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Oil-Refrigerant R134a Mixture Non-Isothermal Two-Phase Flow Through the Radial Clearance of Rolling Piston Compressors

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ABSTRACT

The purpose of this work is to present a non-isothermal two-phase model to predict the ester-oil/refrigerant-R134a mixture flow through the radial clearance of rolling piston compressors in order to estimate the refrigerant gas leakage. The mass, momentum, and energy equations are analytically integrated through the cross section direction to result in a set of one-dimensional equations, which are numerically integrated in the longitudinal direction to provide pressure, temperature, and refrigerant mass fraction profiles along the flow. The model also allows the calculation of local quality, void fraction, density, and kinematic viscosity. In addition, the refrigerant mass flow rate is estimated based on the mixture mass flow rate and the refrigerant mass fraction along the flow. The flow is divided in two regions: a single-phase region, where the refrigerant mass fraction is lower than the refrigerant solubility in the oil, and a two-phase region, where the refrigerant mass fraction becomes equal to the refrigerant solubility. Due to the low mass flux, the homogeneous model is used to simulate the flow in the two-phase region. Results are obtained for inlet pressures between 450 and 600 kPa, inlet temperatures around 35°C, and minimal clearances (the minimal dimension of the radial clearance) between 10 and 60µm. The results depict that there is a sudden pressure drop at the minimal clearance region, which is accompanied by a large temperature reduction due to the refrigerant outgassing. The results obtained with the non-isothermal model are compared to those calculated by using an isothermal model, showing that the refrigerant mass flow rates evaluated by the isothermal model are over estimated in 24% in average.

1. INTRODUCTION

The volumetric efficiency of the rolling piston compressor is related to the gas refrigerant leakage, clearance space, suction gas heating, return of the gas through the discharge valve and lubricant flow. It is well known that the leakage through the radial clearance in this type of compressor constitute the major contribution for the internal mass flow losses. It accounts for almost 70% of the total internal gas leakage (Krueger, 1988). Therefore, a good understanding of all phenomena involved in this leakage is of fundamental importance in order to estimate and enhance the volumetric efficiency of the compressor. Former researches have reported that the radial clearance is completely filled with the lubricant oil. As this lubricant oil is always in contact with the refrigerant gas inside the compressor, in fact the radial clearance is filled with a mixture composed by the oil and the refrigerant, because these fluids are mutually miscible. Most previous researchers have modeled this leakage as an isothermal single phase flow of pure refrigerant gas or pure lubricant oil. Pandeya and Soedel (1978), Yanagisawa and Shimizu (1985), Xiuling *et al.* (1992), Zhen and Zhiming (1994), Huang (1994), Lee and Min (1988), and Leyderman and Lisle (1995) are some of the representative literature on this subject.

However, Costa *et al.* (1990) have accomplished an experiment in order to visualize the flow through the radial clearance, in which they have firstly observed the existence of a liquid film along the clearance. Secondly, they also have noticed a large amount of tiny bubbles just downstream the minimal value of the clearance, also denominated by minimal clearance, δ . They have concluded that a flow model with phase change would be more appropriate to study this flow.

Gasche (1996), having in mind the experiment performed by Costa *et al.* (1990), has developed several flow models to predict the flow through the radial clearance. However, all those flow models have been considered isothermal, and one should ask if the isothermal flow model is suitable for predicting this flow, or if the energy equation was

introduced in the flow modeling, the refrigerant gas leakage calculation would significantly be altered. The purpose of this work is to respond those questions.

2. MATHEMATICAL MODELING

The geometry for studying the oil-refrigerant mixture flow through radial clearance is shown in Figure 1.

