Impact of Lubricant in the Evaporator as a function of Oil Circulation Rate in Variable Speed Heat Pumps working with R290

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ABSTRACT

Modulating the speed of the compressors and adapting its capacity to the required load has led to a decrease in the annual energy consumption in many applications. However, in the compressor, having adequate lubrication at low speeds usually implies discharging too much lubricant at higher speeds. Although necessary for compressor operation, lubricating oil acts as a contaminant in the rest of the system. Consequently, manufacturers have to consider the oil circulation rate (OCR), as it limits the speed range of the compressor.

In a previous study, a compressor was tested over its speed range (30 to 110Hz), working with R290 and POE68 as a lubricant. Experimental data confirm the increase of OCR with speed and suggests that working with R290-POE68 could imply higher OCR values compared to other refrigerant-oil mixtures.

In this study, the impact of the OCR in the performance of a R290 heat pump evaporator was theoretically assessed. Two contributions were studied: The heat transfer coefficient (HTC) reduction in the evaporator and the effect of refrigerant being solved in oil. It was found that OCR values of 5% could decrease the coefficient of performance (COP) by 6%.

1. INTRODUCTION

Although necessary in the compressor, oil is considered a contaminant in the rest of the system. Oil changes the 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 lead to not having enough oil inside the compressor for correct lubrication, limiting its useful life. Consequently, it is essential 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:

$$OCR = \frac{\dot{m}_{oil}}{\dot{m}_{oil} + \dot{m}_{ref}} \tag{1}$$

Where $\dot{m}_{\mathrm{oil}}$ is the oil mass flow circulating in the system and $\dot{\mathrm{m}}_{refr}$ is the refrigerant mass flow.

In a previous study (Ossorio & Navarro-Peris, 2021), a variable speed scroll compressor working with R290 and POE68 was tested and the evolution of OCR and $\dot{m}_{\mathrm{oil}}$ was studied as a function of variables such as T_{cond}, T_{evap} and compressor speed. The results at high speeds shown OCR values higher than 5%, reaching peak values close to 7-8%. In that study it was also found a strong direct dependence of $\dot{m}_{\mathrm{oil}}$ with speed. This dependence was in accordance with the model of (Ribeiro & Barbosa, 2016), which considered the spinning shaft of the compressor as a centrifugal pump.

Lottin studied the effect of oil circulation in the system (Lottin et al., 2003) and concluded that a 5% of OCR could reduce COP more than 15%. Lottin followed a thermodynamic approach to quantify the effect of HTC, considering the refrigerant and oil as a zeotropic mixture and applying three different HTC correlations: Yam and Lin (Yam et al., 1997), Gungor and Winterton (Gungor & Winterton, 1986) and finally Bivens and Yokozeki (Bivens & Yokozeki, 1994). These results were obtained for HFCs, which have liquid densities higher than hydrocarbons. As OCR is given as a mass fraction, it is possible that, for the same oil flow, R290 systems will present higher values of OCR compared to the HFCs and consequently, Lottin's results can't be applied to hidrocarbons directly.

In his book Engineering Data Book III (John R Thome, 2010), Thome also reviewed in-depth the effect of oil in the evaporator HTC. Thome states that, at high vapor qualities, the HTC is sharply reduced due to the significant increase in local liquid viscosity due to high local concentrations of oil. Furthermore, he proposes a procedure to determine the actual HTC considering the oil. First, he uses the flow boiling model of Kattan, Thome and Favrat (Kattan et al., 1998) which is based on flow pattern maps and has been widely recommended in the literature. And then a correction factor is introduced which is dependent on the viscosity of the oil.

Another effect that has to be taken into account when analyzing the effect of oil in the system is the refrigerant solubility in oil. Refrigerant solved in the oil does not evaporate, and thus, it does not provide cooling capacity. The effect of the solubility in the evaporator was studied by (Youbi-idrissi et al., 2003; Youbi-Idrissi et al., 2004; Youbi-Idrissi & Bonjour, 2008). Youbi-Idrissi proposes a methodology to create pressure-enthalpy-quality diagrams for the combined mixture refrigerant-lubricant. This methodology considers that, as the evaporation of the refrigerant takes place, the fraction of oil in the liquid phase increases and, when it is significant, the mixture temperature starts increasing compared to the expected saturation temperature. This apparent superheat can affect the control on the thermostatic expansion valves and the enthalpy calculations based on temperature and pressure at the output of the evaporator. Different authors have used this methodology to different mixtures (Sun et al., 2021; Zhelezny et al., 2009).

