Study of oil circulation in variable speed scroll compressor working with propane

Rubén Ossorio, Emilio Navarro-Peris*

Instituto Universitario de Investigación en Ingeniería Energética (IUIIE)
Universitat Politècnica de València, Camino de Vera s/n, ed. 8E cubo F 5°, Valencia, 46022, Spain
Phone: 34-96-3877000 Ext. 88280, Fax: 34-96-3879126

Abstract:

The use of hydrocarbons as refrigerants is being more common in the recent years, nevertheless the solubility of propane in POE oils is significantly higher than in HFCs. This fact has made that POE oils used with hydrocarbons usually have a higher viscosity than the commonly used with HFCs. This approach solves the lubrication problem in the compressor but the way in which oil could affect the system performance is not so widely studied. In this work, an experimental analysis about the oil circulation rate of a variable speed compressor working with propane has been done. The study has analysed the oil circulation at different compressor speeds and evaporating temperatures. The obtained results have shown that the oil circulation rate increases with the compressor speed having values higher than 5% at the higher speeds. This fact can penalize the COP of the system in more than 15% in these working conditions.

Keywords: Variable Speed Compressor, OCR, Oil separator, Compressor efficiency,

Nomenclature:

C	regression coefficient
Cp	heat capacity
displ	compressor's displacement
h	enthalpy
$\dot{m}$	mass flow
MO	Mineral Oil
μ_c	compressor efficiency
μ_v	volumeric efficiency
N	rotational speed
OCR	Oil Circulation Rate
P	pressure
P_{electr}	electrical power
P_{loss}	power lost to ambient
POE	polyolester
UV	Ultra Violet
ϑ	specific volume

Subscripts:

1	at compressor inlet
2s	at compressor outlet considering isentropic compression
Coriolis	measured with coriolis sensor
Calorimeter	measured with calorimetric balance
e	effective
Ineff	not producing refrigerating effect
in / out	at evaporator inlet / outlet conditions
max	maximum
oil	lubricant oil
refr	refrigerant
total	sum of refrigerant and oil

1. Introduction

Variable speed compressors have spread all over the refrigeration and heat pump systems thanks to its capability to adapt the cooling capacity to the demand, improving also the comfort conditions (Qureshi and Tassou, 1996). This technology introduces several advantages compared with On/Off cycling. However, one of the problems that this technology has to face is a correct lubrication in all speed range.

Lubrication using oil is necessary in the vast majority of compressors for the correct lubrication of the moving parts, to ensure a seal in the compression chamber and to lower the compressor temperature. In order to ensure a sufficient quantity of oil in the key spots different technologies arises, in scroll compressors, a drill into the shaft acts as a pump to arise the oil that lays in the bottom part to the main bearing and to the compression chamber which is typically in the upper part (Branch and Lepak, 2019). This pumping action depends mainly on the rotational speed of the shaft. Consequently, when designing variable speed compressors, having enough pumping action at low speeds implies having too much pumping action at high speeds which also implies an excess of oil discharged out of the compressor. Ribeiro and Barbosa (Ribeiro and Barbosa, 2016) suggested a model of a centrifugal oil pump based on the study of (Kim and Lancey, 2003) to estimate the oil mass flow discharged by the compressor. However, the model used hard to find geometric data and a correction factor using 6 correlation coefficients.

Oil, although being necessary in the compressor, it is considered as a contaminant in the rest of the system. Oil changes thermodynamic properties of the refrigerant, reduces the heat transfer coefficient in the heat exchangers and, in worst cases, can block the expansion device (Kruse and Schroeder, 1985). In addition, an excess of oil migrating to the system could led to not having enough oil inside the compressor for a correct lubrication, limiting its effective life. Consequently, it is important to estimate the quantity of oil circulating in the system and, to do so, Oil Circulation Rate (OCR) is the main parameter used in the literature, which is expressed as

$$OCR = \frac{\dot{m}_{\mathrm{oil}}}{\dot{\mathrm{m}}_{\mathrm{oil}} + \dot{\mathrm{m}}_{refr}} \quad (1)$$

Where $\dot{m}_{\mathrm{oil}}$ is the oil mass flow circulating in the system and $\dot{\mathrm{m}}_{refr}$ is the refrigerant mass flow. Accordingly to Lottin study (Lottin et al., 2003a), values of OCR up to 0.5% have negligible effect in the system. However, above this percentage, the performance of the unit decreases significantly. In particular, with OCR of 5%, the COP can be reduced up to 15%.

Lotting studied the effect of oil circulation in the system (Lottin et al., 2003b) and concluded that the most influenced element was the evaporator as the overall heat transfer coefficient (HTC) decrease with high concentrations of oil which typically occur when vapor quality exceeds 70%. To quantify this effect, Lottin followed a thermodynamic approach considering the refrigerant and oil as a zeotropic mixture and applying three different HTC correlations: Gungor and Winterton (Gungor and Winterton, 1986) and finally Bivens and Yokozeki (Bivens and Yokozeki, 1994) .

To limit the OCR, oil separators are used. These devices are placed at the discharge line of the compressor and separate the oil droplets from the refrigerant using centrifugal forces and return the separated oil to the compressor. In this way, oil circulation on the heat exchangers and expansion devices are significantly decreased as the typical efficiency of these devices is higher than 90%. In small systems though, the oil separator is discarded as it is not economically advantageous. Note that, in these cases, a noticeable decrease of the performance can occur compared with expected values announced by the manufacturer as the latter are reported using oil separator and testing OCR is never displayed.

The procedure to determine the oil circulation rate (OCR) is described in ASHRAE 41.4 standard (ASHRAE, 2015) and it is based on taking samples from the liquid line of the refrigeration circuit. Summarizing, the steps of the procedure are the following: take a refrigerant sample and weight it, then the refrigerant in the sample is slowly evacuated letting it evaporate through a capillary tube, create vacuum in the sample cylinder, weight it again, clean the interior with acetone, let it evaporate… The described methodology results in accurate results when all the steps are rigorously followed but it is a time-consuming procedure with tedious steps which limits the number of tests that can viably be done. Additionally, it is needed a scale with great accuracy and resolution and, what it is more important, the sampling is invasive as it affects the overall quantity of oil and refrigerant in the studied system (Wujek et al., 2007).

With such a cumbersome procedure, generating a wide map of OCR values depending on the working conditions for a variable speed compressor can be very problematic as the required number of tests are significantly increased as a consequence of including the speed in the study.

