

DEVELOPMENT AND EXPERIMENTAL TEST OF NEW GAS SEALS FOR A ROTARY-PISTON ENGINE

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Abstract. *Rotary engines have been proposed as an alternative to reciprocating internal combustion engines. Among many rotary geometries, rotary-piston engines present sealing advantages. However, some points of their sealing grid are simply neglected even in the absence of experimental data validating their efficiency. Hence, it's important to better understand the leakage mechanisms in order to avoid sealing problems in rotary-piston projects. This paper presents an experimental methodology to test new rotor sealings configuration for a rotary-piston engine. Two different leakage paths were assessed: radial and circumferential leakages. Non-contact sealing and a mechanical sealing configuration were designed and tested. The results have shown better sealing efficiency using mechanical configuration and higher leakage on the circumferential direction.*

Keywords: rotary engines, gas leakage, sealing, experimental test

1. INTRODUCTION

The automotive industry is always looking for more efficient engines. Nevertheless, the improvements in reciprocating piston engines efficiency has had a slow progress over the last decades. Most of recent improvements are due to secondary systems such as turbochargers and micro hybridization. In other hand, rotary engines stand out by their high-power density, multiple strokes per revolution and simple exhaust / intake ports systems. The most known and the only commercial rotary engine manufactured until today was the Wankel engine (Fig. 1a). However, the Wankel engine has had several sealing problems and high oil consumption, which is associated to the rotor geometry and the complex trochoid movement (Yamamoto, 1981; Picard, 2014). In the face of this disadvantages, other rotary engine architecture was widely proposed and they are currently being revisited as the twin-rotor piston engines (TRPE) (Guarato et al., 2015; Guarato et al., 2016; Silva, 2018) (Fig. 1b).

Twin-rotor piston engine (TRPE) geometry has two main advantages over the Wankel engine: the working chambers are separated by pistons, which have more transversal area for installing the gas seals and, the circular displacement which allows a constantly perpendicular contact between piston seals and their running surfaces. In addition, the oil scrapping movement become much simpler. In relation to the inter-rotor seals, there is no consensus of which is the best configuration. Some patents do not present any sealing system, while others have tried to propose mechanical contact seals (Free, 2001; Sakita, 2009). In any case, few prototypes were built and there is no experimental data of their performance.

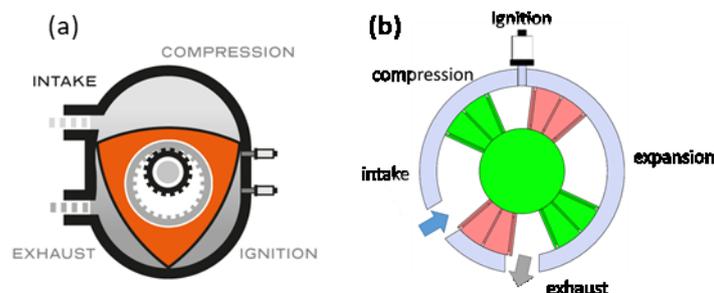


Figure 1. Wankel (a) and Twin-rotor piston (b) engines.

Guarato et al. (2015) presented the working principle of a TRPE and proposed the implementation of Miller cycle on this engine configuration. A 20% increase on break thermal efficiency was obtained while using Miller cycle in relation to Otto cycle for the same engine. Guarato et al. (2016) presented a mechanism which allows an effortless

variation of the compression ratio (CR) in a rotary-piston engine. CR can be changed 8:1 to 68:1, which makes the engine suitable for different fuels at optimum conditions of combustion.

Silva (2018) built a prototype of a TRPE and performed pressure and flow measurement tests. It was observed sealing problems supposed in the rotor-rotor interface and clearances between the combustion chamber components. Since then, new sealing configurations have been developed by the authors of this work.

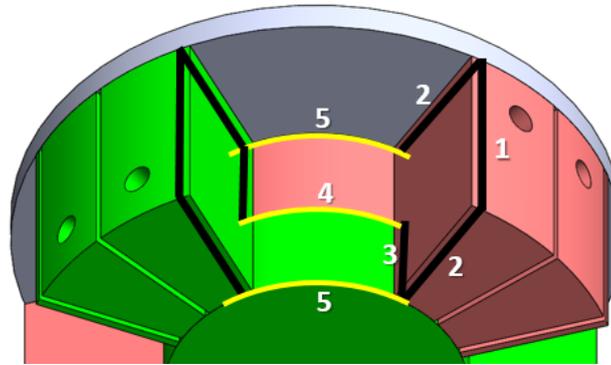


Figure 2. TRPE sealing grid.

