International Journal of Refrigeration 145 (2023) 436–445

Contents lists available at ScienceDirect

International Journal of Refrigeration

journal homepage: www.elsevier.com/locate/ijrefrig

Impact of lubricant in the performance of variable speed heat pumps working with R290

Impact du lubrifiant sur les performances des pompes à chaleur à vitesse variable fonctionnant au R290

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A R T I C L E I N F O

Keywords:
OCR
Heat transfer coefficient
Solubility
Oil effect

Mots clés:
Débit de circulation de l'huile
Coefficient de transfert de chaleur
Solubilité
Effet de l'huile
Compresseur à spirale
Propane

A B S T R A C T

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 – 110 Hz), 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 was theoretically assessed. Three contributions were studied: The heat transfer coefficient (HTC) reduction in the evaporator, the effect of refrigerant being solved in oil and finally, and the impact of oil in the compressor. It was found that OCR values of 5% could decrease the coefficient of performance (COP) by more than 20%.

1. Introduction

Lubrication is necessary for most compressors for the lubrication of their moving parts. It also ensures a seal in the compression chamber and acts as a coolant, transferring heat from the bearings to the crankcase sump (ASHRAE, 2010). Different technologies arise to provide a sufficient amount of oil in the key spots. In scroll compressors, it is common to use a drill into the shaft that acts as a pump to lift the oil from the oil sump to the main bearing and the compression chamber, which is typically located 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, having an excess of oil discharged out of the compressor. 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 six correlation coefficients.

Although necessary in the compressor, oil is considered as 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). 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 $\dot{m}_{oil}$ is the oil mass flow circulating in the system and $\dot{m}_{refr}$ is the refrigerant mass flow.

In a previous study (Ossorio and Navarro-Peris, 2021), a variable speed scroll compressor working with R290 and POE68 was tested and

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https://doi.org/10.1016/j.ijrefrig.2022.10.003
Received 3 May 2022; Received in revised form 8 September 2022; Accepted 4 October 2022
Available online 7 October 2022

Nomenclature

A	area (m^2)
h	enthalpy (kJ kg^{-1})
displ	compressor displacement (m^3 s^{-1})
HTC	Heat Transfer Coefficient (W m^{-1} K^{-1})
LMTD	Log Mean Temp. Diff. (K)
$\dot{m}$	mass flow (g s^{-1})
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^{-1})
x_e	vapor quality (-)
x	mass concentration (-)
η	efficiency (-)
ρ	density (kg m^{-3})
μ	viscosity (cPs)

Subscript

1	suction
2	discharge
air	air
cond	condensing
evap	evaporator/evaporating
exp	expanded
in	inlet
l	liquid
o	reference
out	outlet
pure	pure
s	isentropic
sat	saturating
ref	refrigerant
v	vapor
vap	vaporization

the evolution of OCR and $\dot{m}_{oil}$ 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 $\dot{m}_{oil}$ with speed. This dependence was in accordance with the model of Ribeiro and Barbosa (2016), which considered the spinning shaft of the compressor as a centrifugal pump.

Lottin studied the effect of oil circulation in the heat pump (Lottin et al., 2003), he 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 et al. (1997); Gungor and Winterton (1986) and finally Bivens and Yokozeki (1994). 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 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 analysing 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 solubility in the evaporator was studied by Youbi-idrissi et al. (2003); Youbi-Idrissi et al. (2004); Youbi-Idrissi and 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 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).

Apart from the effect on the evaporator, Lottin et al. (2003) also took into account an increase in compressor consumption due to oil. If both contributions are taken into account, the unit's performance can decrease significantly. In particular, with OCR values of 5%, the coefficient of performance (COP) can be reduced by more than 15%. These results though, were obtained for hydrofluorocarbons (HFC)s, which have liquid densities higher than hydrocarbons such as R290. 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 hydrocarbons directly.

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 and Navarro-Peris (2021) will be analysed. Three significant contributions will be explored independently: HTC reduction, refrigerant solved in oil and oil being heated in the compressor. For the HTC, the flow boiling model of 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 solubility of R290 in POE68, it has been characterized empirically and the methodology proposed by Youbi-Idrissi was followed to calculate the effect in the cooling capacity of the evaporator. And finally, the effect of heating the oil in the compressor will be analysed and compared with the values obtained by Lottin et al. (2003). The increase of pressure drop at the end of the evaporator and in the liquid line due to the presence of oil has not been considered in this study.

2. Methodology

As explained in the introduction, the HTC reduction, the solubility effect, and the effect of the oil in the compression will be studied independently. Their contribution to lower the performance 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.

