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IMPROVING COMPRESSOR DESIGN THROUGH MODELING OF PHASE CHANGE AND TWO-PHASE FLOW PHENOMENA

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ABSTRACT

This paper deals with (i) the occurrence of phase change and two-phase flow phenomena inside reciprocating compressors and (ii) modeling approaches to solve specific problems in which these phenomena arise. The kernel of the discussion resides on the existence of a two-component (lubricant oil and refrigerant) mixture subjected to sudden compression and depressurization operations inside the compressor. The following phenomena and their implications are addressed in detail in the manuscript: (a) refrigerant absorption by lubricant oil and (b) lubrication subjected to refrigerant degassing from the lubricant mixture.

1. INTRODUCTION

Several vapor-liquid two-phase flow and phase change phenomena take place inside reciprocating compressors. Due to the very low vapor pressure of the oil, the vapor phase is comprised solely by refrigerant whereas the liquid phase is a mixture of lubricant oil and dissolved refrigerant. The solubility of the refrigerant in the liquid mixture depends on local pressure and temperature conditions. Therefore, as significant pressure and temperature gradients are experienced within the compressor, the associated variation of local solubility leads to refrigerant being locally absorbed or locally released (in the form of vapor bubbles) by the liquid mixture.

The effect of variation of the liquid mixture physical properties as a function of the dissolved refrigerant content has been under scrutiny for several years (Spauschus, 1963a,b; Martz and Jacobi, 1994). This manuscript addresses the implications of local phase change patterns that, when associated with such variations of physical properties, may have a significant impact on the design of compressor components and also on the performance of the cycle as a whole.

This paper is structured so as to present a compilation of studies on phase change and two-phase flow of refrigerant-lubricant mixtures carried out under the auspices of a joint research program between Embraco and the Federal University of Santa Catarina. Two major studies will be addressed. The first study is concerned with refrigerant absorption by an initially stagnant layer of lubricant oil. The approach was to solve coupled mass, momentum and diffusion equations to properly account for density instabilities in the liquid phase. The second study is dedicated to lubrication modeling in the presence of refrigerant degassing and subsequent bubble formation in the liquid. The geometry addressed in this study is that of an infinite partial bearing. In what follows, each of the two studies is described in detail.

2. REFRIGERANT ABSORPTION BY LUBRICANT OIL

2.1 The problem

The goal of this study is to investigate the process of refrigerant absorption by a layer of lubricant oil under isothermal conditions. Ultimately, such an analysis is motivated by the determination of the system equalizing pressure, which has a direct impact on the amount of copper required for compressor start-up, and hence on the overall cost of the compressor. The study is a step beyond the pioneering analyses of Fukuta *et al.* (1995) and of

Yokozeki (2001) in the sense that it deliberately accounts for density instabilities and natural convection effects in the liquid layer (the liquid refrigerant is heavier than the oil).

The geometry of the problem is described in Figure 1. A given amount of pure oil is placed inside a test cell of known dimensions. Upon removal of moisture and non-condensable gases, the oil is brought into contact with pure refrigerant vapor inside the sealed test cell. As the oil absorbs the refrigerant, the system pressure decreases.

2.2 Experiments

The experimental apparatus is depicted in Figure 2. The test (equilibrium) cell is kept immersed in a temperature controlled bath. The internal volume of the test cell is 467.14 cm³, and a specified amount of oil is inserted into the cell. Vacuum is then applied to the cell. A pressure regulating valve dictates the amount of gas allowed into the cell. Experiments were performed using a R12/Mineral oil and a R134a/Polyol Ester Oil. The reader is referred to Silva (2004) for additional information regarding the experimental procedure.

