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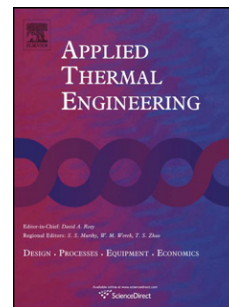
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Experimental evaluation of HCFC-22 replacement by the drop-in fluids HFC-422A and HFC-417B for low temperature refrigeration applications

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ABSTRACT

The Regulation CE-1005/2009 has banned the use of virgin HCFC-22 in Europe from the 31st December 2009, because it contains chlorine, although its use is allowed up to the 31st December 2014 if it is recycled. Now, manufacturers are developing different free-chlorine drop-in solutions to replace the HCFC-22 in the existing equipment by non-ozone depleting substances in order to exhaust its remaining life. This work focuses on the experimental evaluation of two HCFC-22 drop-in solutions for low temperature applications, the HFC-422A and the HFC-417B, in a double-stage vapour compression plant driven by a compound compressor in an evaporating temperature range between -31 to -17°C and in a condensing temperature range between 30 to 48°C. The experimental results show that there is a reduction in cooling capacity and in COP with regard to the HCFC-22 operation, which is analysed and discussed in the paper.

KEYWORDS

Drop-in; R22; two-stage system; compound compressor; R422A; R417B

NOMENCLATURE

Nomenclature

COP	<i>coefficient of performance</i>
h	<i>enthalpy ($\text{kJ}\cdot\text{kg}^{-1}$)</i>
\dot{m}	<i>refrigerant mass flow rate ($\text{kg}\cdot\text{s}^{-1}$)</i>
P_c	<i>compressor power consumption (kW)</i>
\dot{Q}_c	<i>cooling capacity (kW)</i>
q_o	<i>specific refrigerating effect ($\text{kJ}\cdot\text{kg}^{-1}$)</i>
t	<i>compression ratio</i>
T	<i>temperature ($^{\circ}\text{C}$)</i>
\dot{V}_G	<i>compressor displacement ($\text{m}^3\cdot\text{s}^{-1}$)</i>
VCC	<i>volumetric cooling capacity ($\text{kJ}\cdot\text{m}^{-3}$)</i>
w	<i>specific compression work ($\text{kJ}\cdot\text{kg}^{-1}$)</i>

Greek symbols

v	<i>specific volume ($\text{m}^3\cdot\text{kg}^{-1}$)</i>
η_v	<i>volumetric efficiency</i>
ε	<i>subcooler thermal effectiveness</i>
λ	<i>latent heat of phase change ($\text{kJ}\cdot\text{kg}^{-1}$)</i>
Δ	<i>increment</i>

Subscripts

$Disc$	<i>compressor discharge</i>
H	<i>high-compression stage</i>
i	<i>inlet, intermediate pressure/temperature</i>
k	<i>condenser</i>
l	<i>saturated liquid</i>
L	<i>low-compression stage</i>
o	<i>outlet</i>
O	<i>evaporator</i>
s	<i>isentropic</i>
Sl	<i>subcooler liquid line</i>
Sv	<i>subcooler vapour line</i>
$subc$	<i>subcooler</i>
Suc	<i>compressor suction</i>
v	<i>saturated vapour</i>

1. INTRODUCTION

The use of HCFC-22 virgin refrigerant has been banned by law by the Regulation CE-1005/2009 from the 31st December 2009 in European countries, being authorised the refilling of systems with HCFC-22 until the 31st December 2014 only if it comes from a recycling process, because it contains chlorine atoms in its composition. At present, the HCFC-22 represents a 97.2% of the HCFC substances used in the refrigeration sector [1] and it is supposed it will be a shortage of this substance in a few years because of the need for refilling the refrigeration plants currently operating with the HCFC-22. In this situation, two families of fluids stand out as long-term alternative refrigerants, the natural substances (hydrocarbons, ammonia and carbon dioxide) and the HFC artificial refrigerants, however, neither of those fluids can operate without important modifications in the existing HCFC-22 systems.

In the last years, manufacturers have been developing different free-chlorine refrigerant mixtures to substitute the HCFC-22 in order to exhaust the remaining life of the existing refrigerating equipment with minor modifications. In air conditioning, the HFC-422D and the HFC-417A are the most recommended drop-in fluids [2], whose heat transfer characteristics were investigated by Rosato *et al.* [3] and Fernández-Seara *et al.* [4], and whose performance was tested by Aprea *et al.* [5] and Torrella *et al.* [6]. Among the drop-in substances for commercial refrigeration in medium and low evaporating temperatures, the HFC-422A and the HFC-417B correspond to two possible substitutes, being their main properties those shown in Table 1. Both fluids are ternary blends of HFC-134a and HFC-125, with different proportion between them, and with a small proportion of different hydrocarbons (iso-butane for the HFC-422A and n-butane for the HFC-417B) in order to improve the oil mixing behaviour of mineral (MO) and alkylbenzene (AB) lubricants. Furthermore, they are completely compatible with polyol-esters (POE) oils. Although they were designed to match as far as possible the HCFC-22

typical characteristics, according to the manufacturers, it can be expected a reduction of capacity and efficiency of the plants with regard to the HCFC-22 when operating with any of the drop-in refrigerants, which corresponds to a disadvantage for this fluids. However, they state that the variations may vary depending on the system. This affirmation was verified by Arora and Sachdev [7] from a theoretical second law approach, however, no experimental results have been found.

Table 1. Physical, environmental and safety characteristics of HCFC-22, HFC-422A and HFC-417B [10, 14]

In order to cover the lack of the experimental research with the HCFC-22 drop-in refrigerants in medium and low evaporating temperatures, this work presents the energy evaluation of three refrigerants (HCFC-22, HFC-422A and HFC-417B) in the same refrigeration plant, which corresponds to a double-stage vapour compression plant driven by a compound compressor [8, 9]. The evaluation was made for an evaporating temperature range between -31 to -17°C for a constant condensing temperature of 40°C , and for a condensing temperature range between 30 to 48°C for a constant evaporating temperature of -25°C . In this communication, the energy performance of the HCFC-22 drop-in fluids is presented and analysed.

2. THERMODYNAMIC PROPERTIES AND THEORETICAL PERFORMANCE

Despite the drop-in fluids are designed in order to match as close as possible the properties of the refrigerant they substitute, there are always small variations on the thermodynamic properties. In the case of the fluids R422A and R417B, whose pressure-enthalpy diagram is plotted in Figure 1, they have a slight higher pressure for a given phase change temperature than the R22, however, the compression ratios are similar. On the other side, both drop-ins have a temperature glide (Table 1), which makes them not recommended for flooded evaporators. One advantage of them is the specific volume of saturated vapour, which is lower up to a 40% for the R422A and up to a 33% for the R417B with regard to the R22 for an evaporating level of -30°C . Accordingly, if the specific volume is lower, an important increment on the refrigerant mass flow rate through the plants is to be expected. Finally, the last important variation with regard to the R22 is the latent heat of phase change, which constitutes a disadvantage, being it reduced up to a 27% for the R422A and up to a 26% for the R417B inside the evaluation range. This reduction of the latent heat of phase change will tend to reduce the cooling capacity provided by the plants.

Figure 1. Pressure-enthalpy diagram of the R22 and the drop-in fluids [9].