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

In this study, the procedure, correlations and correction factors proposed by 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

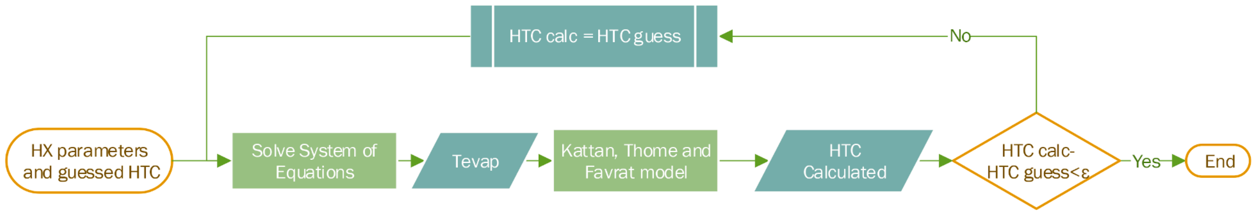

Fig. 1. Diagram of the iterative procedure to obtain the heat transfer coefficient and T_{evap}.

been estimated, the decrease of the evaporating temperature is calculated with the LMTD method using the system of equations (2).

To exemplify this methodology, it will be applied to a virtual heat pump which presents a cooling capacity close to 10 kW at air temperatures of 0 °C. The evaporator of the heat pump has been considered to be designed with an inner tube diameter of 8.6 mm and 2.5 m^2 of total exchange area in the refrigerant part. The number of circuits are designed to ensure a vapor speed of 8.6 m s^{-1} 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 7000 m^3 h^{-1} of air at different $T_{air,in}$. For the tested range of temperatures, a constant cp of 1.012 kJ kg^{-1} 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 and an isenthalpic expansion. All the thermodynamic properties have been calculated using Refprop10 (Lemmon and McLinden, 2002) Equation system (2).

Refrigerant mass flow is calculated with Eq. (3) assuming a displacement of 0.00506 m^3 s^{-1} (46 cm^3 at 110 Hz), a volumetric efficiency following Eq. (4) and the inlet density is expressed as a function of T_{evap} assuming SH = 0 K.

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

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

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. It will be calculated using the Kattan, Thome and Favrat correlation (Kattan et al., 1998) based on flow pattern maps. This correlation requires as inputs the refrigerant properties at the evaporation temperature and parameters of the heat exchanger as; inner tube diameter, heat flux [W m^{-2}], and mass velocity [kg m^{-2} s^{-1}]. 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 Fig. 1.

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 (2010) proposes the following correction factor to quantify the reduction of HTC:

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

Being μ_{ref} 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. (5), 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 (987 kg m^{-3}) and A = 4.54e^{-4} and B = 3730 are the fitting coefficients calculated using the viscosity catalog data at 40 °C and 100 °C.

Once the iteration process has been solved, and the reduction in T_{evap} due to the decrease in HTC has been calculated, the potential COP reduction is calculated using the compressor model Eq. (8) proposed by Shao et al. (2004). The coefficients were fitted using experimental data and their value is displayed in Table 1:

$$W_c \ or \ \dot{m}_t = a_0 + a_1 T_{cond}^2 + a_2 T_{cond} + a_3 T_{cond} T_{evap} + a_4 T_{evap}^2 + a_5 T_{evap} \tag{8}$$

Being COP defined as:

$$COP = \frac{\dot{Q}_{evap}}{\dot{W}_c} = \frac{\dot{m} \ \Delta h_{evap}}{\dot{W}_c} \tag{9}$$

Where $\dot{W}_c$ and $\dot{m}$ are obtained with the compressor model (8) and Δh_{evap} is obtained at a certain T_{cond} and T_{evap}, assuming SC = SH = 0 K.

2.2. Impact of lubricant solubility on the evaporator

The first step to analyze the impact of the solubility is to characterize the solubility of refrigerant in oil. To do so, an experimental campaign was designed which provided the results of Fig. 2 (with x_{ref} being the mass fraction of refrigerant in the liquid mixture):

The experimental data were 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:

$$P = P_{sat}\left(1 - e^{-k \cdot x_{ref}} + e^{-k}\right) \tag{10}$$

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 and to zero when pure oil. Note also that the relation between x_{ref} and the vapor quality is given by Eq. (6).

The solubility formula presented explains how pressure is affected by the lubricant. Regarding the effect of oil on enthalpy, the method proposed by Youbi-Idrissi et al. (2004) has been applied. This method is based on a three terms equation as a function of vapor quality; the first term quantifies the enthalpy of the not evaporated refrigerant, the second the enthalpy of the oil and the third the enthalpy of the vaporized refrigerant:

$$h_t = (1 - x_e - OCR) \cdot h_{l,ref} + OCR \cdot h_{oil} + x_e \cdot h_{v,ref} \tag{11}$$

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

Table 1
Fitting parameters and R2 of the compressor models for consumption and mass flow.

	a_0	a_1	a_2	a_3	a_4	a_5	R^2
Wc	2.32E+03	3.23E-01	3.44E+01	6.91E-01	−3.70E-01	−1.07E+01	0.9984
$\dot{m}$	4.93E+01	−9.51E-05	−1.92E-02	−4.85E-05	1.75E-02	1.57E+00	0.9994

