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Optimization of Compression Chamber to Reduce the Viscous Friction in the Piston-Cylinder Clearance

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ABSTRACT

The compression chamber of a hermetic reciprocating compressor is, normally, formed by a cylinder and a piston which present a constant clearance during the piston displacement inside the compression chamber. This configuration has the advantage of a good sealing, however, during the most part of compression, the pressure difference between compression chamber and internal compressor ambient is small enough to not provide any leakage from chamber to internal ambient. In this situation, during the most part of compression process, the sealing efficiency is not necessary, but the high viscous friction, due the small clearance between piston and cylinder, generates a considerable waste of energy by viscous friction.

This article presents an analysis about a geometric proposal for the compression chamber that guarantees small clearance, necessary to the correct sealing of compressed gas, in a small region near the upper piston position. The second part of compression chamber is characterized by an increasing in the piston-cylinder clearance along the chamber length. The motivation for this configuration is to provide the adequate sealing only in the region where the leakage process is crucial to compressor performance, furthermore, the clearance growth provides a reduction on viscous friction between piston and cylinder, and both effects generate an expressive increase in the coefficient of performance.

1. INTRODUCTION

The new energy efficiency standards of hermetic reciprocating compressors to domestic refrigeration are demanding the development of new solutions for all areas related to compressors. Some areas, as thermodynamics, electrical and also mechanical, are the main focus of compressors designers. The typical mechanism of a reciprocating compressor is described in the Figure 1. In this figure is possible to verify the presence of an eccentric shaft, a piston and a connecting rod linked the piston and the shaft. The presence of lubricant oil to guarantees the reliability of this system for decades, without any maintenance, also provides viscous friction losses in the shaft's bearings and in the clearance between piston and cylinder.

The friction viscous losses on piston-cylinder clearance is one of the most important mechanical losses of the compressor, and this region has a special characteristic, because very small clearance are necessary not only to provide lift force to the piston, but also to avoid the leakage of gas during the compression process. The Figure 2 presents the mass flow through the piston-cylinder clearance, m_f, for R134a and R600a compressors, as a function of the piston distance from valve plate, y_p, and a function of the crankshaft angle, θ.

Analyzing the Figure 2 is possible to verify that leakage occurs expressively only in a short period of time during the all process, characterized from the gas aspiration in the suction until the discharge process, been the main time during the final steps of compression process. Another point of view of the phenomenon is to analyze the leakage as a function of piston distance from valve plate. From this perspective, is possible to check that when the piston is far from valve plate more than 5mm, both R134a and R600a, the mass flow through the piston-cylinder clearance is almost zero.

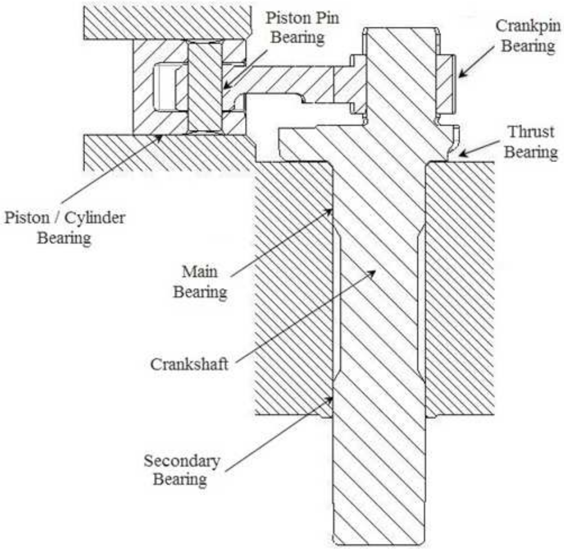


Figure 1: Schematic drawing of reciprocating compressor´s mechanism.

