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Test results of performance and oil circulation rate of commercial reciprocating compressors of different capacities working with propane (R290) as refrigerant

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Abstract

In this experimental investigation five R407C positive displacement hermetic reciprocating compressors, covering different capacities, displacement, stroke-to-bore ratios and number of cylinders, have been characterized using propane as refrigerant by means of a specifically designed characterization test rig. Test results have been systematically compared with their R407C reference performance data to obtain a complete picture on changes on the volumetric efficiency and compressor efficiency amongst others. The compressors used POE oil as lubricant and additional oil circulation rate (OCR) tests at steady state conditions were done to evaluate possible effects and differences to the traditionally used mineral oils.

Keywords: Hermetic compressor; Experiment; Propane; Comparison; R407C; Performance; Efficiency; Volumetric efficiency

Compresseurs à piston en application commerciale utilisant le propane (R290) comme frigorigène : résultats des études sur la performance et la vitesse de circulation de l'huile

Mots clés : Compresseur hermétique ; Expérimentation ; Propane ; Comparaison ; R407C ; Performance ; Efficacité ; Rendement volumétrique

1. Introduction

Within the global search after new environmentally safe alternatives to CFC and HCFC-based refrigerants, HFC's are seen by many as an almost unavoidable choice. However, HFC's have mainly two important disadvantages: first, despite their null ODP they show a high GWP, this fact has led some UE countries to put limits to their use; secondly, they are new chemical products that have never been present in the atmosphere and, for they have long mean life, nowadays unknown long term effects may arise. For these reasons research has been conducted to find other, environmentally unquestionable, alternatives to HFC's based on gases actually present in the atmosphere. In this context, hydrocarbons (HC's) and in a particular, propane

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Nomenclature

T_{ev}	evaporation temperature	$\Psi_{v,1}$	volumetric capacity
T_{cond}	condensation temperature	$\dot{V}_{sw}$	compressor swept volume
R_c	pressure ratio	v_1	refrigerant specific volume
$\dot{Q}_{ev}$	cooling capacity	ε	coefficient of performance
$\dot{m}$	mass flow rate	$\dot{P}_{el}$	electric power consumption
h_i	enthalpy in i point of the refrigeration cycle	ψ_{is}	compressor efficiency
$\eta_{v,1}$	compressor volumetric efficiency referred to compressor inlet conditions	ε_{is}	ideal cycle performance
		OCR	oil circulation rate

(R290), are long time known as excellent refrigerants and their reintroduction in refrigeration systems is being supported by different research groups, companies and institutions.

From the theoretical side, propane is a pure HC which offers excellent thermodynamic properties, good compatibility with most materials already present in refrigeration plants and null ODP and GWP, see Ref. [1] for a more systematic account of the advantages of HC's in refrigeration systems, see also [2] for a comparative account of the advantages and disadvantages with respect to the most used refrigerants. There are also many experiences with R22 drop-in systems as well as specifically designed equipment, see for example [1,6–8], which support the suitability of R290 in different types of refrigeration plants.

Regarding the main drawback of hydrocarbons, their flammability, new standards and codes have been recently released ([3,4]) or are in preparation to allow its safe use (see e.g. the documents around the IEC 60335 standard for use of flammable gases in electrically driven refrigeration equipment, [5]). Also special attention should be paid to the one released by the European Committee for Standardization, [3], fixing certain restrictions to the use of HC's as refrigerant (mainly to be used in indirect systems) and setting maximum allowable charge figures for given applications.

According to statistics of heat pumps tests at the WPZ Töss Switzerland, [9], after R407C, propane is the second most used refrigerant in heat pumps, being carried by about 12% of the total number of heat pumps tested in 2002 (31% of the air–water systems). It should be though noted that the share of propane was even higher in 1999 (14%), when R22 was the mostly used refrigerant. This means that the substitution of R22 has not given advantage to R290, but to R407C which, despite its zeotropic character and its poorer performance compared to R22 and R290, is being widely used as de-facto replacement for R22 in heat pumps and air conditioning systems, because of its adequate pressures and temperatures.

One of the most serious obstacles to a more widespread use of R290 in refrigeration systems lies in the main compressor manufacturers being unwilling to give support to these applications. Beyond the safety issue, one of the reason for this may lie in the lack of information on how compressors behave with propane in a wide range of conditions.

