24th ABCM International Congress of Mechanical Engineering
December 3-8, 2017, Curitiba, PR, Brazil

COBEM-2017-2893
PERFORMANCE OF A DOMESTIC WINERY ACCORDING TO THE INTERNAL LOCALIZATION OF THE REFRIGERATION SYSTEM

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Abstract. *Refrigeration and air conditioning systems is a recurring theme in academia and industry. In this sense, twelve combinations of layout, fan velocity and diameter as well as initial temperature, were studied for a small refrigerator employed as a domestic wine cellar. The chamber utilizes a forced convection fan installed in the ceiling and a thermoelectric plate with a coupled fan, installed on the back wall. Simulations were done on a commercial code that works with finite element method. Results obtained led to the conclusion that, for this type and size of refrigeration chamber, the best positioning of the ventilating units and thermoelectric cell is the center of the inner walls, respectively upper and posterior. The simulations showed that the increase of fan velocity, even as the increase of its diameter, did not promote better refrigeration. Contrarily to the mainstream found in the literature, the positioning of the cold source higher also did not promote better results. The friction of the wind-jet with the walls seems to have promoted unwanted heating. New simulations with other layouts are recommended in order to increase the accuracy of the conclusions.*

Keywords: *Refrigeration, Forced Convection*

1 INTRODUCTION

Improving refrigeration and air conditioning systems is a recurring theme in the academic and industrial medium. Typically, the aim is to reduce the energy consumption with a greater capacity of refrigeration, added to the recent environmental restrictions, which includes less noise and the non-liberation of polluting gases that affect the ozone layer.

Tizzei et al (2011) presented a study about the strategies of temperature control in refrigerated chambers with the utilization of the fuzzy set theory. They emphasized that the system of temperature control has to be adequate in order to diminish the energy consumption and to obtain a better conservation of food, medicines, etc. According to them, large ranges of temperature change induce to the faster deterioration of products.

The control strategy like on-off type, normally used in conventional refrigeration systems, causes premature failure of the compressor, high consumption of electricity, and permanent unwanted oscillations in temperature inside the storage chamber (Buzelin, 2003). Thus, temperature control by varying the compressor speed has emerged as a viable alternative for the reduction in energy consumption, preservation of equipment, and proper maintenance of the temperature inside the storage chamber. Several scientific studies point to the benefits of this type of control, such as that performed by Hua et al. (2009), which highlights the increase in system performance and reduction of electricity expending.

In terms of work to generate cold, domestic refrigeration cycles usually employ vapor compressors, but many others unities of cold generation can be utilized. Some investigations have been developed considering thermoelectric systems, as those presented by Rocha (2009) and Silva and Araújo (2016).

Thermoelectric phenomena are known since the ends of the nineteenth century, although, its application for refrigeration purposes only appeared in the nineteen fifteenth decade, when it was done quick advances in materials for the Peltier cooling, notably, because of the introduction of semiconductor thermocouples (Rowe, 1995). In that time, there was a general opinion that this kind of refrigeration could make superfluous all the others technologies. Afterwards, as the progress seemed slow, doubts have arisen about the possibility of the thermoelectric cooling have any real employability and, in fact, most equipment disappeared from the market.

In terms of cooling source localization, Rocha (2009) studied the efficiency of a domestic refrigerator per the position of the thermoelectric unity. With the aid of a commercial simulator, he compared several layouts, under the optics of the natural and forced convection (with or without forcing fan). His conclusions showed that the best position for the thermoelectric plate was the center of the ceiling of the chamber, with a forcing fan coupled to it in order to promote the forced convection.

Silva and Araújo (2016) investigated the performance of a domestic winery with a thermoelectric plate positioned on the middle of the back wall with a fan also coupled on it. Additionally, they installed another fan on the ceiling to promote forced convection. Such a layout has shown that the zone best refrigerated was the inferior quadrant, at the deep of the chamber. Notwithstanding, the authors recommended future investigations with the repositioning of the cooling and vents unities.

Considering any refrigerated enclosure as having analogous behavior to a frigorific chamber, Rossi et all (2012) studied the distribution of cooling sources in a restaurant. Their conclusion was that increasing the quantity of diffusers results their dimensions diminish in order to keep constant the total insufflation area and the best condition of thermal comfort.

Ferreira and Santos (2016) developed similar work and concluded that for a conventional restaurant, the best localization for diffusers is the ceiling above the area where costumers have lunch, as well as the best area for return grilles is also in the top wall, but in the deep of the enclosure, nearby the food ramp. Authors conclusions were dependent of the facilities distribution, inside the restaurant.

2 MATERIALS AND METHODS

The proposal of this work is to evaluate the thermal behavior of a domestic winery, in a tropical environment, according to the positioning of a thermoelectric unit with a coupled fan and another independent fan, used to force convection, via simulation by using a commercial code that works with finite element method. Two Layouts were tested, according to Tab. 1: one with the forcing fan at the center of the ceiling and the thermoelectric unit at the center of the back wall and another with the forcing fan nearby the door of the chamber and the thermoelectric unit at the top of the back wall. Fan diameter and velocity as well as initial temperature were varied too, resulting in twelve possibilities.

THE PROTOTYPE

A prototype was constructed to collect experimental values (although these values will be not utilized herein but in future works). The structure was made of MDF (Medium Density Fiberboard), plus insulation of EPS (Expanded Polystyrene) and aluminum sheet as finishing. Fig. 1 shows walls composition with dimensions. Fig. 2 shows the intended disposition of the wine bottles inside the chamber and main inner dimensions. Fig. 3 shows prototype construction details.

