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Analysis of variable speed vapor injection scroll compressors working with several refrigerants: Empirical correlation for the characterization and optimization of the intermediate conditions

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ABSTRACT

Vapor-injection scroll compressors (SCVI) are present in the market for many years. There are several studies in the literature dealing with their proper characterization and the development of correlation in order to estimate properly its intermediate working conditions. The main objective of these correlations is the prediction of the compressor behavior for any condition based on a reduced number of test. Derived from that a correlation would be better as less parameter it will require, as this will imply less number of test in order to determine it.
Nowadays, variable speed SCVI are being installed more widely but not so much studies are available for these type of compressors. In this work, based on previous correlations obtained for constant speed SCVI, a correlation for variable SCVI has been obtained. In order to do that, a set of experimental calorimetric tests has been conducted for different condensing and evaporating temperatures and 3 different compressor speeds. From these data, it has been possible to obtain a correlation depending on four parameters able to predict the injection ratio with an error lower than 10 %. The correlation has been checked with other sources of data available in the public literature obtaining similar deviations from the measured values. Finally and based on these results, the optimum intermediate operation conditions for an injection heat pump with an economizer has been determined in different external conditions. From the optimization process, an improvement in COP up to 5 % could be expected from the theoretical analysis.

1. INTRODUCTION

Nowadays, scroll compressors are widely used in residential and commercial air-conditioning, refrigeration and heat pump applications as well as in automotive air conditioning. This kind of compressor technology is orbital motion, positive-displacement machines that compress with two inter-fitting, spiral-shaped scroll members. They have no dead space, the contact between the flanks of scrolls and in their bases and upper edges is almost perfect and constant; therefore, it has very good axial and radial compliance. Consequently, scroll compressors present advantages such as high compressor and volumetric efficiencies, low vibrations and noise, low torque variations and leakage (ASHRAE Handbook, 2008). Several advanced technologies have been studied in order to enhance the compressor performance and save energy. One of these technologies is the vapor-injection and the other is the use of the variable speed motor.

Vapor-injection technic in scroll compressors has rapidly developed in recent years. The refrigerant injection can either increase the capacity and performance efficiency of the system or decrease the discharge temperature of the compressor to extend the working envelope, especially for refrigeration and heat pump systems working with low evaporating temperatures or high condensing temperatures.
The efficiency of a variable speed compressor with refrigerant injection depends on ambient conditions and compressor frequency as well. Therefore, a loss analysis of a variable speed compressor with refrigerant injection should include the effects of compressor frequency on the performance.

Park *et al.* (2002) developed a model of a variable speed scroll compressor with vapor-injection working with R22. The model was validated considering only the no injection condition showing deviations of the predicted compressor capacity and electrical power lower than 10 % with respect to the experimental ones. The model was then used to investigate the influence of geometrical (injection hole diameter and position) and thermodynamic (refrigerant pressure and quality or superheat) injection parameters on compressor working parameters as a function of rotational frequency. An optimal configuration leading to an increase of COP equal to 12 % COP was found.

Dardenne *et al.* (2015) developed a semi-empirical model of a hermetic, variable-speed vapor-injected R-410A scroll compressor based on the semi-empirical model of a fixed speed scroll compressor presented by Winandy and Lebrun (2002). The model was validated using 63 experimental test conditions. The model requires 10 parameters fitted from experimental data to simulate the process that the refrigerant undergoes from suction and injection ports to discharge port. The model includes the leakage in the compression process and computes the suction and injection refrigerant mass flow rates, the compressor power, and the discharge temperature within ±5 %, ±10 %, ±5 %, ±5 K, respectively.

The current study is focused on the estimation of the intermediate conditions of the variable speed scroll compressor with vapor-injection. The study is based on a previous work (Tello-Oquendo *et al.*, 2017), where a correlation was identified for the intermediate conditions for fixed speed scroll compressor with vapor-injection (Equation 1). This correlation relates the injection mass flow vapor-injection ratio with the intermediate pressure ratio throughout a linear function. The advantages of this correlation are the simplicity and accuracy to estimate the injection mass flow rate as a function of the intermediate pressure and the suction mass flow rate. In addition, the cited correlation represents an intrinsic characteristic of the compressors which depends on the compressor design and the location and size of the injection port. The correlation is independent of the injection mechanism of the cycle, and it was evaluated with experimental data for several intermediate conditions (intermediate pressure and injection superheat) independently.

$$\frac{\dot{m}_{inj}}{\dot{m}_e} = A + B\,\frac{P_{inj}}{P_e} \qquad (1)$$

Other authors proposed more complex correlations to mapped the vapor-injection ratio by using a dimensionless-PI correlation, which has ten coefficients fitted from experimental data (Lumpkin *et al.*, 2018).

In this study, a variable speed vapor-injection scroll compressor (VS-SCVI) was characterized by several working conditions and 3 different frequencies. From the experimental data of the compressor, a correlation of the intermediate conditions of the compressor is obtained based on the previous correlation obtained for constant speed SCVI. The correlation is checked with other sources of data available in the public literature obtaining similar deviations from the measured values. Based on these results, the optimum intermediate operation conditions for an injection heat pump with an economizer has been determined in different external conditions.