The use of HFC's has also brought changes in the types of compressor lubricants used because of the different solubility patterns that these refrigerants show when mixed with mineral oils. For this reason new oils like POE have been developed to allow the refrigeration system to work in a more efficient way with HFC's. There is also a considerable lack of information related to the interaction of these new oils with other refrigerants like R290.

The aim of the present work was to develop a comparative study between R290 and R407C for five different reciprocating compressors working with POE oil. The compressors and test conditions were selected to cover a wide range in capacities as well as working conditions. With this philosophy three compressors with different number of pistons (one, two and four) and a stroke of 38.1 mm, and two compressors, with one and two pistons, and with a stroke of 30.23 mm were tested. In the analysis we have tried to separate those changes that are based on the refrigerant thermodynamical properties and those related to the way the compressor works with the different refrigerants. In addition, oil circulation rate measurements were made to evaluate possible anomalies or differences in behavior of these refrigerants with POE oil.

2. Test setup

Compressors selected to be tested are shown in Table 1. The standard POE oil included by the manufacturer in the compressors was used in all tests. In all propane tests almost pure (99.95%) propane was used to ensure traceability of our results.

Table 1
Tested compressor

Stroke (mm)	Number of pistons	Dead space ratio	Compressor name
30.23	1	0.037	SO
38.10	1	0.029	LO
30.23	2	0.037	ST
38.10	2	0.029	LT
38.10	4	0.029	LF

The compressor rating procedure was performed according to the relevant standards in the field such as the ISO-917, [10], and American ANSI ASHRAE 23-1993, [11], as shown in the scheme in Fig. 1. According to the aforementioned standards, the circulating refrigerant mass flow is the determining parameter to be measured and primary and confirming measurements are to be made. The primary test procedure chosen is the secondary refrigerant calorimeter method. A Coriolis-type mass-flow meter was used as the confirming test method. In all cases, confirming tests were carried out simultaneously with the primary mass-flow rate determination.

Several PID control loops (compressor inlet and outlet pressure, superheat and subcooling controls) were incorporated to allow a precise adjustment of the refrigerant conditions at compressor inlet (evaporating temperature and superheat) and outlet (condensing temperature) with a precision of 1 kPa. The rig is fully automated, and designed to make possible to reach any allowable test conditions without manual adjustments.

The mass flow rate directly measured by means of a Coriolis-type (Fisher–Rosemount Micro-Motion CMF025M) was compared with the secondary refrigerant calorimeter based result. The instrument accuracies of pressure transmitter (Fisher–Rosemount 3051) and temperature tranmitter (RTD-PT 100) are 0.02% and 0.05 °C, respectively.

Oil circulation rate (OCR) measurements were done following standard ANSI/ASHRAE 41.4-1996, [12]; care was taken that the oil level at the sight glass was stable during the test.

Safety was a major concern during the design of test facility. Specific procedures and standards regarding the handling and use of flammable gases were taken into account. Specific measures included the use of intrinsically safe electric material, specific propane sensors, the use of emergency switches and alarms and appropriate air renewal procedures to ensure non-critical concentrations in case of leakage.

3. Test and evaluation procedure

Comparing the performance of a given compressor with different refrigerants, there are changes caused just by differences in the thermophysical refrigerant properties whereas others could in principle be optimized with a better compressor design. According to the rating procedures described in the above section, compressor capacity is given as:

$$\dot{Q}_{ev} = \dot{m}(h_1 - h_{3*}) \quad (1)$$

where $\dot{m}$ is the mass flow rate and $(h_1 - h_{3*})$ the enthalpy difference between two specific states of the refrigerant. According to the ISO procedure, see Ref. [10], these states are the inlet of the compressor, 1, and the saturated liquid state at compressor outlet pressure, 3*. The last is not a real state of the refrigerant in the compressor test loop. In this sense the rated capacity and COP do not depend on the actual cycle, but only on the conditions upstream and downstream of the compressor. In this work, instead of the ISO standard, this point was fixed at 8.3 K degrees of subcooling, to compare the obtained results with the available catalog data for R407C.