Figure 2 presents the TRPE sealing lines representing the clearances which the combustion gas can leak through. New seals geometries were designed in order to construct a continuous sealing grid, covering all lines and their junctions. Special attention has been given to lines 4 and 5, the clearances between rotors flat surfaces. These lines seem to be simple to seal using a ring or a bushing but their contact with pistons seals (lines 2 and 3) cannot be easily assured. Two different leakages occur in the sealing lines 4 and 5: a circumferential and a radial leakage (Fig. 3).

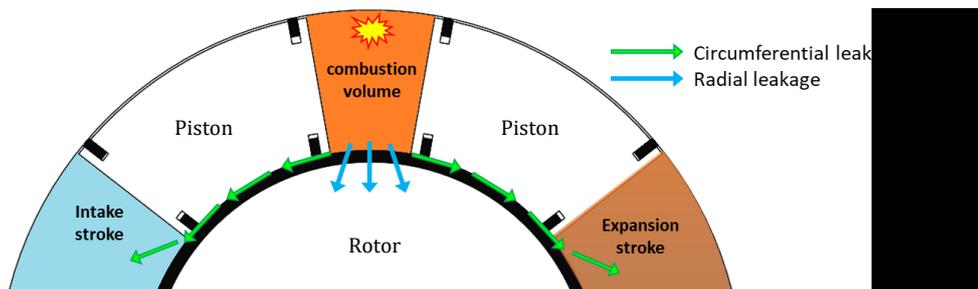


Figure 3. Leakage paths through rotors seals.

The radial leakage crosses the primary sealing surface toward the rotor center. The circumferential leakage contours the rotor seal and flows from the high-pressure volume to the low-pressure volumes, which are equivalent to intake and exhaust strokes in the actual engine. These different leakages must be compared in order to detect the weakest sealing grid point and then improve it. The objective of this paper is to improve sealing design on this TRPE and to validate it through an experimental measurement of radial and circumferential leakages.

2. PROJECT AND VALIDATION OF ROTOR SEALS

2.1 Development of new seal configuration

The development of the new seals started with brainstorming sessions where different geometries were proposed. The ideas were based both on previous sealing systems existing on the literature and on original ideas. Wankel engine's seals was used as the main reference because they were subject of many studies and had many years of development (Jones, 1972; Yamaguchi, 2003; Ohkubo et al., 2004). In addition, the Wankel engine's chamber has similarities to the TRPE's chamber: a displacer (rotor/pistons) with 90° corners; a curved central housing perpendicular to two flat side housings; fixed and adjacent regions for each one of the times of the thermodynamic cycle, delimited only by discontinuous sealing lines; radial pressure and centrifugal force acting on the gas seals. Even neglecting some problems, e.g. discontinuities and clearances, there are some sealing grids proposed for TRPE (Wittry, 1995; Dec, 2010) that were investigated to design the seals presented in this work. Theoretic analyses of seals clearances and displacement were carried out using commercial softwares such as Solidworks®. The solution expected must have the

smaller leakage area as possible. Seals simplicity is another important selection criterion to reduce the manufacturing cost and to facilitate the assemblage.

The solutions for rotor seals converged on to two configurations: a non-contact sealing with bronze bushings and, a mechanical sealing using axial rings. The efficiency of the rotor seals configurations depends on the tolerances achieved in the manufacturing process. An experimental leakage test was carried out for comparing which sealing configuration was more efficient. Aiming reduce the cost and the complexity of the experiment, both configurations were compared in a simplified test chamber by measuring different leakage flows. The best rotor seal configuration had to be adapted to the TRPE prototype. The piston seals will be tested directly on the TRPE prototype and are not detailed in the present work.

2.2 Description of the test chamber

Comparing the rotor seals performance directly on the TRPE prototype implies on manufacturing expensive components and instrumenting difficult access clearances. Hence, a simplified test chamber was designed and manufactured for emulating TRPE seals and to perform experimental tests using compressed air. The simplified test chamber was designed in a way that different seal configurations can be easily interchanged and radial and circumferential leakage can be distinguished.

