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TURBULENT NATURAL CONVECTION IN A RECTANGULAR ENCLOSURE USING LARGE EDDY SIMULATION

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***Abstract**. Turbulent natural convective heat transfer in a square enclosure is studied in this work. The flow is considered to be two-dimensional, incompressible, and unsteady. Large Eddy Simulation (LES) with sub-grid model is applied to the turbulence. The flow is governed by the Navier-Stokes equations where the Boussinesq approximation is taken into consideration. The boundary conditions on the walls are isothermal temperature and convection. The equations are discretized using the Galerkin finite element method for a non-structured mesh. The local Nusselt number is calculated according to the range of some geometrical and thermal parameters. The accomplishment of this work can enable the study of the flow in the interior of refrigerators. Therefore, this study may be used as an initial step in the design of this sort of equipments. Finally, the results of the present work are compared to the numerical and experimental ones found in the literature.*

***Keywords**. turbulence, finite element method, enclosures, LES, natural convection.*

1. Introduction

The study of natural convection in enclosures has many engineering applications. The buoyancy force induced by density differences in a fluid causes natural convection. Natural convection within enclosures occurs in many practical situations ranging from simple space heating of domestic rooms to parts of industrial and nuclear installations. For example, this kind of flow occurs in building technology, cooling of electronic equipments, solar collectors, material processing, manufacturing, and so on.

So far, only a few works that use Large-Eddy Simulation (LES) to study the flow in a cavity with an internal heated body and isothermal boundary conditions have been reported.

A large eddy simulation (LES) is a promising approach in the analysis of unsteady three-dimensional turbulent flows with high Reynolds numbers. A direct simulation of turbulence gives us more accurate and precise data than experiments but it is unsuitable for the high Reynolds number flows because of computational limitations. It is known that the LES enables an accurate prediction of turbulence, but spends much less CPU time than the direct simulation.

The basic idea of the LES is to calculate only the larger scale than the grid size scale, called grid scale (GS), structures of turbulence, but to model smaller scale, called sub grid scale (SGS) structures. The governing equations for the GS quantities are derived by a spatial average or a filter procedure which removes SGS fluctuation from a Navier-Stokes equation.

In the literature, a large number of theoretical and experimental investigations are reported on natural convection in enclosures.

The flow in closed cavities where both the superior and inferior surfaces are isolated and the lateral ones are under different temperatures becomes turbulent when the Rayleigh number Ra is over 5×10^6 for an aspect ratio equal to 2.6, according to Chenoweth and Paolucci (1986) and Bispo et all (1996). For values of $Ra = 2.5 \times 10^7$, the flow is already under the turbulent regime.

Bispo et all (1996) studied turbulent natural convection in a cavity simulating an evaporator. On the upper horizontal surface, isotherm temperature was imposed and on the other surfaces, a constant convection boundary condition was defined.

Rasoul and Prinos (1997) studied the natural convection in enclosures. It was mainly devoted to the classical Rayleigh-Benard problem (hot bottom wall and cold top wall). They also studied the case of a square cavity with one vertical wall that is heated and the opposite one that is cooled. In this study, the effect of the inclination was studied in detail for various Ra numbers, ranging from 10^3 to 10^6 (laminar regime). The dependence of the mean Nusselt number on the Ra number was examined.

Cesini et all (1999) have analyzed the natural convection heat transfer from a horizontal heated cylinder within a rectangular cavity. In that work, conductive heat transfer was imposed on the upper wall. The flow was considered laminar.

Peng and Davidson (1999) used the finite volume method with the k-ω model to study flows with thermal stratification using turbulence models for low Reynolds numbers. Smoothing functions were applied to eliminate the problem of mesh dependency. This gave rise to correct asymptotic behavior near the wall. The geometry was a cavity with aspect ratio A = 5 and Rayleigh number Ra = 5 x 10^{10} with a heating wall temperature T_c = 77.2 ^oC and cooling wall temperature T_c = 31.4 ^oC.

In the work of Tian and Karayiannis (2000a), it was made an experimental study on the turbulent natural convection in a closed square enclosure filled with air and with one wall heated and the other one cooled. The Rayleigh number adopted is 1.58 x 10^9. Tian and Karayiannis (2000a) measured the temperature and velocity distributions in different locations of the cavity. The Nusselt numbers were investigated. The results obtained provide a benchmark problem with which computational codes can be validated.