PHYSICAL PROPERTIES

Thermal conductivities k_n of the walls materials are available in Tab. 1. The gas inside and outside the chamber was the air with the outer convection coefficient he=0,785 W/m^{2}K for a natural process and the inner, hi=2,7 W/m^{2}K, relative to forced convection. Density ρ was 1,225 kg/m^3, specific heat c_p was 1006,43 J/kgK and dynamic viscosity μ was 17,89 x 10^6 Kg/ms.

INITIAL AND BOUNDARY CONDITIONS

Fans speeds (the forcing fan and the fan coupled to the thermoelectric unit), as well as their diameter, are exposed in Tab. 2. In terms of initial conditions inside the chamber, three initial temperatures were considered, in order to investigate its influence over the temperature behavior. Always, the thermoelectric plate temperature was 269 K and the external temperature was 297 K. Tab. 2 shows twelve conditions checked.

Table 1 – Thermal Conductivity of the materials that compound the wall of the chamber.

Material	**Thermal Conductivity k_n (W/mK)**
MDF	0,14
EPS	0,037
Aluminum	204

Table 2 – Constructive characteristics, fan speeds and initial conditions of temperature of the domestic winery.

Condition	Fan Diameter (mm)	Fan speed (m/s)	Position Fan at back wall	Position Fan at the top	Initial inner Temperature (K)
1	80	2,99	middle	middle	269
2	80	2,99	top	nearby the door	269
3	120	4,80	middle	middle	269
4	120	4,80	top	nearby the door	269
5	80	2,99	middle	middle	283
6	80	2,99	top	nearby the door	283
7	120	4,80	middle	middle	283
8	120	4,80	top	nearby the door	283
9	80	2,99	middle	middle	297
10	80	2,99	top	nearby the door	297
11	120 mm	4,80	middle	middle	297
12	120 mm	4,80	top	nearby the door	297

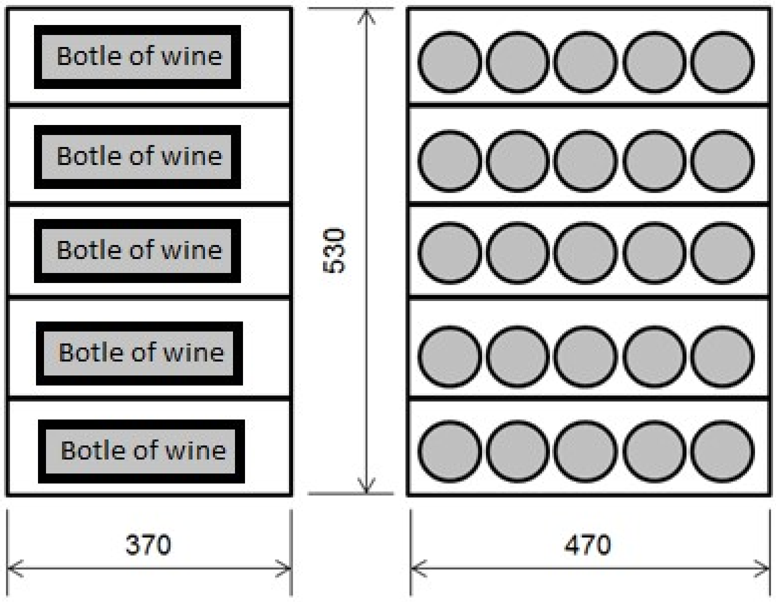

Figure 1. Sketch not scaled of the domestic winery with the intended disposition of the wine bottles and main inner dimensions (mm).

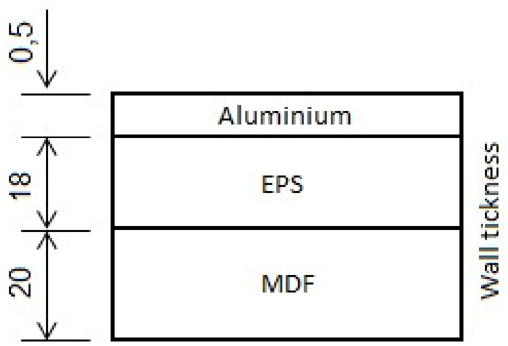

Figure 2. Sketch not scaled of the wall thickness, its composition and main dimensions (mm).