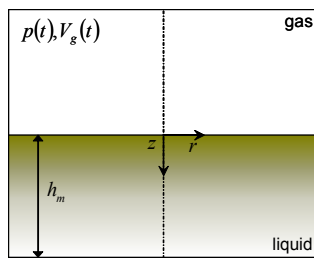


Figure 1: The geometry of the problem

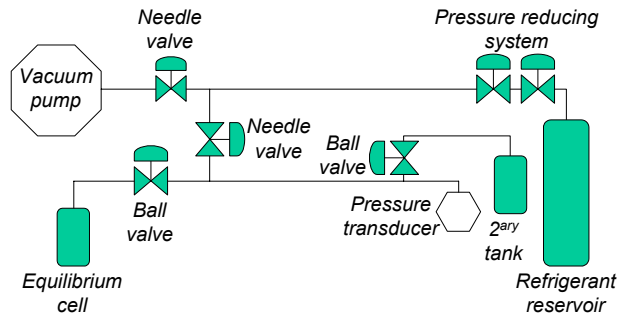


Figure 2: A schematic representation of the experimental apparatus

2.3 Modeling

The methodology addresses the time-dependent absorption of refrigerant by a layer of lubricant oil. The variation of height of the liquid layer is accounted for in the model equations. The liquid phase is modelled as a binary mixture in which the macroscopic motion of fluid is due to natural convection resulting from density differences between the pure liquid refrigerant and the oil. The governing equations are as follows:

(i) Overall mass conservation

$$\frac{\partial \rho_m}{\partial t} + \frac{1}{r} \frac{\partial(\rho_m r v_r)}{\partial r} + \frac{1}{h_m} \frac{\partial(\rho_m v_z)}{\partial Z} = \frac{Z}{h_m} \frac{dh}{dt} \frac{\partial \rho_m}{\partial Z} \quad (1)$$

(ii) Momentum conservation in the r direction

$$\begin{aligned} \frac{\partial(\rho v_r)}{\partial t} + \frac{\partial(\rho v_r v_r)}{\partial r} + \frac{1}{h_m} \frac{\partial(\rho v_z v_r)}{\partial Z} = & -\frac{\partial P}{\partial r} + \frac{1}{r} \frac{\partial}{\partial r} \left(r \mu \frac{\partial v_r}{\partial r} \right) + \frac{1}{h_m^2} \frac{\partial}{\partial Z} \left(\mu \frac{\partial v_r}{\partial Z} \right) \\ & + \frac{1}{r} \frac{\partial}{\partial r} \left(r \mu \frac{\partial v_r}{\partial r} \right) + \frac{1}{h_m} \frac{\partial}{\partial Z} \left(\mu \frac{\partial v_z}{\partial r} \right) + \frac{Z}{h_m} \frac{dh}{dt} \frac{\partial(\rho_m v_r)}{\partial Z} \end{aligned} \quad (2)$$

(iii) Momentum conservation in the z direction

$$\begin{aligned} \frac{\partial(\rho v_z)}{\partial t} + \frac{\partial(\rho v_r v_z)}{\partial r} + \frac{1}{h_m} \frac{\partial(\rho v_z v_z)}{\partial Z} = & -\frac{1}{h_m} \frac{\partial P}{\partial Z} + \frac{1}{r} \frac{\partial}{\partial r} \left(r \mu \frac{\partial v_z}{\partial r} \right) + \frac{1}{h_m^2} \frac{\partial}{\partial Z} \left(\mu \frac{\partial v_z}{\partial Z} \right) \\ & + \frac{1}{h_m} \frac{1}{r} \frac{\partial}{\partial r} \left(r \mu \frac{\partial v_r}{\partial r} \right) + \frac{1}{h_m^2} \frac{\partial}{\partial Z} \left(\mu \frac{\partial v_z}{\partial Z} \right) + \frac{1}{h_m} \frac{\partial}{\partial Z} \left(\mu \frac{\partial v_z}{\partial r} \right) + \frac{Z}{h_m} \frac{dh}{dt} \frac{\partial(\rho_m v_z)}{\partial Z} + \rho g_z \end{aligned} \quad (3)$$