2. EXPERIMENTAL SETUP

The compressor rating procedure was performed based on the characterization methodology for vapor-injection scroll compressors proposed in a previous work (Tello-Oquendo *et al.*, 2017) and on the European Standard (UNE-EN 13771-1, 2003). Based on this standard, the refrigerant mass flow rate is the determining parameter to be measured, and primary and confirming measurements have 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.

Figure 1 shows the scheme of the calorimetric bench used for the compressor characterization. The condenser mass flow rate is directly measured using a Coriolis-type (Fisher–Rosemount Micro-Motion CMF025M), C-1 in Figure 1. Several PID control loops 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 calorimetric bench is fully automated and designed to reach any allowable test conditions without manual adjustments. The instrument accuracies of pressure transmitter (Fisher–Rosemount 3051) and temperature transmitter (RTD-PT 100) are 0.02 % and 0.05 ºC, respectively.

In order to test the vapor-injection compressors, the calorimetric bench was modified to simulate the injection conditions and the interaction with the economizer. Part of the liquid (injection mass flow rate) is derived from the condenser outlet and is expanded to the intermediate pressure in an expansion valve (EEV-1 in Figure 1). After the expansion valve, the injection mass flow rate is vaporized in a heat exchanger using a secondary circuit of a water-glycol mixture. Electric resistors control the temperature of the water-glycol mixture in order to fix the injection superheat. With this arrangement, the system is capable of independently controlling the intermediate pressure and the injection temperature. The injection line is also equipped with a Coriolis-type mass flow meter with uncertainty of ±0.025 g/s (C-2 in Figure 1), a pressure transducer with a precision of 0.2 %, a RTD with a precision of 0.1K, visors and an electrovalve located before the expansion valve (EEV-1 in Figure 1).

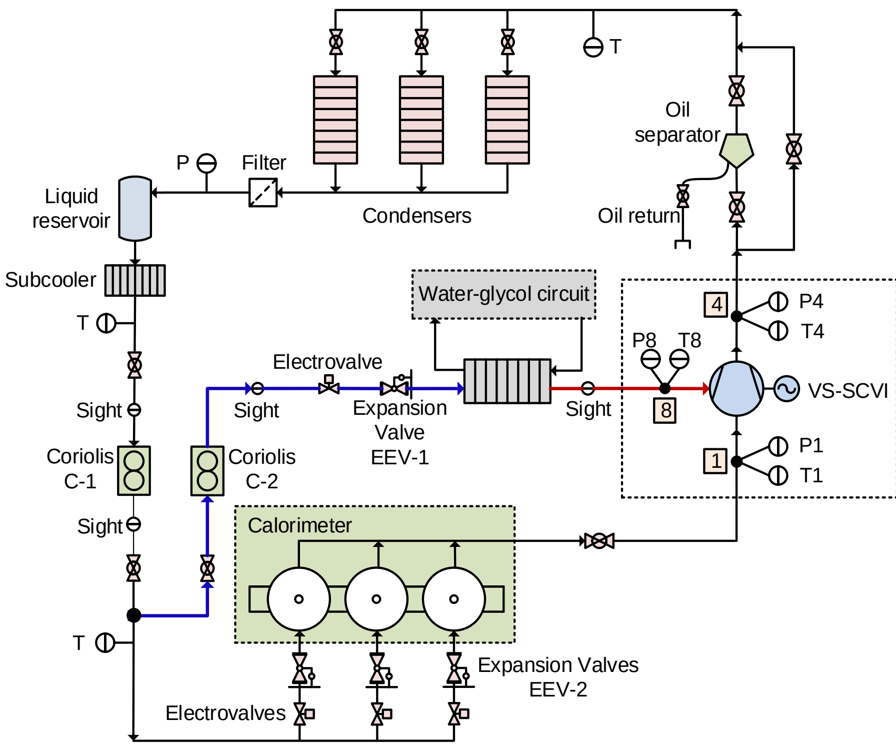

Figure 1: Calorimetric bench schematic

The evaporator mass flow rate is calculated by equation (2) and is compared with the secondary refrigerant calorimeter based result.

$$\dot{m}_e = \dot{m}_c - \dot{m}_{inj} \tag{2}$$

The calorimetric bench was designed to control the operating conditions of the vapor-injection compressor at the points: (1), (4) and (8) (see calorimetric bench schematic in Figure 1). The compressor used for the characterization was a variable speed vapor-injection scroll compressor (VS-SCVI), of 17.28 m^3/h (swept volume). The VS-SCVI was tested with R-290 as a refrigerant.