To solve the problems of the previously explained procedure other methods have appeared which calculates the OCR non-invasively and in an online configuration (meaning continuously and not having to extract samples). These methods are based in thermal conductivity (Wujek et al., 2007), dielectric constant (Hwang et al., 2008), high speed cameras (Xu and Hrnjak, 2017), UV absorption (Kutsuna et al., 1991), speed of sound (Lebreton et al., 2001) and refraction (Fukuta et al., 2006) among others. However, these techniques require high end and experimental equipment with specific calibrations for each test bench, refrigerant and lubricant.

Regarding existing OCR experimental data in the literature, most of them only provides OCR data for the validation for OCR estimations methods. Therefore, these existing tests do not provide information about how OCR varies with the working condition. Other studies focus on fixed speed compressors and only measure a limited amount points as (Navarro et al., 2005) which tested 5 propane single speed compressors at 3 different conditions each. As other example, Wujek (Wujek et al., 2014) focuses on the effect of the chosen oil in OCR, testing 6 different oils in 4 different conditions (all at the same speed).

Regarding the studies of OCR in variable speed compressors also some studies can be found in the literature. In (Sarntichartsak et al., 2006) two variable speed commercial unit (rotary compressor) in the working range of

30-50Hz using R22/MO and R407C/POE/MO are tested, the behaviour with the speed of the compressor and the charge and type of the oil are analysed. However, the compressor speed range is limited, and the tests are not carried out in a calorimeter so the evaporating and condensing temperature are not controlled. Yoon (Yoon et al., 2011) study a variable speed high shell pressure scroll in a speed range of 30-120Hz working with R410A/PVE. A refractive index sensor was used to measure the OCR and a wide test matrix was proposed in which the influence of speed, superheat and suction pressure on OCR could be retrieved with high accuracy.

As reviewed experimental results of OCR in variable speed scroll compressors are limited and are carried out using HFCs. However, most of them have a high global warming potential and are phased out being substituted by new alternatives more environmentally friendly among which hydrocarbons as propane (R290) stands out. However, propane presents a higher solubility in POE oils than HFCs (Gas-Servei, n.d.) what makes that compressor providers recommend using high viscosity lubricants as POE ISO 68 which inturn makes it difficult to extrapolate the results obtained with HFCs to propane systems.

This study analyses the OCR of a variable speed scroll compressor of 46 cm^3 and a capacity between 3.4-15.7 kW working with R290/POE as a function of the compressor speed. The compressor has been tested in more than 70 different points including evaporation temperatures from -30 ºC to 25ºC and compressor speeds from 30 Hz to 120 Hz. In order to be able to perform all these OCR measurements in a limited amount of time a methodology based on mass flow discrepancies has been implemented. The developed methodology has demonstrated enough accuracy to provide acceptable qualitative results. It should be pointed out that the methodology does not require any experimental set up modification and can be implemented just using the sensors included in the Standard EN13771 ("EN 13771," 2016) for compressor rating. With the obtained experimental results, in the second part of the study, the global oil trend in the tested variable speed compressor and how it affects the compressor performance is analysed.

2. Methodology

2.1 Test bench

The used test bench is the calorimetric test bench represented in Figure 1. It has been designed to satisfy the Standard EN13771 (“EN 13771,” 2016) for compressor rating. It is able to control Condensing and Evaporating temperatures, Subcooling (SC) and Superheat (SH). Additionally, the test bench comprises a set of PID controllers that can keep any operating condition stable within a range of 1kPa and 0.1ºC from its setting without manual adjustments.

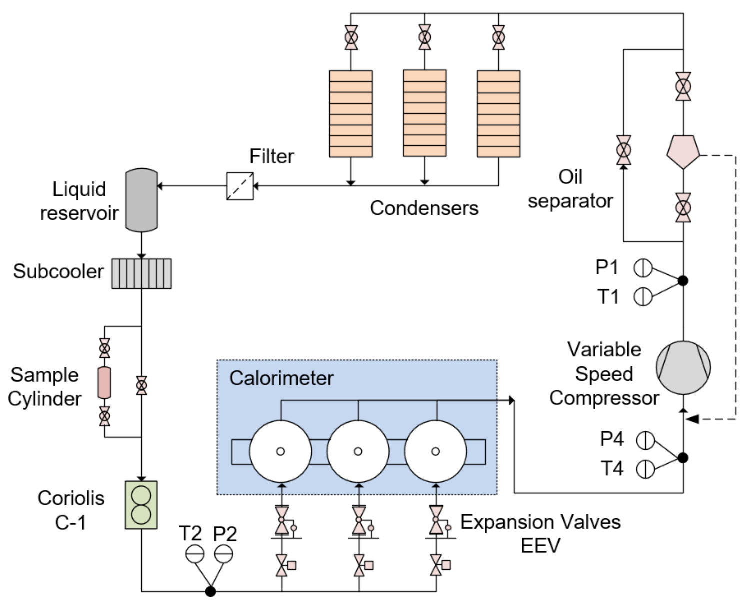


Figure 1: Calorimetric Test Bench

For this study the most important elements of the calorimetric test bench are: the evaporator, the oil separator and the Coriolis mass flow sensor. The evaporator consists of 3 isolated vessels in which the refrigerant exchange heat with an electric resistance. If the vessels are at constant temperature, the electrical power is measured and the heat loss to the ambient is modelled; then, the effective mass flow of refrigerant in the system can be calculated with Eq.(2) (this method is also called calorimetric method):

$$P_{electr} - P_{loss} = \dot{\boldsymbol{m}}_{\boldsymbol{refr}} \,(h_{out} - h_{in}) \qquad (2)$$

Regarding the oil separator, it has been used a Castel 5520/C oil separator at the discharge of the compressor which returns the discharged oil to the suction line of the compressor. Its efficiency has been experimentally tested in the test bench at different compressor speeds and it always showed efficiencies higher thatn 95%. This device can be bypassed so that the same condition can be tested with and without oil separator. And finally, the Coriolis mass flow sensor selected was a Micromotion sensor CMF025M which is placed in the liquid line between the subcooler and the expansion valves.

In addition to the mass flow meter, Pt100 and absolute pressure transducers have been installed all around the test bench to have an accurate estimation of the compression condition. The basic declared uncertainties of the used sensors are displayed in Table 1. Regarding derivate thermodynamic properties as enthalpies, they have been calculated using the REFPROP database (Lemmon and McLinden, 2009).

