Design methodology of an elastomeric reciprocating sealing system: Case of quasi static operating conditions
Méthodologie de conception d'un système d'étanchéité linéaire alternative: Cas d'un fonctionnement quasi-statique

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The reliability of reciprocating sealing systems is pivotal in many applications, such as landing gear braking systems. A reciprocating sealing system is reliable when presenting a good sealing performance during its whole life cycle. Unfortunately, the design of such systems is often based on a trial-and-error approach using the gained experience, making the design cost highly unpredictable.
To overcome this problem, a design methodology of an elastomeric reciprocating system is presented here in the case of quasi static operating conditions. This methodology is based on a contact pressure criterion.
The obtained results validate the methodology and demonstrate that the contact pressure criterion can help for a quick design of reciprocating sealing systems.

La fiabilité des systèmes d'étanchéité alternatifs linéaires est très importante dans plusieurs applications telles que les trains d'atterrissage d'avion. Un système d'étanchéité linéaire est fiable lorsqu'il présente une bonne performance en étanchéité tout au long de sa durée de vie. Malheureusement, la conception de ce type de mécanismes se fait encore sur une démarche empirique fondée sur les acquis d'expérience, ce qui affecte les coûts de conception.
Pour surmonter ce problème, une méthodologie de conception de systèmes d'étanchéité alternatifs linéaires dans le cas d'un fonctionnement quasi statique est présentée ici. Le critère d'étanchéité est basé sur la pression de contact tige/joint.
Les résultats présentés dans cette étude valident la méthodologie utilisée et démontrent qu'un critère d'étanchéité en pression peut aider à la conception rapide de systèmes d'étanchéité linéaires alternatifs.

1 Introduction

Reciprocating sealing systems are devices used as actuators in many industrial applications such as construction machines, production chains and landing gear braking systems. Their reliability is essential for financial, ecological and security reasons. A reciprocating sealing system is reliable when presenting a good sealing performance associated with a sufficient life expectancy.
Tribological study of such systems began around the year 1930 [1]. They can be classified in two groups: those using rod seals and other using piston seals. The difference is in the location of the fluid containment wall. In the first case the containment wall is the rod/seal contact, while in the second case the containment wall is the piston/seal contact.
The main part of a sealing system is a seal, so that most of the studies focus on it. For this reason, the material properties of seals have been improved since 1960, and a great diversity of seal designs have

been proposed. The changes made over the past decades have gradually led to improved performance of existing systems. Despite the many advances in the understanding of sealing systems, their design is often based on a trial-and-error approach using the gained experience, making the design cost highly unpredictable.
In fact, it is difficult to define the sealing concept, provided that it is well known that there is no sealing system without leakage. J Martin [2] defines the sealing as the quality of containment of the fluid inside an equipment. He classifies the sealing types into groups depending on the level of the leakage rate.

However, the mechanism of sealing is already understood [3]. According to Nikas [1], the mechanism of the sealing is related to the contact pressure distribution as illustrated in figure 1.

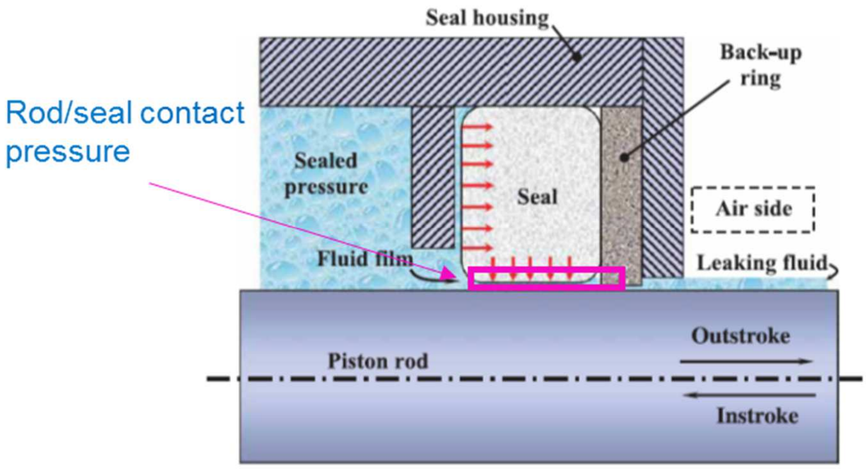

Figure 1 : The seal automatically adjusts to sealed pressure variations by transferring it to the sealing contact via the nearly incompressible material (see arrows) [2]