The procedure of characterization begins with the setting of the condensing pressure, evaporating pressure and the superheat at the evaporator outlet acting on the flow rate of the water condenser, valves EEV-2 and resistors of the calorimeter, respectively. The intermediate pressure is regulated by the electronic expansion valve (EEV-1). The injection superheat is fixed with the water-glycol temperature through a heat exchanger. An injection superheat of 5 K was chosen for testing, since in the majority of systems, the intermediate pressure control is performed with a thermostatic expansion valve. This valve needs a minimum superheat to regulate properly and to ensure that no liquid is injected, therefore, an intermediate superheat of 5 K is appropriate. For all evaporating and condensing temperatures, the values of the considered parameters in the characterization are the superheat at the compressor inlet of 10 K and the injection superheat of 5 K.

3. RESULTS AND DISCUSSION

3.1 Vapor-injection mass flow rate correlation

The analysis of the intermediate conditions of the variable speed vapor-injection scroll compressors (VS-SCVI) was performed with the experimental data obtained for the compressor working with R-290 and also for a compressor working with R-410A from the literature (Dardenne *et al.*, 2015). Figure 2 shows the injection mass flow rate ratio ($\dot{m}_{inj}/\dot{m}_e$) as a function of the intermediate pressure ratio (P_{int}/P_e). For all frequencies, the both plotted ratios show a linear tendency between them, with an R-square correlation factor higher than 0.98. This correlation corresponds to the expression (1) identified in a previous work (Tello-Oquendo *et al.*, 2017).

a) b)

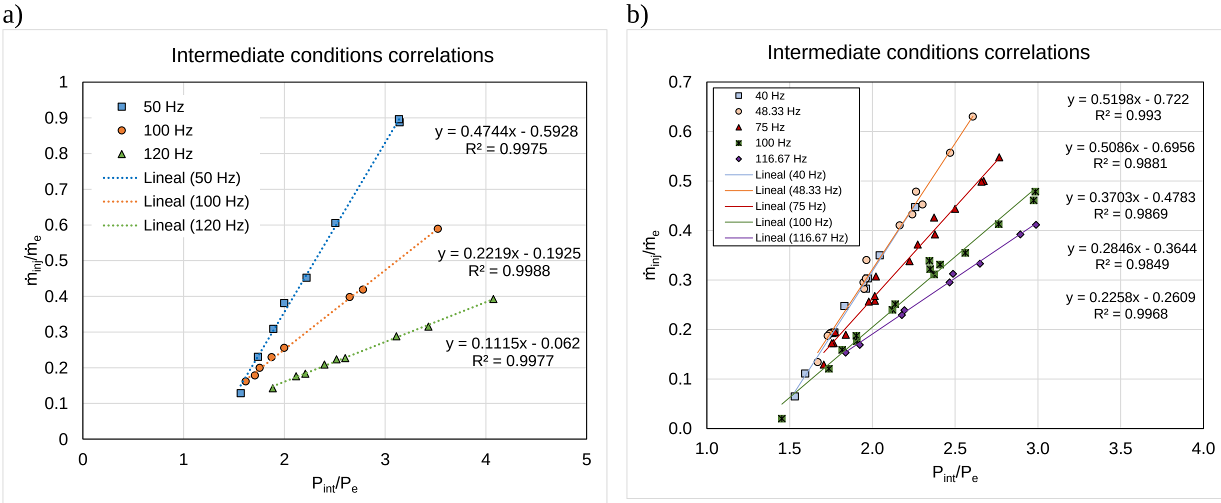

Figure 2: Injection mass flow rate ratio vs intermediate pressure ratio for each frequency. a) VS-SCVI working with R-290. b) VS-SCVI working with R-410A (Dardenne *et al.*, 2015)

The coefficients A and B of the linear correlation (1) can be plotted as a function of the frequency. Figure 3 shows the variation of the coefficients A and B depending on the frequency for both compressors.

a) b)

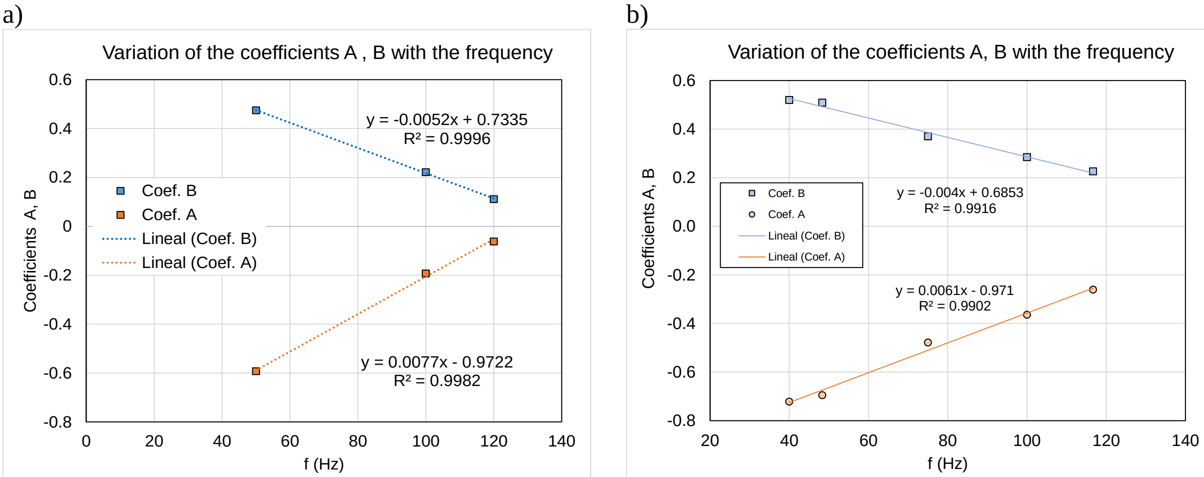

Figure 3: Variation of the coefficients A, B with the frequency. a) VS-SCVI working with R-290. b) VS-SCVI working with R-410A (Dardenne *et al.*, 2015)

