COMPARATIVE STUDY OF THE PERFORMANCE OF A VAPOR-INJECTION SCROLL COMPRESSOR AND A TWO-STAGE RECIPROCATING COMPRESSOR OPERATING UNDER EXTREME CONDITIONS

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ABSTRACT

The current paper presents a comparative study between a vapor-injection scroll compressor (SCVI) and a two-stage reciprocating compressor (TSRC) operating under extreme conditions. The present work is divided into two parts: on the first one, both compressors are compared in terms of compressor efficiency, volumetric efficiency, COP, and cooling capacity using R407C refrigerant; on the second part, the seasonal performances of both compressors working in cooling and heating modes are estimated and analyzed. As a result, the SCVI presents better efficiency and COP, compared to TSRC, for pressure ratios below 7.5. This compressor can be used in air conditioning systems and heat pumps, which work under moderate temperature conditions. For higher pressure ratios, the TSRC has better efficiency which subsequently gives higher system COP. This type of compressors can be used in domestic hot water systems operating under severe climates and low temperature freezing systems (under -20°C).

Keywords: scroll compressor, two-stage compression, reciprocating compressor, vapor injection, extreme conditions.

Nomenclature

h	enthalpy (kJ kg^{-1})	Greek symbols	
H	number of bin hours	ρ	density (kg m^{-3})
$\dot{m}$	mass flow rate (g s^{-1})	η_c	compressor efficiency
P	pressure (bar)	η_v	volumetric efficiency
P_a	compressor consumption (kW)	Subscript	
P_r	pressure ratio	c	condensing
Q_c	cooling capacity (kW)	cond	condenser
Q_h	heating capacity (kW)	e	evaporating
Q_{econo}	economizer capacity (kW)	evap	evaporator
SCOP	seasonal coefficient of performance	inj	injection
SCVI	scroll compressor with vapor injection	int	intermediate
T	temperature (ºC)	s	isentropic
TSRC	two-stage reciprocating compressor		
$\dot{V}$	swept volume (m^3 h^{-1})		

1. INTRODUCTION

Refrigeration and heat pumps units working with single-stage vapor compression system reduce significantly its efficiency when they work with large difference between evaporating and condensing temperatures, these systems have several limitations such as:

- High compressor discharge temperature. The high temperature can produce thermal instability in the lubricating oil.
- Cooling / heating capacity loss.
- Low COPs obtained. Carnot and compressor efficiencies decreases dramatically at high temperature lifts. This behavior makes heat pump systems on a great disadvantage with conventional heating boiler.
- Large compressor displacement needed. The volumetric efficiency decrease rapidly with high pressure ratios. This means that to obtain a given capacity, the compressor displacement has to increase with the subsequent impact on compressor cost.

In order to overcome these limitations, the most widely used solution is the two-stage compression with vapor injection. This technique refers to injecting vapor refrigerant to the intermediate location of the compressor. Several advantages were found for vapor injection technique, the most important are the following:

- Capacity improvement in severe climates (heating at lower than 0°C and cooling of higher than 35°C of ambient temperature).
- The system capacity can be varied by controlling the injected refrigerant mass flow rate, which permits some energy savings by avoiding intermittent operation of the compressor.
- The compressor discharge temperature of a vapor injection cycle is lower than that of a conventional single-stage cycle (Xu, et al., 2011).

Vapor-injection Scroll compressor (SCVI) is one of the most used compressors in heat pump systems with vapor injection technique and there are several recent researches about this technology. Ma, et al., (2003) developed an experimental investigation of air-source heat pump for cold regions using a SCVI with economizer. The prototype was able to work smoothly under ambient temperatures as low as -15°C, the heating capacity and COP were improved and the discharge temperature was steady and remained below 130°C in these temperature conditions; similar studies were developed by Ding, et al., (2004) and Ma and Chai, (2004).

Bertsch and Groll, (2008) simulated, designed and constructed an air-source two-stage heat pump using a SCVI working with R410A as the refrigerant. They tested the heat pump and verified that it is able to operate at ambient temperatures as low as -30°C to 10ºC and supply water temperatures of up to 50°C in heating mode. At the same ambient temperature, two-stage mode operation approximately doubled the heating capacity compared to single-stage mode operation. The discharge temperatures of the compressors in two-stage mode stayed below 105°C. Ma and

Zhao, (2008) compared the heating performance of a heat pump with a flash tank coupled to a SCVI and a system with an economizer cycle using R-22. The heating capacity and COP of the flash tank cycle were higher by 10.5% and 4.3%, respectively, relative to those of the economizer cycle at air temperatures of 45°C in the condenser and -25°C in the evaporator. Wang, et al., (2009a) suggested a model to optimize the refrigeration system using a SCVI with proposition of universal control and design methods; with this model, they investigated the effects of gas injection on the system and components parameters. Wang, et al., (2009b) studied the performance of a heat pump system using a SCVI with the cycle options of both flash tank and internal heat exchanger (economizer) configurations. They found that cooling capacity gain of around 14% with 4% COP improvement at the ambient temperature of 46.1°C in cooling mode, and about 30% heating capacity improvement with 20% COP gain at the ambient temperature of -17.8°C in heating mode compared to the conventional heat pump system, using R410 refrigerant. Feng, et al., (2009) studied a heat pump water heater using a vapor injection system with a mixture of R22/R600a. They found that, compared with using R22 as the refrigerant, the heating capacity and COP of the system with the mixing refrigerant of R22/R600a were higher in 30% and 32% respectively, when the vapor injection was used. In addition, the compressor discharge temperature could be controlled below 100°C. Navarro, et al., (2013) analyzed a SCVI working on wide range of operating conditions; the study included a compressor characterization, a comparison of performance with a single-stage compressor of similar volume, and the determination of the influence of the intermediated conditions on the compressor performance. The main conclusions were the following: heating capacity and COP of the SCVI increases up to 20% and 10% respectively. The discharge temperature of SCVI were always lower, especially in extreme conditions (temperature decreases 10K).

On the other hand, the two-stage reciprocating compressor (TSRC) is another kind of compressor mainly used in systems with two-stage compression with vapor injection. Several previous works were found in the literature about this compressor, most of them in transcritical CO_2 heat pump applications because these systems work with higher compression ratios. Cavallini, et al., (2005) proposed a computer code for the simulation of a two-stage transcritical carbon dioxide cycle. The system used a TSRC with economizer for the injected refrigerant. They found that the COP increased up to 25% for typical air conditioning applications compared with experimental results of a two-stage transcritical carbon dioxide cycle using two reciprocating compressors in series with intercooling. Rigola, et al., (2006) presented a numerical study of a TSRC working with CO_2 as refrigerant in order to show the availability of this one for small freezing applications with similar or even better COP than R134a conventional sub-critical cycles. The TSRC presented a reduction on power consumption due to decrease on pressure ratios of both compression chambers in comparison with one-stage compressor, an important compressor outlet gas temperature decrease and finally a COP improvement from 20% to 15% at -35°C and -10°C of evaporating temperatures, respectively. Cecchinato, et al., (2009) analyzed and optimized different transcritical CO_2 cycles. As a result, for extreme operating conditions (the lowest evaporating temperature and the highest external temperature) the double-throttling, two-stage compression with TSRC and economizer presents the greatest improvement, give rise to 70% increase in energy efficiency against the simple solution (single-throttling and single-compression cycle). In addition, they concluded that two-stage compression is mandatory for limiting the temperature at the compressor discharge in extreme temperature conditions. At the same line Agrawal, et al., (2007) investigated the performance of two-stage CO_2 heat pumps with flash gas bypass, flash intercooling and compressor intercooling using numerical analysis and simulation. It was observed that the flash gas bypass system yields the best performance among the three two-stage cycles analyzed. Agrawal and Bhattacharyya, (2007) performed a simulation studies on a two-stage flash intercooling transcritical carbon dioxide heat pump cycle. They conclude that the flash intercooling technique is not economical with CO_2 refrigerant unlike NH_3 as the refrigerant. The COP was considerably lower than that of the single cycle for a given gas cooler and evaporator temperature. Cho, et al., (2009) studied the performance and operating characteristics of a two-stage CO_2 cycle with gas injection. It was found the cooling COP of the two-stage gas injection cycle was maximally enhanced by 16.5% over that the two-stage non-injection cycle in the experiments and the discharge temperature of the second-stage of compression decreased by 5°C to 7°C. Not much information on compressors that work with traditional refrigerants was found.