Uncertainty of the used sensors

Variable	Equipment	Uncertainty
T	RTD-PT	±0,05ºC
P	Fisher-Rosemount 3051	±0,02%
$\dot{m}$	Coriolis CMF025M	±0,025gs^{-1}
P_{electr}	Sineax CAM	±20W

Table 2: Uncertainty of Sensors

2.2 Description of the methodology to estimate the oil mass flow rate (Discrepancy Method)

According to Standard EN13771, as the mass flow is the main variable for characterizing a compressor, two different mass flow estimations have been simultaneously carried out to lower the uncertainty and to have a validation of the results. As it has been stated previously, the used methods in this study were: 1) a coriolis mass flow meter in the liquid line 2) an estimation of mass flow using an energy balance in the evaporator (calorimetric method).

However, when testing a compressor without oil separator, a homogeneous discrepancy between the results of both methods appears (specially at high speeds) in which the Coriolis mass flow meter always shows a higher value of the mass flow rate.

It has been assumed that this discrepancy is caused by oil circulation in the system which is registered by the Coriolis sensor but not in the heat balance performed in the evaporator. In that way, the measurement of the Coriolis provided the sum of the refrigerant massflow $\dot{\mathrm{m}}_{\mathrm{refr}}$ and the mass flow of the oil $\dot{\mathrm{m}}_{\mathrm{oil}}$. On the other hand, as the evaporating $\dot{\mathrm{m}}_{\mathrm{refr}}$ is the only massflow that has cooling capacity, the result of the energy balance in the evaporator only considered $\dot{\mathrm{m}}_{\mathrm{refr}}$.

Under these assumptions the oil mass flow in the system could be easily achievable subtracting both estimations as in Eq.(3).

$$\dot{m}_{\mathrm{coriolis}} - \dot{m}_{\mathrm{calorimeter}} = \left(\dot{\mathrm{m}}_{\mathrm{oil}} + \dot{\mathrm{m}}_{refr}\right) - \dot{\mathrm{m}}_{refr} = \dot{\mathbf{m}}_{\mathbf{oil}} \tag{3}$$

Note that this assumption is an approximation as part of the refrigerant will remain dissolved in the oil and will not evaporate in the evaporator (Youbi-Idrissi and Bonjour, 2008)(Youbi-idrissi et al., 2003). This is why the mass flow of refrigerant dissolved into the oil should be added to the result of Eq.(3). With this approach, the result of subtracting the estimation of both methods would be the ineffective mass flow: $\dot{\mathbf{m}}_{\mathbf{ineff}}$.

The OCR is a parameter typically used to estimate the impact of the oil circulation on the system performance. However, different systems with the same OCR can behave differently depending on the solubility of the refrigerant on the oil which depends also on temperature and pressure. Consequently, to add the effect of oil solubility to the OCR concept a redefinition can be carried out in which the ineffective mass flow is used instead of the oil mass flow as shown in eq. (4).

$$OCR_e = \frac{\dot{\mathrm{m}}_{\mathrm{ineff}}}{\dot{\mathrm{m}}_{\mathrm{total}}} \tag{4}$$

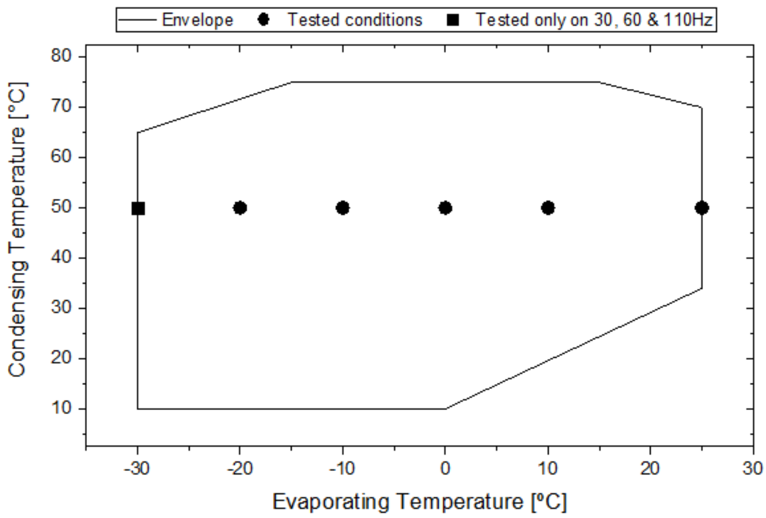
2.3 Experimental campaign

The used compressor is a scroll variable speed compressor with a speed range from 15 to 120 Hz, with a swept volume of 46cm^3 and working with propane. Regarding the lubricant oil, the compressor uses ISO 68 PolyOlester Oil (POE) which is accumulated in a carter located in the bottom part of the compressor at suction condition.

In order to validate the proposed discrepancy method a first experimental campaign was designed to compare the results with the weighting procedure described in the ASHRAE 41.4 standard (ASHRAE, 2015). A Mettler Toledo scale was used with a resolution of 0.01g and a Swagelok stainless steel sample cylinder which was placed in the liquid line as displayed in Figure 1.

This first experimental campaign consisted on 6 different tests in which the compressor was set to work in the same compression condition (Tevap=0ºC | Tcond=50ºC) but varying the working speed.

Once the methodology was validated, to study the compressor OCR as a function of the working conditions the experimental campaign shown in figure 2 has been carried out. It includes the variation of 3 parameters: speed, evaporation temperature and the existence of oil separator. The frequencies of 30, 45, 60, 85, 110 and 120 Hz have been tested at evaporation temperatures from -30 to 25ºC. Assuming that condensing temperature has a second order effect in OCR and in order to reduce the number of tests it has been decided to maintain the condensing temperature constant in this study.



Tested Speeds [Hz]
30
45
60
85
110
120

Figure 2: Envelope of the compressor and tested conditions

In total there are a set of 6 different conditions of evaporating temperature for each of the 6 chosen speeds and each condition is tested with and without oil separator. In total 66 different tests were carried out.

The methodology for each test was:

1. setting the compression condition and speed
2. letting it stabilize and then log for 15 minutes with oil separator
3. bypassing the oil separator
4. letting the system stabilize and then log for 15 minutes without oil separator.

It should be noted that the stabilization includes also the stabilization of the level of oil inside the compressor which was monitored using a sight tube. The results of each test are the average of the measured variables along the logging time.

