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CONVECTION ANALYSIS IN A LID-DRIVEN SQUARE CAVITY PARTIALLY FILLED WITH A SOLID HEAT-CONDUCTING BLOCK: BLOCKAGE EFFECTS

Vinícius Gustavo Poletto
Fernando César De Lai, MSc
Admilson Teixeira Franco, Dr
Silvio Luiz de Mello Junqueira, Dr
Federal Technological University of Parana, Curitiba, Pr.
vinsptt19@gmail.com
fernandodelai@gmail.com
admilson@utfpr.edu.br
silvio@utfpr.edu.br

Abstract. *In this work, convection heat transfer inside a lid-driven non-homogeneous porous square cavity is numerically investigated. The porous medium solid constituent is a heat conductive square block centered inside the enclosure. Non equilibrium conditions are imposed by heating the top-sliding surface while the side walls are kept insulated. Due to the presence of a gravitational force and the lid momentum supply to the fluid, forced and natural convection compete. The investigation of how the heat transfer is affected by the presence of a square obstacle placed inside the cavity is the objective of the present work. Conservation equations are solved separately for the fluid and for the solid via the finite volume method. QUICK interpolation scheme is set for numerical treatment of the advection terms and SIMPLE algorithm is applied for pressure-velocity coupling. The parameters range simulated cover values for the Reynolds number up to 1000 and for the Grashof number up to 10^7. Results show that increasing Gr or decreasing Re lead to fluid stagnation, yielding a diffusive heat transfer regime. In this case, enlarging the block dimension hinders the heat transfer as it alters the fluid path adjacent to the moving lid. Alternatively, increasing the Re or decreasing Gr intensifies the fluid circulation enhancing the heat transfer. The introduction of a solid block in this case might benefit heat transfer up to a limiting blockage size.*

Keywords: *mixed convection, lid-driven cavity flow, heterogeneous porous media, natural convection, forced convection.*

1 INTRODUCTION

Convection in porous media is essential in a series of industrial activities and occurs in a number of day-to-day situations, as pointed out by Nield and Bejan (1993). Bejan (2014) lists some engineering applications in the construction, electronics, food and energy industry.

The influence of the convection nature (free or forced) over heat transfer relies on how thermal disequilibrium conditions are applied in accordance with body forces and how flow is induced. In this work, a lid-driven heterogeneous porous cavity is subject to a temperature gradient in a stable gravitational condition. This problem emerges as a collection of previous studies about natural convection in heterogeneous enclosures conducted by House *et al*. (1990) and Merrikh and Lage (2005); mixed convection in lid-driven cavities studied by Iwatsu *et al*. (1993) and mixed convection in lid-driven heterogeneous cavities performed by Islam *et al*. (2013).

One of the first numerical studies about natural convection in heterogeneous enclosure was performed by House *et al* (1990) in a square horizontally-heated enclosure filled with a solid heat-conducting block. The author's goal was to understand the effects of variation of Rayleigh number, blockage size and the solid to fluid thermal conductivity ratio over the convection process. Subsequently, Lee *et al*. (2005) simulated the natural convection in a square enclosure heated from the base that was partially filled with a solid heat-conducting block. In the sequence, Merrikh and Lage (2005) extended the scope of the analysis to cavities filled with a collection of solid-heat-conduction blocks. A correlation to predict the blockage interference in the boundary layer was proposed. De Lai *et al*. (2011) carried out similar simulations concerning the variation of porosity and the solid to fluid thermal conductivity ratio over the convection heat transfer. In general, it was concluded that an increase in porosity enhances heat transfer. Junqueira *et al*. (2013) studied cavities with aspects ratios different than unity. The influence of the aspect ratio over the heat transfer relies on the flow parameters and the number of block placed inside the cavity. The Berkovsky-Polevikov correlation for Nu_{av} prediction in Rayleigh-Bérnard convection was adapted by Hongtao *et al*. (2013) for cavities partially filled with solid blocks.