However, up to the knowledge of the authors there is no comprehensive public systematic comparison between SCVI and TSRC in the whole range in which they are able to work. For that reason, we developed a comparative study between them in terms of COP, compressor efficiency, volumetric efficiency and cooling capacity. Based on these results and the EN14825 normative for the determination of SCOP, the seasonal performance in cooling and heating mode for both compressors have been calculated. This analysis has allow the determination of the application range for each technology.

The comparative study was performed working with high compression ratios and large differences between evaporation and condensation temperatures. We used laboratory-collected data of the SCVI and catalog data of the TSRC, both using R407C refrigerant.

2. EXPERIMENTAL SETUP

Fig. 1 shows the schematic of the test rig used to collect the SCVI data. The system consists of three circuits: the heat pump circuit, the water circuit for the condenser and the air circuit for the climatic chamber.

The heat pump is an air to water refrigeration injection circuit installed in the climatic chamber. The climatic chamber is able to control the air temperature and humidity conditions in a range from -25°C to 50°C. The water circuit is in charge of controlling the condensation pressure.

Evaporating temperature and superheat are controlled by the climatic chamber temperature control and the electronic expansion valve EEV-2 (see Fig. 1).

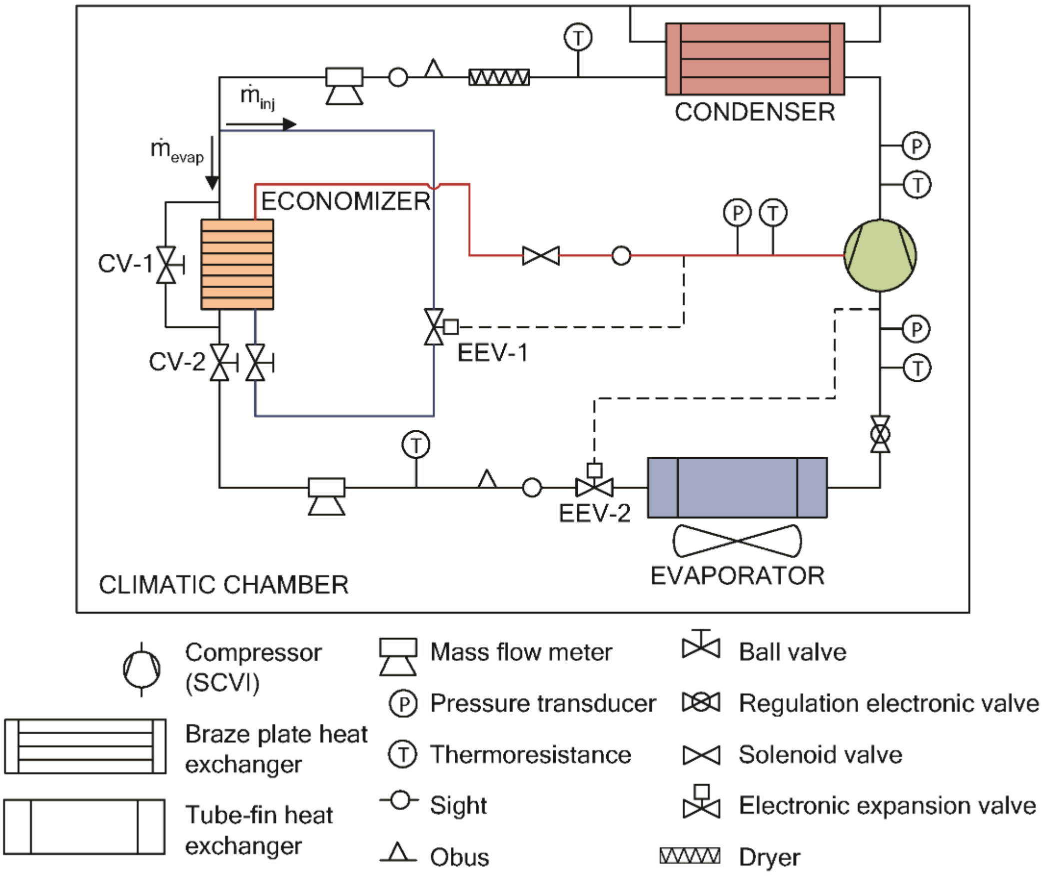

Fig. 1- Test rig schematic.

The heat pump is equipped with a brazed plate heat exchanger (economizer). Fig. 1 shows the economizer heat exchange between the injection refrigerant mass flow ($\dot{m}_{inj}$) and the refrigerant mass flow of the evaporator line ($\dot{m}_{evap}$). The secondary electronic expansion valve (EEV-1 on Fig. 1) mainly controls the intermediate superheat. For a given compressor, once the intermediate superheat is supplied, the intermediate pressure will be defined by the heat transfer between $\dot{m}_{evap}$ line and $\dot{m}_{inj}$ line in the economizer and thus, for a given compressor, a given pressure ratio and an economizer size, the intermediate pressure is approximately fixed.

Regarding the instrumentation, the system is equipped with two coriolis mass flow meters with an accuracy of 0.05%, an electrical power meter with an accuracy of 0.1%, three pressure transducers with an accuracy of 0.2%, and 5 RTD with an accuracy of 0.1 K.

The system is controlled by 4 PID loops, they are able to fix the condensing pressure, evaporating pressure, low pressure superheat and intermediate pressure superheat actuating over water condenser flow rate, climatic chamber temperature, valve EEV-2, valve EEV-1 and economizer size, respectively.

The system is also able to work with the traditional single stage cycle by closing a solenoid valve placed in the injection line.

3. EXPERIMENTAL CAMPAIGN

The SCVI was tested working with R407C as a refrigerant. The laboratory tests were performed using the following parameters: low-pressure superheat of 5K, intermediate superheat of 5K and 0K of subcooling.

Table 1 shows the test matrix defined for the SCVI; it was designed in order to have a representative range of evaporating and condensing temperatures and to evaluate the performance working on cooling and heating conditions. In this series of tests, the intermediate pressure was fixed by the intermediate superheat and by the economizer size. The used economizer was the indicated by the manufacturer for this compressor size; it corresponds to the heat exchanger illustrated on Fig 1. Additionally, Table 1 indicates the used catalog data of the TSRC.

Tc (°C)	**Te (°C)**						
	-35	**-25**	**-20**	**-15**	**-10**	**-5**	**0**
30	b	a	a, b	a	a, b	a	a, b
50	b	a	a, b	a	a, b	a	a, b
60	b		a, b	a	a, b	a	a, b
70	b		b	a	a	a	a, b

a = SCVI (Data collected in the laboratory)
b = TSRC (Catalog data)

Table 1. Test matrix.

