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Abstract: This paper reports an experimental comparison between the use of the refrigerants R409A and R437A in a heat pump unit designed and developed to work with R12. Although the use of both refrigerants in new equipments were abolished in EU and US according the new F-Gas Regulation of EU and SNAP, they still being used as options for R12 in old equipments, especially in developing countries. Both refrigerants were studied for the same test conditions, according to two groups of tests: group A (variation of the heat source temperature) and group B (variation of refrigerant flow rate). The results obtained showed that the R437A presents a higher discharge pressure and a lower discharge temperature. The heating and cooling capacities of both refrigerants were similar, as well as the exergetic efficiency. For the group A of tests the COP of both refrigerants was similar and for the group B of tests the R409A presented an average COP 15% higher. According to the results obtained it is recommended the use of R409A in old equipments (as transition refrigerant) until the acquisition of equipments operating with refrigerants with low-GWP becomes technically and economic feasible.

Keywords: R409A; R437A; Exergetic study; Energetic performance; Heat pump

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1 Introduction

In 1834, Jacob Perkins built the first refrigeration machine operating a vapour compression cycle. Since then, the refrigerants used have been under a great development.

Natural refrigerants were the first to be used due to their abundance in nature [1].

Later, CFCs were discovered during the 30's and for about 70 years dominated the residential and commercial refrigeration, standing out for their high stability and easy adaptability to the topology of equipment used at the time [2]. R12 was the refrigerant from the CFCs family which was used the most. However, it was also known as one of the refrigerants with worst environmental characteristics: high ozone depletions potential (ODP=1) and global warming potential (GWP=10,600) [3].

The discovery of the influence of the CFC compounds released into the atmosphere in the process of the ozone layer destruction, lead to the signature of the Montreal Protocol in 1985 [4]. This event marked the beginning of environmental concerns in the refrigeration industry and the setting of the phase-out period for refrigerants with an ODP>0, in this case: CFCs (1995 in developed countries and 2010 in developing countries) and HCFCs (2030 in developed countries and 2040 in developing countries). Therefore, it became necessary to seek alternatives to CFCs and HCFCs [5].

In Europe, the phase-out of CFCs and HCFCs compounds has been faster than in the rest of the world, due to the publication of its own legislation. To promote the rapid decrease of the CFCs bank in Europe, some intermediate blends were used. The R409A was one of the most used blends for drop-in replacement in equipments using R12 [6]. At the time, some authors studied the use of R409A. For instance, Havelsky [7] studied and compared the use of R12 (CFC), R409A (HCFC) and R134a (HFC) to medium and low temperature refrigeration systems, containing hermetic and semi-hermetic compressors. Although R409A belonged to the HCFCs family, it was regarded as the most acceptable substitute for the R12, presenting the best compatibility properties (compared to R134a), together with environmental and energetic bene-

Table 1: Recommended lubricants for different refrigerant mixtures.

Refrigerant	Lubricants
HCFCs	MO and AB
HFCs (*e.g.* R134a)	PAG and POI
HFC/HC mixtures (CFCs and HCFCs substitutes)	MO, AB and POE
HCs	MO, AB and POE
HFO	PAG

fits (compared to R12). Its replacement for the R134a would force the exchange of MO by PAG or POI (synthetic lubricants) [7]. The information concerning the type of lubricant used per family/type of refrigerant is summarized in Table 1.

Climatic changes and the increased greenhouse effect led to the signature of Kyoto Protocol in 1997 [8]. Since that moment, the trend moved towards the use of refrigerants with low-GWP (GWP>150), in particular, some HFCs HFEs, HFOs, HCs and natural refrigerants (known as long-term alternatives) [9]. This goal has reached recently in EU with the new F-Gas Regulation and in US with the new SNAP Regulation [10, 11].

Initially, the R134a was chosen as the refrigerant for long-term replacement of the R12 in new air conditioning and refrigeration systems marketed [12].

Environmental concerns are currently the driving factor in the refrigeration industry evolution. Only in the EU-27, the direct emissions of refrigerants were responsible for the emission of 130 million metric tonnes of carbon dioxide equivalents (MMtCO2-eq) in 2012 [9].

According to the Montreal Protocol guidelines, HCFC compounds are also subjected to the decommissioning process. However, they are still being widely used in developed countries (only exceeded by HFCs) and it is still the most used family of refrigerants in developing countries. The presence of CFC compounds also stands out in developing countries, according to forecasts published [13, 14]. Being as it may, HCFCs compounds continue to be used in most of the vapor compression based refrigeration, air conditioning and heat pump systems in developing countries due to:

(i) Their excellent thermodynamic and thermophysical properties adding to its low cost [15];
(ii) The high cost of technology associated to the use of low-GWP refrigerants (mostly pure substances), because of their undesirable properties.

The low-GWP refrigerants still need further research on their properties before being used in a large scale [16, 17]. There are several recent studies referring this group of refrigerants [18–22]. However, some low-GWP refrigerants are already used in new equipments.

R409A contributes to the destruction of the ozone layer (ODP=0.05) and has a medium contribution to the greenhouse effect (GWP=1909), according the UNEP classification. Due to its contribution in the destruction of the ozone layer, the R409A is subjected to the process of decommissioning according the Montreal Protocol.

R437A is, then, proposed by several companies as the most suitable option to be used as a drop-in replacement for R12 and R409A. One of its advantages is that it does not contribute to the ozone layer destruction (ODP=0) and has a smaller contribution to the greenhouse effect (GWP=1805).

Through the use of HFCs Mixtures (like R437A) it is intended to extend the useful life of an existing installation, without the need to change or adjust any individual component of the existing cooling system, since there are no compatibility problems between the new refrigerant, the used lubricants and the installation materials, thus making then a low-cost replacement.

Actually, there is a lack of information in literature relative to R437A [23]. Nevertheless it is still sold by manufactures as the ideal drop-in replacer (of the HFC family) for R12, R409A and other HCFC blends in existing stationary refrigeration systems equipped with reciprocating compressors.

Among the few studies found in literature containing the R437A, Habka & Ajib [24] analyzed the behavior of some mixtures applied on Organic Rankine Cycle supplied by a geothermal heat source (80–120°C). Cremaschi, *et al.* [25] explored some low-GWP options to replace the R22 in air-conditioning and heat pump systems for residential and small/medium size refrigeration plants, and included the R437A on the study. Among the studied refrigerants, the R437A presented the biggest environmental impact (as was expected), a good material compatibility and the fifth best energetic performance.

This paper covers the concerns above mentioned. The aim of this work is to understand if the R437A is a viable drop-in alternative for R12 and R409A on existing installations, while the acquisition of equipment using low-GWP refrigerants are not technically and economically feasible.