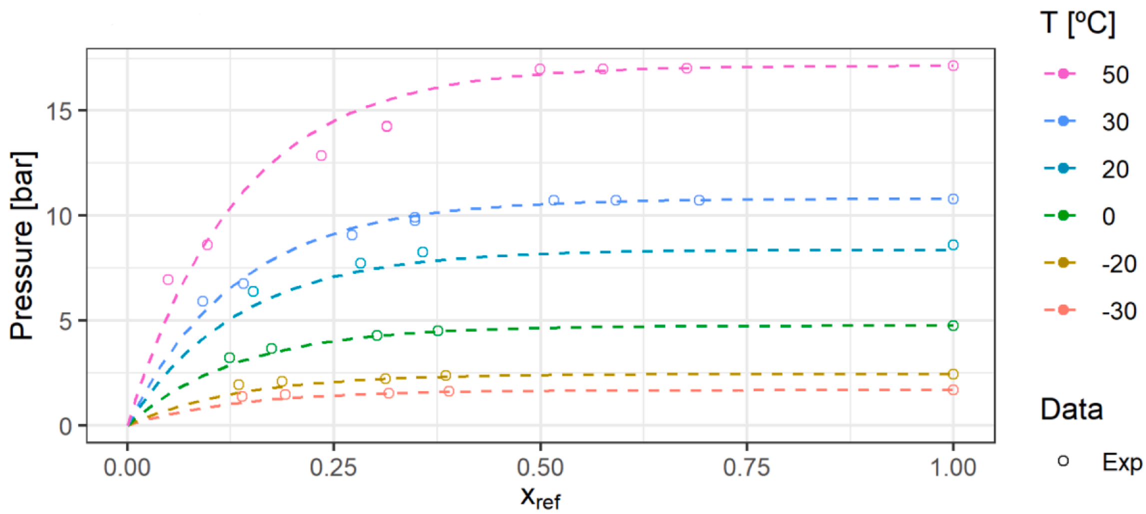

Fig. 2. Liquid-vapor-equilibrium diagram of R290 and POE68.

which is calculated as:

$$x_e = \frac{m_v}{m_{l,ref} + m_{oil} + m_v} \tag{12}$$

In Eq. (11) h_{oil} represents the specific enthalpy of the oil, which is calculated as:

$$h_{oil} = h_o + \int_{T_o}^{T} C_{p,oil}\, dT \tag{13}$$

h_o and T_o are the reference enthalpy and temperature which have values of 200 kJ kg^{-1} and 273.15 K 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):

$$C_{p,oil} = \frac{0.75529 + 0.0034\, T}{\sqrt{\rho_{oil}/998.5}} \tag{14}$$

Where ρ_{oil} is the density of oil at 15.6 °C, which has the value of 984.97 kg m^{-3} for the chosen POE.

Once pressure and enthalpy have been calculated as a function of the vapor quality x_e, 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:

$$NEQ = \frac{m_{l,ref}}{m_{l,ref} + m_v} = 1 - \left(\frac{x_e}{1 - OCR}\right)_{evap,outlet} \tag{15}$$

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

$$SH^* = T_{evap,out} - T_{sat,\ pure} \tag{16}$$

- Calculate the enthalpy ratio R_h, which represents the enthalpy difference reduction in the evaporator at a specific outlet temperature when oil is circulating in the system.

$$R_h = \frac{\Delta h_{ref,oil}}{\Delta h_{r,pure}} \text{ with } \Delta h_{r,oil} = h_{evap,out}(T_{out}, P_{out}, OCR) - h_{evap,in}(T_{in}, P_{in}, OCR) \tag{17}$$

2.3. Impact of lubricant in compressor

During the compression process, the oil in the compression chamber also gets compressed and heated, and additionally, in this process, part of the refrigerant that was solved vaporizes. Those processes require energy, which has to be supplied by the compressor. Consequently, if the oil is considered, the formula of the ideal compression work should be expanded with new terms:

$$W_c = \dot{m}_{ref}\,(h_{2s} - h_1) + \dot{m}_{oil} c_p (T_{2s} - T_1) + \dot{m}_{oil} \vartheta_{oil} (P_2 - P_1) + \dot{m}_{exp} \Delta h_{vap} \tag{18}$$

The first term represents the isentropic compression work of the refrigerant. The second and the third stand for the work needed to compress and heat the oil to the discharge conditions. Remark that as ϑ_{oil} is very low and can be easily neglected And finally, the last term is related with the vaporization of the refrigerant solved into the oil due to an increase of temperature during the compression.

3. Results

3.1. Impact of lubricant on HTC

An example of the effect of oil in HTC and its evolution with the vapor quality is displayed in Fig. 3. It is the result of applying the (Kattan et al., 1998) flow map model and the (Thome, 2007) correction for oil presence. The input parameters are: a $T_{evap}$of −10 °C, an OCR of 5%, a tube inner diameter of 8.6 mm, a heat flux of 4000 W m^{-2} and 72