4. METHODOLOGY

This study was conducted in two parts. On the one hand, a comparative study of the performance of the SCVI and the TSRC has been made. It was performed with the compressors working with high compression ratios and large difference between evaporation and condensation temperatures (see Table 1). On the other hand, a seasonal performance comparison between both compressors working in cooling and heating mode was carried out. These studies are presented in subsections 4.1 and 4.2 respectively.

4.1. Comparative study of the performance of the SCVI and the TSRC.

In order to compare the compressors performance, both the laboratory collected data of the SCVI and catalog data of the TSRC were used. The comparison was made in terms of compressor efficiency, volumetric efficiency, cooling capacity and COP. The swept volume of the SCVI was 17.1 m^3/h and for the TSRC were 20 m^3/h for the low stage and 12.6 m^3/h for the high stage of compression. The thermophysical properties of refrigerant were calculated through NIST Database (Lemmon, et al., 2010).

4.1.1. Parameters estimation of the SCVI.

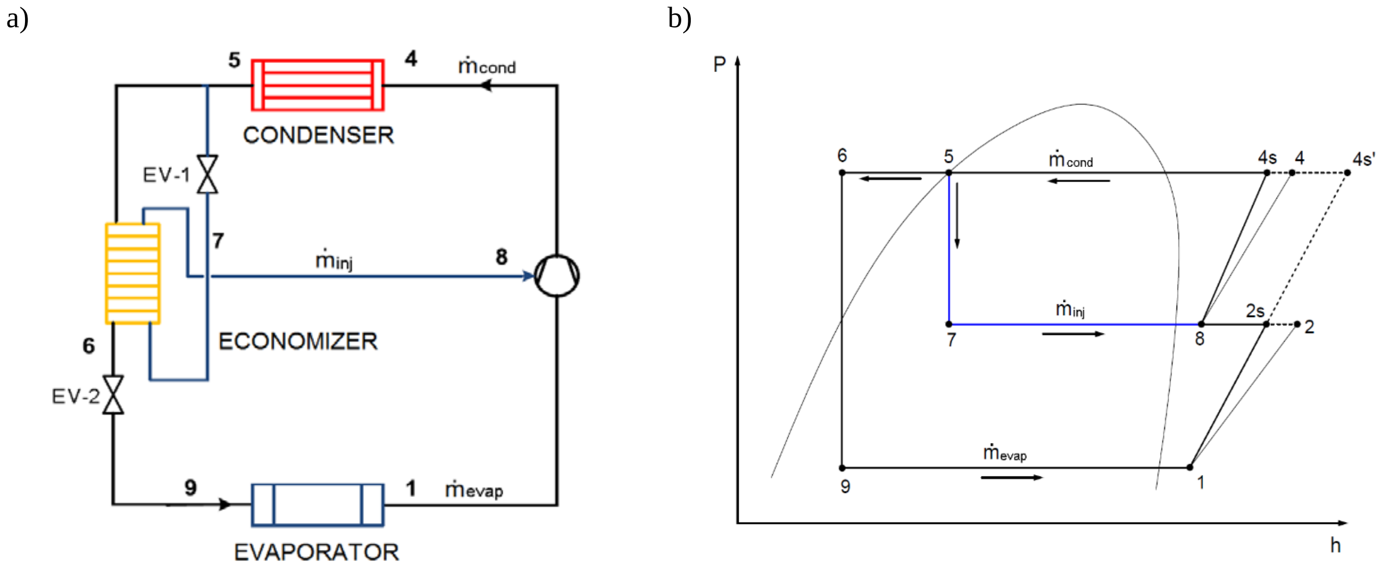

Fig. 2- Vapor injection cycle with economizer of the SCVI. a) Schematic of the cycle. b) p-h diagram.

The cooling capacity, volumetric efficiency and COP of the SCVI were defined by the following equations:

$$Q_c = \dot{m}_{evap}\,(h_1 - h_9)\,, \tag{1}$$

$$\eta_v = \frac{\dot{m}_{evap}}{\rho_1\,\dot{V}_1}\,, \tag{2}$$

$$\mathrm{COP}=\frac{Q_c}{P_a},\tag{3}$$

where $\dot{m}_{evap}$ represents the evaporator mass flow rate of the refrigerant, the number 1 is located at the compressor inlet and the number 9 is located at the evaporator inlet, as can be seen in the schematic of the cycle on Fig. 2(a); P_a represents the compressor consumption.

The SCVI efficiency was defined by the equation (4). This arbitrary expression represent the ideal power consumption of compressor when the refrigerant coming from suction port and injection ports is compressed to the discharge pressure in isentropic process divided by real indicated compressor work. According to Wang, et al., (2009) and Navarro, et al., (2013), based on experimental results, the equation (4) describes suitably the efficiency parameter.

$$\eta_c=\frac{\dot{m}_{evap}\,(h_{4s'}-h_1)+\dot{m}_{inj}\,(h_{4s}-h_8)}{P_a},\tag{4}$$

where $h_{4s'}$ represents the enthalpy at the compressor discharge pressure considering an isentropic compression from the compressor inlet pressure, and h_{4s} represents the enthalpy at the compressor discharge pressure considering an isentropic compression from the intermediate pressure of injection (see the p-h diagram of the Fig. 2(b)). However, for comparison purposes, the equation (5) was used as a reference expression to calculate the compressor efficiency of both compressors.

4.1.2. Parameters estimation of the TSRC.

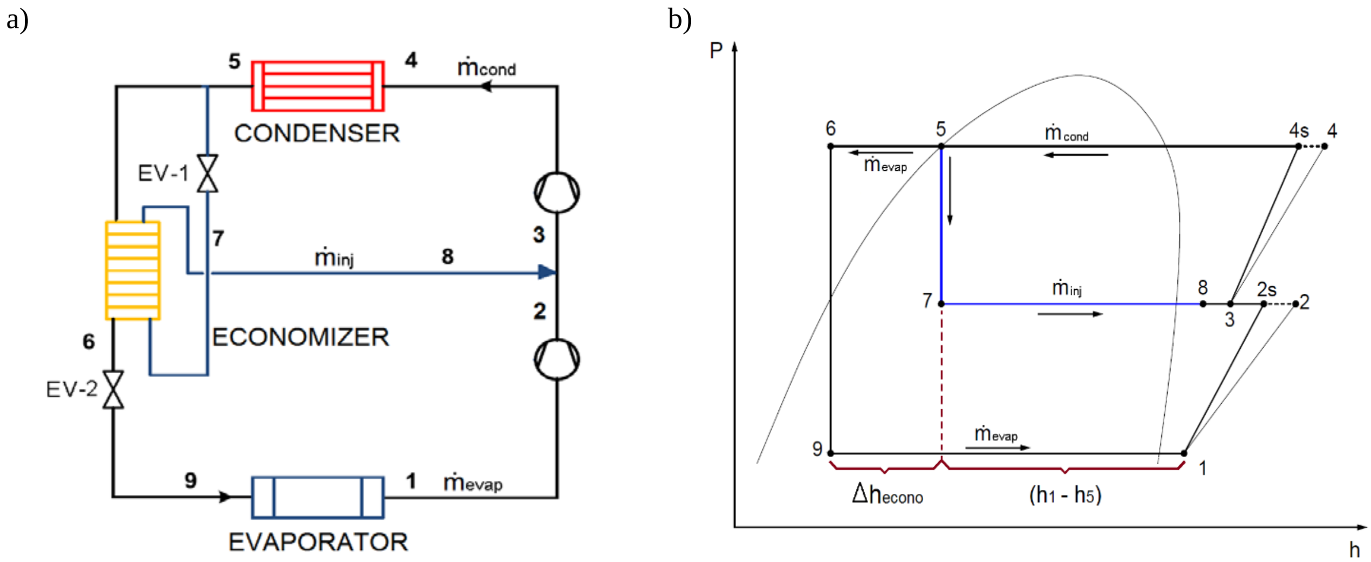

Fig. 3- Vapor injection cycle with economizer of the TSRC. a) Schematic of the cycle. b) p-h diagram.