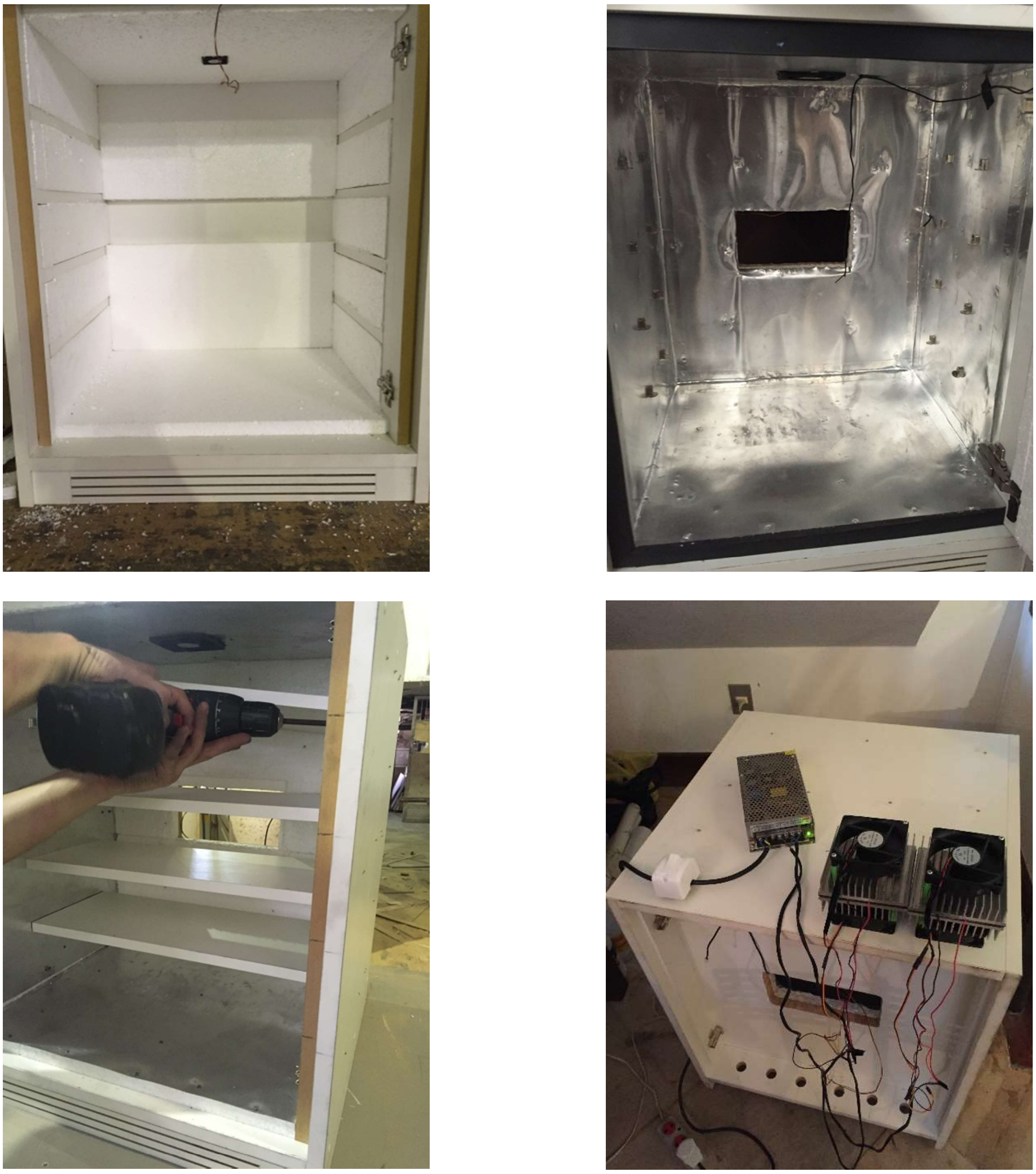

Figure 3. Constructive details. At the top left, the chamber with the inner insulation. At the top right, the finishing in aluminum. At the bottom left, shelves assembly and at the bottom right, the back of the chamber with thermoelectric plates to be installed.

EQUATIONS GOVERNING THE PROBLEM

Considering v_x and v_y velocities, respectively in x and y directions, for a two dimensions Cartesian approach, according to the plans shown in Fig. 4, mass conservation equation form was Eq. (1). Premises for momentum Eq. (2) and (3) were: the fluid was assumed to be Newtonian with constant physical properties. For energy Eqs. (4) and (5), assumptions were: constant physical properties, ρ and k, and no energy generation.

Mass Conservation:

$$\frac{\partial v_x}{\partial x} + \frac{\partial v_y}{\partial y} = 0 \tag{1}$$

Momentum equations:

$$\rho\left(\frac{\partial v_x}{\partial t} + v_x \frac{\partial v_x}{\partial x} + v_y \frac{\partial v_x}{\partial y}\right) = \rho g_x - \frac{\partial P}{\partial x} + \mu\left(\frac{\partial^2 v_x}{\partial x^2} + \frac{\partial^2 v_x}{\partial y^2}\right) \tag{2}$$

$$\rho\left(\frac{\partial v_y}{\partial t}+v_x\frac{\partial v_y}{\partial x}+v_y\frac{\partial v_y}{\partial y}\right)=\rho g_y-\frac{\partial P}{\partial y}+\mu\left(\frac{\partial^2 v_y}{\partial x^2}+\frac{\partial^2 v_y}{\partial y^2}\right) \tag{3}$$

Energy Equation:

$$\rho c_p\left(\frac{\partial T}{\partial t}+v_x\frac{\partial T}{\partial x}+v_y\frac{\partial T}{\partial y}\right)=k_{fluid}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)+\mu\Phi \tag{4}$$

Where,

$$\Phi=2\left[\left(\frac{\partial v_x}{\partial x}\right)^2+\left(\frac{\partial v_y}{\partial y}\right)^2\right]+\left(\frac{\partial v_x}{\partial y}+\frac{\partial v_y}{\partial x}\right)^2 \tag{5}$$

These equations were implemented in finite element method by a commercial code. The software was inducted to generate the refined mesh as evinced in Fig 5. All the work of convergence was done by this package, and results are shown in the next section.

