Experimental analysis of R-450A and R-513A as replacements of R-134a and R-507A in a medium temperature commercial refrigeration system

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ABSTRACT

This work presents the experimental evaluation of R-513A (GWP=573) and R-450A (GWP=547) as R-134a (GWP=1301) drop-in replacements and as R-507A (GWP=3987) retrofits in a commercial direct expansion refrigeration system for medium temperature applications (2ºC). The evaluation covered 24-hour tests using a single-stage cycle with semi-hermetic compressor, an electronic expansion valve customized for each refrigerant and a commercial vertical cabinet with doors placed inside a climatic chamber. The tests were performed at three water dissipation temperatures (23.3, 32.8 and 43.6ºC). Experimental results indicate that R-513A and R-450A can operate with R-134a plants, with increments in energy consumption between -1.6 to +1.2% for R-513A and from +1.3 to +6.8% for R-450A, whereas in comparison with R-507A, R-513A offered reductions in energy consumption between 4.4 to 8.2% and R-450A between 0 to 3.3%. The paper analyses the modification of the operating pressures/temperatures and the energy indicators using the four refrigerants.

KEYWORDS

R-450A; R-513A; R-134a; R-507A; commercial refrigeration; energy analysis;

NOMENCLATURE

COP	coefficient of performance
E	energy consumption, kWh
GWP	Global warming potential, 100 years horizon
h	specific enthalpy, $kJ \cdot kg^{-1}$
HR	relative humidity, %
P	pressure, bar
P_C	power consumption, W
s	specific entropy, $kJ \cdot kg^{-1} \cdot K^{-1}$
T	temperature, ºC
t	time, s
VCC	volumetric cooling capacity, $kJ \cdot m^{-}3$
x_v	vapour title

GREEK SYMBOLS

λ	latent heat of phase-change, $kJ \cdot kg^{-1}$
v	specific volume, $m^3 \cdot kg^{-1}$

SUBSCRIPTS

air	cabinet return air to evaporator
dis	discharge
in	inlet
l	saturated liquid
O	evaporating level, evaporator
out	outlet
prod	product
suc	suction
K	condensing level
v	saturated vapour
w	water

1. Introduction

Commercial refrigeration sector includes hermetic stand-alone, condensing unit, and multipack refrigeration systems. According to UNEP (2011), this sector accounts for approximately 32% of the world HFC consumption or 40% of the HFC consumption in the refrigeration/AC sectors. Hermetic stand-alone systems, quantified in 32 million units plus 20.5 million vending machines, usually rely on R-22, R-134a, R-404A, R-507A or recently R-290. Condensing units, generally composed by one or two compressors, one condenser and one receiver assembled into a unit that serves one or more services, are quantified in 34 million units, generally using R-22, R-134a, R-404A, R-507A and R-407C. In addition, centralized refrigeration systems, consisting of a rack of multiple compressors contained in a machinery room providing service to multiple display cabinets, involve more than 500000 plants worldwide (sales area around 500 m^2). They generally use direct expansion systems with large quantities of refrigerant charge. If the medium temperature services are devoted plants, the main used refrigerants are R-22 and R-134a in EEUU, and R-134a in Europe. If the medium temperature services are joined with the low temperature appliances, the most widespread refrigerants are R-22 and R-507A in EEUU, and R-404A and R-507A in Europe. These systems are prone to frequent leakage rates, especially through the mechanical joints. Among them, centralized systems are the ones with largest direct contribution to the global warming, because of their large refrigerant charges and high annual leakage rates, quantified to be between 15 to 25% (Schwarz et al., 2011).

In October 2016, the 28th Meeting of the Parties to the Montreal Protocol adopted the Kigali Amendment on hydrofluorocarbons, including the HFCs on the Montreal Protocol (UNEP, 2016). This is an historic agreement who aims to reduce the use and production of HFC worldwide. The Kigali Amendment's objective is to reduce the emissions of HFC gases over 80 billion tonnes of carbon dioxide equivalent (CO_{2e}) by 2050. It establishes a phase-down schedule of HFC substances based on the overall CO_{2e} direct emissions, so it depends on the GWP value of the HFC substances. This amendment follows the same action line as the F-Gas Regulation of Europe (European Commission, 2014), who has limited the GWP value of the substances that could be used in different refrigeration applications according to different time lines. Regarding commercial refrigeration, the most important limitations are the limit of GWP of 2500 for stationary equipment from 2020 on and the limit of GWP of 150 for multipack centralised refrigeration systems with rated capacity of more than 40kW from 2022 on, except for the primary circuits of cascade systems, which GWP limit has been fixed in 1500.

The previous agreements and regulations condition the use of high-GWP refrigerants in commercial refrigeration systems (Llopis et al., 2015). In fact, focussing in medium temperature applications from -10 to 10ºC, the use of the most widespread refrigerant in this sector, the R-134a with a GWP of 1301, will be limited. In the same trend of agreements and regulations, refrigerants manufacturers have designed different drop-in mixtures or pure synthetic fluids with lower GWP values that could be used instead of R-134a (Table 1) in different regions of the world with different horizon time lines.

Table 1. R-134a drop-in substitutes for medium temperature applications in commercial refrigeration

The first group of R-134a substitutes corresponds to mixtures of refrigerants between R-134a and HFOs [R-1234yf and R-1234ze(E)] with GWP values below 600 designed to be drop-in substitutes of R-134a in existing plants. The main characteristic of this group is that they present an A1 security classification (ASHRAE, 2016), thus being designed to replace R-134a in centralized systems with direct expansion evaporators. One mixture in this group is R-450A, corresponding to a mixture of R-134a and R-1234ze(E) with mass proportions of 42 and 58%, respectively, obtaining a GWP of 547. Schultz and Kujak (2013) tested this mixture in a 230-ton water cooled screw compressor water chiller under standard conditions (water production at 6.67ºC and cooling water at condenser at 29.44ºC). They measured reductions in capacity between 12 to 15% and reductions in EER from 1 to 4% regarding R-134a. Mota-Babiloni et al. (2015) analysed the drop-in process of R-450A in a refrigeration plant using an open-type compressor, shell-and-tube heat exchangers and thermostatic expansion valve in evaporating levels from -13 to 7ºC and condensing temperatures from 27 to 57ºC. Compared to R-134a, they measured reductions in capacity of 6% in average and average increments in COP of 1%. The second mixture in this group is R-513A, mixture of 56% of R-1234yf and 44% R-134a by mass, with a GWP of 573. Kontomaris et al. (2012) tested R-513A as drop-in of R-134a in a 1969 kW centrifugal chiller, measuring 0.6% increase in energy consumption. Shapiro D. (2012) with a commercial bottle cooler/freezer driven by a 270W hermetic compressor and capillary tube performed pull-down and half-pull down tests using two set points in the thermostat (-5.6 and 3.3ºC) for two environment temperatures (23.9 and 26.7 ºC). He measured variations of R-513A regarding R-134a in capacity from -7.8 to 2.4% and of COP between 0.1 to -9.7%. Schultz and Kujak (2013), in the chiller described above measured that capacity of R-513A matched that of R-134a but measured reductions of COP between 3 to 4%. And recently, Mota-Babiloni et al. (2017) tested R-513A in an R-134a plant driven by a 550W hermetic compressor, water brazed-plate heat exchangers and thermostatic expansion valves. They measured increments in COP of 5% in average and slight increments on cooling capacity.

The second group of R-134a substitutes corresponds to pure refrigerants with GWP below 150, chosen limit by the European Regulation (2014) for centralized systems in Europe. Within this group, the hydrocarbons could be included, however due to its A3 security classification, they are not considered for large systems. The options to replace R-134a currently analysed are the HFC-152a, with a GWP of 137, and HFOs R-1234yf and R-1234ze(E), with GWP below the unit. The low GWP values of these fluids are attached to flammability characteristics. R-152a obtains A2 and HFOs A2L Ashrae 34 security classifications (ASHRAE, 2016). Due to their reduced security classification, taking into account the current security regulations, they are not considered as replacements of R-134a in centralized direct expansion system but only in indirect configurations. Regarding the experimentation with these fluids as substitutes of R-134a: In commercial equipment with low refrigerant charge, Shapiro D. (2012), Schultz and Kujak (2013) and Aprea et al. (2016a, 2016b, 2017a) and Sanchez et al. (2017a) tested R-1234yf and R-1234ze(E) as R-134a replacements in commercial equipment with hermetic compressors. Their results indicate reductions in capacity in relation to R-134a, especially for the R-1234ze(E), and slight modifications on the COP. Also with hermetic compressors, Cabello et al. (2015) tested R-152a as R-134a substitute, measuring reductions in capacity from 1.13 to 9.75% but increases in COP around 11.7% and of 13.2% when adding an internal heat exchanger. Experimentation with larger systems was performed by Kabeel et al. (2016) with R-1234ze(E) in a walk-in cold room with a semihermetic compressor and thermostatic expansion valve. They measured 2-13% reduction in capacity but COP increments from 7 to 33% in relation to R-134a. Also, Mota-Babiloni et al. (2014) using an open-type compressor measured reductions in capacity from 9 to 39% and reductions in COP from 6 to 7% regarding the operation with R-134a with the substitutes R-1234yf and R-1234ze(E). Finally, concerning large systems, Cabello et al. (2017) analysed the R-134a drop-in process with R-152a in the primary circuit of a HFC/CO_2 cascade, obtaining similar energy behaviour than with R-134a.

