Numerical investigation of refrigerant outgassing in the screw pump of a hermetic reciprocating compressor oil supply system

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Abstract. This paper reports an investigation on the phenomenon of refrigerant outgassing in the screw pump of a hermetic reciprocating compressor oil supply system. The interfacial mass transfer between the refrigerant and the lubricating oil is modelled by means of a cavitation model based on the Rayleigh-Plesset equation, considering a fixed mass fraction of refrigerant dissolved in the lubricant. The influence of compressor speed and compressor crankcase pressure is evaluated by comparing the oil volumetric flow rate and the oil volume fraction field for each simulation. The results reveal that significant outgassing of refrigerant may take place in the oil pump, which can lead to noise generation.

1. Introduction

Lubrication plays a major role in assuring compressor reliability and energetic efficiency. In most reciprocating compressors, oil is essential to lubrication of bearings, but it also contributes to cooling components, sealing the piston-cylinder clearance, protecting against corrosion and reducing the equalizing pressure during the off-cycle. A properly designed oil supply system must provide a sufficient flow rate of oil to the bearings in a short time after the compressor startup, with acceptable mechanical losses. The fulfillment of these aspects requires the understanding of the complex two-phase flow of oil-refrigerant mixtures in the oil supply system.

The need for efficient compressor lubricating systems has intensified in the last decade, when variable capacity compressors climbed in popularity. This type of compressor regulates capacity by adjusting the compressor rotational speed, which can reach very low levels when thermal load is small. The threshold for lowering the rotational speed is mainly determined by the compressor ability to sustain sufficient oil supply to the compressor bearings.

In a typical hermetic reciprocating compressor, the oil is located in the sump at bottom of the compressor shell and pumped to the bearings using the rotation of the main shaft. The oil supply system, depicted in Figure 1, is comprised of a partially submersed oil pump that sucks the oil into the interior of the shaft, transporting it to the bearings via a helical groove (feeding channel) machined on the outer surface of the shaft. After leaving the helical groove, the oil is delivered to the piston-cylinder clearance and the compressor internal environment, returning afterwards to the sump. The two most common types of oil pumps are the centrifugal pump and the screw pump. The screw pump considered in this paper works similar to a screw extruder and it is formed by a static central pin and an external barrel that rotates with the shaft. The oil flow is driven upwards by viscous forces.

The interactions between the lubricating oil and the refrigerant gas is another important aspect to consider in the oil supply system design. In hermetic compressors the oil and the gas are in contact with each other, exchanging mass, momentum and heat. When dissolved in the oil, the gas alters the mixture properties, reducing its viscosity when compared to the pure oil, which is detrimental to the oil lubricity. The solubility of the gas in the oil depends on pressure and temperature, so when pressure decreases in the oil supply system the refrigerant comes out of solution, giving rise to bubbles that may lead to noise generation.

The first studies on hermetic compressors oil supply systems employed analytical approaches, modelling the oil flow via analogous electrical circuits. Kim *et al*. [1] used this analogy for the reciprocating compressor, while Kim and Lancey [2] applied their analytical model to a rolling piston compressor and found maximum deviation of 26% between measurements and the results obtained with the model.