A different way to write the above expression is to

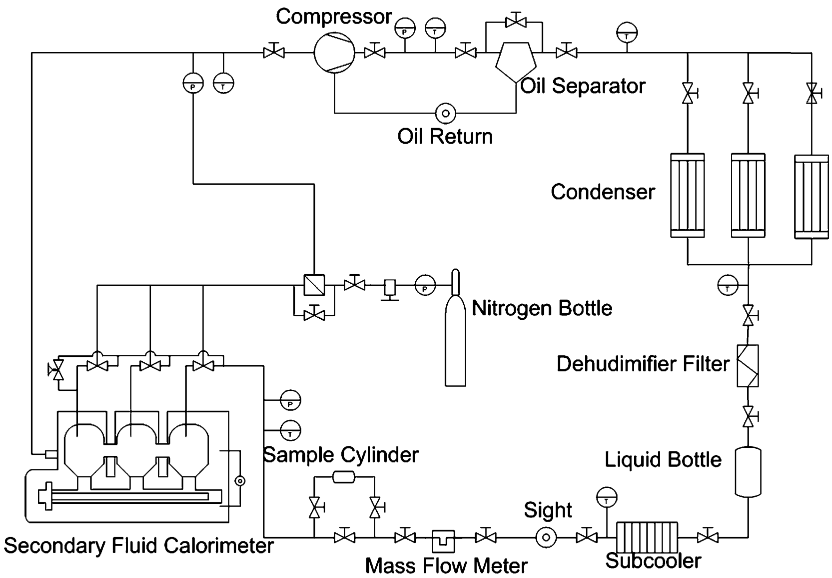

Fig. 1. Schematic of the experimental set up for the characterization of compressors.

consider it from the point of view of the compressor:

$$\dot{Q}_{ev} = \eta_{v,1}\psi_{v,1}\dot{V}_{sw} \tag{2}$$

being $\dot{V}_{sw}$ the compressor swept volume, $\psi_{v,1}=(h_1-h_{3*})/v_1$ the volumetric capacity (or refrigerating effect) of the given fluid for the given cycle and $\eta_{v,1}$ the compressor volumetric efficiency referred to compressor inlet conditions. v_1 is the refrigerant specific volume at compressor inlet for given evaporation temperature and superheat.

A coefficient of performance (COP), ε, is defined as:

$$\varepsilon = \frac{\dot{Q}_{ev}}{\dot{P}_{el}} = \psi_{is}\varepsilon_{is} \tag{3}$$

with $\dot{P}_{el}$ the electric power consumption of the compressor. The coefficient of performance of the ideal cycle with isentropic compression $\varepsilon_{is}=(h_1-h_{3*})/(h_2-h_1)$ is again a refrigerant property, whereas the compressor efficiency, $\psi_{v,1}=\dot{m}(h_{2s}-h_1)/\dot{P}_{el}$, combines compressor related and refrigerant features.

From these considerations it follows that propane possesses a higher capacity per mass unit, but, due to its much lower density, a lower capacity per volume flow unit (specific volumetric capacity). On the other hand discharge temperatures are also expected to be substantially lower on a compressor working with propane.

The ideal results indicate that, at equal volume flow, when switching to propane, a capacity decrease of about 9% for high evaporation temperatures, and a capacity increase around 3% for low evaporation temperatures should be expected. For the same range of conditions, the coefficient of performance of the machine would be expected 6% higher with propane.

All this facts can be seen in Figs. 2 and 3 that exemplifies the ideal cycle performance data of R290 and R407C, for 8.3 K of subcooling and 11.1 K of superheat.

The matrix used for the compressors characterization was chosen taking into account the conditions in which

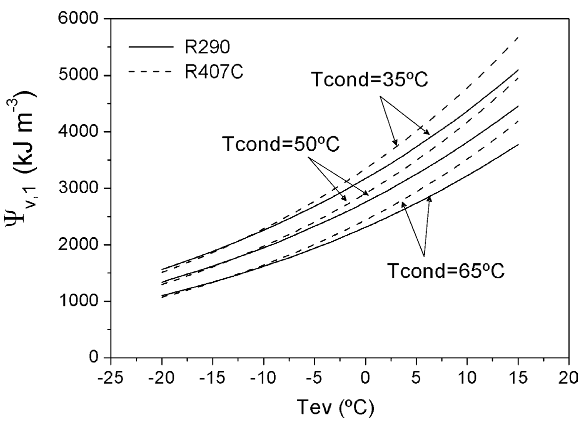

Fig. 2. Specific volumetric capacity of the ideal cycle with isentropic compression based on the thermophysical properties of both refrigerants.