The main goal of the studies is the quantification of fluid losses. Experimental studies focus on the development of experimental setup for the measurement of friction and the fluid flow quantification while the numerical one attempt to assess the fluid thickness in the contact and quantify the amount of leaking fluid. No existing strategy seems to take into account the tribological triplet developed by M. Godet [4], while this approach has proved useful in a large number of other tribo-systems [8, 9].
The design of a reciprocating sealing system is a complex task, which requires a great deal of expertise. It requires knowledge in material sciences, tribology, manufacturing technology, production engineering, fluid engineering, chemistry and physics, and an overall appreciation of engineering systems [3]. Furthermore, the accumulated studies to date have showed a great number of parameters affecting the performance of a sealing system and the authors are not aware of a simplified and yet comprehensive design methodology. This does not prevent the will to innovate in this field of sealing systems design, since 200 patents are reviewed in Sealing Technology Newsletter each year [1].
The present paper intends to propose a methodology witch can help for the design of an elastomeric sealing system, in the case of quasi static operating conditions. For illustrative purpose, the design of the piston actuator of a landing gear braking system will be used as a case of application.

2 Design strategy of a piston actuator of landing gear braking systems

Parameters affecting the sealing performance are presented in the following table 1. Each of them is linked to the contact pressure in some way.

Sealing Parameters	Link with contact pressure
Seal shape and piston shape	Volume seal deformation
Operating parameters, seal ageing	Seal extrusion + ageing
First body properties (rod, seal), operating temperature,	Seal deformation + rod/seal friction
Boundary conditions of the tribosystem,	Mechanism contribution on the contact pressure
Third body in the contact	Local change of the contact pressure

Table 1 : Main parameters affecting the sealing performance

Hence, the contact pressure can be defined as a signature of the sealing mechanism and considered as the main parameter for the design of the piston.
In quasi static condition, it is known that the strict sealing performance with reciprocating seal, the contact pressure (P_c) should be greater than the fluid pressure (P_f) [5, 6]:

$$P_c > P_f \qquad (1)$$

A dual approach is therefore proposed, based on a numerical analysis and an experimental validation as illustrated in Fig. 2.

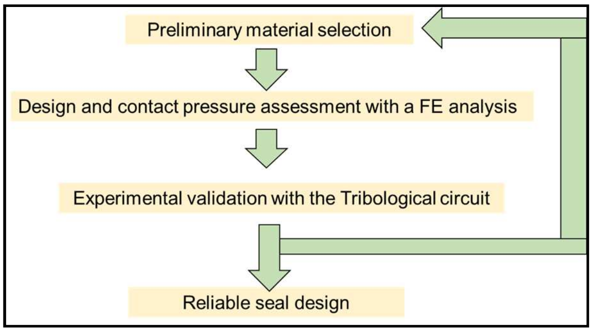

Figure 2: design methodology of a reciprocating sealing system in quasi static operating conditions

The material selection is based on the following criteria: [3]

- Fluid resistance, chemical environment, temperature for the seal
- Fluid resistance, chemical environment, temperature for the anti-extrusion seal
- High hardness for the rod coating. The methodology of the selection of a coating will be developed in another paper.

Details about materials and selection criteria may be found in the Seals and Sealing handbook [3].

The numerical design can be performed with a finite element (FE) model, while the experimental validation is achieved through the tribological circuit [7] identification.