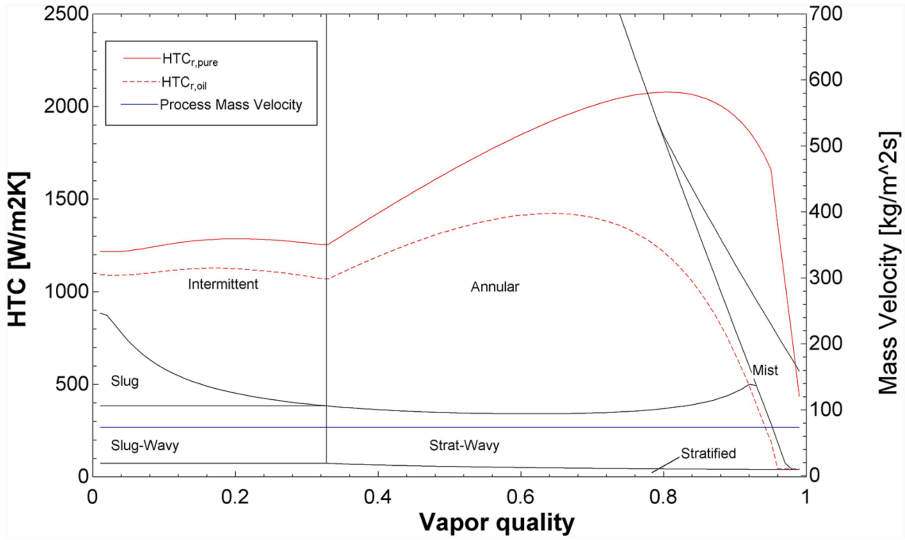

Fig. 3. Evolution of heat transfer as a function of vapor quality.

kg m^{-2} s^{-1} mass velocity.

In Fig. 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 m^{-2} K^{-1} respectively (a HTC_{ratio} of 0.63) for the example case.

The impact of oil in T_{evap} can be estimated solving the equation system (2) with the iterative procedure described in Fig. 1. If we repeat this methodology for different $T_{air,in}$ and OCR values, the results displayed in Fig. 4 are obtained. In the right figure, a slice of the left figure is presented, fixing a $T_{air,in} = 0$ °C.

Fig. 4 (left) shows that the HTC ratio depends much more on the OCR than on $T_{air,in}$. And Fig. 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 Fig. 5:

As expected, when OCR increases, HTC decreases and T_{evap} decreases too. For OCR values of 5% the expected reduction in T_{evap} is 1.3 K at $T_{air,in}$ of 0 °C. This decrease is more significant at higher OCR and lower $T_{air,in}$. With Eqs. (8) and (9) it is possible to convert this decrease in T_{evap} into a decrease of COP. This conversion is displayed in Fig. 6:

For OCR values of 5% the expected reduction in *COP* is 4.3% at $T_{air,in}$ of 0 °C. This decrease is more significant at higher OCR and lower $T_{air,in}$ reaching a peak value of 9.3%.

3.2. *Impact of solubility on performance*

Knowing how to calculate the effect of oil in pressure (10) and specific enthalpy (11) 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 Fig. 7.

In the new diagram, the isoquality 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. This

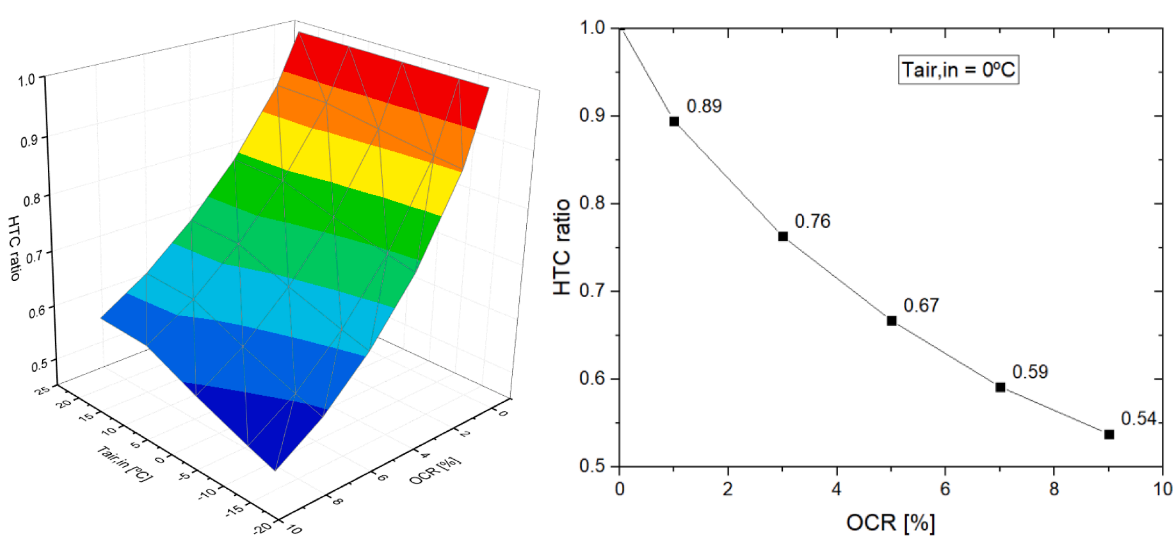

Fig. 4. HTC ratio evolution with: OCR and $T_{air,in}$ (left), HTC ratio evolution with OCR, assuming $T_{air,in} = 0$ °C (right).