According to the published energy tests, it can be said that both groups of refrigerants are able to operate in existing plants designed for R-134a. However, only the drop-in fluids with A1 security classification (R-450A and R-513A) could be considered now as reduced GWP substitutes of R-134a in centralized existing systems. As described in the literature review, these fluids have been only tested in steady-state conditions in refrigeration plants or in water chillers with real operation. Accordingly, this work aims to contribute presenting the real analysis (transient operation) of these fluids in a centralized commercial refrigeration system for medium temperature. In this case, we present the experimental evaluation of R-450A and R-513A as R-134a and R-507A and R-404A drop-in substitutes in a medium temperature commercial refrigeration system. Only R-507A is included in the comparison since its performance is equivalent to that of R-404A (Arora and Kaushik, 2008; Llopis et al., 2010). The system, driven by a semi-hermetic compressor, uses a direct expansion system with an electronic expansion valve adapted to the pressure-temperature characteristics of each refrigerant mixture. The paper presents the analysis of the results of 24-hour energy tests maintaining the product temperature of a commercial fresh food cabinet (T = 2ºC) for three dissipation inlet water temperatures (23.2, 32.8 and 43.6ºC) using on/off operation of the compressor regulated by pressure switches. For the analysis, the refrigerants R-450A and R-513A have been contrasted to R-134a, direct replacement option, and with R-507A, mixture of refrigerant largely used in Europe in commercial booster systems working with two temperature levels. This last case illustrates a possible retrofitting of R-404A and R-507A existing systems with R-450A or R-513A.

In the paper, first, an initial thermodynamic comparison of the refrigerants is presented. Then, the experimental plant used for the tests and its measurement system is described. Finally, the results of the energy consumption tests are presented and discussed.

2. Thermodynamic properties and theoretical performance

R-513A corresponds to a azeotropic mixture of R-1234yf and R-134a with mass proportions of 56 and 44%, respectively, whereas R-450A is a non-azetropic mixture of R-1234ze(E) and R-134a with mass proportions of 58 and 42%, respectively. Both refrigerants, with an A1 Ashrae 34 security classification (ASHRAE, 2016), have been designed as reduced-GWP drop-in substitutes of R-134a in medium temperature applications, thus both are compatible with POE lubricants. R-513A offers a 56% reduction in GWP respect to R-134a and of 86% respect R-507A, whereas R-450A a 58% reduction about R-134a and of 86.8% with respect to R-507A.

Table 2 summarises the main thermodynamic properties and Figure 1 the pressure-enthalpy diagram of the considered refrigerants. R-134a, R-513A and R-450A have similar liquid saturation lines, but the drop-in fluids present a slight reduction in the latent heat of phase change. For an evaporating level of -14ºC (average evaporating temperature of R-134a in the experimental tests), R-513A has a reduction in latent heat of phase change of 11.2% and R-450A of 4.9% regarding R-134a. Differences of the vapour saturation line (Figure 1) and pressure-temperature relation of phase change influence the behaviour of the expansion valves. If they are thermostatic, important differences could be obtained if they are not adjusted; nonetheless, if the valves are electronic they could be reprogrammed for the new refrigerant mixtures. That is of special importance if R-513A and R-450A are to replace R-404A/R-507A, since pressure-temperature relation of phase change largely differs among them. Considering the glide, all the refrigerants are near-azeotropic fluids, being the maximum glide of 0.6K for R-450A. Another important difference is the vapour specific volume. Differences of specific volume of saturated vapour at -14ºC regarding R-134a are of -18.1% for R-513A and of +7.8% for R-450A. These differences introduce modifications in the refrigerant mass flow rate, an increase for R-513A and a reduction for R-450A, although they would be inside the admissible variation range of the expansion devices. However, when the specific suction volume is contrasted with that of R-507A for the same evaporating level, the variation of R-513A is of +93.9% and of R-450A of +155.1%. These large differences will show large reductions in the refrigerant mass flow rate, affecting greatly the R-507A expansion valves that will result large for the operation with R-513A and R-450A, and the R-507A compressors that will become larger for the operation with the drop-ins.

Table 2. Physical, environmental and safety characteristics of R-134a, R-513A, R-450A and R-507A (IPCC, 2013; Lemmon et al., 2013)

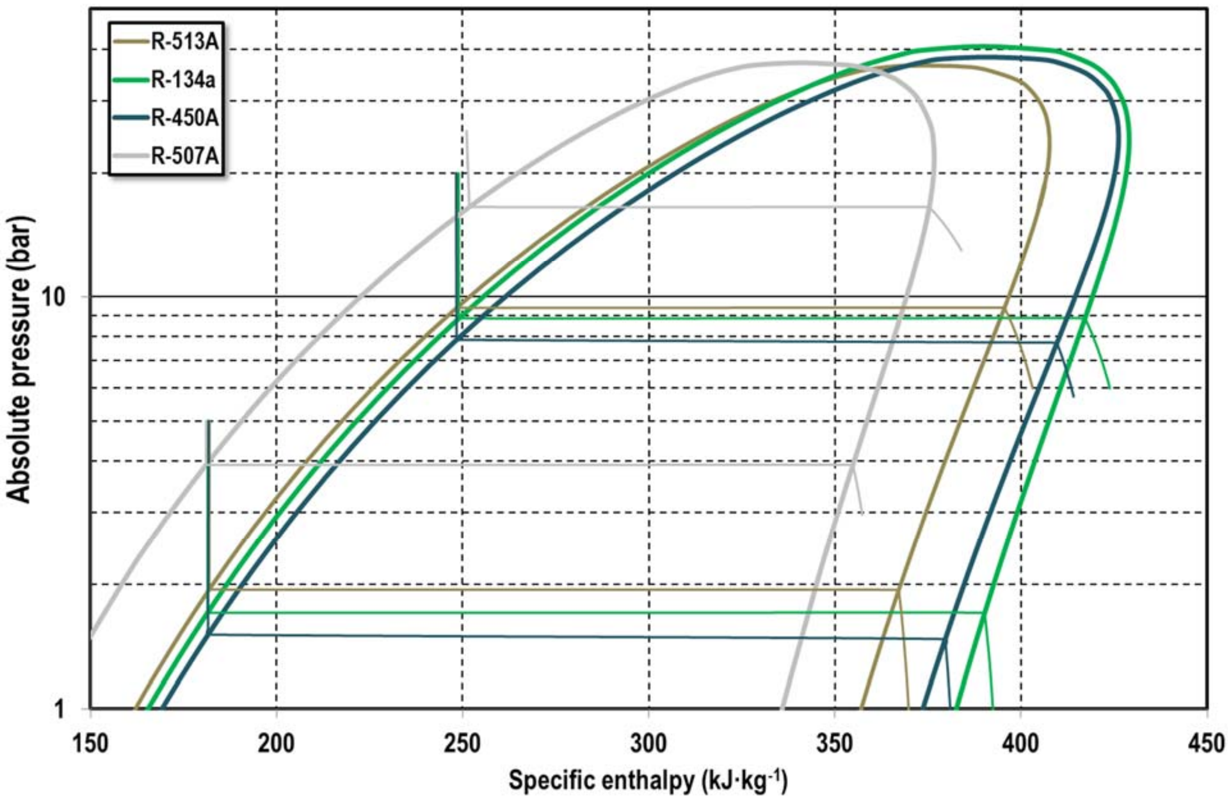

Figure 1. Pressure-enthalpy diagrams, 35 and -14ºC isotherms of R-134a, R-513A, R-450A and R-507A

Figure 2 summarizes the theoretical comparison of the main operating parameters of an ideal single-stage vapour compression system operating with the four refrigerants at an evaporating level of -14ºC and three condensing levels (25, 35 and 45ºC). For the calculation, to consider the glide effect of R-450A, the condensing pressure was evaluated for a 50% vapour title, Eq. (1), and the evaporating pressure using the average enthalpy value at the evaporator, Eq. (2). This criteria was recommended by Radermacher and Hwang (2005). With these conditions, the volumetric cooling capacity, Eq. (3), and the COP, Eq. (4), were computed assuming an ideal compression cycle. Also, Figure 2 presents the comparison of the isentropic discharge temperature and the inlet vapour title at the evaporating process. All the thermodynamic properties were evaluated using Refprop 9.1 (Lemmon et al., 2013).