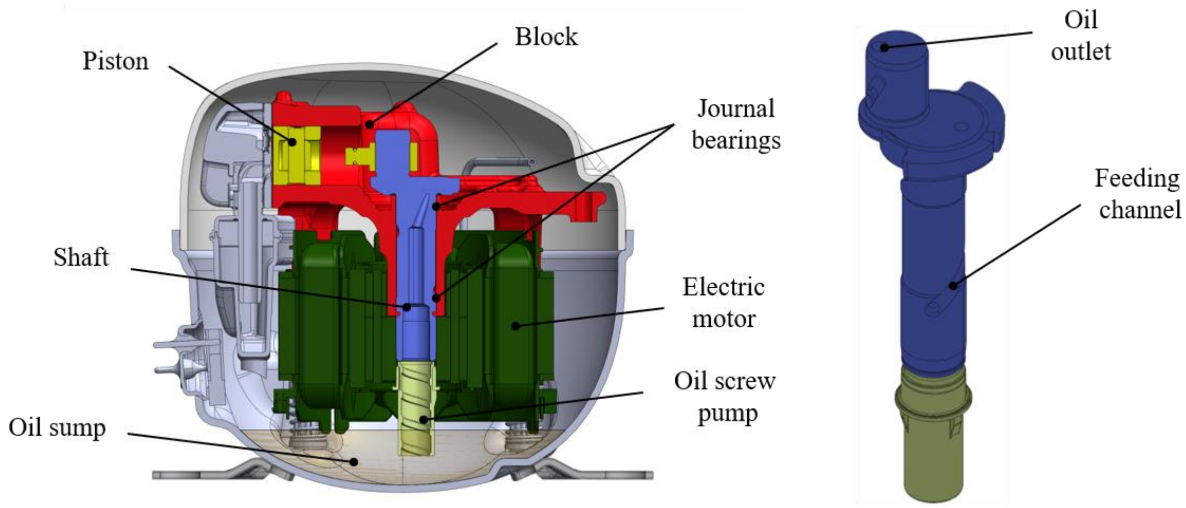

Figure 1. Illustration of a hermetic reciprocating compressor highlighting the oil supply system.

With the continuous advance of computational resources, several studies were able to simulate the oil supply system using CFD, representing a considerable improvement compared to analytical models, since the geometry can be precisely characterized. Lückmann *et al*. [3] developed a CFD model to simulate the oil supply system of a reciprocating compressor operating with a centrifugal oil pump. The proposed model considered isothermal and laminar flow and constant oil and gas properties. The two-phase flow was modelled with a homogeneous VOF model. The simulations considered the transient regime, and predicted the time required for the first drop of oil to reach the top of the oil supply system (defined as climbing time). The oil mass flow rate was also predicted and compared to experimental data.

Alves *et al*. [4] presented a semi-analytical model to predict the oil mass flow rate provided by a screw pump of a reciprocating compressor. The analysis considered pure oil with constant properties in the screw pump, and assumed laminar, two-dimensional, isothermal fully developed flow. The results for the oil mass flow rate obtained with the model were compared with experimental data, whereas the climbing time was compared with the data made available in Lückmann *et al*. [3].

Ozsipahi *et al*. [5] proposed a CFD model to simulate the oil supply system of a reciprocating compressor, and investigated the influence of several geometric parameters on the oil flow rate. The two-phase flow was modelled as isothermal, laminar and the oil and gas properties were assumed constant. The homogeneous VOF model was adopted to track the interface between the phases. Posch *et al*. [6] presented a split-up approach to simulate the oil supply system of a reciprocating compressor, separating the oil pump, the feeding channel and the upper part into different domains. Each domain was modeled with different approaches and the results of each domain were coupled to predict the oil mass flow rate provided by the system. The results were compared to experimental data, showing good agreement. The computational time was considerably reduced for simulations that considered design changes in the feeding channel and the upper part. However, if the design change was located in the oil pump, the split-up approach did not offer significative time saving.

Although several models have been presented to investigate the oil supply system of hermetic compressors, none of them have accounted for the interactions between the lubricating oil and the refrigerant gas. All the available models consider both fluids to be immiscible, with constant properties and no mass transfer (refrigerant absorption and outgassing). This paper considers the analysis of refrigerant outgassing in the screw pump of a hermetic reciprocating compressor oil supply system by using a CFD two-phase model. The interfacial mass transfer between the refrigerant and the lubricating oil is modelled by means of a cavitation model based on the Rayleigh-Plesset equation, considering the mass fraction of refrigerant dissolved in the lubricant oil constant, corresponding to the solubility of refrigerant in oil at the crankcase pressure and temperature. The effects of the compressor speed and compressor crankcase pressure are assessed by comparing the oil mass flow rate and the oil volumetric fraction field for each simulation.