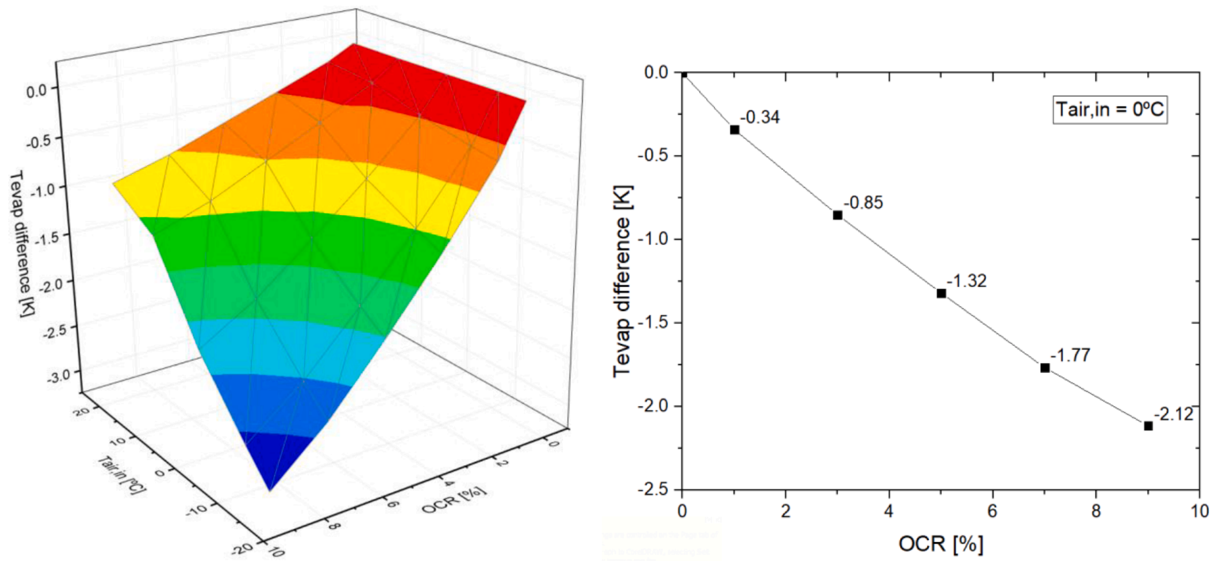

Fig. 5. Evolution of the drop in T_{evap} with OCR and $T_{air,in}$ (left). Dependence of the drop in T_{evap} and the decrease of COP with of OCR $T_{air,in} = 0$ °C (right).

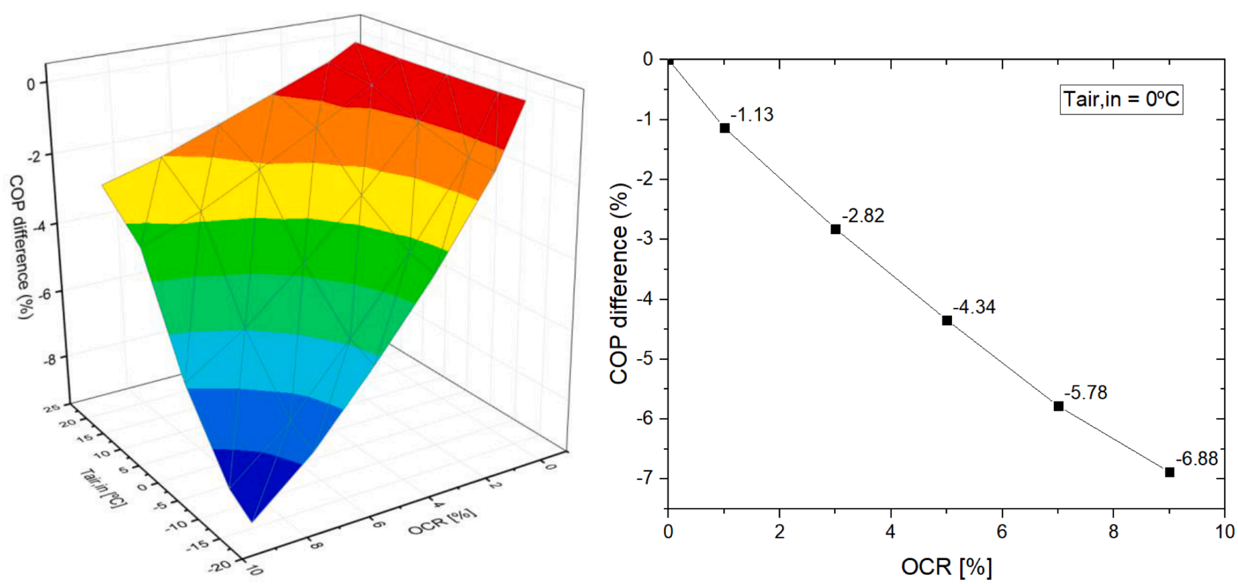

Fig. 6. Evolution of COP with OCR and $T_{air,in}$ (left). Dependence of COP reduction with OCR assuming $T_{air,in} = 0$ °C (right).

apparent superheat means a temperature rise even if there is still refrigerant in liquid state and occurs 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 Fig. 8.

