AN EQUILIBRIUM MODEL FOR THE FLOW OF LUBRICANT OIL AND REFRIGERANT 134a THROUGH THE PISTON-CYLINDER CLEARANCE OF A SMALL RECIPROCATING COMPRESSOR

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Abstract. *A detailed knowledge of lubrication phenomena in reciprocating compressors, particularly the piston assembly lubrication, has a fundamental role in the design of more efficient refrigeration systems. A radial clearance between the piston and the cylinder wall allows piston oscillatory radial movements and the resulting piston misalignment affects both the lubricant film thickness and the lubrication regime. Furthermore, cavitation and restoration of the lubricant film considering the interaction between the oil and the refrigerant are not totally understood. With this in mind, the present work proposes a numerical methodology to predict the flow of a mixture of a polyol ester type lubricant oil and R-134a through the piston-cylinder clearance of reciprocating compressors used in small capacity refrigeration systems. Due to pressure and temperature gradients in the lubricant film, the refrigerant solubility changes and refrigerant evaporates from the liquid mixture, forming a gas/liquid two-phase flow. The model considers the flow of a refrigerant-oil mixture under mechanical and thermodynamic equilibrium in which the Reynolds and the energy conservation equations are solved considering the variation of the physical properties in both phases in order to calculate the film pressure and temperature distributions for a condition in which the piston is static. The pressure and temperature distributions were also used to calculate the average flow properties through a homogeneous two-phase flow model. The analysis shows that the compressibility of the gas phase has a major effect on temperature distribution along the mixture flow. Also, changes in radial clearance resulting from the misalignment between the piston and the cylinder can affect both pressure and temperature distributions and the avarage flow characteristics.*

Keywords*: Reciprocating compressors, oil-refrigerant mixture, piston-cylinder clearance, two-phase flow, numerical modeling*

1. INTRODUCTION

The majority of domestic and small commercial refrigeration systems use reciprocating compressors. In this type of compressor, the increase of the gas pressure is made by the reciprocating motion of a ringless piston inside a cylinder. In order to avoid the metallic contact between the piston skirt and the cylinder bore walls, the lubricant oil forms a thin film in the radial clearance between the piston and the cylinder, reducing the friction during the piston motion. The lubricant oil also has the function of sealing the compression chamber to prevent refrigerant leakage during the compression process. The existence of a clearance allows the occurrence of piston oscillatory and translational secondary movements, which can have a significant impact on the thickness and continuity of the lubricant, and hence on the compressor performance. Another important fact concerning small reciprocating compressors is that the lubricant oil is stored in the compressor sump and kept in direct contact with the gas refrigerant inside the shell. The refrigerant has some degree of solubility in oil that reduces the lubricant viscosity and, consequently, the ability to maintain a continuous lubricant film separating the piston and cylinder surfaces efficiently. Depending on the pressure and temperature gradients in the oil-refrigerant mixture, refrigerant outgassing can take place in the lubricant film. The resulting two-phase flow in the gap is a risk factor in terms of reliability, noise, overall performance and energy consumption. Thus, understanding the many aspects of the interaction between the lubricant oil and the refrigerant is essential for improving the piston lubrication in refrigeration reciprocating compressors.

A literature review shows that the investigation of lubrication phenomena in ringless pistons such as those found in small reciprocating compressors has started only recently. Prata et al. (2000) developed the first model to describe the piston trajectory during its reciprocating motion inside a cylinder lubricated by continuous oil film. The pressure gradient in the film was obtained via a finite volume solution of Reynolds equation, while the piston dynamics was solved using a Newton-Raphson procedure, which provided radial velocities at the top and at the bottom of the piston. Following a similar methodology, Rigola et al. (2003) presented a numerical model to obtain the mass flow leakage through the piston-cylinder radial clearance of a refrigeration reciprocating compressor. In addition to the dynamic equations for the motion of the connecting rod and crankshaft, the authors also used experimental results for the motor torque as a function of its instantaneous angular speed. Their results showed that the refrigerant leakage flow rate varies exponentially with respect to the piston-cylinder clearance. Given the small clearances typically used in these systems, Cho and Moon (2005) analyzed the effect of flexibility of piston and cylinder structures on piston lubrication characteristics. Results indicated that, for severe conditions, elastic deformations of the structure can produce variations of around 10% in the initial clearance, suggesting that the structure flexibility should not be ignored.

Although the previous works have been important in terms of mathematical description of ringless piston dynamics, they have not considered the lubricant film cavitation. In refrigeration compressors, this phenomenon occurs probably due to the presence of refrigerant dissolved in the oil, whose solubility depends on the local pressure and temperature. The effect of refrigerant dissolution on the tribological properties of the oil was evaluated experimentally by Jonsson (1999) for a solution of R-134a in polyol ester oil. Their study indicated that a reduction of the oil viscosity had a negative impact on the bearings lifetime. Solubility of refrigerant in oil is a key parameter in the evaluation and selection of oil-refrigerant mixtures and several theoretical and experimental works (Yokozeky, 2007; Marcelino Neto and Barbosa, 2008) were published recently dealing with the characterization of phase equilibrium of these mixtures. Moreover, other works have investigated the transient absorption of refrigerant in oil aiming to estimate the molecular diffusion coefficient for the most used oil-refrigerant pairs (Fukuta et al., 2005; Gessner and Barbosa, 2006).