2. Modelling

The CFD model presented in this paper considers only the screw oil pump of a hermetic reciprocating compressor. The software ANSYS Meshing v18.2 was employed to generate the mesh and ANSYS Fluent v18.2 was adopted for the flow simulations. The model considers the following assumptions: (i) isothermal laminar flow, (ii) oil and gas thermophysical properties are constant, (iii) the pressure at the oil pump outlet is equal to the reference pressure. The simulations employed isobutane (R600a) as refrigerant gas, with density ρ = 1.356 kg/m^3 and viscosity μ = 8.213 × 10^{-6} Pas, and lubricating oil ISO 5. The liquid phase was composed of lubricating oil and a fixed amount of dissolved refrigerant gas. The properties of the oil-gas mixture were calculated using oil manufacturer data [7], assuming a temperature of 55ºC and reference pressure of 62.8 kPa, resulting in ρ = 823.7 kg/m^3, μ = 2.11 × 10^{-3} Pa.s. The refrigerant mass fraction dissolved in the oil was approximately equal to 2.3%, calculated with the equation proposed by Seeton and Hrnjak [8].

Before simulating the two-phase flow, a single-phase simulation employing only oil was carried out. The resulting pressure and velocity fields of this simulation were used as initial conditions for the two-phase case. The Mixture Model approach was used to model the multiphase flow, as described in the ANSYS theory guide [9]. This approach presents two advantages when compared to the VOF model: (i) it allows interpenetration between the phases, and (ii) it applies the concept of slip velocity to allow the phases to move at different velocities. Additionally, the refrigerant outgassing was modelled with a cavitation model, in which case the VOF model is not recommended. The cavitation model developed by Zwart *et al*. [10] available in Fluent v18.2 was adopted. This model calculates the interphase mass transfer per unit volume based on the Rayleigh-Plesset equation as follows:

$$\begin{aligned} &\text{if } p \le p_{\mathrm{sat}} \\ &\qquad \dot{m}_{\mathrm{e}} = F_{\mathrm{vap}} \frac{3\alpha_{\mathrm{nuc}}(1-\alpha_{\mathrm{v}})\rho_{\mathrm{v}}}{R_{\mathrm{b}}} \left(\frac{2}{3}\frac{p_{\mathrm{sat}}-p}{\rho_{\mathrm{l}}}\right)^{1/2} \\ &\text{if } p > p_{\mathrm{sat}} \\ &\qquad \dot{m}_{c} = F_{\mathrm{cond}} \frac{3\alpha_{\mathrm{v}}\rho_{\mathrm{v}}}{R_{\mathrm{b}}} \left(\frac{2}{3}\frac{p-p_{\mathrm{sat}}}{\rho_{\mathrm{l}}}\right)^{1/2} \end{aligned} \tag{1}$$

For the nucleation site density (α_{nuc}) and the bubble radius (R_{b}) the default values were set, i.e., 5 ×10^{-4} and 1 μm, respectively. The interphase mass flow rate per unit volume ($\dot{m}_{\mathrm{e}}$ or $\dot{m}_{\mathrm{c}}$) is predicted as a function of the local static pressure (p). If p is lower than the saturation pressure of the mixture

(p_{sat}), the refrigerant is desorbed from the mixture in the form of bubbles; otherwise the refrigerant is absorbed into the mixture. The parameters F_{vap} and F_{cond} in equation (1) represent empirical calibration coefficients and their values and significance will be discussed in the next section. Finally, ρ_{v} and α_{v} are the gas density and gas volume fraction, respectively.

Figure 2 shows the computational domain and boundary conditions adopted in the simulations. Two gauge pressure boundary conditions were imposed: one at the pump inlet, where the total pressure was set at 25 Pa (the hydrostatic pressure in the oil column); and the other at the pump outlet, where the static pressure is set to zero. The domain outer wall rotates with a prescribed angular speed while the inner wall remains static, both walls comply with the no-slip condition. The simulations were solved with properly selected underrelaxation factors, a coupled pressure-velocity scheme, a second-order momentum scheme and a first-order scheme to discretize the volume fraction. The transient regime was modelled with a first-order implicit scheme.