Comprehending the role of the driven forces associated to the forced and free convection is fundamental to interpret the heat transport along the cavity here investigated. Although the main motivation of Ghia *et al*. (1982) was to accomplish numerical validation, their results for an isothermal flow in a lid-driven cavity are still widely employed in numerical methods for benchmarking. In the sequence, Koseff and Street (1984) built an experiment to reproduce the geometry of the enclosure and evaluated the validity of flow assumptions such as two-dimensional and steady flow. Mixed convection in a lid-driven cavity was numerically analyzed by Iwatsu *et al*. (1993). His results considered that the sliding-lid temperature was higher than the one at the bottom and, therefore the fluid was subjected to a gravitationally stable condition. In this case, the lid movement was imperative to establish the fluid flow, as if the lid speed were zero there would be no flow. On one hand, an increase in the lid velocity intensifies the flow circulation and, therefore, enhances heat transfer. On the other hand, an increase in buoyant forces leads to fluid stagnation yielding a diffusive heat transfer regime. Further contributions concerning the understanding of mixed convection in lid-driven cavity were provided by Morzynski and Popiel (1988), Moallemi and Jang (1992), Iwatsu *et al*. (1992), Franco and Ganzarolli (1996), Prasad and Koseff(1996), Mansour and Viskanta and Cheng (2011).

Furthermore, Islam et al. (2013) performed numerical simulations to study the effect of placing a solid isothermal block in a lid-driven cavity. As the gravitational condition was unstable, an increase in the Richardson number yielded in heat transfer enhancement. Additionally, the enlargement of the block resulted in the heat transfer beneficiation. Oztop *et al*. (2009) and Khanafer and Aithal (2013) performed similar simulations for sliding-lid cavities filled with cylinders reaching similar conclusions.

In this work, convection inside a lid-driven heterogeneous cavity is numerically simulated in order to investigate the influence of the introduction of a heat-conducting solid block over the heat transfer process.

2 PROBLEM DESCRIPTION

A schematic representation of the porous cavity with length *L* is shown in Figure 1. The origin of the cartesian coordinate system is placed at the lower left corner with *u* and *v* being the velocities in *x* and *y* directions, respectively. The cavity upper horizontal surface slides with constant speed U_0 in *x*-axis positive direction. The cavity bottom surface temperature is T_C. As reference, the sliding-lid is kept at temperature T_H in such way that $\Delta T=T_H-T_C=1$. The vertical walls are adiabatic.

The porous media is represented in porous scale (continuum model) with two visually distinguished constituents: the fluid and the solid. Such approach, also referred as heterogeneous, requires the fluid-solid interface to be geometrically mapped to proper assigning properties for each constituent. Usually, the interface morphology in real porous media is very complex to map mathematically and some sort of geometric simplification is required. In this problem, the solid constituent is idealized as a centered, squared, heat-conducting, solid block of dimension *b*. It is important to consider the relation of *b* with the cavity porosity. Nield and Bejan (1993) define porosity as the ratio between the pore and total porous media volume. Thus, the porosity of the cavity is $\varphi = (1\text{-}b^2)/L^2$. As if *b* tends to zero, porosity tend to one meaning that the flow occurs in a cavity free of blockage interference.

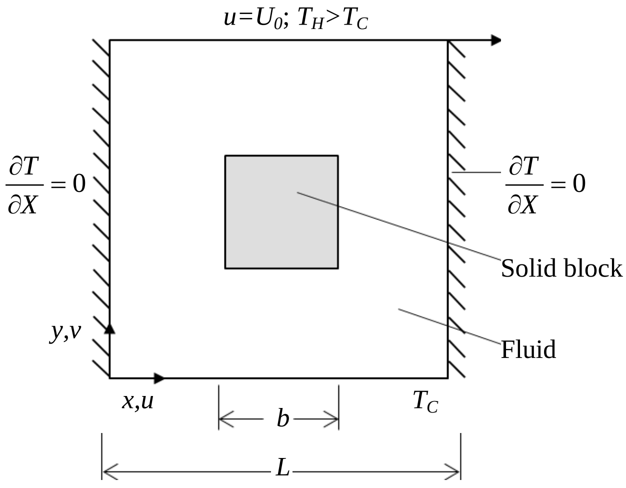

Figure 1. Boundary conditions and domain representation.