Figure 4 shows the main correlation between the TRPE prototype components and the simplified test chamber parts. The simplified test chamber is composed of a rotor (a), a central housing (b), two side housings A (d) and B (c). The simplified test rotor is a 1:1 scale model of TRPE rotor. The housing B plays the role of the second coaxial rotor and the housing A acts as the TRPE internal side housing. The central housing simulates pistons bottom surface where the piston seals (f) and springs (g) are mounted. The side housings and rotor geometry depend on the seal configuration which are tested. Figure 4 illustrates only the bronze bushings (e) configuration of the simplified test chamber.

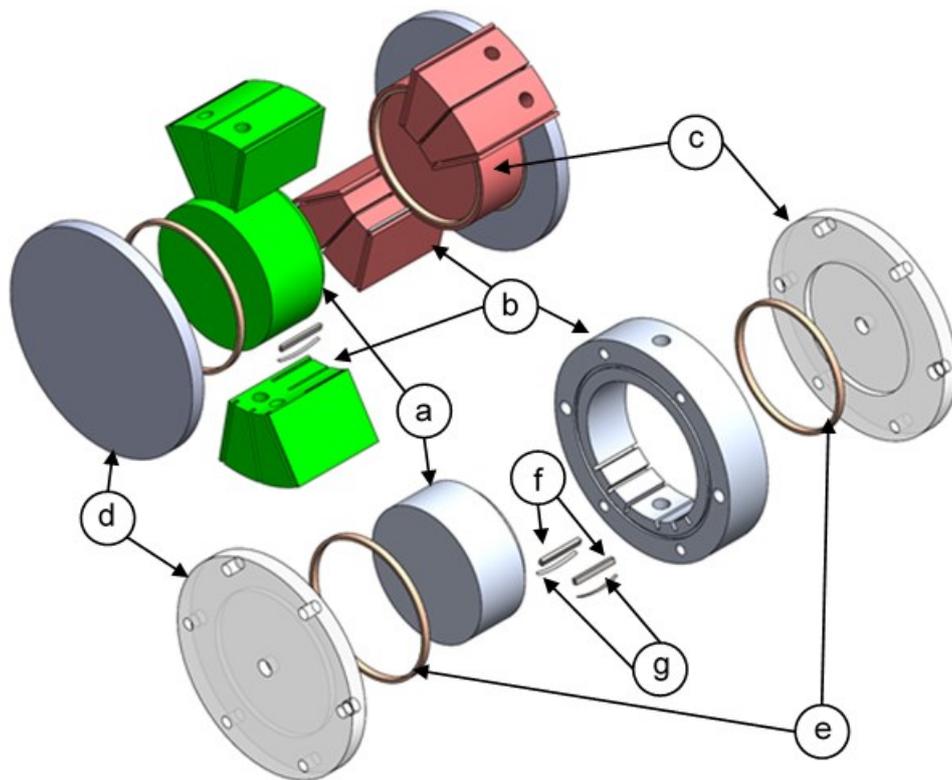


Figure 4. Exploded view of TRPE parts (in the left) and their analogous parts in the simplified test chamber (in the right).

When the simplified test chamber is assembled, the four main TRPE rotor clearances are reproduced, as shown in the Fig. 5(a). “Inlet” is the high-pressure volume where air is injected, simulating the combustion volume between two pistons seals. “R/R” (rotor/rotor) and “R/H” (rotor/ housing) clearances are formed between flat surfaces of rotors and housing, confined inside the rotor seals. “C” (circumferential) clearance is all the crevices outside rotor seals after pistons seals: axial clearance between seals, flat surfaces of rotors and housing; clearance between central housing and rotors cylindrical surface; and piston seals end clearances.

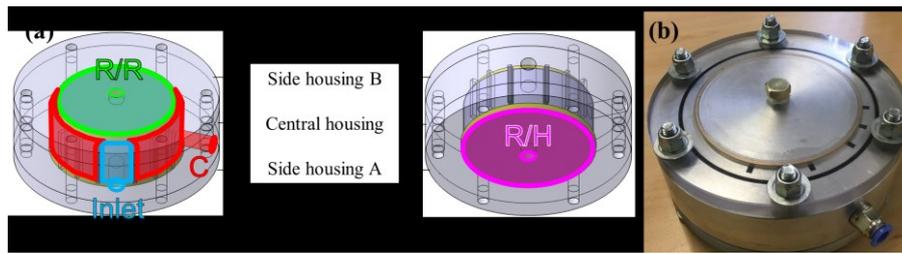


Figure 5. Emulated clearances (a). Manufactured non-contact sealing test chamber (b)

Each test chamber housing has a threaded hole for the air outlet that leads the air leakages to the atmosphere. The central housing has an extra hole for the air inlet. The manufactured non-contact sealing test chamber is presented in Fig. 5b.