In the present study, some tests were performed to evaluate the thermodynamic behavior and performance of both refrigerants under study (R409A and R437A) for a wide range of conditions in a small capacity heat pump unit. For this purpose, six parameters were evaluated,

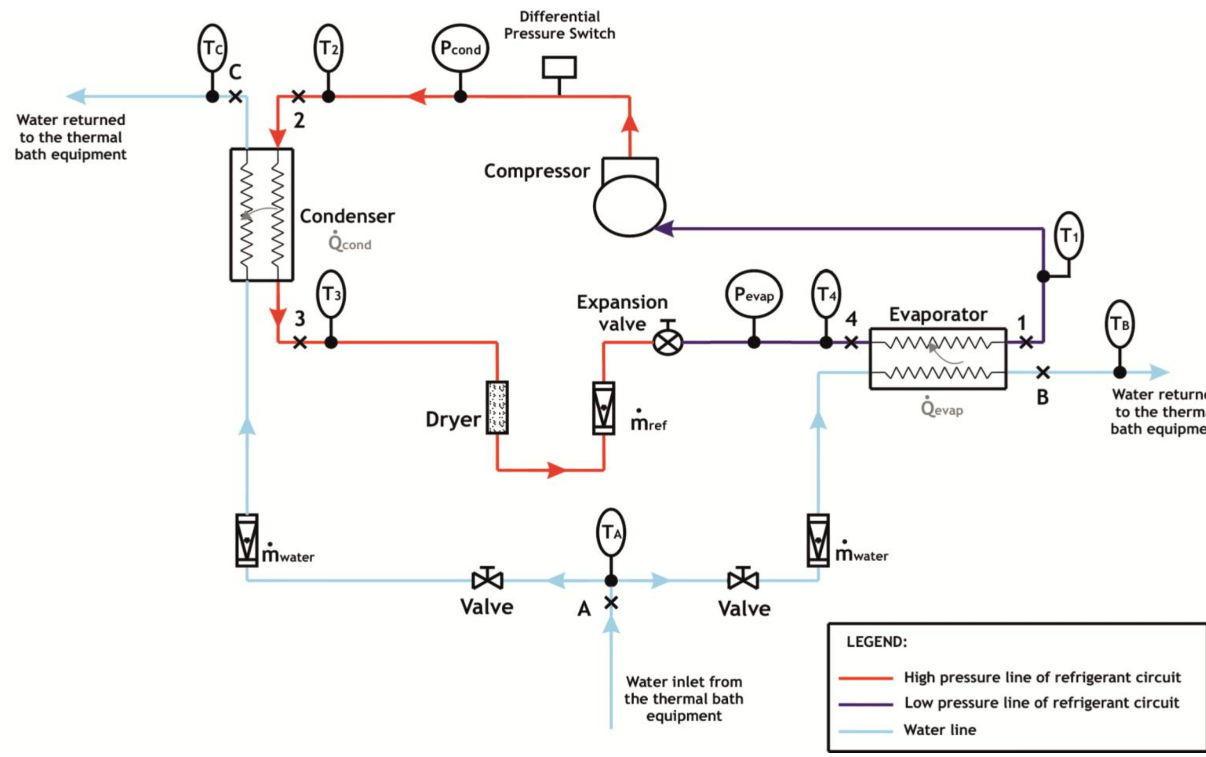

Figure 1: Simplified scheme of the experimental set-up.

namely the discharge pressure and temperature, the cooling and heating capacity, COP and exergetic efficiency, for a wide range of the heat source temperature (T_A) and refrigerant flow rate ($\dot{m}_{ref}$) on the system.

2 Experimental set-up

In this section the characteristics of the experimental set-up are presented, as well as of the refrigerants under study, the set of tests performed in the experimental procedure and the regarded parameters to classify its thermal and energetic performance.

2.1 Characterization of the experimental set-up

The experimental facility built includes two major devices: a heat pump unit and a thermal bath.

A small capacity heat pump unit operating by direct expansion was used to test both of the refrigerants under study: R409A and R437A. This equipment was originally designed to work with R12.

A thermal bath device was used to control the desired heat sources temperature, *i.e.*, control the water inlet temperature that baths the coils of both heat exchangers of the heat pump unit.

Figure 1 shows the simplified scheme of the experimental set-up, including all the instrumentation used.

Some features of the experimental set-up shown in Figure 1 are presented as follows:

(i) As the installation consists in a small refrigeration system, it does not need a charge buffer. It is therefore a critical charge circuit;
(ii) The compressor is of hermetically sealed reciprocating type, being the electric motor windings directly cooled by refrigerant in circulation. This component was not changed during the entire experimental study;
(iii) A manually adjustable expansion valve allows the control of the refrigerant flow rate on the circuit;
(iv) The heat exchangers (condenser and evaporator) are of water-cooled type. An auxiliary water system (light blue line in Figure 1) bathes its serpentines, allowing the rejection and supply of heat to the thermodynamic cycle of refrigeration. Water circulates in a closed loop and reaches the condenser and evaporator at the same temperature (T_A);
(v) Two manual valves are located in the water circuit, allowing the control of the water flow rate through the condenser and evaporator independently;

Table 2: Summary table with the instrumentation equipments used in the experimental set-up.

Parameters	Measuring apparatus	Number of devices	Function in the system
Temperature	Mercury thermometers	7	Measure the temperature of refrigerant and water at the inlet and outlet of condenser and evaporator.
Pressure	Mechanical manometers	2	Measure the pressure in the high-pressure line (*i.e.*, the discharge pressure, measured after the compressor) and the low-pressure line (measured after the expansion valve).
Mass flow rate	Rotameter	3	Measure the flow rates of refrigerant and water across the evaporator and condenser.

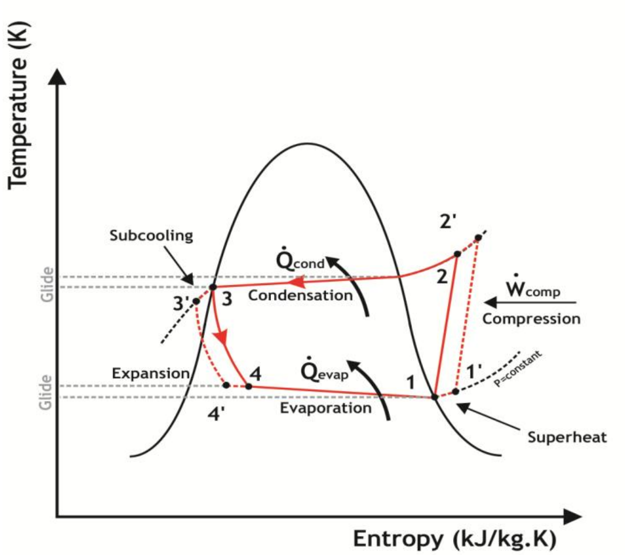

Figure 2: T-s diagram of the vapour-compression refrigeration cycle considered in Fig. 1.