(iv) Mass conservation of oil within the liquid layer

$$\begin{aligned} \frac{\partial(\rho_m w_r)}{\partial t} + \frac{\partial(\rho_m v_r w_r)}{\partial r} + \frac{1}{h_m} \frac{\partial(\rho_m v_z w_r)}{\partial Z} = \frac{1}{r} \frac{\partial}{\partial r} \left(\rho_m r D_{RO} \frac{\partial w_r}{\partial r} \right) \\ + \frac{1}{h_m^2} \frac{\partial}{\partial z} \left(\rho_m D_{RO} \frac{\partial w_r}{\partial Z} \right) + \frac{Z}{h_m} \frac{dh}{dt} \frac{\partial(\rho_m w_r)}{\partial Z} \end{aligned} \quad (4)$$

(v) Instantaneous liquid layer thickness

$$h_m = \frac{m_o}{\int_0^1 \left[(1 - w_R) / \left(\frac{w_R}{\rho_R} + \frac{1 - w_R}{\rho_O} \right) \right] A dZ} \quad (5)$$

(vi) Initial and boundary conditions

$$\begin{aligned} w_R = 0 \quad \text{for } t = 0 \\ \frac{\partial v_r}{\partial Z} = 0; \quad v_z = -\frac{D_{RO}}{(1 - w_R) h_m} \frac{1}{\partial Z} \frac{\partial w_R}{\partial Z}; \quad w_R = w_{eq} \quad \text{for } Z = 0 \\ v_r = 0; \quad v_z = 0; \quad \frac{\partial w_R}{\partial Z} = 0 \quad \text{for } Z = 1 \\ v_r = 0; \quad v_z = 0; \quad \frac{\partial w_R}{\partial r} = 0 \quad \text{for } r = R \\ v_r = 0; \quad \frac{\partial v_z}{\partial r} = 0; \quad \frac{\partial w_R}{\partial r} = 0 \quad \text{for } r = 0 \end{aligned} \quad (6)$$

The density and viscosity of the liquid mixture, ρ_m and μ_m , were calculated from an interpolation of experimental data provided by the oil manufacturers. An iterative algorithm was implemented in order to solve the governing equations. The instantaneous pressure gradient is computed from the interfacial mass balance (Eq. 7). The proposed iterative algorithm encompasses the determination of the molecular diffusivity of the refrigerant in the oil, D_{RO} , through best-fit procedures using the experimental and calculated instantaneous pressures.

$$\frac{V_g}{RT} \frac{dP}{dt} = A \left(\rho_{RV} \frac{dh}{dt} - \frac{\rho_{m,int} D_{RO}}{(1 - w_{eq}) h_m} \frac{\partial w_R}{\partial Z} \Big|_{int} \right) \quad (7)$$

2.4 Results

Figures 3.a and 3.b present curves of pressure decay under constant temperature for the two mixtures. The molecular diffusion coefficient is obtained using two different data fitting approaches: (i) a constant diffusion coefficient (blue curves) and (ii) a diffusion coefficient that shows a linear dependence on the local mass fraction (red curves). As can be seen, the general agreement between the experimental data and the calculated profiles is good in both cases, indicating the adequacy of the methodology.

Profiles of calculated interfacial mass flux of absorbed refrigerant are shown in Figures 4.a and 4.b. In both cases, the general trend is a decrease in the rate of absorption with time. The zones of instability in both curves are associated with the onset of natural convection, which promotes mixing of the liquid and accelerates the absorption process.