3 CONTINUUM MODEL EQUATIONS

As it was outlined, applying the continuum model implies the mapping of the solid-fluid interfaces and assigning individual properties to the fluid and solid constituents. Thus, each constituent must have its own set of equations. The lid-driven flow inside the enclosure is considered two-dimensional, steady and laminar. Properties of both solid and fluid (considered Newtonian) domains are constant. The exception is for the fluid specific mass ρ_f that varies linearly with temperature in the buoyancy term according to the Boussinesq-Oberbeck approximation (Bejan, 2014).

$$\rho_C - \rho = \rho_C \beta (T - T_C) \tag{1}$$

where ρ_C and ρ are the fluid specific mass at temperatures T_C and T, respectively. The fluid volume expansion coefficient at constant pressure is denoted by β. Viscous heat dissipation and thermal radiation are not taken into account for heat transfer calculations.

Considering the length L, the lid-sliding velocity U_0 and the temperature difference $\Delta T=T_H-T_C$ as scales, problem non-dimensional variables are set as: $(X,Y)=(x,y)/L$, $(U,V)=(u,v)/U_0$, $P=p/(\rho_f U_0^2)$ and $\theta=T-T_C/(T_H-T_C)$. Therefore, Equations (2) - (5) are the fluid conservation equations namely, the mass balance, x-direction momentum conservation, y-direction momentum conservation and energy conservation. Equation (6) is the solid energy conservation equation.

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \tag{2}$$

$$U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{1}{Re}\left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right) \tag{3}$$

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

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

$$\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} = 0 \tag{6}$$

where $Re=L.U_0/\upsilon$, $Ri=Gr/Re$, $Gr=g.\beta.(T_H-T_C).L^3/\upsilon^2$ and $Pr=\upsilon/\alpha$, are, respectively, the Reynolds, Richardson, Grashof and Prandtl non-dimensional groups. The Richardson number is responsible to account the prevalence of either the forced or the free convection.

The following boundary conditions are applied.

$$\text{For } X=0 \text{ and } X=1\text{: } U = V = 0; \frac{\partial \theta}{\partial X} = 0 \tag{7}$$

$$\text{For } Y = 0 : U = V = 0; \theta = 0 \tag{8}$$

$$\text{For } Y = 1 : U = 1; V = 0; \theta = 1 \tag{9}$$

The fluid and solid conservation equations are coupled by boundary conditions applied at the block surfaces. One important condition to be satisfied is the continuity of the temperature field inside the enclosure. It is expected to the block to interfere in the temperature field. However, the fluid and solid temperatures at the blockage surfaces must be equal. The heat flux calculated at the block surfaces must also be equal in the solid and the fluid domain. The subscripts f and s indicate the fluid and the solid constituent, respectively.

$$\theta_f = \theta_s ; \left.\frac{\partial \theta}{\partial \eta}\right|_f = K \left.\frac{\partial \theta}{\partial \eta}\right|_s \tag{10}$$

where η is a normal vector defined at every point of the block surfaces. The parameter K stands for the solid-fluid thermal conductivity ratio, $K=k_s/k_f$, where k_s and k_f are the solid and fluid thermal conductivity, respectively. In this paper, the thermal and conductivity ratio is set as unity, meaning that the heat diffusion in the solid and the fluid domains are equal.

Results are exposed by the isotherms and stream functions plots. According to Kimura and Bejan (1983), the stream lines can be physically unified to satisfy the conservation of mass equation:

$$\Psi = \Psi_{i,j} = \Psi_{i,j-1} + \int_0^1 U dY = \psi_{i-1,j} + \int_0^1 -V dx \tag{11}$$

where $\Psi_{i,j}$ is the stream function value at the center of the *i,j* control volume. Thus, $\Psi_{i,j-1}$ and $\Psi_{i-1,j}$ are regarded, respectively, to the underneath and the left control volumes. Stream functions are Ψ=0 at walls. In this paper, it is shown the modulus of the minimum stream function value $|\Psi_{min}|$. However, for convenience, the stream function intensity is simply referred as Ψ. Furthermore, the stream function intensity is also denoted as flow circulation intensity.

Heat transfer is analyzed by means of the hot surface-averaged Nusselt number defined as follows:

$$Nu_{av} = \int_0^1 -\left.\frac{\partial \theta}{\partial Y}\right|_{Y=1} dX \tag{12}$$

The surface-averaged Nusselt number is also defined as $Nu_{av}=L.h_{av}/k_f$ with h_{av} being the surface-averaged convection heat transfer coefficient that leads to the heat flux calculation as $q''=h_{av}.(T_H-T_C)$. The Nu_{av} calculated at the hot or cold surface must be the same.