To analyse what the variation in capacity when replacing the R22 by its drop-in fluids will be, it has been calculated with the Refprop database [10] the theoretical volumetric cooling capacity (VCC) (Equation 1) and the theoretical COP (Equation 2) of the double stage vapour compression cycle with subcooler used in the experimental evaluation [11]. It has been considered that the refrigerant at the exit of the evaporator, condenser and subcooler are saturated, that the inter-stage pressure is equal to the geometric pressure between

condensation and evaporation, the compression processes are ideal and the subcooler operates with constant thermal effectiveness of 80%.

$$VCC = \frac{q_o}{v_u|_{T_o}} = \frac{h_l|_{T_k} \cdot (\varepsilon - 1) - \varepsilon \cdot h_l|_{T_i} + h_v|_{T_o}}{v_u|_{T_o}} \quad (1)$$

$$COP = \frac{q_o}{w_s} = \frac{h_l|_{T_k} \cdot (\varepsilon - 1) - \varepsilon \cdot h_l|_{T_i} + h_v|_{T_o}}{w_{s,L} + \frac{h_v|_{T_i} + h_l|_{T_k} \cdot (\varepsilon - 1) - h_l|_{T_i}}{h_v|_{T_i} - h_l|_{T_k}} \cdot w_{s,H}} \quad (2)$$

The results are presented as percentage variation with regard to the R22 values in Figure 2 for the VCC and in Figure 3 for the COP. As can be observed in Figure 2, in the case of the R422A the variation in capacity with regard to the R22 operation can be positive or negative depending on the condensing level, although no important variations are expected. However, with the R417B, the theoretical results predict a reduction in the VCC for the evaluated condensing temperatures. On the other side, regarding the theoretical COP variations, for both fluids is expected a reduction with regard to the R22 operation between 10 to 20%. This reduction will be higher when higher the condensing temperature is and lower the evaporating level is.

Figure 2. Theoretical VCC variation of the ideal double-stage cycle with regard to the R22.

Figure 3. Theoretical COP variation of the ideal double-stage cycle with regard to the R22.

According to a theoretical analysis when replacing the R22 by the analysed drop-in fluids (R422A and R417B) for medium and low evaporating temperatures, it can be expected a variation in capacity that can be higher or lower for the R422A and lower for the R417B, and a reduction in COP for both fluids. However, these trends have been predicted by a theoretical

analysis of the double-stage cycle with subcooler and no real effects of the compressor efficiencies and variation of the subcooler thermal effectiveness have been taken into account. Furthermore, as stated by the manufacturers the trends can be different depending on the refrigeration system. Accordingly, this study has been complemented with the experimental analysis on a refrigeration plant, which is presented in the next section.

3. EXPERIMENTAL PLANT AND TEST PROCEDURES

3.1. Experimental test plant

The experimental plant used for the evaluation of the drop-ins performance corresponds to a double-stage refrigeration plant driven by a compound compressor, which was initially designed for the operation with the R22. The schematic plant's diagram and its temperature-entropy cycle are shown in Figure 4 and Figure 5 respectively. The plant is driven by a compound compressor of 4kW, condenser and evaporator are brazed-plate heat exchangers, incorporates a liquid receiver at the condenser exit and controls the evaporation process with a thermostatic expansion valve with external equalization. It also incorporates a liquid subcooling system at the inter-stage level composed by a brazed-plate subcooler that is controlled by a thermostatic expansion valve. More information about the cycle operation can be found in [8, 9, 11, 12].

Figure 4. Schematic plant's diagram

Figure 5. Real Ts cycle for R22 at $T_o=-30^{\circ}\text{C}$ and $T_k=40^{\circ}\text{C}$

The test plant is completely instrumented with 17 T-type thermocouples ($\pm 0.1^\circ\text{C}$), with 10 pressure gauges (± 0.1 bar at low pressure and ± 0.3 bar at medium and high temperature), with two Coriolis mass flow meters ($\pm 0.22\%$ of reading) to measure refrigerant mass flow rate, two magnetic volumetric flow meters ($\pm 0.33\%$ of reading) to measure volumetric flow rates of secondary fluids and a digital wattmeter ($\pm 0.5\%$ of reading) to measure the compressor power consumption. The heat transfer rates in the evaporator and condenser were validated with the heat transferred by the secondary fluids, which are a (50/50% by vol.) water/ethylene-glycol mixture in the evaporator and water in the condenser. The refrigerant properties were evaluated with the Refprop database [10].

3.2. Test procedures and measurements

The objective of the energy evaluation, which is presented in this work, was to study the performance of the refrigerants when operating at the same evaporating and condensing temperatures. Accordingly, the base of comparison recommended by Radermacher and Hwang [13] was followed. This criterion considered the condensing temperature for a vapour title in the condenser of 50% and the evaporating temperature the corresponding to the mean enthalpy value in this heat exchanger, that is, the average enthalpy of the refrigerant at the inlet and outlet of the evaporator. Considering this criterion, the three refrigerants were evaluated for a wide range of evaporating and condensing temperatures (

FLUID	Evaporating temperature test		Condensing temperature test	
	T_o ($^\circ\text{C}$)	T_k ($^\circ\text{C}$)	T_o ($^\circ\text{C}$)	T_k ($^\circ\text{C}$)
R22	-31.4 to -17.3	40.0 ± 0.5	-25.0 ± 0.56	31.0 to 47.2
R422A	-30.6 to -19.8	40.0 ± 0.3	-25.0 ± 0.20	33.8 to 46.6
R417A	-31.7 to -17.6	40.0 ± 0.2	-25.0 ± 0.18	34.8 to 48.5

Table 2) while keeping the compressor speed at its nominal value (1450 rpm).

FLUID	Evaporating temperature test		Condensing temperature test	
	T_o (°C)	T_k (°C)	T_o (°C)	T_k (°C)
R22	-31.4 to -17.3	40.0 ± 0.5	-25.0 ± 0.56	31.0 to 47.2
R422A	-30.6 to -19.8	40.0 ± 0.3	-25.0 ± 0.20	33.8 to 46.6
R417A	-31.7 to -17.6	40.0 ± 0.2	-25.0 ± 0.18	34.8 to 48.5

Table 2. Temperature evaluation range of the refrigerants.

The refrigerants were tested using the POE lubricant oil present in the plant (viscosity index 120), since was the oil used in previous analysis [8]. The plant's refrigerant charge was that that ensured the line at the exit of the condenser to be filled with liquid refrigerant at a condensing temperature of 25°C and an evaporating temperature of -20°C, which corresponds to the extreme plant's operation condition which needs more refrigerant. These charges of refrigerant are presented in

FLUID	R22	R422A	R417B
Refrigerant charge (kg)	8.75	8.14	8.37

Table 3 for the fluids. It can be observed that the refrigerant charges with the drop-in fluids are smaller than for the R22, which can be considered as an advantage for the drop-ins. To avoid uncertainties related with the use of the same lubricant, the tests were ordered in time as follow: R22, R417B, R422A, R417B and R422A. With the drop-ins the tests were repeated twice and no appreciable variations in the energy performance was found.

FLUID	R22	R422A	R417B
Refrigerant charge (kg)	8.75	8.14	8.37

Table 3. Refrigerant charge with the different fluids.

Regarding the operation of the plant, the unique modification was the adjustment of the thermal expansion valve of the evaporator, which was regulated in order to maintain the same degree of superheat at the evaporator exit for all the refrigerants.