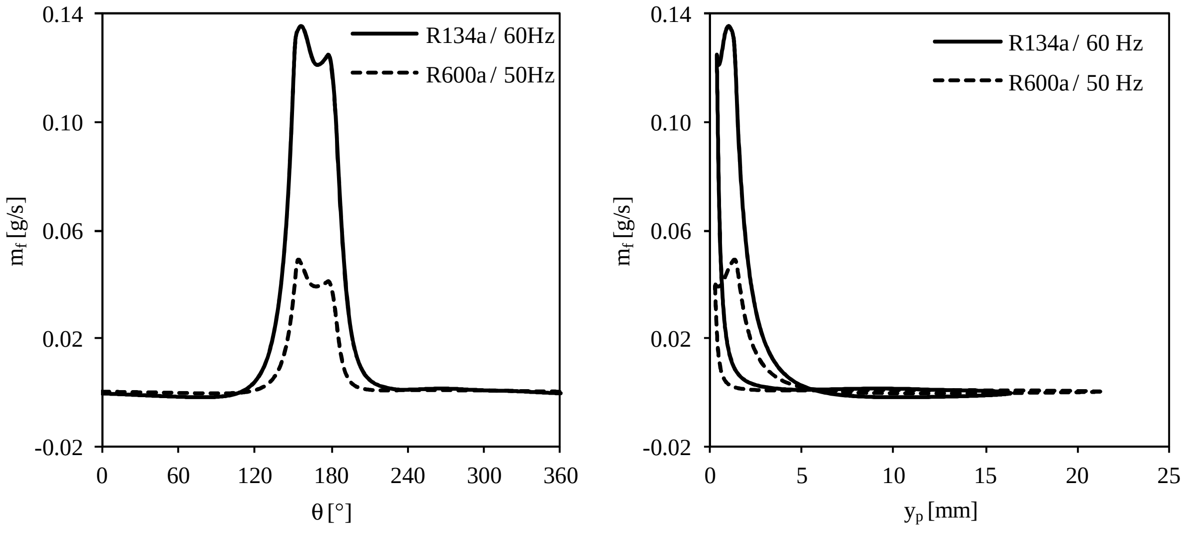
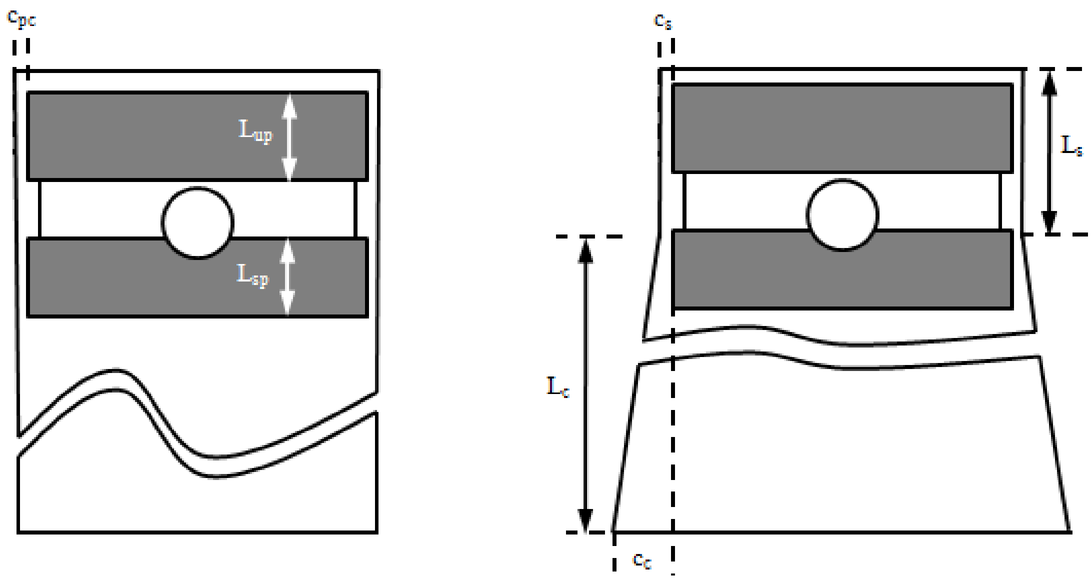


Figure 2: Mass flow through piston-cylinder clearance as a function of: a) crankshaft angle and b) piston distance from valve plate.

