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NUMERICAL SIMULATION OF STATIC DOMESTIC REFRIGERATORS

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Abstract. *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. 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 was to develop a computational fluid dynamics (CFD) model for domestic refrigerators working on natural convection regime. The Finite Volume Methodology was chosen as numerical procedure for discretizing the governing equations. 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. The model was applied to regular static refrigerator, where the cabinet was considered an empty three-dimensional rectangular cavity without shelves and drawers. 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. The cooling capacity of the evaporator was also computed for each temperature investigated.*

Keywords*: Natural Convection, Temperature Distribution, Static Domestic Refrigerators.*

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 life of people, 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 behavior 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 2cm. 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 pointed out 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) e Hermes e 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 solve numerically the air flow and heat transfer inside a static refrigerator in order to analyze the temperature and velocity distribution.

2. MATHEMATICAL MODEL

2.1. Physical model

A 350 liters commercial refrigerator working on natural convection (static refrigerator) was modeled as an empty three-dimensional rectangular cavity without shelves and drawers, as shown in Fig. 1. Figure 1a depicts the geometry of the cabinet and Fig. 1b shows a typical wall of the refrigerator, which is typically composed by a thin steel plate, insulation, and a thin polyurethane plate. It is also shown in the same figure the thermal resistance considered to specify the boundary conditions.

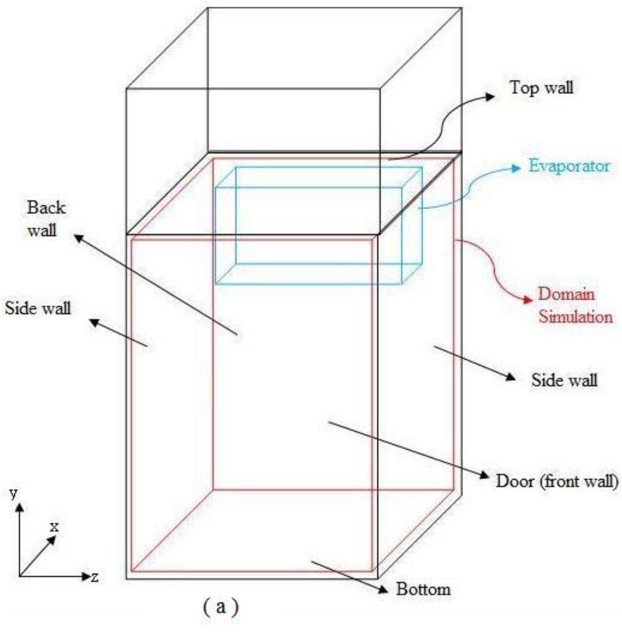


Figure 1. (a) Geometry of the cabinet and (b) Thermal resistances between the external environment and the interior of the cabinet.

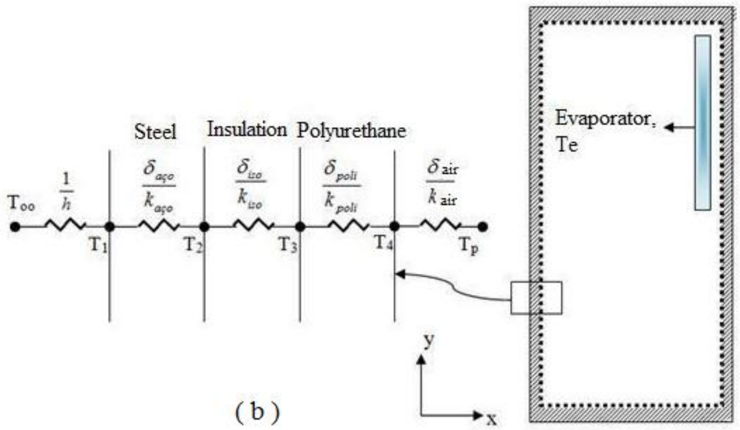


Figure 1. (a) Geometry of the cabinet and (b) Thermal resistances between the external environment and the interior of the cabinet (continuation).