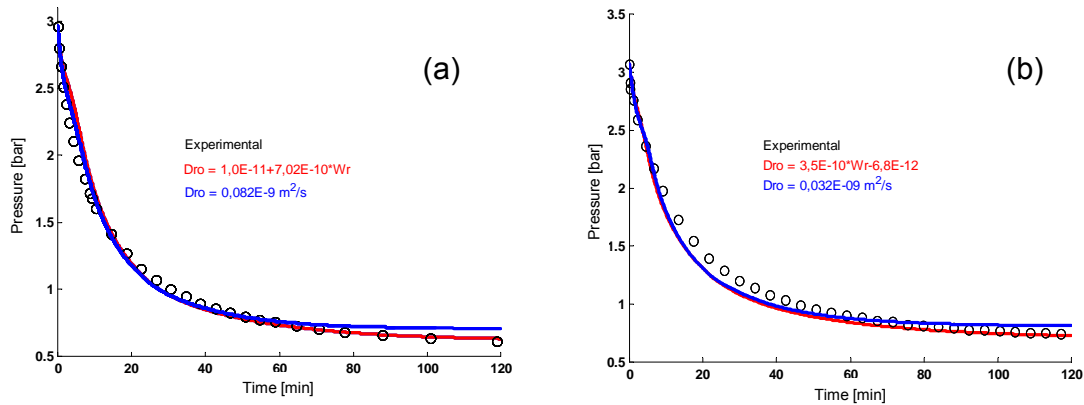


Figure 3: Experimental and calculated pressure decay curves as a function of the molecular diffusivity formulation. (a) R12/Mineral oil at 23.5°C, (b) R134a/Polyol Ester oil at 23.4°C.

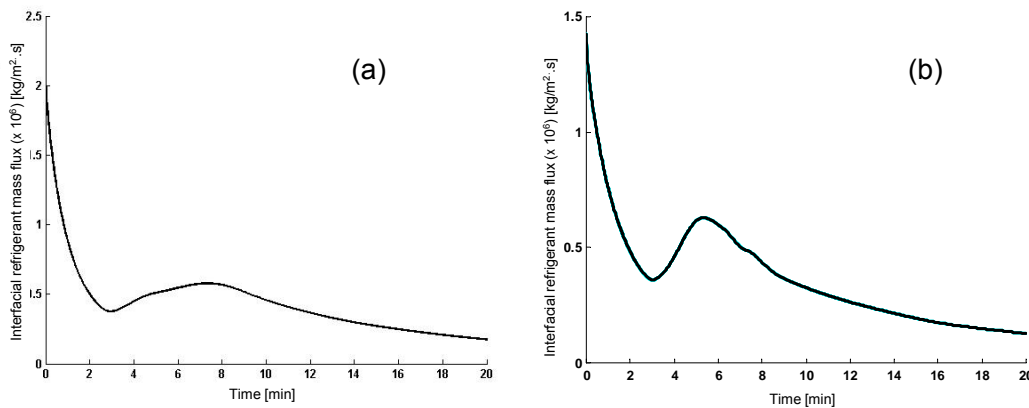


Figure 4: Calculated interfacial mass flux of absorbed refrigerant. (a) R12/Mineral oil at 23.5°C, (b) R134a/Polyol Ester oil at 23.4°C.

3. LUBRICATION IN THE PRESENCE OF DEGASSING

3.1 The Problem

Studies on the performance of high-speed journal bearings indicate that a consideration of the presence of non-condensable gas bubbles entrained in the film led to important advances in comprehending the behavior of journal bearings lubricated by an aerated film (Qi et al., 1997; Chun, 2002). In refrigerant-oil mixtures, depending on the amount of dissolved refrigerant and on the rate of decompression, a significant amount of refrigerant vapor may be released in the form of bubbles. Compared with the non-condensable bubbles situation, the level of difficulty is greatly increased as the local amount of free gas is not known and one must rely on thermodynamic calculations to determine the local gas fraction.

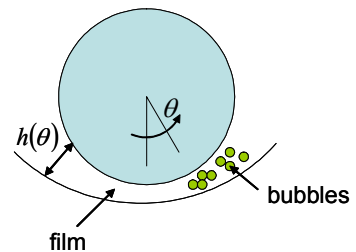


Figure 5: The problem geometry

The problem under consideration is shown in Figure 5. It consists of an infinite partial journal bearing filled with a refrigerant-oil mixture of known overall composition. The problem is considered symmetrical, one-

dimensional, isothermal and steady-state. Thermodynamic equilibrium is also assumed.