Nevertheless, knowing only the thermodynamic and thermophysical properties is not enough to elucidate the behavior of oil-refrigerant mixtures flowing through compressor channels. The pressure drop caused by viscous flow reduces the refrigerant solubility leading to refrigerant outgassing from the mixture. Visual experiments conducted by Lacerda et al. (2000) and Castro et al. (2004) confirmed this fact and also reported the appearance of an expressive region of foam flow for the highest amount of gas released in straight long tubes of about 3 mm diameter. In the light of these findings, numerical models of oil-refrigerant two-phase flows were proposed by Grando and Prata (2003), Barbosa et al. (2004) and Dias and Gasche (2006), all of which obtained good agreement with experimental results from Lacerda et al. (2000) and Castro et al. (2004).

Grando et al. (2006a) proposed a lubrication model for journal bearings considering the oil-refrigerant mixture and the existence of a gas/liquid two-phase flow in the lubricant film. They obtained different results for the bearing load and friction power consumption when the Reynolds classic cavitation model was compared with a model that allowed thermodynamic non-equilibrium of the oil-refrigerant two-phase flow. In a later study, Grando et al. (2006b) extended their previous model to the piston dynamics analysis of refrigeration compressors, and reported that parameters such as the piston trajectory, power consumption and gas leakage are dependent on the amount of refrigerant dissolved in the oil. Analogously to the two-phase lubrication model applied to journal bearings, the choice of the cavitation criterion had crucial role on the numerical results for the piston lubrication problem.

The present work proposes a new numerical methodology to predict the flow of a mixture of polyol ester type lubricant oil and refrigerant R-134a through the piston-cylinder clearance of refrigeration reciprocating compressors. The model considers no relative axial movement of the piston with respect to the cylinder and that only the pressure difference between piston top and bottom sides moves the mixture through the clearance. This approach is convenient because it enables one to focus only on the flow aspects (and not on the piston movement). As the mixture flows, the pressure drop causes a reduction of the refrigerant solubility, giving rise to refrigerant outgassing. Two equilibrium hypotheses are adopted: (i) thermodynamic equilibrium of the liquid mixture, by which the local concentration of the refrigerant in the mixture is assumed to be equal to the local solubility, and (ii) mechanical equilibrium, whereby both gas and liquid phases are assumed to be at the same local pressure. Pressure and temperature distributions along the clearance axial and circumferential lengths are obtained via a coupled Finite Volume solution of the Reynolds and the energy conservation equations. The results obtained for piston alignment and misalignment conditions revealed some useful information which could help to understand the interaction between oil and refrigerant on piston lubrication.

2. PHYSICAL AND MATHEMATICAL MODELLING

Figure 1 shows a schematic diagram of the oil-refrigerant mixture flowing through the piston-cylinder clearance in which the piston is static. For this condition, the flow is driven only by the pressure difference imposed between the inlet at the piston top side (p_{in}) and the outlet at the piston bottom side (p_{out}). The piston misalignment, exaggerated in Fig. 1, affects both axial and circumferential cross sectional areas and the pressure distributions in these directions.

The liquid mixture at the inlet is maintained at a fixed thermodynamic state dictated by pressure, temperature and refrigerant mass fraction (w_r), which is defined as the amount of refrigerant dissolved in the liquid mixture by Eq. (1),

$$w_r = \frac{m_{lr}}{m_l} \tag{1}$$

where m_{lr} and m_l are, respectively the liquid refrigerant and total liquid phase masses. As the mixture flows the pressure drop reduces the refrigerant solubility, $w_{sat}(p,T)$, which is defined as the maximum quantity of refrigerant that can be dissolved in the liquid for the local condition of pressure and temperature. From this point on, it is assumed that the relationship between the inlet refrigerant mass fraction and the local solubility can lead to two different situations. While the local solubility is higher than the inlet refrigerant mass fraction, the flow is locally single-phase with no gas released from the mixture (the refrigerant mass fraction keeps constant). On the other hand, when the local solubility becomes equal to the refrigerant mass fraction, the excess of refrigerant evaporates from the mixture giving rise to a bubbly gas-liquid two-phase flow, as detailed in Fig. 1; the remaining liquid phase is assumed be in thermodynamic equilibrium so that the refrigerant mass fraction is always equal to the local solubility. Moreover, since the refrigerant outgassing is an endothermic process and the flow is nearly adiabatic, there is a temperature drop along the flow.

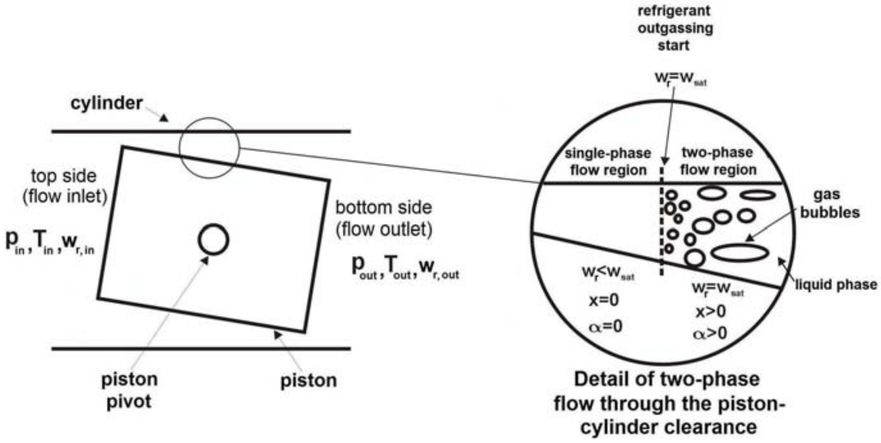


Figure 1. Schematic diagram of the flow of oil and refrigerant through the piston-cylinder clearance