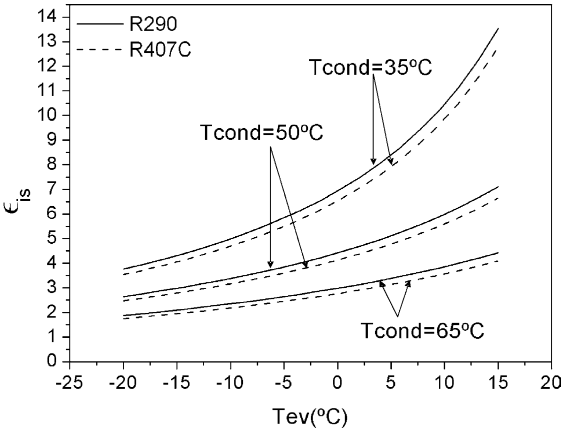

Fig. 3. Coefficient of performance of the ideal cycle with isentropic compression based on the thermophysical properties of both refrigerants.

these compressors would work (relatively high condensation temperatures and evaporation temperatures) in order to provide service-representative results. It covered a set of condensation temperatures ranging from 35 to 65 °C and −20 to 15 °C evaporation temperature. A standard superheat of 11.1 K was chosen. The detailed used test matrix can be seen in Fig. 4. The unfilled circles indicate the points in which OCR measurements were performed. For SO compressor additional measurements of OCR at 35 and 50 °C were done to analize some possible dependences on working conditions.

After a careful characterization of the calorimeter losses, test results showed a high consistency between the result from the primary and confirmation test method as specified in standards. The mean discrepancy between both methods was less than 1% in most cases.

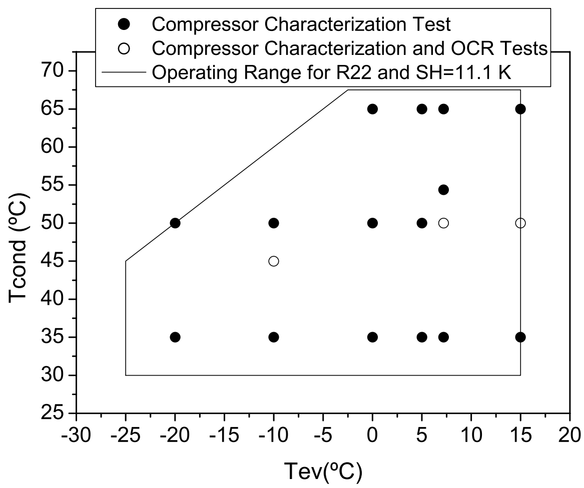

Fig. 4. Test matrix.

4. Results and discussion

Measured values of cooling capacity and COP, corresponding to a condensation temperature of 50 °C, are shown in Figs. 5–7. Table 2 offers a more detailed comparative analysis at two standard conditions which can be regarded as significant, labelled as MT (−10 °C of evaporation temperature, 45 °C of condensation temperature and 11.1 K of superheat) and ARI (7.2 °C of evaporation temperature, 54.4 °C of condensation temperature and 11.1 K of superheat). In the referred tables as well as the curves, propane based experimental results are compared to R407C performance data obtained from the manufacturer's catalog.

Some additional tests were developed to check that the tested compressors actually behave as expected from the catalogue when they are working with R407C. The results show a reasonable consistency between catalogue and experimental data. The dew point definition was used to evaluate the properties in R407C.

Although in general terms the actual performance curves respect the tendencies that may be expected from the theoretical refrigerant curves, Figs. 2 and 3, there are quantitative differences arising from differences in the volumetric and compressor efficiency figures.

Measured cooling capacities, Fig. 5, show that the evaporation temperature at which R407C capacity equals R290 capacity is higher than expected from the volumetric specific capacity figures of Fig. 2. This can be understood from volumetric efficiency data that for the same temperature working conditions is higher for propane.

In Figs. 8 and 9 propane shows higher volumetric efficiency than R407C for low pressure ratios, this tendency is inverted as the pressure ratio goes up. As the number of cylinder grows, for the long stroke version, see Fig. 8, propane maintains a better volumetric efficiency at comparatively higher pressure ratios. In general terms volumetric efficiency depends more on compressor stroke than on compressor size (number of cylinders). This is expected from the ideal compressor behaviour due to the fact that compressors with larger stroke possess a larger dead space ratio, although some other factors like leakages must be taken into account.