2.3 Experimental protocol

The side housings comporting the sealing configuration to be tested (either the non-contact sealing with bronze bushings or, the mechanical sealing using axial rings) are mounted on the central housing. Two central housing grooves are filled with piston seals and their springs. A pressure vessel is filled with compressed air at 4 bar using a compressor with delivery pressure up to 8 bar. The vessel with compressed air is connected to the test chamber inlet. One of the chamber's outlet is open while the others remain closed with blanking plugs. The vessel output valve is open and the compressed air flows through the test chamber seals to the atmosphere. The vessel pressure variation is recorded until it reaches the atmospheric pressure. This procedure is repeated for each outlet (R/H, R/R and C) and the pressure drops are compared. The radial leakage is observed when R/R or H/R outlets are open. The circumferential leakage is observed by opening C outlet. The experiment scheme for measuring the air leakage is presented on Fig. 6.

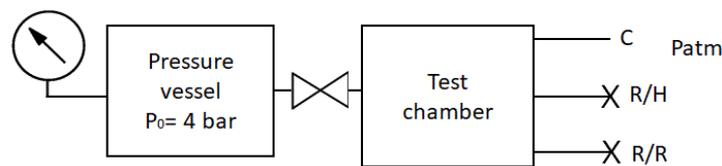


Figure 6. Leakage experiment scheme.

3. RESULTS AND DISCUSSIONS

3.1 Improvement of piston seals design

The previous TRPE prototype (Silva, 2018) had four L-shaped seals which were mounted on the piston corners and overlapped at the central section of each piston sealing line. In this arrangement, half of each seal ran on a different surface. As the seal is a single piece, it could not conform to unequal surfaces due to perpendicularity deviation of the housings, different wear rate and different thermal distortion.

Therefore, the proposed sealing grid for the piston has one rectangular seal for each piston surfaces. Hence, each seal can better conform to its respective running surface. The new junction between the piston seals occurs now at the corner of the pistons. Different joints were investigated such as: butt joints, angled joints and overlapped joints. The butt joint is the simplest to manufacture, but it does not eliminate the butt clearances between the ends of the seals, which are necessary for compensation of manufacturing tolerances and thermal expansion. The angled butt joints may seem adequate to eliminate end clearances as the ones used on the Wankel two-piece apex seal. However, this configuration only works if a side piece is split from the main piece of each piston seal, in a way that both parts slide over each other on an angled contact and close the end clearances. Manufacturing and assembling the split pieces for each piston seal would be unrealistic due to high cost and complexity. In addition, the sharp corners reduce the split pieces' mechanical resistance.

The overlapped joint, as shown on the Fig. 7, was chosen as the best option for this project. With this junction, the butt clearances between the seals remains but now covered by the overlapped extremities of the seals. The clearance between the piston seals ends and the housings are reduced to small orifices at the corners of the assembly. These orifices may appear at the bottom end of the pistons seals running on the side housing as the rotors accelerate. The reason is that these seals are free to move radially and are pushed outward by body force, closing the top end clearances and opening the bottom ones. Even with these remaining clearances, the leakage flow normal to the piston section is

reduced and it is deviated to the grooves. In this case, the gas flows through each seal flank clearance, after that pressurizes the volume under the seal, then flows through the butt clearances in the direction of seals height and finally reaches the low-pressure chambers. All these flow restrictions may prevent pressure loss in the combustion chamber.

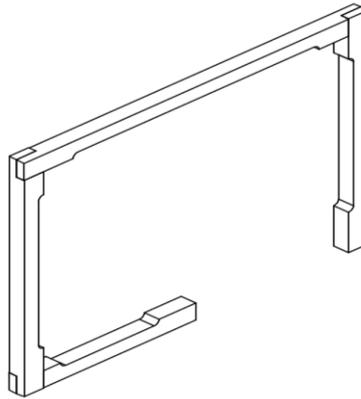


Figure 7. Assembly of the overlapped piston seals.