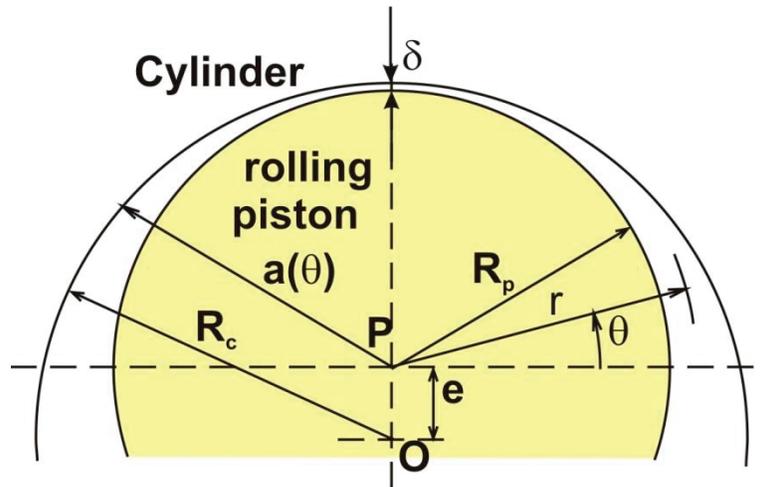


Figure 1: Geometry of the problem

The parameter $a(\theta)$ is given by:

$$a(\theta) = \sqrt{e^2 \sin^2 \theta + R_c^2 - e^2} - e \sin \theta \quad (1)$$

where the eccentricity, e , is calculated by $e = R_c - a(\theta = \pi/2)$. The minimal clearance, δ , is given by the difference between $a(\theta = \pi/2)$ and R_p , that is, $\delta = a(\theta = \pi/2) - R_p$.

Neglecting the curvature effect of the clearance on the flow, because $\delta \ll R$, one can transform the cylindrical geometry to Cartesian geometry to represent the radial clearance, as shown in Figure 2.

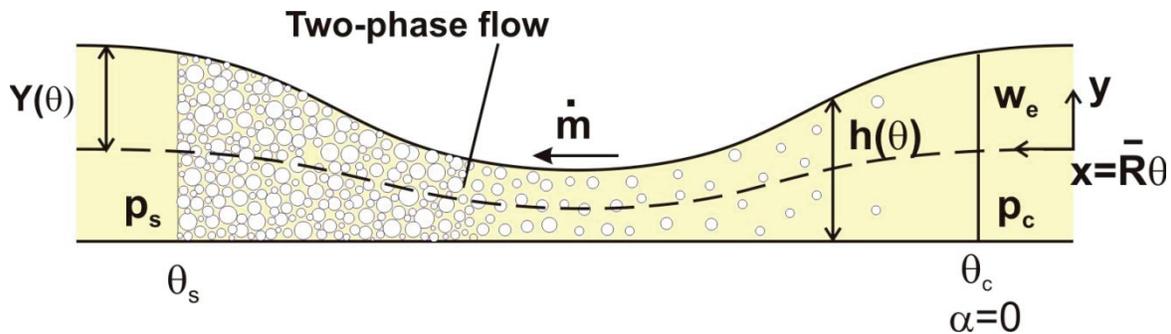


Figure 2: Cartesian geometry of the radial clearance

The height of the clearance is calculated by subtracting R_p from $a(\theta)$, which results in the following equation:

$$h(\theta) = \sqrt{e^2 \sin^2 \theta + R_c^2} - e \sin \theta - R_p \quad (2)$$

As can be seen in Figure 2, the radial clearance is considered filled with the mixture from $\theta = \theta_c$, representing the compression side of the compressor, to $\theta = \theta_s$, representing the suction side.

The following assumptions were considered for modeling the flow: (a) the flow is one-dimensional and fully developed, (b) flow is steady-state, (c) the clearance is insulated and impermeable, (d) the liquid phase is formed by oil and liquid refrigerant, while the vapor phase is considered to be formed by refrigerant vapor only, (e) for the two-phase flow region, the liquid phase is considered always saturated, and (g) the mixture is treated as an ideal mixture.

2.1 Flow characterization

Lacerda *et al.* (2000), Poiate Jr. (2006) and Castro (2006) have accomplished experiments of flow visualization of oil-refrigerant mixture flows along horizontal tubes. In these experiments the authors have used a saturated mixture at the inlet of the flow, that is, the refrigerant concentration was equal to its solubility in the oil. Despite of that, they have observed a single-phase mixture flow in the inlet region of the flow. Only in some extension of the flow they have visualized the two-phase flow, which means that there are two flow regions along the tube: a single-phase flow in the inlet region of the tube and a two-phase flow downstream the tube.