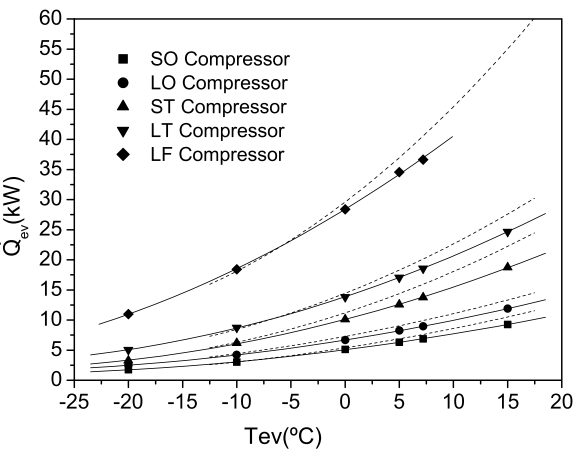

Fig. 5. Cooling capacity vs. evaporation temperature for 50 °C of condensation temperature, the straight lines are for R290 while the dash are for R407C.

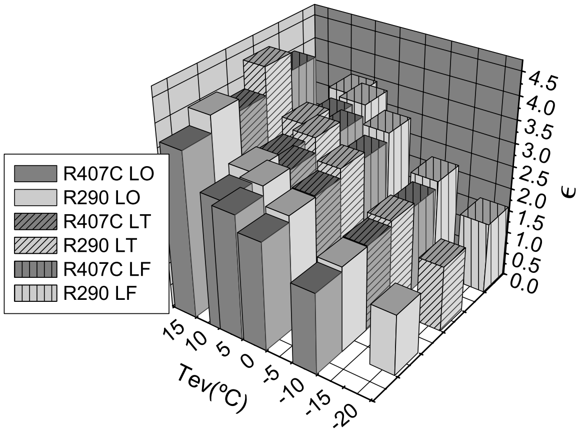

Fig. 6. COP for long stroke compressors at 50 °C of condensation temperature.

It is important to clarify that the plotted lines in Figs. 8 and 9, and the next ones Figs. 10 and 11 only intend to represent the general trend of data for both refrigerants. The represented functions show other test condition dependencies like inlet temperature although the pressure ratio is the most important one.

Relating to COP, a mean relative improvement of about 9% was achieved using propane instead of R407C. This is an important and significant result, which also relies partly on theoretical refrigerant behavior and is quite sensitive to operation conditions: the improvement is larger at high evaporation temperatures, but falls down to less than 6% at

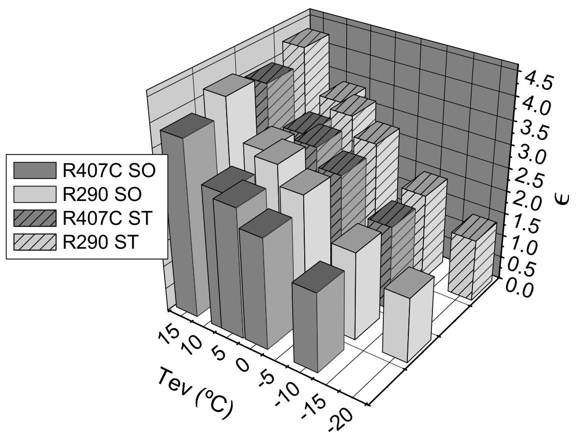

Fig. 7. COP for short stroke compressors at 50 °C of condensation temperature.

Table 2
Experimental compressor comparative data for MT and ARI points

Compressor		MT conditions				ARI conditions			
		Cooling (kW)	Vol. eff.	COP	Comp. eff.	Cooling (kW)	Vol. eff.	COP	Comp. eff.
SO	R407C	3.41	0.60	1.99	0.53	6.94	0.70	2.60	0.55
	R290	3.34	0.63	2.22	0.58	6.37	0.73	2.84	0.59
	%	−2.1	4.7	11.0	9.0	−8.6	3.5	8.8	6.5
LO	R407C	4.86	0.68	2.03	0.54	9.13	0.74	2.53	0.54
	R290	4.68	0.69	2.15	0.56	8.36	0.77	2.83	0.59
	%	−3.8	1.5	5.7	6.6	−8.8	4.5	11.2	9.0
ST	R407C	7.06	0.62	2.10	0.56	14.59	0.74	2.78	0.59
	R290	6.90	0.64	2.23	0.58	12.82	0.74	3.03	0.63
	%	−2.2	2.8	6.0	3.9	−12.9	−0.13	8.6	3.4
LT	R407C	9.38	0.66	2.16	0.57	18.52	0.75	2.71	0.58
	R290	9.60	0.71	2.30	0.60	17.25	0.79	3.02	0.63
	%	2.3	7.9	6.3	4.6	−7.14	5.6	10.8	8.9
LF	R407C	19.93	0.70	2.21	0.59	37.39	0.75	2.82	0.60
	R290	20.10	0.74	2.40	0.63	34.90	0.79	3.08	0.64
	%	0.8	6.0	8.2	7.6	6.9	4.8	8.8	6.6