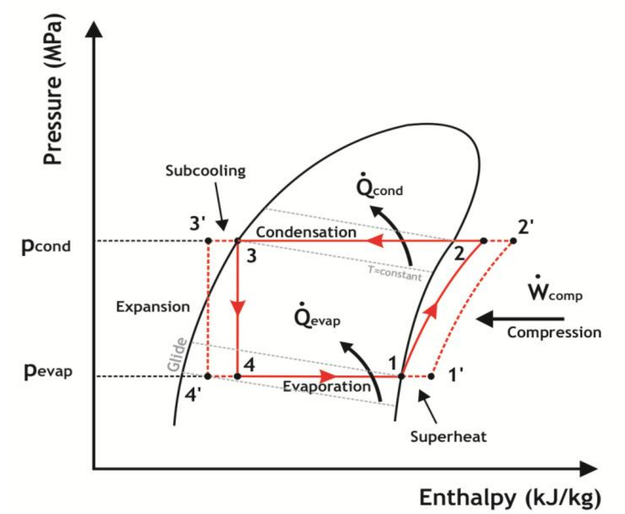

Figure 3: P-h diagram of the vapour-compression refrigeration cycle considered in Fig. 1.

(vi) A silica-gel dryer is used to extract moisture from refrigerant and prevent amendments in its thermal properties;

(vii) A pressure switch was installed on the refrigerant circuit with the purpose of preventing the discharge pressure (P_{cond}) exceeding 1.6 MPa and ensuring the safety of the installation.

The elements used in the instrumentation of the experimental set-up are shown in Table 2.

The T-s and P-h diagrams of the refrigeration cycle corresponding to experimental set-up shown in Figure 1 are illustrated in Figure 2 and Figure 3.

2.2 Characterization of the refrigerants under study

A refrigerant is characterized through the study of its properties. The properties that characterize a refrigerant are related with the following parameters:

(i) Environmental (ODP and GWP);
(ii) In operation (thermophysical properties, thermal energy transfer and efficiency);
(iii) Safety (flammability, toxicity and easily detectable in case of leakage);
(iv) Applicability (compatibility with materials and lubricants) and availability (commercially availability and low cost).

Some characteristics of the refrigerants under study for normal temperature and pressure conditions (NTP) are shown in Table 3, collected from the data sheets belong-

Table 3: Comparison of the characteristics of the refrigerants under study (at NTP conditions).

			Refrigerants		
			HCFC-409A		HFC-437A
Properties	Physical	Mixture composition	60% R22 25% R124 15% R142b		78.5% R134a 19.5% R125 1.4% R600 0.6% R601
		Specific mass liq./vap. sat. [kg m^{-3}]	1216 / 4.97		1176 / 5.38
	Environmental	ODP	0.05		0
		GWP_{100}	1909		1805
	Thermodynamic	$T_{BOILING}$ [°C]	−34.5		−32.33
		$T_{CRITICAL}$ [°C]	106.8		96.41
		$P_{CRITICAL}$ [MPa]	4.69		4.086
		Glide [°C]	7.1		4.28
		h_{VAP} [kJ kg^{-1}]	220.6		213.13
		Thermal conductivity liq./vap. sat. [W m^{-1} C^{-1}]	0.08 / 0.013		0.077 / 0.014
		Volumetric heat capacity liq./vap. sat. (= $c \times \rho$) [kJ m^{-3} C^{-1}]	1569 / 4.13		1689 / 4.63
	Security (ASHRAE)		A1		A1
	Lubricant		MO / AB / POE		MO / AB / POE
	Temperature range		MT / HT	M	MT / HT

ing to the Linde and DuPont companies. Environmental properties (ODP and GWP) were taken from the 2010 annual report of UNEP [26] and the 4th assessment report of IPCC [27].

From the analysis of Table 3, it can be concluded:

(i) R409A contributes to the destruction of the ozone layer (ODP>0), being in the decommissioning process worldwide, according the Montreal Protocol guidelines. The GWP of R437A is a little bit lower;

(ii) Both refrigerants are zeotropic blends, constituted by more than one substance with different volatilities. The temperature difference between the beginning and end phase change of a refrigerant in the evaporator and condenser is named Glide (see Figure 2 and Figure 3). R409A has a Glide 2.82°C higher. Higher values of glide will lead to an increased concentration of the refrigerants with higher evaporation temperatures and consequently, an increased saturation temperature;

(iii) According to their boiling temperatures, both refrigerants are suitable to be used in applications of high (above 0°C) and medium (0 to −25°C) temperature;

(iv) The R409A presents better thermal properties in the biphasic zone (*e.g.*, higher enthalpy of vaporization). On the other hand, the R437A presents better thermal properties in the liquid and saturated vapor zone (*e.g.*, volumetric heat capacity);

(v) Both refrigerants present a security classification of A1 type, according the ASHRAE guidelines. Thus, presenting a lower level of toxicity and flammability;

(vi) As all the HCFCs and HFCs mixtures, they are compatible with MO, AB and POE oils and materials (elastomers, plastics and metals) used on equipments projected to CFCs and HCFCs.

2.3 Tests performed

To understand if the R437A is a viable drop-in alternative for R409A in pre-existing installations, it was necessary to evaluate the thermodynamic behavior of both refrigerants. For this effect, an energetic and exergetic analysis of the thermodynamic cycles was performed, resorting to the data collected on several experimental tests.

The experimental work consisted of two groups of tests (A and B), using both refrigerants:

(i) **Group A** – A study of the heat source temperature influence (*i.e.*, the water inlet temperature (T_A) influence) on the thermodynamic cycle performance to a constant refrigerant flow rate of 12 kg s^{-1}. The

Table 4: Summary table with the tests carried out in the experimental study.

		Groups of tests									
		A					B				
Quantity	T_A [°C]	15	20	25	30	35			25		
	$\dot{m}_{ref}$ [kg h^{-1}]			12			4	8	12	16	20

purpose of this group of tests is to understand the thermodynamic behaviour of both refrigerants to a variation of an environmental parameter (foreign to the cooling circuit), in this case, with the heat source temperature, between 15–35°C;

(ii) **Group B** – A study of refrigerant flow rate ($\dot{m}_{ref}$) influence on the thermodynamic cycle performance to a constant water inlet temperature of 25°C. The aim of the second group of tests is to gain understanding of the thermodynamic behaviour of both refrigerants to a variation of a system optimization parameter, in this case, the refrigerant flow rate, between 4–20 kg s^{-1};

The sets of experimental tests carried out are shown in Table 4.