New experimental results on turbulent natural convection in square cavities were shown by Tian and Karayiannis (2000b). The lateral surfaces are also kept at different temperatures. The authors presented results from the turbulent quantities including the fluctuation T, u, v and the Reynolds tensor. These results are also considered to be a benchmark problem to which computational code results can be contrasted.

Peng and Davidson (2001) studied the turbulent natural convection in a closed enclosure whose vertical lateral walls are maintained at different temperatures. Both the Smagorinsk and the dynamic models are applied to the turbulence simulation. Peng and Davidson (2001) modified the Smagorinsk model by adding the buoyancy term to the turbulent viscosity calculation. This model which is called the Smagorinsk model with buoyancy term. The computed results are compared to experimental data and show a stable thermal stratification under a low turbulence level (Ra = 1.58 x 10^9).

A study on the streamfunction and temperature distributions in a refrigerator was developed by Cortella et all (2001) using the finite volume method. The computational code was based on the vorticity-streamfunction formulation by incorporating the turbulent model LES. The turbulent fluxes were estimated according to the vorticity transfer theory (VTT).

It was performed in the work of Oliveira and Menon (2002a), a numerical study of turbulent natural convection in square enclosures. The finite volume method and the large eddy simulation were used. The enclosure lateral surfaces were kept at different temperatures and the upper and lower surfaces were isolated. The flow is studied for low Rayleigh numbers Ra = 1.58 x 10^9. Three turbulence LES models are used: the Smagorinsk model, the Smagorinsk model with the buoyancy term, and the model based on the vorticity transfer theory (VTT). In all cases, it was considered Pr = 0.7 and Ra = 1.58 x 10^9.

A natural convection heat transfer study in closed rectangular enclosures was accomplished by Oliveira and Menon (2002b). This study considered a turbulent regime and a k-ω turbulence model. The equations were discretized by the use of the finite volume method with Cartesian grid and colocalized arrangement. The conservation equations were for the unsteady regime. However, the results of the cases studied were presented when the regime achieved the steady regime. The local and average Nusselt numbers were evaluated for Rayleigh numbers between 10^5 and 10^{10}. The Prandtl number was 0.71 and the aspect ratios were A = 5; 2; 1 and 0.5.

In the present work, a numerical analysis is performed for turbulent natural convection in a single horizontal square cavity where the vertical lateral walls are isothermal, while the lower and upper horizontal walls are adiabatic. There is a conductive square body within the cavity. The objective of the heat transfer analysis is the investigation of the Nusselt number distribution on the vertical walls for various Rayleigh numbers. Comparisons are made not only with experimental and numerical results found in Tian and Karyiannis (2000), Oliveira and Menon (2002a), but also with the numerical studies by Lankhorst (1991) and Cesini et all (1999).

2. Problem description

Figure (1) shows the geometry with the fluid and solid domains Ω_f and Ω_s, respectively. The typical mesh used in this work is also presented. It will be considered a square cavity whose upper and lower horizontal surfaces S_2 and S_4 are adiabatic. The vertical surfaces are isothermal with temperatures T_c on S_3 and T_h on S_1. The solid body, which is centered in the cavity, is a square with an aspect ratio $A_c = H_c/H = 0.25$, where H_c is the internal body height and H is the characteristic dimension of the cavity. It is considered that a thermal diffusivity α_s of the internal body be twice the value of the air thermal diffusivity α. The initial condition is: initial temperature $T = T_o$ in Ω_f and $T = T_h$ in Ω_s; $\psi = \omega = 0$ in $\Omega_f \cup \Omega_s$, where ψ, ω, and Ω are the streamfunction, the vorticity, and the total computational domain, respectively. All the physical properties of the fluid are constant except the density in the buoyancy term where it obeys the Boussinesq approximation. It is assumed that the third dimension of the cavities is large enough so that the flow and heat transfer are two-dimensional.

The mesh is non-structured. The computational domain $\Omega = \Omega_f \cup \Omega s$ was divided into 6,924 triangular elements and 3,663 nodal points.