The problem 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:

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

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{1}{\rho_0}\frac{\partial p}{\partial x}+\nu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \tag{2}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{1}{\rho_0}\frac{\partial p}{\partial y}+\nu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)+g\beta(T-T_0) \tag{3}$$

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=-\frac{1}{\rho_0}\frac{\partial p}{\partial z}+\nu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \tag{4}$$

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

where u,v,w are the component velocities in the x, y, z directions, respectively, ρ and ρ_0 are the local and reference densities, respectively, 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_∞, 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. Figure 2 shows schematically all the boundary conditions.

Table 1. Boundary conditions for refrigerator compartment.

Boundary	Temperature	Velocity
Top wall	Convection heat flux, T_∞ = -15°C, h=0.005 (W m^{-1} K^{-1})	No slip
Left side wall	Convection heat flux, T_∞ = 25°C, h=5.0 (W m^{-1} K^{-1})	No slip
Right side wall	Convection heat flux, T_∞ = 25°C, h=5.0 (W m^{-1} K^{-1})	No slip
Back wall	Convection heat flux, T_∞ = 25°C, h=5.0 (W m^{-1} K^{-1})	No slip
Front wall	Convection heat flux, T_∞ = 25°C, h=5.0 (W m^{-1} K^{-1})	No slip
Bottom	Insulation, h=0 (W m^{-1} K^{-1})	No slip

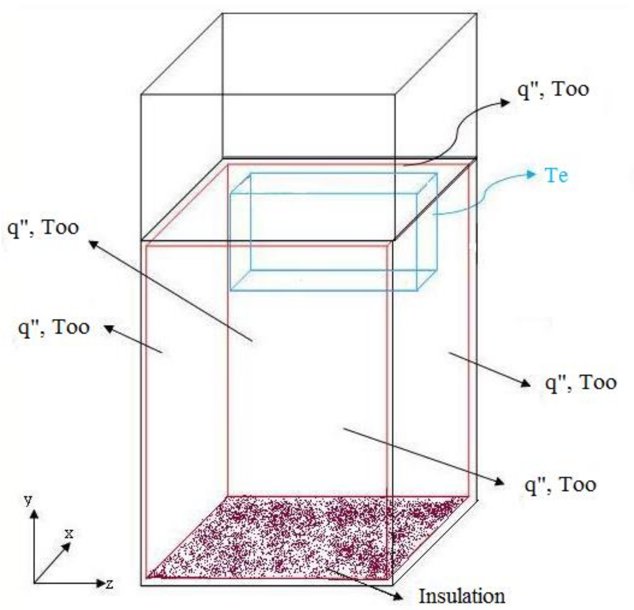
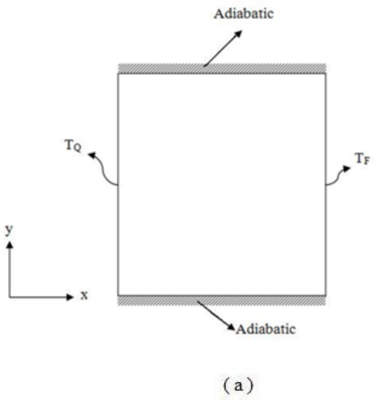


Figure 2. Boundary conditions.

2.2. 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. Three non-uniform meshes containing 20x30x20; 40x60x40, and 60x90x60 points were tested. The total amount of heat transferred to the cabinet increased just 0,2% as the mesh was refined from 40x60x40 to 60x90x60 points, but the computational time increased 6 times, from 2 to 12 days of computation. In order to save computational time, the 40x60x40 points non-uniform mesh, refined near the walls and the evaporator, was chosen to solve the problem.

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). Figure 3 depicts the numerical velocity profile in the vertical direction, v, for several vertical positions, y, showing good agreement with the experimental data. As can be seen in the Fig. 3b, the thickness of the experimental hydrodynamics boundary layer is just slightly larger than the numerical results. Figure 4 shows similar comparison for the temperature distribution, where one can also observe a good agreement between the results.