In this contribution, the effect of oil in variable speed heat pumps working with R290 and POE68 will be studied. The expected impact on the performance of the obtained OCR measurements in (Ossorio & Navarro-Peris, 2021) will be analyzed. Two major contributions will be explored independently: HTC reduction and refrigerant solved in oil. For the HTC reduction, the flow boiling model of Kattan, Thome and Favrat (Kattan et al., 1998) based on flow pattern maps will be used. The effect of the oil in the model will be taken into account with the proposed correlation in Engineering Data Book III (Thome, 2010). Regarding the impact of solubility, the miscibility of R290 and POE68 has been characterized empirically and the methodology proposed by Youbi-Idrissi was followed to calculate the effect in the cooling capacity of the evaporator.

2. METHODOLOGY

As explained in the introduction, the HTC reduction and the solubility effect will be studied independently. Their contribution to lower the performance of the heat pump will be added afterward.

2.1 Impact of lubricant on HTC

A theoretical analysis has been carried out to estimate the impact of OCR in the global system. The reduction of the evaporating temperature due to a decrease of the HTC in the evaporator is assessed. It should be noted that the goal of this study is not to give an exact value but an approximate order of magnitude of the effect that could have the oil. Consequently, simplifications and assumptions will be made.

In this study, the procedure, correlations and correction factors proposed by Thome (Thome, 2010) will be followed to calculate the drop of the overall HTC due to the presence of oil. Once the difference in HTC has been estimated, the decrease of the evaporating temperature is calculated with the LMTD method using the following system of equations:

$$\begin{cases} Q_{evap} = U \cdot A \cdot LMTD \\ Q_{evap} = \dot{m}_{ref} \cdot (\mathrm{h_{evap,out}} - \mathrm{h_{evap,in}}) \\ Q_{evap} = \dot{m}_{air} \cdot c_{p_{air}} \cdot (T_{air,in} - T_{air,out}) \end{cases} \tag{2}$$

To characterize the evaporator, it has been considered to be designed with an inner tube diameter of 8.6mm and 2.5m^2 of total exchange area in the refrigerant part. The number of circuits are designed to ensure a vapor speed of 8.6m/s and the HTC of the refrigerant and air sides have been considered equivalent by design (U = 0,5 HTC). Typically the air part has a much lower HTC but it has been assumed that it was compensated by increasing the exchange area in the air part introducing fins.. Regarding the secondary fluid, a fan circulates 7000m^3/h of air at different $T_{air,in}$. For the tested range of temperatures, a constant cp of 1.012kJ/kg for the air has been considered. Additionally, no superheat (SH) has been considered at the evaporator's output and the inlet quality has been calculated as a function of T_{evap} considering no subcooling (SC) in the condenser.

Refrigerant mass flow is calculated assuming a displacement of 0,00506 m^3/s (46cm^3 at 110Hz), a volumetric efficiency following equation (4) and the inlet density is expressed as a function of T_{evap} assuming SH=0K.

$$\dot{m}_{ref} = displ \cdot \eta_v \cdot \rho_{suc}(T_{evap}) \tag{3}$$

$$\eta_v = 9.648e^{-1} + 2.522e^{-3} \cdot T_{evap} - 2.923e^{-5} \cdot T_{evap}^2 \tag{4}$$

The main unknows of the presented system are T_{evap}, Q_{evap}, and $T_{air,out}$. Consequently, the only parameter that remains undetermined is the heat transfer coefficient of the refrigerant. In this study The Kattan, Thome and Favrat correlation (Kattan et al., 1998) based on flow pattern maps will be employed. This correlation requires as inputs the refrigerant properties at the evaporation temperature and parameters of the heat exchanger as; tube inner diameter, heat flux[W/m^2] and mass velocity [Kg/m^{2}s]. With this information, the correlation provides a HTC for each local vapor quality. The resulting overall HTC will be assumed to be equivalent to the average HTC calculated along the complete evaporator. Note that the model inputs for the determination of HTC depend on the evaporating temperature, which, in turn, is an unknown of the system. Consequently, an iterative procedure is needed, which is described in **Figure 1**.