The traditional piston and cylinder configuration, which uses constant clearance for all cylinder length, has an intrinsic inefficiency from the mechanical losses perspective. Even in the situation far enough from the leakage critical area, the small clearance keeps the sealing and in consequence, the elevated viscous friction losses. This paper presents an optimized configuration for the piston-cylinder construction, which has a straight cylinder region near from valve plate and a conic shape for the rest of compressor cylinder. This new configuration allows the presence of very small clearance only in the region where the sealing is crucial to compressor performance, this situation occurs in the straight part of the cylinder. Finally, when the sealing process is not more necessary, in the conic region of the cylinder, the clearance is continually increased to reduce the viscous friction losses. The Figure 3 shows a schematic illustration for the current and the optimized configuration.



(a) (b)

Figure 3: Schematic configuration of piston-cylinder: a) current geometry and b) optimized situation.

It is possible to verify in the Figure 3 a reasonable number of parameters which must be evaluated to find the best configuration that allows the correct sealing during the compression process and same time, reduces the mechanical losses when the piston is far from the valve plate. Even from the current piston-cylinder configuration, there are three parameters that must be analyzed: i) upper piston length, L_{up}; ii) skirt piston length, L_{sp} and iii) piston-cylinder clearance, c_{pc} . These three parameters are important also to the optimized configuration, but not sufficient. The new parameters are the clearance at straight cylinder region, c_s, the clearance at end of conic cylinder region, c_c, the straight cylinder region length, L_s, and finally, the conic cylinder region length, L_c.

The parameters analyzed in this paper are the straight cylinder region length, the conic cylinder region length and the clearance at straight cylinder region. The last parameter related to optimized geometry, the clearance at end of conic cylinder region, is not evaluated at this work and the value chosen to guarantee the restriction of piston lift during the movement inside the conic region. The others parameters, common to both configuration, were kept as the same values of current solution.

2. MATHEMATICAL MODEL

The measurement of flow through the piston-cylinder is complicated to do using a functional compressor, due that, the most common approach to predict this phenomenon is using numerical simulation. The present paper uses an integral model to solve the complete reciprocating compressor, considering the suction and discharge mufflers and valves, the compressor chamber and the mechanical losses into the mechanism. The results achieved are the cooling capacity, the power consumption, the coefficient of performance, COP, and specific information about the piston-cylinder region, as mechanical losses and cooling capacity loss by mass flow leaked by the clearance. The next sub-sections will be described in more details the models used in this work.

2.1 Mass flow through the piston-cylinder clearance

The gas could flow from inside the cylinder to the internal compressor ambient by two processes. The first one is the effect of the increasing pressure inside the cylinder that eliminates the oil film and establishing a two phase flow. The second is the leakage that occurs due the refrigerant dissolved in the lubricating oil when the piston descends, the model used in this paper focus only in the first process described.

The flow through the piston-cylinder is governed by two different mechanisms: i) the difference between the pressure inside the cylinder and the shell and ii) the rate at which momentum is transmitted from the piston to the fluid inside the clearance between piston and cylinder during the ascending movement of the piston.

The Figure 4 presents a schematic illustration of the velocity profile in the piston-cylinder clearance. In this picture is possible to verify the effect of piston movement over the fluid velocity behavior, due the piston velocity, even with a pressure potential driven the flow from the cylinder to the cavity, the regions near to piston wall present a flow in the opposite direction. The velocity profile present in the Figure 4 is different for each piston position and pressure gradient and must be evaluated along all the compression process.

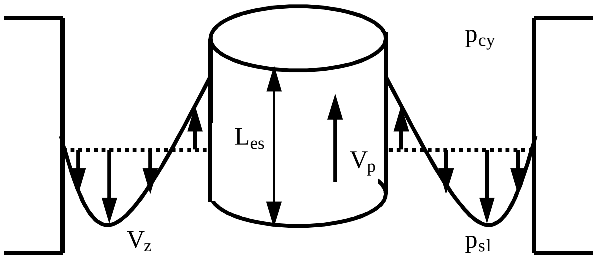


Figure 4: Velocity profile inside the piston-cylinder clearance.