$$P_K = f(T_K, x_v = 0.5) \tag{1}$$

$$P_O = f\left(T_O, \frac{h_{l,PK} + h_{v,PO}}{2}\right) \tag{2}$$

$$VCC = \frac{h_{v,PO} - h_{l,PK}}{v_{v,PO}} \tag{3}$$

$$COP = \frac{h_{v,PO} - h_{l,PK}}{h_{dis,S}(P_K, s_{v,PO}) - h_{v,PO}} \tag{4}$$

On the one hand, R-513A parameters variation regarding R-134a are slight, R-513A has higher VCC (4.3% at 25ºC, 2.6% at 35ºC and 0.3% at 45ºC) and reduction in COP (-2.5% at 25ºC, -3.7% at 35ºC and -5.4% at 45ºC). However, regarding R-507A the differences are larger. R-513A has lower VCC than R-507A (-41.9% at 25ºC, -40.3% at 35ºC and -37.7% at 45ºC) but offers increments in COP (5.3% at 25ºC, 8.0% at 35ºC and 12.7% at 45ºC). On the second hand, R-450A differences from R-134a are reductions in the VCC (-13.5% at 25ºC, -14.1% at 35ºC and 14.7% at 45ºC) and in COP (-0.6% at 25ºC, -0.9% at 35ºC and -1.3% at 45ºC). Again, R-450A against R-507A presents large differences in VCC (-51.8% at 25ºC, -50.0% at 35ºC and

-47.0% at 45ºC) but increments in COP (7.4% at 25ºC, 11.2% at 35ºC and 17.6% at 45ºC). From a theoretical approach, it is observed that R-450A and R-513A will offer small variations on capacity, positive for R-513A and negative for R-450A, and slight reduction in COP for both, trends in agreement with the experimental results published in literature (Mota-Babiloni et al., 2017; Mota-Babiloni et al., 2015). Nonetheless, regarding R-507A, both substitutes will suffer a drastic reduction in capacity but improvements in COP. Another important aspect to be considered in real plants, especially in direct expansion systems such as cabinets in supermarkets, is the vapour title at the evaporator inlet, which will affect the final evaporating level in the heat exchanger and thus the suction pressure of the compressor. In average, R-513A presents an increment of 11% of the vapour title at the inlet of the evaporator regarding R-134a and a decrement of 12% respect to R-507A. That would mean for a specific evaporator that the average evaporating level of R-513A would be lower than of R-134a and higher than R-507A. R-450A presents 3.4% difference of vapour title regarding R-134a and -18% respecting R-507A, being the expected trend similar. Finally, the theoretical analysis show that both R-513A and R-450A would present slight reductions in the compressor discharge temperature.

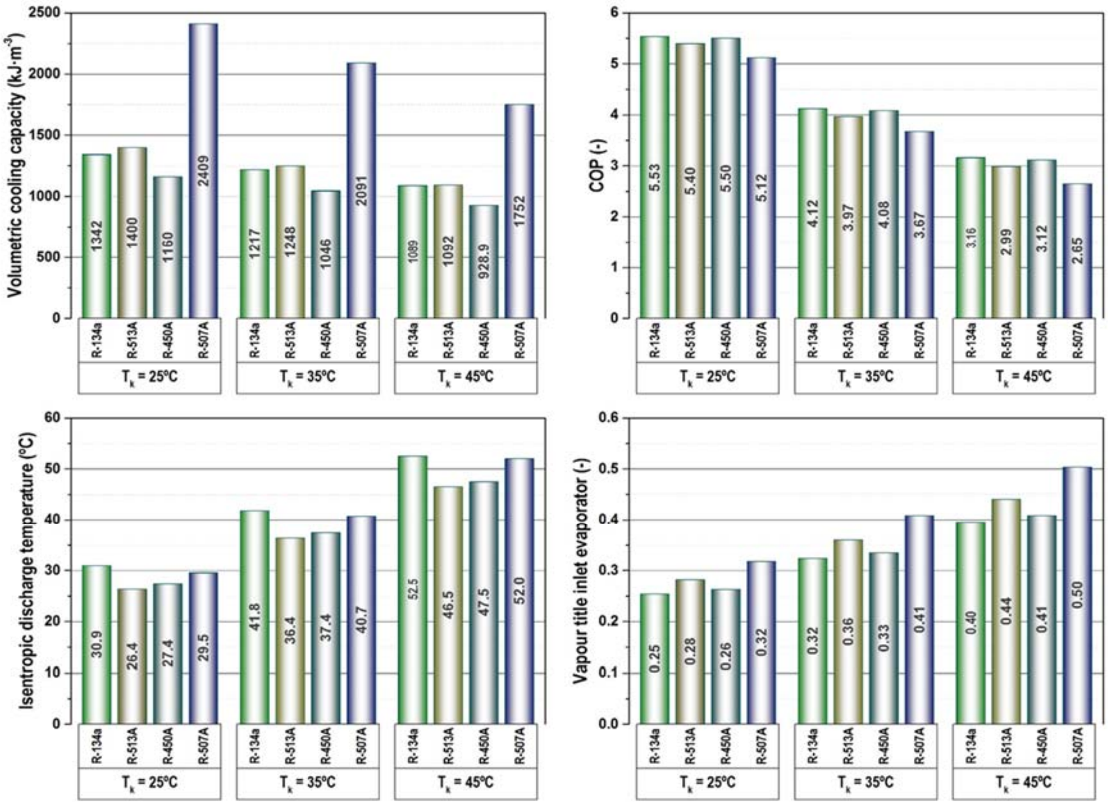

Figure 2. Thermodynamic VCC, COP, isentropic discharge temperature and vapour title at inlet of evaporator (T_O=-14ºC)

3. Test methodology and experimental set-up

This section describes the test methodology used to evaluate the refrigerant mixtures as well as the experimental set up and uncertainties of the measurement system.

3.1. Test methodology

Refrigerants were evaluated using 24-hour energy consumption tests in a small-scale supermarket with a medium temperature cabinet, equal to that used by Sánchez et al. (2017b). The experimental plant is described in subsection 3.2 and the measurement system in subsection 3.3. This test considered stable operation of the plant during 24-hours where the average product temperature inside the cabinet was kept at 2ºC. The energy evaluation was performed for three water dissipation temperatures in the condenser (23.3, 32.8 and 43.6ºC) covering a wide range of operating conditions.

3.2. Experimental plant

The experimental plant used for the evaluation of the refrigerant mixtures is schematized in Figure 3. It corresponds to a single stage cycle driven by a semihermetic compressor, a condenser and liquid receiver that serves a commercial vertical cabinet for fresh food with doors placed inside a climatic chamber. The heat rejection of the system is performed using a water loop, as described by Sanz-Kock et al. (2014).

Semihermetic compressor, designed for R-134a operation, has a nominal power of 1.5kW, a displacement of 6.51 $m^3 \cdot h^{-1}$ at 1450 rpm and uses POE BSE32 lubricant oil. Condenser is a brazed plate heat exchanger (B25-TH-40) with a heat transfer area of 2.39 m^2. Medium temperature cabinet is a glass door vertical type with dimensions: 1875 mm long, 2071 mm height and 890 mm width. It incorporates a controller that regulates operation of the heat exchanger (customizable electronic expansion valve, NTC sensor at the pipe surface and a pressure gauge at the exit of evaporator) and the defrosting period (every 8 hours). The evaporator is finned-tube of 1520 mm length, tube diameter of 5/8'', with three circuits (1st 13.13 m, 2nd 13.09 m and 3rd 13.13 m of tube length) with a total internal volume of $7.8 \cdot 10^{-3}$ m^3. It has 195 aluminium fins (130 x 300 mm) of 0.3 mm thickness. Defrosting is made with 2000W electrical resistors. End of defrosting is controlled by the temperature prove at the surface of the evaporator and finishes when it reaches 5ºC. Behaviour of the cabinet is analysed with one combined temperature and humidity sensor, 1 wattmeter and 5 M-test packages using internal T-type thermocouples. The climatic chamber is class 3 (ISO 23953-2:2015, 2015). It maintains indoor temperature and relative humidity using PID regulators.