Based on Figure 3, the coefficients can be correlated with a linear expression as a function of the frequency. In order to find a general correlation between the intermediate conditions taking into account the frequency, the correlation (1) can be rewritten for variable speed vapor-injection scroll compressors.

$$\frac{\dot{m}_{inj}}{\dot{m}_e} = (A_0 + fA_1) + (B_0 + fB_1)\frac{P_{inj}}{P_e} \tag{3}$$

Figure 4 represents the comparison of the experimental and predicted injection ratio ($\dot{m}_{inj}/\dot{m}_e$) for the two compressors. The mean average error and root mean squared error formulas are listed in Equations (4) and (5), respectively. The results of the correlation for the combined data for the two VS-SCVI are shown in Table 1.

$$MAE = \frac{100}{n}\sum_{i=1}^{n}\left|\frac{X_{pred,i} - X_{exp,i}}{X_{exp,i}}\right| \tag{4}$$

$$RMSE = \frac{100}{\bar{X}_{exp}}\sqrt{\sum_{i=1}^{n}\frac{\left(X_{pred,i} - X_{exp,i}\right)^2}{n}} \tag{5}$$

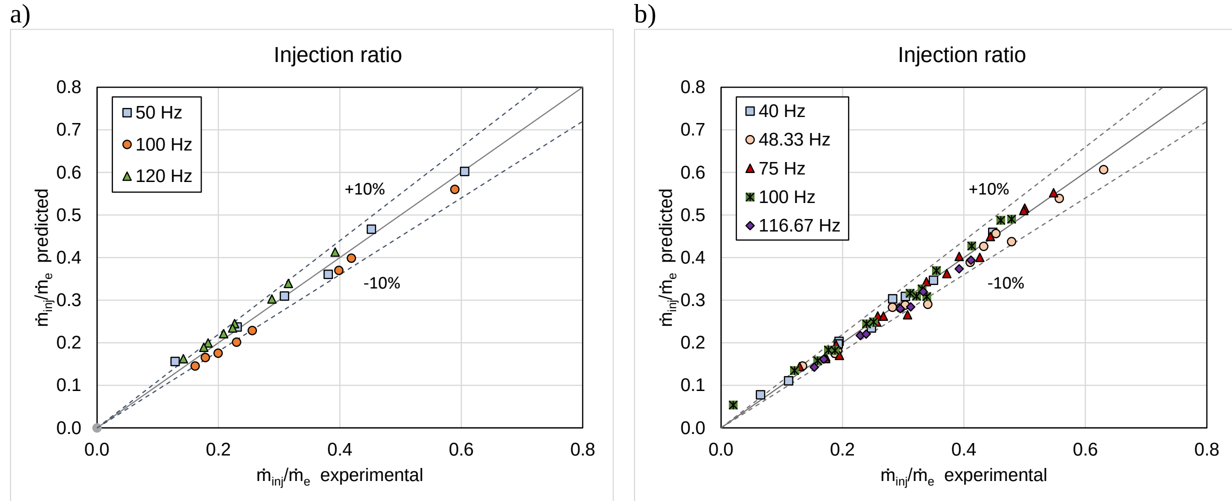

Figure 4: Comparison of the experimental and predicted injection mass flow ratio. a) VS-SCVI working with R-290. b) VS-SCVI working with R-410A (Dardenne *et al.*, 2015)

Table 1: Results of the intermediate conditions correlation

Compressor	MAE (%)	RMSE (%)	R^2
VS-SCVI (R-290)	6.91	5.63	0.9957
VS-SCVI (R-410A)	7.36	5.59	0.9919

The present correlation (3) provides accurate results in the prediction of the injection ratio as a function of the intermediate pressure ratio and the frequency. The main advantage of the present correlation is the simplicity and accuracy to estimate the intermediate conditions of the VS-SCVI. This correlation can be used in a cycle model in order to predict the injection mass flow rate and the intermediate pressure for a given injection mechanism and compressor frequency.

Moreover, this correlation shows better results than others correlations found in the literature, such as one presented by Lumpkin *et al.* (2018), in which the results of the injection ratio prediction were MAE=9.8 %, RMSE=6.7 % and R^2=0.9755 from the same experimental data of Dardenne *et al.* (2015). In addition, the cited correlation is more complex and it has more parameters.