4. RESULTS AND DISCUSSION

This section is devoted to present and analyse the experimental results of the evaluation of the refrigerant R22 and its recommended drop-ins (R422A and R417B) in medium and low evaporating temperatures in the double-stage refrigeration plant. The evolution of the main plant's parameters is presented in Figures 6 to 12 and the values and percentage variation from the R22 operation, for an evaporating temperature of -30°C and condensing of 40°C , are presented in

	t	t_L	t_H	$v_{Suc,L}$ ($\text{m}^3 \cdot \text{kg}^{-1}$)	$\eta_{V,L}$	m_o ($\text{kg} \cdot \text{s}^{-1}$)	m_k ($\text{kg} \cdot \text{s}^{-1}$)	ϵ_{subc}	Δh_{subc} ($\text{kJ} \cdot \text{kg}^{-1}$)	$h_{o,o} - h_{k,o}$ ($\text{kJ} \cdot \text{kg}^{-1}$)	q ($\text{kJ} \cdot \text{kg}^{-1}$)
R22	8.54	2.90	2.95	0.134	0.567	0.026	0.032	0.904	37.9	144.9	18.2
R417B	8.88	3.06	2.90	0.091	0.521	0.036	0.050	0.872	36.5	81.4	11.2
R22 variation (%)	3.9	5.7	-1.7	-32.1	-8.2	35.9	52.4	-3.5	-3.8	-43.8	-37.3
R422A	8.53	3.03	2.82	0.081	0.492	0.038	0.051	0.820	34.9	77.5	10.2
R22 variation (%)	-0.2	4.4	-4.4	-39.5	-13.2	43.0	55.8	-9.3	-7.9	-46.5	-44.5

Table 4.

4.1. Refrigerant mass flow rate

The parameter that most influences the cooling capacity and therefore the efficiency of the plant is the refrigerant mass flow rate through the low compression stage, which is the same that flows through the evaporator (Equation 3). It mainly depends on the compressor displacement at the low-stage, which is constant for the three refrigerants, and on the volumetric efficiency and the specific suction volume of the refrigerant at the low-stage. [11]

$$\dot{m}_o = \frac{\eta_{v,L}}{v_{suc,L}} \cdot \dot{V}_{G,L} \quad (3)$$

The experimental measurements of the refrigerant mass flow rate through the evaporator are presented in Figure 6 for the evaporating temperature variation test at a constant condensing temperature of 40°C (uncertainty 1.55%). As can be observed in Figure 6, there is an important increment on the refrigerant mass flow rate through the evaporator when any of the drop-in fluids are used. The increment is similar for both fluids although at low evaporating levels there is a slight increment for the R422A. The main reason of this rise of mass flow rate is the reduction of the specific suction volume at the compressor suction for the drop-in fluids. On the other side, small variations on the volumetric efficiency at the low-stage have been measured, this parameter being reduced when using any of the drop-in fluids (

	t	t_L	t_H	$v_{suc,L}$ ($m^3 \cdot kg^{-1}$)	$\eta_{v,L}$	m_o ($kg \cdot s^{-1}$)	m_k ($kg \cdot s^{-1}$)	ϵ_{subc}	Δh_{subc} ($kJ \cdot kg^{-1}$)	$h_{o,o} - h_{k,o}$ ($kJ \cdot kg^{-1}$)	q (kJ)
R22	8.54	2.90	2.95	0.134	0.567	0.026	0.032	0.904	37.9	144.9	18
R417B	8.88	3.06	2.90	0.091	0.521	0.036	0.050	0.872	36.5	81.4	18
R22 variation (%)	3.9	5.7	-1.7	-32.1	-8.2	35.9	52.4	-3.5	-3.8	-43.8	-
R422A	8.53	3.03	2.82	0.081	0.492	0.038	0.051	0.820	34.9	77.5	18
R22 variation (%)	-0.2	4.4	-4.4	-39.5	-13.2	43.0	55.8	-9.3	-7.9	-46.5	-

Table 4).

Figure 6. Evaporator refrigerant mass flow rate. $T_k=40^\circ\text{C}$

This increment on the refrigerant mass flow rate respect to the R22 will force in some cases to replace the orifice of the thermal expansion valves in existing R22 plants.

4.2. Cooling capacity and specific refrigerating effect

The second energy parameter of the plant corresponds to the cooling capacity (Equation 4), which in this plant depends on the refrigerant mass flow rate through the evaporator and the specific refrigerating effect. The specific refrigerating effect is function of the enthalpy value at the exit of the evaporator, the enthalpy of the liquid at the exit of the condenser and the enthalpy increment due to the liquid subcooling in the condenser. This last parameter is determined by the thermal effectiveness of the subcooler and the enthalpy difference between saturated liquid at the condensing and inter-stage temperatures (Equation 5), where the enthalpy difference due to the reheating of the refrigerant at the subcooler has been neglected [11].

$$\dot{Q}_o = \dot{m}_o \cdot q_o = \dot{m}_o \cdot (h_{o,o} - h_{k,o} + \Delta h_{subc}) \quad (4)$$

$$\Delta h_{subc} \cong \varepsilon_{subc} \cdot (h_l|_{T_k} - h_l|_{T_i}) \quad (5)$$

In Figure 7, the specific refrigerating effect is presented for the three fluids at different evaporating levels and a constant condensing temperature of 40°C. As can be observed there is a large reduction on this value with regard to the R22 operation when using any of the drop-in refrigerants. This reduction mainly depends on the latent heat of phase change of the refrigerants, which is reduced for the drop-ins (Table 1). However, the enthalpy difference due to the subcooling process, and accordingly the thermal effectiveness of the subcooler, is similar for the three fluids (

	t	t_L	t_H	$v_{suc,L}$ ($m^3 \cdot kg^{-1}$)	$\eta_{V,L}$	m_o ($kg \cdot s^{-1}$)	m_k ($kg \cdot s^{-1}$)	ε_{subc}	Δh_{subc} ($kJ \cdot kg^{-1}$)	$h_{o,o} - h_{k,o}$ ($kJ \cdot kg^{-1}$)	q_o ($kJ \cdot kg^{-1}$)
R22	8.54	2.90	2.95	0.134	0.567	0.026	0.032	0.904	37.9	144.9	182.8
R417B	8.88	3.06	2.90	0.091	0.521	0.036	0.050	0.872	36.5	81.4	117.9

R22 variation (%)	3.9	5.7	-1.7	-32.1	-8.2	35.9	52.4	-3.5	-3.8	-43.8
R422A	8.53	3.03	2.82	0.081	0.492	0.038	0.051	0.820	34.9	77.5
R22 variation (%)	-0.2	4.4	-4.4	-39.5	-13.2	43.0	55.8	-9.3	-7.9	-46.5

Table 4).

Figure 7. Specific refrigerating effect vs. evaporating temperature. $T_k=40^\circ\text{C}$.

According to the results presented in Figure 6 and Figure 7, there is a trend to increment the cooling capacity (increment of the refrigerant mass flow rate) and other to decrement this parameter (specific refrigerating effect), whose combination lead to the evolution of the cooling capacity of the plant. This parameter is presented, with an average uncertainty of 3.2%, in Figure 8 for the evaporating temperature variation test ($T_k=40^\circ\text{C}$) and in Figure 9 for the condensing temperature variation test ($T_o=-25^\circ\text{C}$). From the experimental results, it is obvious that there is a reduction with regard to the R22 on the cooling capacity provided by the plants when using any of the drop-in refrigerants. These reductions are of a 12.1% for the R417B and of 11.6% for the R422A when operating at a $T_o=-30^\circ\text{C}$ and a $T_k=40^\circ\text{C}$, and are smaller than the ones anticipated with the theoretical analysis (increment of 1% for the R422A and reduction of 6.7% for the R417B, Figure 2). This difference between the theoretical and experimental results can be associated to the variation of the volumetric efficiency at the low stage and the small differences on the subcooler thermal effectiveness (

	t	t_L	t_H	$v_{SUC,L}$ ($\text{m}^3 \cdot \text{kg}^{-1}$)	$\eta_{V,L}$	m_o ($\text{kg} \cdot \text{s}^{-1}$)	m_k ($\text{kg} \cdot \text{s}^{-1}$)	ϵ_{subc}	Δh_{subc} ($\text{kJ} \cdot \text{kg}^{-1}$)	$h_{o,o} - h_{k,o}$ ($\text{kJ} \cdot \text{kg}^{-1}$)	q ($\text{kJ} \cdot \text{s}^{-1}$)
R22	8.54	2.90	2.95	0.134	0.567	0.026	0.032	0.904	37.9	144.9	11

R417B	8.88	3.06	2.90	0.091	0.521	0.036	0.050	0.872	36.5	81.4	11.1
R22 variation (%)	3.9	5.7	-1.7	-32.1	-8.2	35.9	52.4	-3.5	-3.8	-43.8	-1.1
R422A	8.53	3.03	2.82	0.081	0.492	0.038	0.051	0.820	34.9	77.5	11.1
R22 variation (%)	-0.2	4.4	-4.4	-39.5	-13.2	43.0	55.8	-9.3	-7.9	-46.5	-1.1

Table 4).