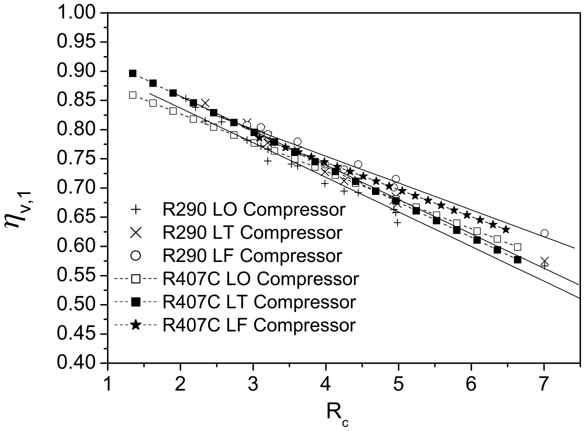

Fig. 8. Volumetric efficiency vs. pressure ratio for compressors with the longer stroke. The plotted curves show a general trend of the experimental data.

low evaporation temperatures which may be expected from the theoretical isentropic COP data.

In fact compressor efficiency, Figs. 10 and 11, shows a complex behavior, being higher for propane at low and medium pressure ratios and lower at high pressure ratios.

Considering the compressor design, efficiency increases with the number of cylinders for the same cylinder volume (Figs. 10 and 11) and decreases when increasing the stroke while keeping the cylinder diameter constant.

A further aspect which was carefully considered were possible changes in the electric motor efficiency. Looking for instance at the operating conditions included in Table 2, the electric efficiencies at those points, showing a rather flat dependency on load, do not differ in more than 1% at most conditions when comparing R290 with R407C conditions. This leads us to conclude that the improvement in overall

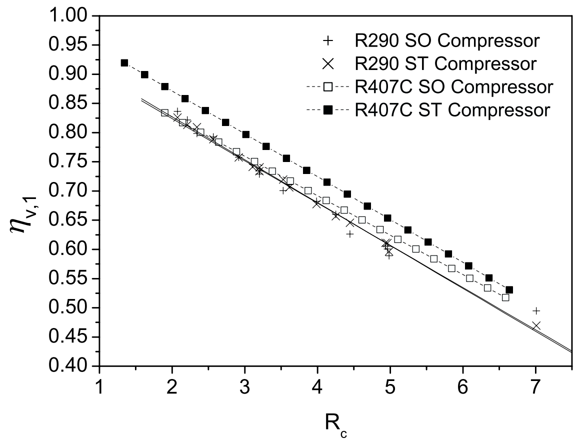

Fig. 9. Volumetric efficiency vs. pressure ratio for compressors with the shorter stroke. The plotted curves show a general trend of the experimental data.

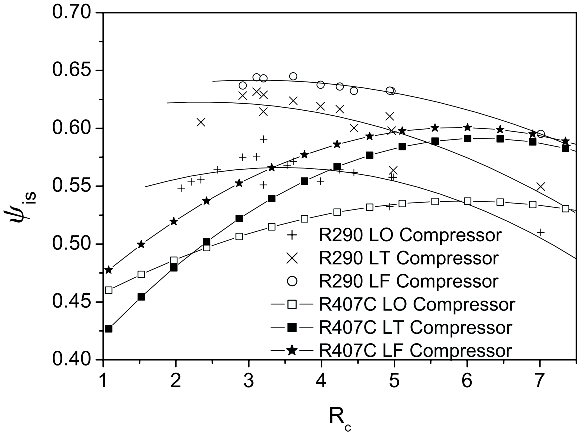

Fig. 10. Compressor efficiency vs. pressure ratio for compressors with the larger stroke. The plotted curves show a general trend of the experimental data.

