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An Experimental Study on the Use of Vacuum Insulation Panels in Household Refrigerators

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ABSTRACT

The aim of this research was to study experimentally the thermodynamic behavior of a typical household refrigerator, by varying the vacuum insulation panel (VIP) coverage area and positioning around the cabinet. To this end, 16 samples of the same refrigerator model were constructed with distinct VIP arrangements and subjected to exhaustive reverse heat leakage) (RHL) and energy consumption tests. As expected, it was found that the energy consumption is dependent on both the coverage area and the positioning. The data gathered was analyzed statically in order to determine the most promising areas for the installation of the panels. Energy savings of 6% and 11% were observed when the panels were installed on the doors and on the rear wall, respectively. Overall, it was found that a coverage area of 56% reduces the energy consumption by 21%.

1. INTRODUCTION

The growing energy for demand, the depletion of natural resources and global warming are the main factors behind the increasingly severe energy consumption policies. Between 2005 and 2012, the ratio of Brazilian houses with at least one refrigerator rose from 87.9 % to 94.5 %, with a corresponding growth of 4039 GWh in the energy consumption. In 2014, for instance, refrigerators and freezers accounted for 23 % of the Brazilian residential electrical energy consumption (EPE, 2014). An alternative to diminish the energy consumed by such products is to decrease their thermal load. Part of this load is due to door openings and the storage of products. The dominant part, however, is due to heat transfer through the cabinet walls. This part, which corresponds to approximately 60 % of the total thermal load (Boughton *et al.*, 1996), can be reduced by diminishing the temperature difference between the external and internal air or by increasing the insulation thickness. However, none of these options are really effective, because the temperatures are established by the energy standards and the thickness affects the net storage volume. Insulation materials with a lower thermal conductivity are thus desirable. However, considerable reductions in the thermal conductivity of the current polyurethane foams are not expected in the near future. The vacuum insulation panels (VIPs), with thermal conductivities of the order of 1/5 those of the PU foams, seem to offer the best alternative, despite the costs, handling, storage and application issues. In a numerical study carried out with a 510-liter refrigerator Fine *et al.* (1992), for instance, obtained energy savings of 20% with a VIP coverage area of 70%. A comparative analysis between a refrigerator mounted with PU foam expanded with CFC-11 and another with vacuum panels (47 % coverage area) revealed a 10 % energy consumption reduction (Fine *et al.*, 1994). Reverse heat leakage tests were also performed with two 570-liter top-mount refrigerators, one with PU and another with VIPs on the freezer or the doors (Vineyard *et al.*, 1998). It was found that the thermal loads of the prototypes with panels only on the doors (15 % coverage area) and only on the freezer (22 % coverage area) were, respectively, 11 and 15 % lower than that of the configuration without panels. In another study (Melo and Vieira, 2003), the panels were distributed in order to cover 22 % of the total area of a top-mount refrigerator, representing 56 % of the freezer area and only 3 % of the fresh-food compartment area. It was verified that the global thermal conductances of the configurations with and without panels were the same, which was associated with a possible redistribution of the heat flux towards regions with lower thermal resistances (i.e. PU only). Hammonds and Evans (2014) also studied prototypes of top-mount refrigerators insulated with VIPs and observed that the heat transfer through the rear wall decreased from 9.0 W to 3.6 W due to the presence of the panels. Therefore, it is clear that both the coverage area and panel distribution are determining factors for the best use of this technology. This study offers a unique experimental analysis, correlating the energy consumption with different VIP insulation alternatives, using, to this end, 16 samples of the same product.