Some other considerations taken into account during the experimental procedure:

(i) The default water flow rate set across each heat exchanger was $\dot{m}_{water}$ = 40 kg h^{-1};
(ii) The compressor was not changed during the entire experimental study;
(iii) The dead-state conditions regarded on exergetic performance calculation of the heat pump unit were: T_0 = 26°C and p_0 = 0.092 MPa, since they correspond to the thermodynamic conditions of the surrounding environment, *i.e.*, the room conditions where the installation set-up was tested. These quantities were continuously monitored during the experimental tests having been observed that they did not vary significantly in time;
(iv) Results were collected when the steady-state conditions were achieved. The temperature of the water at the evaporator outlet, the discharge pressure and the refrigerant flow rate were continuously monitored in order to confirm the moment in which they stabilized:

$$\frac{\partial T_B}{\partial t} = \frac{\partial p_{evap}}{\partial t} = \frac{\partial \dot{m}_{ref}}{\partial t} = 0 \qquad (1)$$

To achieve the steady-state conditions, it was necessary to wait for approximately 60 minutes on average, from the start of each test;

(v) The equal gas pressure criterion was applied in order to define the mass of the new refrigerant to be introduced in the installation, corresponding to the introduction of 0.52 grams of R437A in the installation. After the refrigerants exchange and before the tests with R437A, the heat pump installation remained at rest during 1 week, in order to understand if the pressure of new refrigerant varied during that period. It was found that for 1 week the pressure of new refrigerant remained stable.

Moreover, there are literature studies that address the effect of refrigerant charges on the system performance [28], but this was not the focus of this investigation.

Table 5 summarizes the characteristics of the instrumentation used in the experimental set-up. Table 6 summarizes the accuracy propagation to all the derived quantities.

2.4 Evaluated parameters

To compare the performance characteristics of both refrigerants under study, some parameters were determined through the use of data obtained experimentally. Parameters such as discharge pressure and temperature, cooling and heating capacity, COP and exergetic efficiency were considered.

The corresponding (T-s) and (P-h) diagrams of vapour-compression cycle considered are presented in Figure 2 and Figure 3.

Applying the 1st law of thermodynamics (*i.e.*, an energetic balance) to the refrigeration system under study, it allows the calculation of its energy efficiency (COP):

$$COP = \frac{a}{b} \qquad (2)$$

where:
a - energetic objective of the cycle
b - energy introduced in the cycle to comply the energetic objective

For the calculations the following assumptions were assumed:

(i) All processes are of steady state and steady flow;

Table 5: Summary table with the characteristics of the instrumentation used in the experimental set-up.

Parameters	Measuring apparatus	Variable measured	Range	Accuracy
Temperature	Mercury thermometers	T_0	20÷50°C	± 0.05°C
		T_1	−10÷70°C	± 0.25°C
		T_2	−20÷100°C	± 0.5°C
		T_3	−10÷50°C	± 0.25°C
		T_4	−35÷50°C	± 0.5°C
		T_A/T_C	0÷50°C	± 0.1°C
		T_B	−2÷6°C	± 0.05°C
Pressure	Mechanical manometers	p_{evap}	−0.1÷0.8 MPa	± 0.01 MPa
		p_{cond}	0÷2.7 MPa	± 0.05 MPa
Mass flow rate	Rotameters	$\dot{m}_{ref}$	3÷25 kg h^{-1}	± 0.5 kg h^{-1}
		$\dot{m}_{water}$	10÷200 kg h^{-1}	± 5 kg h^{-1}

Table 6: Summary table with the uncertainty propagation to all the derived quantities.

Quantity		Uncertainty
Cooling capacity	$\dot{Q}_{evap}$	0.42%
Heating capacity	$\dot{Q}_{cond}$	0.58%
Compression work	$\dot{W}_{comp}$	2.56%
COP$_R$		2.60%
COP$_{HP}$		2.63%
η_R		0.02%
η_{HP}		0.04%

(ii) Kinetic and potential energy effects and, chemical or nuclear reaction of the refrigerant in circulation are negligible.

According to Equation (2), the COP of a refrigeration system and a heat pump unit are as follows:

$$COP_R = \frac{\dot{Q}_{evap}}{\dot{W}_{comp}} = \frac{\dot{m}_{ref}\,(h_1 - h_4)}{\dot{m}_{ref}\,(h_2 - h_1)} \quad (3)$$

$$COP_{HP} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp}} = \frac{\dot{m}_{ref}\,(h_2 - h_3)}{\dot{m}_{ref}\,(h_2 - h_1)} \quad (4)$$

Where $\dot{Q}_{evap}$ is the cooling capacity, $\dot{W}_{comp}$ is the compression work, $\dot{Q}_{cond}$ is the heating capacity, $\dot{m}_{ref}$ is refrigerant flow rate, h is the specific enthalpy and $h_{1,2,3,4}$ meaning the specific enthalpy at the 1, 2, 3 and 4 states.

Applying the 2nd law of thermodynamics (*i.e.*, an exergetic balance) to the refrigeration system under study, allows the calculation of its exergy efficiency (η):

$$\eta = \frac{c}{d} \quad (5)$$

where:
c - exergetic objective of the cycle
d - exergy introduced in the cycle to comply the exergetic objective

Taking into account Equation (5), the η for a refrigeration system and a heat pump unit are given by:

$$\eta_R = \frac{\dot{m}_{ref}\,(\psi_4 - \psi_1)}{\dot{m}_{ref}\,(\psi_2 - \psi_1)} = \frac{\dot{m}_{ref}\,[(h_4 - h_1) - T_0\,(s_4 - s_1)]}{\dot{m}_{ref}\,[(h_2 - h_1) - T_0\,(s_2 - s_1)]} \quad (6)$$

$$\eta_{HP} = \frac{\dot{m}_{ref}\,(\psi_2 - \psi_3)}{\dot{m}_{ref}\,(\psi_2 - \psi_1)} = \frac{\dot{m}_{ref}\,[(h_2 - h_3) - T_0\,(s_2 - s_3)]}{\dot{m}_{ref}\,[(h_2 - h_1) - T_0\,(s_2 - s_1)]} \quad (7)$$

Where ψ is the Darrieus function, T_0 is the specific entropy and is the temperature of dead-state.

3 Results and discussion

Table 7 summarizes the results obtained during the experimental tests and the results of some parameters calculated from them. The parameters that allow the comparison of the performance of refrigerants under study are presented and discussed below.

3.1 Discharge pressure

Discharge pressure and temperature measured at the compressor outlet are the maximum pressure and maximum temperature reached in a refrigeration system. These are important parameters for the installation safety and reliability, affecting directly its performance (since it influences the stability between lubricants and compressor components and the solubility of lubricants in refrigerants).