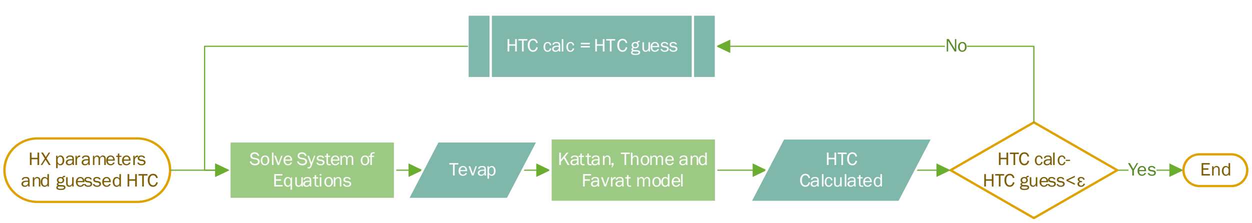

Figure 1: Diagram of the iterative procedure to obtain the Heat Transfer Coefficient and T_{evap}

For calculating the new evaporating temperature affected by the oil, the system is solved again but changing HTC. As mentioned, HTC is greatly affected by the presence of oil, especially at the end of the evaporator, where the local concentration of lubricant is higher. Thome in (Thome, 2010) proposes the following correction factor to quantify the reduction of HTC:

$$HTC_{ratio} = \frac{HTC_{ref+oil}}{HTC_{ref}} = \left[\frac{\mu_{refr}}{\mu_{oil}}\right]^{0.26 \cdot x_{oil}} \tag{5}$$

Being μ_{refr} and μ_{oil} the dynamic viscosity of the refrigerant and the oil respectively at the evaporating temperature. And being x_{oil} the local oil mass concentration in the liquid phase calculated with Eq.(6), being x_e the local vapor quality:

$$x_{oil} = 1 - x_{ref} = \frac{OCR}{1 - x_e} \tag{6}$$

Regarding the dynamic viscosity of the lubricant in Eq(7), it was estimated using the correlation of Guzman-Andrade:

$$\mu_{oil} = A \cdot e^{B/T} \cdot \rho_{oil} \tag{7}$$

Where T is the evaporating temperature in K, ρ_{oil} the lubricant density (987kg/m^3) and A=4.54e^{-4} and B=3730 being fitting coefficients calculated using the viscosity catalog data at 40ºC and 100 ºC.

2.2 Impact of lubricant solubility on the evaporator

The first step to analyze the impact of the solubility is to characterize the miscibility of refrigerant in oil. To do so, an experimental campaign was designed which provided the following results (with x_{ref} being the mass fraction of refrigerant in the liquid mixture):

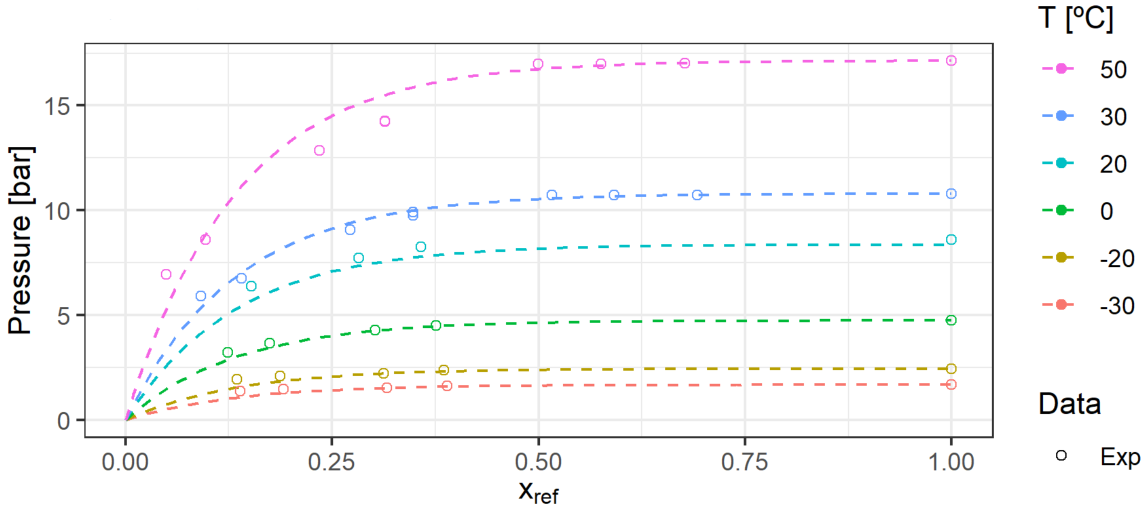

Figure 2: Liquid-Vapor-Equilibrium diagram of R290 and POE68

The experimental data was fitted using a new correlation based on the Weibull distribution, which appears to fit well even when reduced experimental data is available and there is no accurate value of molar weight to obtain the molar concentration. The equation used is the following:

$$P = P_{sat}(1 - e^{-k \cdot x_{ref}} + e^{-k}) \tag{8}$$

Being P_{sat} the saturation pressure of pure refrigerant at a specific T, x_{ref} the mass fraction of refrigerant in the liquid and k the only fitting coefficient that, for the studied mixture, equals 7.5. To be noted is that, by definition, mixture pressure tends to P_{sat} with pure refrigerant. Note also that the relation between x_{ref} and the vapor quality is given by Eq.(6).

The miscibility formula presented explains how pressure is affected by the lubricant. Regarding the effect of oil on enthalpy, the method proposed by (M. Youbi-Idrissi et al., 2004) has been applied. The method is based on the following equation to calculate the mixture enthalpy:

$$h_t = (1 - x_e - OCR) \cdot h_{l,r} + OCR \cdot h_{oil} + x_e \cdot h_{v,r} \tag{9}$$

where $h_{l,r}$ and $h_{v,r}$ are the specific enthalpy of pure refrigerant at the condition of liquid and vapor phase, respectively. And x_e is the vapor quality calculated as:

$$x_e = \frac{m_v}{m_{l,r} + m_{oil} + m_v} \tag{10}$$

In Eq.(9) h_{oil} represents the specific enthalpy of the oil, which is calculated as:

$$h_{oil} = h_o + \int_{T_o}^{T} C_{p,oil}\, dT \tag{11}$$

h_o and T_o are the reference enthalpy and temperature which have values of 200kJ/kg and 273.15K respectively. $C_{p,oil}$ is the specific heat capacity of the oil which is given as a function of temperature and density by the ecuation of Liley and Gambill cited in (Mermond et al., 1999).

$$C_{p,oil} = \frac{0.75529 + 0.0034\,T}{\sqrt{\rho_{oil}/998.5}} \tag{12}$$

Where ρ_{oil} is the density of oil at 15.6ºC, which has the value of 984.97 kg/m^3.

Once pressure and enthalpy have been calculated as a function of the vapor quality X, it is possible to:

- generate a P-h-X plot of the mixture.
- calculate the non-expanded quantity of refrigerant (NEQ) at a certain vapor quality:

$$NEQ = \frac{m_{l,r}}{m_{l,r} + m_v} = 1 - \left(\frac{x_e}{1 - OCR}\right)_{evap,outlet} \tag{13}$$

- calculate the apparent superheat SH*, with $T_{sat,pure}$ calculated at the evaporating pressure considering pure refrigerant

$$SH^* = T_{evap,out} - T_{sat,pure} \tag{14}$$

- calculate the enthalpy ratio $R_{h,}$ which represents the enthalpy difference reduction in the evaporator at a certain outlet temperature when there is oil circulating in the system.

$$R_h = \frac{\Delta h_{r,oil}}{\Delta h_{r,pure}} \quad with\ \Delta \mathrm{h_{r,oil}} = h_{evap,out}(T_{out}, P_{out}, OCR) - h_{evap,in}(T_{in}, P_{in}, OCR) \tag{15}$$

3. RESULTS

3.1 Impact of lubricant on HTC

The impact of oil in the HTC and in T_{evap} can be estimated solving the equation system (2) with the iterative procedure described in Figure 1.

The effect of oil in HTC and its evolution with the vapor quality is displayed in Figure 3. It is the result of applying the (Kattan et al., 1998) flow map model and the (Thome, 2007) correction for the oil presence. The input parameters are: an OCR of 5%, a tube inner diameter of 8.6mm, a heat flux of 4000 W/m^2 and 72 Kg/m^{2}s mass velocity.

