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Challenges on Converting an Upright Ice-Cream Freezer from R404a to R290 Complying with 150g Refrigerant Charge Restriction

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ABSTRACT

Glass door vertical refrigeration systems for commercial applications located in places with an elevated flux of people throughout the day have a high door opening rate, turning the maintenance of the inner temperatures below -18°C a huge challenge. Usually evaporators with wide surface area are used in order to improve the heat exchange, demanding greater refrigerant charge. Other factors, namely the evaporator's heat exchange efficiency loss caused by the ice formation between fins and the power supply oscillations, contribute to the system's capacity losses, affecting the fresh food flavor and preservation time as well as the end customer's expectation.

The present work shows the results obtained through the optimization and conversion of vertical ice-cream freezer from R404a to R290 with limit of 150 g. The use of variable cooling capacity compressor and system component's optimization were fundamental to reach the operation requirements and maintain the internal temperature even in critical operating conditions as door opening tests, with extra benefit of energy consumption reduction.

1. INTRODUCTION

News on climate change is on the rise. A countless number of researches on climate change have been done. Studies show the CO_2 concentration at the atmosphere has been varying cyclically during the human history; however, the CO_2 concentration today has never been higher, rising continuously since the Industrial Revolution.

Many works indicate evidences directing to climate change related to the atmospheric CO_2 concentration rise. Some evidences relate it to the sea level rise, the seawater temperature and acidity rise and the retreat of glaciers and ice caps around the world. According to the National Aeronautics and Space Administration (2016), evidences also indicate an increase on the frequency of extreme climate events since 1950.

Several factors contribute to the man-made CO_2 emission to the atmosphere. The electricity generation is by far the main one though, as fossil fuels are usually burned during generation to some degree. The electrical energy generated in the USA, for instance, is mainly (67%) created by burning fossil fuels, such as coal, natural gas and petroleum, according to the U.S. Energy Information Administration (2016).

Refrigeration also impacts the environment; however, the refrigeration universe is going through a genuine revolution seeking to lower its impact on the environment. The refrigerants, which obviously play an important role in refrigeration, have been transformed throughout history in order to pass the different environmental regulations established. Nowadays the contribution from the refrigerants to the greenhouse effect is broadly known. This contribution is measured in terms of GWP index – Global Warming Potential. GWP is a comparison between the amount of heat retained by a certain amount of gas and the mass of CO_2 necessary to retain the same amount of heat. Furthermore, the GWP is measured accordingly to the amount of time the gas will stay in the atmosphere. Periods of 20, 100 and 500 years are commonly used.

Additionally, more efficient refrigerators lead to a reduction on the environment impact. This happens because a portion of the electrical energy generation usually comes from burning fossil fuels. Hence, a reduction on energy consumption will contribute to a certain reduction on CO_2 emission, depending on the local energy sources.

For these and other reasons, the USA and the Europe Union pioneered environmental laws promoting the use of low-GWP refrigerants for household and commercial applications. One of the alternatives which have been shaping up as environmental and technically adequate is the usage of natural refrigerants. They are found naturally on Earth and are commonly represented by the hydrocarbons (ethane, propane, isobutane and cyclopentene), ammonia, water, air and CO_2 itself. These options have a low GWP compared to largely used refrigerants, such as the HFCs – hydrofluorocarbons.

The natural refrigerants composed by hydrocarbons demand some close attention during their handling, since they are flammable. Furthermore, customized electrical components must be used for their application, reducing the probability of hazardous events in case of refrigerant leaks. Common electrical components can generate sparks causing a flame formation and must to comply with EN 60079-15 normative. Moreover, the maximum charge for natural refrigerants is 150 g per circuit to avoid refrigerant gas concentration increase and become dangerous and the system must to comply with IEC 60335-2-89:2010 standard. The charge limitation on natural refrigerants is a challenge during refrigerant change for light commercial equipments, as they normally have large condensers and evaporators, due to the necessity of a higher cooling capacity. They may also have a liquid receiver, making the case even worse, overburden or even deterring the migration.