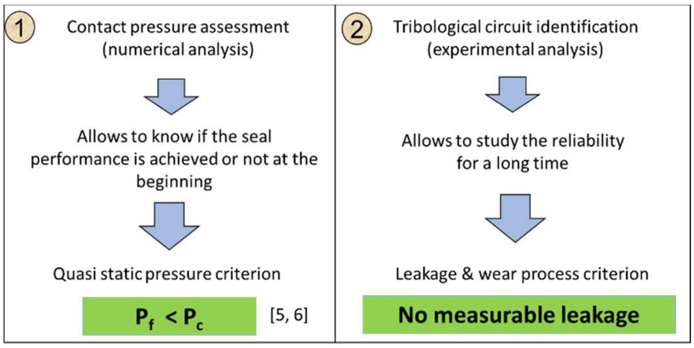

Figure 3 Dual strategy of reciprocating sealing system design

The methodology previously presented is applied on piston actuators of landing gear braking systems. The sealing system is composed of a rod at the center of the piston, a seal, and an anti extrusion seal. The piston operates as a single acting cylinder containing a spring which ensures the instroke of the piston. The fluid used in the system is a phosphoric ester at room temperature. The material of the seal is an EPDM and that of the anti-extrusion seal is a PTFE.

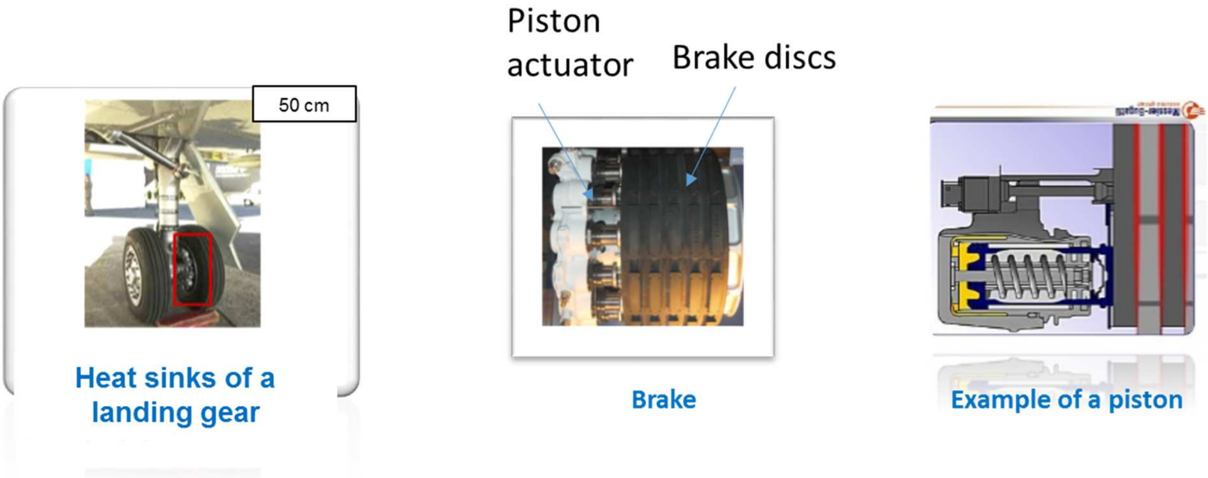

Figure 4 : Piston actuator of landing gear

The system operates in quasi static conditions with an average speed of the seal of 2mm/s during the outstroke. During the braking phase, the hydraulic pressure reaches 12MPa. This phase is preceded by the outstroke and followed by the instroke. The following paragraph will present the numerical design followed by an experimental validation.

3 Finite Element Analysis

The reciprocating system of the hydraulic piston is illustrated in Fig. 5.

