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EXPERIMENTAL EVALUATION OF REFRIGERANTS R290, R32 AND R410A IN A REFRIGERATION SYSTEM ORIGINALLY DESIGNED FOR R22

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ABSTRACT

The present article focuses on the climate performance of three different refrigerants used to replace the hydrochlorofluorocarbon (HCFC), R22, for common refrigeration applications. The original system consists of a small commercial refrigeration system, which provides nominal refrigerating capacity of 15 kW. The condition of replacement of R22 by these alternative fluids (R290, R32 and R410A) represented a drop-in operation; there were no changes in the basic cycle components during the tests, with the exception of the lubricating oil. Experimental tests were carried out in steady state condition and throughout the tests the compressor had its entire speed range explored for different levels of modulation of the electronic expansion valve (EEV), thus enabling the realization of a complete thermodynamic analysis. The use of the hydrofluorocarbon (HFC), R32, which has a low global warming potential (GWP) and zero ozone-depleting potential (ODP) guaranteed good efficiency to the refrigeration system. The results showed regular conditions of efficiency of the experimental facility operating with the blend R410A. Finally, the use of the hydrocarbon (HC) resulted in the maximum values for the coefficient of performance, COP.

INTRODUCTION

Climate-friendly refrigerants with excellent thermal properties are important to reduce the degradation of the environment and guarantee good performance to refrigeration systems. The ODP of a refrigerant is the measure of its relative ability to destroy stratospheric ozone. It depends on the percentage of chlorine or bromine atoms in the molecule and the lifetime of the compound in the atmosphere. The direct emission of refrigerants and indirect emission of CO_2 from the applications in Heating, Ventilation, Air Conditioning and Refrigeration (HVACR) are related primarily but not exclusively, to two global environmental issues: the depletion of the ozone layer and the global warming.

NOMENCLATURE

EEV		Electronic expansion valve
F-gas		Fluorine gas
GWP		Global warming potential
MAC		Mobile air conditioning
ODP		Ozone depletion potential
POE		Polyolester oil
PVE		Polyvinyl ether oil
VSC		Variable-speed compressor
Special characters		
Δh	[kJ/kg]	Specific enthalpy difference
ΔT	[ºC]	Temperature difference
F	[Hz]	Frequency
h	[kJ/kg]	Specific enthalpy
$\dot{m}$	[kg/s]	Mass flow rate
p	[Pa]	Pressure
$\dot{Q}$	[kW]	Refrigerating capacity
T	[°C]	Temperature
$\dot{W}$	[kW]	Energy consumption
Subscripts		
CD		Condensation
DC		Discharge
EV		Evaporation
EVAP		Evaporator
LV		Liquid-Vapor (latent heat of evaporation)
REF		Refrigerant
SC		Subcooling
SH		Superheat

Since then, the Montreal (1987) and Kyoto (1997) protocols through various measures seek to eliminate and reduce the emission of these gases. Therefore, there is the immediately necessity to replace traditional refrigerants characterized by a high GWP for alternative refrigerants less aggressive to the environment.

The HCFCs' cost tends to be cheaper, although the accelerated phase-out will increase the prices and the availability will lessen. Alternatives to HCFCs have been investigated for decades by chemical companies and equipment manufacturers through industrial collaboration and the most-used replacements that have emerged to date are HFC blends;

however, many candidates still do not attend the market requirements, due to the climate effects of HFC refrigerants having high GWPs.

Current research and industry trends show that HCFCs and HFCs will be gradually replaced by HFC blends or by natural refrigerants [1].

The fluorine gas (F-gas) regulation and the mobile air conditioning (MAC) directive of the European Union supported the development of low GWP refrigerants [2]. The new non-ozone-depleting low-GWP refrigerants have a maximum value of 150 for the GWP/100-year time horizon. Some of these will have the potential for broader applications, however they present relative lower efficiency inside the existing systems and they have high costs thereof.

In general, natural fluids such as water, HCs, ammonia and carbon dioxide are refrigerants that have zero ODP and also have a very low GWP. Furthermore, these natural substances are found abundantly in nature, cooperating to ensure their competitiveness in the global market. HCs tend to be less widely available and ammonia is sourced from specialist suppliers.