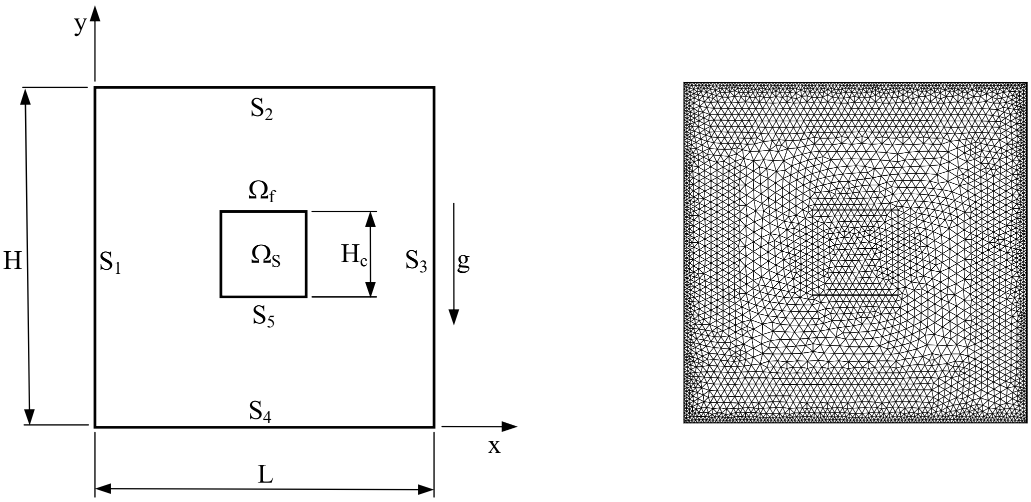

Figure 1. Geometry and mesh arrangement studied in the present work.

2.1. Problem Hypotheses

The following hypotheses are employed in the present work: unsteady regime; turbulent regime; two-dimensional flow; incompressible flow; constant physical properties of the fluid, except the density in the buoyancy terms.

3. Equations

The governing conservation equations are:

$$\frac{\partial u_i}{\partial x_i} = 0\,, \tag{1}$$

$$\frac{\partial u_i}{\partial t} + \frac{\partial u_i u_j}{\partial x_j} = -\frac{1}{\rho}\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left\{\nu\left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right]\right\} + g\beta(T - T_0)\delta_{2j}\,, \tag{2}$$

$$\frac{\partial T}{\partial t} + \frac{\partial u_j T}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\alpha\frac{\partial T}{\partial x_j}\right] + S\,, \tag{3}$$

where x_i are the axial coordinates x e y, u_i are the velocities components, p is the pressure, T is the temperature, ρ is the fluid density, ν is the kinematic viscosity, g is the gravity acceleration, β is the fluid volumetric expansion coefficient, δ_{2j} is the Kronecker delta, α is the thermal diffusivity, and S is the source term. The last term in Eq. (2) is the Boussinesq buoyancy term where T_0 is the reference temperature $T_0 = (T_h + T_c)/2$. T_h and T_c are the temperatures on the vertical wall and on the internal body.

In the large eddy simulation (LES), a variable decomposition similar to the one in the Reynolds decomposition is performed, where the quantity φ is split as follows:

$$\varphi = \overline{\varphi} + \varphi'\,, \tag{4}$$

where $\overline{\varphi}$ is the large eddy component and φ' is the small eddy component.

The following filtered conservation equations are shown after applying the filtering operation to Eq. (1) and (3). This is done using the volume filter function presented in Krajnovic (1998). The density is constant.

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0\,, \tag{5}$$

$$\frac{\partial \bar{u}_i}{\partial t} + \frac{\partial \overline{u_i u_j}}{\partial x_j} = -\frac{1}{\rho}\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j}\left\{\nu\left[\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i}\right]\right\} + g\beta\left(\bar{T} - T_0\right)\delta_{2j}\,, \tag{6}$$

$$\frac{\partial \bar{T}}{\partial t} + \frac{\partial \overline{u_j T}}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\alpha\frac{\partial \bar{T}}{\partial x_j}\right] + S\,. \tag{7}$$

In Equations (5) to (7), $\overline{u_i u_j}$ and $\overline{u_j T}$ are the filtered variable products that describe the turbulent momentum transport and the heat transport, respectively, on the large and sub-grid scales.