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 Fig. 7, if we have $T_{evap} = -10$ °C and SH = 1 at the outlet of the evaporator, an enthalpy of 580 kJ kg^{-1} is expected (according to the pure refrigerant diagram). However, if an OCR of 5% is taken into account, the outlet enthalpy would be closer to 540 kJ kg^{-1} for 21 °C at the output of the evaporator. If the enthalpy at the inlet of the evaporator is 336 kJ kg^{-1} for both cases (assuming T_{cond} = 50 °C and SC = 0), the enthalpy difference ratio Rh would be 0.84 (applying Eq. (17)). The effect of the apparent SH and OCR on the Rh is shown in Fig. 9(up).

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 higher the apparent SH at the output of the evaporator the lower the not evaporated quantity (NEQ), this relation is shown in Fig. 9 (down).

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% for SH = 10 K 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 on the cooling capacity, we will focus on the effect of NEQ with the following equation:

$$\frac{Q_{evap,oil}}{Q_{evap,pure}} = \frac{\dot{m}_{ref,exp}}{\dot{m}_{ref}} \frac{\Delta h_{ref,pure}}{\Delta h_{ref,pure}} = 1 - NEQ \quad (19)$$

If we substitute in the formula the values obtained for SH = 10 K and OCR = 5%, a capacity ratio of 98.89% is achieved. Which is translated to a reduction in COP of 1.11%. In Fig. 10, the results for different OCR values are displayed, keeping the rest of the variables constant:

The graph shows an almost linear dependence between OCR and the COP reduction reaching a peak value of 2% for OCR values of 9%.

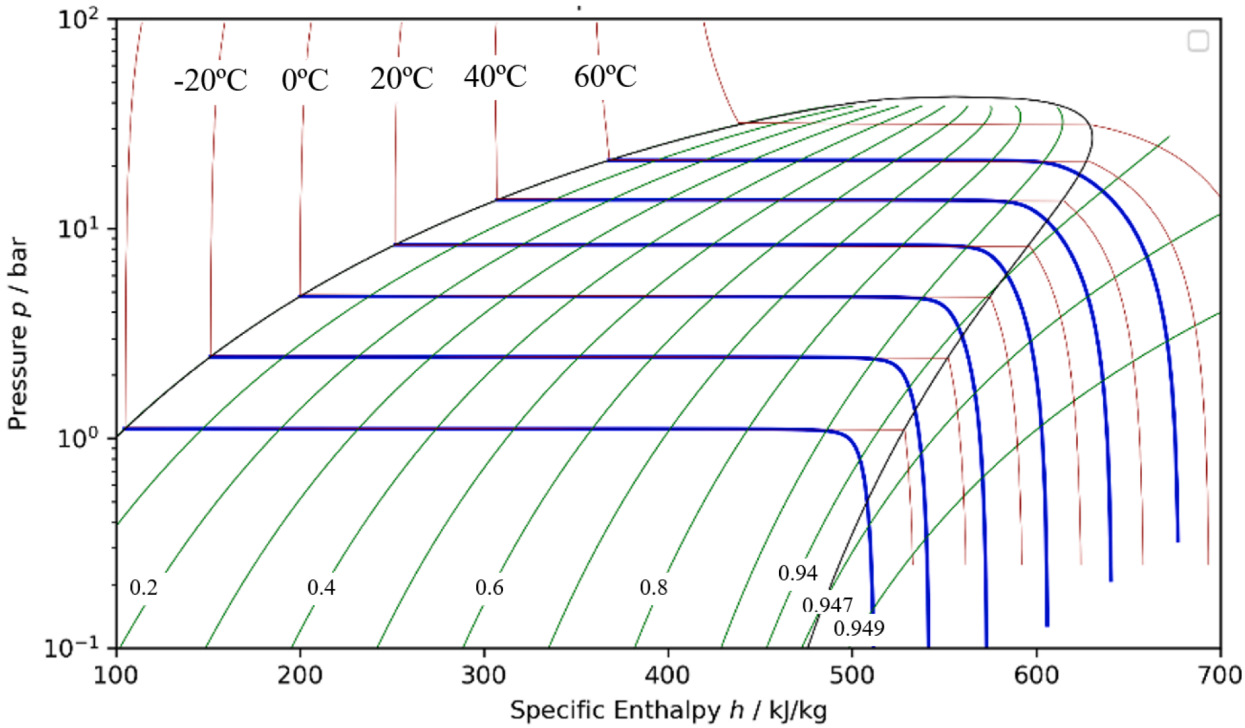

Fig. 7. P-h-X diagram of R290 and POE68 (OCR = 5%).