From the catalog data, the first stage volumetric efficiency and COP of the TSRC were estimated with the equations 2 and 3 respectively.

The compressor efficiency of the TSRC is defined as:

$$\eta_c = \frac{\dot{m}_{evap}\,(h_{2s} - h_1) + (\dot{m}_{inj} + \dot{m}_{evap})\,(h_{4s} - h_3)}{P_a}, \tag{5}$$

where h_{2s} is the enthalpy at the intermediate pressure, considering an isentropic compression from the compressor inlet pressure. h_{4s} represents the enthalpy at the compressor discharge pressure, considering an isentropic compression from the intermediate pressure. h_3 represents the enthalpy at the mixing point of the evaporator mass flow rate with the injection mass flow rate (see the schematic of the cycle and p-h diagram of the Fig. 3).

The catalog data about the intermediate working conditions of the TSRC was limited; therefore, several considerations were taken into account in order to calculate the intermediate pressure and the injection mass flow rate:

- The intermediate superheat of 5K.
- The volumetric efficiencies of the first and second stages follow the same linear function: $\eta_v = A * P_r + B$ where A and B values are obtained from the Fig. 4. This figure illustrates the first stage volumetric efficiency values as a function of the pressure ratio for each set of evaporating and condensing temperatures.

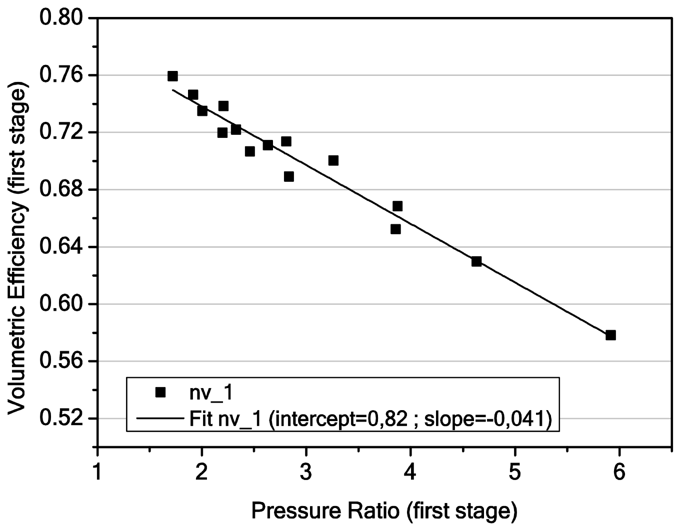

Fig. 4- First-stage volumetric efficiency of the TSRC as a function of pressure ratio.

The refrigerant properties at point 3 (see the p-h diagram of the Fig. 3(b)) were calculated using the gases mixtures equation:

$$h_3 = \frac{\dot{m}_{inj}\,h_8 + \dot{m}_{evap}\,h_{2s}}{\dot{m}_{evap} + \dot{m}_{inj}}, \tag{6}$$

where $\dot{m}_{inj}$ represents the injection mass flow rate of the refrigerant and h_8 represents the enthalpy at economizer outlet.

The intermediate pressure was calculated using an iterative process. It involves the change of the intermediate pressure up to ensure that the refrigerant mass flow injected resulting from the power delivered by the economizer has been equal to the refrigerant mass flow rate injected could support with the second stage volumetric efficiency. It means that the refrigerant mass flows rate calculated with the equations (7) and (8) must be equal:

$$\dot{m}_{inj} = \frac{P_{econo}}{h_8 - h_7}, \tag{7}$$

$$\dot{m}_{inj} = \eta_{v2}\, \rho_3\, \dot{V}_2 - \dot{m}_{evap}, \tag{8}$$

where η_{v2} represents the volumetric efficiency of the second stage of the compression, ρ_3 is the density at mixing point 3 (as can be seen in the schematic of the cycle of Fig. 3(a)), and $\dot{V}_2$ represents the swept volume by the compressor in the second stage.

4.2. Comparative study of the seasonal performance in heating mode.

In real systems, the compressors could work in different temperature conditions from the design temperature points considered in the testing campaign (see test matrix on Table 1). For heating applications such as domestic heating water applications and radiator heating systems, the compressors could work with condensing temperatures of 40°C, 50°C, 60°C even 70°C. In this context, it is necessary to study the performance of the compressors working in these temperature conditions, also considering that the external ambient temperatures vary throughout the year. Therefore, we estimated the seasonal coefficient of performance (SCOP) and the weighted efficiency of the compressors.

The SCOP was estimated using the method of the EN14825 normative. This procedure uses the map of the outdoor bin temperatures and their respective number of bin hours along the year.

For heating applications, the outdoor conditions were divided in three seasons: warmer, average and colder. This allows us to determine the compressor performance working in locations with different external climate conditions. Table 2 shows the internal conditions considered for cooling and heating mode analysis.

Cooling mode	Heating mode
Inside temperatures (evaporator)	Inside temperatures (condenser)
Te (°C)	Tc (°C)
0	40
-10	50
-20	60
-30	70

Table 2. Inside conditions for calculating the SCOP in cooling and heating mode.

The heating SCOP was calculated for each condensing temperature of the inside conditions in Table 2 with the following expression:

$$\text{Heating SCOP}_{Tc} = \frac{\sum_{j=1}^{n} H_j\ Qh_{(T_j)}}{\sum_{j=1}^{n} H_j\ Pa_{(T_j)}}, \tag{9}$$

The condensing temperature corresponding to each bin temperature was estimated as:

$$Te_j = T_j - 15K, \tag{10}$$

The $Qh_{(T_j)}$ and $Pa_{(T_j)}$ were calculated using third order ARI polynomial described in the equation 11:

$$X = C_1 + C_2\ S + C_3\ D + C_4\ S^2 + C_5\ S\ D + C_6\ D^2 + C_7\ S^3 + C_8\ S^2\ D + C_9\ S\ D^2 + C_{10} D^3, \tag{11}$$

where, C_1 to C_{10} represent the specific coefficients for each compressor. X represents the compressor power consumption, the compressor efficiency or the heating capacity. S represents the evaporating temperature and D represents the condensing temperature. The coefficients C_1 to C_{10} were calculated using polynomial regressions and the known compressor data for each parameter.

The weighted efficiency in heating mode for each condensing temperature was estimated using the following expression:

$$\left(\text{Weighted efficiency}\right)_{Tc} = \frac{\sum_{j=1}^{n} H_j\ \eta_{c(Tj)}}{\sum_{j=1}^{n} H_j}, \tag{12}$$

where, $\eta_{c(Tj)}$ represents the compressor efficiency at the corresponding operating point (Te_i, Tc).

4.3. Comparative study of the seasonal performance in cooling mode.

For cooling applications such as: refrigerated storage for frozen foodstuffs, refrigerated containers, and the like, the compressors could work with several evaporating temperatures of 0°C , -10°C, -20°C to -30°C. The cooling SCOP was estimated using the method of the EN14825 normative. The outdoor temperatures are not divided into seasons.