In this work, the liquid mixture is considered always saturated with refrigerant. This means that any pressure drop produces the reduction of the refrigerant solubility in the oil, which promotes the immediate outgassing, giving origin to the two-phase flow. In order to simulate the single-phase region visualized in the experiments, one assume that the mixture enters under-saturated at the inlet of the clearance, having a concentration lower than the solubility of the refrigerant in the oil for the inlet pressure and temperature, that is:

$$w_e = Factor \cdot w_{sat}(p_e, T_e) \quad (3)$$

where w_e is the refrigerant concentration in the oil at the tube inlet, $w_{sat}(p_e, T_e)$, is the solubility (concentration for the saturated mixture) at the inlet pressure and temperature, and *Factor* is a parameter to measure the under-saturation amount. Based on the experiments of those authors, the *Factor*=0.9 was used in this work.

Consequently, the flow modeling will produce a single-phase flow having a constant concentration, lower than the local solubility in the inlet region of the clearance. Due to the frictional pressure drop, the local solubility decreases until the inlet concentration is reached, w_e . After this point, the mixture becomes saturated, and any further pressure reduction gives origin to refrigerant vapor bubbles and, therefore, to a two-phase flow.

2.2 Governing equations

Considering a one-dimensional, non-isothermal steady flow, the governing equations of the flow problem are the continuity, momentum, and energy equations. The lumped form of the continuity equation for this problem states that the mass flow rate is constant. Momentum and energy equations are given by:

$$\frac{dp}{d\theta} = -\frac{12\bar{R}\mu\dot{m}}{\rho H_p h^3(\theta)} \quad (4)$$

$$\frac{dT}{d\theta} = \frac{\frac{1}{\rho} \frac{dp}{d\theta} - \frac{d}{d\theta} (x_{i_{vl}}) - \left(\frac{\partial i_l}{\partial p} \right)_T \frac{dp}{d\theta}}{\left(\frac{\partial i_l}{\partial T} \right)_p} \quad (5)$$

where $\bar{R} = [R_p + a(\theta)]/2$, \dot{m} is the mixture mass flow rate, μ is the local absolute viscosity, ρ is the local density, H_p is the clearance width, x is the local quality, and i is the local enthalpy. The subscripts v and l stand for vapor and liquid phases, respectively.

2.3 Physical and thermodynamic properties calculation

In order to calculate the local pressure gradient using Equation (4), two physical properties are needed: density and absolute viscosity of the mixture. In the single-phase flow region, these properties are calculated for the liquid mixture as a function of the inlet concentration, given by Castro (2006).

However, for the two-phase flow region, ρ and μ must be substitute by the average properties of the two-phase flow, $\bar{\rho}$ and $\bar{\mu}$. The flow density is given by:

$$\bar{\rho} = \alpha \rho_v + (1 - \alpha) \rho_l \quad (6)$$

The vapor phase (v) is considered to be formed only by refrigerant, due to the lower vapor pressure of the oil. Instead, the liquid phase (l) is considered to be formed by a saturated oil-refrigerant mixture. The parameter α is the void fraction, which is calculated by Equation (7) if the homogeneous model is used to represent the two-phase flow.

$$\alpha = \frac{1}{[1 + (1/x - 1) \rho_v / \rho_l]} \quad (7)$$

As the liquid mixture is always saturated by refrigerant, the local quality, x , is calculated by Equation (8).

$$x = \frac{w_e - w_{sat}(p, T)}{1 - w_{sat}(p, T)} \quad (8)$$

where w_{sat} is the local refrigerant solubility, given as a function of the local pressure and temperature of the mixture, $w_{sat} = w(p, T)$.

There are several ways for calculating the average viscosity of the flow. Some of them are attributed to Akers *et al.* (1959) *apud* Yan and Lin (1998), Cichitti (1960) and Dukler (1964) *apud* Collier (1981), Davidson (1943) *apud* Chang and Ro (1996), Isbin *et al.* (1958) *apud* Whalley (1987) and Lin (1991) *apud* Wongwises and Pirompak (2001). Dias (2006) has shown that the correlation proposed by Dukler (1964), given by Equation (9), has produced good results for oil-refrigerant R134a mixture flows in tubes and it is used in this work. In this equation v is the specific volume.

$$\bar{\mu} = \frac{xv_v \mu_g + (1 - x)v_l \mu_l}{xv_v + (1 - x)v_l} \quad (9)$$

For a given mass flow rate, \dot{m} , an initial pressure, p_e , and an initial temperature, T_e , the pressure and temperature profiles along the clearance can be obtained integrating Equations (4) and (5), respectively, by using the Euler method.