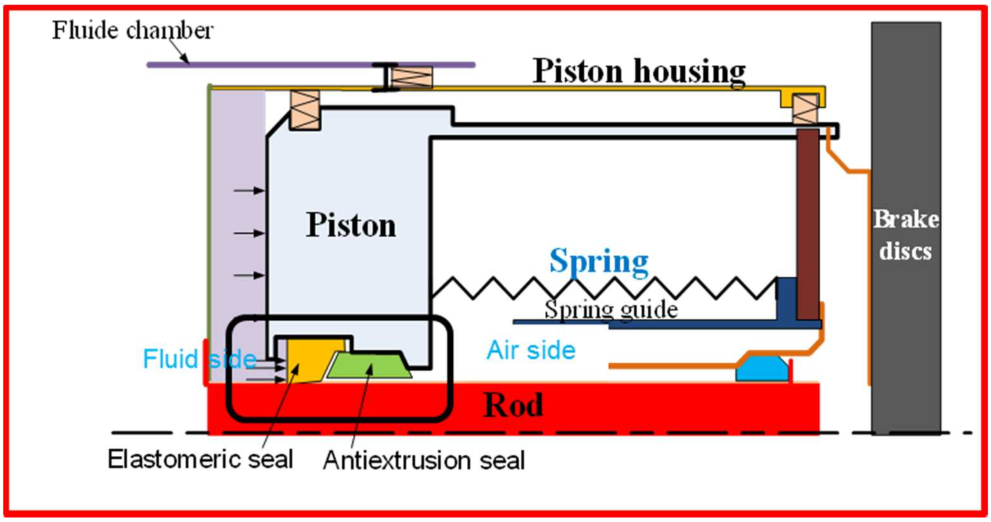

Figure 5 : axisymmetric illustration of the piston actuator

An axisymmetric model of the system has been developed and implemented in the software Abaqus Explicit due to severe non linearity of the problem. It is presented in Fig. 6. The rod diameter is about 6mm. The simulation is performed in 3 steps. At the beginning, the radius of the seal is larger than the allowed space between the rod and the piston (see fig. 6). The first and second steps are intended for the assembly of the seal. The first step is a radial compression of the seal and the second one is a relaxation of the seal, so that we obtain a surface to surface contact between the rod and the seal on one side and the seal and the piston on the other. The seal is tight at this position and its deformation generates a residual contact pressure on the rod. The hydraulic pressure applied on the fluid side of the seal allows for the motion of the piston with the compression of the spring. This motion is stopped by the brake discs resistance illustrated here by the final position of the piston. During the instroke, the hydraulic pressure decreases and the motion of the piston is ensured by the relaxation of the spring.

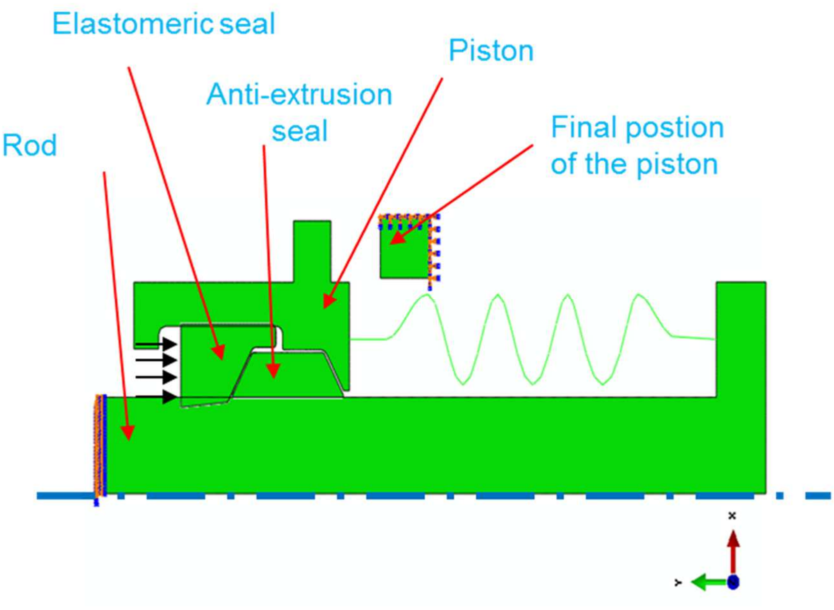

Figure 6: Axisymmetric model of the piston in Abaqus 6.12

Each part of the system has a specific material. The piston and the rod are made of steel modelled as a linear elastic material, the elastomeric seal is in EPDM, modeled as a Mooney-Rivlin hyperelastic material of parameters C10=1, 45; C01=0, 45; D1= 0,011 and a density of 900 Kg/m3. The Rayleigh damping parameters are 100 for alpha and 10E-8 for beta.

The friction coefficients used to perform the simulation are presented in the following figure. A penalty contact method is used to perform the simulation. A parametric study of each friction coefficient has been performed in the range of the values associated to each of them in the figure. This parametric study aimed to see the influence of the friction coefficient at each interface of the system on the contact pressure at the rod/seal interface. The details of the results will be presented elsewhere, but the authors concluded that the friction coefficient at the rod/seal interface are those which influence the most the rod/seal contact pressure.