The cooling SCOP was calculated for each evaporating temperature of the inside conditions in Table 2 with the following expression:

$$\text{Cooling SCOP}_{Te} = \frac{\sum_{j=1}^{n} H_j\ Qc_{(T_j)}}{\sum_{j=1}^{n} H_j\ Pa_{(T_j)}}, \tag{13}$$

where, H_j represent the number of bin hours occurring at the corresponding outdoor temperature T_j. The condensing temperature corresponding to each bin temperature was estimated as:

$$Tc_j = T_j + 15K, \tag{14}$$

The $Qc_{(T_j)}$ and $Pa_{(T_j)}$ were calculated using third order ARI polynomials for each evaporating temperatures (Equation 11).

The weighted efficiency in cooling mode for each evaporating temperature was estimated using the following expression:

$$(\text{Weighted efficiency})_{Te} = \frac{\sum_{j=1}^{n} H_j \, \eta_{c(Tj)}}{\sum_{j=1}^{n} H_j}, \tag{15}$$

where, $\eta_{c(Tj)}$ represents the compressor efficiency at the corresponding operating point (Te, Tc_j).

5. RESULTS AND DISCUSSION

5.1. Performance comparison of the SCVI and TSRC.

5.1.1. Comparison of the compressor efficiency.

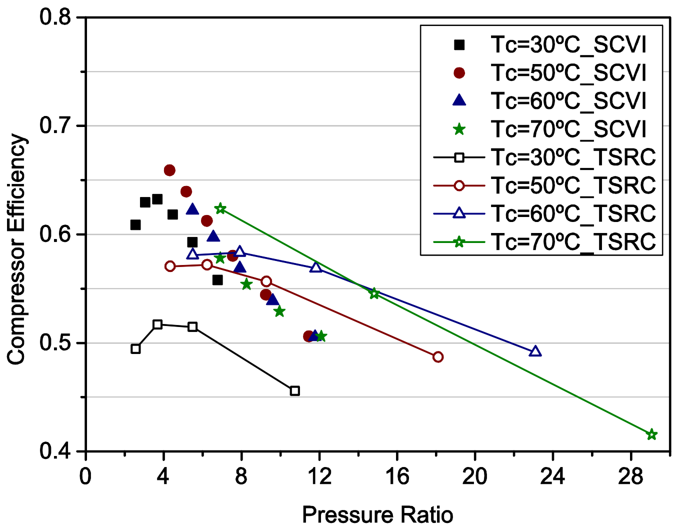

Fig. 5- Compressor efficiency as a function of pressure ratio at several condensing temperatures, for the SCVI and the TSRC.

Fig. 5 depicts the compressor efficiency as a function of the pressure ratio for the SCVI and the TSRC for several condensing temperatures (30, 50, 60 and 70°C). The SCVI presents the optimum point working with pressure ratios about 4 at condensing temperature of 30ºC. For higher pressure ratios and condensing temperatures, the compressor efficiency decreases rapidly up to 0.5 approximately.

Regarding to the TSRC, the curves are flatter; the optimum point moves to the right for pressure ratios of 3.7, 6.2 and 7.9 condensing at 30ºC, 50ºC and 60ºC respectively. For higher pressure ratios, the efficiency decreases smoothly, getting to work with a wide range of pressures.

On the other hand, comparing both compressors, Fig. 5 shows that SCVI improves the efficiencies for pressure ratios up to approximately 7.5, the efficiency improved by 23% to Rp=2.6 at the point (0°C, 30°C) and 7% to Rp=5.5 at the point (0°C, 60°C). The TSRC presented better efficiencies working with pressure ratios above 7.5; the efficiency improved 2.5% to Rp=7.9 at point (-10°C, 60°C) and 12.5% for the Rp=11.8 at the point (-20°C, 60°C).

Moreover, the TSRC was able to work with higher pressure ratios (up to 29) and the SCVI can work with pressure ratios up to 12, both with efficiencies exceeding 0.45.

5.1.2. Comparison of the volumetric efficiency.

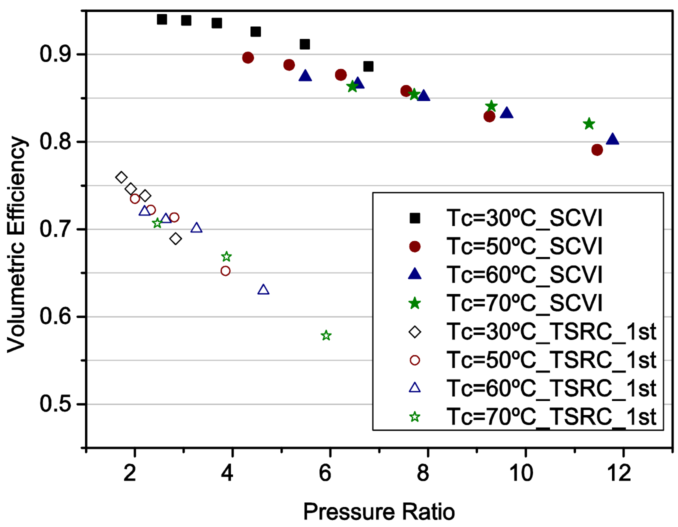

Fig. 6- Volumetric efficiency as a function of pressure ratio at several condensing temperatures, for the SCVI and the TSRC.

Fig. 6 represents the volumetric efficiency of the compressors as a function of the pressure ratio. The TSRC volumetric efficiency curves correspond to the first stage of compression.

The SCVI presents flatter curves of volumetric efficiency, with 1.5% of slope. The best efficiency is reached when the compressor works with pressure ratio between 2.6 and 3.6 for Tc=30°C, however, the SCVI presents high volumetric efficiency values (above 0.8) for any operating point. This feature is due to the SCVI has no harmful dead space and it does not have inlet and outlet valves. The contact between the flanks of scrolls and in their bases and upper edges is almost perfect and constant; due to it has very good axial and radial compliance.

Regarding to TSRC, the volumetric efficiency curves have a larger slope (4.3%). For the first stage, the volumetric efficiency droops to 0.58 for pressure ratio about 6 and condensing temperature of 70°C. Comparatively the SCVI has better volumetric efficiency than the TSRC for any pressure ratios, the SCVI can achieved improvements of volumetric of 32% and 52% for pressures ratios of 2.6 and 6 respectively. According to Navarro, et al., (2007a) and Navarro, et al., (2007b), for a reciprocating compressor, the mechanical and electric losses represent approximately 55% of the total volumetric efficiency losses for all pressure ratios. At low pressure ratios (1.5-2.5), the pressure losses are noteworthy (more than 25%). At high pressure ratios (5-7), the leakages become a significant factor (more than 10%). The relative influence of heat transfer losses between the inlet and outlet remains constant at all pressure ratios, and its overall influence on volumetric efficiency is significant (15%).

5.1.3. Comparison of the cooling COP.

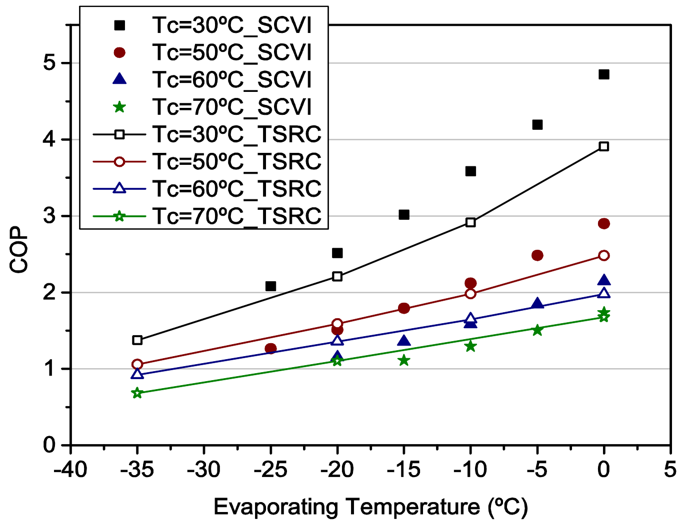

Fig. 7- Cooling COP as a function of the evaporating temperature at several condensing temperatures, for the SCVI and the TSRC.