During the last years, studies comparing the performance of synthetics and natural refrigerants in various applications were published. In spite of considerable efforts to improve the thermal properties of the alternative fluids and to develop new designs or control strategies for the systems, in some cases, inconsistent experimental results can be found [3].

This paper aims to contribute to knowledge about the feasibility of using alternative refrigerants instead of traditional fluids of refrigeration systems in operation. The analyses referred to drop-in tests, where different refrigerants were tested in the same experimental facility under a very specific refrigeration application.

The idea was to use the refrigerant R22 as working fluid, and then replace it by the refrigerants: R290, R32 and R410A. The experimental facility provides instrumentation and control strategies which allowed evaluating the behavior of the mass flow rate of refrigerant, the refrigeration capacity, the power consumption of the compressor and the COP, of each of the four systems.

MATERIAL AND METHODS

Domanski and Yashar [3] pointed important issues to experimentally compare refrigerants. They considered that the class of experimental facility used to test refrigerants on the present research (breadboard apparatus) equipped with a variable-speed compressor (VSC) can provide biased results. It is important to assure that all system components are optimized for each individual refrigerant because the refrigerant's performance in a system is strongly affected by hardware design.

The main objective of this research was not to compare the potentials of different refrigerants, we were not interested in checking all the possibilities of use or creating physical correlations for a pure substance, and there is no sufficient equipment for these specific analyses. However, five years of massive work on the basic equipment, preview knowledge about operation limits and regular updates of the present experimental facility were factors that allowed us to apply the drop-in methodology.

First stage of tests

The first experimental phase was represented by 52 tests, divided into four experimental designs and performed in steady state. The analysis of these tests was performed using the response surface methodology (RSM). According Calado and Montgomery [4] response surfaces are used when the response variables are influenced by many independent variables and the objective is to improve these responses.

The objective was to explore the entire frequency range of operation of the compressor and the modulating effect of VEE. Thus, it has become possible to evaluate the behavior of the refrigeration capacity of the four systems.

The first experimental design (13 tests) was created to analyze the conditions of operation of the original experimental facility containing R22. The other plans were created in order to prove that the system with R290, or R32 and R410A can also operate in a similar way to the system with R22.

Second stage of tests

In addition, was carried out a second stage of experimental tests, 12 tests at steady state. The analysis of these tests was performed by a comparative chart, where COP for each refrigerant was confronted.

The four fluids operated under two conditions of evaporation, -15°C, -10°C and -5°C, and the variation of the saturation temperature in the evaporator was only possible due to precise control of pressure at the evaporator inlet through modulation (opening or closing) of electronic expansion valve. The condensation condition was similar in all tests.

Analyzed Refrigerants

The thermodynamic characteristics of R22 make it suitable for a wide range of applications in commercial and industrial refrigeration systems. It is the most common HCFC used as a refrigerant. A fluid composed of hydrogen, chlorine, fluorine and carbon. It destroys the ozone layer, despite being less stable than chlorofluorocarbons (CFC), and it is also a greenhouse effect gas. As per the Montreal Protocol, R22 has to be phased out by 2030 in non-Article 5 countries (developed countries) and 2040 in Article 5 countries (developing countries). In 2007, the Parties to the Montreal Protocol decided to accelerate the HCFC phase-out schedule [2].

The three adopted fluids are refrigerants that have been developed, or can be used in existing systems that contained the R22. Hydrocarbons have complete chemical compatibility with almost all lubricants commonly used in refrigeration. However, silicone lubricant additives and silicates are not compatible with hydrocarbons [5]. The R438A contains an HC component so that it is soluble with existing mineral oil. The R410A and R404A are immiscible with the traditional mineral oils, and require the use synthetic oils for miscibility and oil return. The working fluids used to transfer the heat from low temperature reservoir to high temperature reservoir were:

- R32: An HFC refrigerant, its ODP is zero, and its GWP is much lower than other HFCs. It is considered a refrigerant with lower flammability, safety designation A2 by ASHRAE Standard 34 [6]. Currently, R32 is commonly used as a component of R407C and R410A [7]. HFCs are considered long-term alternatives with regard to ozone depletion, but as greenhouse gases, HFCs are covered by the Kyoto Protocol, and a number of countries are implementing regulations to control their use [2].