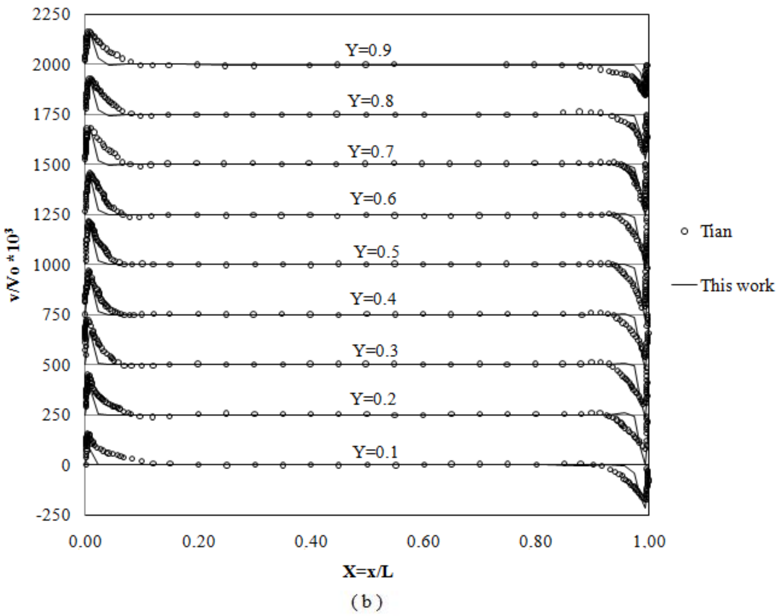


Figure 3. (a) Schematic of the air filled square cavity and (b) Vertical velocity distribution at different heights.

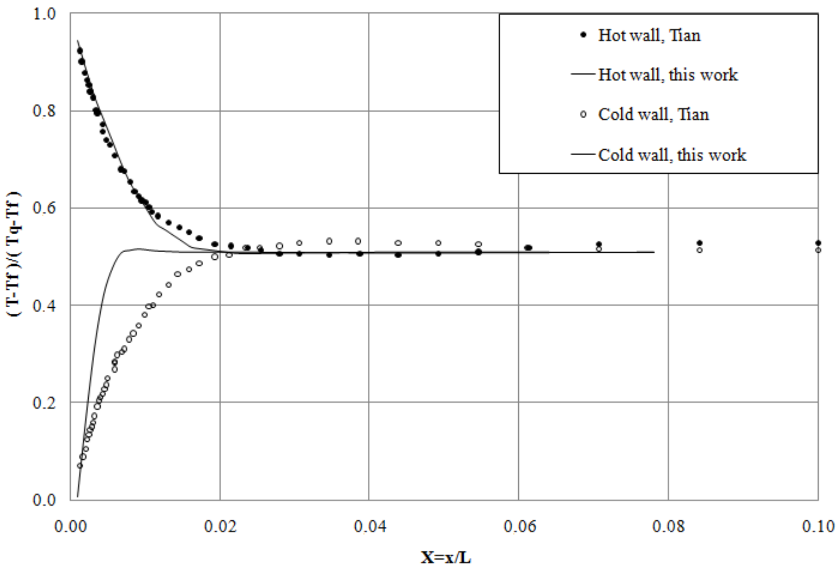


Figure 4. Comparison of the temperature distribution near the walls at Y=0.5.

2. RESULTS AND DISCUSSIONS

In order to analyze the velocity and temperature fields of the air flow inside the cabinet the evaporator temperature, T_e, was varied from -20°C to 0°C. The Rayleigh number for the evaporator surface is of the order of 10^7. Therefore, considering only the evaporator surface the flow can be considered as laminar flow, because the Rayleigh number for the transition is of the order of 10^8. Figure 5 depicts the steady state temperature distribution in the y direction in the plane X = 0.58 and Z = 0.27 for both evaporator temperatures. As expected, one can observe that the larger the evaporator temperature the larger the air average temperature inside the cabinet. In addition, the lower temperatures are noticed in the bottom region of the cabinet. As the evaporator temperature is reduced from 0 to -20°C, the lower air temperature reduces about 17°C.