After carrying out a mesh sensitivity test using the oil mass flow rate as the convergence criterion, a mesh with roughly 8×10^5 nodes was chosen for all simulations. Two convergence criteria were established: scaled absolute residual values lower than 10^{-4}, and oil mass imbalance lower than 1%. The cases were run in a computer with a i7-7700k processor and 64GB DDR-1333 RAM memory, using 5 cores parallel processing. The average wall clock time for the simulations was roughly 30 hours.

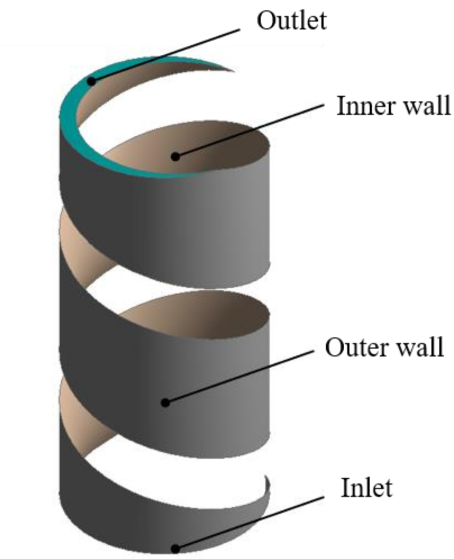

Figure 2. Computational domain with boundary conditions locations.

3. Results

The simulation model was employed to investigate the influence of several parameters on refrigerant outgassing. The empirical calibration coefficients F_{vap} and F_{cond} were the first parameters to be evaluated. Theses coefficients account for the time scale of each process and, since the vaporization process is generally faster than condensation because of bubble nucleation, F_{vap} must be greater than F_{cond}. In fact, the ANSYS Fluent v18.2 default values are $F_{\text{vap}} = 50$ and $F_{\text{cond}} = 0.01$, which were calibrated with water flow data. In the present case, because of a binary diffusion mass transfer resistance between R600a and the lubricant, refrigerant outgassing is assumed to be much slower than vapor cavitation. Thus, smaller values for F_{vap} and F_{cond} (in comparison with the default) have been adopted. Figure 3 shows results for the oil flow rate (Q) for different pairs of coefficients (F_{vap} and F_{cond}). As the values of the coefficients increase, the oil flow rate decreases, due to more refrigerant outgassing. When $F_{\text{vap}} = 50$ and $F_{\text{cond}} = 0.01$, the oil flow rate is 47% smaller than the case in which no cavitation model is employed (dashed line in Figure 3). The overall gas volume fraction, that is, the fraction of the domain occupied by gas, is displayed in Figure 4 as a function of the coefficients. As expected, an increase in the amount of gas in the domain is observed. This result can be verified in Figure 5, which displays the oil volume fraction field in the pump midsection for the cases with smaller (Figure 5a) and larger (Figure 5b) pairs of coefficients. In Figure 5b, the amount of free refrigerant gas is such that a stratified flow pattern in established, whereas in Figure 5a only a very small amount of refrigerant outgases, which accumulates in the lower pressure region located at the top of the screw cross section.

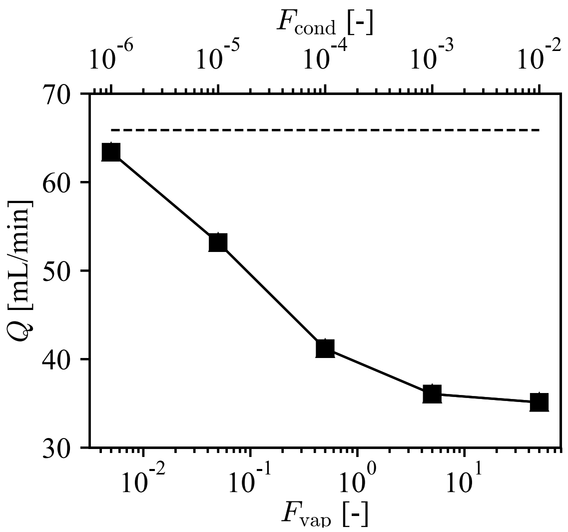

Figure 3. Oil volumetric flow rate (Q) as a function of empirical coefficients (F_{vap} and F_{cond}) and n = 1500 rpm. Dashed line indicates the result without cavitation.

