INFLUENCE OF THE EVAPORATOR POSITION IN A STATIC DOMESTIC REFRIGERATOR

Denise Kinoshita, denikino@aluno.feis.unesp.br
José Luiz Gasche, gasche@dem.feis.unesp.br
Faculdade de Engenharia de Ilha Solteira – Unesp, Avenida Brasil, 56, Ilha Solteria, 15385-000, Brasil

Abstract. *The temperature and velocity distributions of the air inside the cabinet of domestic refrigerator affect the quality of food products. If the consumer knows the locations of warm and cold zones in the refrigerator, the products can be placed in the right zone. In addition, the knowledge of the thickness of thermal and hydrodynamic boundary layers near the evaporator and the other walls is also important. If the product is too close to the evaporator wall, freezing can occur, and if it is too close to warm walls, the products can be deteriorated. The aim of the present work is to develop a steady state computational fluid dynamics (CFD) model for domestic refrigerators working on natural convection regime. The Finite Volume Methodology is chosen as numerical procedure for discretizing the governing equations. The SIMPLE – Semi-Implicit Method for Pressure-Linked Equations algorithm is used to solve the system of algebraic equations. The model is applied to a commercial static refrigerator, where the cabinet is considered an empty three-dimensional rectangular cavity with one drawer at the bottom of cabinet, but without shelves. In order to analyze the velocity and temperature fields of the air flow inside the cabinet the evaporator temperature, Te, was varied from – 20 °C to 0 °C, and nine different evaporator positions are evaluated for evaporator temperature of - 15 °C. The cooling capacity of the evaporator for the steady state regime is also computed for each case. One can say that the vertical positioning on the evaporator inside the cabinet plays an important role on the temperature distribution inside the cabinet.*

Keywords*: Domestic refrigerator, natural convection, evaporator positions*

1. INTRODUCTION

Since ancient times, human being needed to preserve perishable food. Precarious and unhygienic processes were used for a long time in order to solve this problem. However, a satisfactory conservation was never achieved. It was found that one of the main factors influencing the food deterioration was the temperature of the storage environment. With the advance of the industrial revolution and through much research effort it was possible to develop efficient cooling systems that are now used in several applications, from large stores to domestic refrigerators.

Currently, domestic refrigerators are essential products in everyday people life, making them objects of constant study and improvement, especially in relation to energy saving and storage quality of refrigerated products. The air temperature and velocity distributions inside the cabinet of domestic refrigerators affect the quality of food products. If the consumer knows the location of warm and cold zones in the refrigerator, the products can be placed in the right zone. Thus, an inadequate distribution of temperature and air velocity can cause a premature deterioration of these products. In addition, the knowledge of the thickness of thermal and hydrodynamic boundary layers near the evaporator and the other walls is also important. If the product is too close to the evaporator wall, freezing can occur, and if it is too close to warm walls, the products can be deteriorated.

In the last 10 years numerical and experimental analysis has been addressed in the literature to investigate the behaviour of the air temperature and velocity fields inside domestic refrigerators working with natural and forced convection.

Fukuyo *et al*. (2003) presented a new system to improve the thermal uniformity and the cooling rate of food inside the cabinet of domestic refrigerators working with forced convection. The numerical solution of the three-dimensional flow was obtained using the computational fluid dynamics based on the Finite Volume Method. The new system increased the thermal uniformity, reducing by half the temperature difference over the conventional system and increased the cooling rate of foods.

Ding *et al.* (2004) studied numerically and experimentally various ways to improve the thermal uniformity within domestic refrigerators working natural and forced convection. The numerical analysis was performed using a commercial code to solve the three-dimensional turbulent flow. Through changes in the configuration of the cabinet increased uniformity in the temperature field. It was shown that the distance between the shelves and the walls plays a key role for the uniform temperature inside the refrigerator.

Gupta *et al*. (2007) have studied numerically the flow and heat transfer inside a household refrigerator frost-free type. The numerical solution of the three-dimensional flow was obtained using a commercial code based on finite volume method with unstructured meshes. The numerical results were validated by experimental data obtained by the authors. Modifications were suggested in the cabinet configuration to improve the performance of the refrigerator.

Laguerre *et al*. (2005) have performed an experimental investigation of the natural convection heat transfer of air in a closed cavity with vertical heated and cooled walls with application in domestic refrigerators. The result of temperature distribution confirmed the theory that there is temperature stratification, hot zone on top and cold zone on bottom. The thickness of the thermal boundary layer was approximately 2 cm. The temperature field in the boundary layers and in the central region of the cavity was measured with the empty cavity, and also with the cavity provided with artificial obstacles simulating the foods in order to study the influence of obstacles on the temperature field. It was observed that the presence of obstacles significantly modifies the heat transfer.

Laguerre *et al*. (2007) have accomplished a numerical and experimental investigation to obtain velocity and temperature fields inside a refrigerator for three different cases: empty refrigerator, refrigerator with shelves, and refrigerator supplied with products. They have used a commercial code based on the finite volume method, assuming constant temperature in the evaporator and laminar airflow. The comparison of the temperature distribution between the experimental and numerical results showed good agreement when radiation was taken into account.