3.2 Optimization of the intermediate conditions for a VS-SCVI

In this section, the optimum intermediate conditions for heating applications are analyzed, considering an air to water heat pump with a VS-SCVI (R-410A). The compressor performance was obtained from Dardenne *et al.* (2015) and the correlation (3) was used to estimate the intermediate conditions (P_{int}, and $\dot{m}_{inj}$) depending on the frequency.

In a real system with a temperature lift in the secondary fluid of the condenser (ΔT_w), a water inlet temperature ($T_{in,w}$), and by assuming a condenser with infinite heat transfer area, the optimal COP_h of the system is obtained for a unique subcooling. According to Pitarch *et al.* (2017), the optimal subcooling is obtained when the condition of having two pinch points of 0 K between refrigerant and secondary fluid take place at the same time in the condenser of infinite heat transfer area (see the temperature profile of Figure 5), this condition constitutes another thermodynamic constraint of the cycle, which must be satisfied to optimize the intermediate pressure. Therefore, the optimal intermediate conditions of the two-stage vapor compression cycles have to be analyzed in terms of the working conditions and the subcooling.

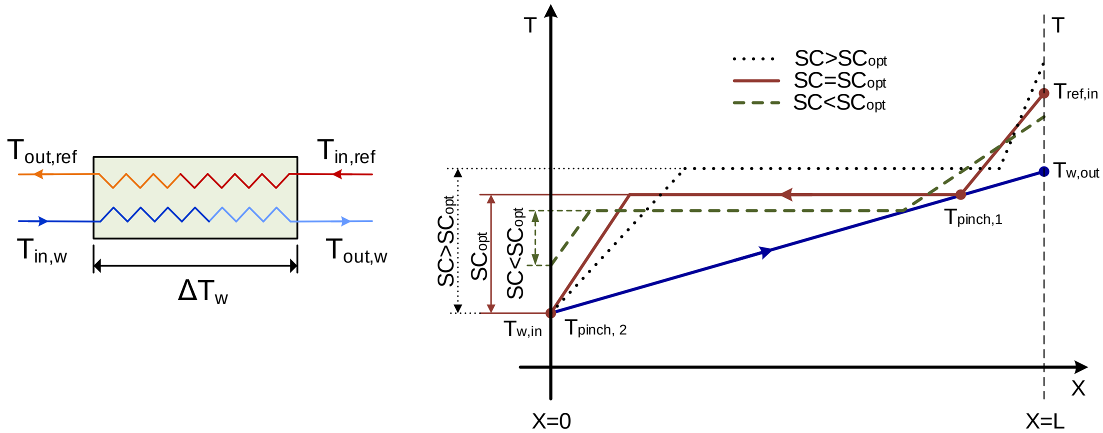

Figure 5: Temperature profile of water and refrigerant into a condenser with optimum subcooling.

In the condenser, the refrigerant temperature is limited by the secondary fluid temperature profile. The optimum subcooling in a condenser with infinite area can be obtained solving the system of Equations (6) - (10). Equation (6) is the energy balance in the condenser, Equation (7) is the energy balance taking the internal pinch point as a reference and the Equation (8) represents the equality of temperatures of refrigerant and secondary fluid in the internal pinch point. The assumption of the infinity heat transfer area in the condenser implies having the condition of Equation (10).The optimal subcooling and the corresponding condensing temperature are outputs of the model.

$$\dot{m}_w \, Cp_w \left(T_{w,out} - T_{w,in}\right) = \dot{m}_c (h_4 - h_5) \tag{6}$$

$$\dot{m}_w \, Cp_w \left(T_{pinch,1} - T_{w,in}\right) = \dot{m}_c \left(h_{pinch,1} - h_5\right) \tag{7}$$

$$T_{pinch,1} = T_c \tag{8}$$

$$h_5 = h(P_5, T_d(P_5) - SC) \tag{9}$$

$$T_{pinch,2} = T_{w,in} = T_5 \tag{10}$$

If the system works with a subcooling lower than the optimum one, the condensing temperature is limited by the internal pinch point ($T_{pinch,1}$). If the system works with a subcooling larger than the optimum one, the temperature of the refrigerant is limited by the external pinch point ($T_{pinch,2}$) and the condensing temperature increases in order to fulfill the heat balance in the condenser (Tello-Oquendo *et al.*, 2018).

In this study, a heat pump equipped with a VS-SCVI (R-410A) was simulated for heating applications for very high temperature ($T_{w,out}$=65 ºC) for old radiators. The evaporating temperatures considered vary between -20 ºC to 10 ºC. The heat pump uses an economizer in the injection mechanism. The economizer size is defined by the temperature approach of 5 K (EN 12900, 2014). For all working conditions, the values of the considered parameters in the simulation are the superheat at the compressor inlet of 10 K, injection superheat of 5 K. By fixing these cycle parameters, the system has two independent variables for optimizing the COP, the compressor frequency, and the subcooling. All the thermophysical properties of the refrigerant at the different points are calculated with the NIST REFPROP database (Lemmon *et al.*, 2010).