Some hypotheses are necessary to become possible find an analytical solution to mass conservation equation: i) the phenomena is a quasi-steady flow; ii) the boundary conditions variations are instantaneously transmitted throughout all fluids; iii) the oil film has constant thickness; iv) the flow of both Newtonian fluids is incompressible; v) the end effects on the flow are disregarded; vi) inertia forces are negligible compared to viscous and pressure forces; vii) one-dimensional flow and viii) laminar flow. The analytical solution to mass conservation, considering the hypotheses described is presented in Equation (1). For more details see Ussyk (1984) and Ferreira and Lilie (1984).

$$m_f = \pi \rho \overline{V}_z R_c^2 (1\text{-}k) \tag{1}$$

where m_f is the mass flow through the piston-cylinder clearance, $\overline{V}_z$ is the mean gas velocity inside the clearance, ρ is the fluid density, R_c is the cylinder radius and k is the ration between the piston and cylinder diameters.

The mean gas velocity inside the clearance is defined according Equation (2).

$$\overline{V}_z = V_p \frac{1\text{-}k^2 + 2k^2 \ln k}{2 \ln k \left(1\text{-}k^2\right)} + \frac{\left(p_{cy}\text{-}p_{sl}\right) R_c^2}{8 \mu L_{es}} \left(\frac{1\text{-}k^4}{1\text{-}k^2} + \frac{1\text{-}k^2}{\ln k}\right) \tag{2}$$

where V_p is the piston velocity, μ is the fluid viscosity and L_{es} is the effective sealing length. Finally, p_{cy} e p_{sl} are the pressures inside the cylinder and in suction line, respectively.

The optimized piston-cylinder geometry has different values of clearance for each piston position along the compression process and due that, it is necessary to calculate the equivalent mass flow for the upper and the skirt piston length. To obtain this equivalent mass flow through the clearance is considered that the leakage is limited by the lower value calculated for upper and skirt region of the piston, so, a weighted average, presented in Equation (3), is calculated to predict the mass flow for the optimized geometry.

$$m_{fe} = \frac{m_{fu} + m_{fs}}{m_{fu} . m_{fs}} \tag{3}$$

where m_{fe}, m_{fu} and m_{fs} are the equivalent, the upper piston and the skirt piston mass flow through the piston-cylinder clearance, respectively. The mass flows through the clearance for upper and skirt of the piston are calculated using the Equation (1).

2.2 Viscous friction loss in the piston-cylinder clearance

The model used to predict mechanical loss inside the clearance between piston and cylinder considers the same hypotheses described in the sub-section 2.1. The momentum equation is analytic solved considering that only oil is filling the gap between piston and cylinder. The Equation (4) is the final format of the equation to predict the mechanical loss due viscous friction in the piston-cylinder clerance.

$$P_c = \frac{2\pi\mu_o L_{es} V_p^2}{ln\, k} + \pi R_c^2 V_p \left(\frac{p_{cy}\text{-}p_{sl}}{2}\right)\left(\frac{2k^2\, ln\, k + 1\text{-}k^2}{ln\, k}\right) \tag{4}$$

where P_c is the mechanical loss due viscous friction and μ_o is the oil viscosity.

The mechanical loss presented in the Equation (4) is the simplest method to predict this inefficiency in the piston-cylinder clearance, but is sufficient for the present analysis, for more complete piston dynamic the author recommend the work of Prata *et al.* (2001).

Is important to verify that ratio between the piston and the cylinder diameter is a direct function of the piston-cylinder clearance and for the optimized case, differently for the current solution, is not constant all over the cylinder length. So, for each piston position along the compression process is necessary to evaluate the ratio between the piston and cylinder diameter, *k*. The approach used in this work is take the clerance in the upper and lower part of each piston region and uses the average value obtained.

2.3 The compressor model

The model presented to predict the leakage and the mechanical loss in the piston-cylinder clearance was implemented in a global compressor model already validated experimentally. This global compressor model solves the shaft and eccentric bearings considering the short bearing model approach and determines the piston position using appropriate equation for this particular crankshaft mechanism. For more details see Ussyk (1984).

The mass, momentum and energy equation are numerically solved for compressor chamber, suction and discharge mufflers, using the finite volume method. The suction and discharge valves are modeled as one-degree-of-freedom model and the effective force and flow area are used to predict the mass flow through the valves. For more details about this topics check Todescat *et al.* (1992) and Deschamps *et al.* (2002).