Given this background, commercial refrigeration systems are designed to maintain an adequate internal temperature, besides having a weak impact on the environment, by using low-GWP refrigerants and decreasing energy consumption levels. Commercial refrigeration systems are commonly located on highly crowded areas and are prone to extensive door openings. The additional thermal load caused by the door openings requires the application to have a specifically designed system in order to guarantee the quality of the stored products. The hot and humid air entering the equipment triggers a severe condition of operation. The extra heat from the air increases the internal temperature and the humidity build up internally causing ice to accumulate on the evaporator, dropping the heat exchange effectiveness mainly by blocking the air flow near the fins. Frequent defrosting cycles are necessary to prevent this from happening.

This work aims to present different results obtained during a migration to a natural refrigerant. A large, 556 liters, vertical freezer with a glass door was used. The equipment can hold up to 100 kg of ice cream. The reference for the average internal temperature was -18°C, since it is adequate to frozen food preservation and it is normally specified by the equipment's designers. The migration made was from the refrigerant R404a (HFC) to R290 (natural), aiming to avoid many modifications to the system, and respecting the 150 g limit for the natural refrigerant charge. Tests simulating the equipment usage were made, by submitting it to periodic door openings. Finally, the environmental impact of this equipment during its lifespan was calculated.

2. Experimental procedures

The tests on the system were done inside a chamber able to control temperature and humidity. The fluctuations on temperature were under +/- 0.5K and on humidity +/- 5%. The air flow inside the chamber was vertical downwards, with a velocity of 0.3m/s fluctuating +/- 0.05m/s.

The refrigeration system was positioned inside the chamber 300mm away from the side walls and 100mm from the back wall. The data acquisition system was set to operate at 0.033Hz. The compressors chosen used variable frequency drive.

The tests used Tylose specimens to simulate an ice cream load. This material is commonly used during tests on refrigeration systems. The specimen load pattern chosen was to fill each one of the five shelves with 17 1kg and 6 0.5kg packages, two of them containing a thermocouple sensor in its geometrical center. The first shelf contained one instrumented package on its back-left corner and one on the front-right corner (Standard A). The second shelf contained one instrumented package on its back-right corner and one on the front-left corner (Standard B). The following shelves were loaded alternating the standards. A total of 100kg of specimens were used, 5kg of them were instrumented. The pattern is shown on Figure 1. Mi are the instrumented packages, the index i representing the package number, going from 1 to 10.

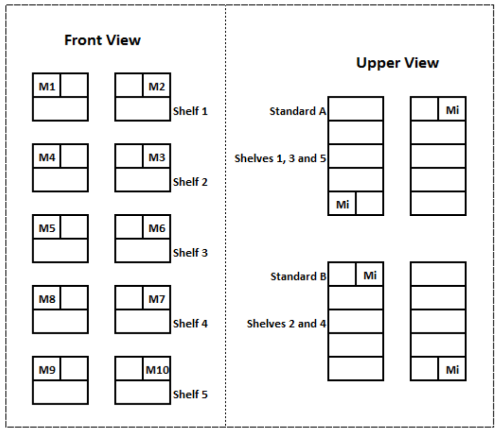

Figure 1: Pattern for fridge load

The system had a 60-mm-wide polyurethane thermal insulation and 220V as the nominal voltage. The system used a finned evaporator with forced air flow. The expansion device was a thermal expansion valve, tuned to keep a 5K superheat on the evaporator. It used a finned condenser with forced air flow. The method used to defrost was hot gas, due to a better efficiency. Details about the defrost procedure will be discussed on the following item.

2.1 Refrigerator configuration

During the tests two configurations of the system were used. The condenser, evaporator, load of specimens and most of the physical elements of the system were kept the same. The two configurations and its differences are shown on Table 1.