Figure 6 shows the heat pump performance working at several frequencies, for a heating application when the water inlet temperature is 40 ºC, the water outlet temperature is 65 ºC, and for several evaporating temperatures. These results are obtained considering the condition of the two pinch points in the condenser of infinite heat transfer area. As expected, the heating capacity increases with the compressor frequency, as well as the compressor consumption due to the refrigerant mass flow through the compressor increases. Nevertheless, the frequency does not have a big influence on the optimum COP. This behavior can be explained by the low variation of the compressor efficiency with the frequency shown in Figure 6 (d) (data obtained from Dardenne *et al.*, 2015).

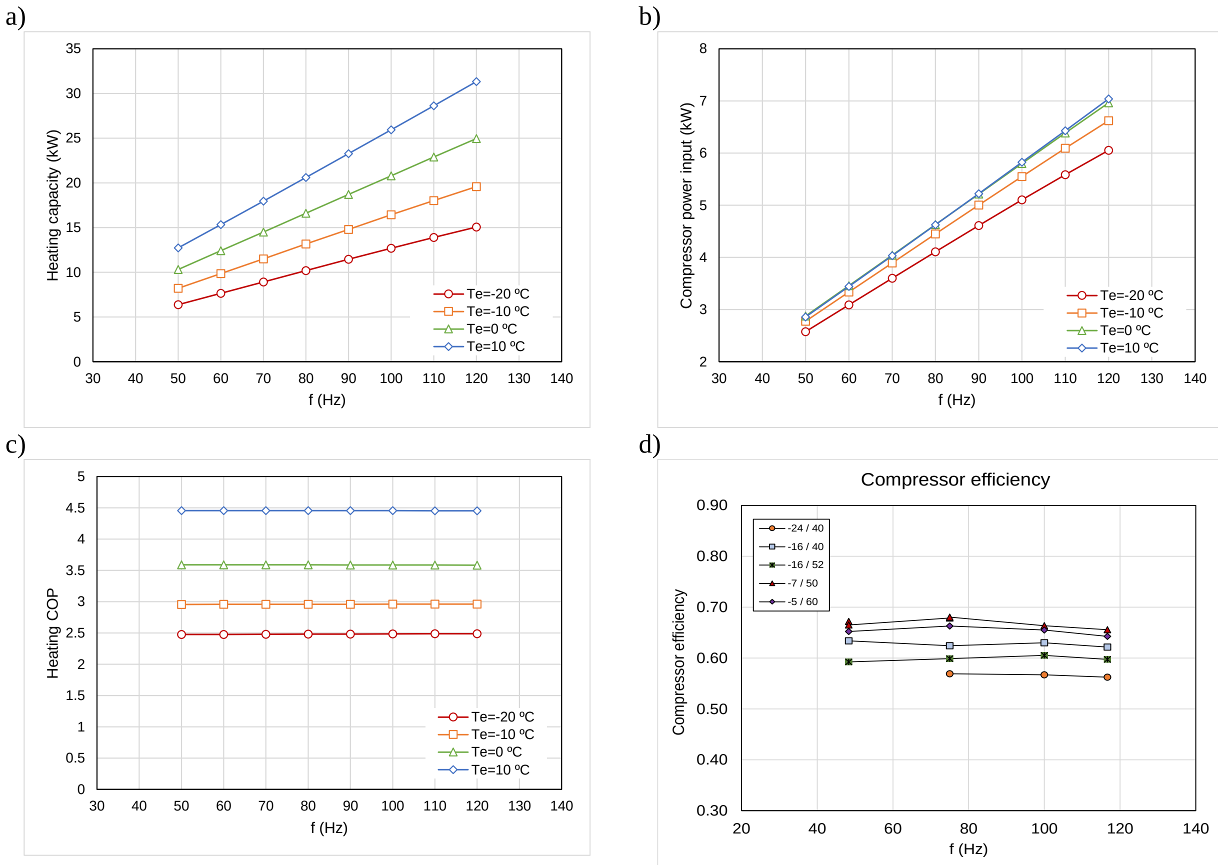

Figure 6: a) Heating capacity. b) Compressor power input. c) Optimum heating COP d) Compressor efficiency (Dardenne *et al.*, 2015). All parameters as a function of the compressor frequency.

Figure 7 shows the optimum intermediate conditions of the compressor and the condensing temperature and subcooling for several compressor frequencies. The optimal intermediate pressure increases as the compressor frequency increase for low evaporating temperatures. For higher evaporating temperatures (10 ºC) the intermediate pressure does not vary for all frequencies. Although the injection mass flow rate increases with the intermediate pressure, the injection ratio decreases for low evaporating temperatures and is almost constant for higher evaporating temperatures, as can be seen in Figures 7 (b) and (c). On the other hand, Figure 7 (d) shows the subcooling and the condensing temperatures that optimize the COP. The optimum subcooling is around 16 K and is smoothly higher for high evaporating temperatures.