Laguerre *et al*. (2008) have proposed an experiment to study the airflow inside domestic refrigerators. The experiment was carried out using a transparent refrigerator model, which makes it possible to visualize and measure the airflow using Particle Image Velocimetry (PIV). As expected, they observed circular airflow in the cavity, downward flow near the cold wall and upward flow near the other walls. The maximum velocity was observed near the bottom of the cold wall.

Amara *et al*. (2008) have studied numerically and experimentally the flow in a three-dimensional household refrigerator. The authors used the same technique used by Laguerre et al. (2008) to obtain the experimental data of velocity fields and compared with the numerical results obtained by a commercial code based on the Finite Volume Method. The influence of temperature and size of the cold wall were studied. The results showed a good agreement between the measured and the calculated fields.

Several other authors have focused on the performance and energy consumption of domestic refrigerator due to the their wide use, Laguerre *et al*. (2002), James *et al*. (2008), Hermes *et al*. (2009), Hermes and Melo (2009).

Despite the importance of this problem, only a few theoretical and experimental studies in this regard have been carried out mainly on static (conventional natural convection) refrigerator, which are currently widely used in Brazil.

The objective of this work is to study numerically the influence of the evaporator positioning on the temperature and velocity distributions inside the cabinet of a domestic refrigerator working on natural convection. Nine different evaporator positions are evaluated for evaporator temperature of -15 °C. The cooling capacity is also calculated for evaporator temperatures varying from -20 to 0 °C.

2. MATHEMATICAL MODEL

A 350 liters commercial refrigerator working on natural convection (static refrigerator) was modeled as an empty three-dimensional rectangular cavity with one bottom drawer, but without shelves, as shown in Fig. 1. Fig. 1a depicts the geometry of the cabinet, with lateral opening in the acrylic lid, while Fig. 1b shows the dimensions of the cabinet. Fig. 1c shows a typical wall of the refrigerator, which is composed by a thin steel plate, insulation, and a thin polyurethane plate. It is also shown in the same Fig. the thermal resistances considered to specify the boundary conditions.