3. Experimental results

3.1 Validation of the methodology

Figure 3 represents the OCR values obtained with both methodologies as a function of compressor speed. The errors are calculated with a 95% of confidence and considering the sensors and dataloggers uncertainty besides repeatability and random errors.

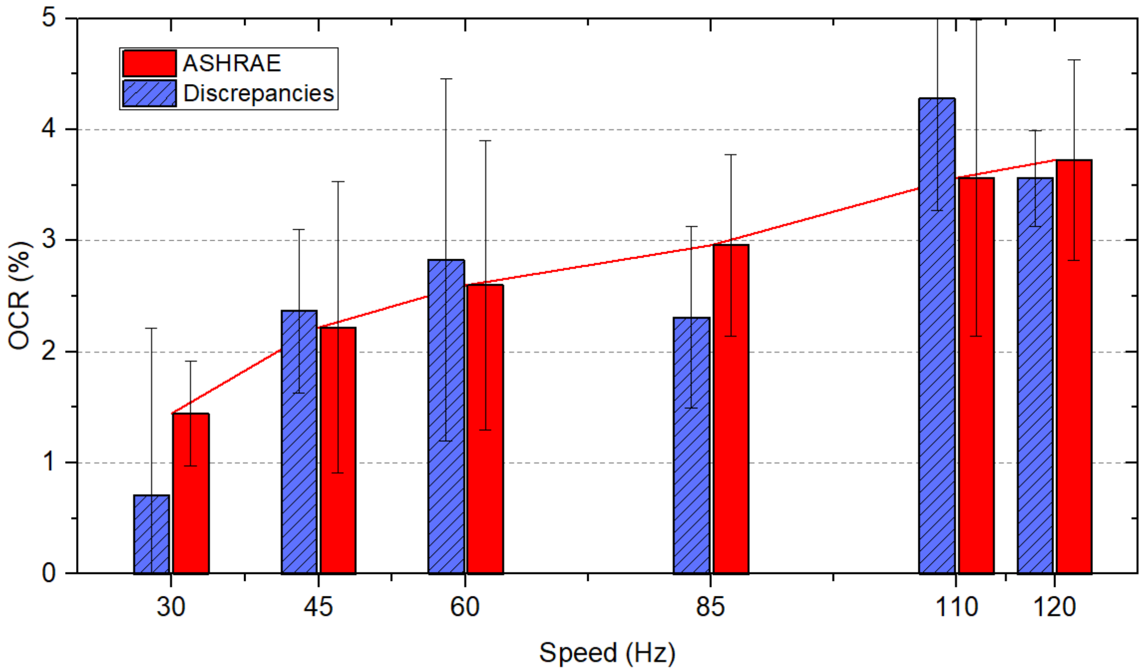


Figure 3: Comparison between OCR against speed for both methodologies ASRHARE and Discrepancies

The results show that despite the higher dispersion of the discrepancy method, both methodologies follow the same trend and there is an overlap in the error bars between both methodologies showing that they are compatible in the tested range.

3.2 Oil circulation results

Based on the previous results, the experimental campaign described in Figure 2 was carried out and the obtained results are described in this section.

Figure 4 shows the evolution of oil mass flow (4a and 4b) and OCR (4c and 4d) as a function of the total mass flow (measured with a coriolis sensor) with and without oil separator. The oil massflow has been calculated with Eq.3 and the OCR with Eq.1. The lines potted in the figures represent the trends shown by tests carried out with the same compressor speed.

Remark that, if the rest of the compression conditions remain constant, an increase of the evaporating temperature implies an increase of the total mass flow (as the suction specific volume increases with the evaporating temperature). Consequently, moving to the right in the X axis implies an increase of mass flow but also an increase of evaporating temperature for each speed.

