REFRIGERANT R134a LEAKAGE THROUGH THE RADIAL CLEARANCE IN ROLLING PISTON COMPRESSORS

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Abstract. *The refrigerant gas leakage through the radial clearance is one of the main sources of volumetric efficiency loss in rolling piston compressors. Therefore, the understanding of the flow characteristics through this clearance is essential in order to improve its volumetric efficiency. Although it seems very simple, several complex phenomena are present in this flow. One important characteristic of this leakage is that the fluid flowing through the clearance is, in fact, a homogeneous liquid mixture composed by oil and refrigerant. This is an important aspect because when the mixture flows through the radial clearance from the high-pressure side to the low-pressure side, the solubility of the refrigerant in the oil decreases, producing a great number of refrigerant gas bubbles along the flow, leading to a two-phase flow of the binary mixture. In this work it is proposed a model to simulate the two-phase flow of a mixture composed by an ester oil and refrigerant R134a through the radial clearance of a rolling piston compressor. The mixture flow is modeled as a stationary homogeneous two-phase flow and the resultant equation is solved by the method of Runge-Kutta. Velocities and pressure profiles, and the amount of gas leakage are presented as a function of the minimal clearance width and the mixture mass flow rate. The influence of the tangential velocity of the cylinder is also investigated. It has been found that the mass flow rate of the refrigerant gas calculated using two-phase flow model can be up to about 37% lower than the mass flow rate of the refrigerant gas estimated using single-phase flow model.*

Keywords*: Gas Leakage, Radial Clearance, Rolling Piston, Compressor, Refrigerant R134a*

1. Introduction

The volumetric efficiency of the rolling piston compressor is related to the refrigerant leakage, clearance space, suction gas heating, return of the gas through the discharge valves an oil lubricant flow. Among them refrigerant leakage is the main cause of the volumetric efficiency loss. A great amount of the refrigerant leakage occurs at the existing clearance between the external surface of the rolling piston and the internal surface of the cylinder, known by radial clearance. Figure 1 shows a typical position of the radial clearance and the main components of the rolling piston compressor pump. Krueger (1988) has estimated that about 30% of total internal loss of refrigerant gas is due to this leakage. Therefore, a good understanding of the gas leakage through the radial clearance is an important issue in order to estimate and improve the volumetric efficiency of the compressor.

A review of the main works published in the last 25 years involving leakage modeling at the radial clearance shows that for most investigations simplified models have been considered. Several authors have dealt with this problem assuming compressible flow of pure refrigerant gas. Pandeya and Soedel (1978), Yanaginawa and Shimizu (1985a), Xiuling *et al.* (1992), Zhen and Zhiming (1994) and Huang (1994) are some of the representative literature on this subject. Other authors have considered the presence of the oil at the radial clearance but have not included the influence of the refrigerant gas dissolved in the lubricant; examples are Lee and Min (1988) and Leyderman and Lisle (1995). Gasche *et al.* (1997) have presented a broad discussion about several phenomena associated with the flow through the radial clearance and have proposed various models to calculate this flow, Gasche *et al.* (1998a), Gasche *et al.* (1988b) and Gasche *et al.* (1999). In all their models Gasche *et al.* assumed that the gas leakage through the radial clearance was caused by either pure oil flow or single-phase oil-refrigerant mixture flow.

Costa *et al.* (1990) performed an experimental visualization of the flow through the radial clearance during a normal operation of the compressor. The results showed two important characteristics of the flow. They first observed that part of the clearance was always filled with a liquid fluid. Besides, they visualized a great number of bubbles in the liquid, most of them concentrated just after the minimal clearance value. Actually, this liquid is not a pure substance but a mixture of the lubricating oil and the refrigerant gas being pumped. In fact, the refrigerant is dissolved in the lubricant oil, resulting in a homogeneous mixture. The amount of refrigerant dissolved in the oil depends on both the oil temperature and pressure. As the oil-refrigerant mixture flows through the clearance, the pressure drops and the solubility of the refrigerant in the oil decreases resulting, instantaneously, in a supersaturated mixture. To reach a new equilibrium condition, part of the refrigerant dissolved in the oil converts into bubbles that break off the mixture flow at the suction chamber, causing a mass loss because it must be recompressed.