Regarding the difference in performance with both fluids, the cooling capacity is slightly higher for the R417B at high evaporating and condensing temperatures, however, the capacity of both drop-ins coincides at low evaporating levels.

Figure 8. Cooling capacity vs. evaporating temperature. $T_k=40^\circ\text{C}$.

Figure 9. Cooling capacity vs. condensing temperature. $T_o=-25^\circ\text{C}$.

4.3. COP

Finally, the last energy parameter in importance is represented by the COP of the plant, which corresponds to the quotient between the cooling capacity and the compressor power consumption (Equation 6). The evolution of the COP is presented, with an average uncertainty of 3.3%, in Figure 10 for the evaporating temperature variation tests and in Figure 11 for the condensing temperature variation test.

The experimental results show that there is an important reduction in COP with regard to the values reached by the R22 when using any of the drop-in fluids. This difference is nearly constant for the same condensing level (Figure 11) and increases when lower the evaporating

level is (Figure 10), being the COP offered with the drop-in fluids similar at low evaporating temperatures.

$$COP = \frac{Q_e}{P_c} \quad (6)$$

Figure 10. COP vs. evaporating temperature. $T_k=40^\circ\text{C}$.

Figure 11. COP vs. condensing temperature. $T_o=-25^\circ\text{C}$.

From the results presented in Figures 10 and 11, it is obvious that the reduction in COP is important when using any of the drop-in fluids. Furthermore, it needs to be highlighted that the experimental reductions (26.0% for the R422A and 20.3% for the R417B) for the operation at $T_o=-30^\circ\text{C}$ and $T_k=40^\circ\text{C}$ are higher than the ones expected from the theoretical analysis (12.9% for the R422A and 11.9% for the R417B).

4.4. Compressor discharge temperature

Finally, although it is not much important from an energy point of view, is presented in Figure 12 the evolution of the compressor discharge temperature for the evaporating temperature variation test. When using any of the drop-in fluids it has been measured a reduction of 25 to 30°C on the compressor discharge temperature inside the test range, which is similar for both tested fluids. This reduction can be considered as an advantage for the drop-in fluids, since it will reduce problems regarding high temperatures and extend the compressor life.

Figure 12. Compressor discharge temperature vs. evaporating temperature. $T_k=40^\circ\text{C}$.

5. CONCLUSIONS

In this work, the HCFC-22 substitution by two recommended drop-in refrigerants (HFC-422A and HFC-417B) in medium and low evaporating temperatures has been analysed from a theoretical and an experimental approach. The experimental evaluation of the fluids was made with the same refrigerating test plant, a double-stage vapour compression plant with subcooler, in an evaporating temperature range from -31 to 17°C and in a condensing temperature range from 30 to 48°C .

The experimental results showed that when using any of the drop-in fluids there is an important increment on the refrigerant mass flow rate through the plants, which in some cases would need to readjust the expansion valves. However, with the substitutes there is an important reduction of the specific refrigerating effect, which tends to reduce the cooling capacity.

Regarding the energy performance with the R22 drop-in fluids it has been measured a reduction in the capacity in the plant, being this reduction higher than expected from a theoretical analysis. Furthermore, regarding the COP in medium and low evaporating temperatures, the reduction with the R22 substitutes is important and higher to the values predicted with a theoretical analysis.

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FIGURES

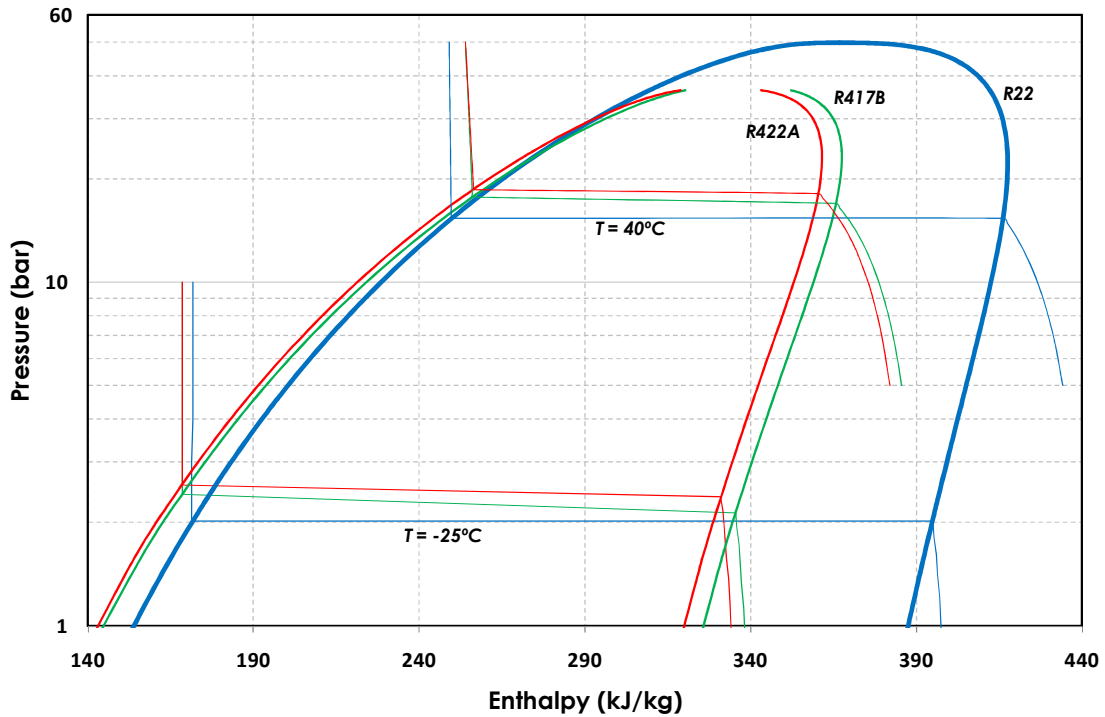


Figure 1. Pressure-enthalpy diagram of the R22 and the drop-in fluids [9]