According to Oliveira and Menon (2002a), the products $\overline{u_i u_j}$ and $\overline{u_j T}$ are split into other terms by including the Leonard L_{ij} tensor, the Crossing tensor C_{ij},, the Reynolds sub-grid tensor R_{ij}, the Leonard turbulent flux $L_{\theta j}$, the Crossing turbulent flux $C_{\theta j}$, and the sub-grid turbulent flux θ_j. The Crossing and Leonard terms, according to Padilla (2000), can be neglected. After the development shown in Oliveira and Menon (2002a), the following conservation equations are obtained:

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0\,, \tag{8}$$

$$\frac{\partial \bar{u}_i}{\partial t} + \frac{\partial\left(\bar{u}_i \bar{u}_j\right)}{\partial x_j} = -\frac{1}{\rho}\frac{\partial \bar{p}}{\partial x_i} + \nu\left(\frac{\partial^2 \bar{u}_i}{\partial x_j \partial x_j}\right) - \frac{\partial \tau_{ij}}{\partial x_i} + g\beta\left(\bar{T} - T_0\right)\delta_{2j}\,, \tag{9}$$

$$\frac{\partial \bar{T}}{\partial t} + \frac{\partial\left(\bar{u}_j \bar{T}\right)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\alpha\frac{\partial \bar{T}}{\partial x_j}\right] + \frac{\partial \theta_j}{\partial x_j}\,. \tag{10}$$

where Pr is the Prandtl number with $\alpha = \nu/Pr$. The tensors τ_{ij} and θ_j that appear in Eq. (9) and (10) are modeled in the forthcoming topics.

3.1. Sub-grid model

Many sub-grid models use the diffusion gradient hypothesis similar to the Boussinesq one that expresses the sub-grid Reynolds tensor in function of the deformation rate and kinematic energy. According to Silveira-Neto (1998), the Reynolds tensor is defined as:

$$\tau_{ij} = -2\nu_T \bar{S}_{ij} - \frac{2}{3}\delta_{ij}\bar{S}_{kk}\,, \tag{11}$$

where ν_T is the turbulent kinematic viscosity and $\bar{S}_{ij}$ is the deformation tensor rate given by:

$$\bar{S}_{ij} = \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i}\right). \tag{12}$$

Substituting $\bar{S}_{ij}$ in Eq. (9):

$$\frac{\partial \bar{u}_i}{\partial t} + \frac{\partial\left(\bar{u}_i \bar{u}_j\right)}{\partial x_j} = -\frac{1}{\rho}\frac{\partial \bar{P}}{\partial x_i} + \nu\left(\frac{\partial^2 \bar{u}_i}{\partial x_j \partial x_j}\right) + \frac{\partial}{\partial x_j}\left\{\nu_T\left[\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i}\right]\right\} + g\beta\left(\bar{T} - T_0\right)\delta_{2j}\,. \tag{13}$$

In a similar way, the energy equation is obtained:

$$\frac{\partial \overline{T}}{\partial t}+\frac{\partial\left(\overline{u}_j\overline{T}\right)}{\partial x_j}=\frac{\partial}{\partial x_j}\left[\left(\alpha+\alpha_T\right)\frac{\partial \overline{T}}{\partial x_j}\right]. \tag{14}$$

where the turbulent thermal diffusivity is calculated as:

$$\alpha_T = \nu_T / Pr_T \, , \tag{15}$$

and Pr_T is the turbulent Prandtl number.

The sub-grid models give the following expression for the turbulent viscosity ν_t:

$$\nu_T = c\,\ell\, q \, , \tag{16}$$

where c is a dimensionless constant, ℓ and q are the scale lengths and the velocity, respectively.

The parameter ℓ is related to the filter size and it is usually used in the two-dimensional case with a rectangular element as:

$$\ell=\overline{\Delta}=\left(\Delta_1\Delta_2\right)^{1/2} , \tag{17}$$

where Δ_1 and Δ_2 are the filter lengths in x and y directions.