3.2 New proposed rotor seals

Two configurations were pre-selected to seal the clearances between rotors and side housings, which are equivalent to lines 4 and 5 on Fig. 2. The first sealing configuration is a mechanical sealing composed of rings axially pressed against the rotors by wave springs (Fig. 8(a)). In this case, the radial leakage is obstructed by the contact of the components. Each ring has a gap and a bigger diameter than its groove, allowing an assembling under elastic deformation. Thus, there is an expansive pressure that keeps the ring outer flank in contact with the groove outer flank. This assembly prevents gas entering in the grooves and creating leakage under the rings. The second one is non-contact configuration with tight-fitting bronze bushings around the rotors (Fig. 8(b)). In this configuration the radial leakage is restricted by tight fits and close running surfaces.

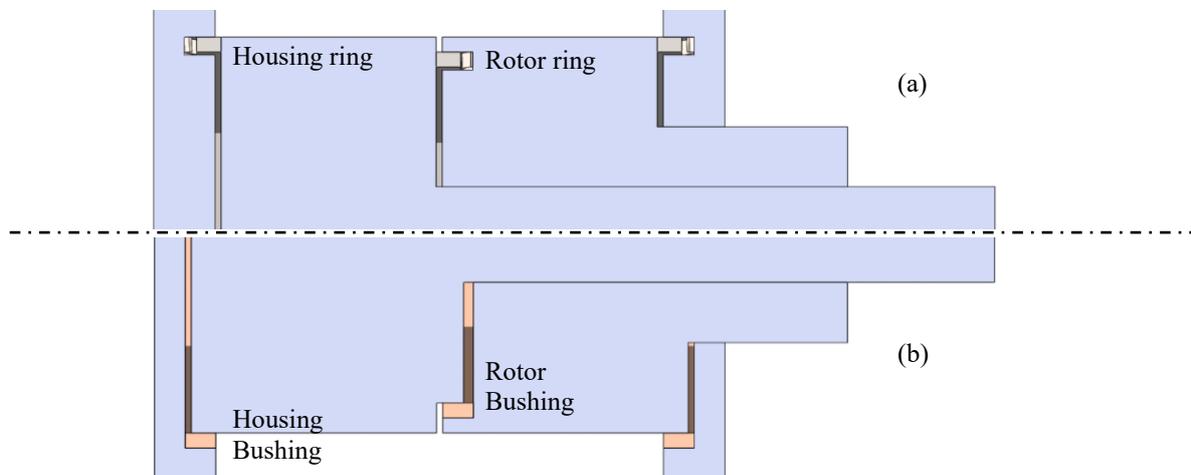


Figure 8. (a) Mechanical sealing configuration (b) Non-contact sealing configuration.

On both configurations, the seals between rotor and housing are mounted on the housings and can be in contact with the piston seals. Thus, the area for circumferential leakage depends on the piston seals end clearances. On the other hand, the inter-rotors seal is mounted in a groove that is machined on one of the rotors. Consequently, the rotor seal is separate from the piston seals by the groove wall. In this region, the circumferential leakage depends on the axial clearances between the rotors.

3.3 Experimental comparison between rotor sealing configurations

The results of the experimental tests are shown on Fig. 9. The circumferential leakage (C) and radials leakages (R/H and R/R) are compared to those of the non-contact sealing configuration (with bushings) and the mechanical sealing configuration (with rings). The mechanical sealing configuration with rings provides a better sealing and substantially

reduces the radial leakages than the non-contact configuration. The “R/H” sealing is the most efficient because it provides the slowest pressure reduction over time.