- R290: Propane is an HC that can be used both for refrigerating and for deep-freezing applications. It is also proposed and actually used in small heat pump and refrigeration systems [8-9]. Colbourne and Suen [10] showed the advantages in using HCs in relation to the use of fluorinated refrigerants. The use of HCs represented performance improvements in the order of 6.0% for domestic refrigeration applications, 15.0% for applications of commercial refrigeration, 8.8% for air conditioning and 9.6% for heat pumps. Park and Jung [11] analyzed the thermal performance of two hydrocarbon refrigerants (R290 and R1270) in an attempt to replace the R22. They used an experimental facility depicting an air conditioning/heat pump system with capacity of 3.5 kW. The test results showed that the coefficients of efficiency of hydrocarbon refrigerants were 11.5% higher than those of R22 in all conditions. This fluid has good thermodynamic properties, but it is flammable. This problem requires additional efforts in design, manufacturing and service of the equipment. Corberán et al. [12] summarized in their work the main safety standards adopted for use of hydrocarbon refrigerants. ASHRAE Standard 34 [6] classified these refrigerants as class 3 (high flammability fluid) whereas ISO 817 and EN 378 classify as A3 class fluid (low toxicity and high flammability) [13]. At the present research it was used the EN 378, the maximum charge calculated was 1.5 kg for this hydrocarbon. Other observed points were: refrigerant authorized places, construction requirements for the mechanical system and external resources associated with the installation, such as ventilation.

- R410A: It is an HFC refrigerant blend designed to be long-term substitute for R22. As well as R22, this mixture is classified by the ASHRAE Standard 34 [6] as A1 (no-flame propagation). It is a zeotropic mixture: in a state change (condensation or evaporation), the temperature varies. The temperature "glide" varies for different blends, and this factor makes some of these refrigerant blends unsuitable as replacements for R22. The standard refrigeration oils for use with R410A as polyolester (POE) and polyvinyl ether (PVE) have insufficient miscibility with R32 [14]. When the relationship between the miscibility of the oil/refrigerant is low, oil tends to remain in the evaporator and does not return to the compressor. This may cause a decrease in system performance, and the problem of oil return can lead to poor lubrication of the compressor. Okido et al. [15] have succeeded in developing new POE oils that eliminate this problem. Yunho et al. [16] compared the performance of R290 with R404A and R410A for a refrigeration system. The COPs of R404A and R410A were 11-12% and 4-9% lower, respectively, than that of R290 under equal operation system capacity. Da Silva et al. [17] evaluated the energy efficiency and climate performance of three different systems used in supermarket applications. A cascade cycle (CO_2/R404A) and also R404A and R22 with direct expansion systems. The impact in the atmosphere of the cascade system operating with CO_2 was much less than the two direct expansion systems.

Differences in thermodynamic properties of these refrigerants can be visualized on a pressure-enthalpy diagram, as shown in Figure 1. Based on the standard reference state for the International Institute of Refrigeration, the value of specific enthalpy is set to 200 kJ/kg and the value of specific entropy is set to 1.0 kJ/kgK for saturated liquid at 0°C (273.15 K), EES 9.482 [18].

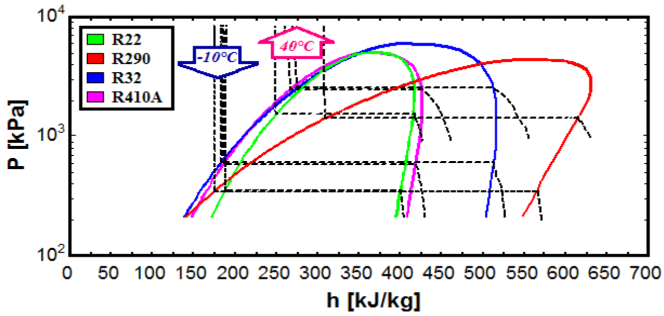

Figure 1 Pressure-enthalpy diagram comparing the refrigerants.