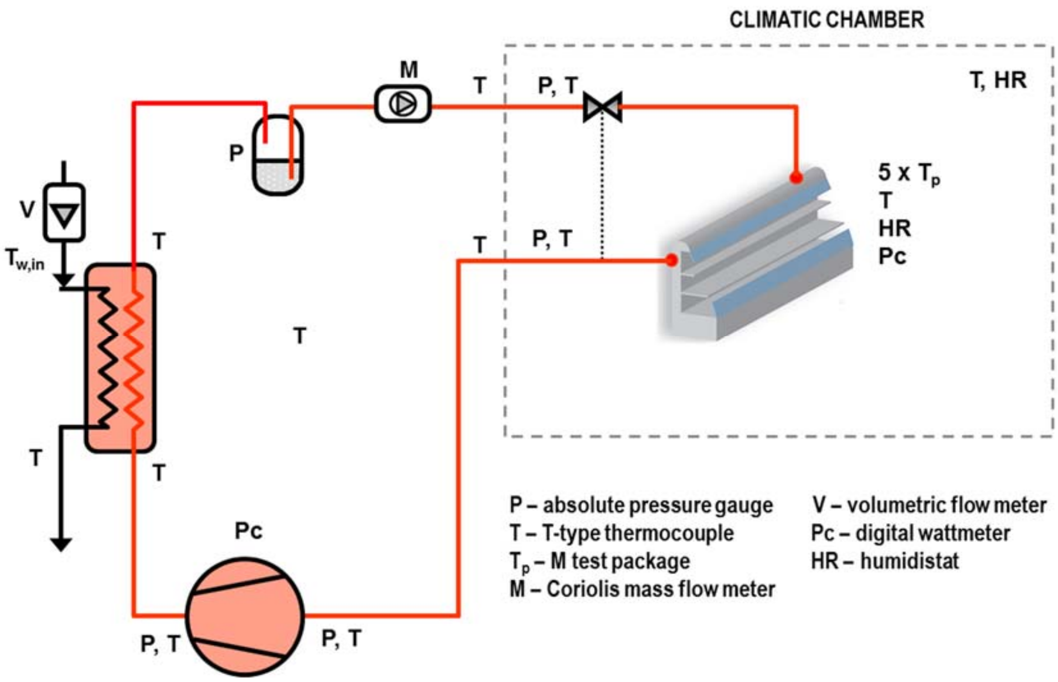

Figure 3. Scheme of the experimental plant and location of measurement devices

3.3. Measurement system and uncertainties

The plant is fully instrumented to be able to measure the main energy parameters of operation. The measurement devices and their uncertainty are detailed in Table 3. The refrigeration cycle incorporates 8 T-type thermocouples, 3 pressure gauges, 1 Coriolis mass flow meter, a volumetric flow meter and a digital wattmeter. The cabinet uses 2 T-type thermocouples, 2 pressure gauges, 1 digital wattmeter, a combined humidity-temperature sensor and 5 M-test packages according to ISO 15502 for measuring the product temperature. The climatic chamber uses a combined humidity-temperature sensor and another T-type thermocouple is used for measuring the environment temperature. All sensors are gathered using two cRIO-9074 data acquisition systems.

Table 3. Number of sensor elements and uncertainties

4. Energy consumption tests

To evaluate the behaviour of the plant with the different refrigerants, 24-hour energy tests were performed. The references for the tests were the average temperature of the M-test packages inside the cabinet, which was set to 2ºC, and the water inlet temperature to the condenser to perform heat rejection, which was set to 23.3, 32.8 and 43.6ºC, covering a wide range of operating conditions.

4.1. Test summary and test conditions

The plant was operated with the compressor at its nominal speed (1450rpm) using ON/OFF control strategy using the pressure switches of the plant. For each refrigerant, the pressure switches were adjusted to obtain a low-cut temperature of -25.0ºC and a cut-in temperature of -4.5ºC. The superheat set point at the evaporator was set to 4.5K. The expansion valve controlled the superheat using a NTC temperature measurement and a pressure measurement at the exit of the evaporator. The driver of the expansion valve was customized for each refrigerant using the pressure-temperature saturation relation calculated with Refprop V.9.1(Lemmon et al., 2013). Defrosting period was set to one each eight hours, ending when the temperature prove at the surface of the evaporator reached 5ºC. The set point of the cabinet was adjusted for each refrigerant and each test condition to obtain an average product temperature of 2ºC (arithmetic mean of temperature of the five M-test packages). The cabinet was placed inside the climatic chamber, where 25ºC of dry bulb temperature and 55% of relative humidity were retained. The external conditions were established using a loop working with water running at a constant volumetric flow rate of 1 $m^3 \cdot h^{-1}$ and different inlet water temperatures to the condenser (23.3, 32.8 and 43.6ºC).

Table 4 summarizes the test conditions during the 24-hour tests at each water inlet temperature. Deviations during the test represent the standard deviations of the parameters. Figure 4 illustrates the 24-hour test for R-513A at a water inlet temperature to the condenser of 32.8ºC.

Table 4. Reference parameters of the evaluation during 24-hour tests

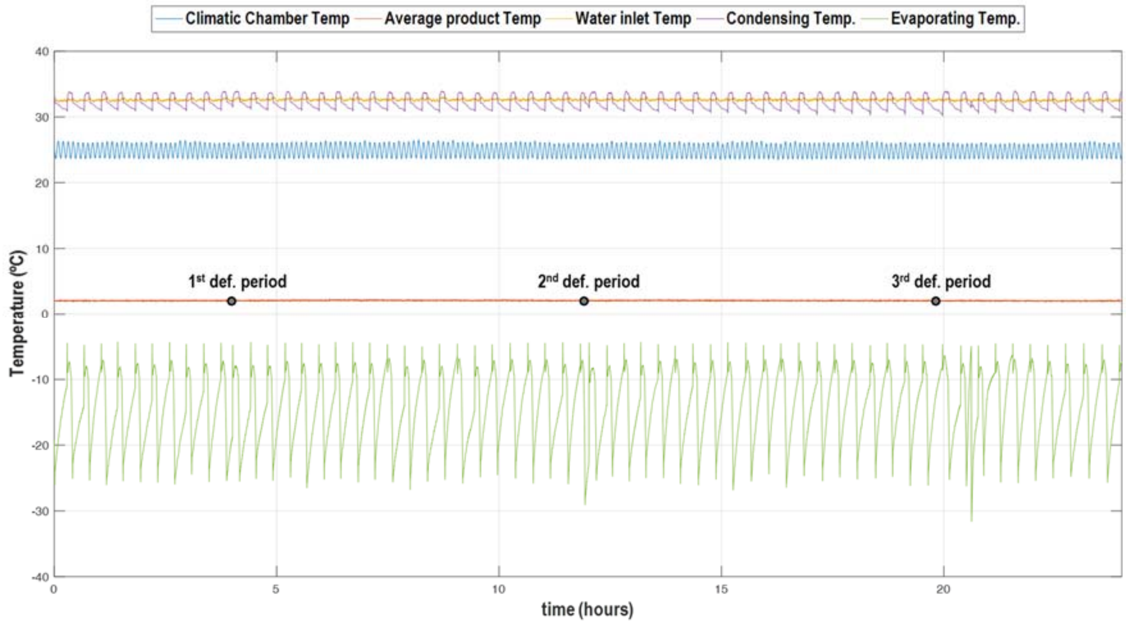

Figure 4. Climatic chamber, average product, water inlet, condensing and evaporating temperatures for R-513A at $T_{w,in}$=32.8ºC during 24-hour test

Following, the temperature indicators in the 24-hour tests are detailed in subsection 4.2 and the operation time and energy indicators in subsection 4.3.

4.2. Temperature and pressure indicators

Figure 5 presents the average values of the condensing and evaporating temperatures during 24-hour tests for the four refrigerants at the different test conditions. The condensing temperature corresponds to the average value calculated using pressure discharge and vapour title of 50%, Eq. (5), only when the compressor was on. The uncertainty of averaged condensing temperature is of ±0.44K. The average value of evaporating level, computed using pressure at the exit and average enthalpy value in the evaporator, Eq. (6), is integrated during 24-hour test, since the fan of the cabinet is continuously in operation. The uncertainty of averaged evaporating temperature is of ±0.12K No significant differences were observed regarding the condensing temperature; however, important variations were measured concerning the evaporating level. R-513A showed a reduction of 3.1K at T_w=23.3ºC regarding R-134a but identical evaporating levels at the other conditions. R-450A presented increments between 2 to 4.7K in the evaporating level. Respect to R-507A both mixtures presented important increments on the evaporating temperature, between 2.4 to 4.1K for R-513A and from 6.9 to 9.8K for R-450A.

$$T_K = f(P_{dis}, X_v = 50\%) \quad (5)$$

$$T_O = f\left(P_{O,out}, h = \frac{h_{O,in} + h_{v,Po,out}}{2}\right) \quad (6)$$