3.1.1. Model sub-grid of vorticity transfer theory (VTT)

The turbulence model implemented in this study can be classified as a large eddy simulation (LES), according to Cortella et all (2001), where the turbulent fluxes are estimated on the basis of the vorticity transfer theory (VTT). In accordance with approach mentioned previously, the turbulent kinematic viscosity is computed as:

$$\nu_T=\left(C\,\Delta\right)^3\left[\left(\frac{\partial\omega}{\partial x}\right)^2+\left(\frac{\partial\omega}{\partial y}\right)^2\right]^{1/2} , \tag{18}$$

where C is the dimensionless constant. From Eq. (18), ω is the vorticity and Δ is the element average dimension given by:

$$\omega=\frac{\partial v}{\partial x}-\frac{\partial u}{\partial y}\, , \; \Delta=\left(\prod_{k=1}^{N} d_k\right)^{1/N} , \tag{19}$$

where, x and y are the axial coordinates, $\vec{x}$ is the position vector of the center of the reference element and d_k (k = 1 to N), the distance from the center of the reference element to the center of the adjacent element. More details on this model can be seen on the work of Métais e Lesieur (1996).

For isotropic turbulence, the dimensionless constant C = 0.2 can be satisfactorily used according to Cortella et all (2001). The turbulent thermal diffusion is estimated from the turbulent kinematic viscosity, by assuming:

$$Pr_T = \nu_T/\alpha_T = 0.4 \, . \tag{20}$$

4. Initial and boundary conditions

From this section on, the upper bars that mean the average values $\overline{T}$ and $\overline{u}$ will be omitted.

Figure (1) pictures the enclosure on which the following initial conditions are imposed:

$$u(x,y,0) = 0,\ v(x,y,0) = 0,\ T(x,y,0) = 0 \, , \qquad \text{in } \Omega, \tag{21}$$

$$u = v = 0,\ T = T_h = 1 \, . \qquad \text{In the solid body} \tag{22}$$

The boundary conditions are:

$$u = v = 0,\ T = T_h = 1, \qquad \text{on } S_1, \tag{23}$$

$$u = v = 0,\ \frac{\partial T}{\partial y} = 0, \qquad \text{on } S_2, \tag{24}$$

$$u = v = 0,\ T = T_c = -1, \qquad \text{on } S_3, \tag{25}$$

$$u = v = 0,\ \frac{\partial T}{\partial y} = 0, \qquad \text{on } S_4, \tag{26}$$

Besides that, the flow field can be described by streamfunction ψ and vorticity ω distributions given by:

$$u = \partial\psi/\partial y\ ,\ \upsilon = -\partial\psi/\partial x\ ,\ \omega = (\partial\upsilon/\partial x) - (\partial u/\partial y), \tag{27}$$

where u and υ are the velocity components in the x and y directions, respectively. Hence, the continuity equation given by Eq. (1) is exactly satisfied. Working with dimensionless variables, it is possible to deal with Rayleigh number Ra, Prandtl number Pr, and the enclosure aspect ratio A given by:

$$Ra = Pr\left[g\beta(T_h - T_c)H^3/\nu^2\right] = 1.58\times10^9,\ Pr = \nu/\alpha = 0.7 \ \text{ and } \ A = H/L = 1.0, \tag{28}$$

where H is the characteristic dimension and L is the cavity width. In the present work, the thermal diffusivity on the solid α_s is considered to be twice as much the fluid thermal diffusivity α, that is, $\alpha_s = 2\ \alpha$.

5. Numerical method

Equations (8), (9), and (10) are solved through the finite element method (FEM) with linear triangular elements. The discretization uses the Galerkin formulation. The system of equations is solved with the Gauss Quadrature. The problem solution follows the steps below:

($1^{\underline{0}}$) the streamfunction field ψ is solved in Eq. (27);
($2^{\underline{0}}$) the wall vorticity is determined in a matrix form, according to Silveira-Neto et all (2000);
($3^{\underline{0}}$) the boundary conditions for vorticity are applied;
($4^{\underline{0}}$) the vorticity in the interior is calculated according to Eq. (27);
($5^{\underline{0}}$) the temperature field is solved through Eq. (10);
($6^{\underline{0}}$) the local Nusselt is obtained using Eq. (29);
($7^{\underline{0}}$) the time and the interaction are increased using the time step Δt and the unity, respectively. Then it turns to the first step ($1^{\underline{0}}$) starting it all over again till it reaches the stop criterion.