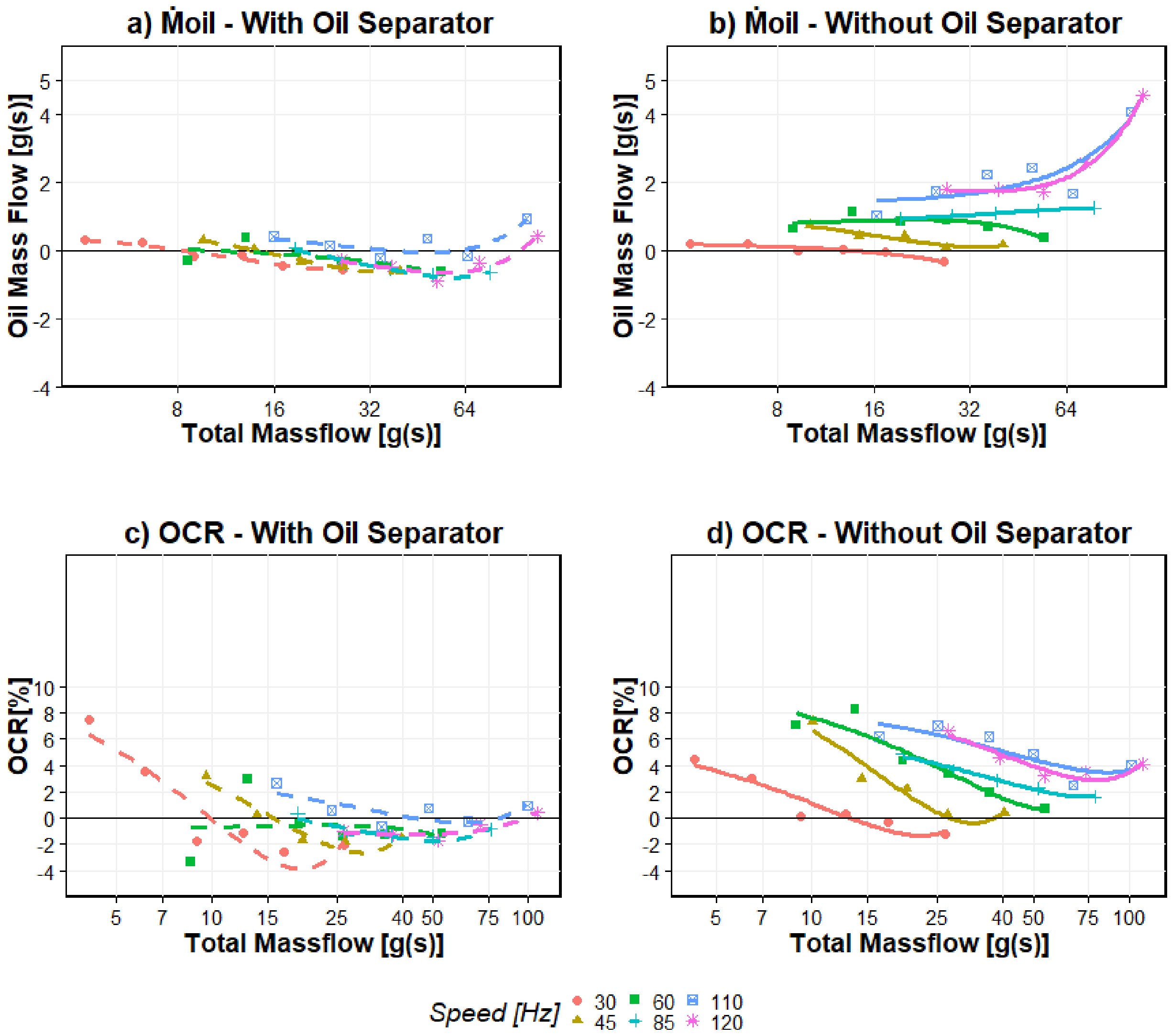


Figure 4: Evolution of $\dot{m}_{oil}$ and OCR against total massflow with and without oil separator a) Results of $\dot{m}_{oil}$ with oil separator b) Results of $\dot{m}_{oil}$ without oil separator c) Results of OCR with oil separator d) Results of OCR without oil separtor

Regarding the oil separator, it keeps a low circulating mass flow of oil close to zero at all the speeds. Remark that at very low mass flows the error in the OCR calculation increases significantly, this fact must be taken into account interpreting the high OCR obtained at low mass flows. The negative values of oil mass flow must be viewed also in the frame of the uncertainty of the sensors and the followed methodology. Nevertheless, this fact does not avoid to extract qualitative conclusions from the data analysed.

With the information shown in figure 4b, it can be concluded that at low speeds the oil mass flow is reduced and tends to zero, even without using oil separator. At 60Hz, the amount of oil mass flow begins to increase significant obtaining oil flows around 1g/s and lastly, at high speeds (110Hz), the oil mass flow increases reaching values close to 4 g/s. This trend is in the same direction of the increase of the total mass flow with the compressor speed.

Analysing the OCR can supply some more information as it allows to normalize the oil mass flow values relatively to the total mass flow. Figure 4d shows an increase of OCR with the speed. Regarding the evaporating temperature, OCR values decrease as evaporating temperature increases. This behaviour is produced because the oil circulation depends mainly on the rotational speed of the shaft which is responsible of pumping up the oil from the carter to the main bearings and the compression chamber. Consequently, the effect of an increase in the evaporating temperature, keeping the speed constant, will increase the refrigerant mass flow but the oil mass flow won't be influenced so much by the evaporating temperature.

In Figure 5 the OCR has been plotted against the speed and each data series represent a different Evaporating temperature. In this plot the effect of evaporating temperature and speed on OCR can be visualized without taking into account the total mass flow, a variable that depends on the chosen compressor.

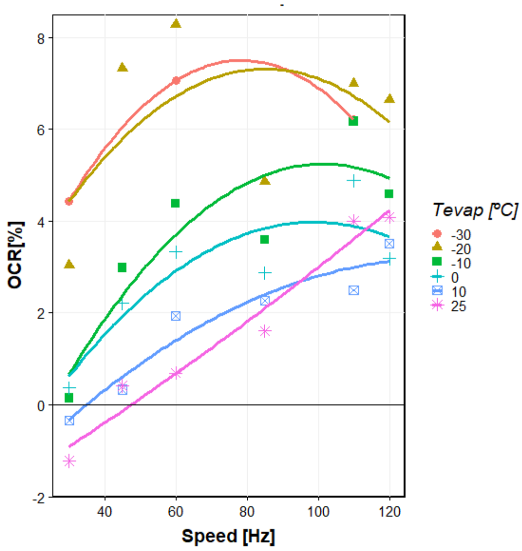


Figure 5: Effect of the Evaporating Temperature and Speed in OCR

In general there is an increase of OCR with the rotational speed of the compressor. This tendency is particularly clear at high evaporating temperatures in which the increase is homogeneous. On the other side, at low evaporating temperatures the tendencies split in two parts: the fist, characterized by a rapid increase of OCR with speed and a second in which the trend tends to change and the OCR starts to reduce. This local maximum could be explained by a saturation in the oil pump mechanism which in turn seems to get reached at lower speeds as the evaporating temperature decreases. Remark that the saturation affects the oil mass flow (not the OCR) and as the refrigerant mass flow keeps rising with speed the OCR starts decreasing. The authors also want to point out that a sharp decrease of OCR has been detected at 85Hz at different evaporating temperatures. The authors do not have strong evidences that could explain this behaviour, but some hypotheses could be a resonant effect at that speed, which is very close from the nominal speed (90Hz), or a set of outliers.

~~The OCR shows an increase with the speed until reaching a maximum in which the oil mass flow discharged by the compressor stops rising, probably because the pumping mechanism has reached a saturation state. This saturation speed seems to be affected by the evaporation temperature, the lower the evaporation temperature the earlier the saturation is reached. Regarding the evaporating temperature, as described in Figure 4d, OCR decreases with it reaching also a maximum at -20ºC in which the OCR saturates~~ reaches a maximum~~.~~

3.3 Oil mass flow correlation

From the experimental data and in other to be able to extrapolate the obtained results to other working conditions an empirical correlation of the oil circulation has been obtained.

Ribeiro and Barbosa (Ribeiro and Barbosa, 2016) suggested a semi-empirical model based on a centrifugal pump and a correction factor composed by 6 fitting coefficients depending on speed and pressure ratio. However, it needed geometric information of the interior of the compressor which is not declared by the manufacturer. This fact and the great number of coefficients makes the model difficult to be applied in our study.