3. NUMERICAL RESULTS

The model described earlier was applied to the flow of a mixture composed by an ester-oil ISO VG10 and refrigerant R134a. The results obtained from the problem solution were refrigerant mass flow rate and longitudinal profiles for pressure, temperature, refrigerant concentration, vapor quality, void fraction and dynamic viscosity. In

addition, a parametric analysis involving inlet pressure and minimal clearance, δ , was performed. A comparison with the results obtained by using the isothermal model was also accomplished.

Figure 3 shows the longitudinal profiles for the main flow variables. One can observe the abrupt variation of the pressure profile, which is caused by the cross section area variation and viscous forces. In addition, one can notice a large temperature drop in Figure 3(b) as result of the liquid-vapor phase change. All other variables behavior comes from the pressure and temperature profiles.

These figures were also prepared to depict the influence of the inlet pressure on both mixture mass flow rate and refrigerant mass flow rate (refrigerant leakage), which is determined by Equation (1). As expected, the refrigerant leakage increases for increasing inlet pressure due to two effects: the mixture mass flow rate increases when the total pressure drop is enhanced, and the inlet refrigerant mass fraction increases for increasing inlet pressure, which causes larger outgassing for the same suction pressure, which also explains the augmentation of the total temperature drop presented in Figure 3(b). Table 1 presents refrigerant leakages for other operational conditions. Another result also expected is the increase of the refrigerant leakage for larger minimal clearance values.

$$\dot{m}_{ref} = \dot{m} \frac{w_c - w_s}{1 - w_s} \quad (10)$$

The results in Table 1 also show that the isothermal model over predicts the refrigerant mass flow rate by 24 % in average. Therefore, for the operational conditions analyzed in the present work, one can conclude that the oil-refrigerant mixture flow through the radial clearance can not be considered an isothermal flow for predicting the refrigerant leakage through the radial clearance in rolling piston compressors.

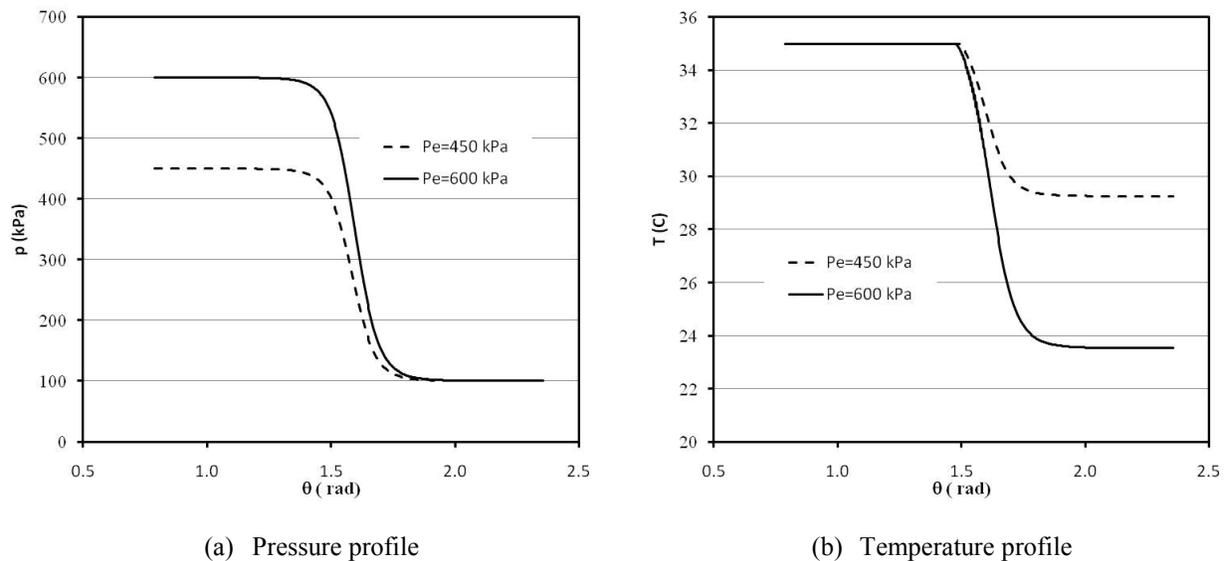
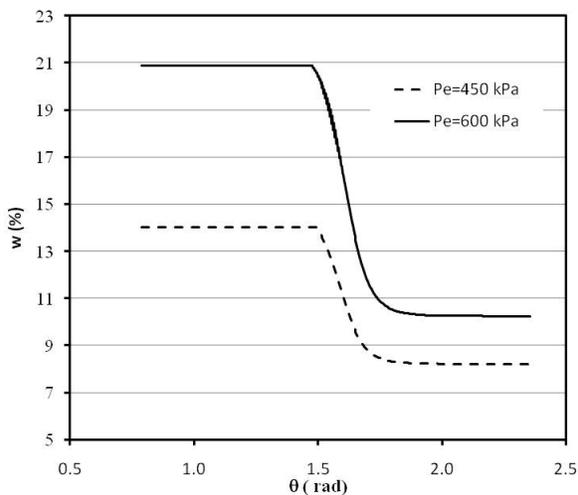
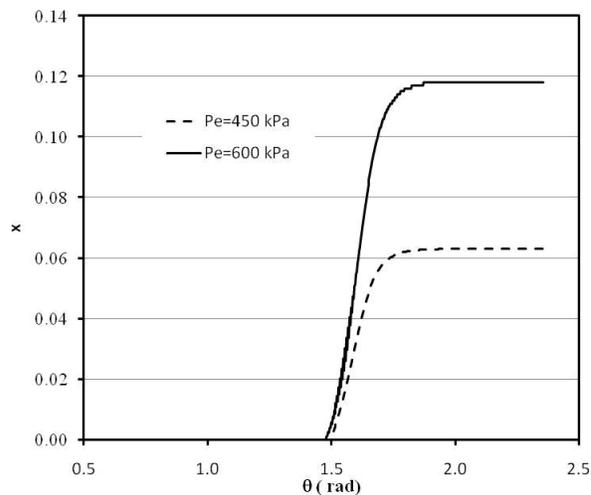