In Figure 3, the black lines represent the limits between flow regimens, which are defined as a function of vapor quality and mass velocity. The horizontal blue line represents the mass velocity of the refrigerant during the process of evaporation. It determines the flow pattern that will appear in the different parts of the evaporator. And finally, the red lines represent the evolution of the HTC during the evaporation, being the dashed line the reduced HTC due to the oil presence. If it is assumed that the evolution of vapor quality is homogeneous along the evaporator length and that the input vapor quality is 0.35, the average HTC considering pure refrigerant and considering the influence of the lubricant would be 1.73 and 1.10 W/m2K respectively (a HTCratio of 0.63) for the example case.

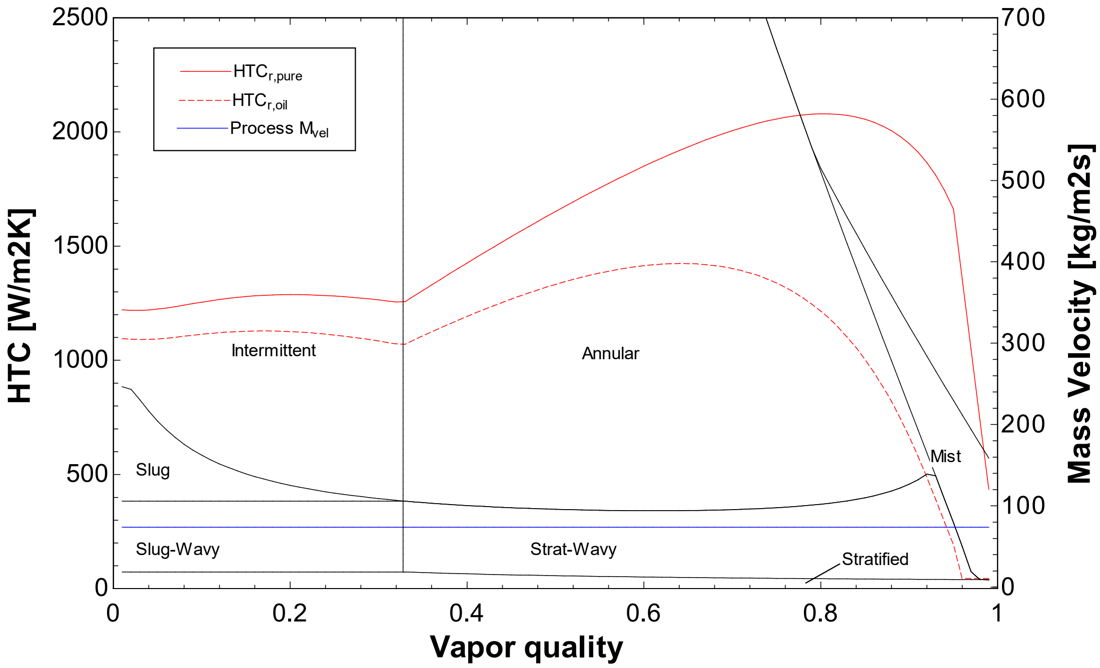

Figure 3: Evolution of heat transfer as a fuction of vapor quality

If we repeat this methodology for different $T_{air,in}$ and OCR values, the results displayed in Figure 4 are obtained. In the right figure, a slice of the left figure is presented, fixing $T_{air,in}$ =0ºC.

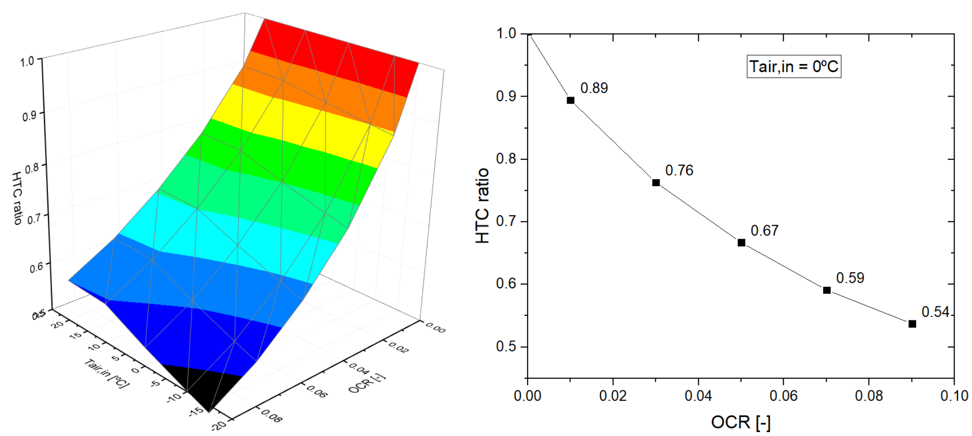

Figure 4: HTC ratio evolution with: OCR and $T_{air,in}$ (left), only OCR, assuming $T_{air,in}$ = 0ºC (right)

Figure 4 (left) shows that the HTC ratio depends much more on the OCR than on $T_{air,in}$. And Figure 4 (right) shows that HTC ratio decreases with OCR . At OCR levels of 5% the HTC ratio is 0.67.