The local Nusselt number Nu on the vertical isothermal cold wall is defined as:

$$Nu = (\partial T/\partial x)_w\ H/(T_h - T_c)\,. \tag{29}$$

6. Numerical method validation

Two geometries are studied here in order to compare the results with the ones found in the literature and then to validate the computational code in FORTRAN. In the first comparison, the study of the natural turbulent flow in a square enclosure with different temperatures for various Rayleigh numbers is carried out in Brito et all (2002). The second comparison is made in the present work considering a laminar flow in a rectangular enclosure with an internal cylinder. In the second validation, the results are compared to the numerical and experimental ones found in Cesini et all (1999).

In the first comparison, it is also used the large eddy simulation (LES), however, the sub-grid model adopted in Brito et all (2002) is different from the one used in the present work. The results in Brito et all (2002) are compared not only to the experimental and numerical ones in Peng and Davidson (2001), but also to the numerical ones in Lankhorst (1991). A good agreement is verified. It is also made a comparison between the results from Brito et all (2002), for the average dimensionless temperature and the experimental ones given by Tian and Karayiannis (2000).

Figure (2) presents the geometry of the flow domain Ω and the mesh used in the second comparison. This geometry is numerically and experimentally analyzed by Cesini et all (1999). Cesini et all (1999) consider a two-dimensional laminar flow. For the numerical simulation made by Cesini et all (1999), a dimension z is adopted in such a way that the flow can be considered two-dimensional. Cesini et all (1999) study a rectangular enclosure where the horizontal surface has a constant convection heat transfer whereas the horizontal lower surface is submitted to isolation. The vertical surfaces are isothermal having a low temperature T_c. On the other hand, the cylinder surface has a high temperature T_h. In the second comparison, the mesh is non-structured with linear triangular elements. The computational domain is divided into 5,790 triangular elements with 3,011 nodal points. The rectangular enclosure height and width are, respectively, H = 1 [m]; L = 0.8771 [m], with a cylinder diameter D = 0.2456 [m]. The ratio aspect used is $A_c = L/D \approx 3.6$. Table (1) shows the average Nusselt numbers Nu_m around the cylinder that are computed through the code of the present work.

Table 1 presents the results for the comparison of the average Nusselt number Nu_m of the present work with those ones in the work of Cesini et all (1999). The maximum deviation is 11.88 % with Rayleigh number equal to 3.4×10^3 using a mesh with 5,790 elements and 3,011 node points. The minor deviation is 7.53 % to Rayleigh number equal 3.0×10^4.

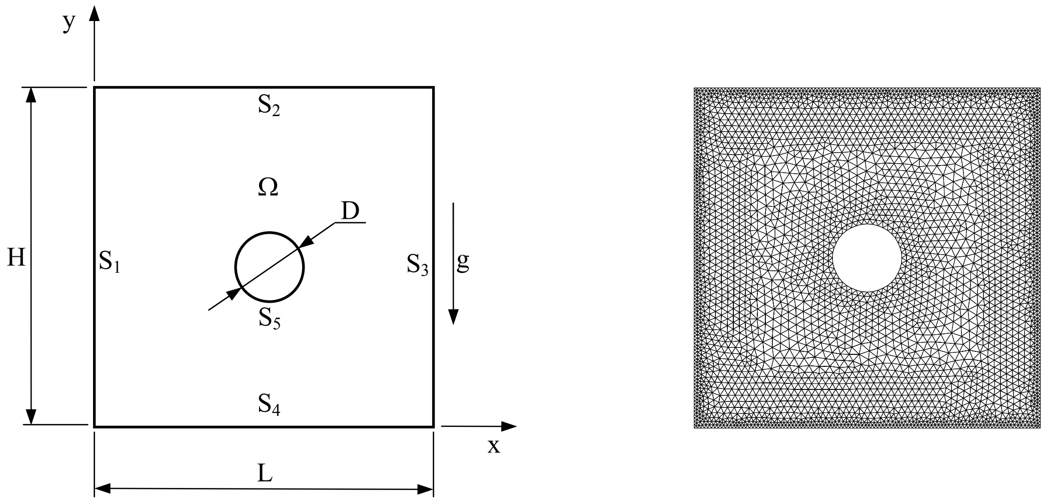

Figure 2. Geometry and mesh used na segunda validação.