2. EXPERIMENTAL WORK

A top-mount refrigerator with a 108-liter freezer and a 329-liter fresh-food compartment, originally insulated with cycloisopentane-expanded PU, was selected for this study. The average thicknesses of the freezer and fresh-food walls are, respectively, 70 and 50 mm. The vacuum panels are 8-mm thick and are constituted by a fiber glass core and a laminated film casing. The panels are self-adhesive and were fixed on the inner surface of the metallic sheet that covers the cabinet, in series with the PU layer. Sixteen samples were obtained by combining four cabinet configurations with four door configurations. The cabinets, classified from 0 to 3, are shown in Figure 1 for the door configuration A. The door configurations are as follows: configuration A, without panels on the doors; configuration B, with a 0.50 x 0.55 m panel on the freezer door and no panels on the fresh-food door; configuration C, with a 1.10 x 0.55 m panel on the fresh-food door and no panels on the freezer door; and finally, configuration D, with panels on both doors.

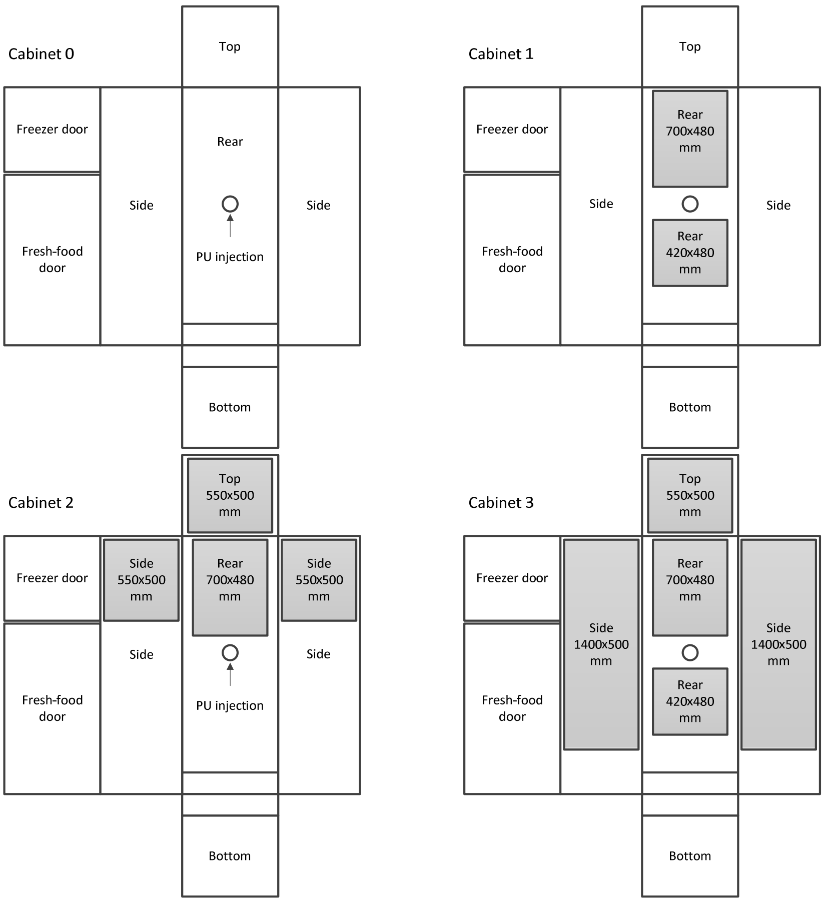

Figure 1: Diagram of cabinets 0 to 3, without panels on the doors

Table 1 shows details of the sixteen configurations, indicating the coverage area distributed by regions or compartments, and the types of doors and cabinets. All samples were tested in a climate chamber with controlled temperature, humidity and air velocity. The compartments temperatures were taken as the average of the values measured with type-T thermocouples installed at their geometric centers at three heights: top, middle and bottom.

The energy consumption tests were carried out based on the recommendations of the ISO 15502 (2005) standard, but with the following modifications to reduce the testing time: i) the compartments were not loaded with packages and ii) an interval of at least 5 h, comprised of only full and periodic compressor cycles, was adopted to integrate the energy consumption. Each refrigerator was submitted to two tests, one with the average temperatures of the compartments above and another with them below the reference values (5 ºC for the fresh-food and -18 ºC for the freezer). The ambient temperature was maintained at 32 ºC. An energy consumption for each compartment was obtained through the interpolation of the values obtained with both tests, based on the reference temperature of each compartment. The final energy consumption was taken as the average between these values.