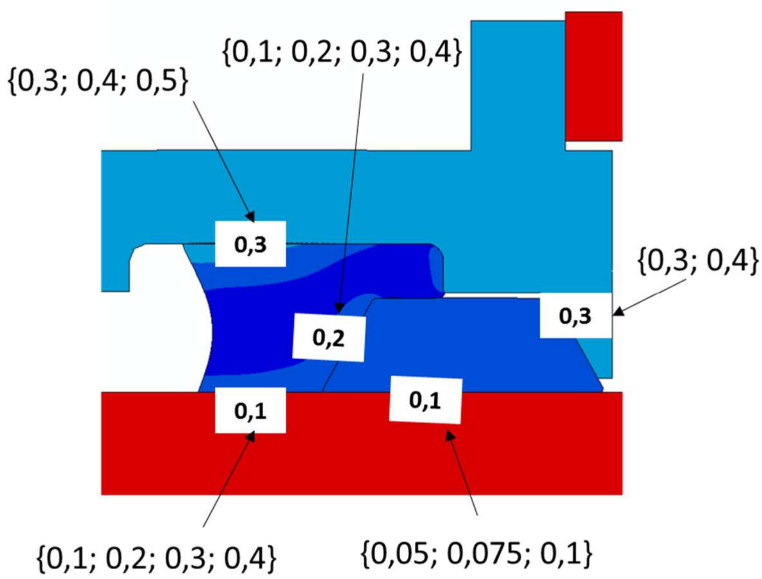

Figure 7 : coefficient of friction at different interfaces

The element types used to mesh the parts are 4-node reduced-integration, axisymmetric, solid element note CAX4R, except for the elastomeric seal composed with CAX4R and CAX3 which is a 3-node linear element, to allow some large deformations in the seal. An analysis of the mesh dependency of the model has been performed in order to ensure its accuracy.

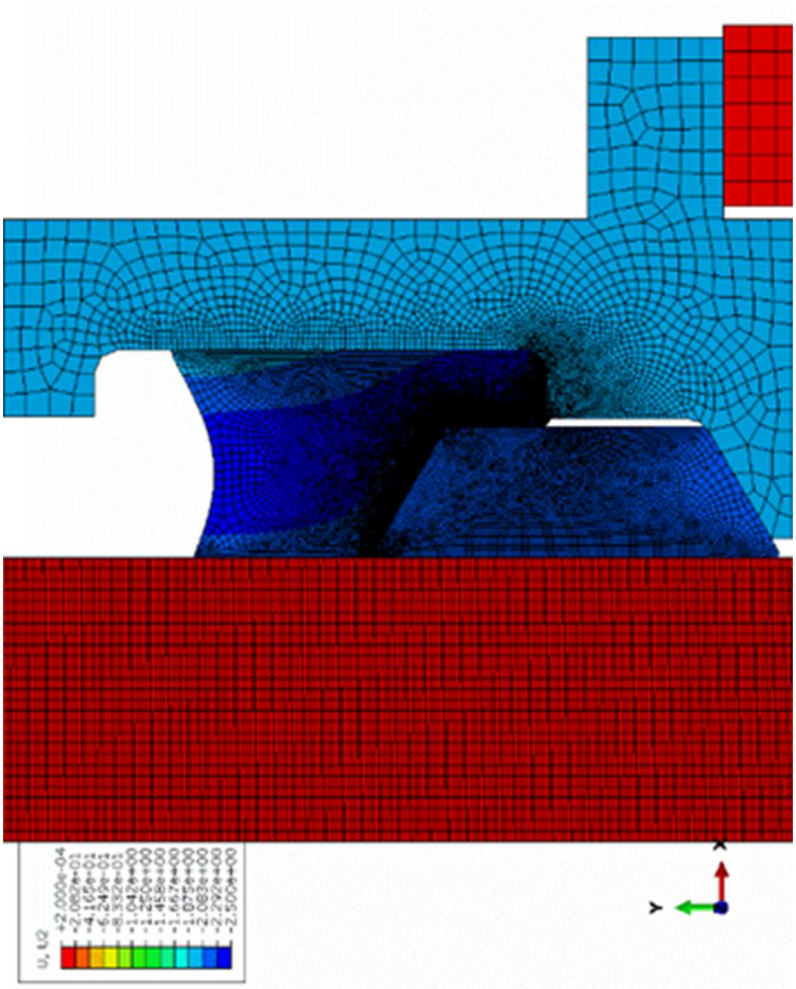

Figure 8 : illustration of meshed model

The developed model presents a stable convergence for hydraulic pressures less than 20 MPa.