4 RESULTS AND DISCUSSIONS

The equations (2) to (6) are solved numerically via finite volume method with a uniform mesh. The SIMPLE algorithm performs the pressure-velocity coupling, as in Patankar (1981). The advection terms are treated numerically by the QUICK interpolation scheme proposed by Leonard (1979) that accounts for changes in the flow direction.

The variables summarized in Table 1 are arbitrated to assure the occurrence of laminar flow. Ghia *et al*. (1982) show that for a sliding-lid cavity free of blockage interference the transition to turbulence starts to take place at Reynolds number of the order of 2300. The non-dimensional block length is defined as *B=b/L* with *B=0* representing the cavity free of blockage interference.

Table 1. Parameters for the numerical tests.

Re	1000
Gr	10^3; 10^6
B	0; 0.3; 0.6; 0.9
K	1
Pr	1

Table 2 show the comparison of Nu_{av} reproduced by this work with results from the literature for the numerical simulation of natural convection in heterogeneous enclosures.

Table 2. Verification of Nu_{av} with literature solutions for natural convection.

B	*K*	*Ra*	Lee *et al.* (2005)	House *et al.* (1990)	Present
0.3	1	10^5	3.880	-	3.845
0.3	1	10^6	6.290	-	6.305
0.5	5	10^5	-	4.601	4.625
0.9	5	10^6	-	3.812	3.811

Table 3 exposes the comparison of Nu_{av} reproduced by the present work with results from the literature for mixed convection in lid-driven cavities.

Table 3. Verification of Nu_{av} with literature solutions for mixed convection.

B	*Gr*	*Re*	Iwatsu *et al.* (1993)	Islam *et al.* (2013)	Present
-	10^2	400	3.620	-	4.082
	10^4		3.620	-	3.843
	10^6		1.220	-	1.181
0.125	10^4	100	-	7.729	7.586
0.250			-	5.600	5.539
0.500			-	5.326	5.336
0.750			-	8.872	8.782

Results of the grid accuracy tests are exposed in Table 4 as the Nu_{av} and the percentage relative error, *EP*.

$$EP = \left| \frac{\Theta_{ref} - \Theta_{cal}}{\Theta_{ref}} \right| \tag{13}$$

Where Θ_{ref} and Θ_{cal} stand for the reference and calculated values of the Nu_{av}, respectively. Initially, combinations of end-of-range parameters are set for the simulations. The 50x50 volume uniform mesh is able to produce results with accuracy around 1% for the majority of cases. However, for Gr=10^6 and Re=1000 the relative error increases significantly. The critical cases occur when the buoyant and inertia forces have the same order of magnitude, resulting in Ri~1. For those cases, additional simulations point out that a uniform 200x200 volume mesh is capable of producing results within 1% accuracy.

Table 4. Grid accuracy tests for a sliding-lid cavity filled with a B=0,9 centered block

Re	*Gr*	*Ri*	Nu_{av} 50x50	Nu_{av} 100x100	*EP* Eq.(13)	Nu_{av} 200x200	*EP* Eq.(13)	Nu_{av} 300x300	*EP* Eq.(13)
100	10^3	0.1	1.162	1.162	0.039	1.162	0.001	-	-
	10^7	1000.0	1.001	1.001	0.020	1.001	0.011	-	-
1000	10^3	0.001	6.075	6.127	0.840	6.170	0.700	-	-
	10^6	1.0	3.765	4.020	6.359	4.159	3.450	3.723	0.961
	10^7	10.0	1.077	1.096	1.737	1.108	1.067	-	-

The grid accuracy test indicates that a 50x50 volume uniform mesh can be used since $Ri \neq 1$. However, for the B=0.9 configuration a 100x100 mesh must be considered to assure the calculation of the velocity and temperature fields in the narrow gap between the block surface and the cavity wall. If Ri=1 the mesh has to be refined to a 200x200 volume uniform mesh to get better results.