3. RESULTS

The optimized piston-cylinder geometry is evaluated for two different situations according the information presented in Table 1.

Table 1: Detailed information of evaluated cases.

Case	Refrigerant	Frequency	T_{evap}	T_{cond}	Displacement
1	R134a	60 Hz	-23.3 °C	+54.4 °C	5.65 cm³
2	R600a	50 Hz	-25.0 °C	+55.0 °C	8.57 cm³

where T_{evap} and T_{cond} are evaporating and condensing temperatures, respectively.

For the two cases described in the Table 1, the parameters analyzed are the straight cylinder region length, L_s, the conic cylinder region length, L_c, and the clearance at straight cylinder region, c_s. The three parameters are presented in dimensionless format to make easier the comparison between the two different kinds of compressor described on Table 1.

The dimensionless variables are L^* and c^* as input parameters and COP^*, P_c^* and ΔCap^* as output data. The L^* is the ration between the straight cylinder region length and the sum of conic and straight cylinder lengths and the c^* is the ratio between the piston-cylinder clearance and the smallest value that is possible to manufacture the compressor. The dimensionless output data are defined as the ration among optimized information for each parameter and the respective reference value for the current geometry.
The Figure 5 illustrates the compressor's efficiency variation according the dimensionless length and clearance are modified. Analyzing the situation which the clearance at the straight region of cylinder is 2.0, indicated by the square symbol in the graphic, is possible to verify that the efficiency is always worst (R600a) and quite similar (R134a) than current piston-cylinder geometry, furthermore, when the length of straight region is reduced, the

efficiency decreases more. For the clearances above 1.6, indicated by the white and black circles in the graphic, the efficiency is always better than the current situation, for both fluids. It is possible to check, for these two clearances, that the smaller values of straight cylinder region increase the compressor's efficiency. The maximum efficiency gain reached with this optimization is 1.4%, keeping the cooling capacity at same level of reference compressor.

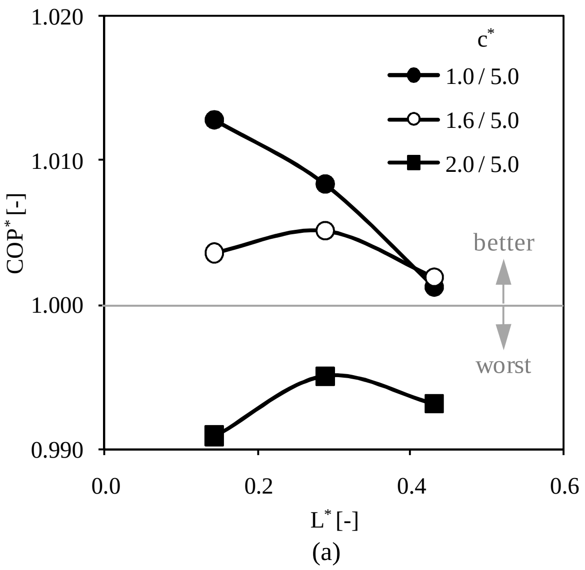


Figure 5: Dimensionless efficiency as a function of dimensionless length: a) R134a and b) R600a.

The explanation for the change of behavior in the efficiency curve when the clearance is modified could be observed in the mechanical loss and leakage behavior, presented respectively in the Figures 6 and 7. The mechanical loss in piston-cylinder clearance has a linear variation with the length and a quadratic one with the clerance, this behavior also could be verified in the Equation (4). For the leakage, the behavior is quite similar to the clearance, but changed for the length, in this situation, the cooling capacity loss variation is proportional to inverse of length.