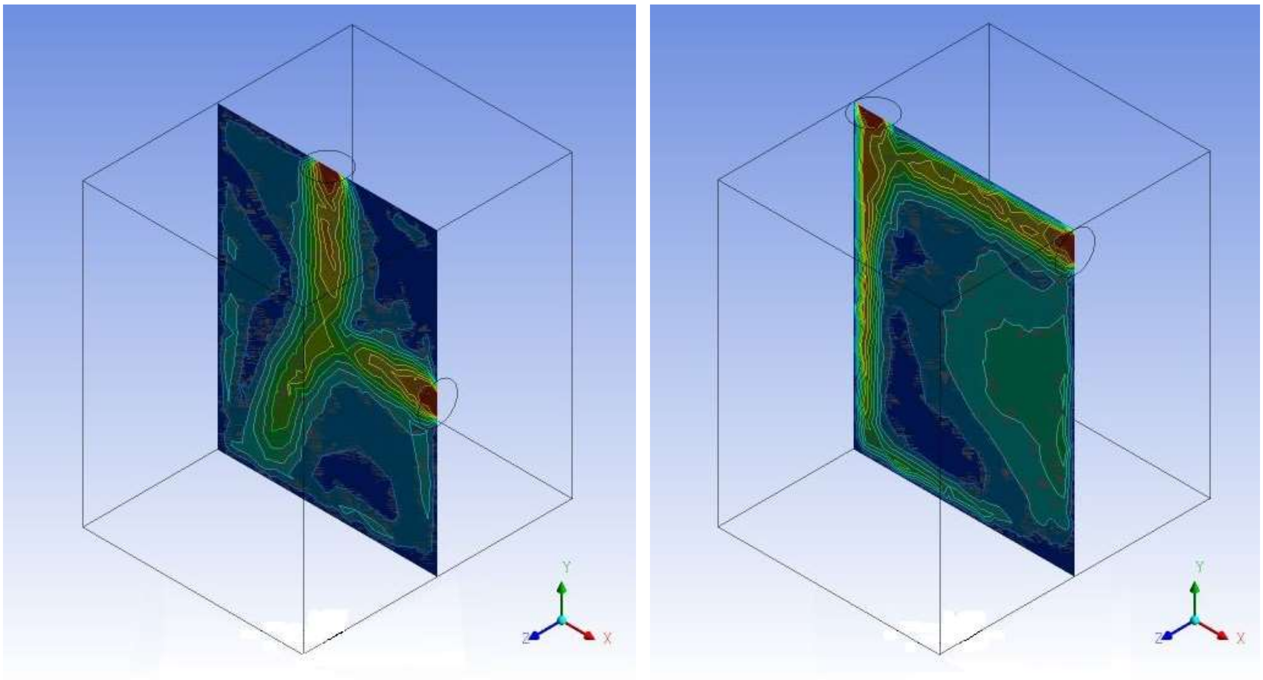

Figure 4. Middle plans where the analysis of velocity field and temperature behavior were performed. At left, fans at the middle positions and at right, fans positioned at the extremes of the plan.

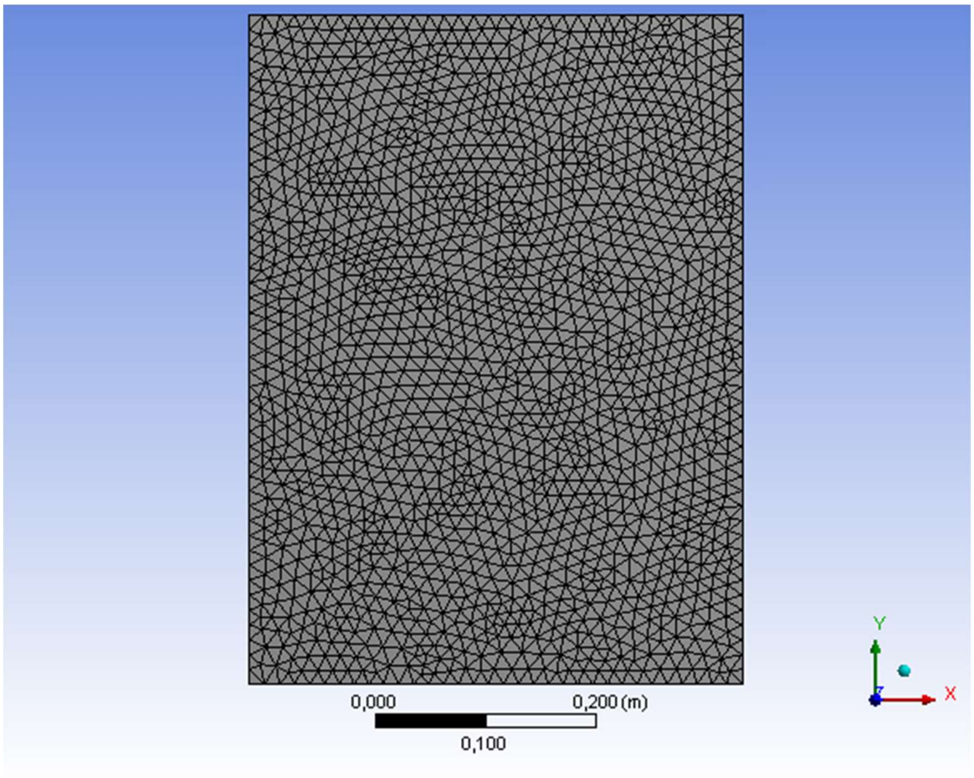

Figure 5. Refined not structured mesh, with 135671 elements and 201085 nodes, that the commercial code developed to treat the case.

3 RESULTS AND DISCUSSION

Velocity fields for fan speeds of 2,99 m/s and 4,80 m/s, with them positioned as explained in Fig. 4, are shown in Fig. 6 to 9.