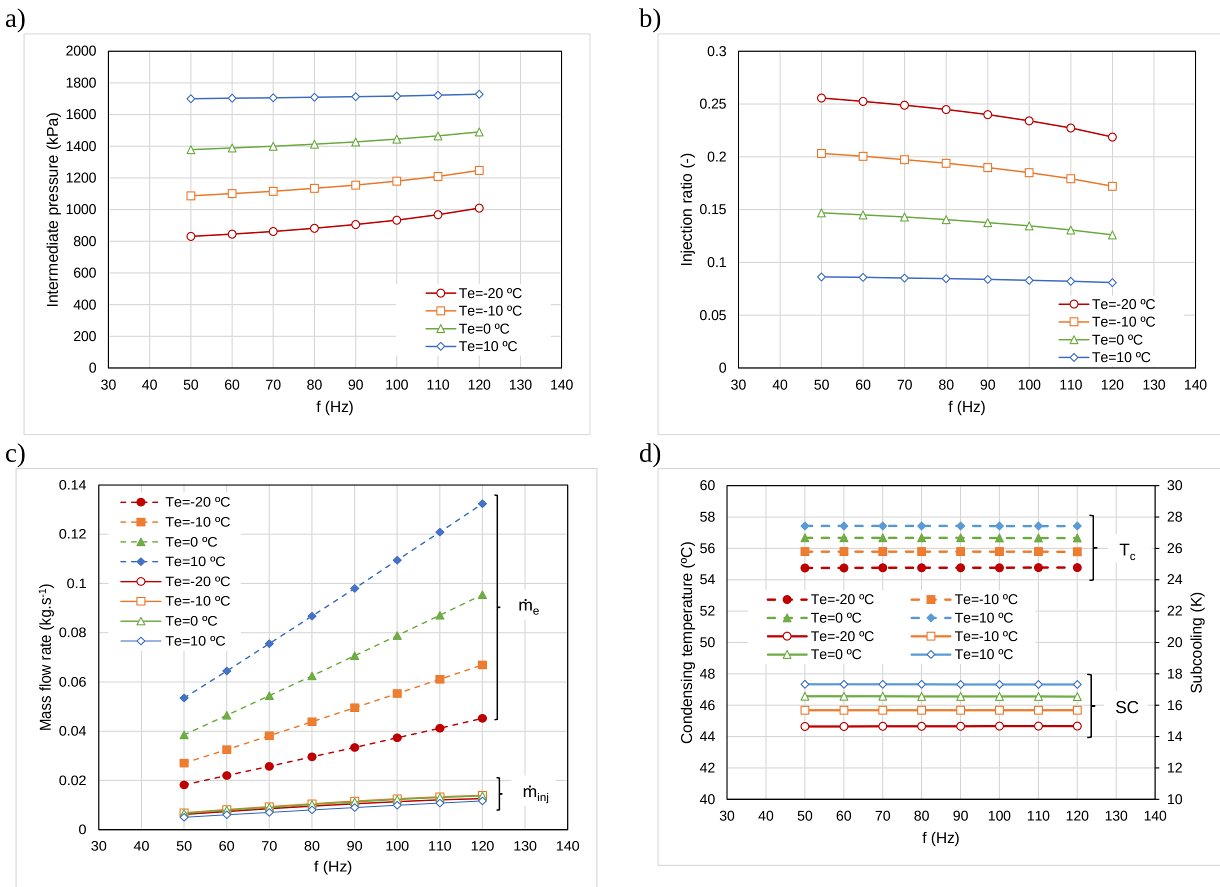

Figure 7: a) Intermediate pressure. b) Injection ratio. c) Mass flow rates d) Subcooling and condensing temperature. All parameters as a function of the compressor frequency.

In order to show the influence of the subcooling on the COP and the intermediate conditions, several simulations have been done for subcooling between [0 – 25] K for the working condition T_e= -20 ºC, $T_{w,in}$=40 ºC, $T_{w,out}$=65 ºC.