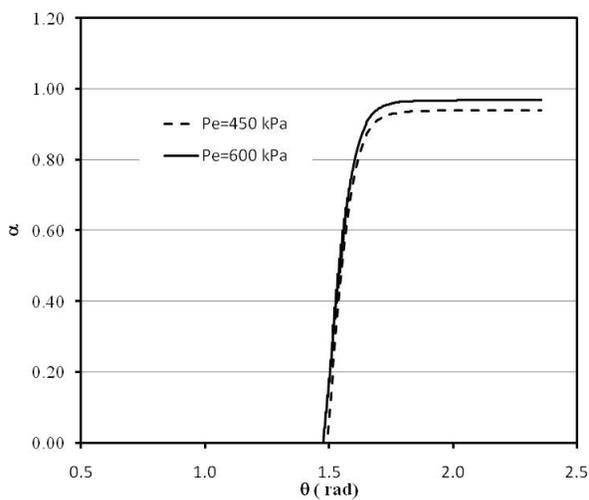
Figure 3: Influence of the inlet pressure for $T_c=35$ °C and $\delta=30$ μm



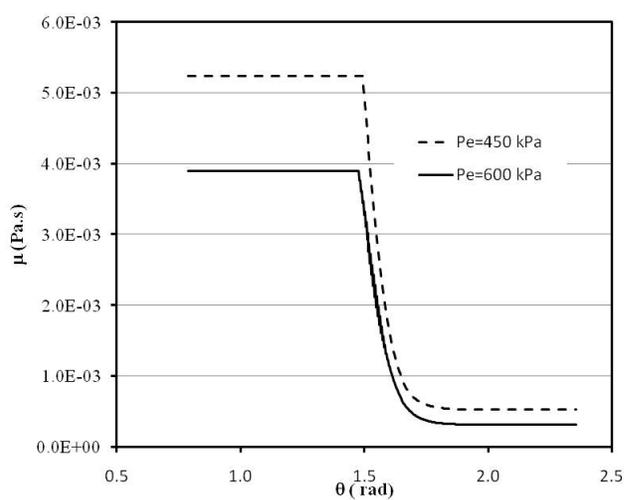
(c) Refrigerant mass fraction profile



(d) Vapor quality profile



(e) Void fraction profile



(f) Dynamic viscosity profile

Figure 3 (cont.): Influence of the inlet pressure for $T_c=35$ °C and $\delta=30$ μ m