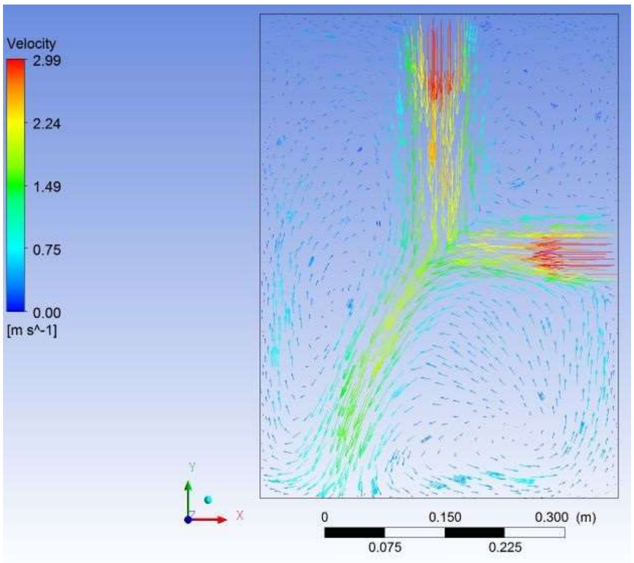

Figure 6. Velocity field in the vertical middle plan of the chamber, with the forcing fan at the middle of the ceiling and the thermoelectric unity, at the middle of the back wall. Fans diameter 80 mm, with speed of 2,99 m/s.

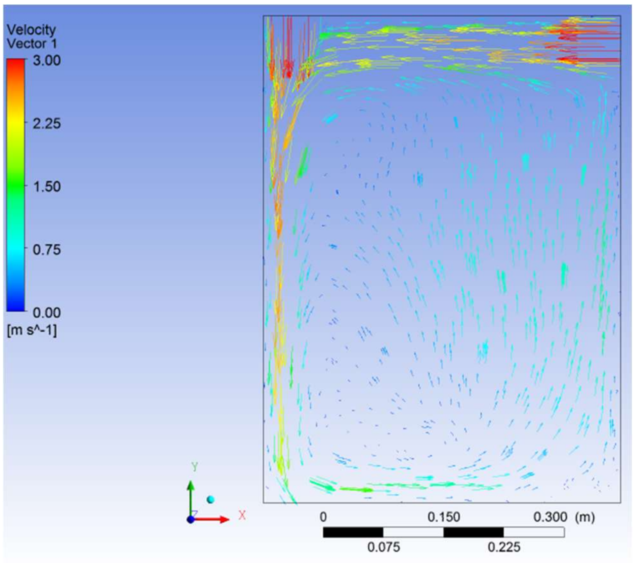

Figure 7. Velocity field in the vertical middle plan of the chamber, with the forcing fan nearby the door and the thermoelectric unity at the top of the back wall. Fans diameter 80 mm, with speed of 2,99 m/s.

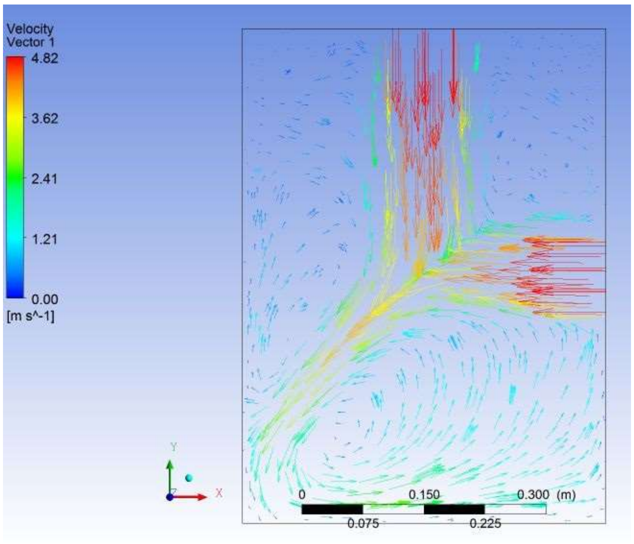

Figure 8. Velocity field in the vertical middle plan of the chamber, with the forcing fan at the middle of the ceiling and the thermoelectric unity, at the middle of the back wall. Fans diameter 120 mm, with speed of 4,80 m/s.

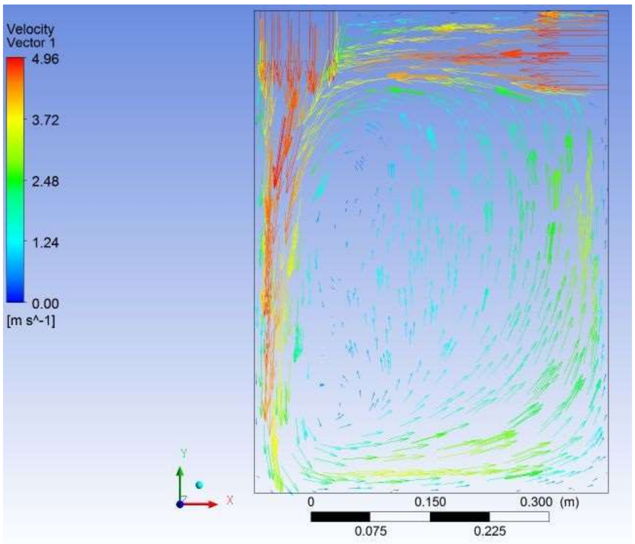

Figure 9. Velocity field in the vertical middle plan of the chamber, with the forcing fan nearby the door and the thermoelectric unity at the top of the back wall. Fans diameter 120 mm, with speed of 4,80 m/s

Velocity field is highly influenced by the layout of the fans and their speeds. Figs. 6 to 9 are self-explanatory. As evinced by Silva and Araújo (2016), fans positioned at the middle of the walls confines the area to be refrigerated at the right bottom quadrant. Three vortices can be observed in this configuration. The speed of the fan units greatly influences the intensity of the spirals. Instead, if fans are positioned at the extreme of each wall, according to figures at right, the refrigerated area will be larger. There will be a single vortex, which in theory seems to improve the performance of the winery.