For Gr=10^3 and Re=1000, implies that Ri=0.001 indicating that the forced convection is predominant. The isotherms and the Nu_{av} for this case are shown in Figure 2 (a) and the streamlines and the values for Ψ are exposed in Figure 2 (b). For the B=0, the fluid circulation decreases from the wall towards the center of the enclosure where temperature is nearly homogeneous. The isotherms are concentrated on the horizontal walls. Velocity and temperature fields are similar to those presented by Iwatsu *et al*. (1993).

For B=0.3, the influence of the block over heat transfer is limited to a slightly increase in Nu_{av}. Although the heat transfer is beneficiated, the flow circulation decreases around 39% in comparison with the B=0 configuration.

The insertion of a B=0.6 block interferes in the temperature field, as heat is conducted horizontally across the block. In spite of the decrease in flow circulation, the influence of the solid body enhances heat transfer as the Nu_{av} reaches a peak. This effect is a consequence of the blockage pushing the flow closer to the walls.

The effects of increasing blockage size to B=0.9 are a quite different. The circulation intensity is reduced in approximately 76% in comparison with the B=0.6 configuration. The flow is hindered, justifying a decrease in Nu_{av}.

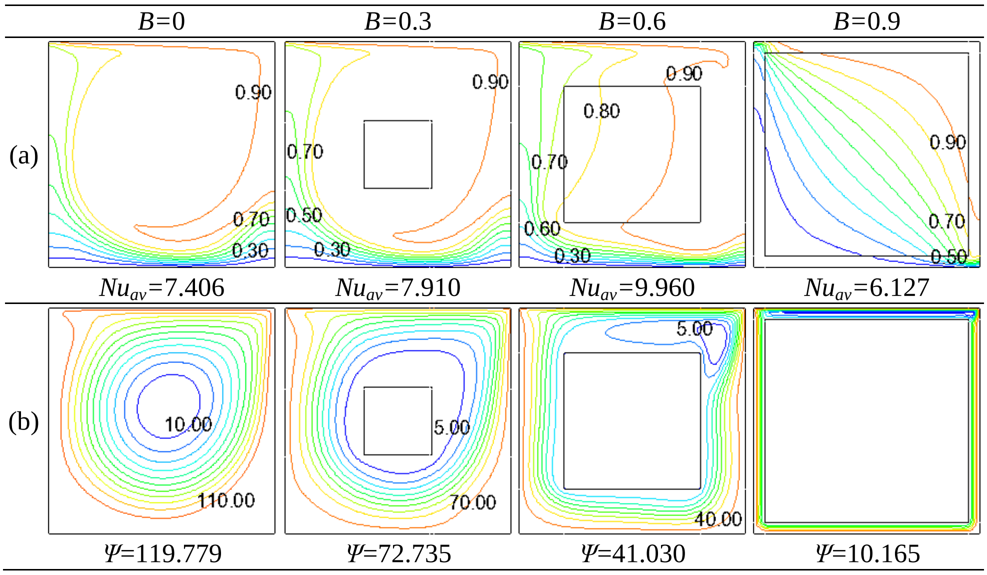

Figure 2. Results for the introduction of a solid heat-conducting block for $Gr=10^3$ and $Re=1000$ ($Ri=0.001$). (a) Isotherms and Nu_{av}. (b) Streamlines and Ψ.

An increase in the buoyancy forces intensity tends to decrease the flow circulation allowing a stagnant-prone situation, as it is explained by Iwatsu *et al.* (1993). Flow stagnation occurs in the $B=0$ configuration in the vicinity of the base for $Ri\sim1$. The volume of quiescent fluid grows upwardly as the buoyant forces are increased. For $Gr=10^6$ and $Re=1000$, $Ri=1$ and the flow stagnation is clearly depicted by the lack of streamlines in the lower hemisphere and the isotherms vertical stratification, as it is shown in Figure 3 (a) and (b). Conversely, the circulations are concentrated in the upper hemisphere of the enclosure.