Two additional parameters used to characterize gas-liquid two-phase flows are the mass quality (x) and the void fraction (α). The mass quality is the ratio between the mass of gas and the total mass of the two-phase mixture within a cross-sectional area of the flow. In terms of the inlet refrigerant mass fraction and the local solubility, it is given by,

$$x = \frac{w_{r,in} - w_{sat}\left(p,T\right)}{1 - w_{sat}\left(p,T\right)} \tag{2}$$

The void fraction is defined as the ratio between the gas volume and the total volume within a cross section, which for hydrodynamic equilibrium conditions (identical *in-situ* velocities in both phases) is calculated in terms of the liquid and gas densities, ρ_l and ρ_g, as follows,

$$\alpha = \frac{1}{\left[1 + \left(\frac{1}{x} - 1\right)\frac{\rho_g}{\rho_l}\right]} \tag{3}$$

Also, according to the above assumption of identical velocities in both phases (homogeneous model), the quality and the void fraction can be used to estimate average flow properties from the properties of each individual phase. Thus, the homogeneous density and viscosity are given by,

$$\bar{\rho} = \alpha\rho_g + \left(1-\alpha\right)\rho_l; \qquad \bar{\mu} = x\mu_g + \left(1-x\right)\mu_l \tag{4}$$

It is worth mentioning that there are several expressions available in the literature for calculating the homogeneous viscosity (Collier and Thome, 1994), and the relationships given by Eq. (4) has been chosen based on previous experience of this research group in the simulation of flashing flows in small diameter tubes.

An illustration of the problem geometry and of the coordinate systems adopted in the mathematical formulation is presented in Fig. 2, in which the piston has radius R and axial length L. Two different coordinate systems are considered: the first consists of cylindrical coordinates $r\theta z$ placed at the piston top center which is used as a reference to the flow parameters along the clearance, the second is a rectangular coordinates XYZ fixed at the cylinder top center and is used to describe the piston misalignment represented by the eccentricities e_Y and e_Z. This approach is very convenient to calculate the system eccentricities, but only if the piston and the cylinder are assumed to be rigid bodies. All the misalignments are caused by piston translation along the YZ plane and rotation around the Y axis; furthermore, the piston rotation center is assumed be positioned at the axial coordinate $y=L/2$. The eccentricities of the piston-cylinder system give rise to a variable film thickness h as a function of piston circumferential and axial coordinates. The model adopts the following main assumptions: (i) laminar and steady-state flow; (ii) properties changes in clearance radial coordinate are negligible; (iii) the mixture is treated as a Newtonian fluid and flows in mechanical and thermodynamic equilibrium; (iv) oil vapor pressure is negligible and (v) piston and cylinder walls are considered rigid and adiabatic.

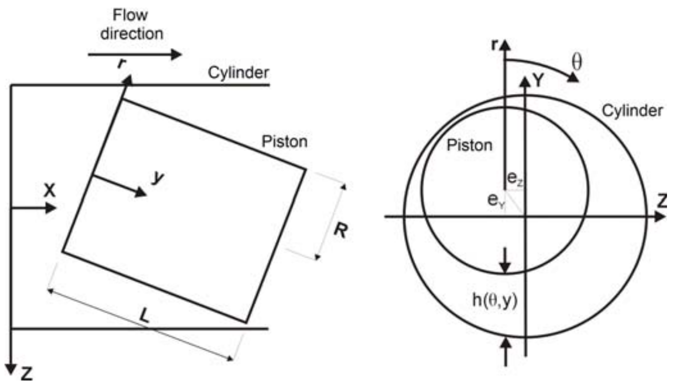


Figure 2. Geometric characteristics of the piston-cylinder clearance and coordinate systems and used in the mathematical formulation

2.1. The Reynolds Equation

The Reynolds equation considers pressure and viscous forces to be dominant over the inertia force and, for the situation where the piston is static, it can be written in terms of the dimensionless axial coordinate ξ as,

$$\frac{\partial}{\partial\theta}\left(\frac{\bar{\rho}h^3}{\bar{\mu}}\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial\xi}\left(\frac{\bar{\rho}h^3}{\bar{\mu}}\frac{\partial p}{\partial\xi}\right)=0 \tag{5}$$

which requires the following boundary conditions:

$$\xi = 0 \rightarrow p = p_{in}; \quad \xi = \frac{L}{R} \rightarrow p = p_{out}; \quad p(0,\xi) = p(2\pi,\xi) \tag{6}$$

Equation (5) requires previous knowledge of the lubricant film thickness, $h(\theta,\xi)$ over the solution domain. Through geometrical relationships from Fig. 2, the film thickness can be calculated as,

$$h(\theta,\xi) = c\left(1-\varepsilon_Y \cos\theta - \varepsilon_Z sen\theta\right) \tag{7}$$

where c is the nominal clearance, and $\varepsilon_Y=e_Y/c$ and $\varepsilon_Z=e_Z/c$ are the dimensionless eccentricities on the Y and Z directions, respectively, written as functions of the piston top and bottom dimensionless eccentricities in Eqs. (8),

$$\varepsilon_Y = \varepsilon_{t,Y} - \frac{\varepsilon_{t,Y}-\varepsilon_{b,Y}}{(L/R)}\xi; \quad \varepsilon_Z = \varepsilon_{t,Z} - \frac{\varepsilon_{t,Z}-\varepsilon_{b,Z}}{(L/R)}\xi \tag{8}$$