Gasche *et al.* (2000) have proposed a model to simulate the two-phase mineral oil-refrigerant R22 mixture through the radial clearance. The results showed that the bubble formation due to the solubility reduction of the refrigerant in the oil changed the flow significantly, reducing the mass flow rate of both mixture and refrigerant gas, mainly at low temperatures. It was found that the mass flow rate of refrigerant gas calculated using the two-phase model was up to 30% lower than the mass flow rate of the refrigerant gas estimated using the single-phase model.

In the present work it is performed an investigation of the gas leakage through the radial clearance using an actual mixture composed by an ester oil and refrigerant R134a. The mixture flow is modeled as a stationary homogeneous two-phase flow model. Comparison with the single-phase model is also accomplished.

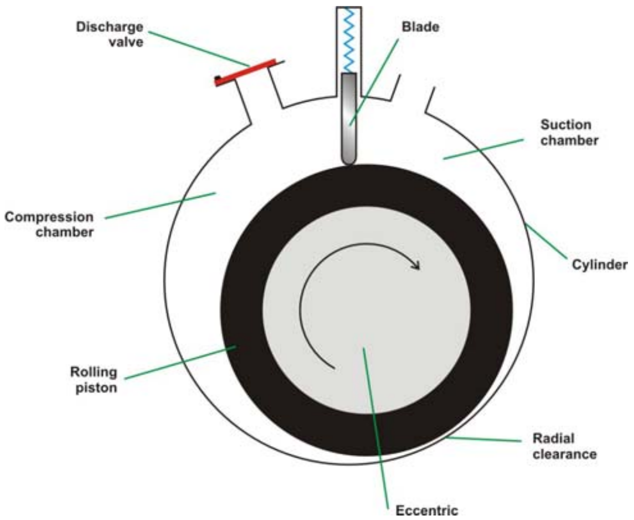


Figure 1. Schematic representation of the rolling piston compressor.

2. Problem Formulation

The geometry used to study the oil-refrigerant mixture flow at the radial clearance is showed in Fig. 2, in which only part of the radial clearance is filled with the liquid mixture. This is in accordance with the visualization performed by Costa *et al.* (1990). Also showed in the figure are the dimensions required in the problem formulation.

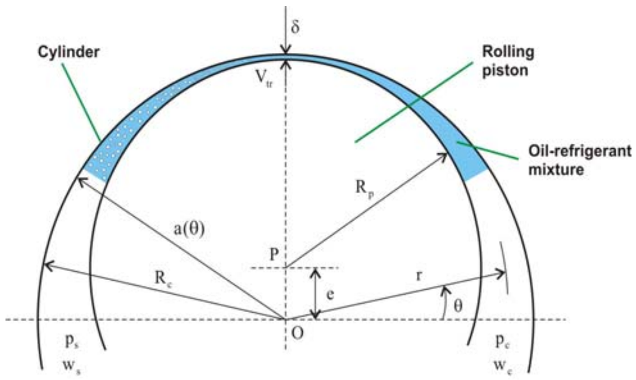


Figure 2. Geometry for the flow at the radial clearance

Momentum and continuity equations govern the isothermal oil-refrigerant mixture flow, which can be modeled as two-dimensional since the channel width, characterized by the rolling piston thickness, H_p, is much larger than the radial clearance height, δ. Furthermore, because δ is very small, fluid acceleration can be ignored and the momentum equation can be written considering only the balance between the viscous and pressure forces, as in typical lubricating problem. The following differential equation is obtained in cylindrical coordinates,

$$\frac{\partial}{\partial r}\left(\mu r \frac{\partial u}{\partial r}\right) = \frac{dp}{d\theta} \tag{1}$$

The velocity profile at any cross section can now be determined integrating Eq. (2) along the radial direction. Noticing that at $r=R_p$, $u=V_{tr}$ and at $r=a(\theta)$, $u=0$, where V_{tr} is the absolute tangential velocity at the external surface of the rolling piston, it is then obtained,

$$u = \frac{1}{\mu}\frac{dp}{d\theta}\left[r - a - (R_p - a)\frac{\ln(r/a)}{\ln(R_p/a)}\right] + \frac{V_{tr}}{\ln(R_p/a)}\ln(r/a) \tag{2}$$

In Eq. (3) the dimension $a(\theta)$ is defined as,

$$\mathbf{a}(\theta) = \sqrt{e^2\,\mathrm{sen}^2\,\theta + R_c^2 - e^2} - e\,\mathrm{sen}\,\theta \tag{3}$$