Table 1: Refrigerant mass flow rate calculated by both isothermal and non-isothermal models

| P_e (kPa) | P_s (kPa) | ΔP (kPa) | T_e ($^{\circ}\text{C}$) | δ (μm) | Isothermal Model | | Non-isothermal Model | | \dot{m}_{ref} difference % |
|----------------|----------------|---------------------|---------------------------------|-------------------------------|---------------------|--------------------------|----------------------|--------------------------|------------------------------------|
| | | | | | \dot{m} (kg/h) | \dot{m}_{ref} (g/s) | \dot{m} (kg/h) | \dot{m}_{ref} (g/s) | |
| 450 | 100 | 350 | 35 | 10 | 0.2100 | 0.00476 | 0.2058 | 0.00361 | 24 |
| | | | | 20 | 1.1846 | 0.02684 | 1.1607 | 0.02035 | 24 |
| | | | | 30 | 3.2566 | 0.07377 | 3.1904 | 0.05593 | 24 |
| | | | | 40 | 6.6705 | 0.15111 | 6.5346 | 0.11457 | 24 |
| | | | | 50 | 11.6289 | 0.26344 | 11.3912 | 0.19969 | 24 |
| | | | | 60 | 18.3076 | 0.41474 | 17.9325 | 0.31427 | 24 |
| 500 | 100 | 400 | 35 | 10 | 0.2528 | 0.00720 | 0.2482 | 0.00545 | 24 |
| | | | | 20 | 1.4255 | 0.04059 | 1.3992 | 0.03071 | 24 |
| | | | | 30 | 3.9183 | 0.11157 | 3.8458 | 0.08443 | 24 |
| | | | | 40 | 8.0255 | 0.22852 | 7.8765 | 0.17288 | 24 |
| | | | | 50 | 13.9905 | 0.39836 | 13.7301 | 0.30135 | 24 |
| | | | | 60 | 22.0249 | 0.62714 | 21.6141 | 0.47451 | 24 |
| 550 | 100 | 450 | 35 | 10 | 0.3016 | 0.010062 | 0.2980 | 0.00806 | 20 |
| | | | | 20 | 1.7005 | 0.05986 | 1.6796 | 0.04542 | 24 |
| | | | | 30 | 4.6738 | 0.16452 | 4.6164 | 0.12479 | 24 |
| | | | | 40 | 9.5723 | 0.33696 | 9.4543 | 0.25553 | 24 |
| | | | | 50 | 16.6861 | 0.58737 | 16.4798 | 0.44545 | 24 |
| | | | | 60 | 26.2677 | 0.92465 | 25.9426 | 0.70118 | 24 |
| 600 | 100 | 500 | 35 | 10 | 0.3585 | 0.01542 | 0.3586 | 0.01179 | 24 |
| | | | | 20 | 2.0206 | 0.08692 | 2.0210 | 0.06650 | 23 |
| | | | | 30 | 5.5531 | 0.23888 | 5.5542 | 0.18275 | 23 |
| | | | | 40 | 11.3722 | 0.48921 | 11.3744 | 0.37438 | 23 |
| | | | | 50 | 19.8227 | 0.85274 | 19.8281 | 0.65237 | 23 |
| | | | | 60 | 31.2041 | 1.34235 | 31.2125 | 1.02698 | 23 |

4. CONCLUSIONS

In the present work a non-isothermal two-phase model is used to predict the ester-oil/refrigerant-R134a mixture flow through the radial clearance of rolling piston compressors in order to estimate the refrigerant gas leakage. The mass, momentum, and energy equations are analytically integrated through the cross section direction to result in a set of one-dimensional equations, which are numerically integrated in the longitudinal direction to provide pressure, temperature, and refrigerant mass fraction profiles along the flow. The flow is divided in two regions: a single-phase region, where the refrigerant mass fraction is lower than the refrigerant solubility in the oil, and a two-phase region, which is simulated using the homogeneous model. Results are obtained for inlet pressures between 450 and 600 kPa, inlet temperatures around 35 $^{\circ}\text{C}$, and minimal clearances between 10 and 60 μm . The main conclusion of the work is that the refrigerant mass flow rate through the clearance, which represents the refrigerant leakage, is over predicted

by 24 % in average if the isothermal model is used. Therefore, the non-isothermal model must be used in order to better estimate the refrigerant leakage through the radial clearance in rolling piston compressors.

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