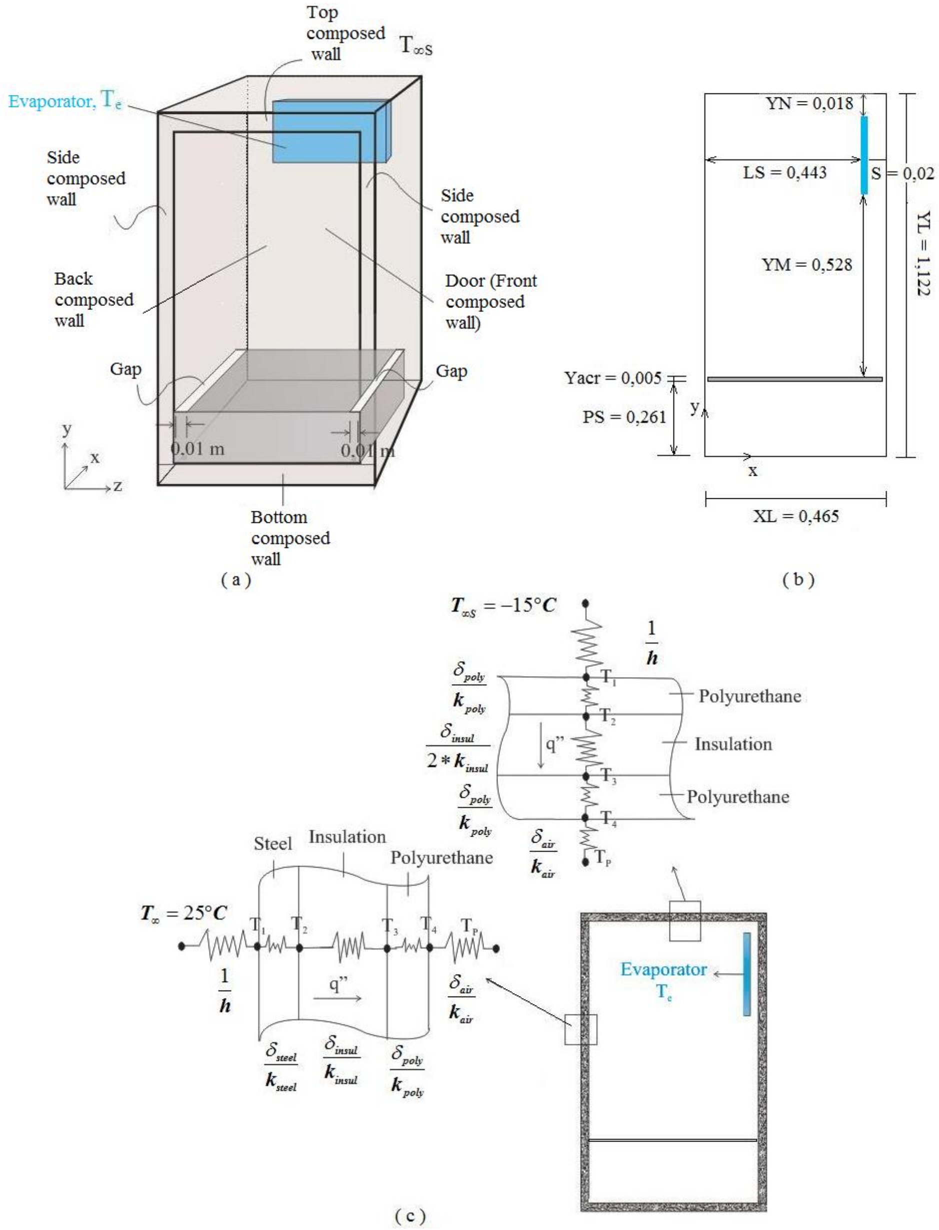

Figure 1. (a) Geometry of the cabinet, (b) dimensions (m) of the cabinet, and (c) Thermal resistances between the external environment and the interior of the cabinet.

The natural convection inside the cabinet is governed by the mass conservation equation, the Navier-Stokes equation, and the energy conservation equation. Using the Boussinesq assumption, the governing equations are given by:

$$\frac{\partial \boldsymbol{u}}{\partial \boldsymbol{x}}+\frac{\partial \boldsymbol{v}}{\partial \boldsymbol{y}}+\frac{\partial \boldsymbol{w}}{\partial \boldsymbol{z}}=0 \tag{1}$$

$$\boldsymbol{u}\frac{\partial \boldsymbol{u}}{\partial \boldsymbol{x}}+\boldsymbol{v}\frac{\partial \boldsymbol{u}}{\partial \boldsymbol{y}}+\boldsymbol{w}\frac{\partial \boldsymbol{u}}{\partial \boldsymbol{z}}=-\frac{1}{\rho}\frac{\partial \boldsymbol{p}}{\partial \boldsymbol{x}}+\nu\left(\frac{\partial^2 \boldsymbol{u}}{\partial \boldsymbol{x}^2}+\frac{\partial^2 \boldsymbol{u}}{\partial \boldsymbol{y}^2}+\frac{\partial^2 \boldsymbol{u}}{\partial \boldsymbol{z}^2}\right) \tag{2}$$

$$\boldsymbol{u}\frac{\partial \boldsymbol{v}}{\partial \boldsymbol{x}}+\boldsymbol{v}\frac{\partial \boldsymbol{v}}{\partial \boldsymbol{y}}+\boldsymbol{w}\frac{\partial \boldsymbol{v}}{\partial \boldsymbol{z}}=-\frac{1}{\rho}\frac{\partial \boldsymbol{p}}{\partial \boldsymbol{y}}+\nu\left(\frac{\partial^2 \boldsymbol{v}}{\partial \boldsymbol{x}^2}+\frac{\partial^2 \boldsymbol{v}}{\partial \boldsymbol{y}^2}+\frac{\partial^2 \boldsymbol{v}}{\partial \boldsymbol{z}^2}\right)-\boldsymbol{g}[1-\beta(\boldsymbol{T}-\boldsymbol{T}_0)] \tag{3}$$

$$\boldsymbol{u}\frac{\partial \boldsymbol{w}}{\partial \boldsymbol{x}}+\boldsymbol{v}\frac{\partial \boldsymbol{w}}{\partial \boldsymbol{y}}+\boldsymbol{w}\frac{\partial \boldsymbol{w}}{\partial \boldsymbol{z}}=-\frac{1}{\rho}\frac{\partial \boldsymbol{p}}{\partial \boldsymbol{z}}+\nu\left(\frac{\partial^2 \boldsymbol{w}}{\partial \boldsymbol{x}^2}+\frac{\partial^2 \boldsymbol{w}}{\partial \boldsymbol{y}^2}+\frac{\partial^2 \boldsymbol{w}}{\partial \boldsymbol{z}^2}\right) \tag{4}$$

$$\boldsymbol{u}\frac{\partial \boldsymbol{T}}{\partial \boldsymbol{x}}+\boldsymbol{v}\frac{\partial \boldsymbol{T}}{\partial \boldsymbol{y}}+\boldsymbol{w}\frac{\partial \boldsymbol{T}}{\partial \boldsymbol{z}}=\alpha\left(\frac{\partial^2 \boldsymbol{T}}{\partial \boldsymbol{x}^2}+\frac{\partial^2 \boldsymbol{T}}{\partial \boldsymbol{y}^2}+\frac{\partial^2 \boldsymbol{T}}{\partial \boldsymbol{z}^2}\right) \tag{5}$$

where $\boldsymbol{u}, \boldsymbol{v}, \boldsymbol{w}$ are the component velocities in the x, y, z directions, respectively, ρ is the reference density, ν is the kinematic viscosity, g is the gravity, β is the thermal expansion coefficient, k is the thermal conductivity, α is the thermal diffusivity, T and T_0 are the local and reference temperatures, respectively.