Considering the operating pressures of the system, Figure 6 presents the data about suction pressure, Figure 7 of discharge pressure and Figure 8 about compression ratio, evaluated with Eq. (7). These figures present the variation range of the parameter in a bar graph and the average value during the 24-hour test when the compressor is in operation in digits. R-513A and R-450A present similar values and variation range of suction pressure (Figure 6) than R-134a and approximately half value and half variation range regarding R-507A. In discharge pressure, R-513A presents similar discharge pressure than R-134a, but R-450A offers a reduction of discharge pressure in average of 1.21 bar. When compared to R-507A, both mixtures presents approximately half value of pressure. Those data indicate that both refrigerant substitutes will operate correctly in an R-134a system concerning pressure, and will present a strong difference when used with R-507A, having implication in the pressure losses and in the operation of the expansion valves an in the pressure switches regulation. Nonetheless, regarding the pressure ratio, average value during the 24-hour test when the compressor is in operation, no important differences have been measured among all the refrigerants, as depicted in Figure 8.

$$t = \frac{P_{dis}}{P_{suc}} \tag{7}$$

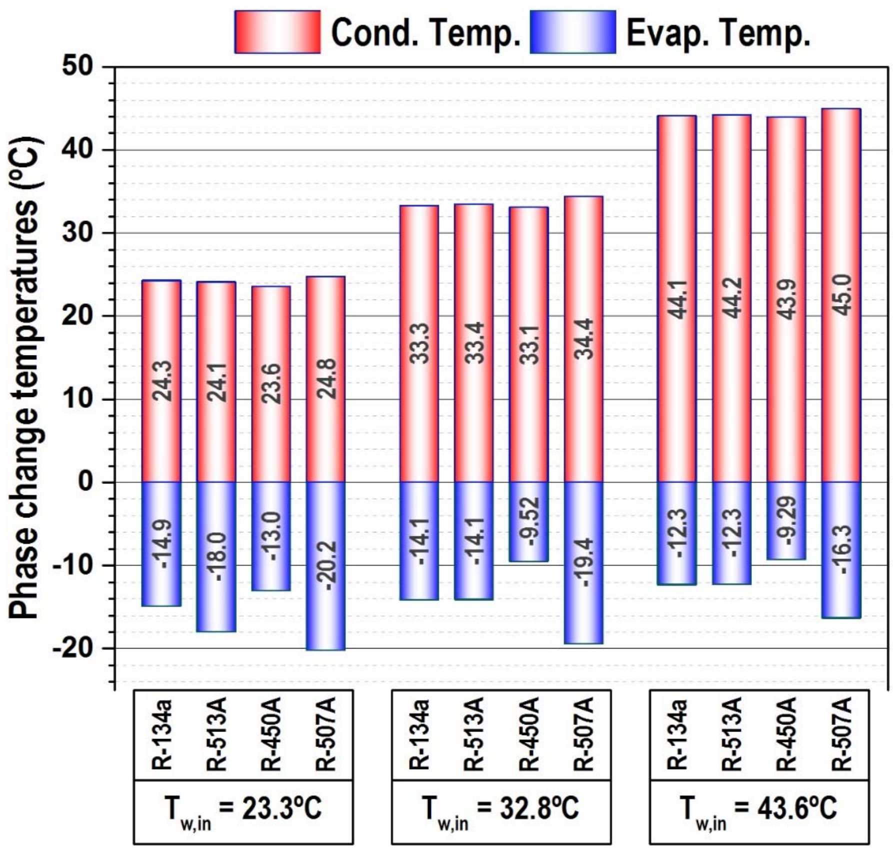

Figure 5. Average evaporating and condensing temperatures 24h test

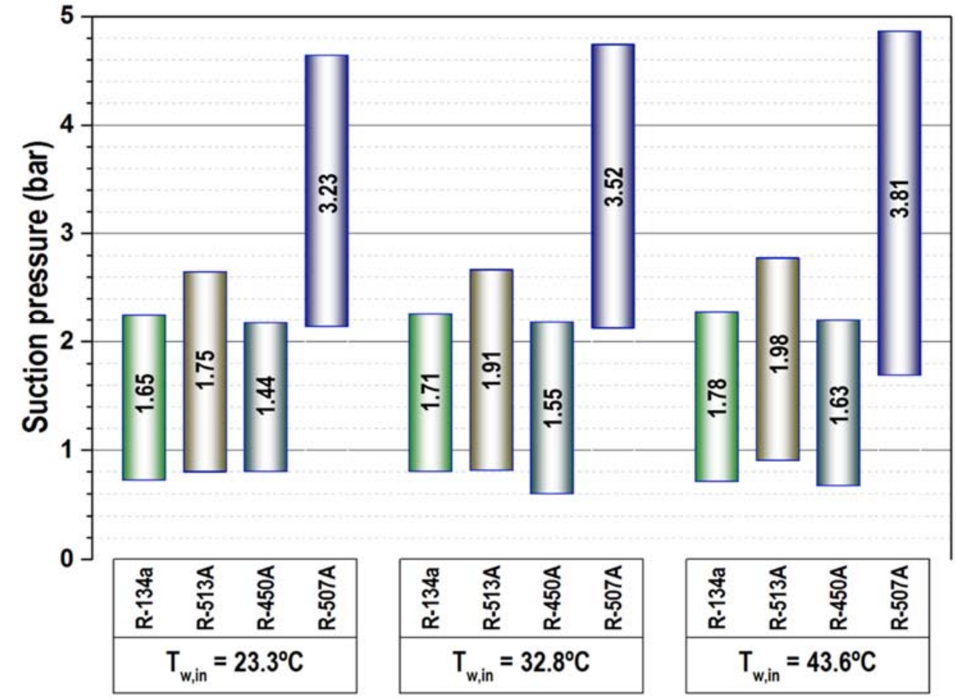

Figure 6. Average (value), maximum and minimum suction pressure during 24h test

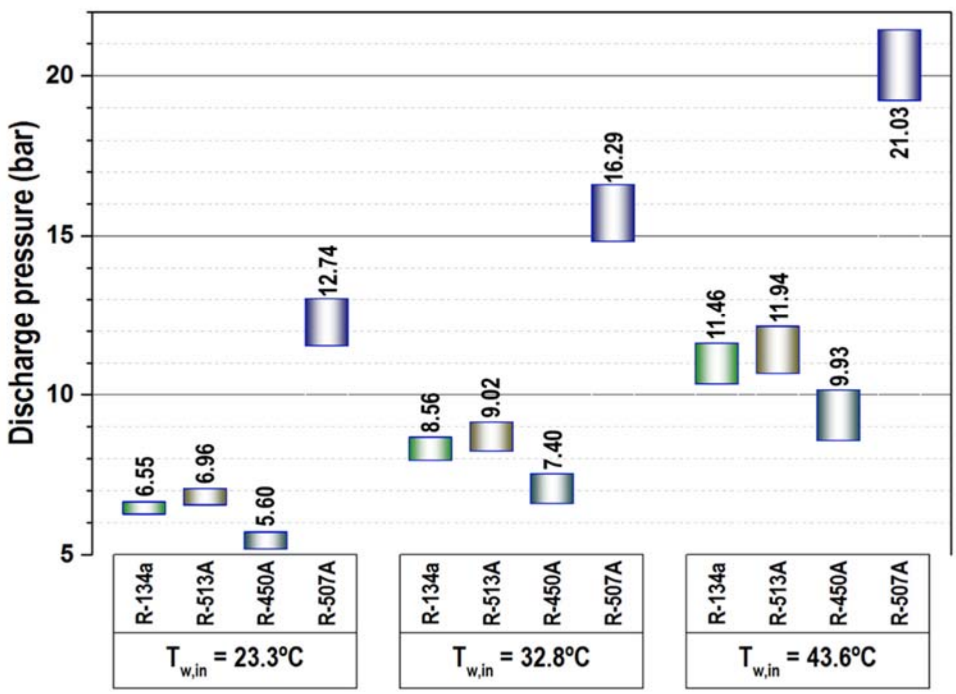

Figure 7. Average (value), maximum and minimum discharge pressure during 24h test