Table 7: Summary table with the results obtained from the experimental tests.

Refrigerant	Group of tests	$\dot{m}_{ref}$ [kg h^{-1}]	T_A [°C]	T_B [°C]	T_C [°C]	T_1 [°C]	T_2 [°C]	T_3 [°C]	T_4 [°C]	P_{evap} [MPa]	P_{cond} [MPa]	Compressor ratio	W_{comp} [kW]	$\dot{Q}_{evap}$ [kW]	$\dot{Q}_{conf}$ [kW]	COP_R	COP_{HP}	η_g	η_{HP}
R409A	A	12	15	5.5	33.5	5	80.5	25	−17.5	0.19	1.04	5.42	0.163	0.557	0.720	3.41	4.41	0.685	0.278
			20	7.5	35.4	7	81	29	−17	0.19	1.18	6.15	0.160	0.547	0.707	3.42	4.42	0.570	0.284
			25	14	40.4	14.5	86	33.5	−16	0.2	1.29	6.39	0.153	0.543	0.697	3.54	4.54	0.488	0.327
			30	19	44	19	88	38	−15	0.21	1.41	6.65	0.157	0.527	0.683	3.36	4.46	0.454	0.354
			35	23.5	48.2	23.5	92	42.5	−13.5	0.22	1.58	7.28	0.140	0.510	0.650	3.64	4.64	0.356	0.352
	B	4	25	20	31.5	20.5	65.5	27.7	−29.5	0.11	1.08	9.64	0.024	0.196	0.220	8.00	9.00	0.583	0.193
		8		16.5	36	17	78	30	−22	0.15	1.18	7.76	0.076	0.380	0.456	5.03	6.03	0.518	0.283
		12		14	40.4	14.5	86	33.5	−16	0.20	1.29	6.39	0.153	0.543	0.697	3.54	4.54	0.464	0.317
		16		11	45	11.5	92	38	−9	0.26	1.44	5.50	0.222	0.689	0.911	3.10	4.10	0.439	0.393
		20		9	48.3	8	93	41.5	−5	0.30	1.54	5.10	0.311	0.817	1.128	2.63	3.63	0.324	0.477
R437A	A	12	15	5.5	32.5	5	73	20	−16.5	0.11	1.05	9.39	0.167	0.563	0.730	3.38	4.38	0.643	0.245
			20	8	35.5	8	76	24.5	−16	0.19	1.24	6.46	0.160	0.550	0.710	3.44	4.44	0.587	0.278
			25	14	40.6	14	84	29	−14	0.20	1.47	7.28	0.157	0.540	0.697	3.45	4.45	0.534	0.301
			30	18.5	45	18	86	33.5	−13	0.21	1.55	7.30	0.157	0.533	0.690	3.40	4.40	0.465	0.374
			35	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	B	4	25	20	31.8	20	65.5	27	−28	0.11	1.09	9.73	0.029	0.191	0.220	6.62	7.62	0.640	0.156
		8		17	35.6	17	76	28	−20.5	0.15	1.24	8.16	0.089	0.371	0.460	4.18	5.18	0.563	0.256
		12		14	40.6	14	84	29	−14	0.20	1.47	7.28	0.157	0.540	0.697	3.45	4.45	0.506	0.337
		16		9	45.3	9	89	31	−8	0.24	1.67	6.90	0.253	0.684	0.938	2.70	3.70	0.438	0.419
		20		-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-

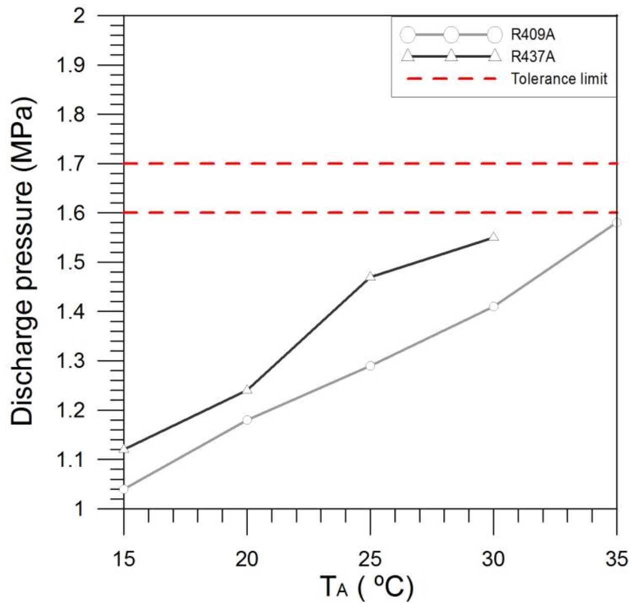

Figure 4: Effect of the heat source temperature on the compressor discharge pressure (Group A of tests).

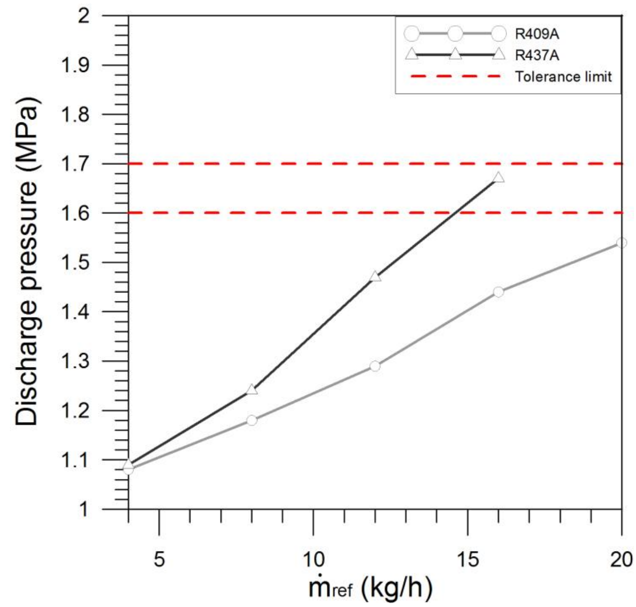

Figure 5: Effect of refrigerant flow rate on the compressor discharge pressure (Group B of tests).

The refrigerant discharge pressure increases significantly with the increase of the heat source temperature (see Figure 4) and refrigerant flow rate (see Figure 5), having the highest growth been verified with R437A for the variation of refrigerant flow rate (an increase of 0.0363 MPa by 1 kg h^{-1} of refrigerant). For lower refrigerant flow rates the difference in discharge pressure values of both refrigerants is much smaller. It can be concluded that the refrigerant flow rate variation has a greater influence on the discharge pressure.