The shown two-phase domes are extremely different, the first point to observe are the distances between each critical point, critical temperature influences refrigerant pressure and vapor density. The saturation temperature, 40°C, was used to exemplify the range of discharge pressure that the compressor will work with. It is important to take a look at the high discharge pressures values of R410A and R32, depending on the tests conditions, these parameters will increase the power consumption of the alternative compressor installed on the present experimental facility. Transport properties like thermal conductivity and viscosity are important parameters for heat exchanger design. The four isothermal curves (-10°C) point to the large difference between the evaporation conditions. The latent heat of evaporation (Δh_{LV}) of the R290 is higher than the others. A test condition where the refrigeration capacity is kept constant for all refrigerants will show that the Δh_{LV} difference had directly influence under the reduction on values of mass flow rate of the R290 compared with the R22.

Experimental Facility

The experimental facility consists of an alternative compressor, two concentric tubes heat exchangers (refrigerant/water), an EEV and all appropriate instrumentation.

The analog signals of temperature, pressure and flow were converted to digital through the Programmable Logic Controller (PLC). Data were monitored and managed through an interface created with the software LABVIEW. The secondary fluid, which is the water that circulates through the condenser, flows through a cooling tower. Moreover, the heat transfer in the evaporator is accomplished using a thermal storage tank that simulates a thermal load through an electrical resistance, with the function of maintaining stable the desired water temperature in the inlet of the evaporator. Figure 2 depicts, schematically, the experimental facility of this study.

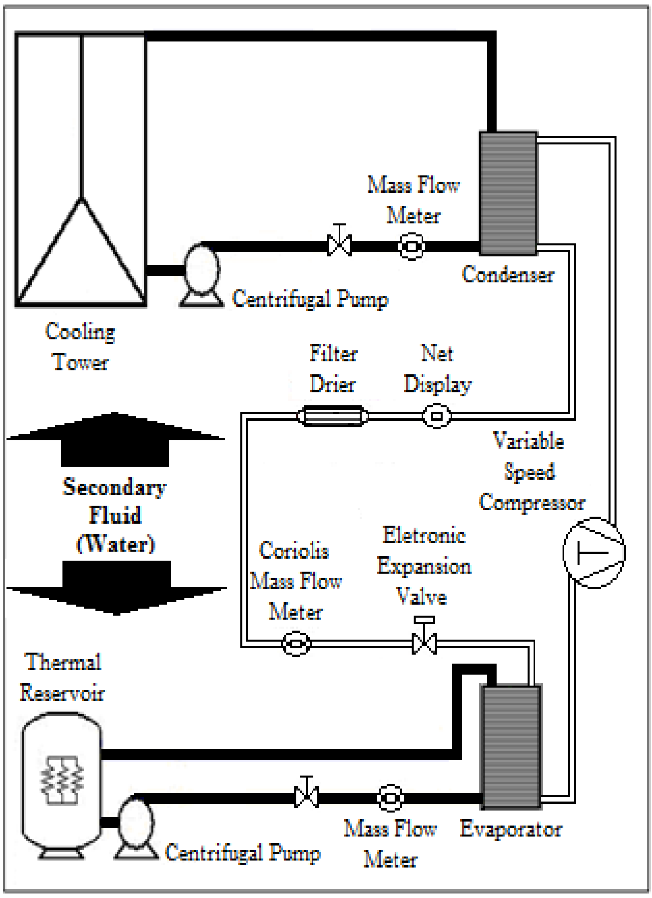

Figure 2 Schematic view of the experimental facility.