Table 1: VIP positioning and respective areas, in m² (cabinet overall area = 5.2 m²)

Configuration		Total	By region			By compartment	
Cabinet	Doors		Doors	Rear	Top and side	Fresh-food	Freezer
0	A	0.00	0.00	0.00	0.00	0.00	0.00
	B	0.28	0.28	0.00	0.00	0.00	0.28
	C	0.60	0.60	0.00	0.00	0.60	0.00
	D	0.88	0.88	0.00	0.00	0.60	0.28
1	A	0.54	0.00	0.54	0.00	0.26	0.28
	B	0.82	0.28	0.54	0.00	0.26	0.56
	C	1.14	0.60	0.54	0.00	0.86	0.28
	D	1.42	0.88	0.54	0.00	0.86	0.56
2	A	1.16	0.00	0.34	0.82	0.06	1.10
	B	1.44	0.28	0.34	0.82	0.06	1.38
	C	1.76	0.60	0.34	0.82	0.66	1.10
	D	2.04	0.88	0.34	0.82	0.66	1.38
3	A	2.21	0.00	0.54	1.68	1.11	1.10
	B	2.49	0.28	0.54	1.68	1.11	1.38
	C	2.81	0.60	0.54	1.68	1.71	1.10
	D	3.09	0.88	0.54	1.68	1.71	1.38

In order to evaluate the quality of the thermal insulation of the walls, the global thermal conductances of each compartment, the so-called *UA* values, were measured by reverse heat leakage tests. The test methodology adopted herein follows closely that proposed by Gonçalves *et al.* (2000) where the product refrigeration system is deactivated and electrical heaters are used to heat the internal air, providing a temperature difference in relation to the external air. Therefore, the heat flux direction is the opposite to the normal direction, hence the name of the test. During the tests the evaporator fan was kept on to minimize the temperature stratification within the compartments and also to maintain the original airflow pattern of the refrigerator. Figure 2 illustrates this procedure.

All tests were conducted on a steady-state basis in an environment at 18 ºC. The compartments of each product were submitted to the operating conditions shown in Table 2, which were chosen in order to create temperature gradients between internal and external environments similar to those found when the refrigeration system is on.

Table 2: Operating conditions for reverse heat leakage test

Condition	1	2	3	4
Freezer [°C]	40.3	43.3	48.3	53.0
Fresh-food [°C]	29.1	42.9	47.9	44.9

The thermal conductances of the fresh-food and freezer (UA_{ff} and UA_{fz}, respectively) were determined by fitting Equation (1), which describes an energy balance over the whole refrigerator, to the temperature and power experimental data using the least-squares method.

$$UA_{fz}\left(T_{fz}-T_a\right)+UA_{ff}\left(T_{ff}-T_a\right)=W_{fz}+W_{ff}+W_f \tag{1}$$

In the above equation W_{fz} and W_{ff} represent, respectively, the power released by the electrical heaters of the freezer and fresh-food compartments, and W_f stands for the evaporator fan power. The average temperatures of the freezer, T_{fz}, fresh-food, T_{ff}, and external ambient air, T_a, were obtained analogously to the energy consumption tests.

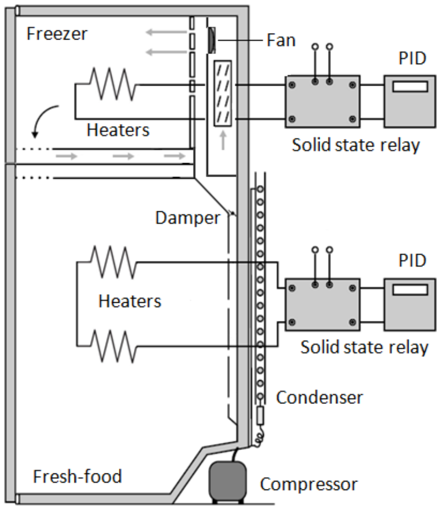

Figure 2: Reverse heat leakage test instrumentation (adapted from Hermes *et al*., 2013)