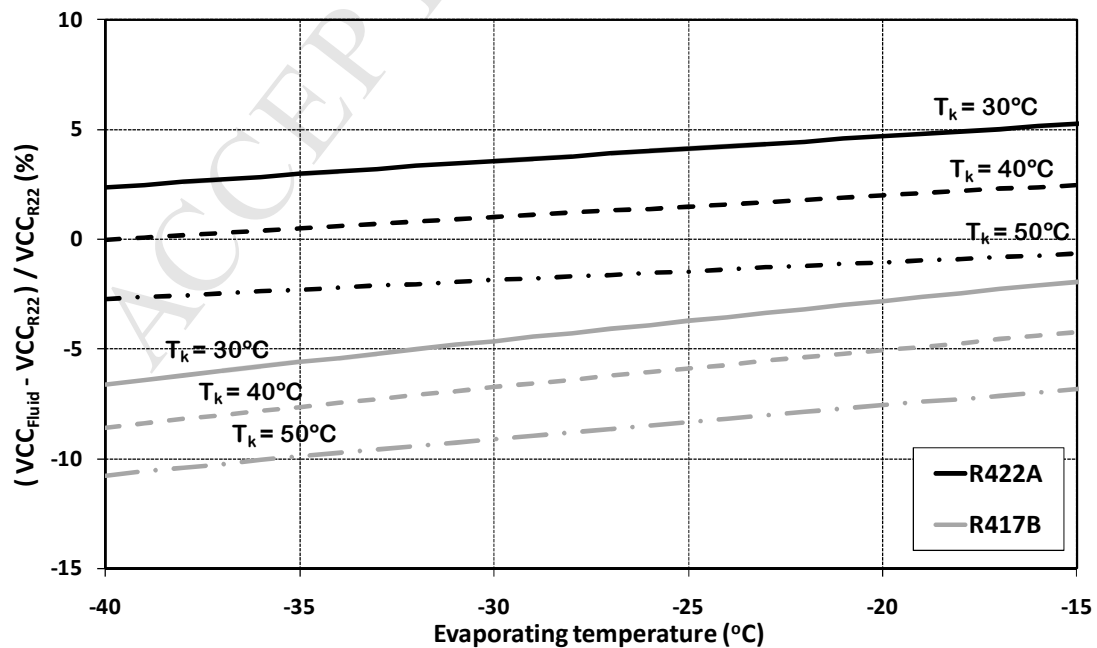


Figure 2. Theoretical VCC variation of the ideal double-stage cycle with regard to the R22.

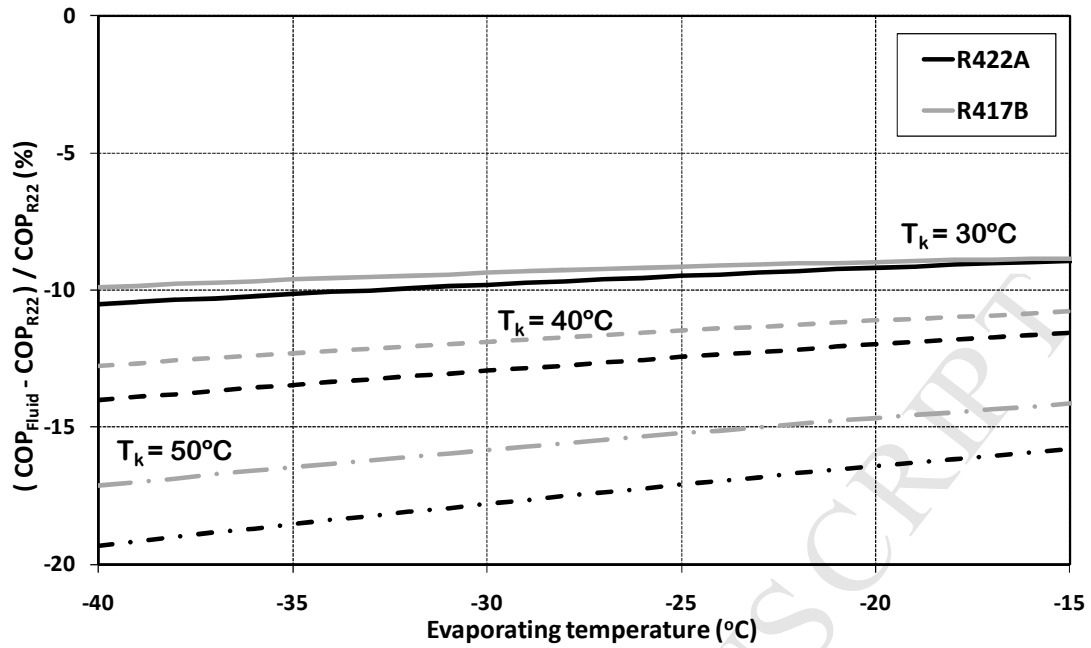


Figure 3. Theoretical COP variation of the ideal double-stage cycle with regard to the R22.