Fig. 7 illustrates the cooling COP according to the evaporating temperature at several condensing temperatures (30°C, 50°C, 60°C and 70°C) for the SCVI and the TSRC.

The system with SCVI present larger slope curves, the cooling COP reaches values of 4.9 for slight temperature conditions (0°C, 30°C), and 1.1 for extreme conditions (-15°C, 70°C). This compressor is able to work up to -25°C of evaporating temperature, condensing to 30°C and 50°C.

The system with TSRC present less slope curves, the cooling COP reaches values of 3.9 for the point (0°C, 30°C) and 1.1 for (-20°C, 70°C). This compressor is able to work up to -35°C of evaporating temperature, condensing up to 70°C.

Comparing both compressors, the SCVI has 14% better COP at the point (-20°C, 30°C), but for higher condensing temperatures as 50°C and 60°C, the COP is lower than TSRC by 5% and 18%

respectively. This reduction is due to the SCVI efficiency decreases for pressure ratios higher than 7.5.

When the compressors work with higher evaporation temperatures like 0°C, the SCVI presents up to 24% better COP than the TSRC for a condensing temperature of 30°C. For higher condensing temperatures as 50°C, 60°C and 70ºC, the COP can be up to 17%, 9% and 3% better respectively. Therefore, the SCVI can be applied in air conditioning systems and heat pumps working under moderate temperature conditions and pressure ratios lower than 7.5, and the TSRC can be applied in domestic hot water systems for severe climates, cooling systems working to very low evaporating temperatures (under -20°C) and large pressure ratios (above 7.5).

5.1.4. Comparison of the cooling capacity.

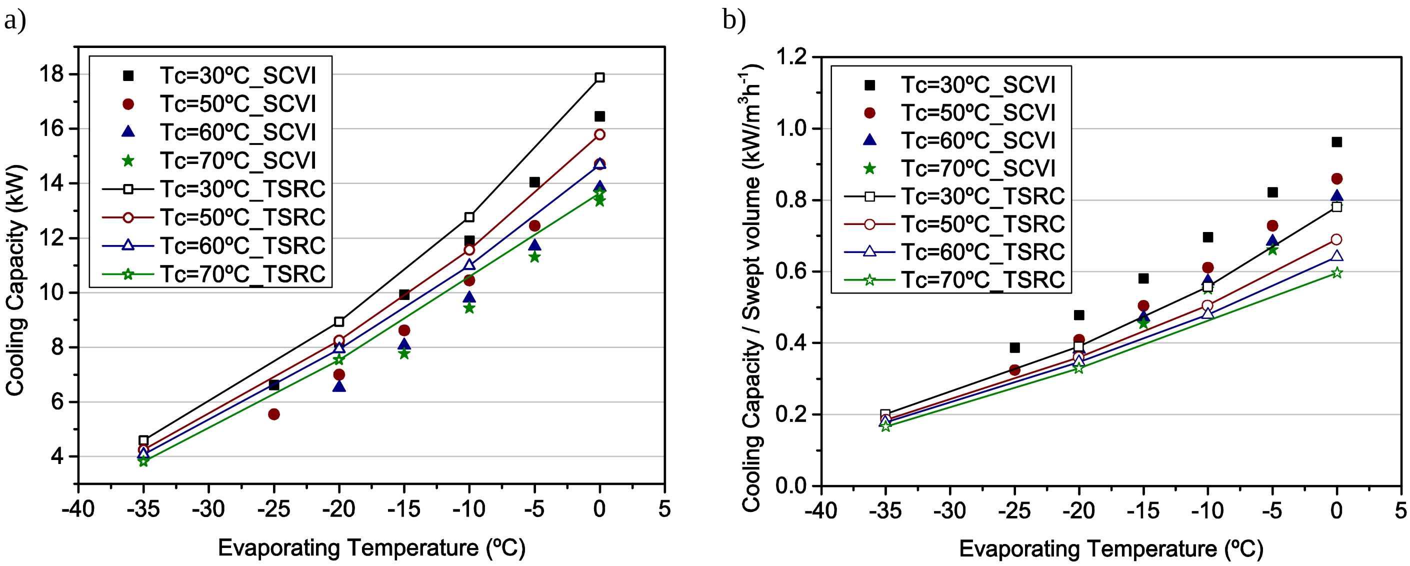

Fig. 8- a) Cooling capacity as a function of evaporating temperature. b) Specific cooling capacity as a function of evaporating temperature at several condensing temperatures, for the SCVI and the TSRC.

Fig. 8 (a) illustrate the cooling capacity as a function of the evaporating temperature. This figure shows that the cooling capacity of the TSRC is slightly higher than the cooling capacity of the SCVI. It is due to the TSRC has a larger swept volume that the SCVI.

In order to compare the cooling capacity independently of the compressor size, Figure 8 (b) shows the specific cooling capacity of both the compressors as a function to the evaporating temperature. This figure illustrates that the system with SCVI produces 23-31% more specific cooling capacity, for high evaporating temperatures (at Te = 0ºC); this is due to the differences between the volumetric efficiency of the compressors shown in Fig. 6. According to Fig. 6, we would expect a large difference between the specific cooling capacities of the compressors for extreme conditions (due to the low volumetric efficiency of the TSRC). However, in Fig. 8 (b) this effect is not observed, on the contrary, the TSRC curves are flatter than the SCVI curves for low evaporating temperatures and the difference between the specific cooling capacities of the compressor decreased up to 20% (at Te = -20ºC). The improvement in the specific cooling capacity is due to the difference in the refrigerant injection between the compressors. This effect can be explained with the difference in the economizer capacity working in extreme temperature conditions as shown in the Fig. 9 (a) and (b).