Table 1: Different refrigerator configuration

Specification	Combination A	Combination B
Compressor	Variable frequency drive reciprocating hermetic compressor with a 12.11cm³ displacement for R404a	Variable frequency drive reciprocating hermetic compressor with a 13.54cm³ displacement for R290
Expansion device	Thermal expansion valve, tuned to R404a	Thermal expansion valve, tuned to R290
Refrigerant charge	650g of R404a	150g of R290

The freezer has a double glass door equipped with electric resistance. Its power was adjusted to avoid condensation on the doors.

2.2 Temperature stabilization criterion and door-closed power consumption tests

The criterion adopted during the power consumption tests was a comparison between the average package temperature during the instant t and t- 24 hours. The maximum temperature fluctuation was +/- 0.5K. The average temperatures were measured between defrost cycles. The temperature and humidity inside the chamber were 35°C and 70%, respectively. These values were set to simulate a critical condition to the equipment, namely the climate condition number 07 for food preserving equipment, complying with the European Committee for Standardization. (2001). After the temperature stabilization the power consumption measurement is done during a 24-hour period between defrost cycles.

2.3 Periodic door opening power consumption tests

The procedure aimed to simulate a real-world usage of the equipment with extensive door openings. The tests with doors opening started 1 hour after the first defrost preceded by the stabilization described on item 2.2. The door was open on a 90° angle. The first opening takes 3 minutes. The following openings take 10 seconds and are spaced in periods of 5 minutes. Overall the procedure has 144 openings during the 12 hour test. The energy consumption was measured during the 12-hour procedure.

2.4 Total Equivalent Warming Impact (TEWI) evaluation

The environmental impact caused by a refrigerator should be evaluated based on direct and indirect effects it has during its manufacturing and operation. The methodology proposed by this work considers both of them. The direct effect, related to the refrigerant charge and the expansion agent, is the GWP of the fluids. The indirect is the possible usage of fossil fuels for generating the electricity used by the system. The Total Equivalent Warming Impact (TEWI) is defined as follows:

$$TEWI = m_{REF}GWP_{REF} + m_{EXP}GWP_{EXP} + \alpha E_{ANUAL}L \quad (1)$$

Where m_{REF} [kg] is the total mass of refrigerant, GWP_{REF} [$kgCO_2$/kgREF] is the GWP of the refrigerant, m_{EXP} [kg] is the total mass of expansion agent, GWP_{EXP} [$kgCO_2/kg_{EXP}$] is the GWP of the expansion agent, α [$kgCO_2$/kWh] is the Carbon Dioxide emission factor, relating the CO_2 released on the atmosphere during the electricity generation, E_{ANUAL} [kWh] is the annual energy consumption of the equipment and L is the lifespan of the equipment, in years.
The TEWI should be evaluated accordingly to an amount of time the gas will affect the environment. In this work a period of 100 years was chosen.
As both combinations A and B use the same expanded insulation and the work aims to in observe the differences between them, Equation (1) can be rewritten without m_{EXP} and GWP_{EXP}, as follows:

$$TEWI = m_{REF}GWP_{REF} + \alpha E_{ANUAL}L \quad (2)$$

The E_{ANUAL} value is defined combining the energy consumption during periods of doors closed and open periodically, as follows:

$$E_{ANUAL} = 365(0.5PC_{24h} + PCP_{12h}) \quad (3)$$

Where PC_{24h} is the result of doors-closed power consumption test and PCP_{12h} the periodic door opening power consumption test.