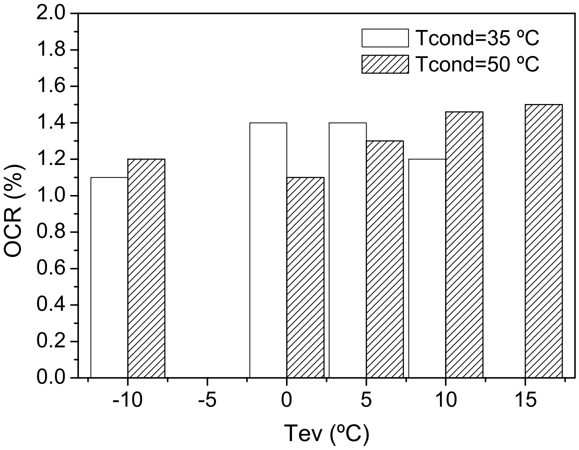

Fig. 12. OCR measurements for SO compressor at different working conditions.

efficiency of the electric motor but on an improvement of the mechanical and thermal characteristics of the compression.

In this sense, the observed differences in compressor efficiency for both refrigerants could be related to the lower temperatures in the discharge for propane than for R407C (between 10 and 15 °C) limiting the amount of irreversible heat transfer between the relatively hot and the relatively cold parts of the circuits inside the compressor. Furthermore, as known from many experimental studies, see for instance [13], propane shows considerable reduced pressure losses at equivalent flow velocities. This leads to reduced losses especially at sections with relatively high velocities (mainly at the inlet and exhaust valves).

Another interesting factor considered in our investigation was oil circulation rate. The obtained results are shown in Fig. 12 and Table 3.

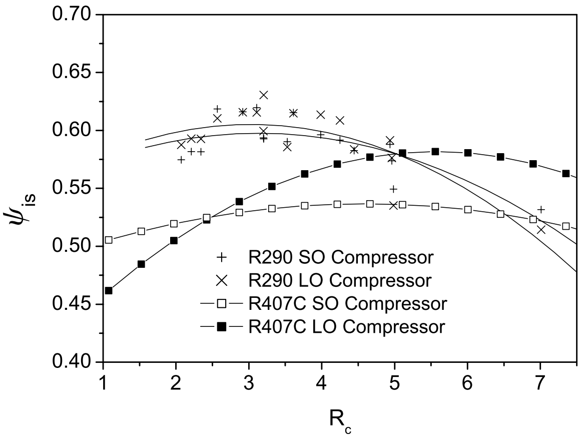

Fig. 11. Compressor efficiency vs. pressure ratio for compressors with o for compressors with the shorter stroke. The plotted curves show a general trend of the experimental data.

From the SO compressor wider OCR test serie, it was observed that the proportion of oil leaving the compressor is quite independent on operating conditions—at least within the tested range (between −10 and 15 °C evaporation temperature for a constant superheat of 11.1 K), Fig. 12. For this reason only three OCR measurement were taken on each of the tested compressors at different operating point, and their associated mean results are shown in Table 3. The resulting quantities of oil do not seem large enough to cause a significant damage or effect on the normal operation of the system working with this R290/POE oil combination. Furthermore they are quite comparable to OCR figures for the R407C/POE oil combination, see for instance [14].

On the other hand there is some variability in OCR values for different compressors.

Table 3
OCR measured values for the different compressors

Compressor		OCR meas	Mean value
SO	(−10, 45)	1.1	1.4
	(7.2, 50)	1.5	
	(15, 50)	1.5	
LO	(−10, 45)	1.5	1.4
	(7.2, 50)	1.7	
	(15, 50)	1.0	
ST	(−10, 45)	0.4	0.35
	(7.2, 50)	0.4	
	(15, 50)	0.3	
LT	(−10, 45)	0.6	0.55
	(7.2, 50)	0.5	
	(15, 50)	0.5	
LF	(−10, 45)	0.6	0.7
	(7.2, 50)	0.8	
	(15, 50)		

5. Conclusions

A large and systematic test campaign including five different compressors and a representative span of operating conditions was conducted to derive general conclusions with respect to the suitability of propane (R290) as refrigerant in combination with modern POE oil. Performance data were systematically compared with those of R407C derived from catalogue data. These measurements and the following analysis lead us to the following general conclusions:

