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HEAT TRANSFER BY NATURAL CONVECTION IN 3D ENCLOSURES

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Abstract. *The heat transfer by transient natural convection in a fluid confined in enclosures is a problem related to different practical situations. The conditioning of products in the food industry, heating or cooling, are examples of application for this type of convection problem. The specific problem considered here concerns the transient heating of a cavity completely full 3D, initially at a temperature T_0. At one point, all the walls of the flask are heated to a temperature T_w greater than T_0. The heat spreads by natural convection in enclosures until its complete thermal equilibrium. This problem is analyzed here by numerical simulations and compared with experimental data. Different situations the initial Rayleigh number is observed for a given geometry of the cavity. The numerical simulations are performed using the program OpenFOAM through the Boussinesq hypothesis. A discussion of the topology, flow patterns and loss of symmetry within the cavity and the effects of natural convection are presented.*

Keywords: *Natural Convection, enclosures, OpenFOAM*

1. NOMENCLATURE

g	Gravitational acceleration
h	Convective heat transfer coefficient
H	Enclosure height
k	Thermal conductivity
K	Dimensionless kinetic energy, $K'/[(1/2)\rho\mathrm{Ra}(\alpha/H)^2]$
Nu	Nusselt number, hH/k
p	Dimensionless pressure, $p'/(\rho u^{*2})$
Pr	Prandtl Number, ν/α
Ra	Rayleigh Number, $[g\beta(T_w - T_0)H^3]/[\nu\alpha]$
t	Dimensionless time, t'/t^*
t^*	Time scale, $(H^2/\alpha)Ra^{-0.5}$
T	Temperature
T_0	Initial temperature
T_w	Wall temperature
$\mathbf{u}$	Dimensionless Velocity, $(\mathbf{u}'/u^*)$
u^*	Velocity scale, $(\alpha/H)Ra^{0.5}$
$\mathbf{x}$	Dimensionless position vector, $\mathbf{x}'/H$
W	Enclosure width

Greek Symbols

α	Thermal diffusivity
β	Volumetric coefficient of expansion
Δt	Dimensionless time step
Γ	Boundaries of the cavity (walls)
Γ_H	Heated boundaries (at T_w)
ν	Kinematic viscosity
Ω	Computation domain (cavity volume)
ρ	Density
θ	Dimensionless temperature, $(T - T_0)/(T_w - T_0)$

Subscripts

0	Initial values at $t = 0$
w	Wall

Superscripts

$*$	Characteristic scale
$'$	Variables with dimension

2. INTRODUCTION

Heat transfer by natural convection in a fluid confined between surfaces (cavity) at different temperatures is a problem that is related to many practical situations: sterilization processes for packaging liquid products in the food industry, solar collectors, cooling of electronic components heating or cooling buildings and heat transfer within the heat storage tanks are examples of application for this type of problem convective.

The experimental work of Lin (1982) also analyzes the patterns of flow and heat transfer in cavities. In this case, the cavity is cubic and all the walls are heated suddenly. The article shows a large number of flow patterns that change over time and heating are dependent on the Rayleigh number. Following the same research line, Tollini (1996) analyzed the transient heat transfer and three-dimensional cavity in a parallelepiped-shaped, with aspect ratio of $A_h = 1,74$, with the same dimensions as a milk box. Comparisons were made between the representative parameters of heat transfer and visualization of the structure of convective flow, for values of Rayleigh number between $1,08 \cdot 10^9$ and $8,24 \cdot 10^9$, experimentally and by simulation number. These two studies are similar to what will be analyzed in this study (Fig. (1a)).

Many articles on transient natural convection in two-dimensional and three-dimensional shells are presented in the literature. Fusegi *et al.* (1991) applied the method of finite difference scheme with third order accuracy to simulate a cubic air cavity for Rayleigh numbers 10^5 and 10^6. The specific effects of the horizontal thermal boundary conditions on the flow structure were analyzed in detail. It was found that the heat transfer through the horizontal walls activities improved convective flow. The numerically speed and temperature profiles provided in the planes of symmetry are consistent with the experimental measurements. Hsieh and Wand (1994) studied the problem of a three dimensional cavity heated in various ways. These works present the experimental flow visualization and transient temperature field. The history of the temperature and flow visualization during heating showed an oscillatory behavior to the initial formation of a stable core zone and recirculating a thermal stratification of the field.