2.2. The Energy Conservation Equation

The first step is to start from the equation for energy conservation of a general phase k. Considering that the effects of viscous dissipation, heat generation and slip between the phases are negligible, together with the equilibrium hypotheses, the vectorial form of the energy equation in terms of the specific enthalpy, i, the conductive heat flux vector, $\mathbf{q}$, and the compressibility of the phase k is written as,

$$\alpha_k\rho_k\frac{Di_k}{Dt} = -\nabla\cdot\left(\alpha_k\mathbf{q}_k\right)+\alpha_k\frac{Dp}{Dt} \tag{9}$$

For the gas phase, the enthalpy is assumed to be a continuous function of the pressure and temperature and then, the gas enthalpy variation can be defined as a function of pressure and temperature variations as follows,

$$di_g = \frac{1-\beta_g T}{\rho_g}dp + c_{p,g}dT \tag{10}$$

in which β_g and $c_{p,g}$ are, respectively, the thermal expansion coefficient and the specific heat capacity of the gas phase. Thus, by using the Fourier's Law to express the conductive heat transfer, considering steady state flow and the further assumption of ideal gas ($\beta_g \cong 1/T$), Eq. (10) for the gas phase becomes,

$$\alpha_g \rho_g c_{p,g} \mathbf{V} \cdot \nabla T = \nabla \cdot \left(\alpha_g k_g \nabla T \right) + \alpha_g \mathbf{V} \cdot \nabla p \tag{11}$$

where k_g is the gas thermal conductivity, ∇T and ∇p are the pressure and temperature gradients vectors and $\mathbf{V}$ is the flow velocity field whose axial and circumferential components are calculated through the flow pressure solution as,

$$u_\theta = -\frac{h^2}{12\bar{\mu} R}\frac{\partial p}{\partial \theta}; \qquad u_\xi = -\frac{h^2}{12\bar{\mu} R}\frac{\partial p}{\partial \xi} \tag{12}$$

Following the same reasoning for the liquid phase as an ideal binary mixture, the enthalpy is function of the pressure, temperature and refrigerant mass fraction. Thus, the enthalpy variation will have an additional term relating the refrigerant mass fraction variation and the solute (liquid refrigerant) and solvent (oil) enthalpies (Bird et al., 2002):

$$di_l = \frac{1-\beta_l T}{\rho_l} dp + c_{p,l} dT + \left(i_{lr} - i_o \right) dw_r \tag{13}$$

where β_l and $c_{p,l}$ are, the thermal expansion coefficient and the specific heat capacity of the liquid phase. Substitution of Eq. (13) in Eq. (9) results in,

$$\alpha_l \rho_l c_{p,l} \frac{DT}{Dt} = -\nabla \cdot \left(\alpha_l \mathbf{q}_l \right) + \alpha_l \beta_l T \frac{Dp}{Dt} - \alpha_l \rho_l \left(i_{lr} - i_o \right) \frac{Dw_r}{Dt} \tag{14}$$

According to Bird et al. (2002), for problems involving simultaneous heat and mass transfer, the diffusive term is composed of one term for the heat conduction due to a temperature gradient and another term for the heat transport with the mass diffusion due the refrigerant mass fraction gradient on the liquid phase. Thus, the diffusive term is written as,

$$\mathbf{q}_l = -k_l \nabla T - \rho_l D_{OR} \left(i_{lr} - i_o \right) \nabla w_r \tag{15}$$

where k_l is the liquid phase thermal conductivity and D_{OR} is the mass diffusion coefficient. Substituting Eq. (15) in Eq. (14) and assuming that the thermal expansion coefficient of the liquid phase is negligible, the resulting steady-state equation for liquid phase energy conservation equation, after some algebraic manipulation, is,

$$\alpha_l \left(c_{p,lr} \mathbf{m}_{lr} + c_{p,o} \mathbf{m}_o \right) \cdot \nabla T = \nabla \cdot \left(\alpha_l k_l \nabla T \right) + \left(i_{lr} - i_o \right) \rho_l D_{OR} \nabla w_r \cdot \nabla \alpha_l \tag{16}$$

where $\mathbf{m}_{lr}$ and $\mathbf{m}_o$ represent the total (advection + diffusion) mass fluxes of refrigerant and oil given respectively by,

$$\mathbf{m}_{lr} = \rho_{lr} \mathbf{V} - \rho_l D_{OR} \nabla w_r; \quad \mathbf{m}_o = \rho_o \mathbf{V} + \rho_l D_{OR} \nabla w_r \tag{17}$$

Adding Eqs. (11) and (16) considering that $\alpha_g = \alpha$ and $\alpha_l = 1-\alpha$, the two-phase flow energy equation becomes,

$$\mathbf{A}.\nabla T = \nabla \cdot \left(\bar{k} \nabla T \right) + B \tag{18}$$

$$\mathbf{A} = A_\theta \mathbf{i} + A_\xi \mathbf{j} = (1-\alpha)\left(c_{p,lr} \mathbf{m}_{lr} + c_{p,o} \mathbf{m}_o \right) + \alpha \rho_g c_{p,g} \mathbf{V}; \quad B = \alpha \mathbf{V} \cdot \nabla p - \left(i_{rl} - i_o \right) \rho_l D_{OR} \nabla w_r \cdot \nabla \alpha \tag{19}$$

$$\bar{k} = \alpha k_g + (1-\alpha) k_l \tag{20}$$

and the gradient vector of a general scalar property ϕ is given by,

$$\nabla \phi = \frac{1}{R}\left(\frac{\partial \phi}{\partial \theta} \mathbf{i} + \frac{\partial \phi}{\partial \xi} \mathbf{j} \right) \tag{21}$$

As well as for Eq. (5) the boundary conditions adopted for Eq. (18) are based on inlet and outlet prescribed temperatures, and on the existence of circularity of the physical domain.