The refrigeration capacity was calculated by the First Law of Thermodynamics, equation (1), for steady state condition, and based on the consideration that the refrigerant is the unique substance inside the control volume.

$$\dot{Q}_{REF} = \dot{m}(\Delta h_{EVAP}) \quad (1)$$

Once Δh_{EVAP} is the difference of enthalpy between the outlet and the inlet of the evaporator. Piezoresistive pressure transducers (with measured uncertainty of 25.0 kPa) and resistance temperature detectors PT-100 (with measured uncertainty of 0.15 °C) were used to measure these properties, thus enabling the determination of the thermodynamic state of the refrigerant at each point of interest from the vapor compression cycle. A Coriolis flow meter was used to measure the mass flow rate of refrigerant ($\dot{m}$) in the main circuit, and the measured mass flow rate uncertainty was 0.0015 kg/s. The power consumption of the compressor was measure with uncertainty of 0.003 kW. The propagation of uncertainty for the refrigeration capacity ($\dot{Q}_{REF}$) and the COP were estimated and these can be viewed on the results phase.

The experimental facility operated originally with the R22 for normal refrigeration applications. It was considered the effect of the increment opening of electronic expansion device, compressor speed, cooling tower operation and refrigerant charge together.

The charge of R22 used for all tests was 3.2 kg; adopted due to earlier analyses, this mass was idealized, as implied in the optimum operation conditions of the system with R22. The first drop-in operation performed with the hydrocarbon R290, it was used only 47% of the mass adopted for R22, i.e. 1.5 kilograms. It was used in all tests with R22 and R290 mineral oil. On the second drop-in of the system, R32 was used in a smaller amount compared to R22, only 1.9 kilograms. It has become necessary to change the mineral oil by the polyolester oil for testing with the R32 and R410A. The final system operation, R410A, was performed with the same charge value of R22. The charges adopted for all fluids supplied the evaporator and assured optimized conditions. Higher charge values resulted in higher working pressures and excessive consumption.

The evaporator installed in the experimental facility may operate at a maximum capacity equivalent to 15 kW and has a tube-in-tube geometry, which allows the use of a secondary incompressible fluid (water) to make the evaporation of the refrigerant. This side of the evaporator circuit is composed of a thermal storage reservoir which simulates the thermal load through an electrical resistance. The electrical resistance has a power of 15 kW and it is controlled by a proportional controller, programmed in the PLC, which has the function of keeping the temperature of water stable at the evaporator inlet. A centrifugal pump with by-pass valves makes possible the control of the mass flow rate of water circulating in the evaporator.

During the tests, the parameters responsible for the simulation of thermal load were kept constant for all refrigerants, i.e. water temperature and mass flow rate at the evaporator inlet were maintained at 20°C and 0.40 kg/s. These values assured the required conditions of evaporation and a value for the minimum degree of superheat above 2 K.

RESULTS AND DISCUSSION

The correct operation of any refrigeration system operating according to the vapor compression cycle requires that some thermodynamic parameters are monitored and controlled; among them are the temperatures of evaporation and condensation, the degree of superheat (measured immediately after the evaporator outlet) and the degree of subcooling.

The power consumed by the compressor can be considered as a limiting parameter for the operation of the four systems. The value of this parameter should not exceed 3.8 kW, thus preserving the life of the frequency converter installed with the compressor.

First stage of results

The tests of systems containing R410A and R32 were performed over a range of lower frequency (32Hz to 47Hz) compared to the range of other fluids (35Hz to 65Hz), higher frequencies were not analyzed due to the fact that the two systems (HFCs) responded with a high consumption of electric power above 3.9 kW. For each of the four systems 13 tests were planned according to the levels of the factors of Table 1.

Table 1 Level values of the operating frequency and the opening of the EEV.

System	Experimental design factors					
	F_{VSC} [Hz]			$Opening_{EEV}$ [%]		
	Lower	Central	Upper	Lower	Central	Upper
R22	40.0	50.0	60.0	50.0	70.0	90.0
R290	40.0	50.0	60.0	50.0	70.0	90.0
R32	35.0	40.0	45.0	35.0	45.0	55.0
R410A	35.0	40.0	45.0	35.0	45.0	55.0

The responses surfaces, Figures 3-6, show the behavior of the refrigeration capacity according to variation of both factors for all refrigerants. It is noteworthy that in all refrigerants, the conditions of evaporation from the main fluid stabilized during the tests at different values, representing a range of evaporation, -20°C to 5°C. The Table 2 refers to the conditions of evaporation and condensation in each of the four systems.