Table 1. Average Nusselt numbers Nu_m, comparisons with the work of Cesini et all (1999).

	$A_c \approx 3.6$			
	Present Prediction	Numerical Cesini et al. (1999)	Experimental Cesini et al.(1999)	(%)
$Ra = 1.3 \times 10^3$	2.148	2.35	2.35	9.40^*
$Ra = 2.4 \times 10^3$	2.518	2.75	2.79	10.80^*
$Ra = 3.4 \times 10^3$	2.735	2.98	3.06	11.88^*
$Ra = 5.0 \times 10^3$	2.980	3.25	—	9.06
$Ra = 1.0 \times 10^4$	3.441	3.74	—	8.69
$Ra = 2.0 \times 10^4$	3.977	4.29	—	7.87
$Ra = 3.0 \times 10^4$	4.343	4.67	—	7.53
$Ra = 4.0 \times 10^4$	4.628	4.99	—	7.82
$Ra = 5.0 \times 10^4$	4.863	5.25	—	7.96
$Ra = 7.5 \times 10^4$	5.320	5.77	—	8.46

* in relation to the experimental data.

7. Results

Figure (3) shows the dimensionless average temperature distribution $(\overline{T} - T_c)/\Delta T$. It is noted that near the isothermal surfaces S_1 and S_3 there is a temperature peak that increases downstream.

For enclosures without an internal body, according to Tian and Karayiannis (2000), the thermal layer thickness is thicker for enclosures with isolated walls. For the ones with horizontal surfaces and with perfect conduction, that is, the isothermal ones, the thickness is thinner. It is verified that within the regions between the internal body and S_1 and S_3, the convective effect is predominant due to the heating and cooling of the fluid from the hot isothermal surface S_1 and from the cold isothermal surface S_3. The fluid flowing clockwise coming from a lower temperature region causes a deformation on the isotherms, hence, increasing the heat transfer. One can also note regions of high temperature gradients near surfaces S_1 e S_3 and also near the lower and upper parts of the internal body. The internal body presence makes a clear temperature stratification happen along the computational domain Ω.

Figure (4) depicts the velocity vectors and the stream function distributions for the sub-grid model vorticity transfer rate that is implemented in the present work. A clear temperature stratification is seen in the enclosure central region. Recirculation cells are also formed everywhere in the region. Small ones are near the inferior vertical surface S_3 and the superior surface S_1. Figure (4) also shows the average velocity vectors calculated in the interval 400-600 t_0 for the case studied in the present work.

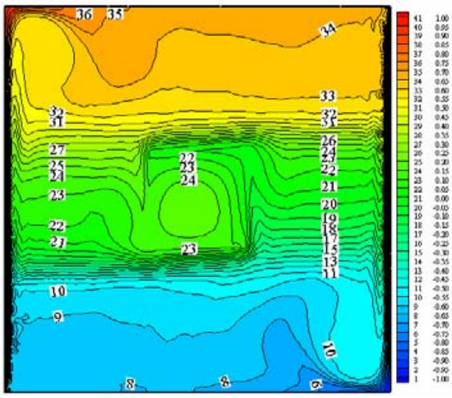

Figure 3. Dimensionless average temperature for time t = 600 t_0.

Figure (5) presents the local Nusselt numbers Nu calculated on the vertical surface S_1. It is noted that the highest heat transfer rates take place in the region below the mean part of the enclosure height, as expected. As the fluid is heated, its density decreases and it goes up due to buoyancy forces, hence, lowering the heat transfer rates since S_1 has a high temperature.