The model used in this study only depends on the rotational speed of the compressor and follows the equation:

$$\frac{\dot{\mathrm{m}}_{\mathrm{oil}}}{\dot{\mathrm{m}}_{\mathrm{oil\_max}}} = \left(\frac{\mathrm{N}}{N_{max}}\right)^2 \quad (5)$$

~~In which the fitting coefficient C equals 2,517 when $\dot{\mathrm{m}}_{\mathrm{oil}}$ is expressed in g/s and N is the rotational speed, being N_{max} the maximum speed declared by the manufacturer. The obtained result for the coefficient is very close to the oil mass flow obtained for the maximum speed, so, for this compressor, it could be assumed that it follows like a "pump affinity law" in which the value of C is $\dot{\mathrm{m}}_{\mathrm{oil\_max}}$.~~

In which $\dot{\mathrm{m}}_{\mathrm{oil\_max}}$ is the fitting coefficient and equals 2,517 when $\dot{\mathrm{m}}_{\mathrm{oil}}$ is expressed in g/s and N is the rotational speed, being N_{max} the maximum speed declared by the manufacturer. Note the similarity of the proposed correlation with the affinity laws of pumps and fans.

In Figure 6, the result of applying the single-coefficient correlation is displayed. For each speed. The dots are the model predictions, the black and solid line represents the evolution of the correlation and the boxplot represents the experimental data for each speed.

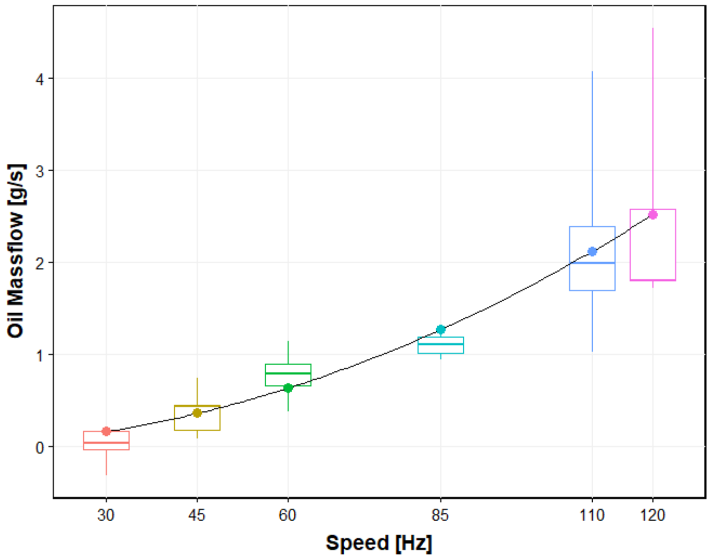


Figure 1: Evaluation of the model estimation of oil mass flow as a function of speed compared with the experimentally estimated values of oil mass flow.

In the graph it can be seen how the model fits properly the evolution of the oil mass flow. The result of the correlation provides a R-square value of 0.85 which is assumed to be acceptable considering the dispersion of the experimental data. Ribeiro and Barbosa (Ribeiro and Barbosa, 2016) considered also the pressure ratio in their model but in this study adding coefficients depending on the pressure ratio didn't improve the accuracy of the model. Consequently, only the coefficient depending on the quadratic term of the speed was considered which, in turn, allows to fit the model performing only one test. The authors consider that, even if only one compressor has been tested, this correlation would apply for other vertical scroll compressors as long as they implement the same oil pump mechanism.

4. Discussion

Once the results of the experimental campaign have been presented, in this section the effects of the oil in the compressor performance are analysed. The analysis will be focus mainly on compressor and volumetric efficiencies when the compressor works a 110 Hz. That condition has been selected because it is the condition in which the compressor is pumping a higher amount of oil and therefore where its effects become more significant.

When considering the compressor efficiency, also known as isentropic efficiency, AHRI Standard 571 and ASHRAE Standard 23.1 define it as:

$$\mu_c = \frac{\dot{m}\,(h_{2s}-h_1)}{P_{electr}} \tag{6}$$

In this equation $\dot{m}$ represents the refrigerant mass flow, however, it is common to consider the total mass flow (oil + refrigerant) for the value of mass flow.

Even if the increase of enthalpy on the oil is not useful for the cycle, the compressor is using some energy to increase its pressure and the temperature. Consequently, the compressor power can be extended with 2 new terms to consider the effects on the oil Eq.(7). This equation could help to compare data of the same compressor with different lubricants (Wujek et al., 2014).

$$\mu_c = \frac{\dot{m}_{ref}\,(h_{2s}-h_1)+m_{oil}c_p(T_{2is}-T_1)+m_{oil}\vartheta_{oil}(P_2-P_1)}{P_{electr}} \tag{7}$$

In conclusion, three different ways of calculating the compressor efficiency can be derived:

1) Assuming the total mass flow (refrigerant+oil) measured with a Coriolis sensor

2) Assuming only the refrigerant mass flow calculated with the calorimetric method and Eq.(2)

3) considering both the refrigerant and the lubricant terms according Eq.(7).

P_{electr} and the enthalpy difference are the same independently of the presence of the oil separator in the system, therefore the differences observed in the different ways of calculating the compressor efficiency will be related to the refrigerant mass flow and the oil term.

In Figure 7, the results of the different possible definitions of compressor efficiency are displayed for different evaporating temperatures and for 110Hz, which is considered high speed with a significant OCR of 5% average.

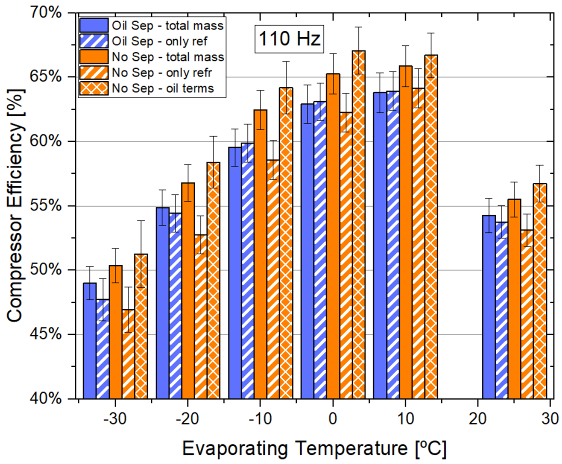


Figure 7: Compressor Efficiency calculated for different Evaporating Temperatures at 110Hz with different approaches

The solid bars represent the compressor efficiency assuming the mass flow measured with the coriolis, the stripped ones considering the mass flow obtained with the calorimeter and the rhomboid bar considering also the oil to calculate de efficiency [Eq. (7)]. Remark that the first two columns (blue) correspond to the test in which the oil separator was connected and they give very similar results. The point at -30 evaporating temperature shows a higher divergence but, in that point, the experimental installation presents higher errors. The results show a homogeneous increase of the compressor efficiency when the compressor is tested without oil separator and the Coriolis mass flow is considered (between solid bars), showing that the oil contribution to the total mass flow is significant. When the mass flow measured with the calorimeter is considered (stripped bars), a decrease of efficiency appears in most cases when no oil separator is used showing that part of the refrigerant remains solved in the oil. Regarding the oil terms of Eq.(7), the weight of the third term (the compression of oil) is negligible but the second one (the increase in temperature of oil) has an important contribution to the global efficiency, increasing the compressor efficiency up to 5%. This apparent increase in efficiency is actually a loss as it is power delivered to the oil instead of to the refrigerant. In case the amount of oil pumped by the compressor could be reduced this loss of efficiency would be reduced too.