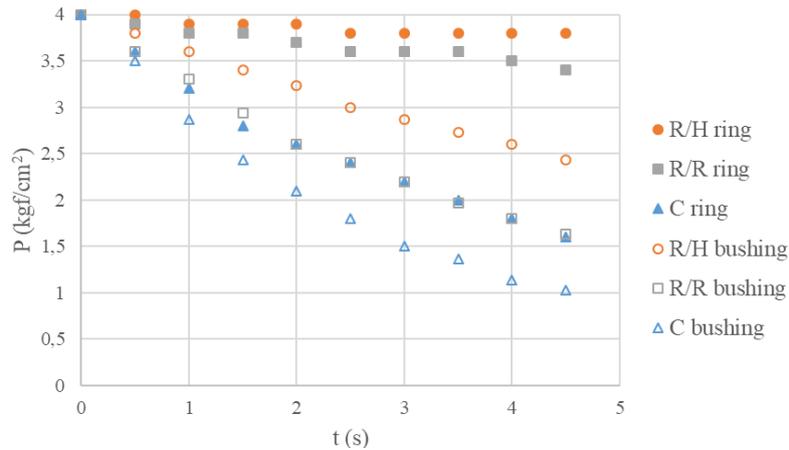


Figure 9. Evolution of vessel pressure due to circumferential (C) and radials (R/H and R/R) leakages.

For both configurations, the circumferential “C” leakage is higher than radial “R/H” and “R/R” leakages. This suggests that gas can reach exhaust and intake chambers before reaching clearances between housings and rotors or inter-rotor clearance. Considering the air pressures of each piston seal against the groove outer flank there is no flow under the piston seals. The “C” leakage depends on piston seals end clearances and rotor axial clearances. The piston seals used on the non-contact configuration were the same used on the mechanical configuration, thus the end clearances did not change. Then, the higher “C” leakage in the non-contact sealing can be attributed to a larger rotor’s axial clearance. This clearance on the non-contact sealing configuration is difficult to adjust since there are more surfaces to be machined.

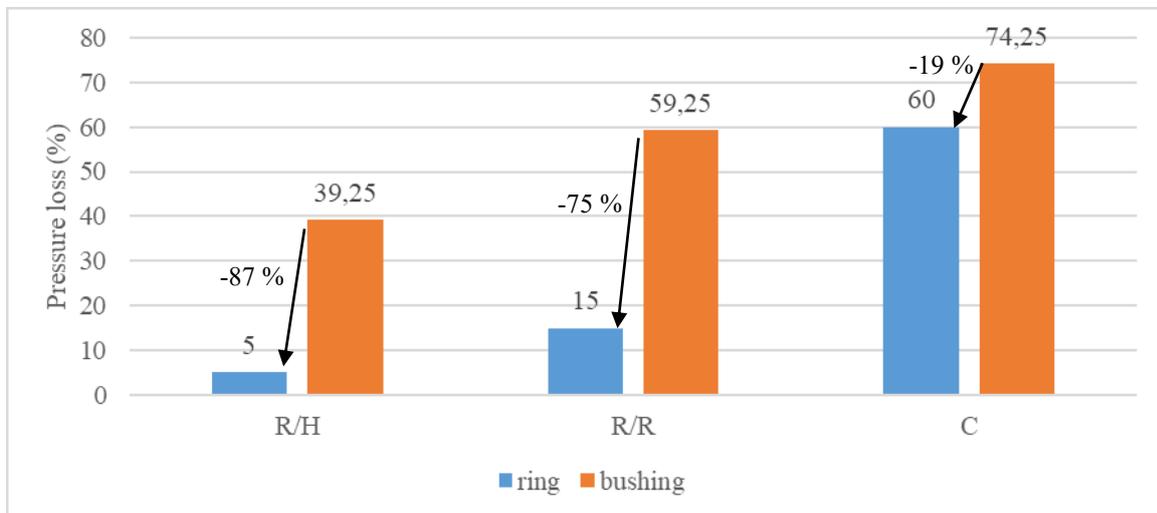


Figure 10. Percentage of pressure loss due to R/H, R/R and C leakages using mechanical (rings) and non-contact (bushings) sealing configurations, after a leakage period of 4,5 s.

Even controlling the manufacturing of test chamber components and assuring tight fits, the non-contact configuration could not seal as well as the mechanical configuration with ring. Figure 10 shows the percentage of pressure loss, in relation to the vessel initial pressure of 4 bar, after a leakage period of 4,5 s. The percentage loss was calculated by the difference between final and initial pressures over the initial pressure. The loss pressure due to R/I leakage is 5 % with rings and 39.25 % with bushings; due to R/R leakage is 15 % with rings and 59.25 % with bushings; due to C leakage the loss is 60 % with rings and 74.25 % with bushings. The mechanical configuration reduces the pressure loss due to R/I, R/R and C leakages of 87, 75 e 19 %, respectively.