Notice that the second term of Eq. (2) represents the contribution of the tangential velocity for the development of the flow. The longitudinal profile of pressure can be obtained through of integration of the velocity profile along the cross section of channel, that is,

$$\dot{m} = \int_{R_p}^{a(\theta)} \rho u H_p\, dr \tag{4}$$

The result of the integration of Eq. (4), is given by the following expression,

$$\frac{dp}{d\theta} = -\frac{\left[\frac{\dot{m}}{\rho H_p} + V_{tr}a\left(\frac{1-(R_p/a)}{\ln(R_p/a)} + \frac{R_p}{a}\right)\right]\frac{2\mu}{a^2}}{\mathbf{1} - \left(\frac{R_p}{a}\right)^2 + \frac{2(1-R_p/a)^2}{\ln(R_p/a)}} \tag{5}$$

Equation (4) can now be integrated from the inlet over the extension of the radial clearance for a given mass flow rate, knowing the inlet pressure, p_c. Using an iterative method, this mass flow rate is chosen in order to give the required suction pressure at the end of the clearance, p_s.

The mixture flow through the radial clearance is taken as a two-phase flow, in which the bubble formation is due to the reduction of refrigerant solubility in the oil, w. In accordance with the experimental visualization performed by Costa *et al.* (1990), the homogeneous model appears to be a good approximation to represent this flow. In the homogeneous model the two-phase flow behaves as a single-phase flow having physical properties whose values are, in some sense, mean values for the flow. Yanagisawa and Shimizu (1985b) proposed the following equations to calculate these mean values at any position along the flow,

$$\rho = \alpha\rho_g + (1-\alpha)\rho_l \tag{6}$$

$$\mu = \alpha\mu_g + (1-\alpha)\mu_l \tag{7}$$

where ρ_g is the gas density and ρ_l is the liquid mixture density; μ_g is the gas viscosity and μ_l is the liquid mixture viscosity. The void fraction α is the ratio of the area occupied by the gas to the area occupied by the liquid mixture at each particular cross section. The void fraction is obtained from the quality, x, using the following equation,

$$\alpha = \frac{1}{1 + \left(\frac{1}{x} - 1\right)\frac{\rho_g}{\rho_l}} \tag{8}$$

To determine the quality, x, it is considered that the liquid mixture remains always saturated with refrigerant. Therefore, the refrigerant in the gaseous phase at any cross section along the longitudinal direction, θ, is determined from de difference between the solubility of the refrigerant at the inlet, w_c, and solubility of the refrigerant at that particular section, w. A mass balance from de inlet to a generic position along the flow yields the following equation for the local quality, x, as function of the local refrigerant solubility, w,

$$x = \frac{w_c - w}{1 - w} \tag{9}$$

where w_c is the refrigerant solubility in the oil at the compression chamber conditions, that is, T_{mix} and p_c.

The equations proposed by Silva (2004) were used to calculate the refrigerant solubility, w, the liquid mixture density, ρ_l, and the liquid mixture viscosity, μ_l, and are given by Eqs. (10), (11), and (12), respectively.

$$w = \frac{a_1 + b_1 p + c_1 T + d_1 p^2 + e_1 T^2 + f_1 pT}{a_2 + b_2 p + c_2 T + d_2 p^2 + e_2 T^2 + f_2 pT} \tag{10}$$

where,

$a_1 = 0.68247268$	$a_2 = 1.0$
$b_1 = 0.0700619$	$b_2 = -0.00313147$
$c_1 = 0.06991081$	$c_2 = 0.05031545$
$d_1 = -0.00012087$	$d_2 = 1.05413714\text{x}10^{-6}$
$e_1 = -0.00171566$	$e_2 = 0.00136449$
$f_1 = 0.00241240$	$f_2 = -6.40745705\text{x}10^{-5}$

$$\rho_l = \frac{\rho_0}{1 + w\left(\frac{\rho_0}{\rho_{rl}} - 1\right)} \tag{11}$$

$$\mu_l = \frac{a_1 + b_1 + c_1 w + d_1 T^2 + e_1 w^2 + f_1 wT}{a_2 + b_2 + c_2 w + d_2 T^2 + e_2 w^2 + f_2 wT} \tag{12}$$

