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.