The effect of oil in T_{evap} is studied in Figure 5:

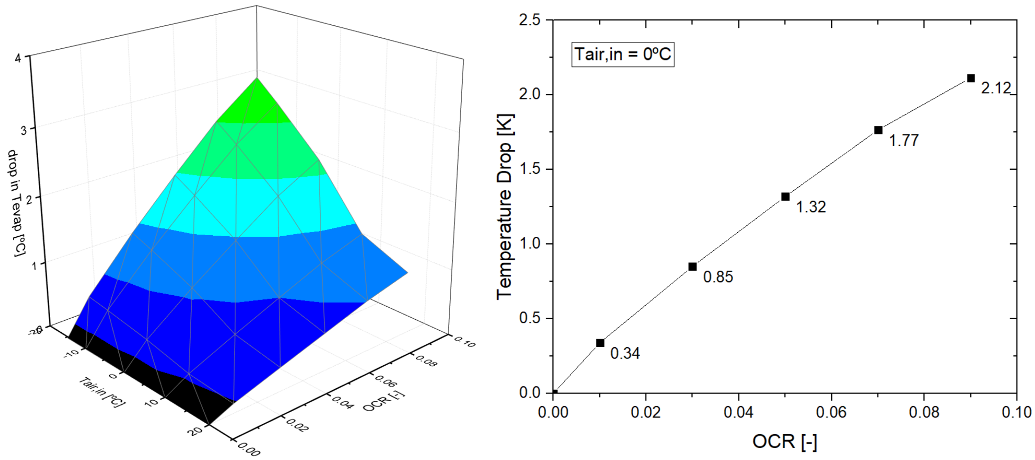

Figure 5: Evolution of the drop in T_{evap} with OCR and $T_{air,in}$ (left). Dependence of the drop in T_{evap} and the decrease of COP with of OCR at $T_{air,in}$ of 0ºC (right)

As expected, when HTC decreases T_{evap} decreases too. For OCR values of 5% the reduction in T_{evap} is 1.3K.

3.2 Impact of solubility on performance

Knowing how to calculate the effect of oil in pressure (8) and specific enthalpy (9) it is possible to create a P-h-X diagram of the refrigerant-oil mixture. The particular case of R290 and a 5% of POE68 is displayed in Figure 6:

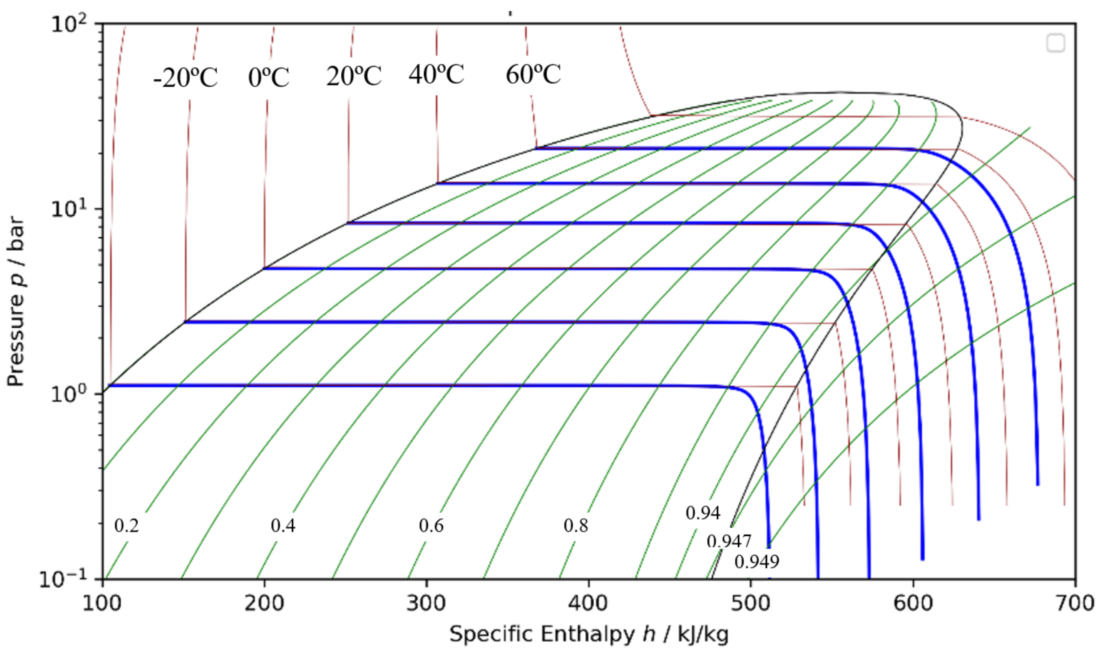

Figure 6: P-h-X diagram of R290 and POE68 (OCR=5%)

In the mixture diagram, the iso-quality lines extend out of the two-phase region of the pure refrigerant and, as the oil is considered to remain in the liquid state in the studied range, no qualities higher than 1-OCR are obtained. To be noted is also that there are two differentiated regions during the evaporation; one in which the mixture behaves as pure refrigerant (quasi isothermal evaporation), and one in which the sensible heat starts being significant and an apparent SH appears (temperature rise even if there is still refrigerant in liquid state. This apparent SH appears when the local liquid concentration of oil is high, at the end of the evaporator, with high vapor qualities. The relation between these variables (vapor quality, liquid concentration of oil and apparent SH) can be studied in Figure 7.

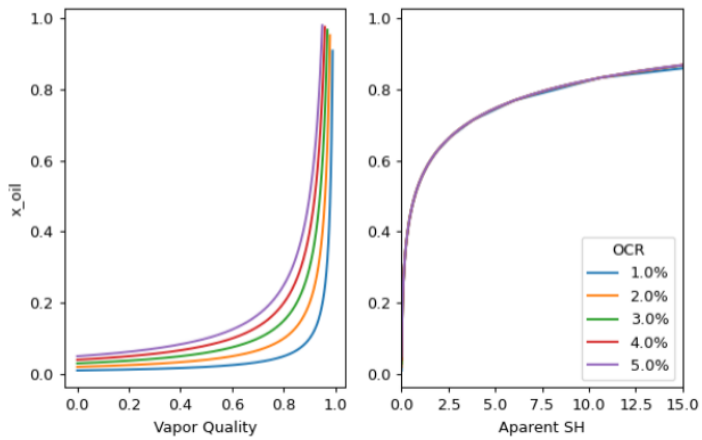

Figure 7: Relation between vapor quality, oil concentration in the liquid and apparent SH

This apparent SH will affect the control algorithm of thermostatic expansion valves and also, it will affect the energy balances in the evaporator. For example, looking at Figure 6, if we have a T_{evap} of 20 ºC and SH=1 at the outlet of the evaporator, an enthalpy close to 600 kJ/kg is expected (according to the pure refrigerant diagram). However, if the mixture is taken into account, the outlet enthalpy would be closer to 570 kJ/kg for 21ºC at the output of the evaporator. If the enthalpy at the inlet of the evaporator is 350 kJ/kg for both cases, the enthalpy difference ratio Rh would be 0.88. The effect of the apparent SH and OCR on the Rh is shown in Figure 8 (left).

Another way of studying the effect of having oil in the evaporator is considering that part of the refrigerant is solved in the oil and never evaporates, which lowers the cooling capacity in the evaporator. The more the apparent SH at the output of the evaporator the lower the not evaporated quantity (NEQ), this relation is shown in Figure 8 (right).