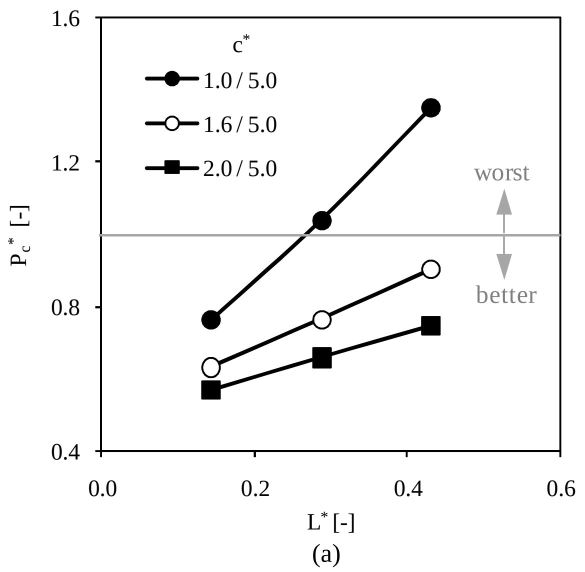


Figure 6: Dimensionless mechanical loss as a function of dimensionless length: a) R134a and b) R600a.

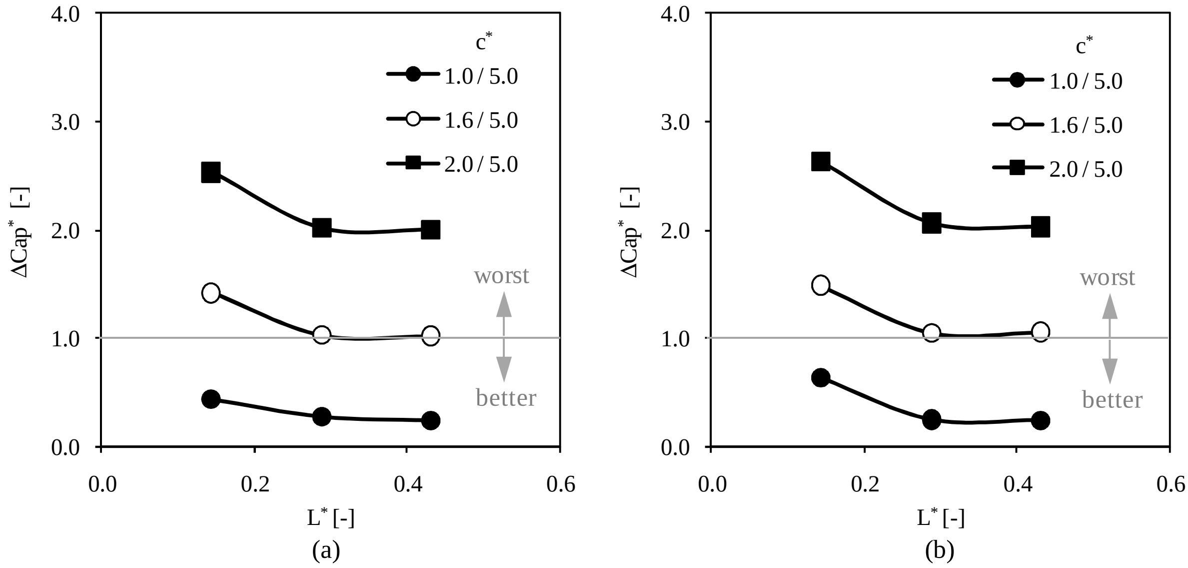


Figure 7: Dimensionless capacity loss as a function of dimensionless length: a) R134a and b) R600a.

The Figure 8 shows the relationship between piston-cylinder mechanical loss and cooling capacity loss. In this figure is indicated a gray area, where the mechanical and cooling capacity losses are lower than the current geometry, in other words, the gray area guarantees that efficiency is always better then straight cylinder configuration. It is important to remember that even out of this gray area is possible to achieve efficiency higher than the reference compressor, the dimensionless clearance with value 1.6 has efficiency better for all lengths evaluated in this work, but in the Figure 8, is out of gray area. This situation could be explained due the relationship between mechanical loss and leakage and its impact on compressor efficiency, for example, when the leakage increases just a few over the reference, but the mechanical loss decrease much more than reference, the compressor efficiency will achieve better value than reference.