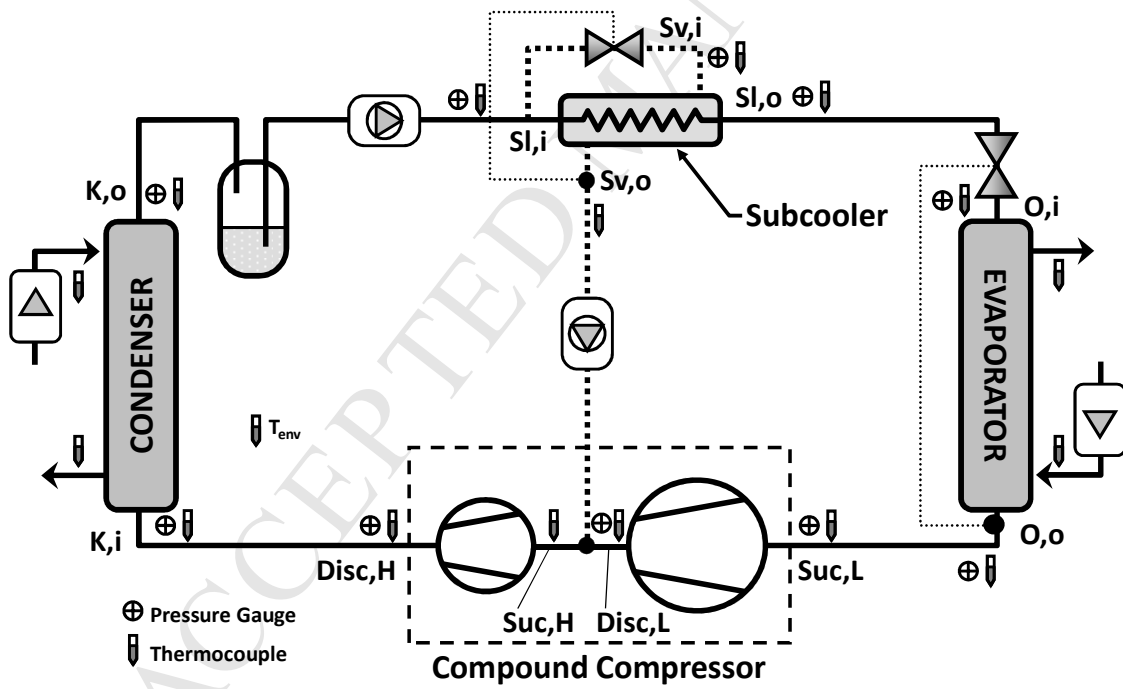


Figure 4. Schematic plant's diagram

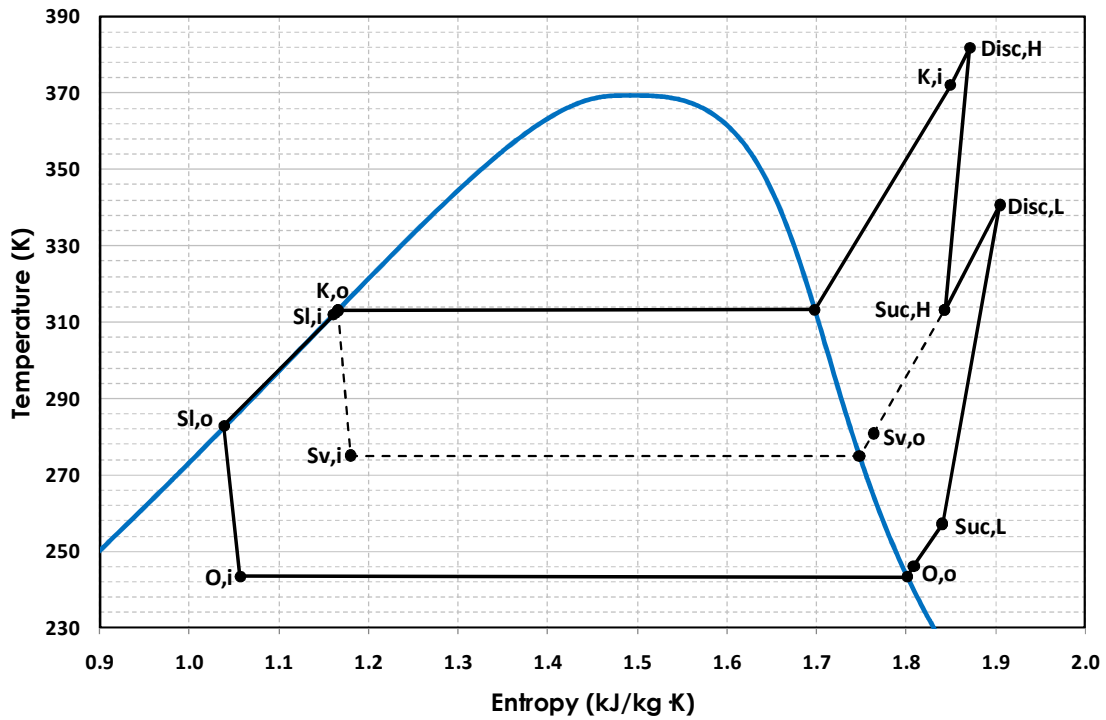


Figure 5. Real Ts cycle for R22 at $T_o=-30^{\circ}\text{C}$ and $T_k=40^{\circ}\text{C}$

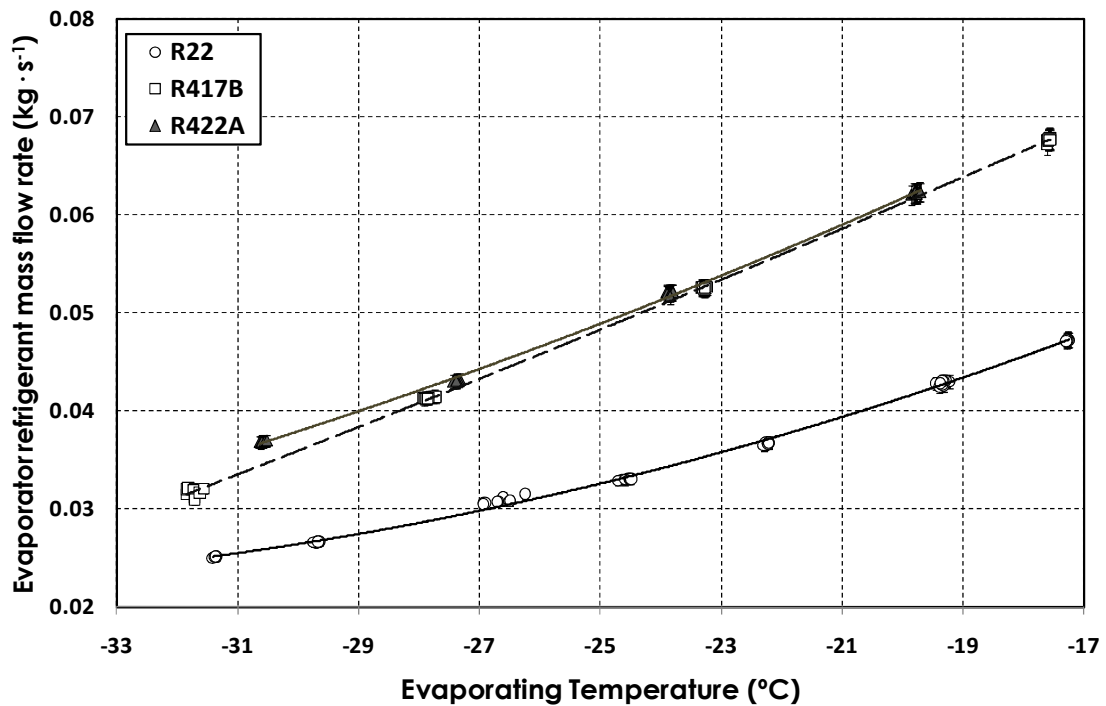


Figure 6. Evaporator refrigerant mass flow rate. $T_k=40^{\circ}\text{C}$

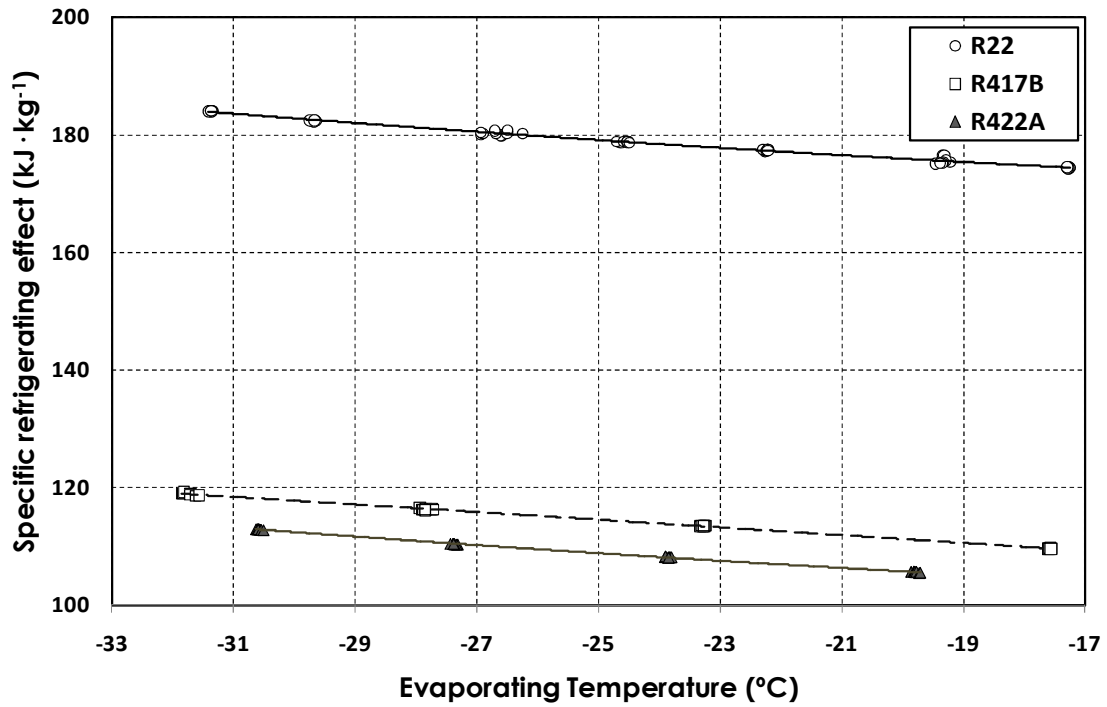


Figure 7. Specific refrigerating effect vs. evaporating temperature. $T_k=40^{\circ}\text{C}$.