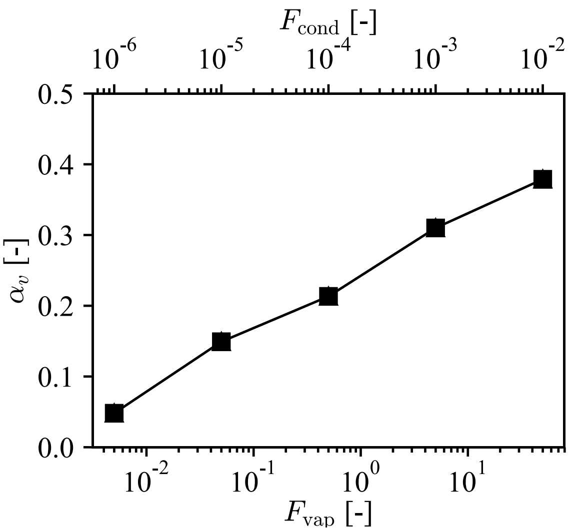

Figure 4. Overall gas volume fraction (α_v) as a function of empirical coefficients (F_{vap} and F_{cond}) and n = 1500 rpm.

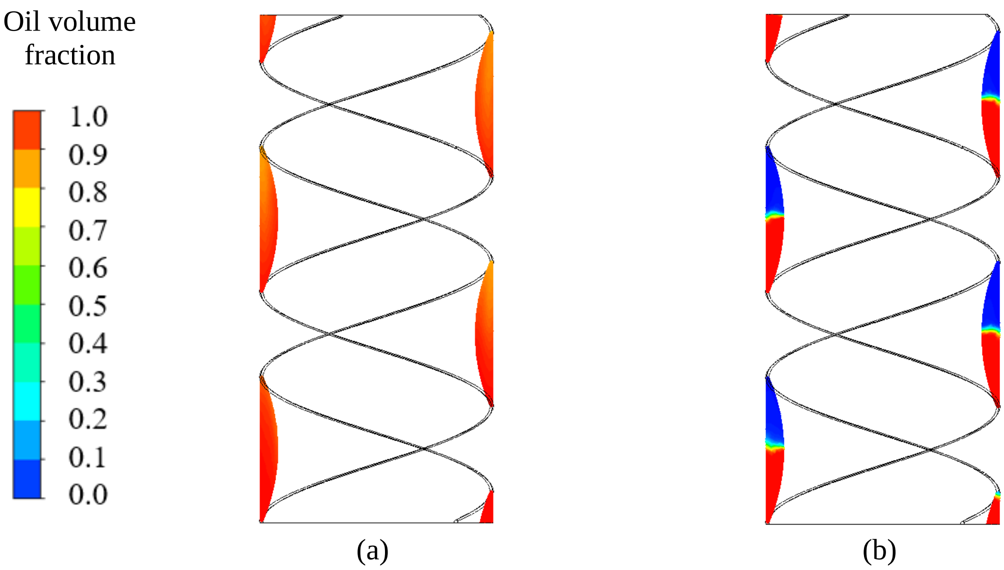

Figure 5. Contours of oil volume fraction in the pump midsection for different values of empirical coefficients: (a) $F_{\text{vap}} = 5 \times 10^{-3}$ and $F_{\text{cond}} = 10^{-6}$, (b) $F_{\text{vap}} = 50$ and $F_{\text{cond}} = 0.01$, both cases n = 1500 rpm.