In most studies of transient heating by natural convection in cavities, an interesting behavior of the flow field is observed: the initial phase, the vertical side walls, boundary layers are formed. Internal recirculation and an oscillating field heat is observed. Thereafter, the central zone is established and a thermal stratification is observed. The interaction between the flow patterns near the bottom wall heated with the primary recirculation zone, is a very interesting feature of this type of flow. The breaking of dynamic stability and symmetry of the flow during the heating time is operated in simulation of the cases.

The numerical results obtained by the finite volume method is used by OpenFOAM (*Open Field Operation and Manipulation*) to find the flow field and analyzing the evolution of integrated two parameters: the coefficient of heat transfer (expressed by the Nusselt number) and the kinetic energy of flow over time. The numerical simulations are carried out in the geometry of a cavity in the form of a cube and the other of a parallelepiped (Fig. (1a)) for different numbers of Rayleigh. The validation of the numerical results will be made by comparing the representative parameters of heat transfer obtained in experiments available in the work of Lin (1982) e Tollini (1996).

3. MATHEMATICAL MODELING

Consider a cavity parallelepiped of side measures $H = L = W$ filled with fluid with density ρ, dynamic viscosity ν and thermal conductivity k. The temperature distribution inside the cavity is initially uniform temperature T_0, with the fluid at rest. The six walls of the cavity are subjected to a temperature T_W, greater than T_0. The resulting natural convection considerably increases the heat transfer to the cavity. After a sufficient period of time, when the temperature reaches the temperature of the cavity T_W, convection inside ceases. The transient temperature distribution and transfer that result from this process are the biggest interest in our work.

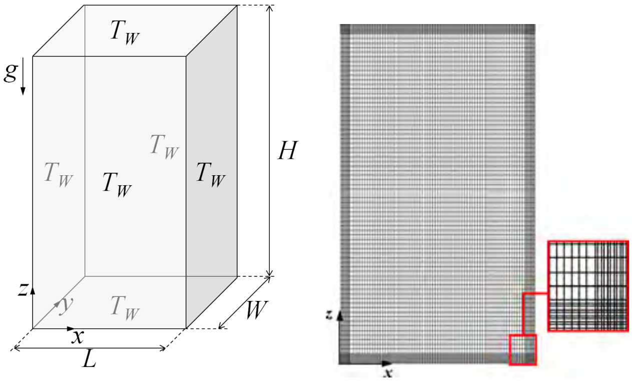
The transient flow into the enclosure is governed by the conservation equations of mass, momentum and energy. Considering the Boussinesq approximation those equations are written in a dimensionless form as:

$$\nabla \cdot \mathbf{u} = 0 \tag{1}$$

$$\frac{\partial \mathbf{u}}{\partial t} + \mathbf{u} \cdot \nabla \mathbf{u} = -\nabla p + \frac{\mathrm{Pr}}{\mathrm{Ra}^{0.5}} \nabla^2 \mathbf{u} - \mathrm{Pr}\, \theta\, \mathbf{e}_z \tag{2}$$

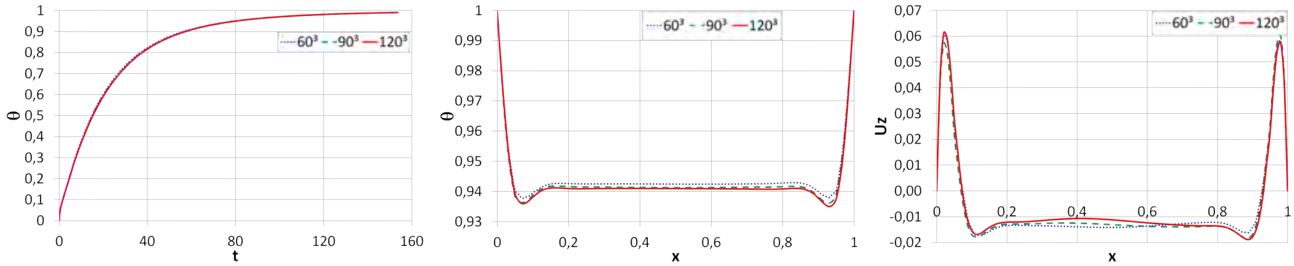
$$\frac{\partial \theta}{\partial t} + \mathbf{u} \cdot \nabla \theta = \frac{1}{\mathrm{Ra}^{0.5}} \nabla^2 \theta \tag{3}$$

Those equations are made dimensionless using the following scales: H for the length, $(\alpha/H)\mathrm{Ra}^{0.5}$ for the velocity and $(H^2/\alpha)\mathrm{Ra}^{-0.5}$ for the time. The dimensionless temperature, θ, is defined as $(T - T_0)/(T_w - T_0)$ and the Rayleigh number, Ra, is given by $[g\beta(T_w - T_0)H^3]/[\nu\alpha]$.