Table 2 Values of the average saturation temperature on heat exchangers.

System	Saturation conditions			
	Evaporator		*Condenser*	
	Average Temperature [°C]	Standard deviation [°C]	Average Temperature [°C]	Standard deviation [°C]
R22	-7.61	8.77	34.12	3.53
R290	-4.04	9.55	36.47	3.13
R32	-9.53	5.78	31.46	0.53
R410A	-14.09	5.90	32.42	0.36

Observing each response surface, note the existence of 13 white dots (experimental points) tangencies the colored surface, 8 points form an octagon and the other 5 are grouped in the central portion.

The behavior of the refrigeration capacity in the system containing R290 was similar to the behavior of the capacity in the system containing the R22, but for the system with R290 the values were higher.

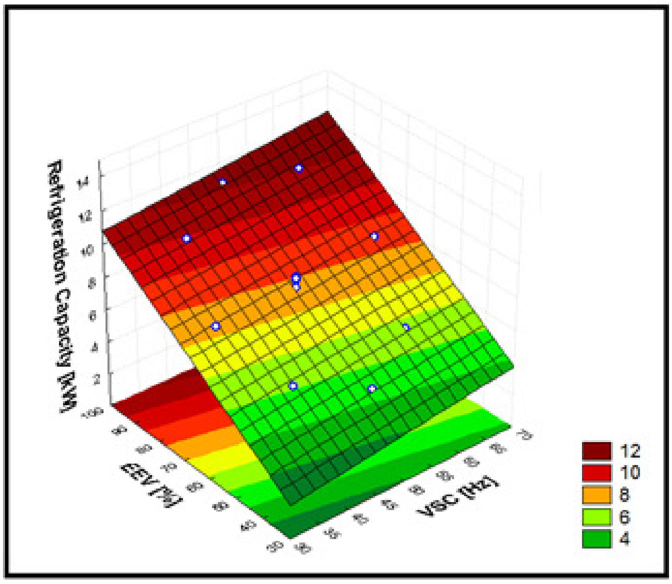

Figure 3 Response surface of refrigeration capacity, R22.

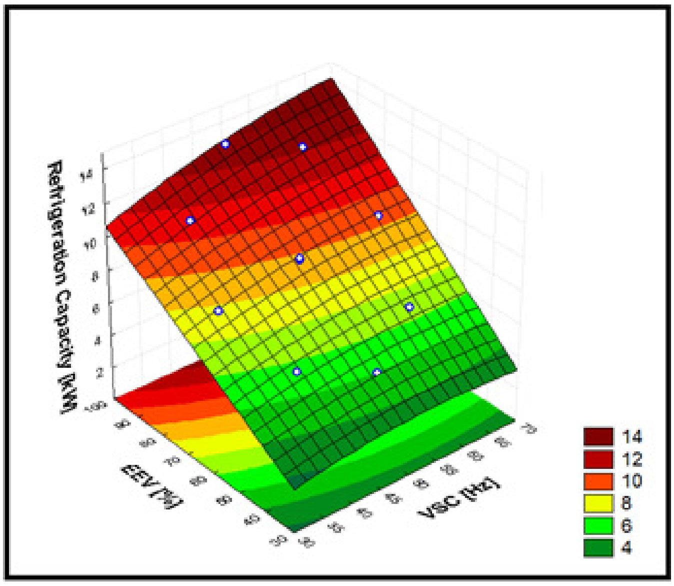

Figure 4 Response surface of refrigeration capacity, R290.

It is noted in Figures 5-6 that the 13 experimental points are located in the lower left portion of the colored surface, this region is considered a security zone for the compressor. Other operating conditions of this system that depart from this region may compromise the frequency inverter that powers the electric motor of the compressor. Furthermore, the elevation of discharge temperature and consequent wear of the mechanical components of the compressor can occur.