Regarding volumetric efficiency, it is defined as the ratio between refrigerant mass flow pumped by the compressor and its swept volume:

$$\mu_v = \frac{\dot{m}\,\vartheta_{succ}}{displ\;N} \tag{8}$$

In this case, the same distinction about the consideration of the mass flow rate in the compressor efficiency can be done.

The compressor also displaces oil and this fact can be taken into account with a new term displayed in Eq.(9)

$$\mu_v = \frac{\dot{m}\,\vartheta_{succ} + \dot{m}_{oil}\vartheta_{oil}}{displ\;N} \tag{9}$$

Figure 8 represents the different results of volumetric efficiency for different evaporating temperatures and for 110Hz. The format used is the same as in Figure 7, the first 2 columns for each evaporation temperature represent tests with oil separator and the other 3 without it. In addition, the solid, stripped and rhomboid bars correspond to the different definitions of volumetric efficiency respectively: 1) considering the mass flow measured with the Coriolis sensor 2) with the calorimetric method 3) including also the oil term (eq. (9).

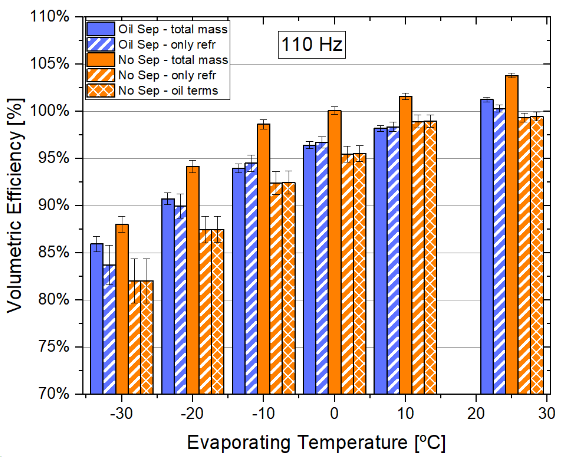


Figure 8: Compressor Efficiency calculated with different approaches for different Evaporating Temperatures at 110 Hz.

When using oil separator (the first two bars), the graph shows that the chosen mass flow estimation does not have a great effect on the result as the oil mass flow is maintained negligible. In contrast, when considering the results without oil separator, a great discrepancy up to 8% appears between the mass flow estimation methods (between the 3rd and 4th bar) which, in some cases, results in a volumetric efficiency higher than one when the Coriolis estimation is used. This variation is due to consider the oil mass flow as vapor refrigerant which have very different specific volume. Regarding the oil term in eq. (9), it does not contribute significantly to the volumetric efficiency (no sensible difference between 4th and 5th bar), this can be explained by the fact that the specific volume of the liquid oil is significantly smaller than the one of the vapor refrigerant. Additionally, another interesting result is that the volumetric efficiency considering the calorimetric mass flow tends to decrease when the oil separator is present (difference between stripped bars). Assuming that the compressor operates in a similar way in both situations, this values indicates that when the oil separator is not present some refrigerant can be maintained dissolved in the oil in the evaporator, not contributing to the refrigerant effect and therefore reducing the performance of the system, hence the introduction of an oil separator in these heat pumps, although increasing the total charge of the system could increase significantly the cooling capacity and the COP. This point also reveals that characterizing a compressor in a test bench with an oil separator could introduce a significant deviation in the estimation of the effective mass flow rate when this compressor is working in a real system not equipped with that device.

The results shown correspond to 110Hz which is considered “high speed” for this compressor but the obtained conclusions are similar for the rest of compressor speeds where noticeable values of OCR have been measured.

5. Conclusions

In this paper an analysis of the OCR in a scroll variable-speed compressor at different operating conditions and working with propane has been carried out. In order to do that it has been used a methodology based on the discrepancy between two mass flow estimation methods (calorimetric and Coriolis sensor). The developed methodology has demonstrated to supply enough accuracy to detect different oil behaviour in the system with the advantage of supplying an online and non-invasive compared to the ASHRAE standard.

Based on this technic the following conclusions related to oil behaviour in the compressor have been obtained:

- Measurements based on Coriolis and on calorimeter can diverge significantly if an oil separator is not present.
- OCR increases as evaporating temperature decreases.
- OCR increases with compressor speed. This is a consequence of the way in which compressor lubrication is performed in scroll compressors, and the OCR values can be significantly high at high speeds.
- A correlation of the oil mass flow with the square of the compressor speed has been found. The correlation allows a first estimation of oil mass flow requiring just one coefficient for vertical scroll compressors.
- At high OCRs, evaluating the compressor and volumetric efficiency with the Coriolis mass flow estimation can give values up to 5% higher than when the calorimeter procedure is used. This deviation can be very relevant when cooling capacity is estimated with a Coriolis mass flow meter.
- The existence of oil being heated in the compression process can decrease the real compressor efficiency up to 5%.
- The evaluation of volumetric efficiency has demonstrated, for the tested compressor, that a system without oil separator could have a reduction in the cooling capacity up to 2% at high speeds associated to the reduction of effective refrigerant mass flow caused by oil. This reduction can be explained by refrigerant which is compressed by the compressor but remains solved in the oil not supplying refrigerant effect.
- The OCR values obtained at high speeds and low evaporating temperatures are significantly high, up to 8%. And according to some estimations from literature (Lottin et al., 2003b), those OCR values could led to COP reductions up to 20%.

Finally, from all the results presented, it can be concluded that the deviations observed in the refrigerant mass flow rate depending on the employed measurement method could have an important influence in the estimation of the cooling capacity of variable speed scroll compressor working with propane. Hence, significant deviations between catalogue capacity values and the real capacity values supplied by a compressor working in real systems could be expected. To avoid this, more information about how the catalogue mass flow has been obtained and about the OCR evolution with the speed would reduce this uncertainty in the design of systems with these compressors.