In the Eq. (12) the constants are given by:

$a_1 = 0.037096749$	$a_2 = 1.0$
$b_1 = 0.0000916029$	$b_2 = 0.053096577$
$c_1 = -0.08000517$	$c_2 = 2.230898924$
$d_1 = -2.739\text{x}10^{-7}$	$d_2 = 0.001165576$
$e_1 = 0.043492955$	$e_2 = -0.30528645$
$f_1 = -0.000060485$	$f_2 = 0.033402141$

where p is the local pressure, obtained from the solution of Eq. (5), and the mixture temperature, T_{mix}, which is considered constant in the present work. The density of pure oil pure is, ρ_o, is calculated by Eq. (13), and the liquid refrigerant density, ρ_{rl}, is calculated by Eq. (14).

$$\rho_0 = 966.43636 - 0.57391608.T - 0.00024475542.T^2 \tag{13}$$

$$\rho_{rl} = 1294.679 - 3.22131.T - 0.0123398.T^2 \tag{14}$$

The refrigerant R134a gas properties were calculated by the following equations:

$$\rho_g = a + bp + cp^2 \tag{15}$$

$$\mu_g = d + ep + fp^2 \tag{16}$$

where p is given in Pa, ρ_g in kg/m^3, and μ_g in Pa.s. These equations were obtained fitting data supplied by the commercial software REFPROP. Table 1 shows the constant values for four mixture temperatures.

Table 1 – Constant values for Eqs. (15) and (16).

Constant	T_{mix} (°C)			
	60	**75**	**90**	**95**
a	0.52639	0.2787	0.16276	0.08222
b	0.03302	0.03317	0.03256	0.0326
c	1.00541×10^{-5}	6.94149×10^{-6}	5.0994×10^{-6}	4.35762×10^{-6}
d	1.31957×10^{-5}	1.37658×10^{-5}	1.43299×10^{-5}	1.45151×10^{-5}
e	5.85636×10^{-11}	1.36752×10^{-10}	1.94095×10^{-10}	2.18475×10^{-10}
f	1.58243×10^{-13}	1.30493×10^{-13}	1.10601×10^{-13}	9.73624×10^{-14}

3. Solution Methodology

The first order differential equation for the pressure, Eq. (5), was solved by using the method of Runge-Kutta. The uniform mesh dimension used was $\pi/180$ rad. According to the Fig. 2, θ varies from zero to 180º, and the average tangential velocity of the rolling piston considering one complete revolution is give by Gasche et al. (2000) as V_{tr}=-0.05 m/s.

The compressor dimensions used are: R_p=17.662 mm, R_c=21.3225 mm, H_p=11 mm, which are referred to a small refrigeration compressor. All the calculations were performed for a mixture composed by refrigerant R134a and the ester oil EMKARATE RL10H.

4. Results and Discussions

The main goal of this work is to determine the mass flow rate of refrigerant gas that leaks from the compression chamber to the suction chamber using the two-phase flow model, and to compare it with the value obtained using the single-phase flow model. The only difference between these two models is in the computation of the physical properties. In the single-phase flow model these properties are taken as constant along the flow and are calculated using a mean value for the refrigerant solubility along the clearance.

For both models the compression pressure was fixed in p_c=1250 kPa, the suction pressure was fixed in p_s=650 kPa, and the tangential velocity in V_{tr}=-0.05 m/s. The refrigerant solubility at the inlet of the flow, w_c, was determined through the Eq. (10) for the compression pressure, p_c, and the temperature of the mixture, T_{mix}.

The Fig. 3 shows the longitudinal profiles of the main variables of the flow for a typical case. From the figure it is seen that as the liquid mixture approaches the minimum clearance ($\theta = \pi/2$) the pressure drops abruptly, reducing the refrigerant solubility in the oil. In turn gas is released from the liquid, increasing both the mixture void fraction and the kinematic viscosity.

The effect of the tangential velocity on the pressure profile was investigated and the results are shown in Fig. 4. In order to exploit the results properly only the inlet region and the minimal clearance region are shown. It is interesting to notice that exists an adverse pressure gradient for positive values of the tangential velocity. This can be explained by analyzing Eq. (5). If V_{tr} is negative, only negative pressure gradient can be obtained. However, for positive values of the tangential velocity, positive pressure gradients are possible if the second term at the numerator is greater than the first term. Obviously, this effect is greater as the tangential velocity increases.