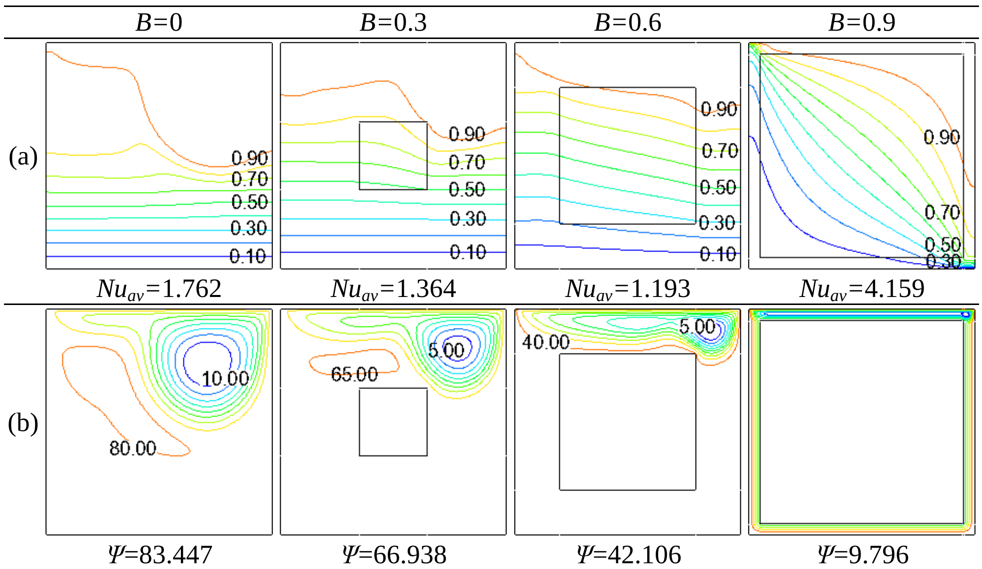

Figure 3. Results for the introduction of a solid heat-conducting block for $Gr=10^6$ and $Re=1000$ ($Ri=1.0$). (a) Isotherms and Nu_{av}. (b) Streamlines and Ψ.

The introduction of a $B=0.3$ solid block restrains flow circulation to the upper channel, reducing the fluid circulation and the values of Nu_{av}. The fluid by the sides and underneath the block remains quiescent and heat is transmitted solely by diffusion. Enlarging the block to $B=0.6$ provokes an accentuated reduction in Ψ causing the heat transfer to be even more mitigated. However, for the $B=0.9$ configuration, the fluid is not capable of recirculation bounded by the block upper surface and the sliding-lid. Alternatively, the fluid circulates around the block establishing a convection regime. In spite of the reduction in Ψ the Nu_{av} is increased in approximately four times in comparison with the $B=0.6$ configuration.

The effects of the solid block insertion over heat transfer are depicted in Figure 4. The fluid stagnation due to the action of buoyant forces is clearly depicted by the rough drop in the Nu_{av}. If there is quiescent fluid near the cavity base, heat is transferred by diffusion and the Nu_{av} tends to unity. The only cavity experiencing convection heat transfer is the *B=0.9* as its Nu_{av} is greater than unity. For this case, flow stagnation takes place only for $Gr=10^7$.

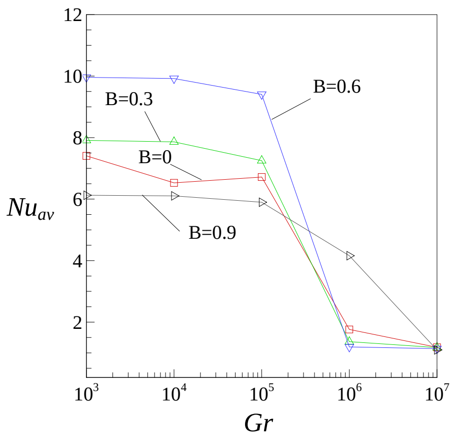

Figure 4. Influence of the blockage size over Nu_{av} over various *Gr* for *Re*=1000.

5 CONCLUSION

In this work a numerical analysis of the convection in a lid-driven square cavity partially filled with a solid heat-conducting block is carried out. It is summarized that the influence of the introduction of a solid block over the convection process depends on flow configuration. If forced convection is predominant, $Ri<1$, the introduction of a solid block might benefit heat transfer up to a limiting blockage size. Blocks larger than the limiting size reduces the Nu_{av} value because the flow circulation is severely decreased. For cavity configurations with $Ri \geq 1$, heat is transferred by diffusion. Introducing a solid block restricts the circulations reducing the values of Ψ and Nu_{av}. Enlarging the block might establish fluid circulation in the cavity, resulting in convection heat transfer and an increase in Nu_{av}.

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