In parallel, to check the effects of the possible aging of the panels on the results obtained, the thermal conductivity of three 0.2 x 0.2 m panel samples with a composition identical to that of the VIPs installed within the cabinets was monitored for almost three years. Sample 1 is 8-mm-thick while samples 2 and 3 are 15-mm-thick. The thermal conductivity was measured with a thermal conductivity meter with an accuracy of 1%. The conductivity was referenced to an average temperature of 32.5 ºC. The panel aging when installed inside the cabinet walls was also evaluated. For this reverse heat leakage tests were repeated within a period of one year.

3. RESULTS AND DISCUSSION

3.1. Energy consumption and reverse heat leakage tests

The product energy consumption and thermal conductances of each compartment are shown in Table 3. To improve the analysis, the total thermal load that the cabinet would have been exposed to with the refrigerating system operating in steady state was also calculated. It was defined as the opposite side of Equation (1) plus the fan power, assuming T_{fz}, T_{ff}, T_a, and W_f equal to -18 ºC, 5 ºC, 32 ºC and 8.2 W, respectively. The effects of the VIP coverage area on the energy consumption and thermal load are presented, respectively, in Figures 2 and 3. The global trend is linear for the energy consumption, indicating a reduction of 2.9 (kWh/month)/m^2. A similar trend was observed for the thermal load. However, there are secondary trends in each figure, which are dependent on the position of the panels. For instance, when analyzing samples 0D and 1A, 1B and 2B, and 1D and 3A the energy consumption was similar for different coverage areas. In addition, for samples 0C and 1A, and 1D and 2B, despite the similar coverage areas, the energy consumption differed. It was also noted that configurations 0C, 0D, 1A, 1B and 2B presented thermal loads nearly identical to that of the original configuration, 0A, which has no panels. A similar situation was identified when comparing samples 1D and 3A. It was also observed that samples 1D and 2B, with nearly the same coverage area, showed different thermal loads. The same effect was verified for samples 1C and 2A. These results suggest that, despite the experimental uncertainties, the location of the panels plays an important role in the energy consumption.

However, explaining the differences between each case can be an arduous task, given the various physical phenomena involved and the approximations adopted during the tests. Thus, the experimental data was correlated using linear equations in which the energy consumption and the thermal load are expressed in terms of the areas and the position of the panels. The correlations were fitted using the least squares method considering up to three independent variables, as shown in Equation (2).

$$y = a_0 + \sum_{i=1}^{3} a_i x_i \tag{2}$$

Table 3: Energy consumption and reverse heat leakage test results

Configuration		Energy consumption	UA_{fz}	UA_{ff}	Thermal Load
Cabinet	Doors	[kWh/month]	[W/K]	[W/K]	[W]
0	A	51.3 ± (1.2)	0.50 ± (0.03)	1.25 ± (0.05)	67.0 ± (2.0)
	B	51.7 ± (1.4)	0.54 ± (0.03)	1.22 ± (0.05)	68.4 ± (1.9)
	C	50.6 ± (1.5)	0.52 ± (0.03)	1.19 ± (0.05)	66.8 ± (1.9)
	D	48.9 ± (1.0)	0.50 ± (0.03)	1.22 ± (0.05)	66.1 ± (1.9)
1	A	48.7 ± (1.4)	0.48 ± (0.03)	1.26 ± (0.05)	66.2 ± (2.0)
	B	47.8 ± (1.1)	0.45 ± (0.03)	1.31 ± (0.05)	66.1 ± (1.9)
	C	49.2 ± (0.5)	0.50 ± (0.03)	1.20 ± (0.05)	65.3 ± (1.9)
	D	45.1 ± (1.3)	0.52 ± (0.03)	1.11 ± (0.05)	64.2 ± (2.1)
2	A	49.7 ± (1.4)	0.41 ± (0.03)	1.25 ± (0.06)	62.5 ± (2.0)
	B	47.9 ± (1.4)	0.54 ± (0.03)	1.20 ± (0.05)	67.1 ± (1.9)
	C	46.6 ± (2.3)	0.48 ± (0.03)	1.20 ± (0.05)	64.6 ± (2.0)
	D	46.8 ± (1.5)	0.45 ± (0.03)	1.23 ± (0.05)	63.9 ± (1.9)
3	A	45.2 ± (0.9)	0.50 ± (0.03)	1.11 ± (0.05)	63.2 ± (1.9)
	B	44.9 ± (1.0)	0.51 ± (0.03)	1.08 ± (0.05)	62.9 ± (1.9)
	C	43.8 ± (0.9)	0.52 ± (0.03)	1.03 ± (0.05)	62.0 ± (1.9)
	D	41.3 ± (0.6)	0.43 ± (0.03)	1.04 ± (0.05)	57.8 ± (2.0)