In order to complete the physical model it is necessary to specify the boundary conditions. As shown in Tab. 1, the non-slip and impermeable boundary conditions for the velocity field are used for all walls. For the energy equation it is used a combination of external heat convection and heat conduction in the composed wall (steel plate, insulation, and polyurethane plate) for all surfaces of the domain. In order to simulate the upper compartment of the refrigerator, the upper external temperature, $T_{\infty s}$, is taken as -15°C. All the other external temperatures are taken as the environmental temperature. The external heat transfer coefficients, h, are chosen to simulate the more appropriate heat transfer condition. The temperature of the evaporator, T_e, is prescribed on the entire surface as a constant value. Tab. 1 and Fig. 2 show all the boundary conditions.

Table 1 - Boundary conditions for the refrigerator cabinet.

Boundary	Temperature	Velocity
Top wall	Convection heat flux, $T_{\infty s}$ = -15°C, h=2 (W m^{-1} K^{-1})	No slip
Left side wall	Convection heat flux, T_∞ = 25°C, h=3 (W m^{-1} K^{-1})	No slip
Right side wall	Convection heat flux, T_∞ = 25°C, h=3 (W m^{-1} K^{-1})	No slip
Back wall	Convection heat flux, T_∞ = 25°C, h=8 (W m^{-1} K^{-1})	No slip
Front wall	Convection heat flux, T_∞ = 25°C, h=3 (W m^{-1} K^{-1})	No slip
Bottom	Convection heat flux, T_∞ = 25°C, h=2 (W m^{-1} K^{-1})	No slip

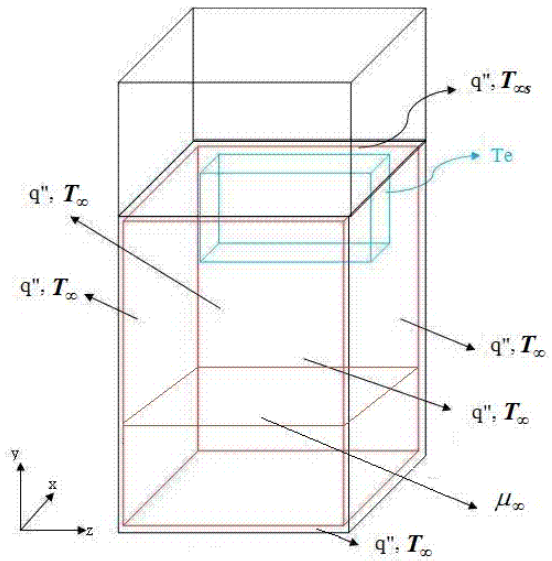

Figure 2. Boundary conditions.

3. SOLUTION METHODOLOGY

The Finite Volume Methodology was chosen as numerical procedure for discretizing the governing equations and the boundary conditions. The SIMPLE-Semi-Implicit Method for Pressure-Linked Equations algorithm applied to a staggered mesh was used for solving the pressure-velocity coupling problem. The Power-Law scheme was employed as interpolation function for the convective-diffusive terms, and the TDMA-Tri-Diagonal Matrix Algorithm was used to solve the systems of algebraic equations.

In order to validate the computational code, the natural convection inside closed cavity with a vertical heated wall and a vertical cold wall is solved and the results are compared with the experimental data obtained by Tian and Karayannis (2000). Fig. 3 depicts the numerical velocity profile in the vertical direction, v, for several vertical positions, y, showing good agreement with the experimental data.