3. SOLUTION PROCEDURE

A scheme of numerical domain discretization of the problem is shown in Fig. 3. Equations (5) and (18) are integrated via a finite volume numerical procedure (Patankar, 1980) for each of the small volumes depicted in Fig. 3. The algebraic equations which result from Eqs. (5) and (18) integration in an intermediate volume assume the forms,

$$a_{P,p}\, p_P = a_{n,p}\, p_N + a_{s,p}\, p_S + a_{e,p}\, p_E + a_{w,p}\, p_W + b_p \tag{22}$$

$$a_{P,T} T_P = a_{n,T} T_N + a_{s,T} T_S + a_{e,T} T_E + a_{w,T} T_W + b_T \tag{23}$$

in which the coefficients of the above equations are calculated with the flow properties at the volume neighborhood.

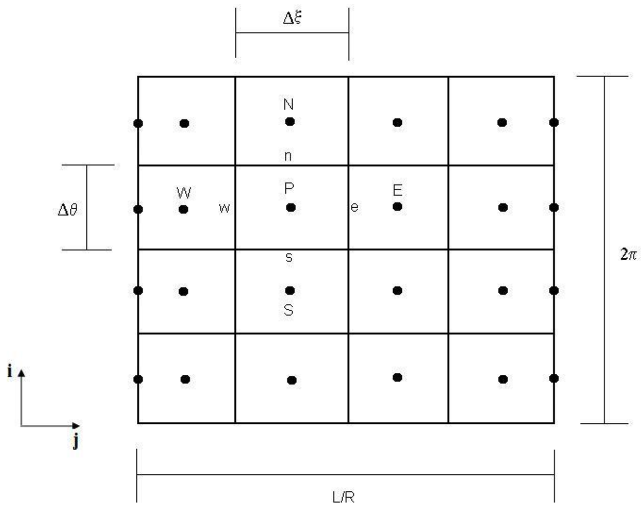


Figure 3. Numerical discretization of the domain

Two non-linear systems of algebraic equations are formed by Eqs. (22) and (23) which required an iterative coupled procedure of solution. Starting from initial guesses for pressure and temperature field, a CTDMA algorithm was used to solve each equation system updating the pressure and temperature fields iteration by iteration until a convergence criterion is satisfied. The sequence of steps followed by the numerical solution was: (i) data input of geometric and computational parameters and mixture properties at the piston top and bottom sides; (ii) estimate initial guesses for pressure and temperature fields along the flow; (iii) calculate the mixture properties over the entire domain; (iv) update the pressure field solving the non-linear system given by Eq. (22); (v) update the temperature field solving the non-linear system given by Eq. (23); (vi) evaluate the difference between the pressure and temperature fields calculated in the current iteration and the ones calculated in the previous iteration. If both differences are less than a previously established tolerance, convergence was achieved; if not, return to step (iii) and iterate until convergence.

4. RESULTS AND DISCUSSION

The oil-refrigerant pair considered was a polyol ester ISO 10 and the hydrofluorocarbon refrigerant HFC-134a, a combination commonly used in household refrigeration compressors. The oil properties were calculated with the same correlations used by Grando and Prata (2003) and Dias and Gasche (2006). The properties of the refrigerant gas and liquid phases were obtained through REFPROP 6.0 (McLinden et al., 1998) database. Table 1 shows the geometric and computational data used in simulations presented on this section. The molecular diffusion coefficient was assumed constant, as suggested by Gessner and Barbosa (2006), given the pressure and temperature range considered here. It is assumed in all cases that the mixture is saturated at the channel inlet (based on the prescribed values of local pressure and temperature). This condition implies that, under the assumption of thermodynamic equilibrium, the very first drop in pressure due to fluid friction provokes the refrigerant outgassing from the mixture leading to the occurrence of the two-phase flow in the clearance.

Table 1. Geometric and computational data used in simulations.

R (mm)	10.5
L (mm)	21.0
c (μm)	5.0
p_{in} and p_{out} (kPa)	1000.0 and 100.0
T_{in} and T_{out} (^{o}C)	60.0 and 55.0
D_{OR} (m^2/s)	3.0×10^{-10}
Mesh	50×50
Tolerances	1.0×10^{-6}

The results for pressure, temperature, refrigerant mass fraction and gas void fraction distributions along the piston dimensionless axial coordinate for the piston alignment condition (i.e., constant clearance) are presented in Fig. 4. Particularly for this condition, the flow properties distributions are symmetric in the circumferential direction.