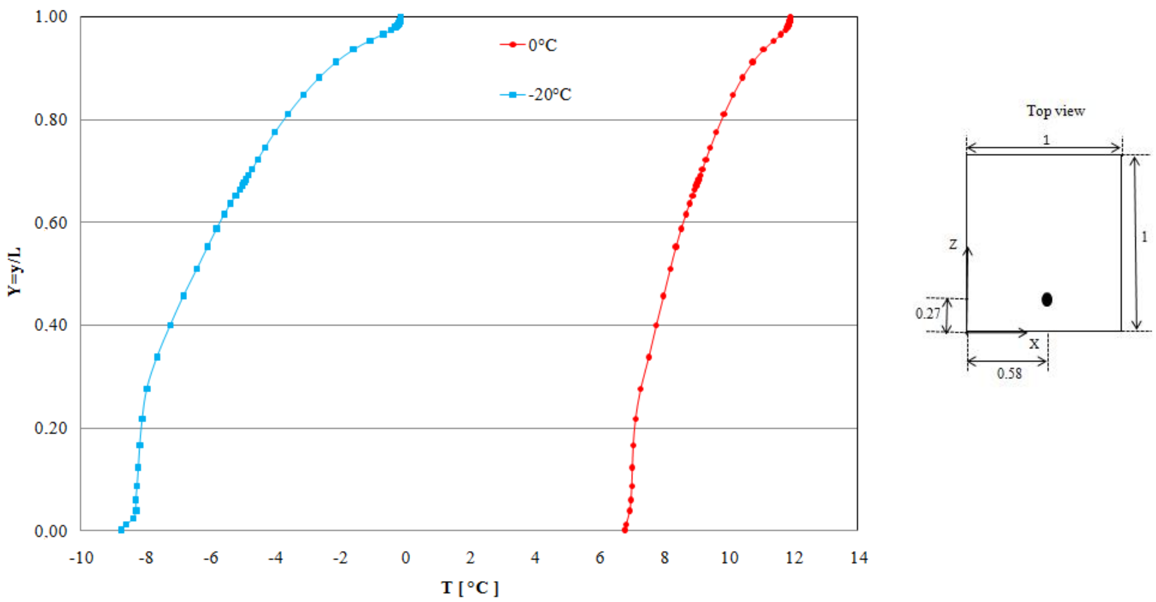


Figure 5. Temperature distribution in the y direction in the plane X = 0.58 and Z = 0.27 for T_e = 0°C and T_e = -20°C.

Figure 6 presents temperature and velocity fields for T_e = -15°C, showing the lowest value (-4°C) at the bottom and the highest value (4°C) at the top, as expected. The velocity field is shown in Fig. 6b, where one can notice that the velocity is practically zero at the central region of the cabinet. Moreover, there is a high velocity (~0.2 m/s) just below the evaporator, where the cold air flows downward to the bottom of the cabinet. In the opposite side, the air velocity is in the opposite direction, reaching about 0.1 m/s.

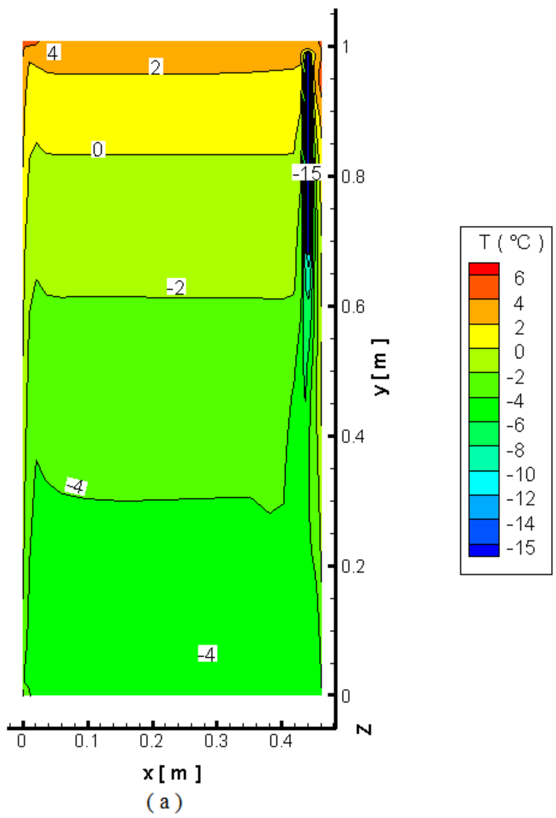
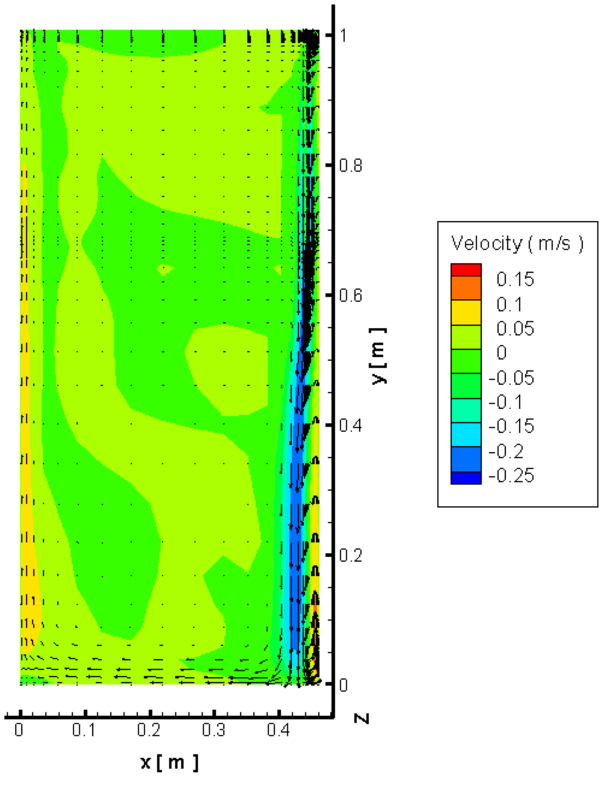