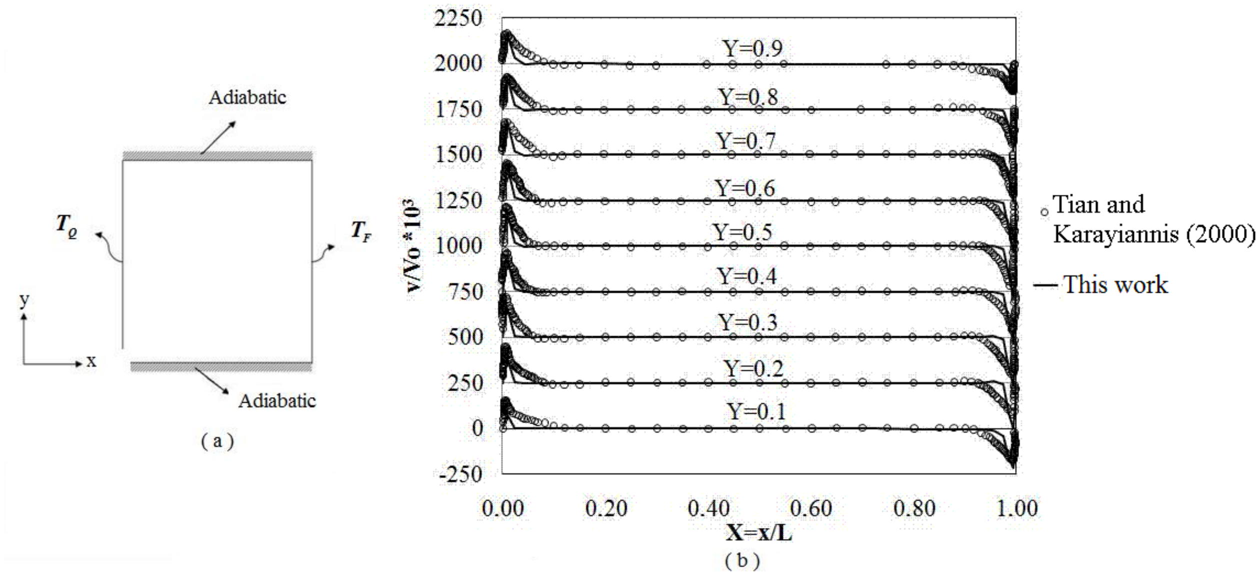

Figure 3 - (a) Schematic of the air filled square cavity and (b) Vertical velocity distribution at different heights.

4. RESULTS AND DISCUTIONS

Three non-uniform meshes containing 20x30x20 (mesh 1); 40x60x40 (mesh 2), and 60x90x60 (mesh 3) points were tested. The results were compared to define the final mesh to be used. As can be seen in Fig. 4, the difference between the results obtained from meshes 2 and 3 is small. Besides, the total amount of heat transferred to the cabinet increased just 0.02 %. However, the computational time increased 5 times (from 2 to 10 days of computation). In order to save computational time, mesh 2 was chosen to generate the results.

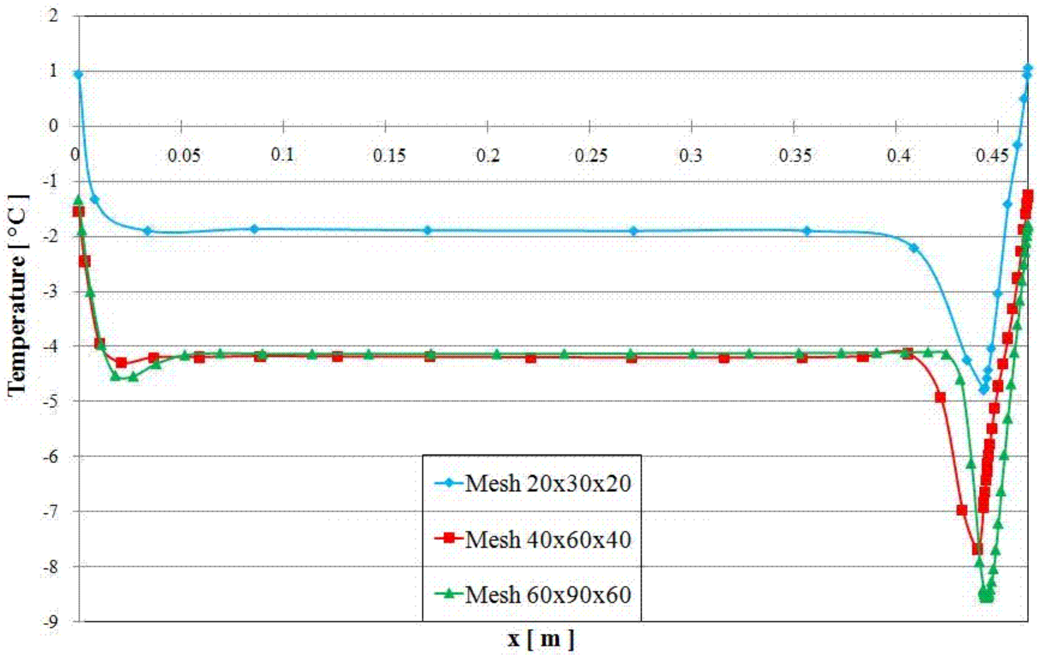

Figure 4 – Temperature distribution for mesh independence test (y = 0.4 m and z = 0.3 m).

After performing the mesh independence test, the influence of the evaporator position inside the cabinet on the temperature distribution was analysed for evaporator temperature, T_e= -15°C. For this temperature, the Rayleigh number is of the order of 10^7, meaning that the flow can be considered as laminar flow. The temperature field is shown in Fig. 5 for nine different positions. The Tab. 2 shows the dimensions for the positioning of the evaporator at z = 0.259m. In order to present detailed results, Fig. 6 shows the temperature profile as a function of the y position, for x = 0.2 m and z = 0.3 m. The horizontal position of the evaporator does not change significantly the temperature field. However, the vertical position plays an important rule on the temperature distribution. The average temperature at the top of the cabinet increases from 2 to 6, and to 12°C as the evaporator movies to the bottom direction, while the average temperature at the bottom of the cabinet, just above the drawer, diminishes from -2 to -4, and to -6°C. Also the temperature inside the drawer decreases from 4 to 2 °C and 0 °C when the evaporator is moved to the bottom.