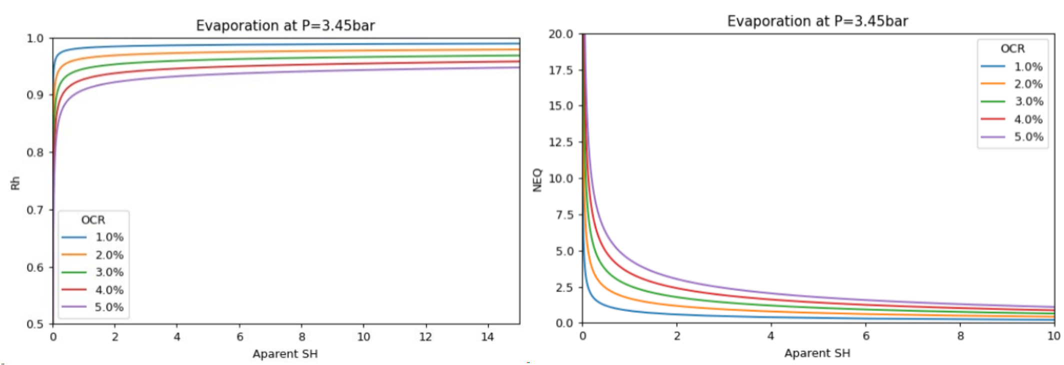

Figure 8: Evolution of Rh (left) and NEQ (right) with apparent SH for different OCR values

This evolution is greatly affected by the solubility of the refrigerant in oil. In this case, even with a low miscible oil we obtain Rh=0.943 and NEQ=1.11% at SH=10K and OCR of 5%. It should be noted that Rh and NEQ are manifestations of the same phenomena so, to study the effect of solubility in the cooling capacity, we should select one or the other.

The effect of Rh on the cooling capacity can be calculated with (16):

$$\frac{Q_{evap,oil}}{Q_{evap,pure}} = \frac{\dot{m}_t}{\dot{m}_r}\frac{\Delta h_{r,oil}}{\Delta h_{r,pure}} = \frac{\dot{m}_t}{\dot{m}_r} \cdot Rh \qquad with\ \frac{\dot{m}_t}{\dot{m}_r} = \frac{1}{1-OCR} \tag{16}$$

And the effect of NEQ with:

$$\frac{Q_{evap,oil}}{Q_{evap,pure}} = \frac{\dot{m}_{r,exp}}{\dot{m}_r}\frac{\Delta h_{r,pure}}{\Delta h_{r,pure}} = 1 - NEQ \tag{17}$$

If we substitute in both formulas the values obtained for SH=10K and OCR=5% a capacity ratio of 98.89% for NEQ and 91.97% for Rh are achieved. Which is translated into a reduction of cooling capacity of 1.11% and 0.72% respectively. (16)

6. CONCLUSIONS

In this study, the theoretical effect of oil presence on the performance of an evaporator was analized. Two effects were studied, the decrease of heat transfer and refrigerant being solved in the oil. The results are studied as a function of the amount of oil circulation in the system which, in scroll compressors working with R290, is superior compared to compressors working with HFCs. This could be explained as propane has lower density and the oil circulation ratio is mass based. In particular, with oil circulation of 5%, which is easily attainable in scroll compressors working at high speeds with R290, the following results arises:

- The decrease of HTC due to the presence of oil can reduce T_{evap} by 1.3K. This drop in T_{evap} can reach 2K at lower $T_{air,in}$ and it increases with higher oil circulations.
- The impact of the solubility of refrigerant in the selected oil can translate in a decrease of cooling capacity of 1.11% and 0.72%. In other situations, this impact can be more significant, especially when lower SH is chosen and when the miscibility of refrigerant and oil is higher.

If we add up the studied effects, a COP reduction higher than 5% is expected. This penalty is lower than the one calculated by Lottin. However, only the impact in the evaporator was studied, in a future study, the effect of oil in the compressor will be addressed too.

NOMENCLATURE

A	area	(m^2)
h	enthalpy	(KJ/kg)
displ	compressor displacement	(m3/s)
HTC	Heat Transfer Coefficient	($W/m^2/K$)
LMTD	Log Mean Temp. Difference	(K)
$\dot{m}$	mass flow	(g/s)
NEQ	Not Expanded Quantity	(%)
OCR	Oil Circulation Ratio	(-)
R_h	enthalpy ratio	(-)
SC	sub cooling	(K)
SH	super heat	(K)
U	global HTC	($W/m^2/K$)
x_e	vapor quality	(-)
x	mass concentration	(-)
η	efficiency	(-)
ρ	density	(kg/m3)
μ	viscosity	(cPs)

Subscript

air	air	o	reference
cond	condensing	oil	oil
evap	evaporator/evaporating	out	outlet
exp	expanded	sat	saturating
in	inlet	ref	refrigerant
l	liquid	v	vapor

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