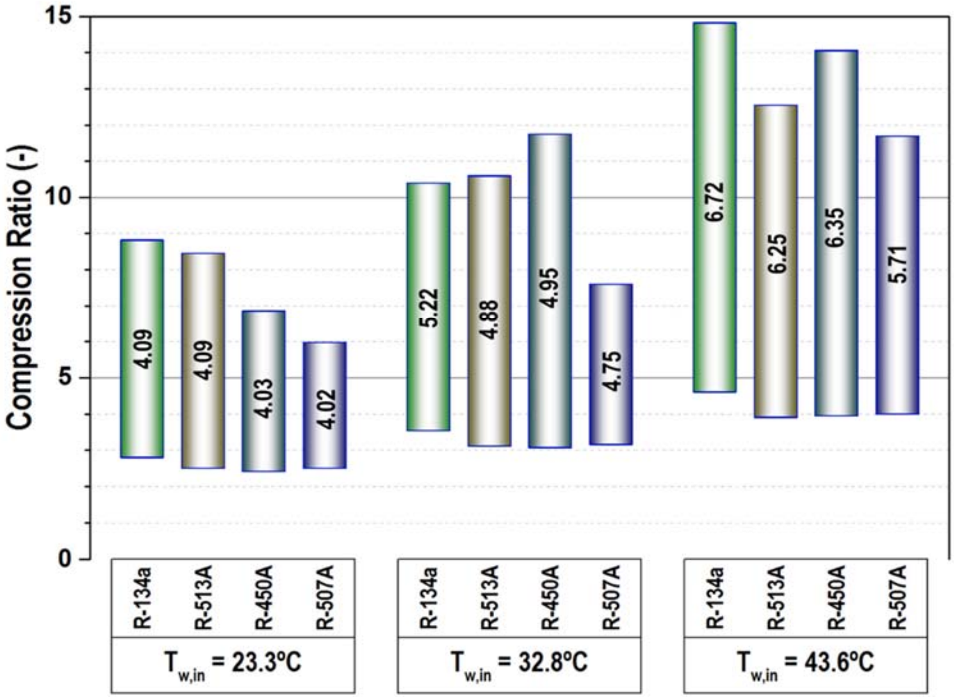

Figure 8. Average (value), maximum and minimum compression ratio during 24h test

Considering temperature indicators, Figure 9 presents the average temperature difference between the evaporating temperature with the inlet air to the evaporator and with the average product temperature. These temperature differences drive the heat transfer from the refrigerant to the inlet air of the cabinet and to the product. On the one hand, it has been measured that R-513A needs 3K plus difference in relation to R-134a at the lowest condensing levels but equivalent at the highest heat rejection levels. However, R-450A presents in average a reduction 3.2K, derived from a higher evaporating level. Our hypothesis is that these differences could be caused by the variations of the inlet vapour title to the evaporator (higher for R-513A and lower for R-450A regarding R-134a, Figure 2), or modifications of the overall heat transfer performance of the evaporator. However, it is not possible in this system to extend the analysis, since the plant is continuously in transient operation. On the second hand, both refrigerant mixtures present reductions of these temperature differences from R-507A, in the case of R-513A a reduction in average of 3.9K and for R-450A a reduction in average of 8K. That indicates that the performance of the evaporator with these refrigerant mixtures increases regarding with the use of R-507A. A hypothesis that justifies this last comment is that R-450A and R-513A have a reduced inlet vapour title to the evaporator. Furthermore, it must be said that the evaporator is an R-134a design. Considering the compressor's discharge temperatures, shown as average value when the compressor is in operation in Figure 10, it can be said that no important differences have been observed between the refrigerants. The refrigerant mixtures R-513A and R-450A present a slight reduction of the discharge temperature regarding R-134a and R-507A, both compatible with the lubricant oil.

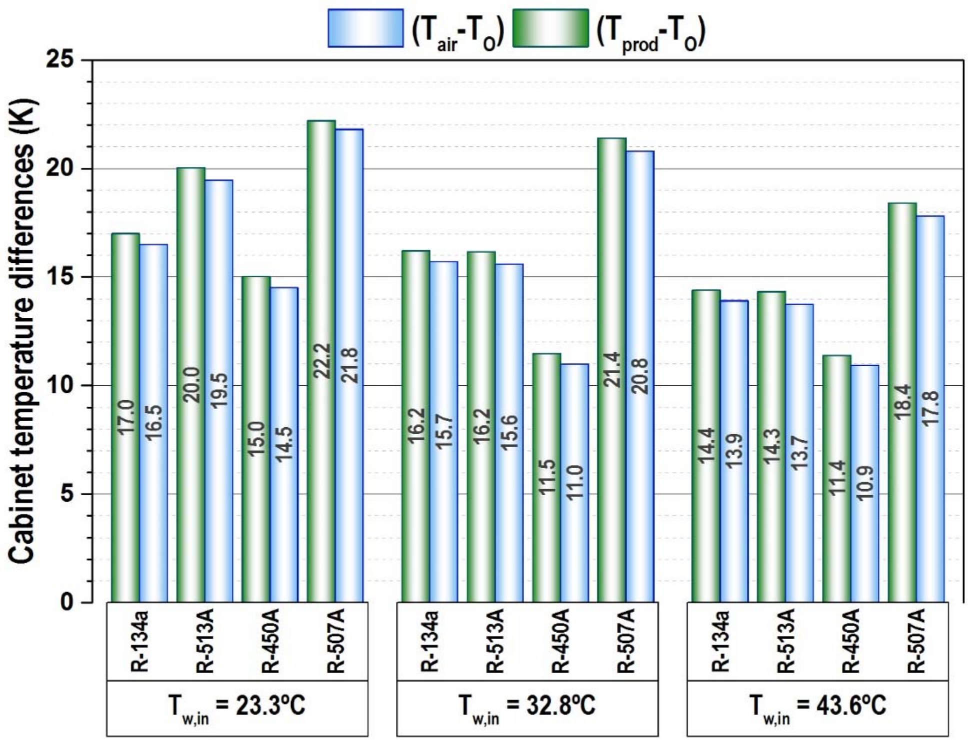

Figure 9. Average temperature differences between the evaporating temperature and air inside the cabinet and product temperatures

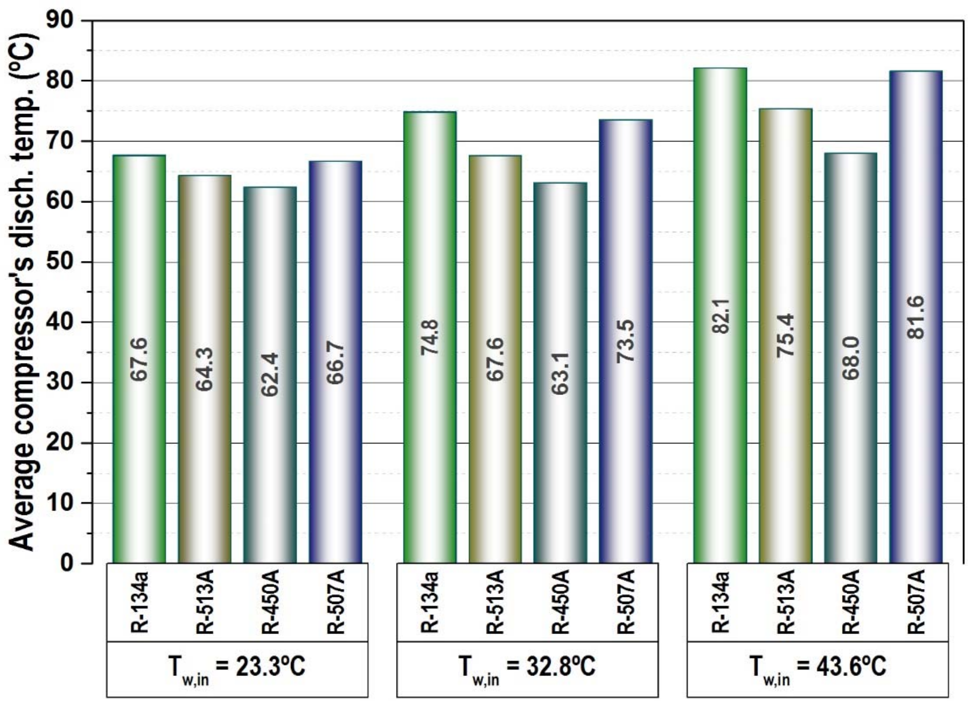

Figure 10. Average compressor's discharge temperature (Compressor ON)

4.3. Energy indicators

Energy consumption is a relation between the operating time and the average power consumption of the elements, being both needed to be compatible between the refrigerants to undertake a drop-in or substitution process. Figure 11 details the fraction time of compressor's operation and expansion valve opening time during the 24-hour tests. R-513A presents practically an identical compressor's operation factor than R-134a, whereas R-450A is in average a 5.8% more time in operation, because of its highest specific vapour volume (Table 2). Compared to R-507A, R-513A is in average in operation 10.9% more time and R-450A 17.6%. Again, these increments are in relation with the increased specific vapour at compression suction. Considering the expansion valve's operation fraction time, it has been observed that R-134a, R-513A and R-450A present slight differences among them, thus the expansion valve being compatible among them. However, the operation fraction time in relation with R-507A differs in average 4% with R-513A and in 6.7% with R-450A. The largest operation time is a relation of the latent heat of phase change. Regarding the defrosting time of the evaporator, no appreciable differences have been measured among the four refrigerants.

Following, Figure 12 presents the average power consumption of the compressor, only when the compressor is in operation and of the cabinet during the 24-hours including three defrosting periods. It has been measured a similar power consumption of the cabinet for all the refrigerants and all the external operating conditions. R-513A presents in average a 4.4% more compressor power consumption than R-134a and a reduction of 38.2% with reference to R-507A. R-450A has in average a reduction of 8.3% respect to R-134a and a reduction of -45.3% in relation to R-507A. In both cases, there is a reduction in the power consumption, being the R-134a or R-507A compressors compatible with the refrigerant mixtures.