(a) (b)

Figure 1: Cavity: (a) Boundary Conditions; (b) Mesh Detail.



(a) $\theta \times t$ (b) $\theta \times x$ (c) $U_z \times x$

Figure 2: Mesh Test.

The homogeneous values for the variables θ and $\mathbf{u}$ are used as initial conditions. The boundary conditions are:

- Non slip conditions for the velocity in the cavity walls, *i. e.*, $\mathbf{u} = 0$.
- Prescribed temperature on the heated walls: $\theta = 1$

Although many problems of heat transfer are observed in simple geometries, as shown in Fig (1a), the analytical solution occurs by solving a system of simultaneous equations nonlinear partial differential and coupled. Due to the complexity of obtaining analytical solutions to such problems, it becomes necessary to use numerical methods for obtaining the flow field and thermal studied domain (Maliska (2004)). To solve the system of equations and nonlinear transient given by Equations (1) - (3), first became a discretization of the equations in space and time through the Finite Volume Method with OpenFOAM. The *solver* of OpenFOAM chosen to solve the problem of convection through the Boussinesq hypothesis is *buoyantBoussinesqPimpleFoam*, it uses a set of files that stores the information needed to solve the problem (case). These files are in a directory, and have the information as to the geometry description, details mesh boundary conditions, parameters for numerical methods and the physical properties of the problem.

In figure (1b) we can see a mesh with 729000 hexahedral volumes (90^3) and with a refinement near the wall that was construct the platform Salomé (*The Open Source Integration Platform for Numerical Simulation*). The decision to choose this discretization was based on comparison with two other meshes: 60^3 and 120^3. As can be seen in Figure (2) there is little variation in the results of temperature and velocity. However, in terms of total execution time of the simulation mesh with 90^3 elements took 174671 seconds, while mesh with 120^3 elements took 447035,2 seconds. For this reason, the mesh with 729000 elements was chosen for simulation.

The Nusselt number global ($\overline{\mathrm{Nu}}$) can be obtained from the heat flux as the heated walls (Bejan (2004)):

$$\overline{\mathrm{Nu}} = \frac{\overline{h}H}{k}, \tag{4}$$

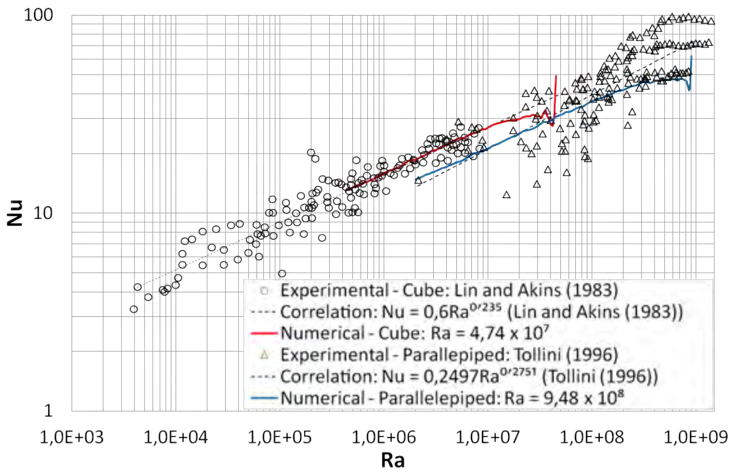


Figure 3: Comparison of Nusselt *versus* Rayleigh with experimental data.

considering $\overline{h} = \frac{\overline{q''}}{(T - T_0)}$ the coefficient of heat transfer and $\overline{q''} = \frac{k}{A_\Omega} \iint\limits_{\Omega} \frac{\partial T}{\partial \eta}\, d\Omega$ heat flux represented by the wall Ω, with area A_Ω toward η, perpendicular to the corresponding wall.