3. Test Results

The temperatures on steady state for the Combination A are depicted on Figure 2. These temperatures were enclosed between two defrost cycles. The higher average temperature measured during the defrost period was -22.6°C, the hottest single package was at -19.5°C. The average temperature before the defrost cycle was -24.4°C and the hottest single package at –21.1°C. During the entire test no package crossed the -18.0°C threshold. The power consumption registered for Combination A during the 24-hour period was 16.15kWh, complying with the criterion established on item 2.2.
Similarly, the temperatures on a stable state for the Combination B are depicted on Figure 3. These temperatures were also enclosed between two defrost cycles. The higher average temperature measured during the defrost period was -21.9°C, the hottest single package was at -20.4°C. The average temperature before the defrost cycle was -23.9°C and the hottest single package at –22.0°C. During the entire test no package crossed the -18°C threshold. The power consumption registered for Combination B during the 24-hour period was 15.70kWh, complying with the criterion established on item 2.2.
During the equipment operation, the highest temperatures recorded happened during the defrost period. On the other hand, the lowest temperatures were measured instants before the defrost period, as expected.
A large similarity is visible when one compares the results between both Combinations A and B. The average temperatures from Combination A were kept below Combination B during the entire test. For the hottest single package temperature, however, Combination B had a favorable result. It is important to notice the temperatures never crossed the -18°C threshold. The Combination B had energy consumption 2.8% lower than the Combination A. Table 2 briefly gathers the results.

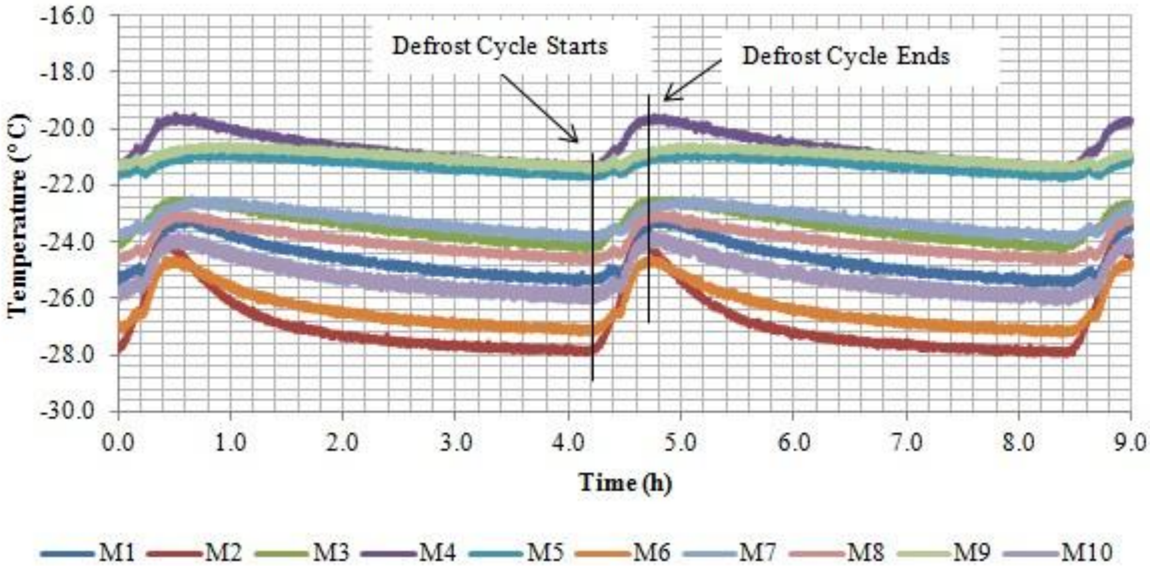

Figure 2: Internal temperature distribution on steady state for Combination A, doors-closed power consumption tests