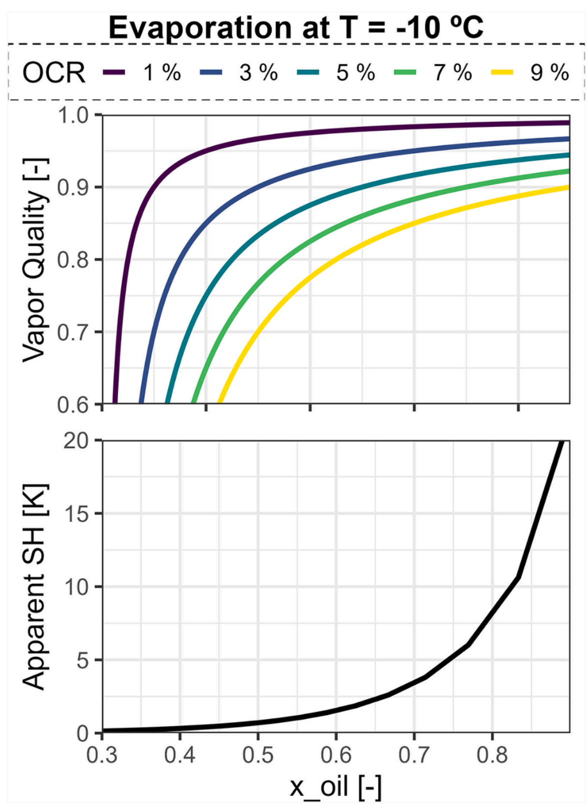

Fig. 8. Relation between vapor quality, oil concentration in the liquid and apparent SH.

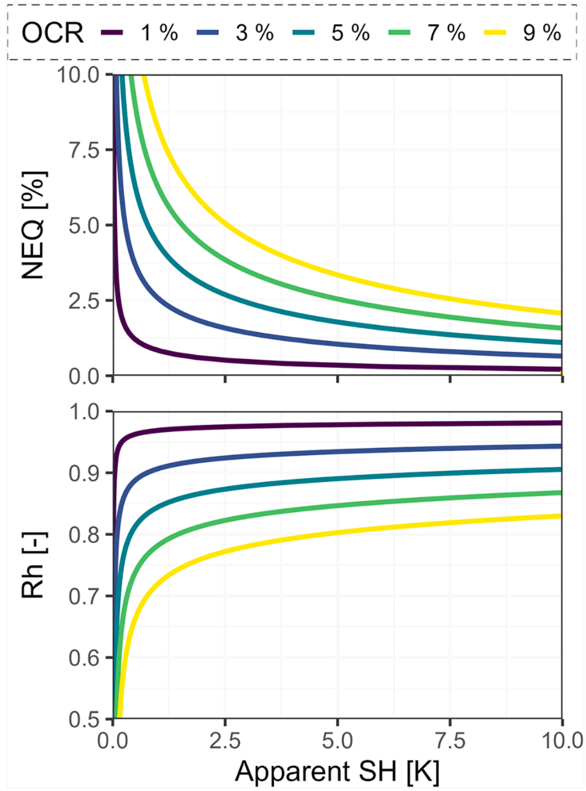

Fig. 9. Evolution of Rh (up) and NEQ (down) with apparent SH for different OCR values.

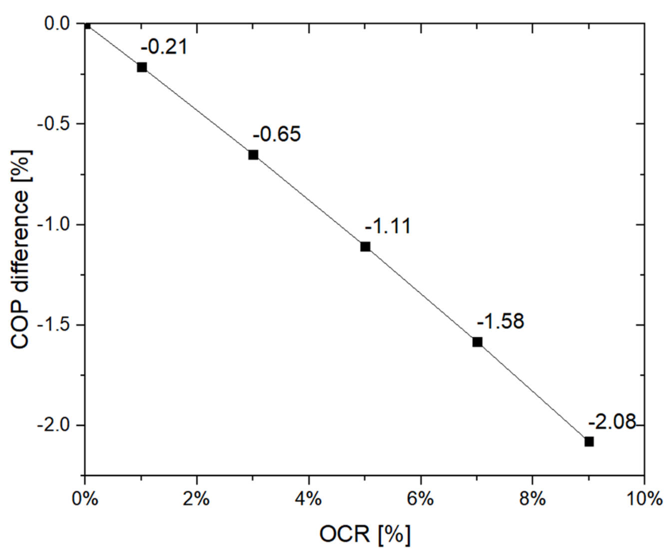

Fig. 10. COP reduction due to refrigerant solved in oil as a function of OCR and the methodology used. (No updated).

3.3. *Impact of oil in compressor consumption*

To exemplify the methodology, the impact of oil will be analysed for a representative compression condition of a tested variable speed compressor (46 cm^3) working with R290 and with 1200 ml of POE68 on it (Ossorio and Navarro-Peris, 2021). The compression condition was: N = 110 Hz, T_{evap}= −10 °C, T_{cond}= 50 °C and SH = 10 K. For that particular condition, an OCR of 5% was obtained experimentally.