In order to assess the predictive ability of the *solver* of OpenFOAM about heat transfer in parallelepiped were compared with values ranging Nusselt to Rayleigh experimental data Lin and Akins (1983) and Tollini (1996) (Fig. (3)). For the cube, was simulated the case of $\mathrm{Ra} = 4,74 \cdot 10^7$ and for the parallelepiped, $\mathrm{Ra} = 9,48 \cdot 10^8$.

The kinetic energy of the flow field (K), dimensionless by $(1/2)\rho\mathrm{Ra}(\alpha/H)^2$, is calculated as the integral over the volume (V) cavity:

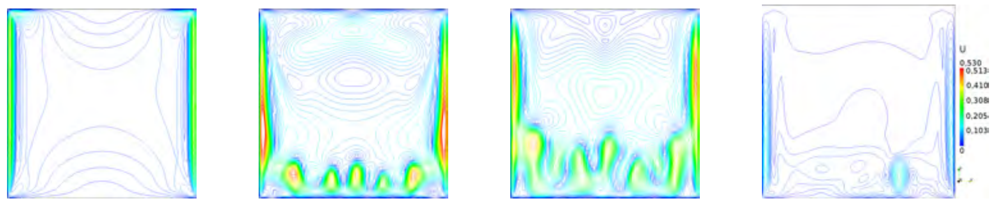
$$\mathrm{K} = \frac{1}{2}\rho \iint\limits_{V} (\mathbf{u} \cdot \mathbf{u})\, dV. \tag{5}$$

4. RESULTS AND DISCUSSION

The results of the simulations elapsed below the cube and the parallelepiped until they reach the temperature T_W, which occurred around $t = 1600$ s for a time step of $\Delta t = 0,1$ s if number of maximum *Courant* is $Co = 0,5$. The images were obtained by post-processing tool in OpenFOAM, named *paraFOAM*, adapted the program Paraview (*Open-source, multi-platform data analysis and visualization application*).

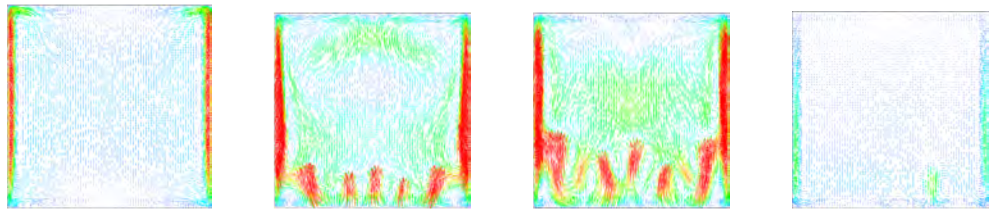
When we began to observe the temperature range found that the coldest region is always at the bottom of the cavity, this characteristic is accentuated in fluid layers closer to the vertical walls, featuring a thermal stratification in the cavity, as shown in Figs. (5 and 7). For more core layers, the temperature stratification along the vertical line of the cavity is not as pronounced. This is explained by the fact that in the central cavity layers, the velocities are relatively low, as seen in Figs. (4 and 6). Secondly, the convective currents explain the location of the cooler region at the bottom of the cavity. With respect to the velocity field is interesting to note that the center of movement of the main movement lies very close to the wall, and that some particles that rise not attain the heated upper layer, deviating from their paths and making a smaller circulation, taking a downward motion toward the lower face.

The movement of fluid within the cavity is the composition of the effects of heat transfer through each of its six sides, since the fluid was at rest and in thermal equilibrium with the outside environment. The bottom, the top and four lateral faces, represent, separately, a different effect on the movement of fluid. The four vertical side faces, on receiving heat, cause a tendency for fluid to rise rapidly through a thin layer along the sidewalls and descend slowly for a large mass in the central cavity. As the somatário areas of the four sides represents 2/3 of the external cavity and the influence of the gravitational acceleration, one may assume that the motion caused by them heat transfer is of paramount importance for the movement of fluid within the cavity.