In addition, under operating conditions, the housings, rotors and pistons will present dissimilar thermal expansions which requires larger clearances. Therefore, the mechanical sealing configuration with ring was chosen instead of tight-fitting bushings to be implemented on the TRPE prototype.

3.4 Junction seals

The tests presented in the previous section shown that the sealing grid is ineffective in preventing circumferential leakage. It was observed that this leakage is sensitive to the rotor axial clearance. Assuming the engine will work with large axial clearances, the absence of a seal between the inter-rotor seal and the bottom piston seals creates a large area for circumferential leakage. Thus, a junction seal was proposed (Fig. 11).

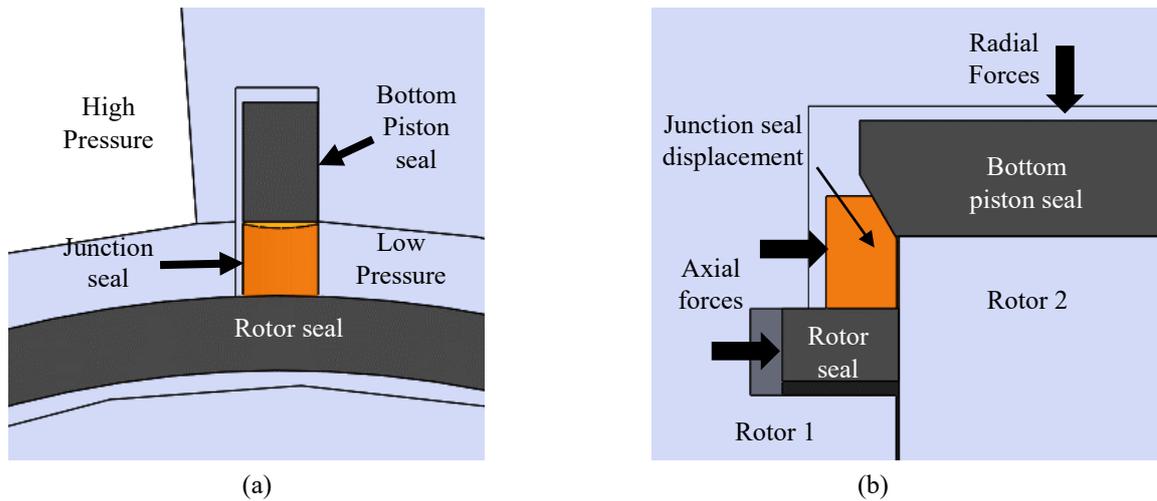


Figure 11. a) Front view and b) lateral view of the junction seal.

Axially pressed by springs and gas pressure, the junction seal slide over an angled cut on the bottom piston seal. As result, the clearance between the rotor seal and the piston seals are closed and a small orifice remains at the corner. For this displacement to occur, the piston seal must be blocked in the other extremity. The seals dimensions and wear rate must be precisely controlled to prevent seals blocking without contacting each other or the rotor. These junction seals are yet to be validated through experimental testes.

4. CONCLUSIONS

This work proposed the development of better seal configuration for a TRPE prototype. Through theoretical analysis and experimental tests, a careful selection of new geometry for piston and rotor seals was made. For the first time, even that qualitatively, the theoretical behaviour of TRPE seals was investigated considering the existence of clearances and displacements. New seals geometries were developed in order to increase the seals conformability and reduce the leakage area. The sealing grid regions with low sealing efficiency were pointed out and improvements were suggested.

Seeking a balance between sealing efficiency and durability, an overlapped piston seal was proposed. This configuration has simple to manufacture and imposes higher leakage restriction compared to a previous grid tested on a TRPE prototype. Two solutions for seal the rotors were proposed and compared: a non-contact and a mechanical sealing configuration.

A simplified test chamber was manufactured to evaluated non-contact and mechanical seals for the rotors. Circumferential leakage, surrounding the rotors' seals, and radial leakage, passing directly through the seals, were compared. The findings on the rotor leakage can be summarized by the following conclusions:

- Circumferential leakage is dominant and its cause is the axial rotors' clearance;
- The gas reaches the exhaust and intake chamber before accumulating in rotors and side housings interfaces;
- Mechanical seals provide better sealing than non-contact tight-fitting seals.

5. ACKNOWLEDGEMENTS

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