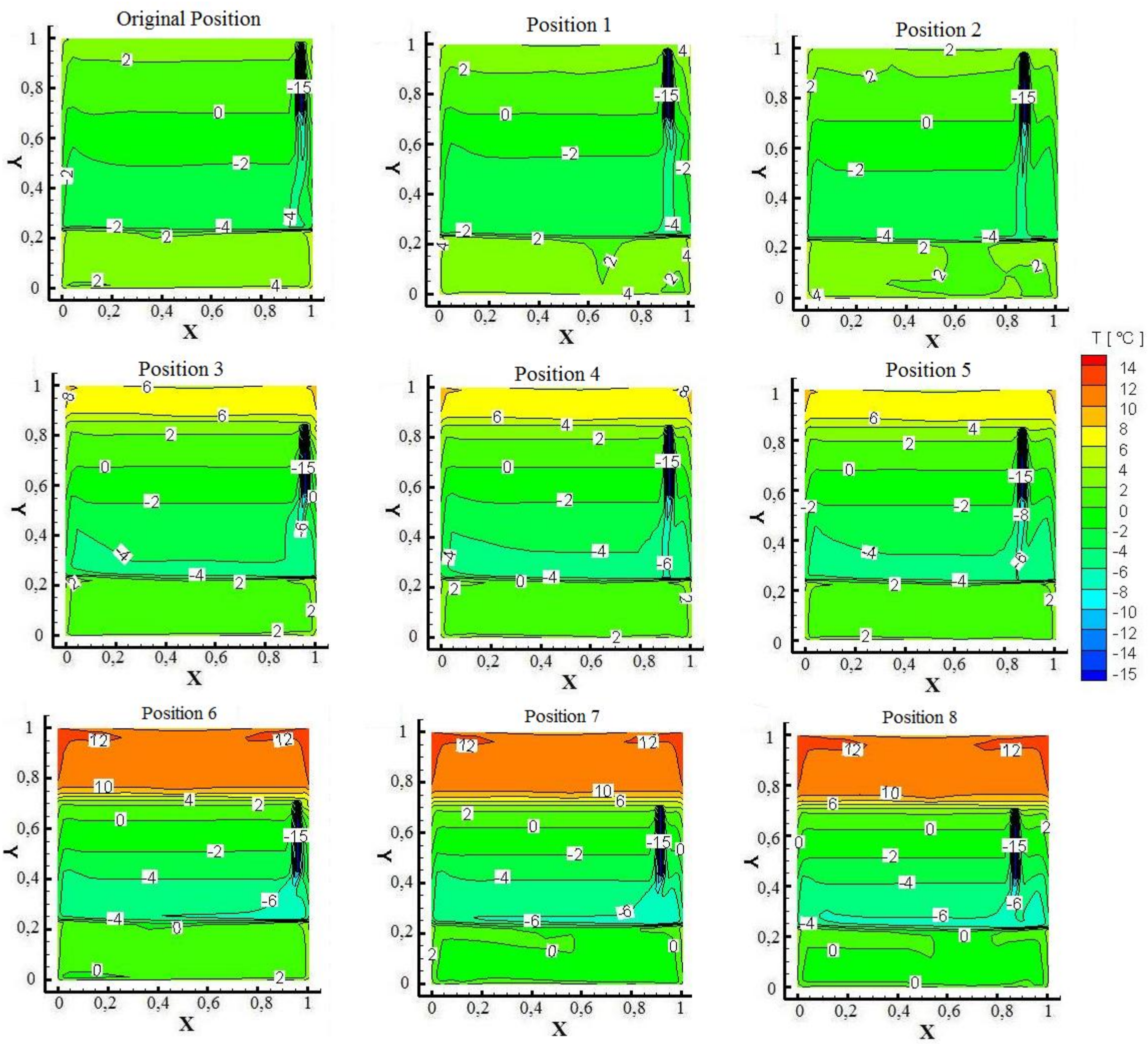

Figure 5. Influence of the nine evaporator positions in the air temperature distribution for Z = 0.5.

Table 2 – Dimensions for the positioning of the evaporator at z = 0.259 m.

Position	LS [m]	YM [m]
Original	0.443	0.528
1	0.423	0.528
2	0.403	0.528
3	0.443	0.373
4	0.423	0.373
5	0.403	0.373
6	0.443	0.218
7	0.423	0.218
8	0.403	0.218