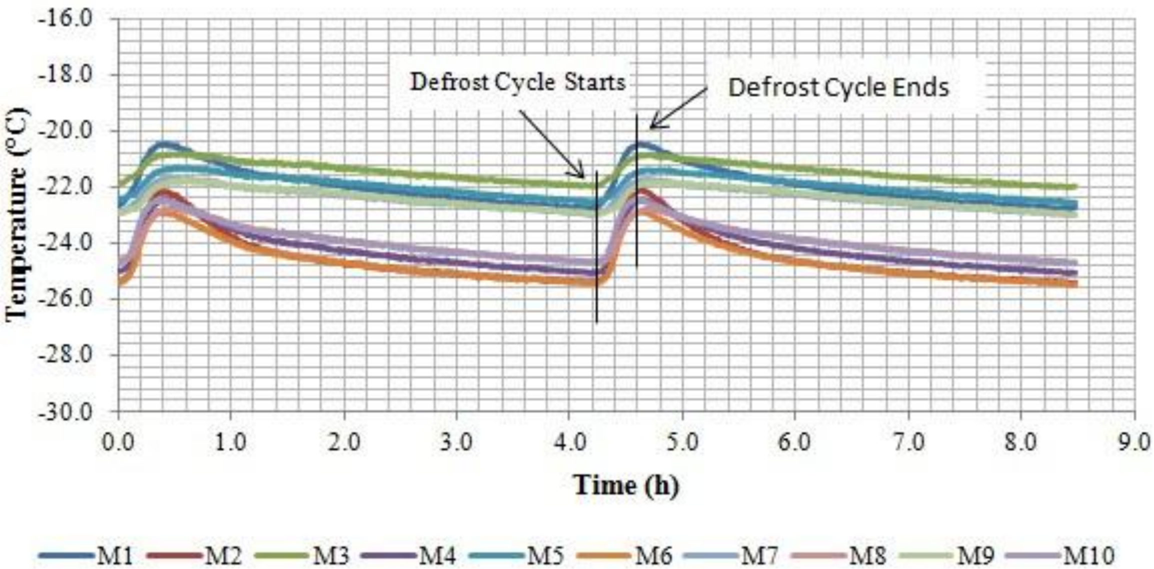

Figure 3: Internal temperature distribution on steady state for Combination B, doors-closed power consumption tests

In order to have a simulation closer to the real world usage the more extreme tests described on item 2.3 were performed. The door openings induce a severe introduction of hot and humid air to the system, strongly impacting the internal temperature and blocking the air flow by the formation of ice on the finned evaporator.

Three points of interest were chosen to analyze the system performance during the door opening procedure. The first one is the moment when the hottest package reaches -18.0°C. The second, the moment the average temperature of the packages reach -18.0°C. These two points carry information on the capacity the refrigerator has to keep or recover the internal temperature. Lastly, it was also evaluated the moment when the higher average temperature during the procedure is reached.

Table 2: Comparative performance during doors-closed power consumption tests

Property	Combination A	Combination B
Highest average temperature during the defrost cycle	-22.6°C	-21.9°C
Highest single package temperature during the defrost cycle	-19.5°C	-20.4°C
Highest average temperature before the defrost cycle	-24.4°C	-23.9°C
Highest single package temperature before the defrost cycle	-21.1°C	-22.0°C
Energy consumption during 24 hours (PC_{24h})	16.15kWh	15.70kWh

Figure 4 depicts the temperature behavior inside the equipment during the periodic door opening test, for Configuration A. The zero mark at the x-axis represents the first door opening and this moment occurs after 1 hour from the start of the defrost cycle. For this combination the average temperature at this moment was -23.3°C and the hottest package was at -20.2°C.