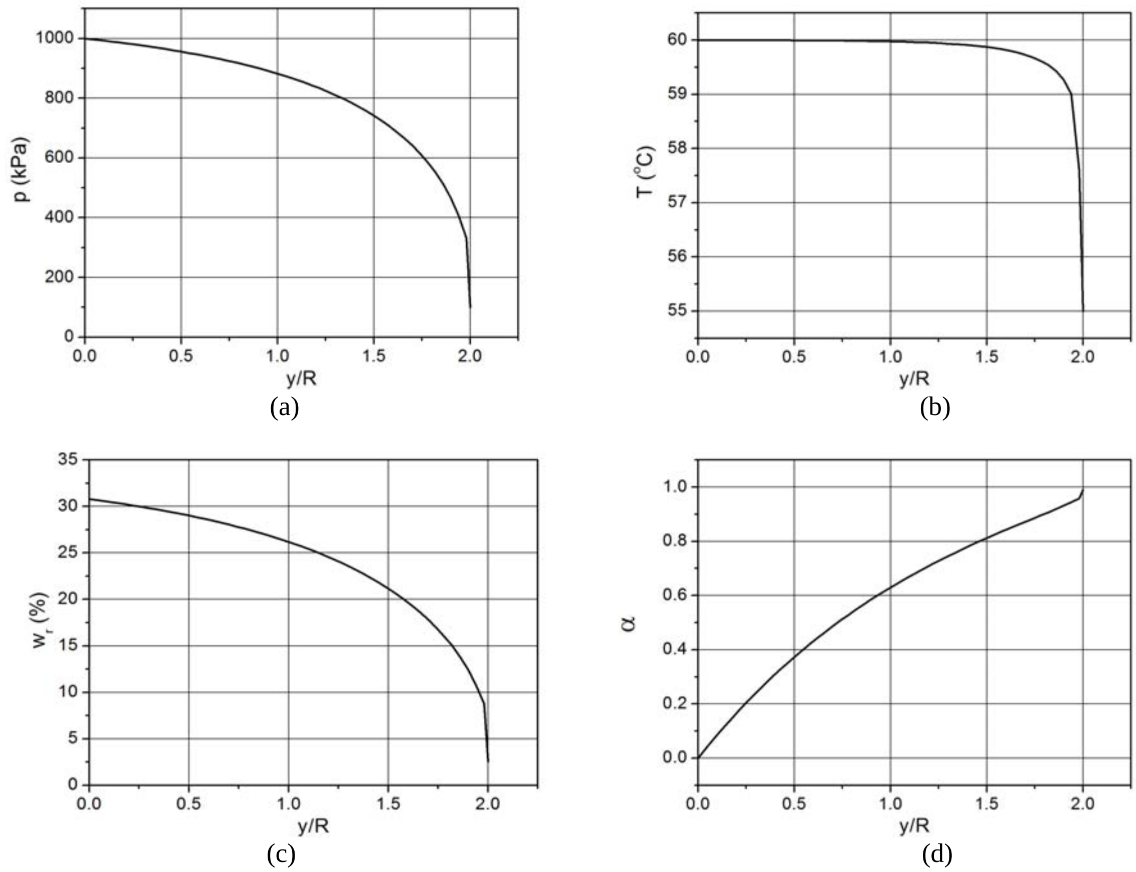


Figure 4. Simulation results for axial profiles of the flow properties (alignment condition)

The behavior observed in Fig. 4 was similar as that reported by Grando and Prata (2003) and Dias and Gasche (2006) for straight tubes: as the flow pressure is reduced by flow viscous forces (Fig. 4a), the refrigerant mass fraction (or solubility) is also reduced (Fig. 4c) causing the increase on volume of gas released from the mixture (Fig. 4d). The first half of the flow is marked by a relatively low pressure drop which results in a void fraction of about 60% and a constant temperature profile (Fig. 4b). For the flow second half region, where the pressure drop is high, a strong temperature drop is also observed due the intense mass and heat transfer from the liquid mixture to the gas phase, once the piston and cylinder walls were considered adiabatic; the flow void fraction reaches up to 99% at the piston bottom.

The next results analyze exclusively the heat transport mechanism considered in energy conservation equation and its influence on the flow characteristics through the clearance for the piston alignment condition. The first analysis investigates the effect of the energy transport due the refrigerant mass diffusion in the liquid mixture (given in Eq. 15) and is shown in Fig. 5. These results indicate that the inclusion of the mass diffusion term in the energy equation have no influence on both flow pressure and temperature distributions probably because of the small magnitude of the molecular diffusion coefficient.

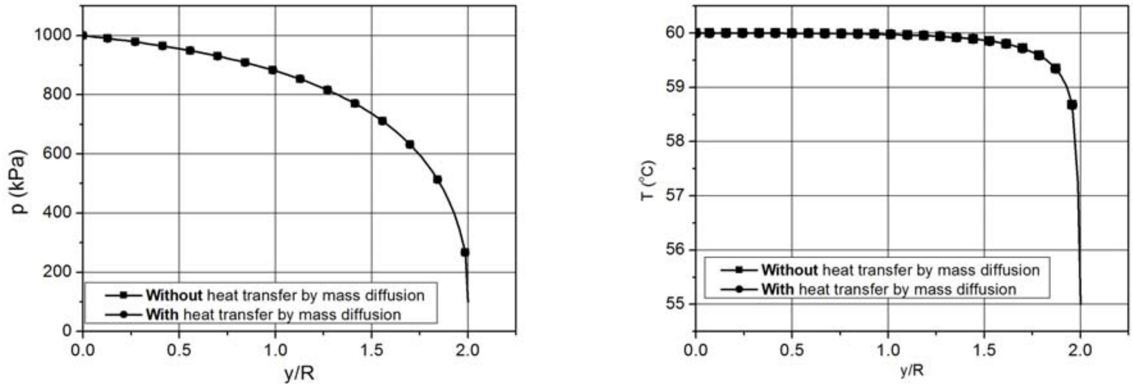


Figure 5. Effect of the mass diffusion modeling on axial properties profiles (alignment condition)

Figure 6 explores the results of the second analysis concerning the energy equation terms in which simulations were performed considering the following conditions: (I) presence of only diffusive terms in energy equation; (II) presence of diffusive and advective terms and (III) presence of all terms in the energy equation (diffusive, advective and gas phase compressibility terms). Results show the major influence on temperature distributions along the dimensionless axial coordinate. When only the diffusive terms are considered, it is interesting to observe that the temperature drop is not constant due the dependence of the mixture thermal conductivity with the flow temperature. Advective terms inclusion leads to a drastic change on temperature profiles when compared with the pure diffusion case, once flow velocities are high enough to suppress energy transport effects by conduction. Adding the gas phase compressibility term originated from the flow pressure drop, the flow advective effect is still dominant until the amount of gas released becomes high enough to smooth the temperature gradient near the piston bottom side.