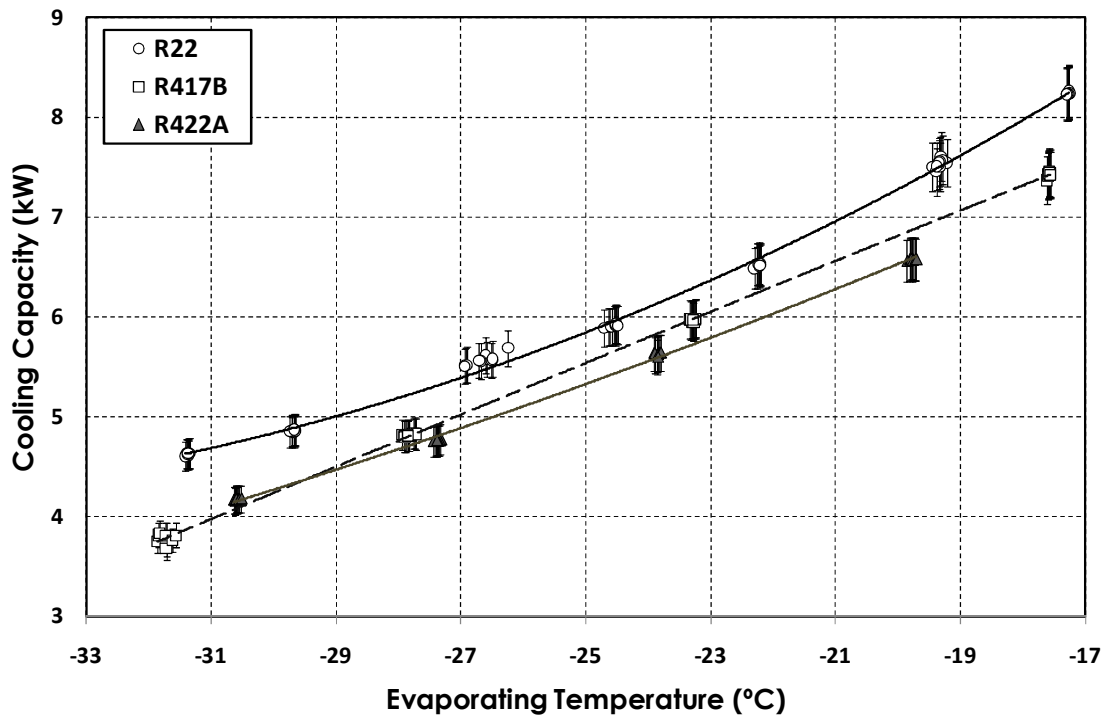


Figure 8. Cooling capacity vs. evaporating temperature. $T_k=40^{\circ}\text{C}$.

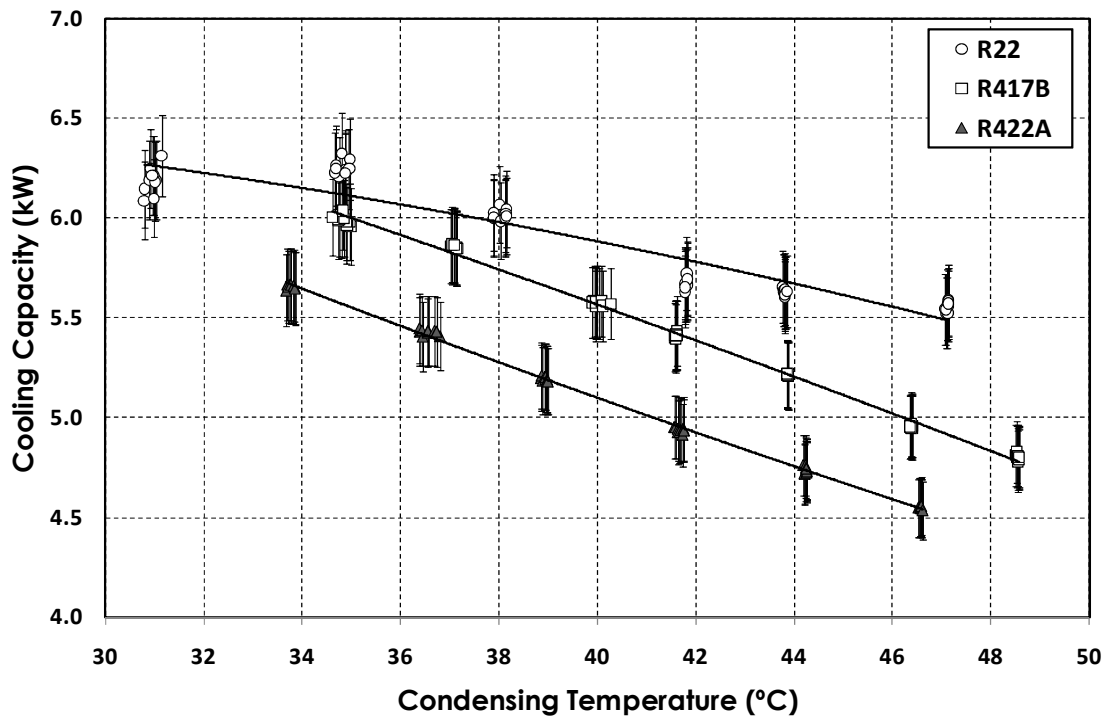


Figure 9. Cooling capacity vs. condensing temperature. $T_o = -25^\circ\text{C}$.

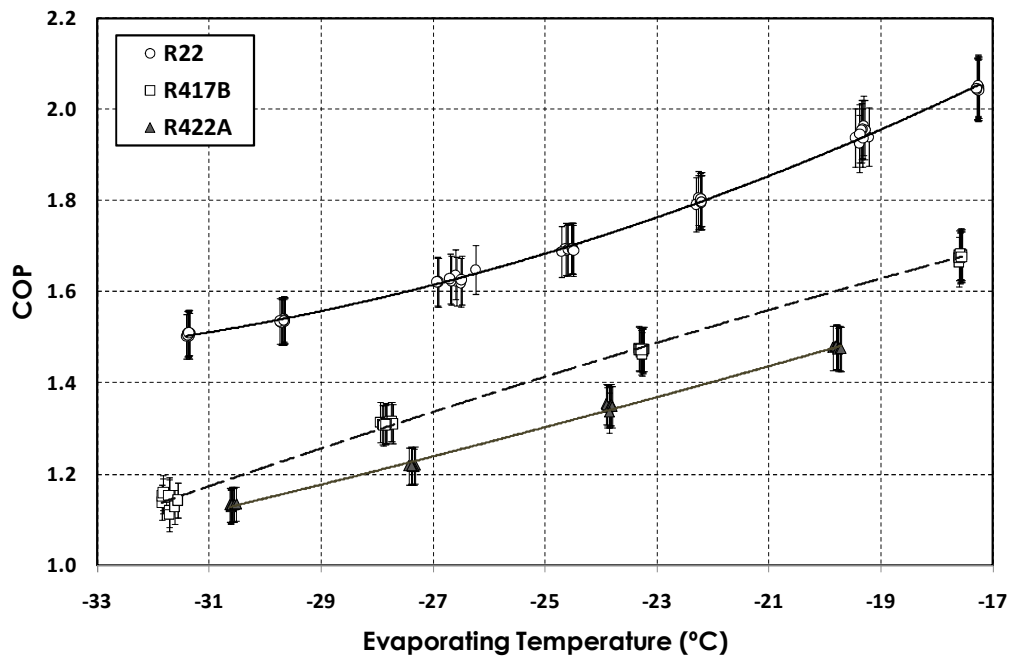


Figure 10. COP vs. evaporating temperature. $T_k = 40^\circ\text{C}$.