For this analysis, it has been assumed $c_{p,oil} = 1738$ J kg^{-1} K^{-1}, $\rho_{oil} = 979$ kg m^{-3}, $\Delta h_{vap} = 580.3$ kJ kg^{-1} . Regarding $\dot{m}_{exp}$, to calculate the maximum effect of this term, it is considered that, at the discharge of the compressor, due to the increment of temperature, the solved refrigerant in the oil is close to zero and $\dot{m}_{exp} = \dot{m}_{ref} \cdot NEQ$ (with NEQ = 1.11% from last section).

$$W_c = \dot{m}_{ref}\ (h_{2s} - h_1) + \dot{m}_{oil} c_{p,oil} (T_{2is} - T_1) + \dot{m}_{oil} \vartheta_{oil} (P_2 - P_1) + \dot{m}_{exp} \Delta h_{vap}$$

The result are displayed in Fig. 11 and shows that, from the total consumption, the 86.4% is used to compress the refrigerant and, from the other 13.6% of the compressor power consumption, 6.6% is used to heat the oil and 7% to vaporize the solved refrigerant. The work used to compress the oil can be neglected as ϑ_{oil} is very low. Note that the compressor model presented is based in the theoretical calculations carried out by Lottin and it is highly dependent on the made assumptions and has not been validated by experimental data. Additionally, this

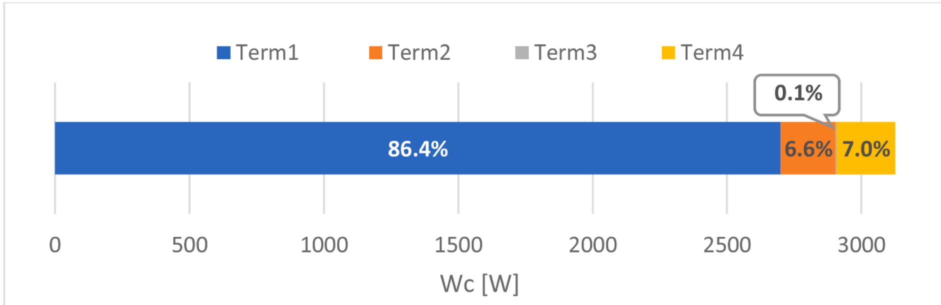

Fig. 11. Contribution to the total compressor consumption of the different terms of Eq. (18).

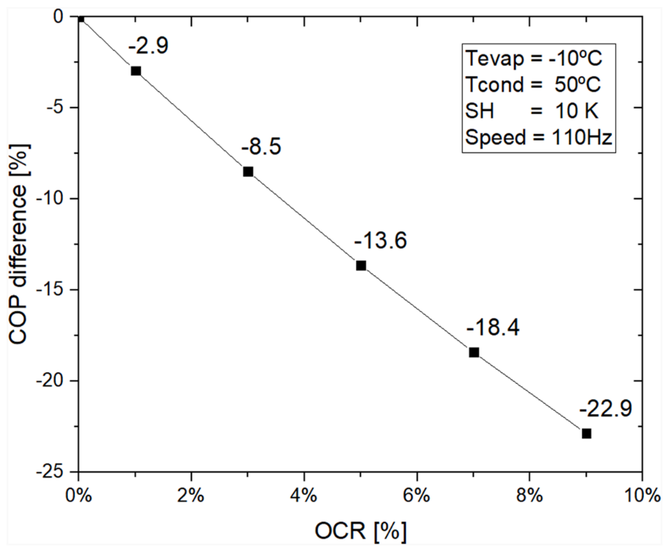

Fig. 12. Evolution of the COP reduction due to the heating of oil during compression for different OCR values.

impact cannot be avoided by introducing an oil separator and consequently it is not easy to study.

Regarding the impact on the performance for the studied case with OCR = 5%, oil increases the compression work by 15.8%, which translates to a decrease of COP of 13.6%. In Fig. 12, the results for different OCR values are displayed, keeping the rest of the variables constant:

The results show that the COP reduction can achieve values higher than 20% for OCR values of 9%.

4. Conclusions

In this study, the theoretical effect of oil presence on the performance of a heat pump is analyzed. Three effects were studied, the decrease of heat transfer in the evaporator, refrigerant being solved in oil and the effect of oil being heated during compression. The results are studied as a function of the amount of oil circulation rate in the system which, in scroll compressors working with R290, is higher than refrigeration systems 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 rate of 5%, value that, according previous works, has been found in scroll compressors working at high speeds with R290, the following results arises:

- The lowering of HTC translates into a COP decrease of 4.3% for a representative working condition. This impact could increase for higher OCR and lower evaporating temperatures.
- The impact of solubility in the evaporator in the evaluated conditions is in the range of 1%. In other situations, this impact can be more significant, especially when lower SH is selected and when the solubility of refrigerant and oil is higher.
- A decrease in COP higher than 5% is expected if we add up the impact of oil in the evaporator due to HTC and solubility.
- The highest losses due to the oil presence are found in the compressor, where a COP reduction close to 15% can be found. These losses appear as energy is wasted heating the lubricant during the compression.

If all contributions are supposed to be independent (to have an estimation of the maximum theoretical impact), a decrease of COP of around 20% could be reached for OCR values easily found at high compressor speeds. With this significant decrease in performance, a redesign of the pumping system of these compressors may be considered to reduce the oil circulation at high speeds and the use of oil separators is highly recommended to reduce the amount of oil reaching the heat exchangers. Additionally, the reduction of discharged oil could help reducing the refrigerant charge of the system, especially important with flammable refrigerant that have charge restrictions as R290.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgements

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. In addition, Ruben Ossorio would like to thank the Spanish government for his PhD scholarship with reference PRE2018–083535.

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