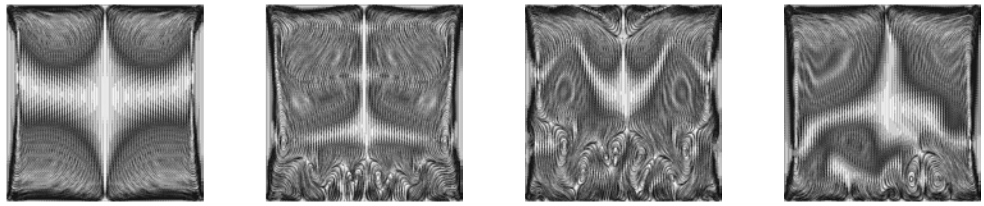
(a) (b) (c) (d)

Velocity Contours.



(e) (f) (g) (h)

Velocity Vectors.

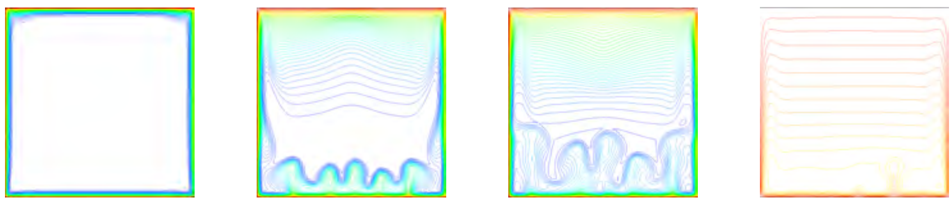


(i) (j) (k) (l)

Streamlines

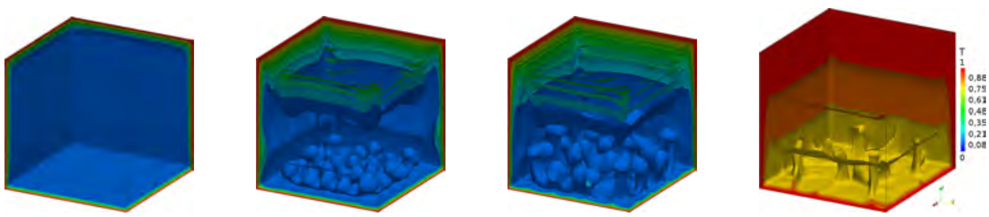
(i) $t = 1,0$ (ii) $t = 4,8$ (iii) $t = 5,8$ (iv) $t = 48,0$

Figure 4: Visualization Velocity Field for the Cube for $\mathrm{Ra} = 4,74 \cdot 10^7$ at $x = H/2$.



(a) (b) (c) (d)

Temperature Contours at $x = H/2$



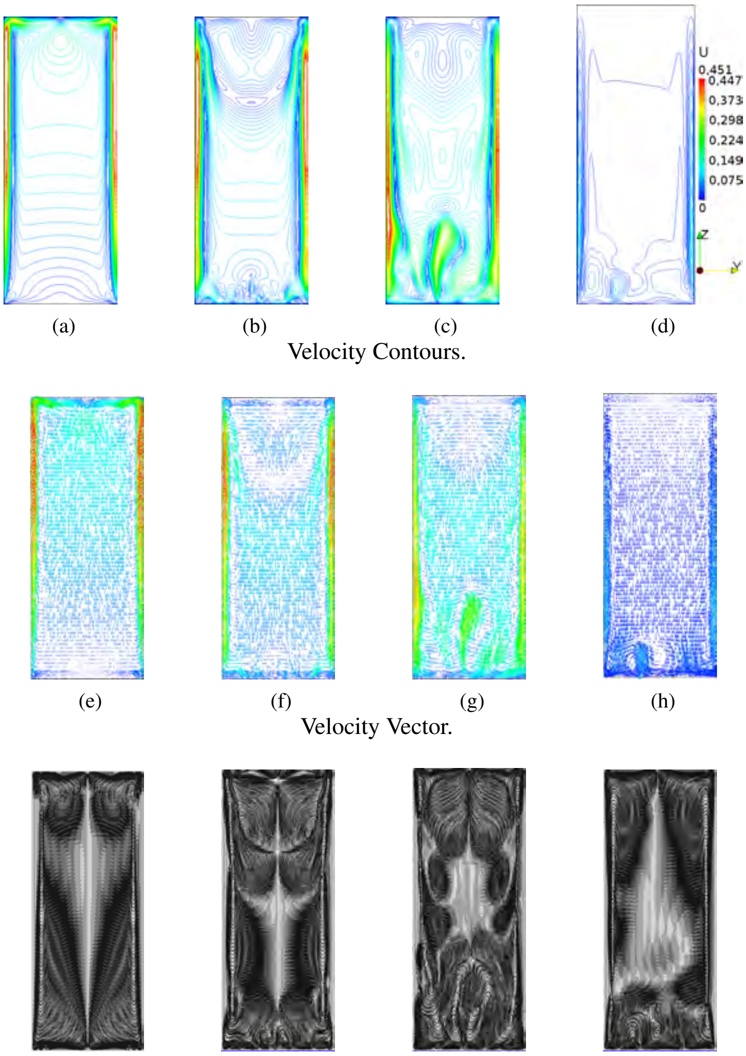
(e) (f) (g) (h)