Figure 5a depicts dimensionless velocity profiles at several positions θ along the flow for V_{tr}=10 m/s, δ=30 μm, and T_{mix}=75 °C. The mass flow rate resulted for the same pressure drop, p_c-p_s, is 2.292 g/s. It is possible to realize that exists a recirculating region at the inlet and exit regions of the flow. Due to the high tangential velocity value, which produces a great mass flow rate at a region near the rolling piston surface, there must be a recirculating region at greater radii in order to satisfy the mass conservation equation. Figure 5b shows similar velocity profiles for the same mass flow rate, mixture temperature, and minimal clearance, but for V_{tr}=-0.05 m/s. In this case, as it would be expected no recirculating region is observed.

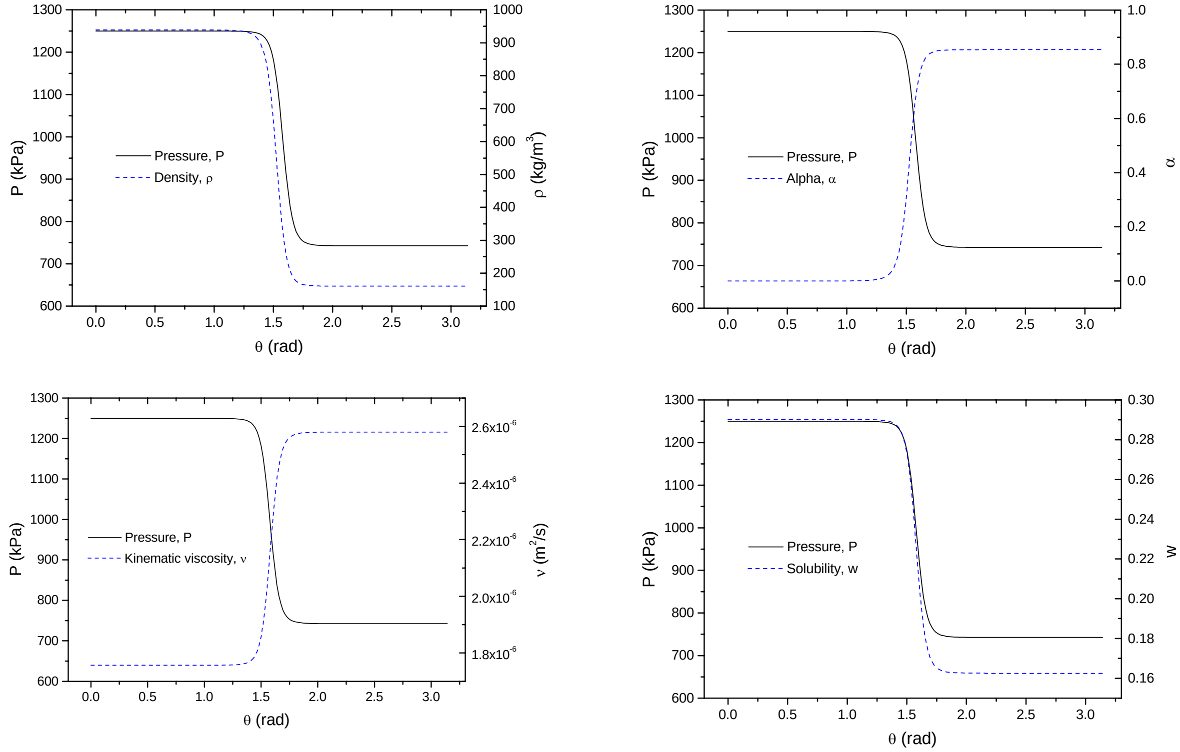


Figure 3. Values of the main instantaneous variables along the flow for δ=30 μm, T_{mix}=75 °C, and V_{tr}=-0.05 m/s.