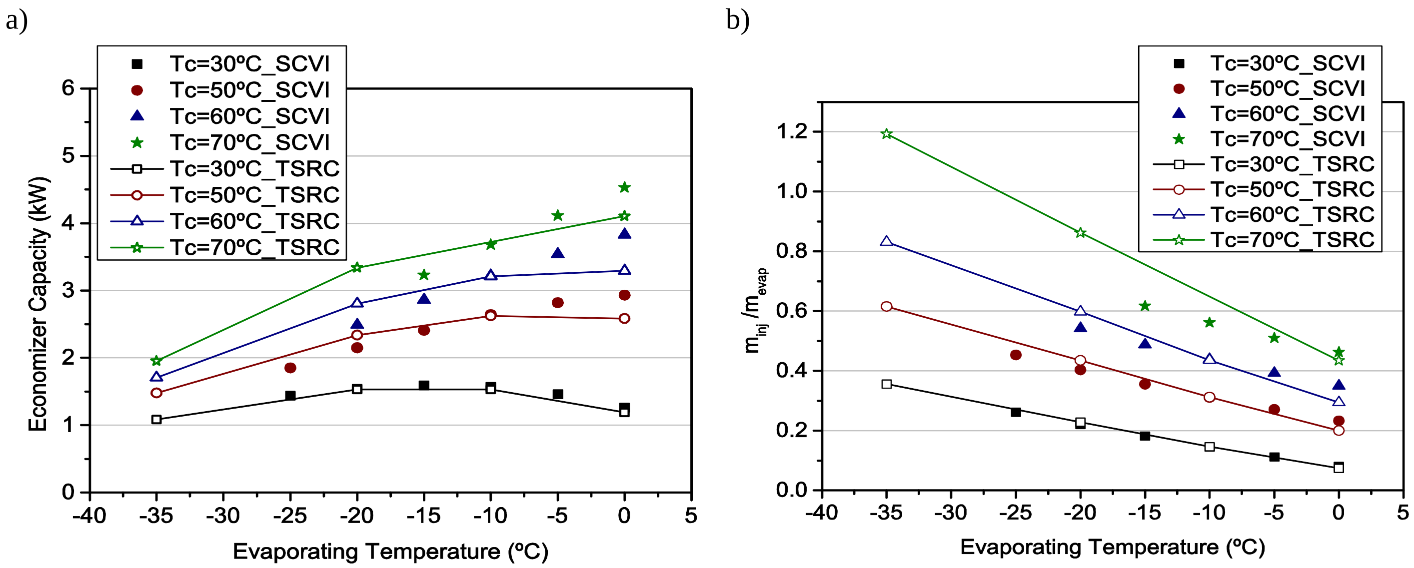

Fig. 9- a) Economizer capacity as a function of evaporating temperature. b) Mass flow injection ratio as a function of the evaporating temperature at several condensing temperatures, for the SCVI and the TSRC.

Fig. 9 (a) shows the economizer capacity for both compressors as a function of the evaporating temperature. This figure shows that the TSRC economizer capacity curves are flatter than the SCVI curves when the compressors work with moderate temperature conditions (Tc=30°C and Te above -10°C). For low evaporation temperatures (-35°C to -20°C), which involves high pressure ratios (above 7.5), the TSRC system presents higher economizer capacity than the SCVI system. Moreover, this effect is shown in the Fig. 9 (b) that depicts the variation of the mass flow injection ratio ($\dot{m}_{inj}/\dot{m}_{evap}$) according to the evaporating temperature. The TSRC mass flow injection ratio is higher as the evaporating temperature decrease and the condensing temperature increase, therefore, the TSRC must inject larger quantity of refrigerant increasing the economizer capacity.

The higher economizer capacity of the TSRC produces a greater subcooling at the condenser outlet, allowing the increasing of the enthalpy difference in the evaporator, which implies the increase in the cooling capacity, as can be seen in the Fig. 3(b). This figure illustrates the p-h diagram of the vapor injection cycle with TSRC. The cooling capacity is calculate as the product between the evaporator mass flow rate and its enthalpy difference, with the equation (16). Therefore, the equation (17) indicates that the cooling capacity improves when the economizer capacity increases, so that, the cooling capacity is significantly compensated working at high pressure ratios. However, despite of higher economizer capacity, the TSRC is not able to reach the specific cooling capacity of the SCVI, as can be seen in Fig. 8(b).

$$Q_c = \dot{m}_{evap} \left[(h_1 - h_5) + \Delta h_{econo}\right] \tag{16}$$

$$Q_c = \rho_1 \, \dot{V}_1 \, \eta_{v1} \, (h_1 - h_5) + Q_{econo} \tag{17}$$

5.2. Comparison of the cooling SCOP.

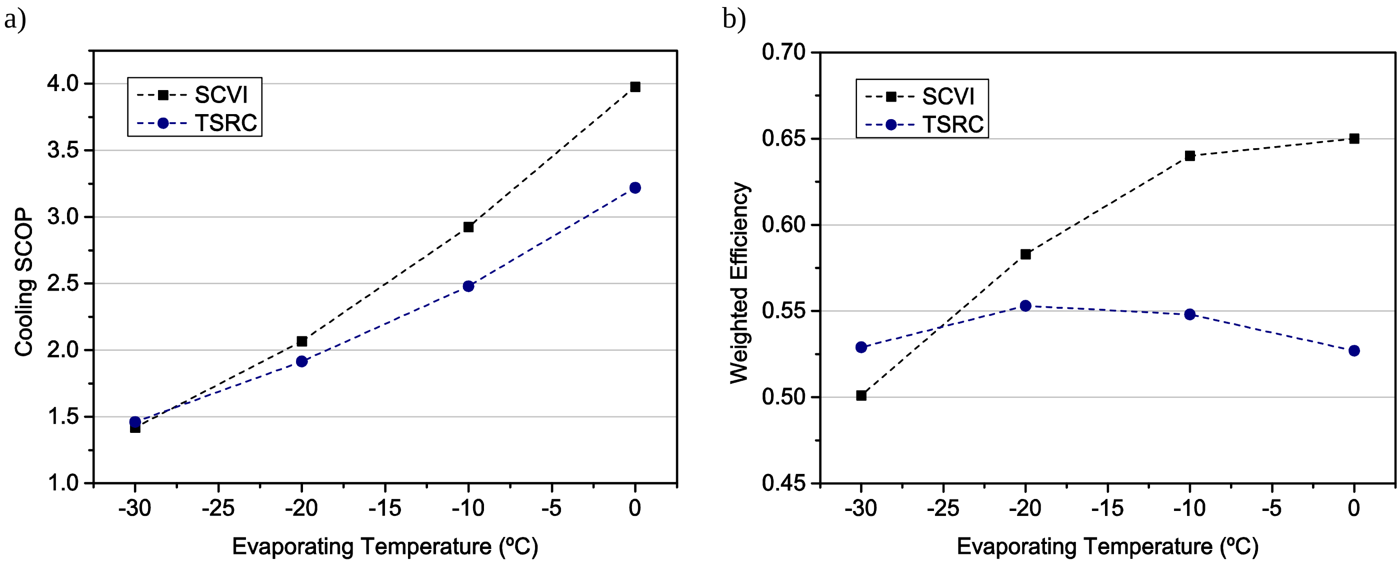

Fig. 10- a) Cooling SCOP as a function of evaporation temperature. b) Weighted efficiency as a function of evaporating temperature, for the SCVI and the TSRC.

Fig. 10(a) depicts the seasonal cooling COP of the SCVI and the TSRC for several evaporating temperatures (0°C, -10°C, -20°C and -30°C). The SCVI reaches cooling SCOP values around 4 working at 0°C of evaporating temperature. For lower evaporating temperatures, the cooling SCOP decreases rapidly to 1.42 working at -30°C. Moreover, the TSRC reaches cooling SCOP values of 3.2 working to 0°C of evaporating temperature. For lower evaporating temperatures, the cooling SCOP decreases to 1.46 working at -30°C. The SCVI cooling SCOP exceeds in 23%, 18% and 8% when it works with evaporating temperatures of 0°C, -10°C and -20°C respectively; in these conditions the SCVI works with better weighted efficiency, as can be seen in Fig. 10(b). Furthermore, the TSRC presents a 3% higher SCOP than the SCVI working at -30°C of evaporating temperature. In these working conditions, the pressure ratios above 7.5 are reached, hence, the TSRC works with better SCOP and the weighted efficiency is higher.

Fig. 10(b) represents the weighted efficiency for cooling mode as function of evaporation temperature. This figure shows that the SCVI has better efficiency when operating at evaporating temperatures of -20°C, -10°C and 0°C. When the compressors works at -30°C of evaporation, the TSRC has efficiency 5.7% above the SCVI, therefore the TSRC has 3% better SCOP. Moreover, the weighted efficiency of the TSRC has a maximum when it works at -20°C of the evaporating temperature; however at that point the SCVI has 5.4% higher efficiency.