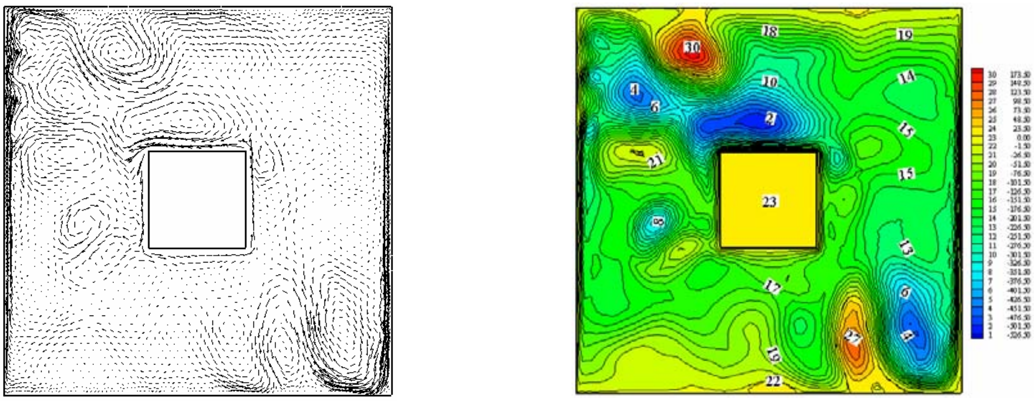

Figure 4. Velocity vectors u and v and stream function ψ for time t = 600 t_o .

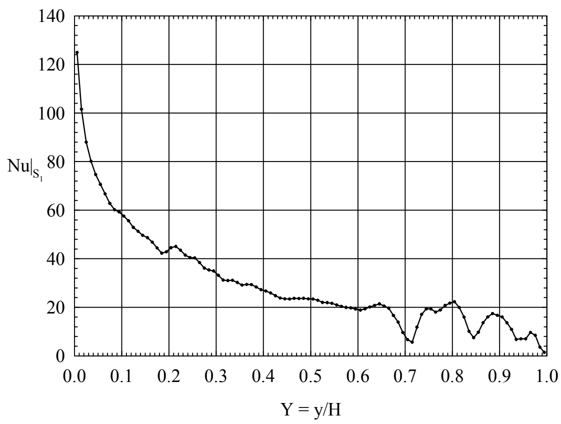

Figure 5. Local Nusselt number Nu on the heated surface S_1 for Ra = 1,58 × 10^9, Pr = 0,7 and t = 600 t_o.

8. Conclusions and comments

In this work, the turbulent natural convection is studied in a square enclosure with an internal body with a high initial temperature. It is used the large eddy simulation with vorticity transfer theory (VTT) sub-grid modeling according to Cortella et al. (2001). The conservation equations are discretized by the Galerkin finite element method with linear triangular elements.

Two cases are used for validation of the computational domain of the present work.

The first comparison shown in the present study is performed in Brito et all (2002) and compared to the results in

Peng and Davidson (2001). In this first comparison, the turbulent flow in an enclosure with vertical surfaces at different temperatures is carried out. The results for average velocities $\overline{u}/U_0$ at x / L = 0.5 are presented. Theses ones are contrasted with the ones from Oliveira and Menon (2002a), Peng and Davidson (2001), and with the experimental ones. A good agreement is verified for the entire enclosure. Also in this first comparison, the dimensionless average temperatures $(\overline{T} - T_c)/\Delta T$ at x / L = 0.5 are obtained. The data achieved in Brito et all (2002), which were used in the first comparison of the present work, are compared to the ones in Oliveira and Menon (2002a), Tian and Karayiannis (2000) and Lankhorst (1991), giving a good agreement for all the vertical part of the region.

In the second comparison, the non-isothermal flow is in an enclosure with a cylinder at a high isothermal temperature and vertical walls at a low isothermal temperature. The horizontal superior surface has a prescribed constant convection, simulating a refrigerator case. The horizontal inferior surface is thermally isolated. This case is studied both numerically and experimentally by Cesini et all (1999). The numerical results presented are compared to the numerical and experimental ones from Cesini et all (1999), showing a good agreement.

For the analyzed results in the present work, it is verified that the model based on TTV formulation, according to Cortella et al. (2001), presents a certain similarity with the case studied by Brito et all (2002), where LES is used with a sub-grid structure function of second order velocities. For that being so, the output found by Brito et all (2002), agrees well with the numerical results in Oliveira and Menon (2002a), which use the sub-grid model TTV and the Smagorinsky sub-grid model. So far, it is not found studies with a similar geometry employed here and with the same turbulence modeling using LES with a sub-grid modeling TTV.

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10. References

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