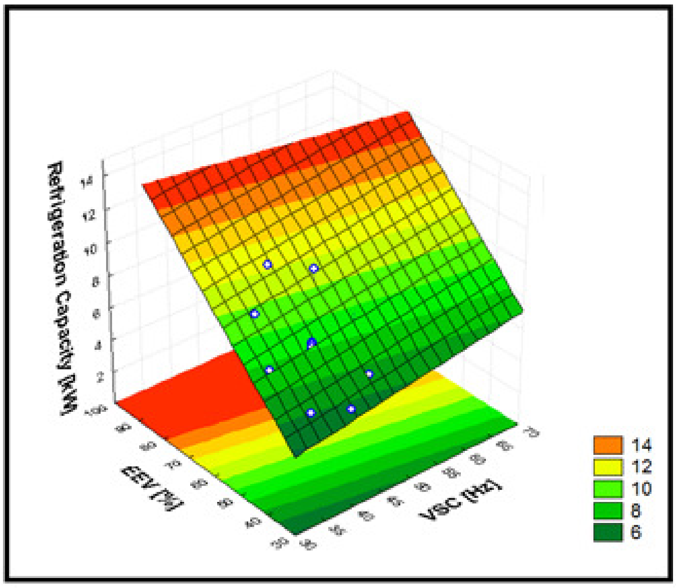

Figure 5 Response surface of refrigeration capacity, R32.

The system with R32, even operating in the security zone, responded with significant values for the rate of heat transfer in the evaporator. It is also observed through a less rigorous analysis, that the systems containing R22 or R290 do not return values as high refrigeration capacity when operating at low speeds and small openings of EEV.

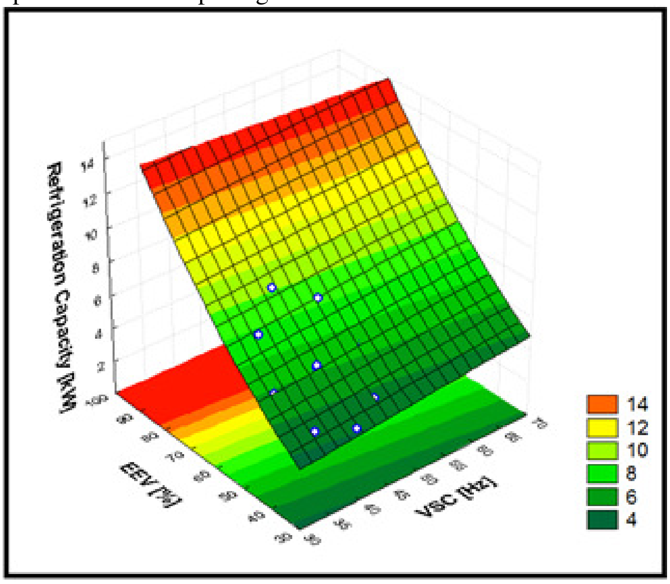

Figure 6 Response surface of refrigeration capacity, R410A.

The system with R410A, which also operated in the security zone, returned lower refrigeration capacities compared to the system containing the R32.

The maximum relative error, based on the uncertainties of the sensors, did not exceed 2% for the results of these 52 tests.

The main conclusion refers to the flexibility of system operation. Based on the fixed water conditions at the evaporator inlet, different refrigeration capacities were achieved due to the use of EEV and VSC for each refrigerant.

These conditions were shown to be suitable for an application where the thermal load be varied during operation of the system.

Second stage of results

In addition, Table 3 makes reference to the second stage of experimental tests. The four fluids operated under different conditions of evaporation.

It should be noted that the values of refrigeration capacity at -15°C stabilized at 7.5 kW for all refrigerants, when the systems operated at -10°C this capacity level rose to 8.9 kW. Finally, the refrigeration capacity reached to 11.0 kW for an evaporation temperature of -5°C.

Table 3 Results of the second experimental stage.