The TSRC can be applied in cooling systems for very low temperatures (lower than -25°C), such as ultrafreezing (baked goods, meats, fish, seafood, vegetables and prepared foods), and systems with large differences between evaporating and condensing temperatures. The SCVI can be applied in systems that work with evaporating temperatures higher than -25°C like cooling systems for supermarkets, reefer containers for refrigeration and transportation of fruits, vegetables, pharmaceuticals, flowers, dairy, meats, photographic material and the like.

5.3. Comparison of the heating SCOP.

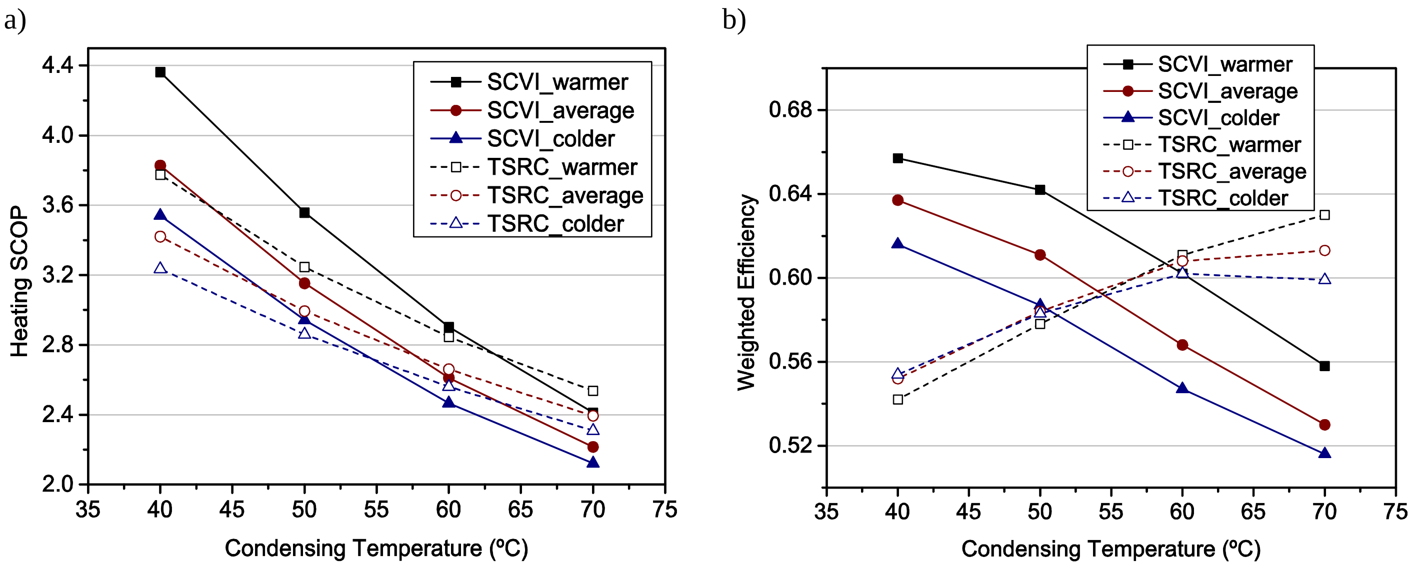

Fig. 11- a) Heating SCOP as a function of condensing temperature. b) Weighted compressor efficiency as a function of condensing temperature, for "warmer", "average" and "colder" heating seasons, for the SCVI and the TSRC

Fig. 11(a) illustrates the heating SCOP for both compressors for several condensing temperatures and climates. For warmer climates, the SCVI presents better cooling SCOP for low and moderate condensing temperatures. The SCVI presents a higher heating SCOP in 16%, 14% and 3% when working with condensing temperatures 40ºC, 50°C and 60°C respectively, and the TSRC improves the heating SCOP by 9% at high condensing temperature (70°C).

In average climates, the SCVI presents 12% and 8% better heating SCOP condensing at 40ºC and 50°C respectively. The TSRC improves the heating SCOP in 3% and 14% for condensing temperatures of 60°C and 70°C respectively.

For colder climates, the behavior is similar to average climate, but the heating SCOP values and the percentages of improvement are lower. The SCVI presents 9% and 3% better heating SCOP for condensing temperatures of 40°C and 50°C respectively. Instead, the TSRC improves the heating SCOP in 4% and 9% condensing at 60°C and 70°C respectively.

Fig. 11(b) represents the weighted efficiency of the compressors working in heating mode. In general terms, the SCVI presents better weighted efficiency condensing at 40ºC and 50ºC for all climates, therefore, the TSRC works with better weighted efficiency condensing at higher temperatures, 60ºC and 70ºC.

The SCVI presented 21% and 11% better weighted efficiency for warmer climates, condensing at 40ºC and 50°C respectively. Instead, the TSRC presented 2% and 13% upper weighted efficiencies condensing at 60°C and 70°C respectively.

For averages climates, the weighted efficiency of the SCVI exceeds in 15% and 5% to the TSRC, condensing at 40ºC and 50°C respectively. Nevertheless, condensing at 60°C and 70°C, the TSRC exceeds in 7% and 15% the weighted efficiencies respectively.

For colder climates, the TSRC presents improvements of 10% and 15% working with 60°C and 70°C condensing temperatures respectively. Weighted efficiencies of the compressors at 40ºC and 50°C condensing temperature were similar.

With these results, the SCVI can be applied for heating in air conditioning systems up to 50°C of condensing temperatures in all kinds of climates. The TSRC can be applied for higher condensing temperatures in systems for domestic water heating for high temperatures (60°C and 70°C), and heating installations in which water input could be up to 80°C.

6. CONCLUSIONS

A comparative study of the performance of a vapor-injection scroll compressor and a two-stage reciprocating compressor was carried out. The comparison was performed in terms of the compressor efficiency, the volumetric efficiency, the COP, and the cooling capacity. The following conclusions have been obtained:

- The SCVI presents better efficiency when working with pressure ratios below 7.5 approximately. For higher pressure ratios, the TSRC works with a better efficiency.
- All over the working range, the SCVI presents better volumetric efficiency than the TSRC, and their relative difference increase as the pressure ratios increase.
- At moderate temperature conditions, the SCVI develops higher cooling COP and capacity.

- At extreme working conditions, the TSRC presents higher cooling COP. The economizer capacity is also bigger, which produces a greater subcooling at the condenser outlet, increasing the enthalpy difference in the evaporator. Therefore, the TSRC has flatter specific cooling capacity curves.

With the results of the seasonal performance study, we conclude that the TSRC can be applied in cooling systems for very low temperatures, such as ultrafreezing and systems that work at very low evaporating temperatures (under -20°C) and large pressure ratios. The SCVI can be applied in systems that work with moderate evaporating temperatures such as cooling systems for supermarkets, reefer containers and refrigeration transportation of merchandise, with up to 24% better SCOP.

For heating applications, the TSRC can be applied in DHW systems and old heating installations with radiators for colder seasons with 9% better SCOP. The SCVI can be applied in air conditioning systems and heat pumps working under moderate condensing temperatures (above 60°C) with 12% better SCOP.

The present study consider only one type of refrigerant (R407C), however, in terms of efficiency and COP, we conclude that for refrigerants with higher pressure ratio (R134A, R32) would be favorable the use of TSRC. On the contrary, for refrigerants with lower pressure ratios (R290, R404A, R410A), would be favorable the use of SCVI.

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