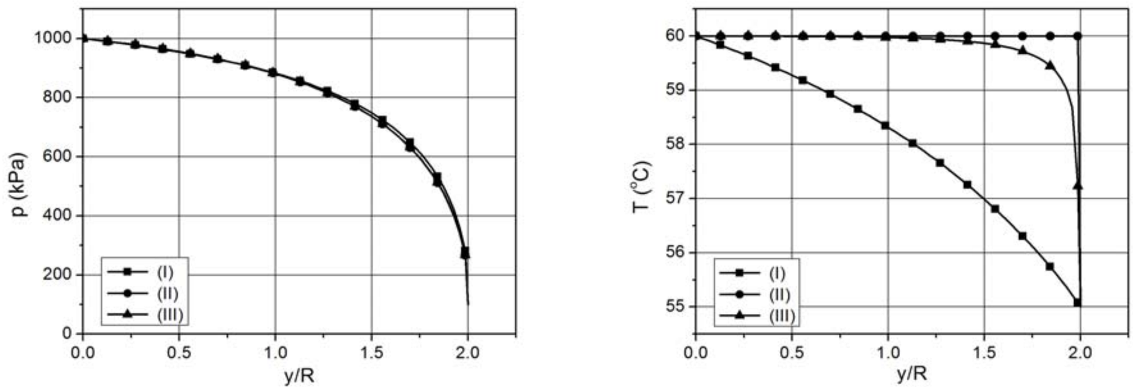
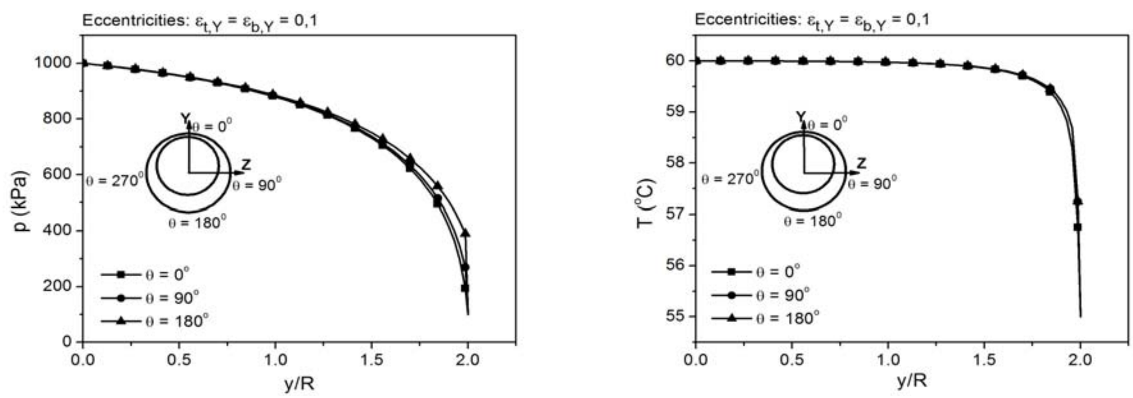
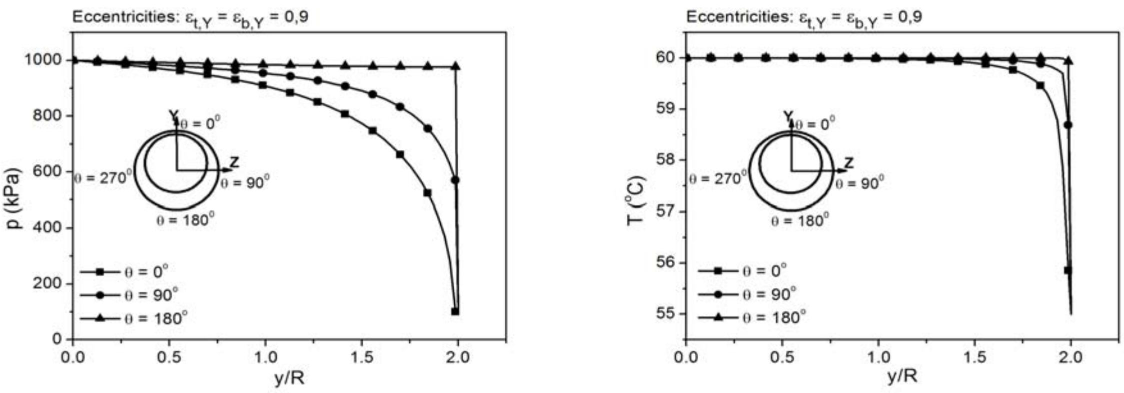


Figure 6. Influence of the energy transport terms on axial properties profiles (alignment condition)

Finally, Figs. 7 and 8 present the model results for the evaluation of the piston-cylinder misalignment of the flow through the clearance. The misalignments are represented by piston top and bottom dimensionless eccentricities. Figure 7 shows the axial pressure and temperature profiles for the situation in which the piston is translated along the positive *Y* axis resulting in the flow of the film thickness that varies only with the circumferential direction.