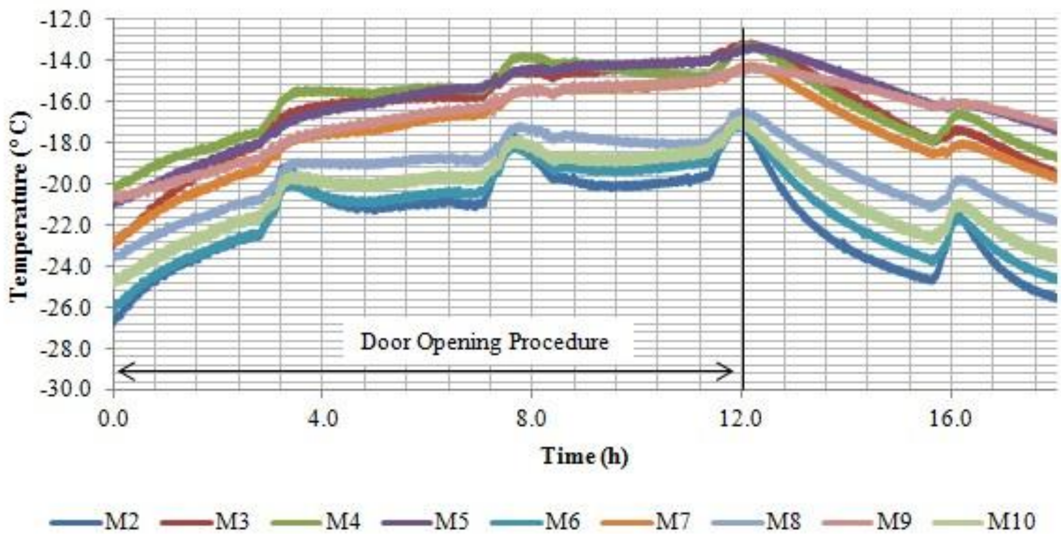

Figure 4: Internal temperature distribution on steady state for Combination A, Periodic door opening power consumption test

After 2.2 hours since the beginning of the procedure the hottest package reached -18.0°C. The average temperature reached -18.0°C at the 6.4 hours mark. Lastly, the hottest average temperature happened moments after the end of the procedure, at the 12.1 hours mark. The average temperature was -15.4°C and the hottest package was at -13.1°C. The energy consumption during the 12-hour period for the Combination A was 17.12kWh, complying with the criterion established on item 2.3.

Figure 5 shows the temperature behavior inside the equipment during the periodic door opening test, for Configuration B and similarly to Combination A, the door opening test starts after 1 hour of last defrost cycle. For this combination the average temperature at the beginning of the procedure was -22.2°C and the hottest package was at -21.0°C.

After 1.7 hours since the beginning of the procedure the hottest package reached -18.0°C. The average temperature reached -18.0°C at the 7.7 hours mark. Lastly, the hottest average temperature happened, again, moments after the

end of the procedure, at the 12.2 hours mark. The average temperature was -16.2°C and the hottest package was at -13.9°C.
The energy consumption during the 12-hour period for the Combination B was 16.25kWh, complying with the criterion established on item 2.3.

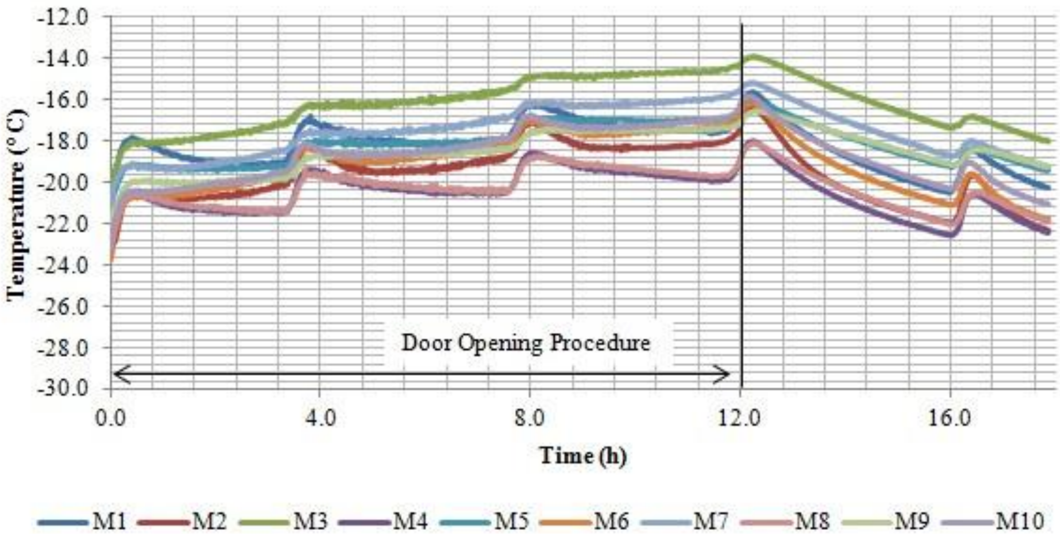

Figure 5: Internal temperature distribution on steady state for Combination B, Periodic door opening power consumption test