However, the velocity field analysis needs to be completed with the thermal behavior analysis, to clarifying which layout is more adequate.

Thermal behaviors according to each condition shown in each line of the Tab. 2 are exposed in Figs. 10 to 21.

A primary analysis of the shapes of all figures showing temperature gradients lead to the supposed easy conclusion that the best performance is that shown in Fig. 12. However, the initial temperature employed in the simulation process greatly influences results. In all Three cases of initial temperature, the positioning of the forcer fan at the middle of the ceiling and the thermoelectric unity at the middle of the back wall, have presented the best results, i.e. the lowest temperature in a greater area of the studied plan.

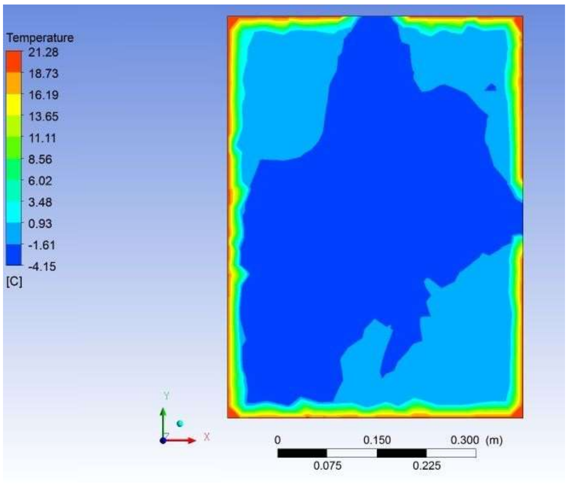

Figure 10. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan at the middle of the ceiling and the Thermoelectric unity, at the middle of the back wall. Fans diameter 80 mm with speed of 2,99 m/s. Initial temperature of 269 Kelvin.

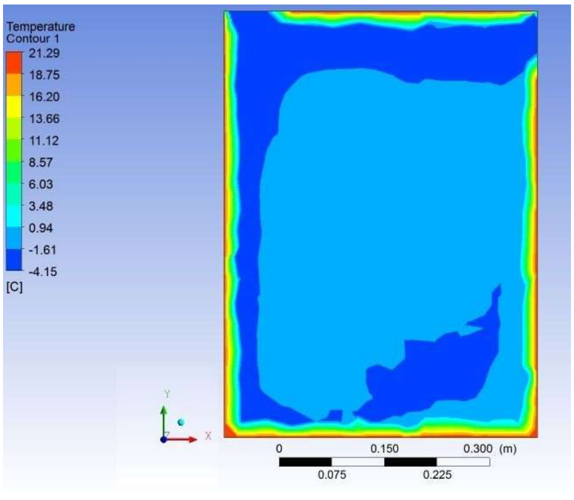

Figure 11. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan nearby the chamber door and the Thermoelectric unity, at the top of the back wall. Fans diameter 80 mm with speed of 2,99 m/s. Initial temperature of 269 Kelvin.

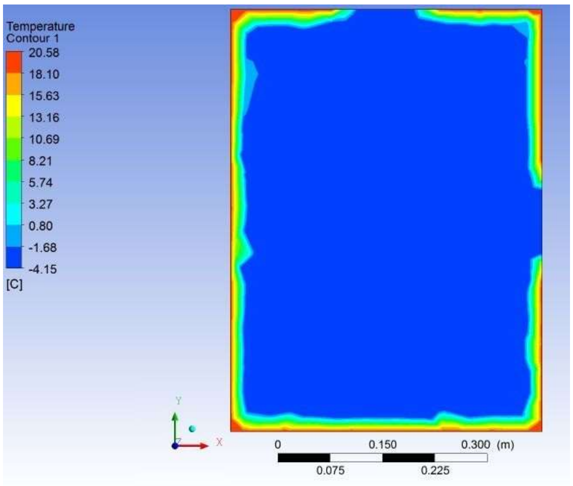

Figure 12. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan at the middle of the ceiling and the Thermoelectric unity, at the middle of the back wall. Fans diameter 120 mm with speed of 4,80 m/s. Initial temperature of 269 Kelvin.

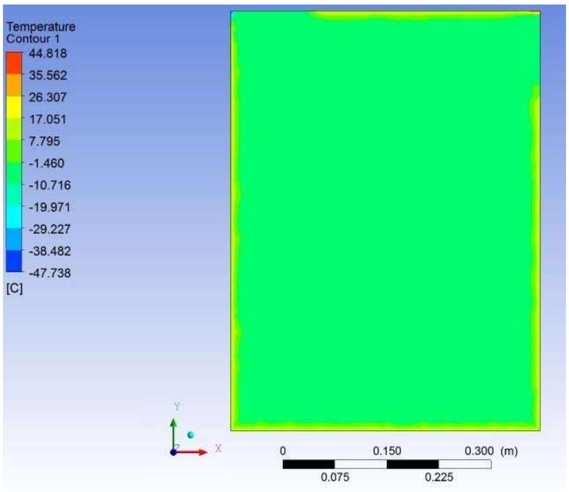

Figure 13. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan nearby the chamber door and the Thermoelectric unity, at the top of the back wall. Fans diameter 120 mm with speed of 4,80 m/s. Initial temperature of 269 Kelvin.