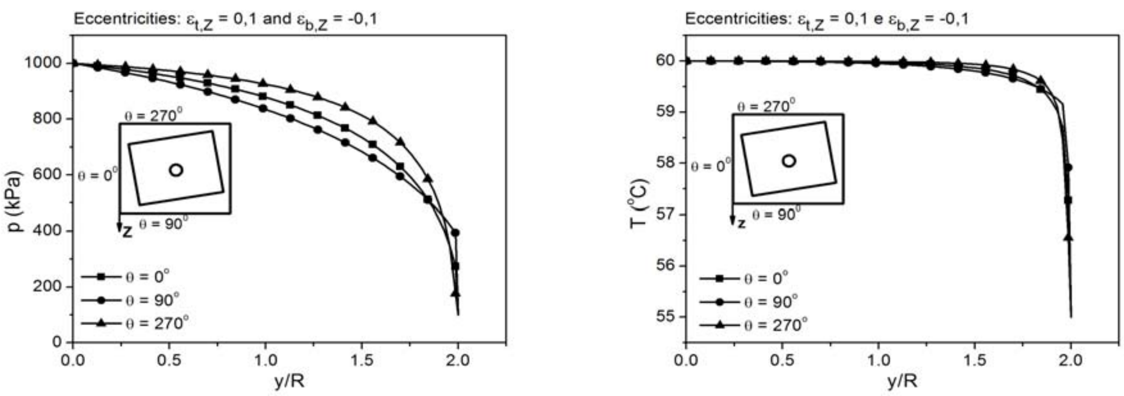
(a)



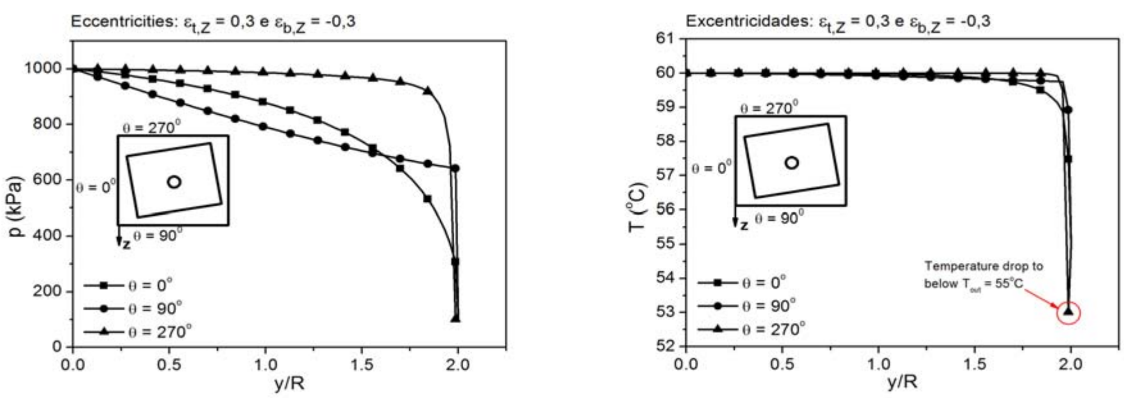
(b)

Figure 7. Pressure and temperature profiles for a misalignment situation caused by piston translation

Comparing the different values of eccentricities (Figs. 7a and 7b), it is noted that the degree of misalignment affects strongly the flow properties distribution. Furthermore, both pressure and temperature drops (Fig. 7b) are more intense at the minimum clearance region, i.e., θ=0°. The reason is the increase on viscous friction caused by reduction of the cross sectional flow area that results in more outgassing from the liquid mixture. Pressure and temperature distributions for misalignment caused by piston rotation in *XZ* plane are shown in Fig. 8 in which there are both axial and circumferential variations of the film thickness. Again, the degree of misalignment has a direct influence on the properties distributions along the clearance; through the analysis of pressure axial distributions one can realize that for the divergent channel formed in $\theta=90°$, the pressure drop is higher near the piston top and it experiments a certain decrease until a lower value at the proximity of the piston bottom side. For the convergent channel, in $\theta=270°$, the behavior is exactly the opposite: the pressure drops in a lower rate near the piston top and increases continuously until reaches a maximum value at the proximity of the piston bottom. A further detail in this region is that the excessive pressure drop caused a temperature decrease to a value below the temperature defined at the flow outlet. These results are also related with the viscous pressure drop increase when the flow cross sectional area becomes smaller.



(a)



(b)

Figure 8. Pressure and temperature profiles for a misalignment situation caused by piston rotation

5. CONCLUSIONS

This work proposed a new model to describe the flow of oil and refrigerant through the piston-cylinder clearance of small reciprocating compressors commonly employed in domestic refrigeration systems. The model dealt with the mechanical and thermodynamic equilibrium hypotheses for the gas/liquid two-phase flow of the oil-refrigerant mixture. From this point, the Reynolds and energy conservation equations were solved numerically over the piston-cylinder geometry to provide the profiles of pressure and temperature for both alignment and misalignment conditions of a stationary piston. The results, which were obtained for a mixture of polyol ester lubricant oil and HFC-134a, showed a general trend similar to that provided by models proposed to describe the same problem in straight tubes. The particularity of the model was the solution of the energy conservation equation to calculate the flow temperature profile along the piston-cylinder clearance. The influence of energy equation terms was evaluated showing that the convective and gas phase compressibility terms exerted much more influence than diffusive heat and mass transfer terms on temperature drops estimated by the model. Finally, an analysis was performed when the piston was submitted to different misalignments indicating that, for a variable film thickness, smaller flow cross sectional areas are related to higher viscous pressure drops and, consequently, higher outgassing rates and temperature drops.

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