R437A presents an average discharge pressure 7.3% and 8.8% higher than R409A for Group A and B of tests respectively. This discharge pressure difference between refrigerants is due to the higher density of saturated vapour presented by R437A (in about 7.6%), the lower pressure in the evaporator with the R437A (10% and 3% on average for Group A and B of tests) and to the higher compression ratios reached on the R437A cycle (19.2% and 8.8% on average for Group A and B of tests) in comparison to the R409A.

An increase of refrigerant charge in the installation will lead to a substantial rise of discharge pressure. This tendency was also observed by previous researchers [25]. It was not possible to perform tests with higher refrigerant charges than the one used in this work (given the conditions imposed and installation characteristics), therefore it was not possible to perform tests for $T_A = 35°C$ (Group A of tests) and $\dot{m}_{ref} = 20$ kg h^{-1} (Group B of tests). For these conditions the discharge pressure exceeded the maximum pressure value allowable in the installation.

3.2 Discharge temperature

Discharge temperature is an important indicator used to study the superheating effect and heat generated by the refrigerant compression process. Both the increase of the heat source temperature (see Figure 6) and the refrigerant flow rate (see Figure 7) cause the discharge temperature to increase. The R409A presents an average discharge temperature 4.9% and 2.2% higher than R437A for Group A and B of tests, as well as a higher average superheating temperature in 4% and 14% for Group A and B of tests, respectively. The highest glide value presented by the R409A can contribute to an increased saturation temperature of the refrigerant and, thereby, to an increased discharge temperature.

The R409A also presents lower phase change temperatures on the evaporator and condenser.

As for the discharge pressure, the variation of refrigerant flow rate is also the factor that most significantly affects the discharge temperature. For lower refrigerant flow rates, the discharge temperature of both refrigerants comes close, as well as the compression ratio of approximately 10 for $\dot{m}_{ref} = 4$ kg h^{-1}.

An increase on the refrigerant charge into the installation would lead to an increase of the discharge temperature. For the studied conditions, the discharge temperature comes very close to the critical temperature of the refrigerants under study. This may introduce changes on the refrigerant and lubricant properties, resulting in decreasing solubility of lubricant oil/refrigerant and degradation

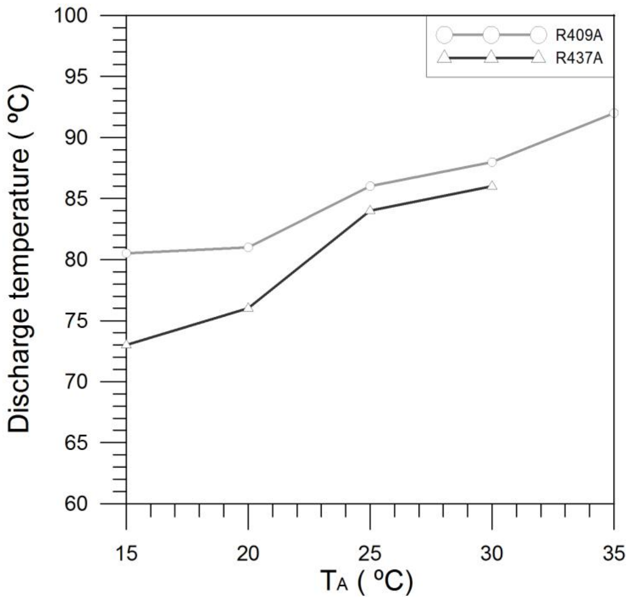

Figure 6: Effect of the heat source temperature on the compressor discharge temperature (Group A of tests).

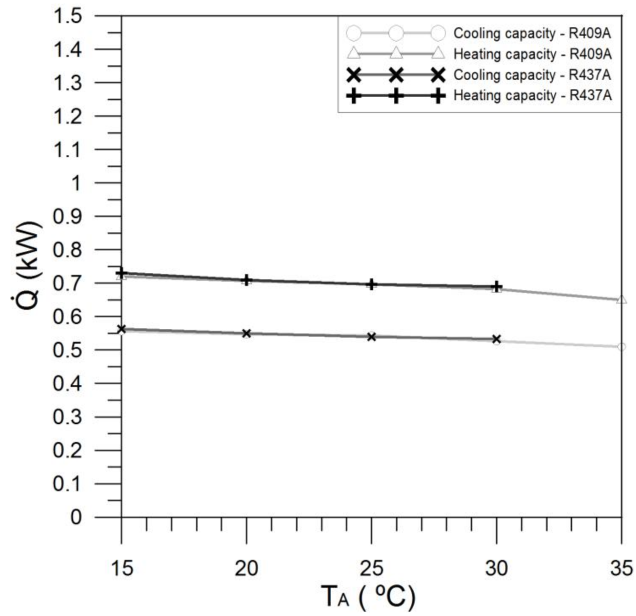

Figure 8: Effect of the heat source temperature on the cooling and heating capacities (Group A of tests).

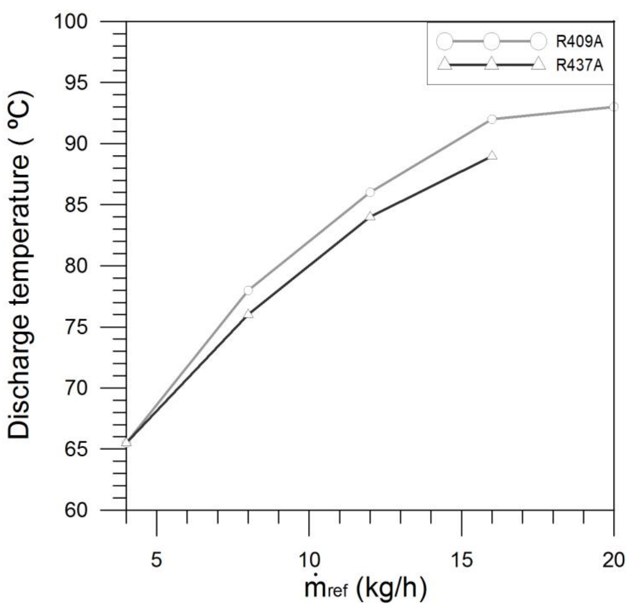

Figure 7: Effect of refrigerant flow rate on the compressor discharge temperature (Group B of tests).