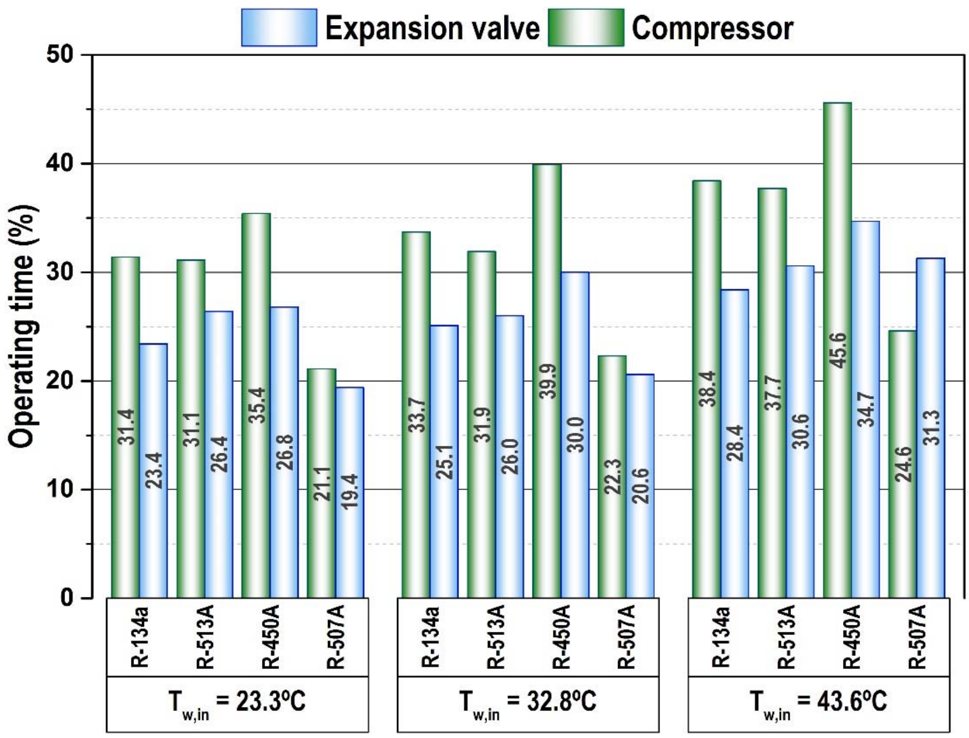

Figure 11. Operating time percentage of compressor and expansion valve during 24-hour test

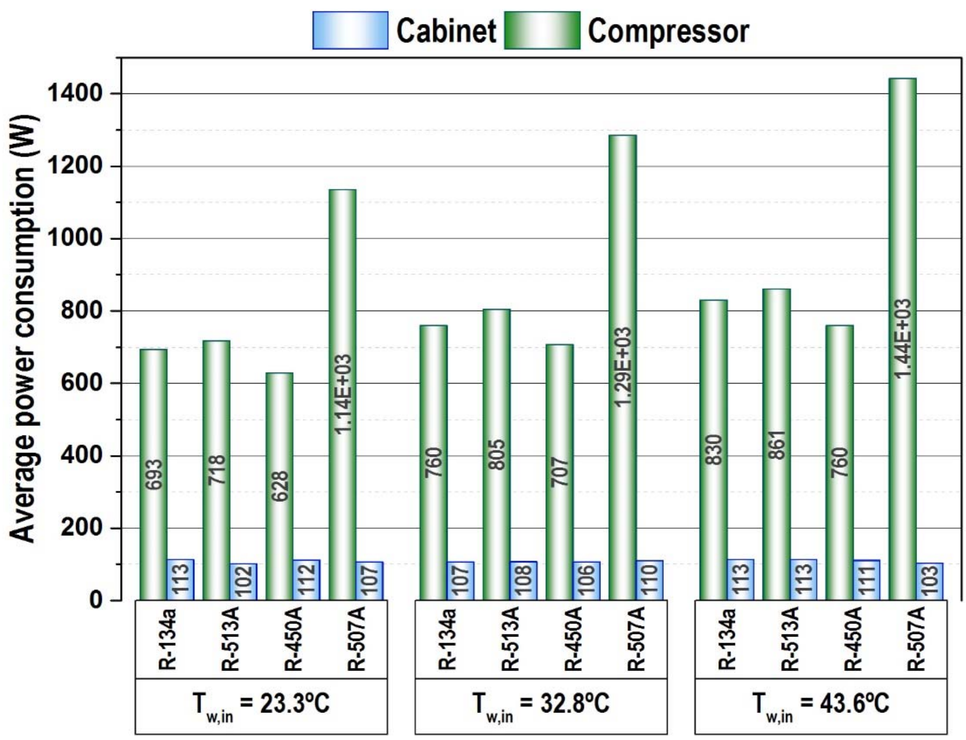

Figure 12. Average compressor's and cabinet's power consumptions during 24-hour test

Finally, to compare the system with the refrigerants from the point of view of energy consumption, the energy consumption (kWh) of each element has been calculated from the power consumption measurements and operating time according to Eq. (8) using a trapezoid integration method. In Eq. (8), 'i' represents each energy consumer, 'P_C' its power consumption and 'j' each sampled data. The expression is evaluated during the 24-hour test.

$$\boldsymbol{E_i = \frac{1}{36 \cdot 10^5} \cdot \int_0^{24h} P_{C,i}(t) \cdot dt = \frac{1}{36 \cdot 10^5} \cdot \sum_{j=1}^{24h} \left\{ \left[\frac{P_{C,i}(j) + P_{C,i}(j-1)}{2} \right] \cdot [t(j) - t(j-1)] \right\}} \quad (8)$$

The results are detailed in Figure 13 for the compressor and cabinet. The uncertainty of energy consumption is below 0.5%. Concerning the cabinet's energy consumption no appreciable differences have been measured among the refrigerants. Considering the compressor's, R-513A presents in average 1.3% increase regarding R-134a and 8.5% reduction respect to R-507A; and R-450A has in average 6.3% increase in relation to R134 and 3.7% reduction regarding R-507A. Finally, considering the total energy consumption of the system:

- R-513A energy consumption in relation to R-134a was of -1.6% at T_w=23.3ºC, 0.3% at T_w=32.8ºC and 1.2% at T_w=43.6ºC, and with reference to R-507A was of -6.1% at T_w=23.3ºC, -8.2% at T_w=32.8ºC and -4.4% at T_w=43.6ºC.
- R-450A energy consumption regarding R-134a was of 1.3% at T_w=23.3ºC, 6.8% at T_w=32.8ºC and 5.8% at T_w=43.6ºC, and in relation to R-507A was of -3.3% at T_w=23.3ºC, -2.3% at T_w=32.8ºC and 0.0% at T_w=43.6ºC.

The energy results indicate that R-513A presents similar energy consumption than the system using R-134a and a reduction in average of 6% using R-507A; whereas R-450A increases the energy consumption in 4.6% in average regarding R-134a and reduces it in 1.9% in average respect to R-507A.

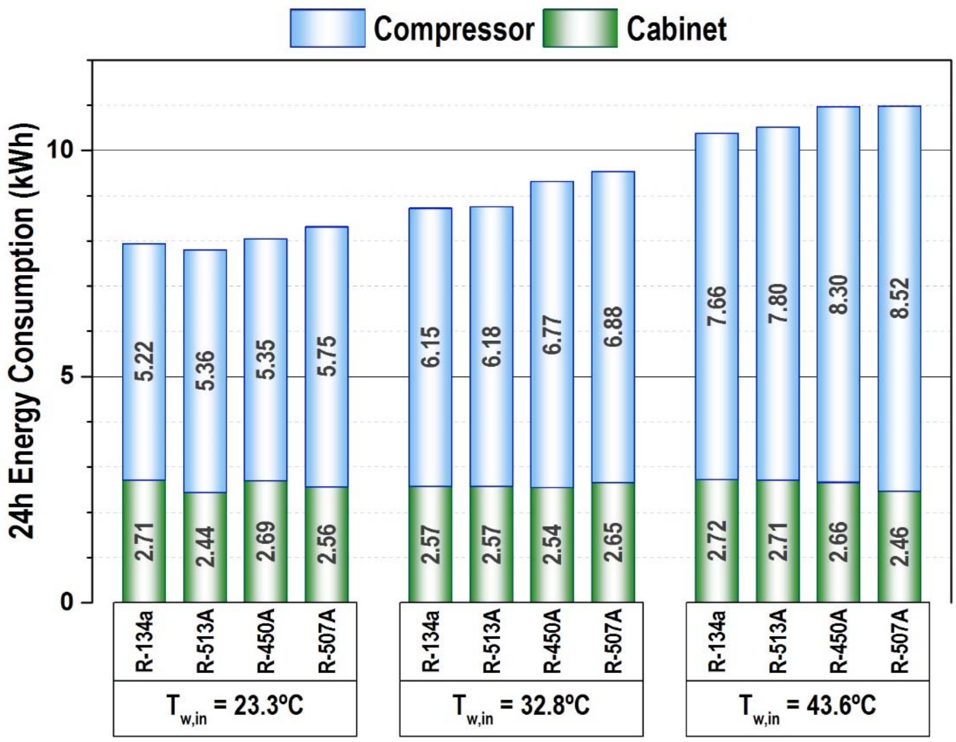

Figure 13. Energy consumption of compressor, cabinet and the system during 24-hour test