Acknowledgment

The authors would like to acknowledge the Spanish "MINISTERIO DE ECONOMIA Y COMPETITIVIDAD", through the project ref- ENE2017-83665-C2-1-P "Maximización de la eficiencia y minimización del impacto ambiental de bombas de calor para la descarbonización de la calefacción/ACS en los edificios de consumo casi nulo" for the given support.

6. References

ASHRAE, 2015. Standard Method for Measuring the Proportion of Lubricant in Liquid Refrigerant. ASHRAE J. 2015.

Bivens, D.B., Yokozeki, A., 1994. Heat transfer coefficients and transport properties for alternative refrigerants. Proc. Int. Refrig. Air Cond. Conf. 299–304.

Branch, S.M., Lepak, D., 2019. OIL CIRCULATION IN A SCROLL COMPRESSOR. Patent.

EN 13771, 2016. . CEN.

Fukuta, M., Yanagisawa, T., Shimasaki, M., Ogi, Y., 2006. Real-time measurement of mixing ratio of refrigerant/refrigeration oil mixture. Int. J. Refrig. 29, 1058–1065. https://doi.org/10.1016/j.ijrefrig.2006.03.010

Gas-Servei, n.d. R-290 (Propane) Data-Sheet [WWW Document]. URL https://www.gas-servei.com/shop/docs/technical-data-sheet-r-290-gas-servei.pdf

Gungor, K.E., Winterton, R.H.S., 1986. A general correlation for flow boiling in tubes and annuli. Int. J. Heat Mass Transf. 29, 351–358. https://doi.org/10.1016/0017-9310(86)90205-X

Hwang, Y., Radermacher, R., Hirata, T., 2008. Oil mass fraction measurement of CO2/PAG mixture. Int. J. Refrig. https://doi.org/10.1016/j.ijrefrig.2007.05.011

Kim, H.J., Lancey, T.W., 2003. Numerical study on the lubrication oil distribution in a refrigeration rotary compressor. Int. J. Refrig. 26, 800–808. https://doi.org/10.1016/S0140-7007(03)00050-1

Kruse, H.H., Schroeder, M., 1985. Fundamentals of lubrication in refrigerating systems and heat pumps. Int. J. Refrig. 8, 347–355. https://doi.org/10.1016/0140-7007(85)90030-1

Kutsuna, K., Inoue, Y., Mizutani, T., Sudo, E., Araga, T., 1991. Real time oil concentration measurement in automotive air conditioning by ultraviolet light absorption. SAE Tech. Pap. https://doi.org/10.4271/910222

Lebreton, J.-M., Vuillame, L., Morvan, E., Lottin, O., 2001. Oil Concentration Measurement In Saturated Liquid Refrigerant Flowing Inside A Refrigeration Machine *, Int.J. Applied Thermodynamics.

Lemmon, E.W., McLinden, O.M., 2009. NIST Reference Fluid Thermodynamic and Transport Properties — REFPROP (student version). Database.

Lottin, O., Guillemet, P., Lebreton, J.M., 2003a. Effects of synthetic oil in a compression refrigeration system using R410A. Part II: Quality of heat transfer and pressure losses within the heat exchangers. Int. J. Refrig. 26, 783–794. https://doi.org/10.1016/S0140-7007(03)00065-3

Lottin, O., Guillemet, P., Lebreton, J.M., 2003b. Effects of synthetic oil in a compression refrigeration system using R410A. Part I: Modelling of the whole system and analysis of its response to an increase in the amount of circulating oil. Int. J. Refrig. 26, 772–782. https://doi.org/10.1016/S0140-7007(03)00064-1

Navarro, E., Urchueguía, J.F., Gonzálvez, J., Corberán, J.M., 2005. Test results of performance and oil circulation rate of commercial reciprocating compressors of different capacities working with propane (R290) as refrigerant. Int. J. Refrig. 28, 881–888. https://doi.org/10.1016/j.ijrefrig.2005.01.010

Qureshi, T.Q., Tassou, S.A., 1996. Variable-speed capacity control in refrigeration systems. AppliedThermalEngineering 16, 103–113.

Ribeiro, G.B., Barbosa, J.R., 2016. Analysis of a variable speed air conditioner considering the R-290/POE ISO 22 mixture effect. Appl. Therm. Eng. 108, 650–659. https://doi.org/10.1016/j.applthermaleng.2016.07.155

Sarntichartsak, P., Monyakul, V., Thepa, S., Nathakaranakule, A., 2006. Simulation and experimental evaluation of the effects of oil circulation in an inverter air conditioning system using R-22 and R-407C. Appl. Therm. Eng. 26, 1481–1491. https://doi.org/10.1016/j.applthermaleng.2006.01.002

Wujek, S.S., Bowers, C.D., Okarma, P., 2014. Effect of Lubricant-Refrigerant Mixture Properties on Compressor Efficiencies. Int. Compress. Eng. Conf. 1–9.

Wujek, S.S., Hrnjak, P.S., Seeton, C.J., 2007. Online measurement techniques for determining oil circulation rate. 2007 Proc. 5th Jt. ASME/JSME Fluids Eng. Summer Conf. FEDSM 2007 2 FORA, 181–186. https://doi.org/10.1115/FEDSM2007-37640

Xu, J., Hrnjak, P., 2017. Quantification of flow and retention of oil in compressor discharge pipe. Int. J. Refrig. 80, 252–263. https://doi.org/10.1016/j.ijrefrig.2017.05.004

Yoon, P., Kang, D., Kim, C., Ahn, S., Chung, B., Kim, B., Lee, J., Hwang, Y., 2011. An experimental study on oil discharge ratio at inverter-driven high shell pressure scroll compressor using R410A/PVE. Int. J. Refrig. 34, 105–112. https://doi.org/10.1016/j.ijrefrig.2010.08.019

Youbi-Idrissi, M., Bonjour, J., 2008. The effect of oil in refrigeration: Current research issues and critical review of thermodynamic aspects. Int. J. Refrig. https://doi.org/10.1016/j.ijrefrig.2007.09.006

Youbi-idrissi, M., Bonjour, J., Marvillet, C., Meunier, F., 2003. Impact of refrigerant – oil solubility on an evaporator performances working with R-407C ` ne-huile sur les Impact de la solubilite ´ vaporateur fonctionnant au R407C performances d ’ un e. Int. J. Refrig. 26, 284–292.