Temperature Isosurfaces of the Cube.

(i) $t = 1,0$ (ii) $t = 4,8$ (iii) $t = 5,8$ (iv) $t = 48,0$

Figure 5: Visualization Temperature Field for the Cube for $\mathrm{Ra} = 4,74 \cdot 10^7$.

The bottom face, in turn, causes a tendency to climb the fluid as a whole, especially through the center of the cavity. Since there is a large body of fluid down the center, driven by the main movement of the fluid, some recirculation patterns are created at the base of the cavity, analogous to flow type Bénard cells, shown in isosurfaces temperature. In fact these structures resemble small thermal rising, mushroom-shaped, as reported by Sparrow *et al.* (1970). Now the top face of



(a) (b) (c) (d)
Velocity Contours.

(e) (f) (g) (h)
Velocity Vector.

(i) (j) (k) (l)
Streamlines.

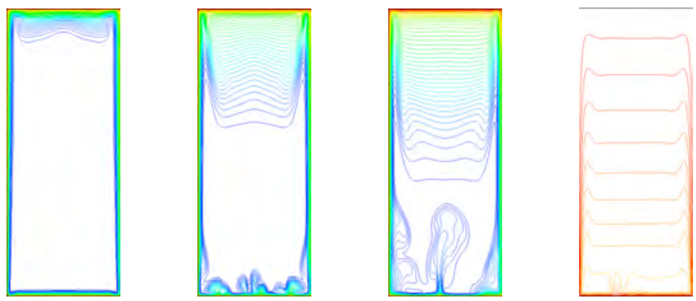
(i) $t = 2,6$ (ii) $t = 5,3$ (iii) $t = 7,9$ (iv) $t = 105,6$

Figure 6: Visualization Velocity Field for the Parallelepiped for $\mathrm{Ra} = 9,48 \cdot 10^8$ at $x = H/2$.

the cavity causes the formation of a thin layer and heated lighter near the top, where speeds are relatively very low. The fluid that goes up the side walls, to reach the heated layer near the top, is directed toward the center of the cavity, starting a slow descent. Thus, it is assumed that the influence on the movement of the upper face of the fluid within the cavity is almost zero, illustrated by velocity vectors.

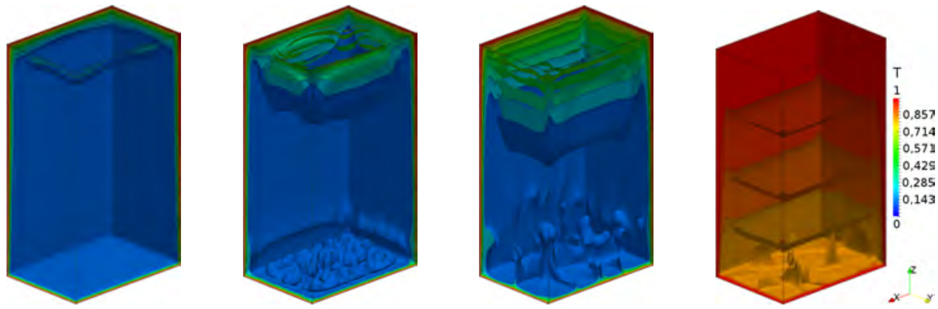
The first 5 min of simulation are the ones that show the effects of heat transfer. The kinetic energy reaches its peak during this period, as shown in Fig. (9). Since speed is an oscillatory behavior of its velocity components, mainly toward x (U_x) and y (U_y) (Fig. (8a and 8b)) while the direction of the axis z (U_z) (Fig. (8c), pointing upwards during this period, shortly after assuming downward movement.