- R407C shows a worse thermodynamical behavior than propane, theoretically as well as in the real compressor. A mean COP improvement of 9% resulted when using propane instead of R407C.
- Regarding the cooling capacity, results expected from the refrigerant properties were confirmed although slight changes were recorded due to differences in the volumetric efficiencies.
- On the basis of the observed behaviour, two main factors can be thought to have an influence: the reduced irreversible heat flow due to the lower discharge temperatures of propane and the reduced pump losses in the inlet and discharge valves.
- Related with the POE oil used to perform these tests, there is no significant difference between the oil behavior with propane and with R407C, at least in standard working conditions. The influence of oil in the refrigeration system should be negligible if the installation is designed correctly to avoid oil traps [15], letting the oil to be drawn by the refrigerant.

During these tests the viability and the efficiency of a refrigeration system working with propane and POE oil has been proven. To look further, additional tests should be performed to check if long term durability problems could arise as a consequence of differences in solubility and viscosity of the oil-refrigerant mixture [16].

Furthermore, a deeper understanding of the rather complex confluence of effects on the basis of the differences observed deserves the employment of the computer codes that are being developed and adjusted in parallel with the described tests. Test results and models, altogether, will show the way to the final aim of this research: a better understanding of what is happening inside a refrigeration compressor (with or without propane).

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References

[1] A new high efficiency air–water reversible heat pump working with propane for commercial applications in south Europe. Final publication report contract JOE3-CT97-0077 program NNEJoule C. CORDIS europena comision.
[2] K. Spindler, The influence of the thermophysical properties of the refrigerants on the performance of heat pumps, Proceedings of the 72nd Eurotherm Seminar, Valencia, Spain, 2003, p. 285–290.
[3] EN 378-1, Refrigerating systems and heat pumps—safety and environmental requirements—parts 1–13. European Committee for Standardization, CEN TC; 2000, p. 113–182.
[4] ACRIB, Guidelines for the use of hydrocarbon refrigerants in static refrigeration and air conditioning systems. Air conditioning and refrigeration industry board. ISBN 1 872719 13 9; 2001.
[5] Modificación de la norma IEC 60335-2-40 sobre requerimientos particulares para el uso de gases inflammables en bombas de calor eléctricas, sistemas acondicionadores de aire y de-humidificadores, translation to spanish of the 61D/125/CDV committee draft for vote to part 2–40 of standard IEC 60335 refering to electrical driven refrigeration equipment using flammable gases as refrigerant.
[6] R.W. James, J.F. Missenden, The use of propane in domestic refrigerators, Rev Int Froid 15 (5) (1992) 95–100.
[7] B. Purkayastha, P.K. Basal, An experimental study on HC290 and a commercial liquefield petroleum gas (LGP) mix as suitable replacements for HCFC22, Int J Refrigeration 21 (1) (1998) 3–17.
[8] J.M. Corberán, J.F. Urchueguía, I. Navarro, J. Gonzálves, A. Calas. Performance of a reciprocating hermetic refrigerant compressor using propane as working fluid. Proceeding of the fourth IIR-Gustav Lorentzen conference on natural working fluids at Purdue. IIF-IIR Commission B1, B2, E1, E2; 2000, p. 225–232.
[9] Wärmepumpentest—und Ausbildungszentrum (WPZ) Töss, Zusammenstellung von Leistunsdaten von Wärmenpumpen, Töss, Schweiz; 2002.
[10] ISO 917:1974. Standard testing of refrigerant compressors; 1974.
[11] ANSI/ASHRAE 23-1993 standard.
[12] ANSI/ASHRAE 41.4-1996 standard.
[13] O. Pelletier, Propane as refrigerant in residential heat pumps, Engineering Licenciate Thesis, KTH, ISBN. 1102-0245; 1998.
[14] E. Navarro, J. Urchueguia, J. Gonzalvez, J.M. Corberan, Experimental investigation on performance and oil circulation rate in reciprocating compressors of different sizes using propane as refrigerant, Proceedings of fourth European thermal science conference, Birminghan, England; 2004.
[15] S. Cem Kesim, K. Albayrak, A. Ileri, Oil entrainment in vertical refrigerant piping, Int J Refrigeration 23 (2000) 626–631.
[16] P. Fernando, H. Han, B. Palm, E. Granryd, P. Lundqvist, The solubility of propane (R290) with commonly used compressor lubrication oils, Proceedings of the international conference on compressors and their systems, London, England, 2003.