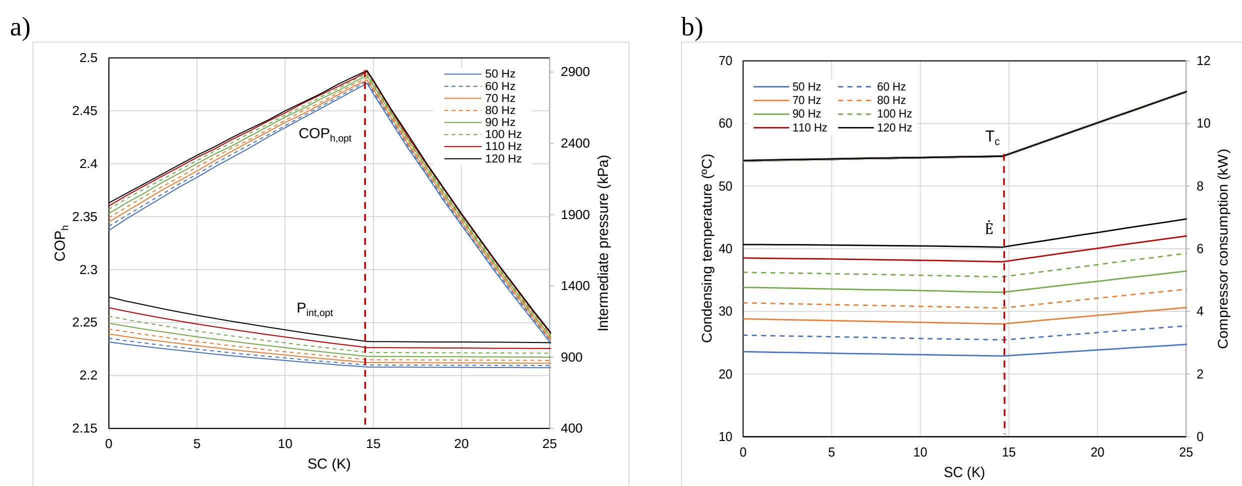

Figure 8: a) Variation of the COP and the intermediate pressure as a function of the subcooling. b) Variation of the condensing temperature and compressor consumption as a function of the subcooling.

Figure 8 (a) illustrates the variation of the COP and the intermediate pressure as a function of the subcooling at the condenser outlet. An optimum point of the cycle was identified when the system works with SC=14.7 K. As the subcooling increases, the temperature at the condenser outlet (refrigerant side) decreases and the energy transfer capacity of the economizer is reduced. Consequently, in order to maintain the superheat at the economizer outlet, the injection ratio decreases along with the intermediate pressure. Thus, the intermediate pressure decreases as the subcooling increases up to the optimum subcooling, as can be seen in Figure 8(a). At this point, the temperature at the outlet of the condenser equals the inlet temperature of the secondary fluid, which corresponds to the thermal limit (see pinch point 2 in Figure 5).

For subcooling values larger than the optimum point, the intermediate pressure is almost constant as well as the injection ratio. As the subcooling increases the condensing temperature increases as well as the compressor consumption (see Figure 8 (b)) producing a reduction of the heating COP despite the increase of the heating capacity. Table 2 summarizes the cycle performance comparison when the system works with the optimal subcooling and cero subcooling.

Table 2: Comparison of the cycle performance when the system works with optimum subcooling and cero subcooling.

Frequency (Hz)	**P_{int} decreasing (%)**	**Heating capacity improvement (%)**	**COP improvement (%)**	**Discharge temperature increasing (K)**
50	17.45	0.47	5.95	7.50
60	18.16	0.80	5.85	7.60
70	18.86	1.17	5.71	7.60
80	19.57	1.49	5.62	7.80
90	20.42	2.05	5.52	7.80
100	21.36	2.51	5.43	7.90
110	22.42	3.19	5.38	8.00
120	23.60	3.86	5.29	8.10

For the studied application, the intermediate pressure decreases up to 23.6 % when the system works with the optimum subcooling with respect to the system working with SC=0 K. The COP improves by at least 5 % in the optimal SC. The heating capacity improves up to 3.9 % for high frequencies. Because of the injection pressure is lower in the optimum subcooling, the injection mass flow rate is lower and the discharge temperature increases up to 8 K for high frequencies.

6. CONCLUSIONS

This paper presents an analysis of variable speed scroll compressors with vapor-injection (VS-SCVI). The following conclusions can be drawn from the study:

- A correlation between the intermediate conditions of VS-SCVI has been obtained from experimental data. The injection ratio was correlated with the injection pressure ratio and the frequency.
- The correlation can predict the intermediate conditions (injection ratio) with an error lower than 10 %. The correlation has been checked with other sources of data available in the public literature obtaining similar deviations from the measured values and better results than other correlations.
- It is important to note that the proposed correlation was verified for frequencies higher than 40 Hz (according to the available experimental data). The future work is evaluate the validity of the obtained correlation for low frequencies.
- For a given application, an optimum subcooling was identified in the condenser considering the temperature lift of the secondary fluid. The optimal subcooling must be considered in the estimation of the optimum intermediate pressure.
- For water heating application ($T_{w,in}$=40 ºC, $T_{w,out}$=65 ºC), using an air to water heat pump with R-410A as a refrigerant, the predicted optimum intermediate pressure is up to 23.6 % lower than optimum intermediate pressure calculated for a SC= 0 K, for high frequencies.
- Under the optimum intermediate conditions, the COP improves by at least 5 % when the system works with the optimum subcooling (14.7 K) with respect to the system working with SC=0 K. With the optimum

subcooling, the heating capacity improves up to 3.8 % and the discharge temperature increases up to 8 K for high compressor frequencies.

NOMENCLATURE

$\dot{E}$	compressor consumption	(kW)
f	frequency	(Hz)
h	enthalpy	(kJ/kg)
$\dot{m}$	mass flow rate	(kg/s)
P	pressure	(kPa)
SC	subcooling	(K)
T	temperature	(ºC)
X	variable	(-)

Subscript

c	condenser
d	dew point
e	evaporator
inj	injection
int	intermediate
opt	optimum
w	water

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