The flow is governed by the Reynolds equation

$$\frac{\partial}{\partial \theta} \left(\frac{\bar{\rho} h^3}{12 \bar{\mu}} \frac{\partial p}{\partial \theta} \right) = R \frac{\partial}{\partial \theta} \left(\frac{\bar{\rho} h U}{2} \right) \quad (8)$$

where, since two-phase flow is allowed to take place, $\bar{\rho}$ and $\bar{\mu}$ are two-phase homogeneous density and dynamic viscosity given by (Carey, 1992)

$$\bar{\rho} = \phi \rho_g + (1 - \phi) \rho_l; \quad \mu = \chi \mu_g + (1 - \chi) \mu_l \quad (9)$$

The gas volume fraction is thus,

$$\phi = \frac{\rho_l \chi}{\rho_g + (\rho_l - \rho_g) \chi} \quad (10)$$

and, by assuming local equilibrium between the refrigerant and the non-volatile oil,

$$\chi = \frac{w_i - w}{1 - w}. \quad (11)$$

4.2 Implementation and Solution

The conditions are such that the inlet pressure is always known and the eccentricity is kept constant at 0.75. The mixture tested was a 2.3% (mass) R134a/polyol ester oil assumed saturated at the inlet. The following solution procedure was adopted to account for two-phase flow in the bearing: (i) an initial pressure distribution is guessed, (ii) the thermodynamic and physical properties along the film are calculated based on the guessed profile, (iii) If $p < p_{sat}(w)$, degassing occurs and there is an increase in the volume fraction of gas (bubbles) through expansion and degassing, (iv) Having estimated the new properties of the mixture, an updated pressure distribution is calculated, (v) the process is repeated until convergence is obtained for the calculated pressures. The procedure was compared with different cavitation conditions.

4.3 Results

Figure 6 shows a comparison of calculated pressure distributions along the bearing for different outlet pressure conditions using the proposed algorithm. The pressure distribution obtained using the Reynolds cavitation condition is also illustrated.

As can be seen, in the cases simulated using the two-phase flow algorithm there is a region of negative pressure that will lead to refrigerant degassing from the liquid mixture. From this point onwards ($\theta \approx 200^\circ$), the lubricant ceases to be single phase liquid and becomes a two-phase mixture. The two-phase flow algorithm enables one to compute the amount of free refrigerant (gas phase) and also the effect of refrigerant degassing on the physical properties of the liquid phase.

Depending on the value assigned to the outlet pressure in the test cases of Figure 6, the pressure curves exhibit a recovery towards the end of the bearing. This recovery is maximum for the case in which the inlet and outlet pressures are equal, i.e., the more physically meaningful situation. Under equilibrium conditions, the pressure recovery is associated with re-absorption of the free refrigerant by the liquid mixture. It is, however, not likely that such an extreme situation will take place in a real bearing due to the finite mass transfer resistances within the liquid phase and to the departure from a quasi-static process. Since it is impossible at present to determine, from a thermodynamic point of view, the amount of refrigerant absorbed by the liquid during the pressure recovery, two extreme cases are studied: (i) One in which the re-absorption is complete (total equilibrium limit) and (ii) another where no re-absorption is allowed (total non-equilibrium limit).

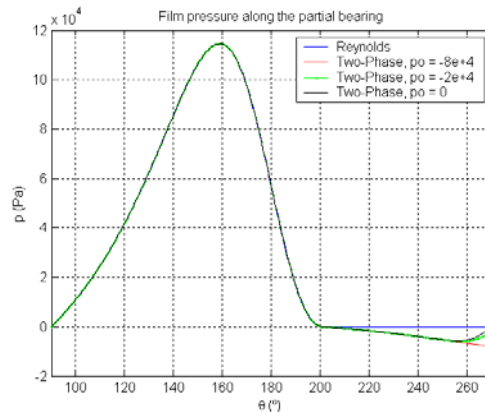


Figure 6: Calculated pressure distributions along the bearing.