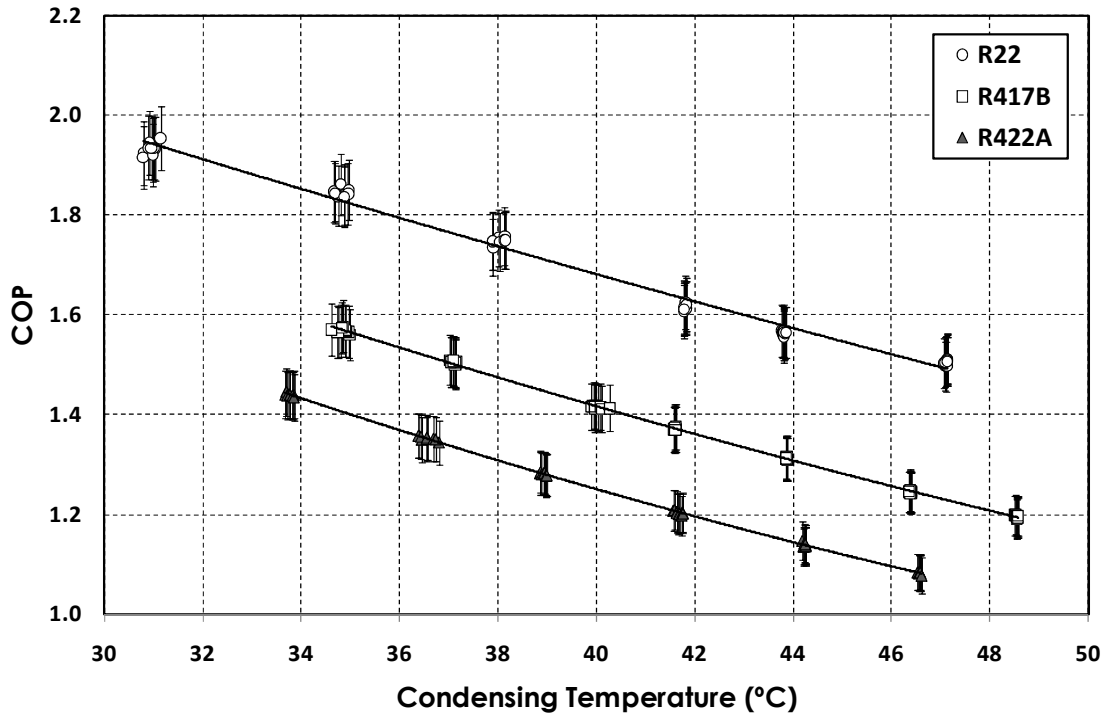


Figure 11. COP vs. condensing temperature. $T_o = -25^\circ\text{C}$.

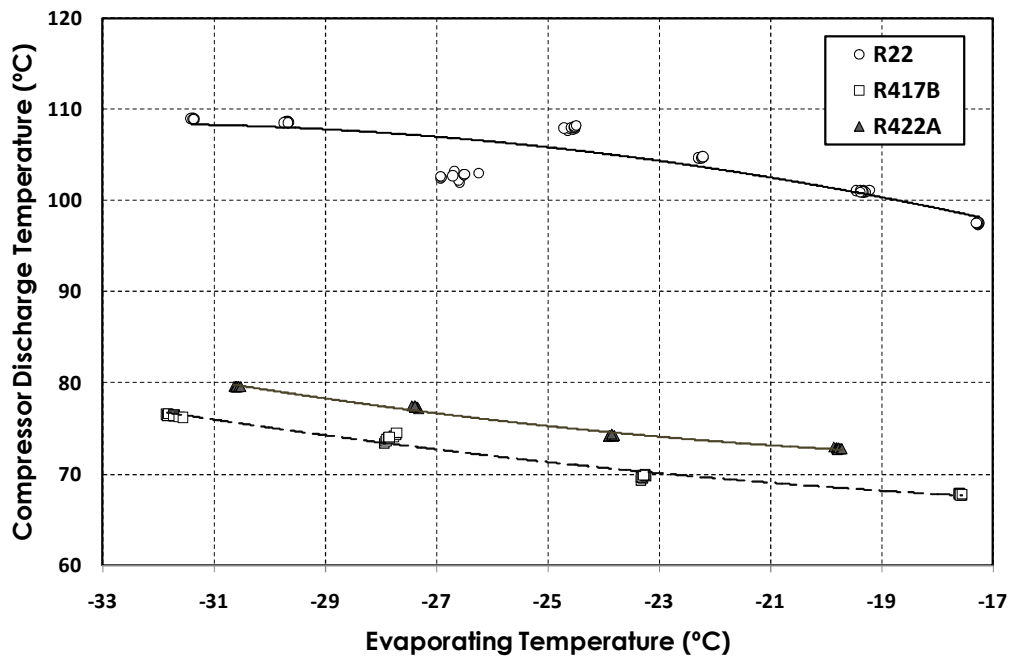


Figure 12. Compressor discharge temperature vs. evaporating temperature. $T_k = 40^\circ\text{C}$.

TABLES

	HCFC-22	HFC-422A	HFC-417B
Composition (% wt)	CHClF ₂	85.10% R125	79.00% R125
		11.50% R134a	18.25% R134a
		3.40% R600a	2.75% R600
Molecular weight (g·mol ⁻¹)	86.47	113.60	113.07
Normal boiling point (°C)	-40.81	-44.03	-41.51
Critical temperature (°C)	96.14	71.73	75.18
Critical pressure (bar)	49.90	37.46	37.78
Glide ^a (°C)	0.00	2.46	3.43
λ (T=-30°C)	226.81	167.44	172.70
λ (T=40°C)	166.60	104.80	111.17
ODP	0.05	0.00	0.00
GWP _{100 years}	1810	3100	3027
ASHRAE safety group	A1	A1	A1

^a Glide evaluated at saturation temperatures under standard atmosphere pressure (101.325 kPa)

Table 1. Physical, environmental and safety characteristics of HCFC-22, HFC-422A and HFC-417B [10, 14]

FLUID	Evaporating temperature test		Condensing temperature test	
	T_o (°C)	T_k (°C)	T_o (°C)	T_k (°C)
R22	-31.4 to -17.3	40.0 ± 0.5	-25.0 ± 0.56	31.0 to 47.2
R422A	-30.6 to -19.8	40.0 ± 0.3	-25.0 ± 0.20	33.8 to 46.6
R417A	-31.7 to -17.6	40.0 ± 0.2	-25.0 ± 0.18	34.8 to 48.5

Table 2. Temperature evaluation range of the refrigerants.

FLUID	R22	R422A	R417B
Refrigerant charge (kg)	8.75	8.14	8.37

Table 3. Refrigerant charge with the different fluids.

	t	t_L	t_H	$v_{Suc,L}$ ($m^3 \cdot kg^{-1}$)	$\eta_{V,L}$	m_o ($kg \cdot s^{-1}$)	m_k ($kg \cdot s^{-1}$)	ϵ_{subc}	Δh_{subc} ($kJ \cdot kg^{-1}$)	$h_{o,o} - h_{k,o}$ ($kJ \cdot kg^{-1}$)	q_o ($kJ \cdot kg^{-1}$)	Q_o (kW)	P_c (kW)	COP	T_{Disc} ($^{\circ}C$)
R22	8.54	2.90	2.95	0.134	0.567	0.026	0.032	0.904	37.9	144.9	182.8	4.83	3.16	1.53	108.2
R417B	8.88	3.06	2.90	0.091	0.521	0.036	0.050	0.872	36.5	81.4	117.9	4.25	3.48	1.22	79.2
R22 variation (%)	3.9	5.7	-1.7	-32.1	-8.2	35.9	52.4	-3.5	-3.8	-43.8	-35.5	-12.1	10.2	-20.3	-29.0 ($^{\circ}C$)
R422A	8.53	3.03	2.82	0.081	0.492	0.038	0.051	0.820	34.9	77.5	112.4	4.27	3.72	1.15	75.1
R22 variation (%)	-0.2	4.4	-4.4	-39.5	-13.2	43.0	55.8	-9.3	-7.9	-46.5	-38.5	-11.6	17.9	-26.0	-33.1 ($^{\circ}C$)

Table 4. Main parameters of the plant and deviation from the R22 operation for $T_o = -30^{\circ}C$, $T_k = 40^{\circ}C$.