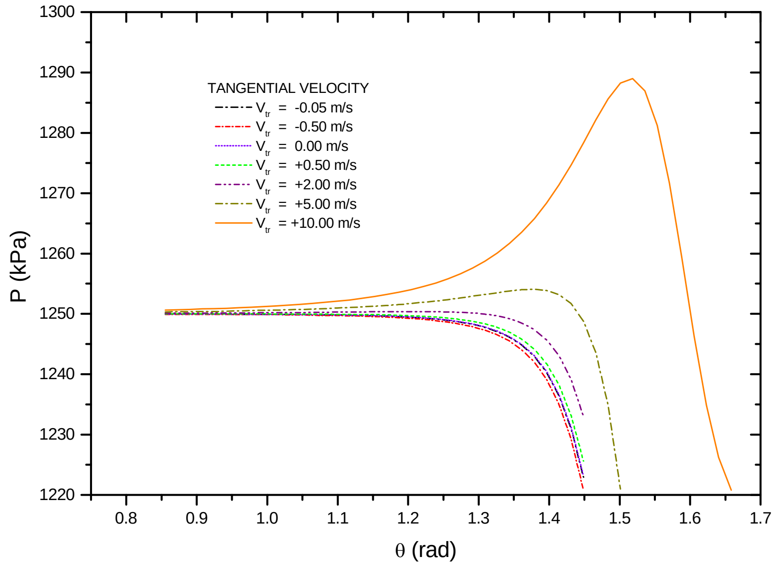
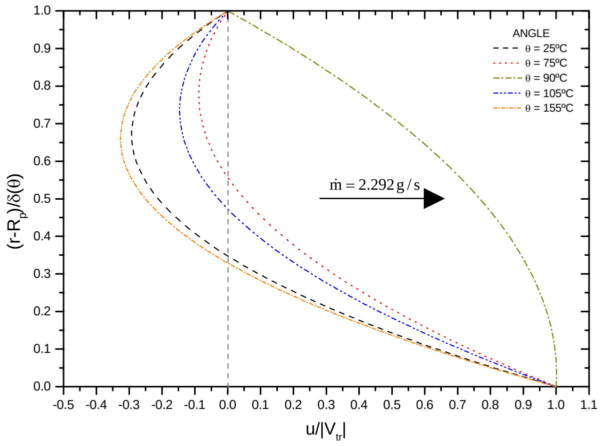


Figure 4. Effect of the tangential velocity on the pressure profiles for δ=30 μm and T_{mix}=75 °C

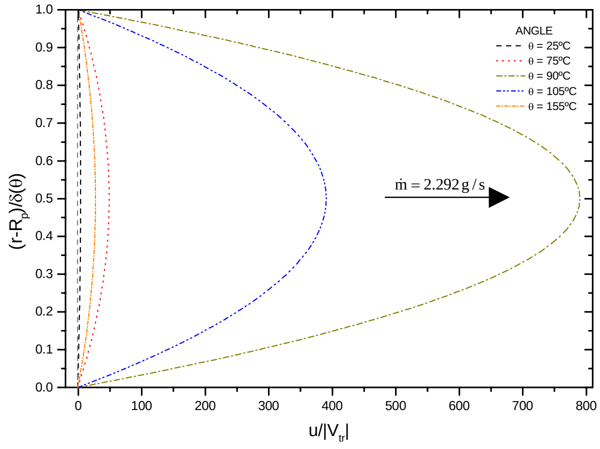
The Tab. 2 presents mass flow rate results for both the mixture and the refrigerant gas. The gas mass flow rate represents the refrigerant mass loss through the radial clearance, and it is computed from a refrigerant mass balance at the flow exit resulting in,

$$\dot{m}_{gas} = \dot{m}_{mix} \frac{w_c - w_s}{1 - w_s} \tag{16}$$

where w_s is the refrigerant solubility at the flow exit, obtained from the pressure at the suction chamber and the mixture temperature. The mixture mass flow rate, $\dot{m}_{mix}$, is chosen such that the pressure difference, p_c-p_s, is satisfied by Eq. (5).



(a) Velocity profiles for V_{tr}=10 m/s



(b) Velocity profiles for V_{tr}=-0.05 m/s

Figure 5. Velocities profiles for δ=30 μm, and T_{mix}=75 °C

Analyzing Tab. 2, it is observed that the mass flow rate do not vary linearly with the radial clearance. Another important aspect is that the mass flow rates calculated using the two-phase flow model are lower than those calculated using the single-phase flow model The same behavior is followed by the refrigerant gas leakage. For the analyzed temperature range, it is noticed that $\dot{m}_{mix}$ presents a minimum value for T_{mix}=75 ^{o}C. This result can be explained by using Fig. 6, which shows the mixture liquid viscosity. It can be seen from this figure that the liquid viscosity is maximum for that temperature, producing the minimum mass flow rate because the flow is governed mainly by the viscous term. Despite of that, the mass flow rate of the refrigerant gas do not follow the same tendency, because the ratio $(w_c\text{-}w_s)/(1\text{-}w_s)$ in Eq. (16) varies from each case, depending on the mixture temperature.