In Equation (2) y can be the energy consumption, EC, or the thermal load, Q_t, and x_i are the areas covered by panels in specific regions of the cabinet. The coefficient a_0 is the average energy consumption or thermal load of the cabinet without panels. The other coefficients, a_i, are sensitivity factors, in (kWh/month)/m^2 or W/m^2, related to the respective areas and they are negative, thus indicating a reduction in both the energy consumption and thermal load due to the presence of the panels. Three distinct analyses were performed. The first considers the total area of panel, A, as the independent variable to evaluate the global impact of the thermal insulation. The second analysis takes into account the separate effect of the panel area in each compartment, where A_{fz} and A_{ff} are the panel areas in the freezer and fresh-food compartments, respectively. Finally, the third analysis tries to explain the impact of the combined application of panels in specific regions of the cabinet, such as the doors, A_p, rear wall, A_t, and side and top walls combined, $A_{l,s}$.

The global effect of the VIP installation was expressed by the following linear equations:

$$Q_t = 68.0 - 2.390A \tag{3}$$

$$EC = 51.6 - 2.915\,A \tag{4}$$

It is interesting to note that the a_0 coefficient of these equations corresponds to the thermal load and energy consumption of sample 0A (without panels). It was also observed that the other coefficients are negative, indicating that the addition of VIPs contributes to a reduction in both Q_t and EC. It was noted that the sensitivity of the VIP area increase is around 20 % higher for the energy consumption correlation. In fact, the addition of VIPs in sample 3D caused a reduction of 14 % in the thermal load and 19 % in the energy consumption. It should be mentioned that the temperature differences used to obtain the thermal conductances of the compartments were taken between the center of each compartment and the external air, and the internal temperatures were intentionally homogenized throughout the tests. However, when the refrigerating system is operating the dominant temperature differences are not uniform and are not those measured between the center of the compartment and the external environment (ideally the temperatures of the inner and outer surfaces of the walls should be measured). Moreover, the internal temperatures are clearly different from those of the reverse heat leakage tests, mainly due to the presence of components that influence them, such as the compressor, condenser and evaporator, which create hot and cold regions next to the surface. Therefore, it can be seen that the reverse heat leakage test does not capture all of the effects of the VIPs, and that the energy consumption tests are more representative.