Next, the sensitivity of refrigerant outgassing in relation to the compressor rotational speed (n) is investigated. The results for the oil volumetric flow rate (Q) are shown in Figure 6, where the 'No cavitation' curve represents the cases in which no cavitation is considered, that is, pure oil flow. On the other hand, the 'Cavitation' curve represents the cases in which the present cavitation model is applied. When refrigerant outgassing is present, the oil volumetric flow rate is considerably reduced, and this reduction increases with the compressor speed, reaching 50% for n = 3000 rpm. Figure 7 shows the overall gas volume fraction (α_v) for each case. As can be seen, the higher the compressor speed, the higher the volume of gas released. Therefore, the local gas volume fraction increases as well, as shown in Figure 8 for the middle section of the screw pump.

Finally, to investigate the influence of the compressor crankcase pressure on refrigerant outgassing, a simulation was carried out with the reference pressure set to 200 kPa instead of 62.8 kPa. In fact, the crankcase pressure is usually higher than the evaporation pressure when the compressor starts, because during the off-cycle refrigerant migrates into the compressor internal environment from the condenser high-pressure line.

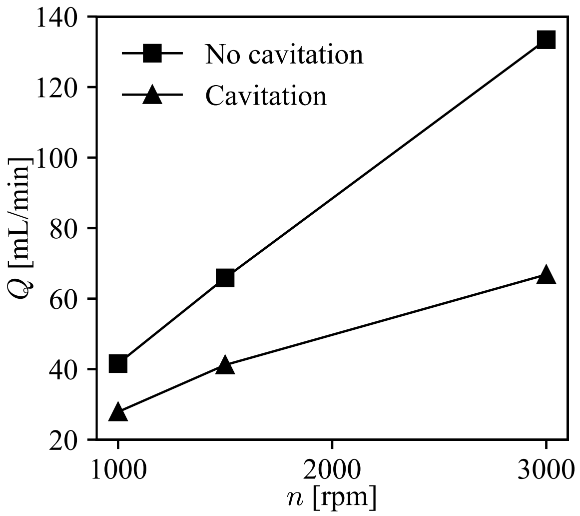

Figure 6. Oil volumetric flow rate (Q) as a function of compressor speed (n) without cavitation model ('No cavitation') and with cavitation model ('Cavitation'), $F_{\text{vap}} = 0.5$ and $F_{\text{cond}} = 10^{-4}$.

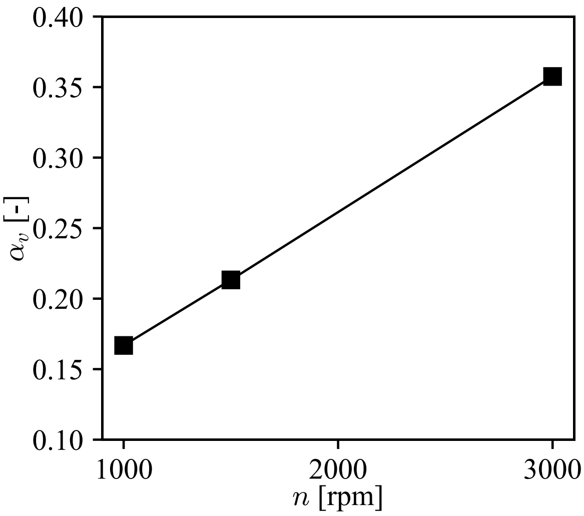

Figure 7. Domain gas volume fraction (α_v) as a function of compressor speed (n), $F_{\text{vap}} = 0.5$ and $F_{\text{cond}} = 10^{-4}$.

With the higher crankcase pressure, the mixture density and viscosity become ρ = 802.4 kg/m^3 and μ = 1.51 × 10^{-3} Pas, respectively, whereas the refrigerant mass fraction dissolved in the oil is equal to 7.8%. The results presented in Table 1 reveal minor differences in the results for the oil flow rate and gas volume fraction brought about by the higher reference pressure. This may be related to the main limitation of the model proposed herein, which assumes a fixed amount of refrigerant gas dissolved in the oil. As a consequence, the increase of the refrigerant mass fraction dissolved in the oil affects only the mixture properties, without intensifying the refrigerant outgassing process.