Comparing the combinations A and B, some similar behaviors appear. The moment when the hottest package reaches -18.0°C happens before the first defrost cycle for both of the combinations, at the 2.2 hours mark for A and 1.7 hours for B. The moment when the average temperature reaches -18.0°C happens before the second defrost cycle of with 7.7 hours mark for B and 6.4 for A. The highest temperature slope happens both before the first defrost cycle for both combinations A and B. The Combination B was capable of maintaining a lower average temperature at the end of the procedure, at -16.2°C versus -15.4°C for Combination A. The hottest packages from Combination B and A were at -13.9°C and -13.1°C, respectively.
It is important to notice that Combination A had five packages which passed -15.0°C, while Combination B had only one. Combination B also took a lead on the consumption measurement. It had energy consumption 5.3% lower than Combination A. Table 3 briefly gathers the results.

Table 3: Comparative performance during periodic door opening power consumption test

Property	**Combination A**	**Combination B**
Hottest package reached -18.0°C	2.2 hours	1.7 hours
Average reached -18.0°C	6.4 hours	7.7 hours
Hottest average temperature	-15.4°C at 12,1 hours	-16.2°C at 12,2 hours
Hottest package temperature	-13.1°C at 12,1 hours	-13.9°C at 12,2 hours
Energy consumption during 12 hours (PCP_{12h})	17.12kWh	16.25kWh

Even though both combinations were not able to maintain an internal temperature below -18.0°C during the procedure both of them kept it lower than -15.0°C, which is acceptable for frozen food preservation. It is worth mentioning the severity of the tests done. The chamber conditions chosen, 35.0°C and 70% humidity were intense and the amount and duration of door openings were extensive.

The environmental impact evaluation was done accordingly to Equation (2) using the parameters from Table 4, which uses GWP_{100REF} according to Forster et al. (2007) and the Intergovernmental panel on climate change (2001). Considering α as 0.676 kg CO_2/kWh, according to the U.S. Energy Information Administration (2007), and a five-year lifespan for the refrigerator, period defined by the manufacturer.

Table 4: Environment impact

Property	**Combination A**	**Combination B**
m_{REF} [kg]	0.65	0.15
GWP_{100REF} [$kgCO_2/kg_{REF}$]	3260	3.3
E_{ANUAL} [kWh]	9194.9	8798.5
$TEWI_{100}$	33198	29739

The $TEWI_{100}$ was reduced by 11.6% after the migration to a natural refrigerant. Both GWP_{100REF} and power consumption cooperated positively to the environmental impact reduction.

4. Conclusions

There was no need to make great changes to the components of equipment when migrating from R404a to R290, just the expansion device and the hermetic compressor were changed in order to accommodate the new refrigerant.
Thermally wise, Combinations A and B behaved similarly, with a few advantages to the Combination B, specially the ability to maintain a considerably smaller number of packages below the -15.0°C mark. Combination B also had a lower temperature average, eliminating any possible doubts about its cooling capacity. Moreover, Combination B consumed a lower amount of energy.
Environmentally wise, Combination B was significantly better than Combination A. The new condition reduced the environmental impact by using a less aggressive refrigerant and having a higher efficiency.

NOMENCLATURE

M	instrumented package	(–)
TEWI	total equivalent warming impact	(–)
m	mass	(–)
GWP	global warming potencial	(–)
α	carbon dioxide factor	(kg/kWh)
E	energy consumption	(kWh)
L	equipment lifespan	(–)
PC	power consumption	(kWh)
PCP	power consumption during door openning	(kWh)

Subscript

i	package number
REF	refrigerant fluid
EXP	expansion fluid
100	TEWI and GWP time horizon
ANUAL	annual time horizon
24h	period of 24 hour
12h	period of 12 hour

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