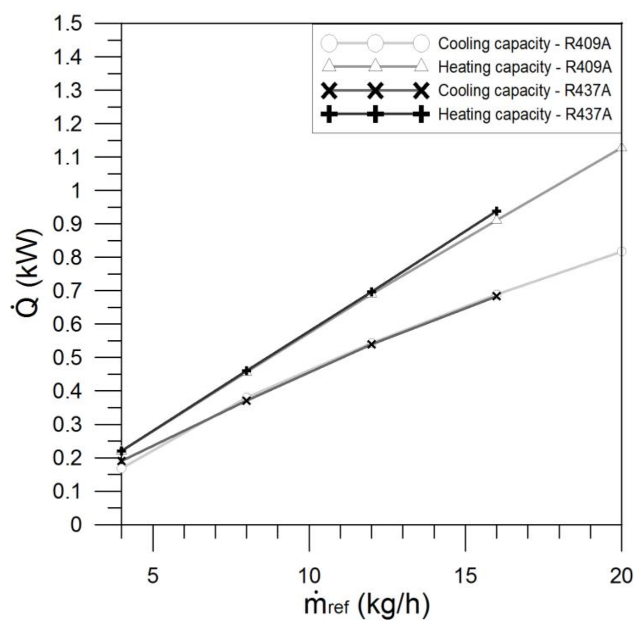

Figure 9: Effect of refrigerant flow rate on the cooling and heating capacities (Group B of tests).

of the insulation materials and refrigerant thermal properties.

3.3 Cooling and heating capacities

Cooling capacity expresses the rate of heat removed from the cold source, producing a decrease of its temperature. On the other hand, the heating capacity expresses the rate of heat supplied to the heat source, producing an increase of its temperature.

For both refrigerants under study, the heating capacities are almost identical, as well as the cooling capacities (see Table 7, Figure 8 and Figure 9). In general, the heating capacity is higher than the cooling capacity, but coming closer to the lower refrigerant flow rate values. With the heat source temperature growth, the heating and cooling capacities suffer a small decrease. This is due to the decreased enthalpy gained by the refrigerant on the evaporator and ceded on the condenser. Increasing the temperature of both thermal sources T_A causes the vertical displacement of the cooling cycle on p-h diagram, which leads to a reduction of the biphasic zone and subsequent stretching to the superheated vapour zone (having the superheating temperature grown from 12 to 33°C for R409A and 18 to 29

This causes a decrease of the refrigerant heat transfer capacity, as much on the evaporator as on the condenser. It can be concluded that the refrigerant flow rate is the factor that most contributes to the variation of cooling and heating capacities, since the increase of the amount of refrigerant in circulation significantly increases the amount of heat carried. The appearance of sub-cooling in the refrigeration cycle was also observed on the R437A, something that had not happened with the R409A.

Also for both refrigerants, the outlet water temperature on the evaporator and condenser were similar, as a consequence of the identical cooling and heating capacities.

3.4 COP

The Coefficient of Performance (COP) is used to quantify the energetic performance of equipments that operates through a vapour compression refrigeration cycle (see Equation (2)). For a refrigeration installation the COP is expressed by Equation (3) and Equation (4) to a heat pump installation (only differing on energetic objective of the cycle). Increasing the heat source temperature does not cause a significant impact on COP as shown in Figure 10. In this case, the resulting COP for both refrigerants is similar, being the average values of COP_R = 3.4 and COP_{HP} = 4.4. This happens because the heat source temperature increase does not cause a significant variation on cooling and heating capacities or on the compression work (having this decreased from 0.163 to 0.157 kW for R409A and 0.167 to 0.157 kW for R437A).

On the other hand, the increase of refrigerant flow rate causes a significant COP decrease, being the higher COP difference between refrigerants presented for lower values of refrigerant flow rate (see Figure 11). For $\dot{m}_{ref}$ = 4 kg h^{-1}, the COP for R409A is 17.5% higher than the presented by the R437A. This difference is attenuated with the refrigerant flow rate increase. Considering the flow rate of 16 kg h^{-1}, the COP for R409A is 12.9% higher than the presented by the R437A. As the cooling and heating capacities of both refrigerants are similar, it can be concluded that

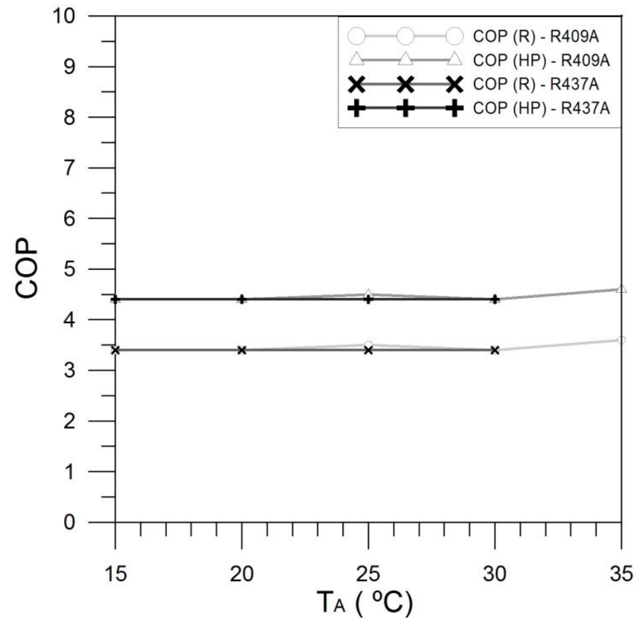

Figure 10: Effect of the heat source temperature on COP (Group A of tests).

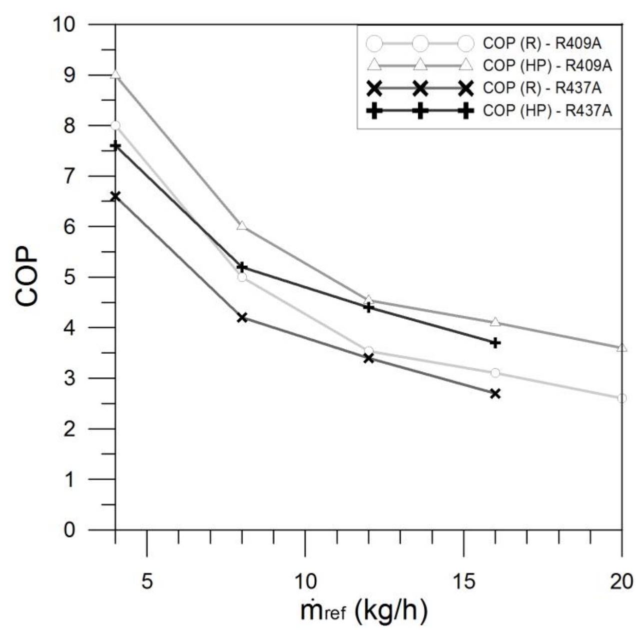

Figure 11: Effect of refrigerant flow rate on COP (Group B of tests).

the compression work required by R437A is higher. The average compression work required by R437A is 11.2% higher than by R409A.