Figure 7 shows profiles of local equilibrium liquid phase refrigerant mass fraction as a function of distance along the bearing for the total equilibrium and the total non-equilibrium cases. Calculated profiles of vapor mass fraction and of vapor volume fraction along the bearing are shown in Figures 8 and 9, respectively. The variation of vapor mass fraction, which corresponds to the mass of gas released from the mixture, is extremely small. The mass of gas is only 0.35% of the total mass of the mixture in the condition of maximum gas released. In terms of gas volume fraction, however, the significant expansion capacity of the gas upon release is confirmed. In volumetric terms, the gas can respond for up to 65% of the volume of the mixture for the present conditions.

As can be seen in Figure 9.b, the gas volume fraction (and hence the homogeneous density) can change even if no additional gas is released. This is due to the high compressibility of the gas. As soon as pressure is recovered towards the end of the bearing, the density of the gas increases, thus reducing void fraction and increasing the apparent density of the liquid-gas mixture.

The net effect of the total equilibrium and total non-equilibrium hypotheses on the pressure distribution along the partial bearing can be evaluated in Figure 10. As a result of differences in physical properties and in two-phase flow parameters, pressures are slightly lower in the total equilibrium case than those for the total non-equilibrium case.

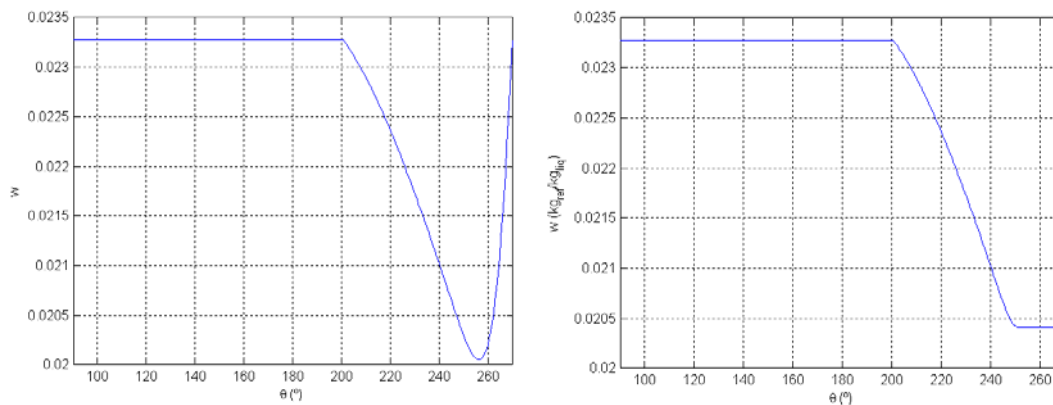


Figure 7: Refrigerant mass fraction. (a) Total equilibrium, (b) Total non-equilibrium.

4. CONCLUSIONS

Two studies involving two-phase flow and phase change phenomena with direct implications on compressor operation and design were presented. In the first study, a 2D analysis of the absorption of refrigerant by lubricant oil was presented, which took into account the variation of the height of the liquid layer and natural convection

phenomena. The second study investigated lubrication in an infinite partial journal bearing in the presence of vaporization of refrigerant dissolved in the oil. Results indicated that, in this case, local thermodynamic equilibrium in the cavitation region does not significantly influence the pressure distribution along the bearing.