Figure 6. (a) Temperature and (b) velocity fields for Te = -15ºC for z = 0.259 m.



(b)

Figure 6. (a) Temperature and (b) velocity fields for Te = -15ºC for z = 0.259 m (continuation).

In order to present detailed results, Fig. 7 shows the temperature distribution for T_e = -15°C in the *x* direction, which measures the distance from the front (where is the door of the refrigerator) to the cabinet back (where is the evaporator) for different heights (distance *y* from bottom to top of the cabinet). As expected, for Y = 0.91 (in the evaporator region) it is observed the lowest temperature of the flow near the evaporator. However, as Y decreases to 0.51 and 0.12 the temperature near the evaporator region increases, while the temperature far from the evaporator decreases due to the recirculation of the flow.

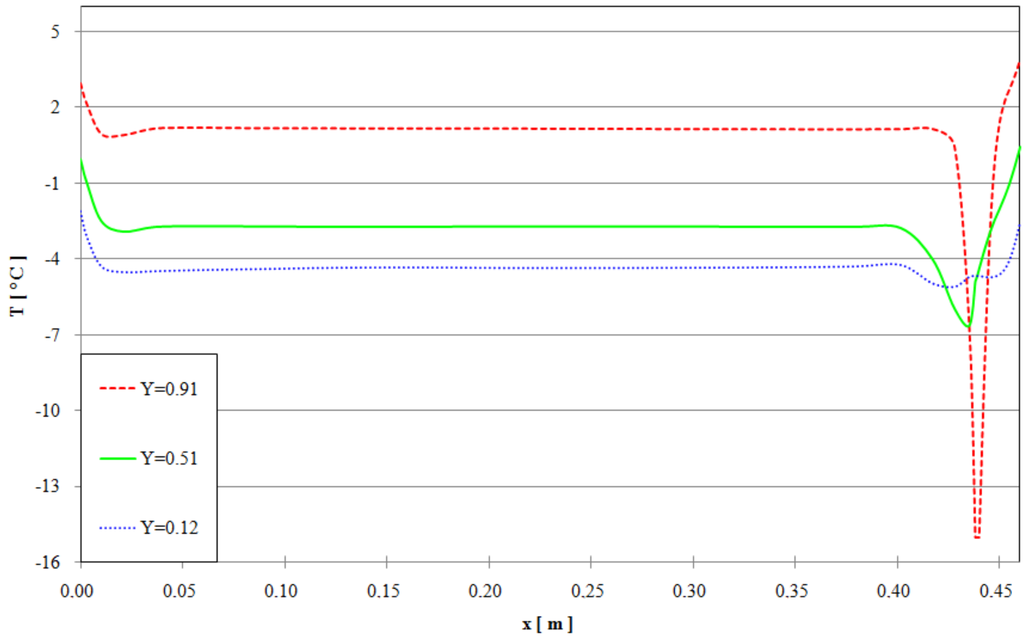


Figure 7: Temperature distribution for T_e = -15°C in the direction at z = 0.582 m.