3.5 Exergetic efficiency

The exergetic efficiency (η) provides the quantification and a qualitative analysis of the several forms of energy ap-

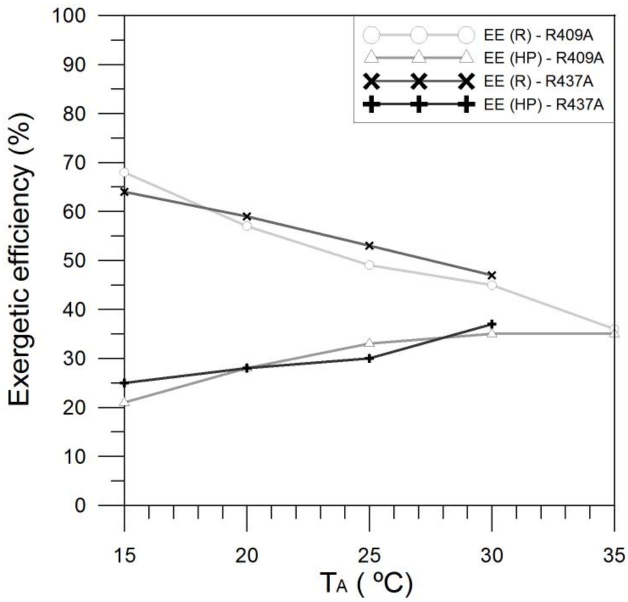

Figure 12: Effect of the heat source temperature on η (Group A of tests).

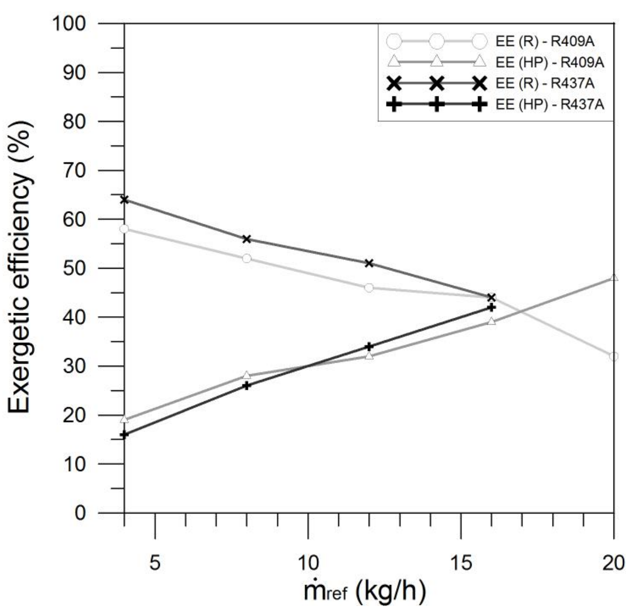

Figure 13: Effect of refrigerant flow rate on η (Group B of tests).

plied to the system. Through the use of this parameter it is possible to determine the irreversibility of a thermodynamic process (Equation (5)). For a refrigeration installation the η is expressed by Equation (6) and Equation (7) to a heat pump installation (only differing on exergetic objective of the cycle). The effects of the heat source temperature and refrigerant flow rate variation on η are shown on Figure 12 and Figure 13, respectively.

Both the increase of the heat sources temperature and refrigerant flow cause the increase of η of the installation operating as heat pump and the decrease of η of the installation operating as refrigerator. Again, for both refrigerants, the curves of η are similar, being the difference between the average value of curves for each refrigerant less than 2%. However, the average η presented by R437A to the flow rate variation in the installation operating as refrigerator is 6.7% higher than the one presented by R409A. According the dead state condition considered ($T_0 = 26°C$), the amount of exergy destroyed it is similar for both refrigerants under study.

4 Conclusion

The Montreal Protocol signature promoted the gradual abolition on substances that deplete the ozone layer, namely, the compounds belonging to the CFCs and HCFCs families. However, these compounds are still used in some parts of the world, especially in developing countries. Most recently, the R437A was found as the best alternative refrigerant blend, due to its suitable properties, to perform a drop-in replacement on equipments using R12 and R409A, in its different applications. This is a non-ozone depleting substance, which is expected to be used as a transition refrigerant due its medium global warming potential. In this study, the performance of R437A was investigated experimentally in a heat pump unit and compared with the performance of the system when the R409A was used as refrigerant and tested for the heat sources temperature and the refrigerant flow rate variation. From the results obtained in the experimental procedure it is possible to perform the following remarks:

(i) The refrigerant flow rate variation was the factor that most influenced the analyzed parameters (discharge pressure and temperature, $\dot{Q}$, COP and η);
(ii) The R437A presented a higher discharge pressure and compression ratio. In this case, it may be necessary to reduce the refrigerant charge in the system, in order to reduce the discharge pressure. Perhaps an optimization of the charge of the new refrigerant on the system could help understanding how to solve this problem;
(iii) The higher discharge temperature was obtained using R409A;
(iv) Both refrigerants have shown similar values of cooling and heating capacities for the studied conditions;
(v) For the heat sources temperature increase the COP of both refrigerants under study was similar. For the refrigerant flow variation, the R409A presented an

average COP 15% higher than the one presented by the R437A. The difference between the COP of the refrigerants decreases with the increase of the refrigerant flow rate;

(vi) Although the R437A presents a lower direct environmental impact (ODP and GWP), the value of indirect GHG emissions (resulting from the energy consumption of the equipment) is higher, according to the lower COP experimentally obtained. This is very important, since the indirect emissions represent the greatest environmental impact in the life cycle of a refrigerant [9];

(vii) The exergetic efficiencies presented by the installation were similar for both refrigerants under study.

Abbreviations List

AB	Alkylbenzene oil
CFCs	Chlorofluorocarbons
EU	European Union
GHG	Greenhouse Gas
GWP	Global Warming Potential
HCFCs	Hydrochlorofluorocarbons
HCs	Hydrocarbons
HFCs	Hydrofluorocarbons
IPCC	Intergovernmental Panel on Climate Change
MO	Mineral oil
ODP	Ozone Depletion Potential
PAG	Polyol ester oil
POI	Polyol alkyl oil

Nomenclature

c Specific heat (kJ kg^{-1} C^{-1})
$\dot{Q}$ Heat transfer rate (kW)
$\dot{W}$ Work rate (kW)
s Specific entropy (kJ kg^{-1} C^{-1})
$\dot{m}$ Mass flow rate (kg h^{-1})
h Specific enthalpy (kJ kg^{-1})
T Temperature (°C)
p Pressure (MPa)

Greek symbols

η Exergetic efficiency
ψ Darrieus function (kJ kg^{-1})

Subscripts

0 Dead-state
cond Condensation
evap Evaporation
ref Refrigerant
R Refrigeration
HP Heat pump

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