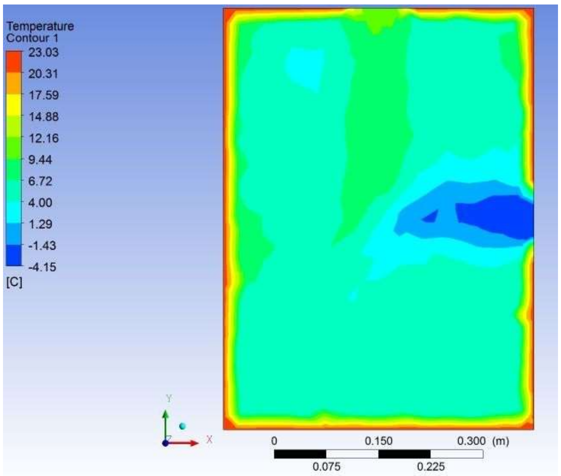

Figure 14. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan at the middle of the ceiling and the Thermoelectric unity, at the middle of the back wall. Fans diameter 80 mm with speed of 2,99 m/s. Initial temperature of 283 Kelvin.

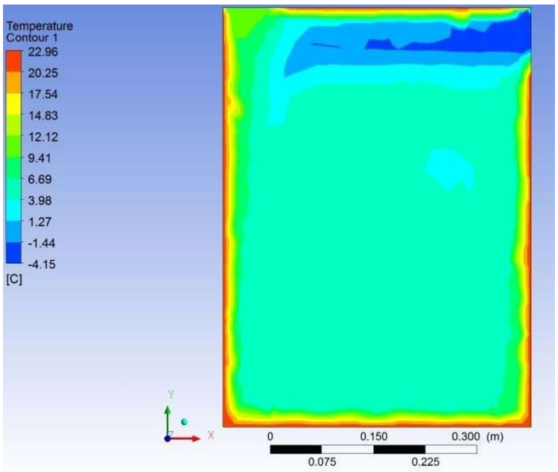

Figure 15. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan nearby the chamber door and the Thermoelectric unity, at the top of the back wall. Fans diameter 80 mm with speed of 2,99 m/s. Initial temperature of 283 Kelvin.

Notwithstanding, in order to the simulation be more faithful to the real phenomenon, the initial temperature should be the typical tropical ambient temperature, because this study seek to represent a domestic refrigerator that works in a domestic Brazilian residential condition. Therefore, the most representative process is that shown in Figs. 18 to 21, which had the temperature of 297 K as initial. In this way, analyzing among the results presented there, the most adequate conclusion is that the best performance is the one with the positioning of the smaller fans in the middle of the upper and posterior sides of the middle plane, shown in Fig. 18, i.e. 2.99 m/s.

The higher velocities imposed higher temperatures in a large internal area, as shown in Figs. 20 and 21. These results do not confirm those achieved by Rocha (2009). In his work, the author states that higher temperatures appear in the upper part of the refrigerated chamber and the positioning of the cold source in that region attenuates this effect.

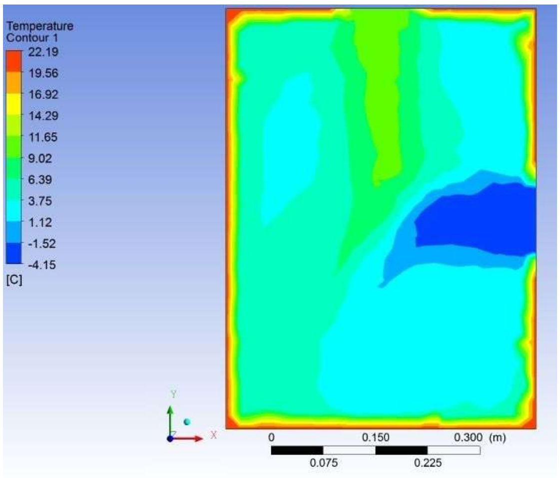

Figure 16. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan at the middle of the ceiling and the Thermoelectric unity, at the middle of the back wall. Fans diameter 120 mm with speed of 4,80 m/s. Initial temperature of 283 Kelvin.

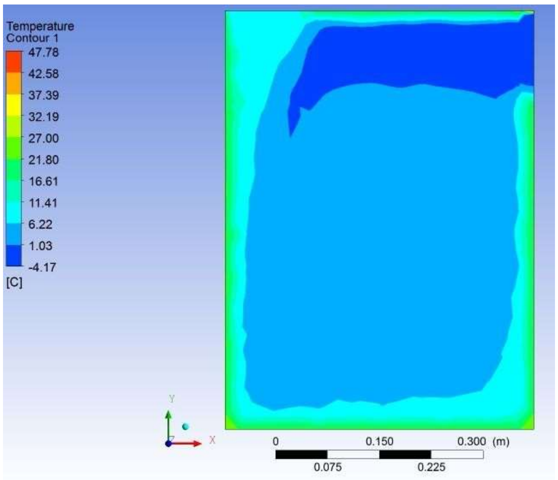

Figure 17. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan nearby the chamber door and the Thermoelectric unity, at the top of the back wall. Fans diameter 120 mm with speed of 4,80 m/s. Initial temperature of 283 Kelvin.

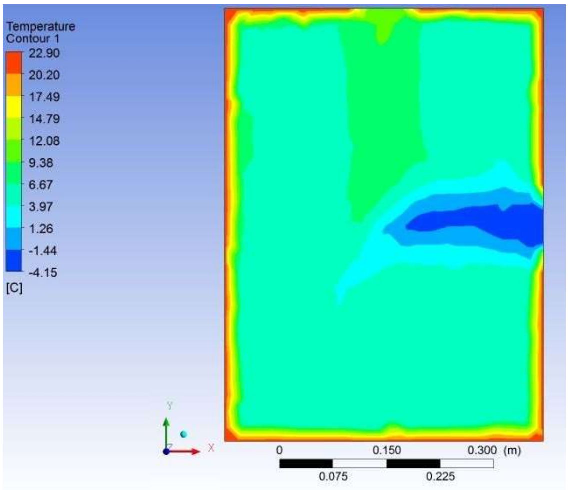

Figure 18. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan at the middle of the ceiling and the Thermoelectric unity, at the middle of the back wall. Fans diameter 80 mm with speed of 2,99 m/s. Initial temperature of 297 Kelvin.