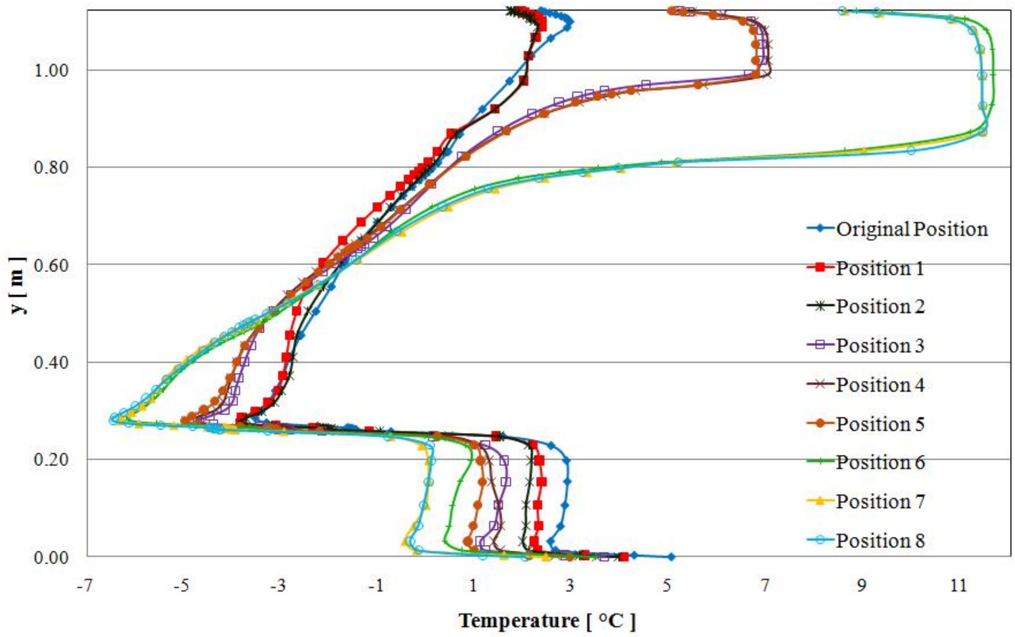

Figure 6: Vertical air temperature profile in the cabinet in various positions for x = 0.2 m and z = 0.3 m.

In addition, Fig. 7 shows that the gap in the lateral drawer improves air circulation inside the cabinet, producing greater cooling of space.

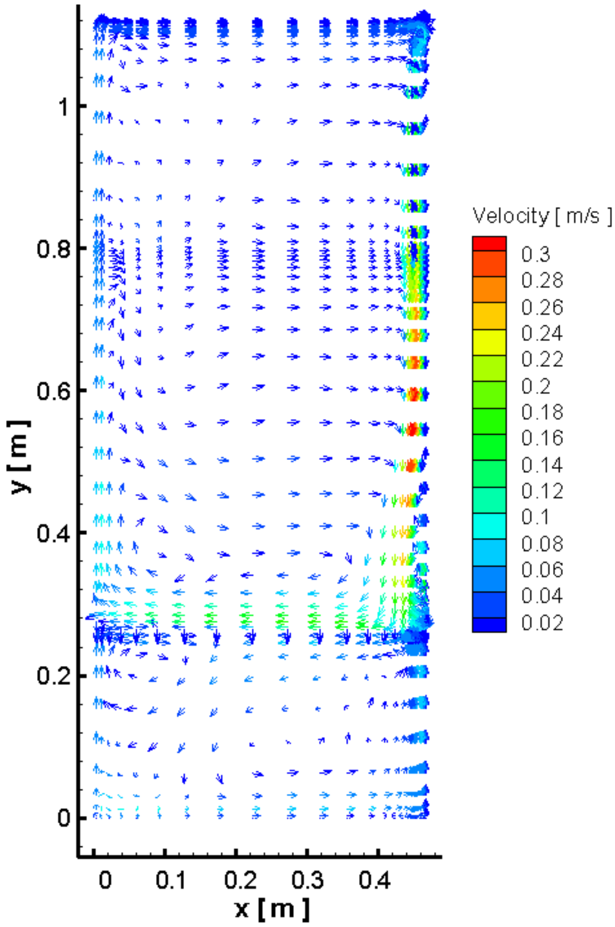

Figure 7: Vector velocity, v, for T_e = - 15 ºC at z = 0.259 m.

The cooling capacity of the evaporator, which represents the total thermal load of the cabinet, was also computed for evaporator temperature varying from -20 to 0 °C. As expected, the thermal load of the cabinet, showed in Fig. 8, decreases for increasing evaporator temperature. In addition, the decrease is linear, which is an unexpected behavior due to the high non-linearity of the problem.