5. Conclusions

This work describes the experimental set up and test methodology to evaluate the refrigerants R-513A and R-450A as drop-in substitute of R-134a and as retrofit of R-507A in a medium temperature direct expansion vapour compression system for commercial used. The system and the refrigerants were tested in laboratory conditions at a product temperature of fresh food of 2ºC at three water inlet dissipation temperatures (23.3, 32.8 and 43.6ºC), thus covering the common conditions of a supermarket. The experimental evaluation covered 24-hour energy consumption tests.

From the energy consumption tests, it has been concluded that:

- No significant differences were observed about the condensing temperature, but the performance of the evaporator varied depending on the refrigerant. R-513A presented equivalent evaporating temperatures than R-134a (except at T_w=23.3ºC) and an increment in average of 3.9K in relation to R-507A. R-450A operated in average at 3.2K higher evaporating level than R-134a and in average 8K than R-507A. Compressor's discharge temperatures were equivalent for all the refrigerants.
- Compressor's operating fraction time of R-513A was practically equivalent than that of R-134a but increased in average 10.9% in relation to R-507A. R-450A was in operation in average 5.8% more time than with R-134a and 17.6% more than with R-507A. The operation time of the valve for R-513A and R-450A was similar to that of R-134a but increased in relation with R-507A operation.

- Compressor's power consumption of R-513A and R-450A was comparable and compatible to that of R-134a, whereas in relation to R-507A it was reduced in average of 38.2 and 45.3%, respectively.
- R-513A energy consumption compared to R-134a was of -1.6% at T_w=23.3ºC, +0.3% at T_w=32.8ºC and +1.2% at T_w=43.6ºC, and in relation to R-507A was of -6.1% at T_w=23.3ºC, -8.2% at T_w=32.8ºC and -4.4% at T_w=43.6ºC.
- R-450A energy consumption regarding R-134a was of +1.3% at T_w=23.3ºC, +6.8% at T_w=32.8ºC and +5.8% at T_w=43.6ºC, and in relation to R-507A was of -3.3% at T_w=23.3ºC, -2.3% at T_w=32.8ºC and 0.0% at T_w=43.6ºC.

Accordingly, it can be concluded from this experimentation that the use of the reduced GWP refrigerants R-513A (GWP=573) and R-450A (GWP=547) as drop-in replacements of R-134a (GWP=1301) is possible. They offer a slight increase on energy consumption, but they will offer important reductions of the direct emissions. And, regarding the retrofit of R-507A (GWP=3987) with these refrigerant mixtures, after replacing or adjusting the expansion valves, they will offer both a reduction in the indirect emissions and an important reduction of the direct effect. In both cases, the substitution is possible.

6. Acknowledgements

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TABLES

Table 1. R-134a drop-in substitutes for medium temperature applications in commercial refrigeration

REFRIGERANT NAME	Composition (% by mass)	GWP (AR4)	GWP (AR5)	SECURITY CLASSIFICATION	Energy tests
R-134a	HFC-134a (100.0)	1430	1301	A1	
150 < GWP < 1300					
R-450A	HFO-1234ze(E)/HFC-134a (58.0/42.0)	601	547	A1	(Schultz and Kujak, 2013) (Mota-Babiloni et al., 2015)
R-513A	HFO-1234yf/HFC-134a (56.0/44.0)	630	573	A1	(Kontomaris K. et al., 2012) (Shapiro, 2012) (Schultz and Kujak, 2013) (Mota-Babiloni et al., 2017)
GWP < 150					
R-152a	HFC-152a (100.0)	124	137	A2	(Cabello et al., 2015) (Sánchez et al., 2017a) (Cabello et al., 2017)
R-1234yf	HFO-1234yf (100.0)	-	<1	A2L	(Mota-Babiloni et al., 2014) (Sánchez et al., 2017a) (Aprea et al., 2016a, 2017)
R-1234ze(E)	HFO-1234ze(E) (100.0)	-	<1	A2L	(Shapiro, 2012) (Schultz and Kujak, 2013) (Mota-Babiloni et al., 2014) (Kabeel et al., 2016) (Aprea et al., 2016b) (Sánchez et al., 2017a)

	R-134a	R-513A		R-450A		R-507A	
Composition (% wt)	$C_2H_2F_4$	56.0%	HFO-1234yf	58.0%	HFO-1234ze(E)	50.0%	HFC-143a
		44.0%	HFC-134a	42.0%	HFC-134a	50.0%	HFC-125
Molecular weight (g·mol^{-1})	102.0	108.4		108.67		98.9	
Normal boiling point (ºC)	-26.07	-29.58		-23.36		-46.74	
Critical temperature (ºC)	101.6	94.9		104.5		70.6	
Critical pressure (bar)	40.59	36.48		38.22		37.05	
Glide at 35ºC[a] (K)	0.00	0.00		0.64		0.03	
Glide at -14ºC[a] (K)	0.00	0.05		0.62		0.01	
λ at T=35ºC[a] (kJ·kg^{-1}·K^{-1})	168.2	147.2		161.9		123.6	
λ at T=-14ºC[a] (kJ·kg^{-1}·K^{-1})	208.8	185.3		198.6		173.8	
v at T=-14ºC (m^3·kg^{-1})	0.116	0.095		0.125		0.049	
$GWP_{100\ years}$ (IPCC, 2014)	1301	573		547		3987	
ASHRAE 34 safety group	A1	A1		A1		A1	

[a]Glide and λ evaluated at pressure corresponding to the phase change temperature with a vapour title of 50%

Table 2. Physical, environmental and safety characteristics of R-134a, R-513A, R-450A and R-507A (IPCC, 2013; Lemmon et al., 2013)

	Temperature	Pressure	Mass flow rate	Volumetric flow rate	Power consumption	Relative humidity	Test package
Refrigeration cycle	8	3	1	1	1	-	-
Cabinet	3	2	-	-	1	1	5
Climatic chamber	1	-	-	-	-	1	-
Others	1	-	-	-	-	-	-
Uncertainty	± 0.5 K	± 0.3 % of full scale	± 0.1 % of measurement	± 0.33 % of measurement	± 0.5 % of measurement	± 2 %	± 0.5 K

Table 3. Number of sensor elements and uncertainties

Table 4. Reference parameters of the evaluation during 24-hour tests

	R-134a			R-513A			R-450A			R-507A		
Parameter	Tw=23.3ºC	Tw=32.8ºC	Tk=43.6ºC	Tw=23.3ºC	Tw=32.8ºC	Tk=43.6ºC	Tw=23.3ºC	Tw=32.8ºC	Tk=43.6ºC	Tw=23.3ºC	Tw=32.8ºC	Tk=43.6ºC
Average water inlet temperature (ºC)	23.2	32.7	43.6	23.3	32.6	43.6	23.3	32.8	43.6	23.1	32.8	43.7
Deviation during test	0.1	0.1	0.1	0.2	0.1	0.2	0.2	0.1	0.2	0.2	0.2	0.2
Average product temperature (ºC)	2.1	2.1	2.1	2.1	2.1	2.1	2.0	2.0	2.1	2.0	2.0	2.1
Deviation during test	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
Average climatic chamber temperature (ºC)	24.8	25.1	24.8	24.9	24.9	24.9	24.9	24.9	24.9	24.8	24.9	24.9
Deviation during test	0.8	0.9	0.8	0.8	0.8	0.8	0.9	0.9	0.9	0.8	0.8	0.8
Average climatic chamber RH (%)	55.1	53.9	54.5	54.3	53.6	53.4	53.3	53.1	53.1	54.1	54.1	54.2
Deviation during test	6.0	6.3	6.0	6.1	6.3	6.2	6.1	6.3	6.3	6.1	6.1	6.1
Environment temperature (ºC)	19.5	19.3	19.0	19.5	17.2	16.9	17.7	15.4	16.3	18.6	18.6	18.6
Deviation during test	0.6	1.3	1.1	1.9	1.5	1.3	2.3	1.5	1.4	1.3	1.3	1.1