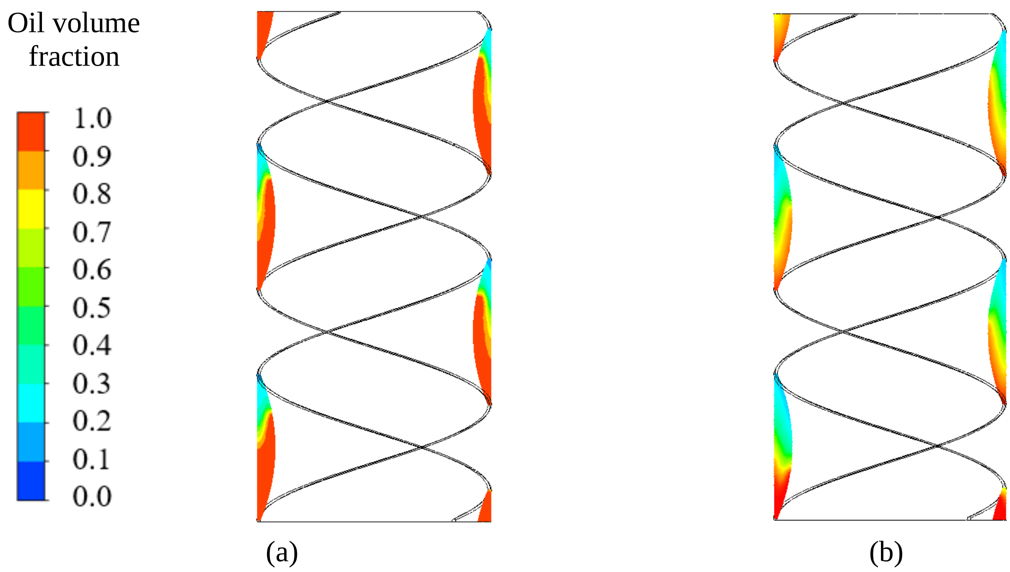

Figure 8. Contours of oil volume fraction for different speed: (a) n = 1000 rpm, (b) n = 1500 rpm, both cases $F_{\mathrm{vap}} = 0.5$ and $F_{\mathrm{cond}} = 10^{-4}$.

Table 1. Results for different refence pressures, with $F_{\mathrm{vap}} = 0.5$ and $F_{\mathrm{cond}} = 10^{-4}$ and n = 1500 rpm.

Reference pressure [kPa]	Oil flow rate [mL/min]	Overall gas volume fraction [-]
62.8	41.19	0.21
200	38.89	0.20

4. Conclusion

This paper presented a two-phase simulation model to investigate the refrigerant outgassing in the screw pump of a hermetic reciprocating compressor oil supply system. The outgassing process was modelled with a cavitation model based on the Rayleigh-Plesset equation, considering a fixed gas mass fraction dissolved in oil, corresponding to the solubility of refrigerant in oil at the crankcase pressure and temperature. Simulations were carried out to investigate the influence of empirical calibration coefficients, compressor speed and reference pressure on the refrigerant desorption phenomenon. The empirical calibration coefficients proved to strongly affect the refrigerant outgassing. The increase of the compressor speed intensified the refrigerant outgassing, with the gas occupying almost 50% of the domain volume when the rotation speed was set to 3000 rpm. On the other hand, the model predicted a very small effect of the pressure inside the compressor environment on the refrigerant outgassing, which is probably related to its limitation of assuming a fixed mass fraction of refrigerant dissolved in the oil regardless of the pressure value. Although the presented model is relatively simple, the findings of the present study highlights the importance of further analysis since refrigerant outgassing may lead to several issues, including noise generation. The adoption of a model that calculates the gas mass fraction in the oil via a transport equation should address the main limitation of the model proposed herein. It would also be important to develop an experimental rig to investigate the two-phase flow and generate data to validate the model.

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