Figure 8 shows the vertical velocity profile, v, in order to show the hydrodynamic boundary layer thickness. As can be seen, the lower the distance from the bottom the larger the boundary layer thickness. In addition, as pointed out before, the higher velocities are observed in the evaporator side, at the bottom region of the cabinet.

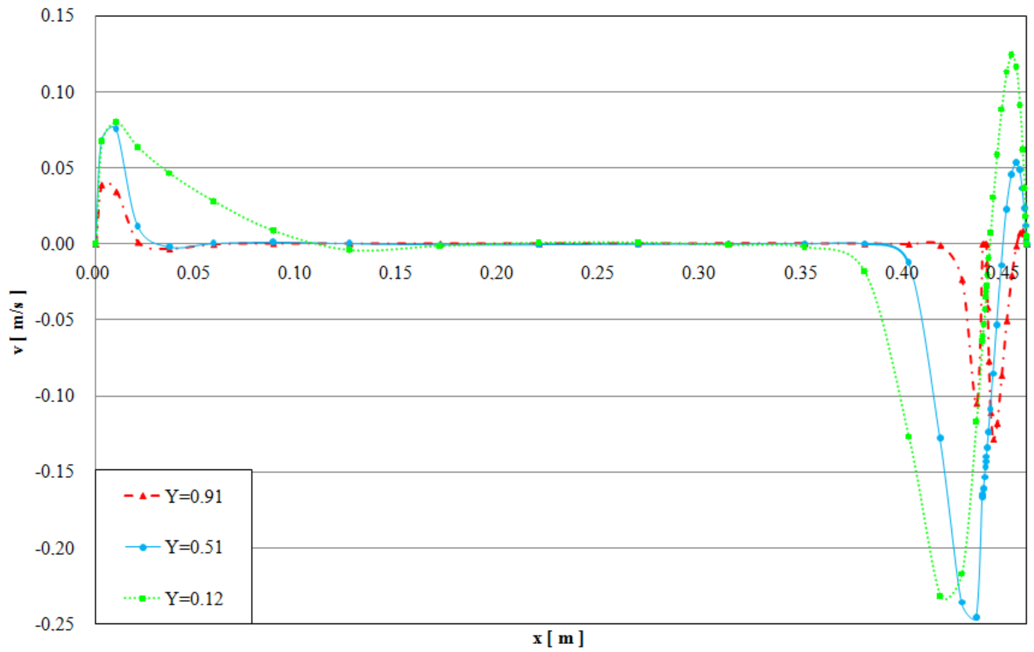


Figure 8: Velocity distribution for T_e = -15°C at z = 0.582 m.

The cooling capacity of the evaporator was also computed for each evaporator temperature. As expected, the evaporator cooling capacity showed in Tab. 2 decreases for increasing evaporator temperature. In addition, for each 5°C of temperature increment, the cooling capacity decreasing is not constant, varying from 13% for the first increment (-20°C to -15°C) to 21% for the last increment (-5°C to 0°C).

Table 2. Evaporator cooling capacity.

Evaporator Temperature (°C)	Cooling Capacity (W)
-20	17.8
-15	15.7
-10	13.6
-5	11.5
0	9.5

3. CONCLUSION

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 three-dimensional rectangular cavity without shelves and drawers. The actual thermal resistance between the external environment and the interior of the cabinet was taken into account.

In order to analyze the velocity and temperature fields of the air flow inside the cabinet, the evaporator temperature was varied from -20°C to 0°C. As conclusion, besides the velocity and temperature fields give important information to consumers about the correct zones to locate the products, the results show that the evaporator cooling capacity decreases only 15% when the evaporator temperature decreases from -15 to -10°C. As result, the refrigeration system could be designed for a larger evaporator pressure, which reduces the compressor energy consumption, without losing significant cooling capacity.

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