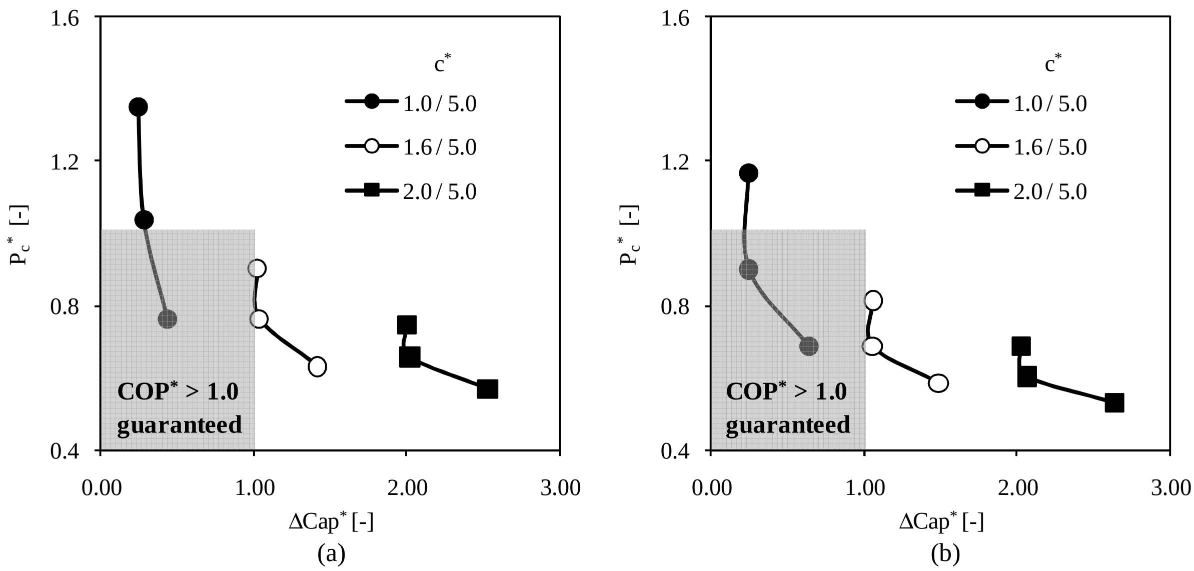


Figure 8: Dimensionless mechanical loss as a function of dimensionless capacity loss: a) R134a and b) R600a.

4. CONCLUSIONS

The paper presented a theoretical analysis of the optimized geometry to piston-cylinder configuration of reciprocating compressors. Numerical models for the mechanical loss and the leakage inside the piston-cylinder clearance were implemented in a compressor global model. Finally, the efficiency of the proposal geometry was evaluated for different levels of clearance and cylinder length, showing that optimized geometry could increase the compressor efficiency more than 1% for specific geometry configuration.

NOMENCLATURE

m_f	mass flow through piston-cylinder clearance	(g/s)
m_{fe}	equivalent mass flow through piston-cylinder clearance	(g/s)
m_{fu}	mass flow through the upper piston clearance	(g/s)
m_{fs}	mass flow through the skirt piston clearance	(g/s)
L_{up}	upper piston length	(mm)
L_{sp}	skirt piston length	(mm)
L_c	conic cylinder region length	(mm)
L_s	straight cylinder region length	(mm)
L_{es}	effective sealing length	(mm)
L^*	ration between the straight cylinder region length and total length	(-)
R_c	cylinder radius	(mm)
k	ratio between the piston and cylinder diameter	(-)
c_{pc}	piston-cylinder clearance	(μm)
c_s	clearance at straight cylinder region	(μm)
c_c	clearance at conic cylinder region	(μm)
c^*	ratio between the clearance and the smallest value possible	(-)
COP^*	ratio between optimized and current coefficient of performance	(-)
ΔCap^*	ratio between optimized and current capacity leakage loss	(-)
y_p	piston position	(mm)
V_p	piston velocity	(m/s)
$\overline{V_z}$	mean gas velocity inside clearance	(m/s)
p_{cy}	pressure inside the cylinder	(Pa)
p_{sl}	pressure in suction line	(Pa)
P_c	mechanical loss due viscous friction	(W)
P_c^*	ratio between optimized and current mechanical loss	(W)
T_{evap}	evaporating temperature	(°C)
T_{cond}	condensing temperature	(°C)
μ	fluid viscosity	(Pa.s)
ρ	fluid density	(kg/m³)
μ_o	oil viscosity	(Pa.s)
θ	crankshaft angle	(°)

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