These results obtained with the two-phase flow model proposed in this work indicate that it is very important to consider the gaseous phase in evaluating refrigerant loss through the radial clearance, because the difference between the models was significant, reaching values up to about 37%.

Table 2. Mixture mass flow rate and rate of refrigerant leakage for various cases investigated.

$T_{mix} = 60^{o}C$			$w_c = 0.42163$ kg - R134a/kg - mix			
$\delta\,(\mu m)$	$\dot{m}_{mix}\,(g/s)$			$\dot{m}_{R134a}\,(g/s)$		
	Single-phase	Two-phase	$\Delta\dot{m}\,(\%)$	Single-phase	Two-phase	$\Delta\dot{m}\,(\%)$
10	0.2528	0.1612	36.2	0.07442	0.04745	36.2
30	3.973	2.52	36.6	1.1699	0.74183	36.6
50	14.225	9.019	36.6	4.18905	2.65593	36.6
$T_{mix} = 75^{o}C$			$w_c = 0.29022$ kg - R134a/kg - mix			
10	0.1815	0.1463	19.4	0.03168	0.02553	19.4
30	2.864	2.292	20.0	0.49998	0.40004	20.0
50	10.257	8.204	20.0	1.79051	1.43229	20.0
$T_{mix} = 90^{o}C$			$w_c = 0.22968$ kg - R134a/kg - mix			
10	0.1926	0.1653	14.2	0.02505	0.0215	14.2
30	3.036	2.5885	14.7	0.39503	0.3367	14.8
50	10.8721	9.265	14.8	1.41463	1.2055	14.8
$T_{mix} = 95^{o}C$			$w_c = 0.21541$ kg - R134a/kg - mix			
10	0.201	0.1745	13.2	0.02424	0.02105	13.2
30	3.1642	2.7315	13.7	0.38156	0.32937	13.7
50	11.331	9.7748	13.7	1.36648	1.17883	13.7

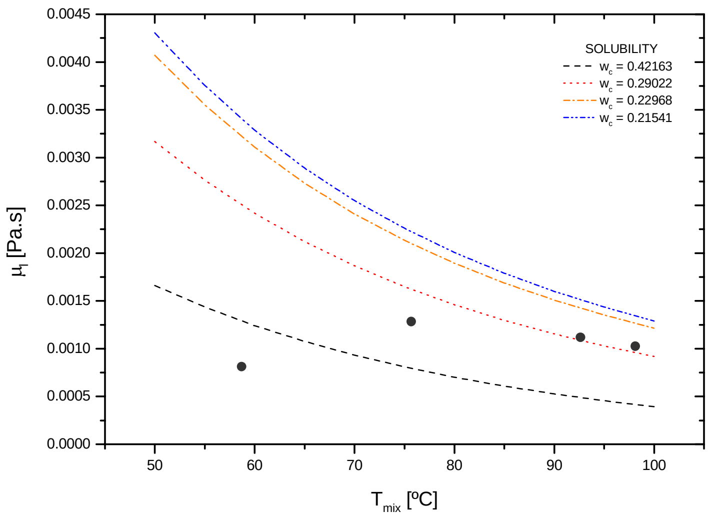


Figure 6. Liquid viscosity for the data of Table 2

5. Conclusion

Steady two-phase flow of oil-refrigerant mixture through the radial clearance in rolling piston compressors has been modeled to estimate leakage of refrigerant gas. An homogeneous model has been used to represent refrigerant outgassing, and the influence of the gaseous phase is included in both density and absolute viscosity, through apparent physical properties. The results obtained with the two-phase flow model show that the bubbles formation changes significantly the flow characteristics, reducing the mass flow rate of both mixture and refrigerant gas, mainly at low temperatures. It has been found that the mass flow rate of the refrigerant gas calculated using two-phase flow model can be up to about 37% lower than the mass flow rate of the refrigerant gas estimated using single-phase flow model.

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7. Responsibility notice

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