System	F_{VSC} [Hz]	T_{CD} [°C]	T_{DC} [°C]	ΔT_{SH} [°C]	ΔT_{SC} [°C]	$\dot{m}$ [kg/s]	$\dot{Q}_{REF}$ [kW]	$\dot{W}_{VSC}$ [kW]	COP [-]
T_{EV} = −15.00 [°C]									
R22	65.0	36.30	100.10	35.00	9.30	0.0390	7.41	2.92	2.54
R290	65.0	37.80	85.00	36.00	12.50	0.0212	7.46	2.71	2.75
R32	45.0	31.90	117.00	33.30	1.30	0.0260	7.50	3.04	2.47
R410A	45.0	32.60	108.00	33.00	3.60	0.0370	7.42	3.12	2.38
T_{EV} = −10.00 [°C]									
R22	65.0	37.70	94.00	27.50	8.70	0.0478	8.84	3.22	2.83
R290	65.0	39.70	79.50	29.00	11.70	0.0262	8.89	2.89	3.08
R32	45.0	32.10	113.30	25.00	1.20	0.0316	8.92	3.28	2.72
R410A	45.0	32.90	103.00	25.80	2.40	0.0459	8.90	3.36	2.65
T_{EV} = −5.00 [°C]									
R22	65.0	39.90	89.00	21.60	8.30	0.0610	10.96	3.51	3.12
R290	65.0	40.30	75.20	22.00	11.20	0.0332	10.97	3.18	3.45
R32	45.0	32.20	104.00	17.60	1.00	0.0395	10.91	3.57	3.06
R410A	45.0	33.00	99.00	19.50	2.30	0.0575	10.93	3.63	3.01

The system containing R22, as well as the system with R290, operated at 65Hz. While others systems operate with lower compressor speeds, 45Hz. The immediate explanation for the use of reduced frequency operation in systems with HFCs is related to the high energy consumption by the current reciprocating compressor installed in the experimental facility. A possible gain in efficiency for these systems would be achieved if we used a scroll compressor; this equipment is more suitable for operations with HFCs that present higher saturation pressure curves.

It is important to highlight that the consumption is greatly influenced by the operating frequency of the compressor. Moreover, no system operated above 3.8 kW, thus respecting the limits of operation and safety of the equipment.

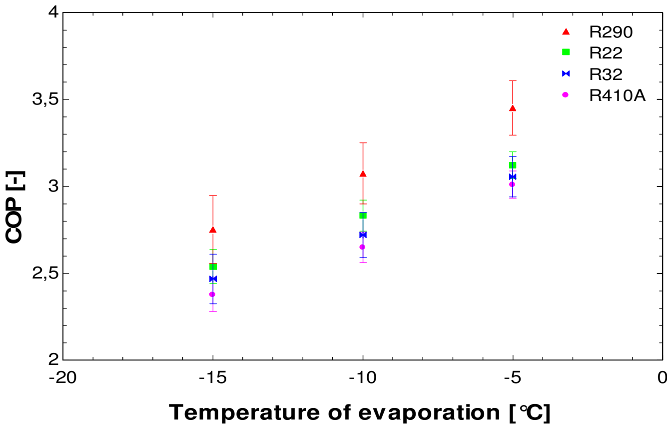

Figure 7 Comparative tests between the four refrigerants, behavior of COP according to conditions of evaporation.

The behavior of the COP, Figure 7, proves that the system containing the HC is excellent for applications in medium evaporating temperatures, exceeding the values of COP from other systems.

CONCLUSION

The four refrigeration systems (R22, R290, R410A and R32) have operated in a flexible way. The VSC operated at different frequencies and EEV modulated between 30 and 100%. The results (response surfaces) obtained in steady state showed that replacement of R22 is really possible. The use of the hydrocarbon R290 resulted in the maximum values of refrigeration capacity, exceeding the R22 in refrigeration applications.

The systems containing HFCs proved to be competent for applications at low speeds, since, by operating at frequencies above 50 Hz, these systems returned high values for energy consumption.

Finally, the analysis of the behavior for all systems at the same refrigeration capacity proved that the system containing the HC has the higher COP for the whole range of evaporation explored in this research.

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