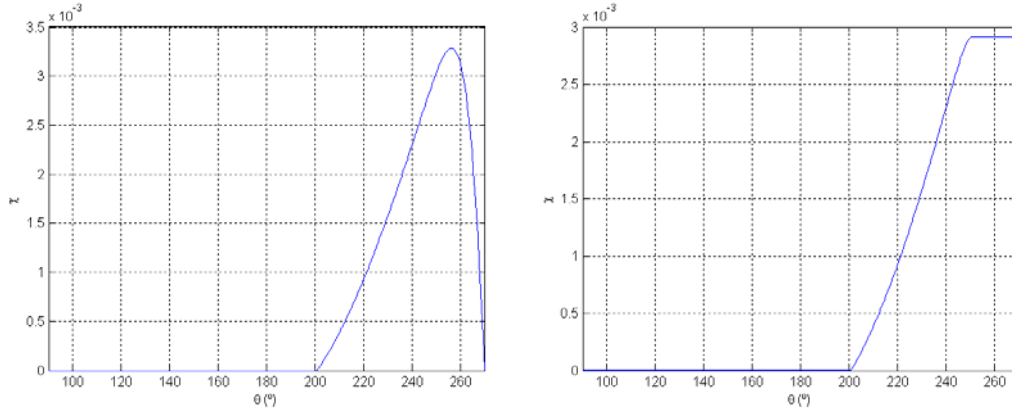


Figure 8: Gas mass fraction. (a) Total equilibrium, (b) Total non-equilibrium.

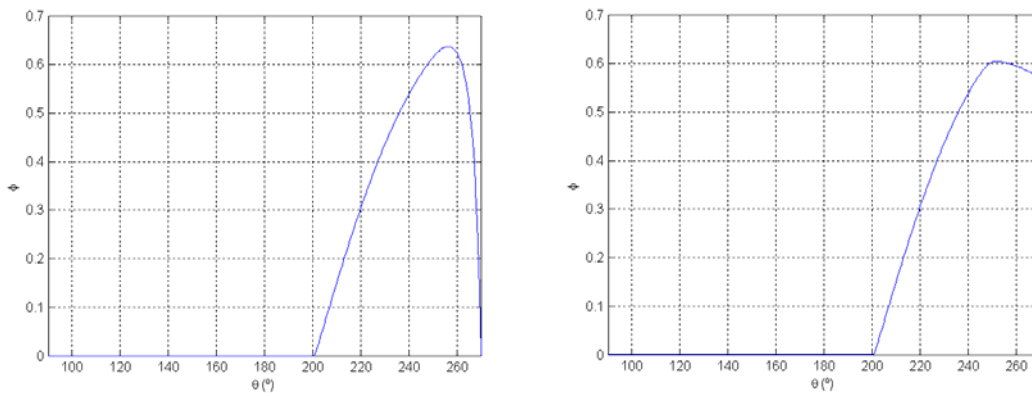


Figure 9: Gas volume fraction. (a) Total equilibrium, (b) Total non-equilibrium.

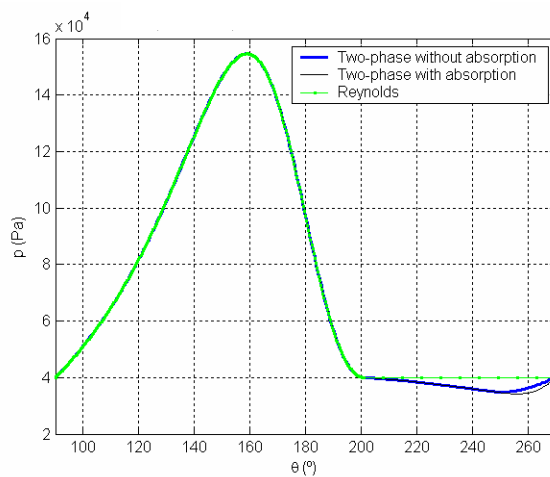


Figure 10: Pressure distribution as a function of equilibrium hypothesis

NOMENCLATURE

| | | | | |
|----------|-----------------------------|----------------------|-------------------|-------------|
| D | molecular diffusivity | (m ² /s) | Subscripts | |
| g | acceleration of gravity | (m/s ²) | g | gas |
| h | height | (m) | l | liquid |
| P | pressure | (bar) | O | oil |
| r | radial coordinate | (m) | R | refrigerant |
| R | test cell or bearing radius | (m) | int | interface |
| t | time | (s) | m | mixture |
| v | velocity | (m/s) | | |
| V | volume | (m ³ /s) | | |
| w | mass fraction | (-) | | |
| z | axial coordinate | (m) | | |
| Z | dimensionless coordinate | (-) | | |
| μ | viscosity | (kg/m.s) | | |
| ρ | density | (kg/m ³) | | |
| θ | angular coordinate | (rad) | | |

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