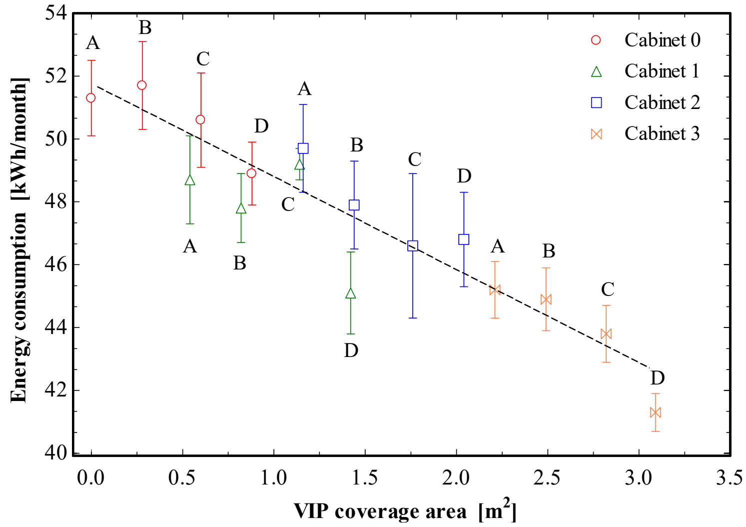

Figure 2: Energy consumption *versus* VIP coverage area

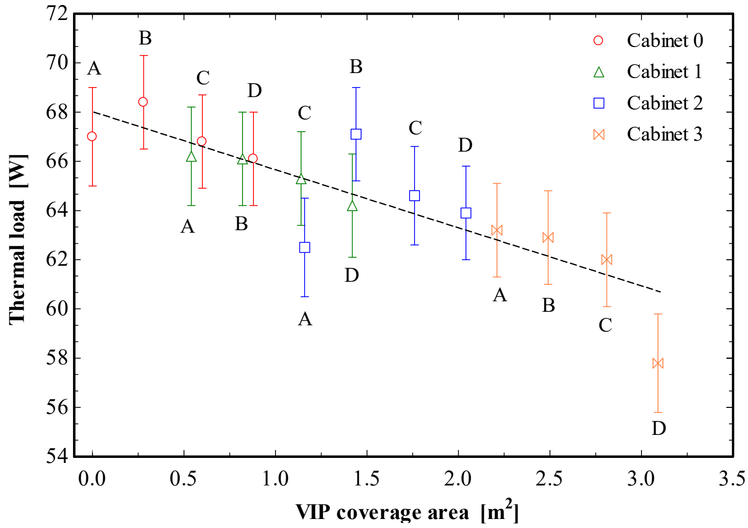

Figure 3: Thermal load *versus* VIP coverage area

Equations (5) and (6) present the correlations that consider the individual effects of the VIPs installation in the freezer and fresh-food compartments. Again, the a_0 coefficient of both equations is consistent with the parameters measured for the reference sample (0A).

$$Q_t = 67.9 - 1.771A_c - 2.966A_r \quad (5)$$

$$EC = 51.5 - 2.291A_c - 3.496A_r \quad (6)$$

The addition of 1.0 m^2 of VIP in the freezer provides a reduction of 2.6 % in the thermal load and 4.4 % in the energy consumption, while the use of 1.0 m^2 in the fresh-food compartment decreases the thermal load by 4.4 % and the energy consumption by 6.8 %. Both correlations indicate that the impact of the VIPs on the fresh-food compartment is higher than that on the freezer. This is related to the thermal resistance modification due to the presence of the panels and the fact that the fresh-food compartment has a thinner wall (~50 mm) than the freezer (~70 mm).

The third analysis uses as independent variables the VIP area in different regions of the cabinet, without distinguishing between compartments. These regions are: i) doors, ii) rear wall and iii) side and top walls. Again, the a_0 coefficient of both equations is in line with the results of the sample without VIPs.

$$Q_t = 68.0 - 2.262A_p - 2.728A_t - 2.328A_{l,s} \quad (7)$$

$$EC = 52.3 - 3.403A_p - 5.681A_t - 2.192A_{l,s} \quad (8)$$

The coefficients a_1, a_2 and a_3 of Equation (7) are very close, which is related to the temperature uniformity traditionally used in the reverse heat leakage tests. On the other hand, the correlation for the energy consumption reveals a higher significance for the rear wall, followed by the doors and finally the side and top walls. For the energy consumption, the effect of the VIPs placed on the rear wall (coefficient a_2) is approximately 40 % higher than the effect on the doors (coefficient a_1) and almost 60 % higher when compared to that of the side and top walls (coefficient a_3). This behavior can be explained by the temperature differences that govern the heat conduction through the rear wall, which are intensified by the lower evaporator temperature and higher condenser and compressor temperatures. For example, for the reference sample, the average evaporating and condensing temperatures are -24.5 ºC and 48.0 ºC, respectively. The ambient and freezer temperatures are, respectively, 32 ºC and -18 ºC. Thus, the temperature difference that governs the heat conduction at the rear wall is close to 72.5 ºC, while the temperature difference at the other walls is close to 50 ºC.