Throughout the simulations performed during the remaining period, the movement of particles does not change much, with only a small increase in the thickness of the upper layer, characterized by low speeds, making therefore the particles that reach the top do not have energy to go to the center before starting the downward movement. The average velocity of fluid particles begin to decline gradually and stable layer on top increased slightly. Furthermore, particles of Benard cells seem to have reduced speed, especially for the apparent decrease in cell size, until the temperature throughout the cavity becomes T_W and the outflow ceases.



(a) (b) (c) (d)

Temperature Contours at $x = H/2$

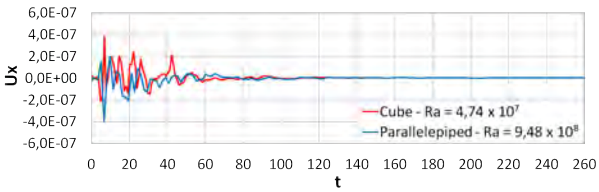


(e) (f) (g) (h)

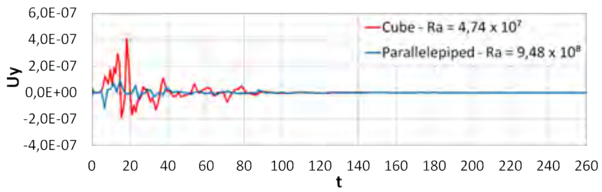
Temperature Isosurfaces in the Parallelepiped.

$(i)\ t = 2,6$ $(ii)\ t = 5,3$ $(iii)\ t = 7,9$ $(iv)\ t = 105,6$

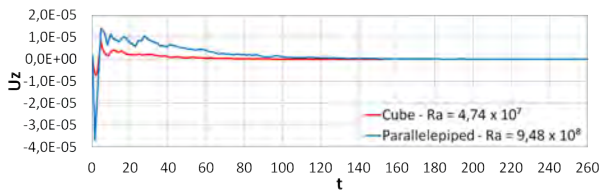
Figure 7: Visualization Temperature Field for the Parallelepiped for $\mathrm{Ra} = 9,48 \cdot 10^8$.



(a) U_x



(b) U_y



(c) U_z

Figure 8: Components of Average Velocity with Time Varying.

5. CONCLUSIONS

Results for the simulation of three-dimensional natural convection in a parallelepiped were presented, through which showed the influence of heat flow per side. Simultaneously, the first few seconds, the hydrodynamic and thermal boundary layers are formed. It can be seen that while it initiates the thermal stratification of the cavity bubbles fluid heated mushroom-shaped, formed in the bottom wall due to the increased temperature gradient on the underside, interacting with the fluid heats the side walls down slowly. The flow of fluid in the cavity has an oscillatory behavior in the first instants of the simulation and it has a connection with the peak kinetic energy of the particles and the formation of symmetrical patterns. These oscillations occur at the time of the formation of bubbles that rise fluid and affect the region of fluid that

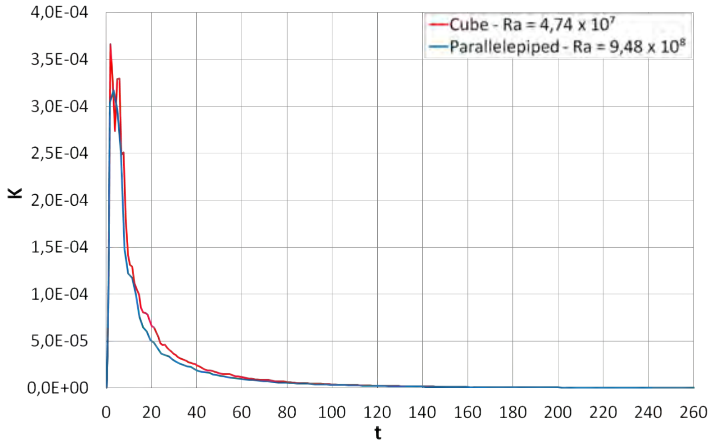


Figure 9: Variation of Kinetic Energy.

are descending. This occurs repeatedly until a certain moment when the kinetic energy decreases rapidly, while the speed and the cavity warms up completely.

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