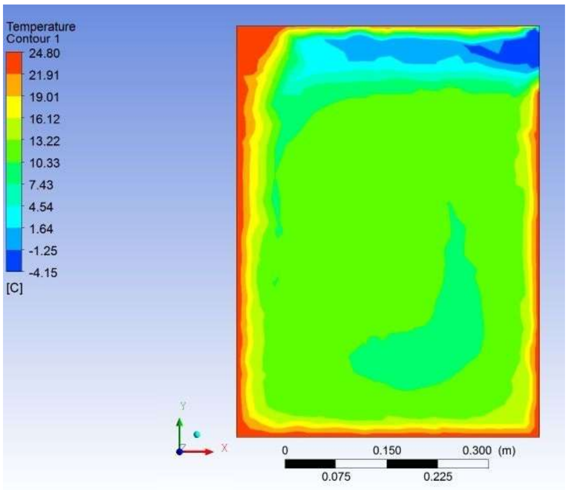

Figure 19. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan nearby the chamber door and the Thermoelectric unity, at the top of the back wall. Fans diameter 80 mm with speed of 2,99 m/s. Initial temperature of 297 Kelvin.

For Rossi et all (2012), the geometry of the insufflators has a direct influence on the trajectory of the air jet, on the distribution of temperatures and on the thermal comfort of an air-conditioned environment. However, the present work seems to contradict this conclusion in part, since smaller fans with lower speeds promoted better results inside the refrigerated chamber.

The possible explanation for this fact may be the size of the chamber. As the compartment is small, greater wind flow resulted in a more intense vortex, a consequent greater friction and with that, greater heating.

Other combinations of positioning and fan speed are needed to widen the range of options and mature the analysis of which condition is the best. For the case studied, more forced convection seems to have no direct influence proportional to the best results. Also, it seems to have promoted higher friction and thus, higher internal temperatures, which is highly undesirable for a home refrigerator.

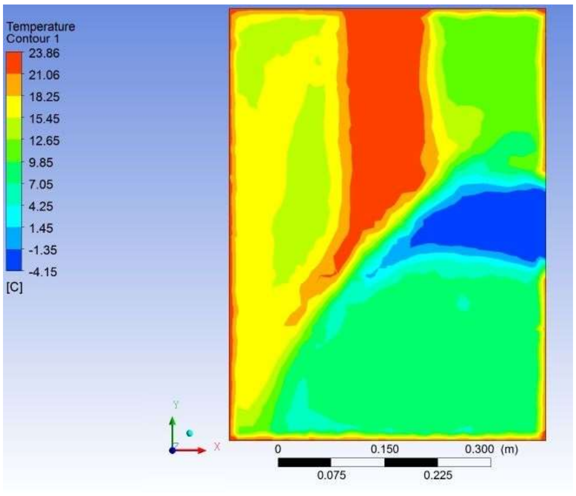

Figure 20. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan at the middle of the ceiling and the Thermoelectric unity, at the middle of the back wall. Fans diameter 120 mm with speed of 4,80 m/s. Initial temperature of 297 Kelvin

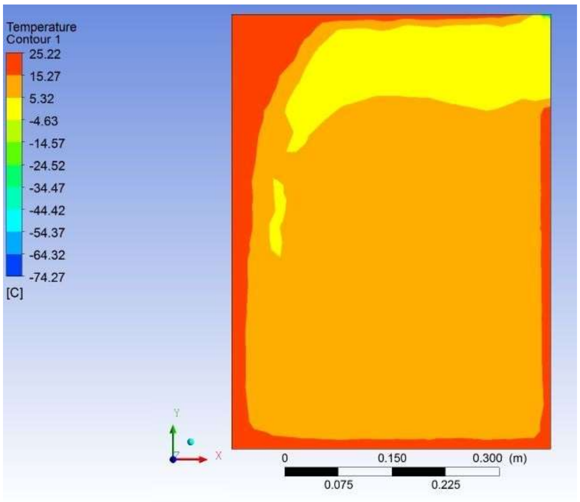

Figure 21. Temperature behavior in the vertical middle plan of the chamber, with the forcing fan nearby the chamber door and the Thermoelectric unity, at the top of the back wall. Fans diameter 120 mm with speed of 4,80 m/s. Initial temperature of 297 Kelvin.

4 CONCLUSIONS

Two positioning layouts of cold source and convection forcer fans, combined with two speed conditions and diameter of fan units were studied in a domestic wine cellar. The vertical mean plane of the refrigerated chamber was taken as representative of the entire internal volume. The performance was verified through simulation in a commercial package.

The best results achieved out between twelve resulting combinations were those with the positioning of the cold source on the back wall, exactly in the center thereof and positioning of a fan for convection at the ceiling, also at the central point.

The results seem to contradict the mainstream found in the literature, which states that the best cooling conditions are achieved with higher fan speeds and with cold source positioning at the top of the chamber. The size of the winery may have influenced the performance of the fans and the cooler, due to the increased friction of the air jets with the inner walls.

Complementary simulations should be carried out in order to increase the number of combinations and verify the accuracy of the conclusions.

5 AKNOLEDGEMENTS

The authors acknowledge “Programa Pesquisa Produtividade - Universidade Estácio de Sá” for funding the research.

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