3.2. VIP aging

Figure 4 shows the thermal conductivity evolution of three VIP samples over time. It can be seen that the thermal conductivity varies linearly, increasing on average by 1.9 mW/(m.K) per year for sample 1 and 1.4 mW/(m.K) per year for samples 2 and 3. This increase, close to 30 %, was expected and is due to the length of the sealing ring, which is small compared to the panel area, thus facilitating the oxygen and nitrogen permeation (Fricke *et al.*, 2006).

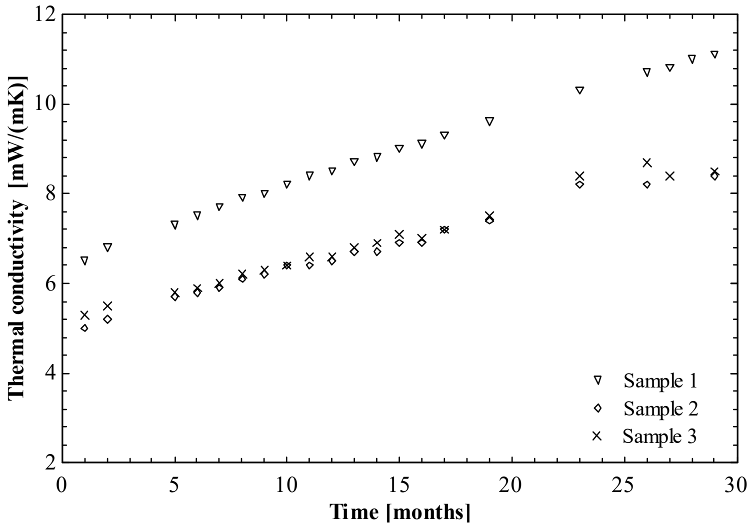

Figure 4: Thermal conductivity of the VIP samples *versus* time

As mentioned above, the VIP aging was also investigated with the panels installed inside the cabinet. To this end, two reverse heat leakage tests were carried out with the configuration 1D: one at the beginning of the project and another after one year. The first test indicated a global thermal conductance of 0.52 ± 0.03 W/K for the freezer and 1.11 ± 0.05 W/K for the fresh-food compartment, while the second showed values of 0.52 ± 0.03 W/K and 1.16 ± 0.06 W/K, respectively. This difference is insignificant and close to the experimental uncertainties. In the refrigerator the panel is installed between the outer metallic sheet of the wall and the PU foam. The PU cells are closed and filled with cycloisopentane, which has molecules larger than those of the gases in the atmosphere. Therefore, even if there are open cells in the PU foam, the permeation of gases will be much lower than that observed for the samples exposed to the ambient air, which explains the virtually imperceptible degradation of the insulation when installed in the cabinet.

4. CONCLUSIONS

Energy consumption tests were conducted with sixteen samples of the same product, mounted with different vacuum insulation panel configurations. The results showed that the global effect of the panels used is a reduction in the energy consumption. It was observed that not only the area but also the position of panels around the cabinet should be considered with regard to the energy consumption. It was found that samples with similar coverage areas presented distinct energy consumptions, while samples with different areas showed similar consumptions. Experiments revealed that, in general, the effect of the panels is greater when they are installed on the fresh-food compartment. The analysis by regions showed that the effect of the panels is greater on the rear wall due to the temperature difference to which it is submitted. The doors were also more impacted by the use of vacuum panels because of their lower thickness compared to the side and top walls. It was observed that the insertion of vacuum panels only on the rear wall reduced the energy consumption by 11 %. It was also noted that the installation of 8-mm-thick panels on 56 % of the refrigerator surface area caused a reduction of 21 % in the energy consumption. Finally, it was shown that the panel aging, obtained from thermal conductivity measurements, does not correspond to that observed in practice.

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