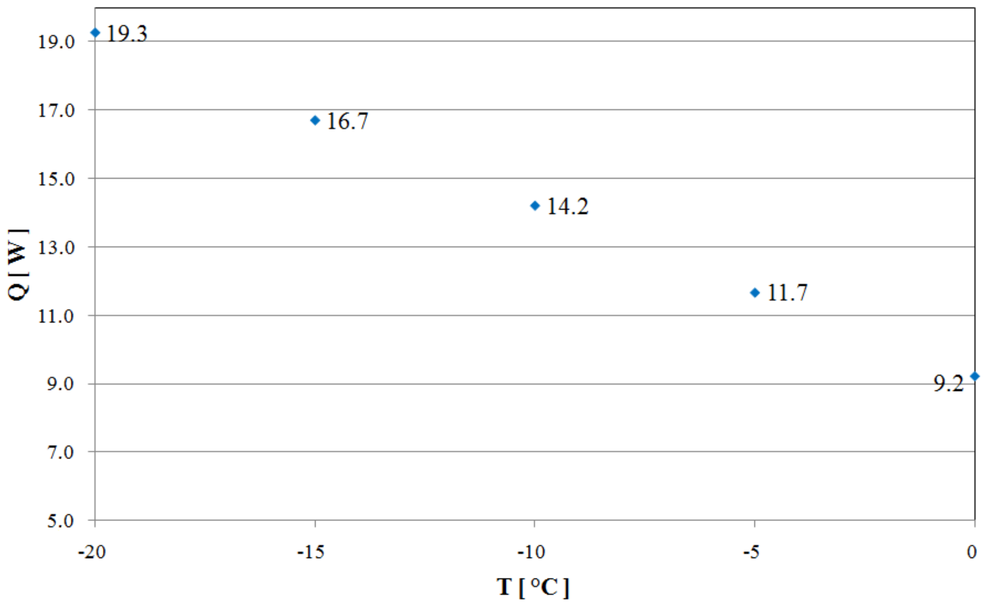

Figure 8 - Cabinet thermal load (evaporator cooling capacity).

5. CONCLUSIONS

In the present study it was developed a Computational Fluid Dynamic model for simulating the flow inside a domestic refrigerator working on natural convection regime. The Finite Volume Methodology was used as numerical procedure for discretizing the governing equation. The cabinet of the refrigerator was considered an empty cavity with one bottom drawer, but without shelves. The influence of the evaporator position on the temperature distribution inside the cabinet was analyzed and the thermal load of the cabinet was estimate for evaporator temperature ranging from -20 to 0 °C. The horizontal position of the evaporator does not change significantly the temperature field. However, the vertical position plays an important rule on the temperature distribution. The results this study shows that the evaporator can be positioned at different heights, depending on the desired temperature distribution within the cabinet. With the lateral gap in the lid drawer the temperature inside it decreases when the evaporator is moved to the bottom. This gap improves the air circulation inside the cabinet, producing greater cooling of space. The thermal load of the cabinet decreases linearly for increasing evaporator temperatures.

6. ACKNOWLEDGEMENTS

This research was supported by resources supplied by the Center for Scientific Computing (NCC/GridUNESP) of the São Paulo State University (UNESP).

7. REFERENCES

Amara, S.B., Laguerre, O., Charrier-Mojtabi, M.C., Lartigue, B. and Flick, D., 2008, "PIV Measurement of the Flow Field in a Domestic Refrigerator Model: Comparison with 3D Simulation", International Journal of Refrigeration, Vol. 31, pp. 1328-1340.

Ding, G.L., Qiao, H.T. and Lu, Z.L., 2004, "Ways to Improve Thermal Uniformity Inside a Refrigerator", Applied Thermal Engineering, Vol. 24, pp. 1827-1840.

Fukuyo, K., Tanaami, T. and Ashida, H., 2003, "Thermal Uniformity and Rapid Cooling Inside Refrigerator", International Journal of Refrigeration, Vol. 26, pp. 249-255.

Gupta, J. K., Gopal, M. R. and Chakraborty, S., 2007, "Modeling of a Domestic Frost-Free Refrigerator", International Journal of Refrigeration, Vol. 30, pp. 311-322.

Hermes, C.J.L. and Melo, C., 2009, "Assessment of the Energy Performance of Household Refrigerators Via Dynamic Simulation", Applied Thermal Engineering, Vol. 29, pp. 1153-1165.

Hermes, C.J.L., Melo, C., Knabben, F.T. and Gonçalves, J.M., 2009, "Prediction of the Energy Consumption of Household Refrigerators and Freezers Via Steady-State Simulation", Applied Energy, Vol. 86, pp. 1311-1319.

James, S.J., Evans, J. and James, C., 2008, "A Review of the Performance of Domestic Refrigerators", Journal of Food Engineering, Vol. 87, pp. 2-10.

Laguerre, O., Amara, S.B. and Flick, D., 2005, "Experimental Study of Heat Transfer by Natural Convection in a Closed Cavity: Application in a Domestic Refrigerator", Journal of Food Engineering, Vol. 70, pp. 523-537.

Laguerre, O., Amara, S.B., Charrier-Mojtabi, M.C., Lartigue, B. and Flick, D., 2008, "Experimental Study of Air Flow by Natural Convection in a Closed Cavity: Application in a Domestic Refrigerator", Journal of Food Engineering, Vol. 85, pp. 547-560.

Laguerre, O., Amara, S.B., Moureh, J. and Flick, D., 2007, "Numerical Simulation of Air Flow and Heat Transfer in Domestic Refrigerators", Journal of Food Engineering, Vol. 81, pp. 144-156.

Laguerre, O., Derens, E. and Palagos, B., 2002, "Study of Domestic Refrigerator Temperature and Analysis of Factors Affecticng Temperature: a French Survey", International Journal of Refrigeration, Vol. 25, pp. 653-659.

Tian, Y.S. and Karayannis, T.G., 2000, "Low Turbulence Natural Convection in an Air Filled Square Cavity. Part I: The Thermal and Fluid Flow Fields", International Journal of Heat and Mass Transfer, Vol. 43, pp. 849-866.

8. RESPONSIBILITY NOTICE

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