The evolution of the hydraulic pressure in time during the operating phases is presented in Fig. 9. This figure shows that the hydraulic pressure reaches 12MPa during the braking phase. On the same figure, the speed variation of a seal point located at the inner zone of the interface gives an idea of the seal sliding on the rod. Further, the deformed shape of the seal and the rod/seal contact pressure are presented in Fig. 10.

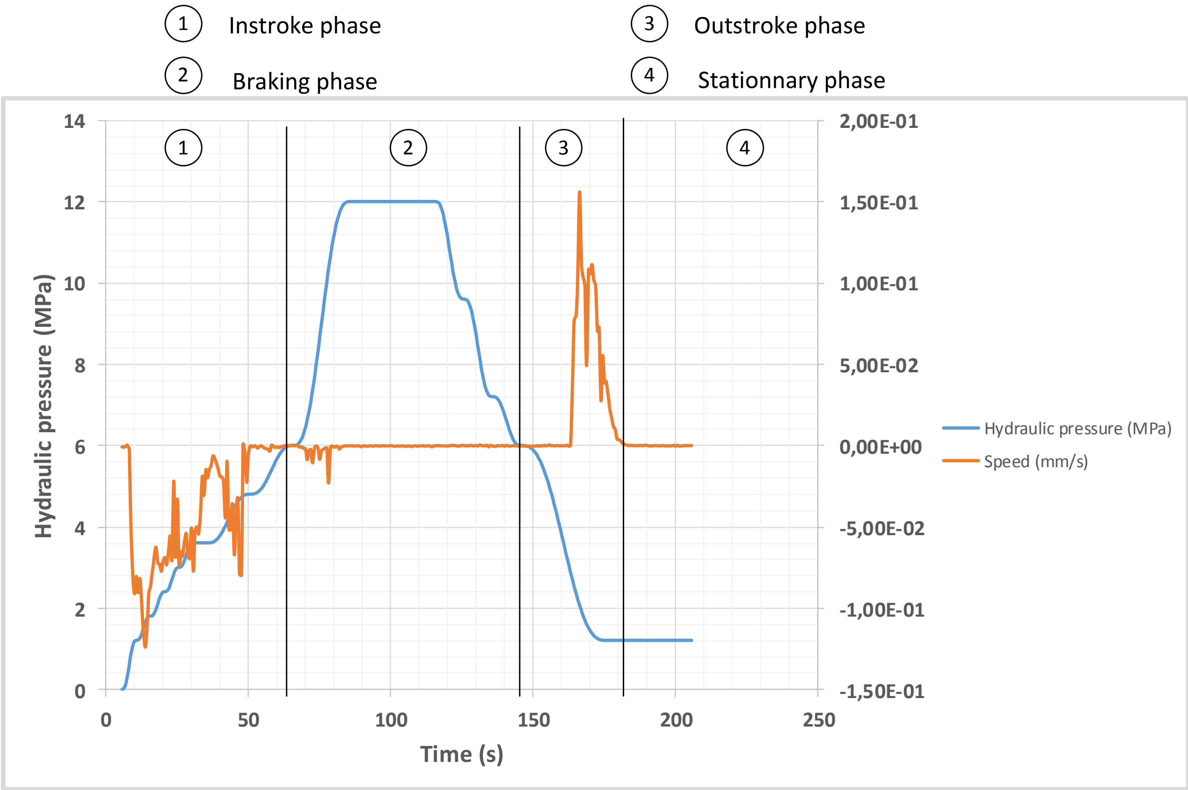

Figure 9: Evolution of the hydraulic pressure and sliding speed of the seal

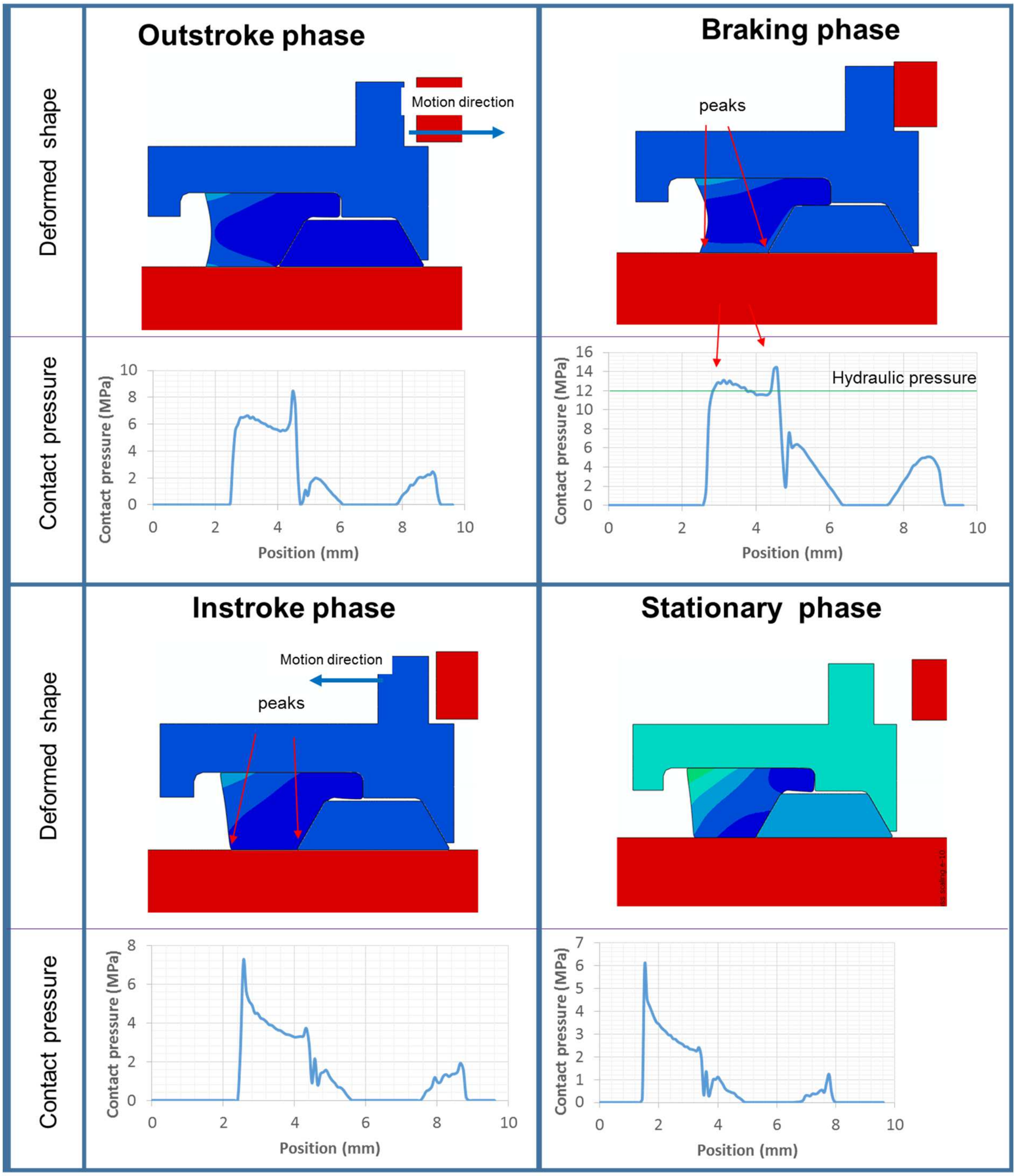

Figure 10 : summary of the simulation results

The pressure applied by the fluid on the left-hand part of the elastomeric seal induces a deformation which generates a contact pressure at the rod/seal interface. This contact pressure depends on the operating phase, as show in Fig. 10. This figure also demonstrates that the rod/seal contact pressure is larger than the hydraulic pressure at two peaks during the braking phase. The first one is on the fluid side and the second one on the anti-extrusion side. This pressure profile shows that the seal design is in agreement with the sealing criterion of Eq. (1), and that both geometry and materials were chosen adequately.

4 Experimental validation

An SEM observation is performed on samples of real pistons after more than 140 000 cycles.

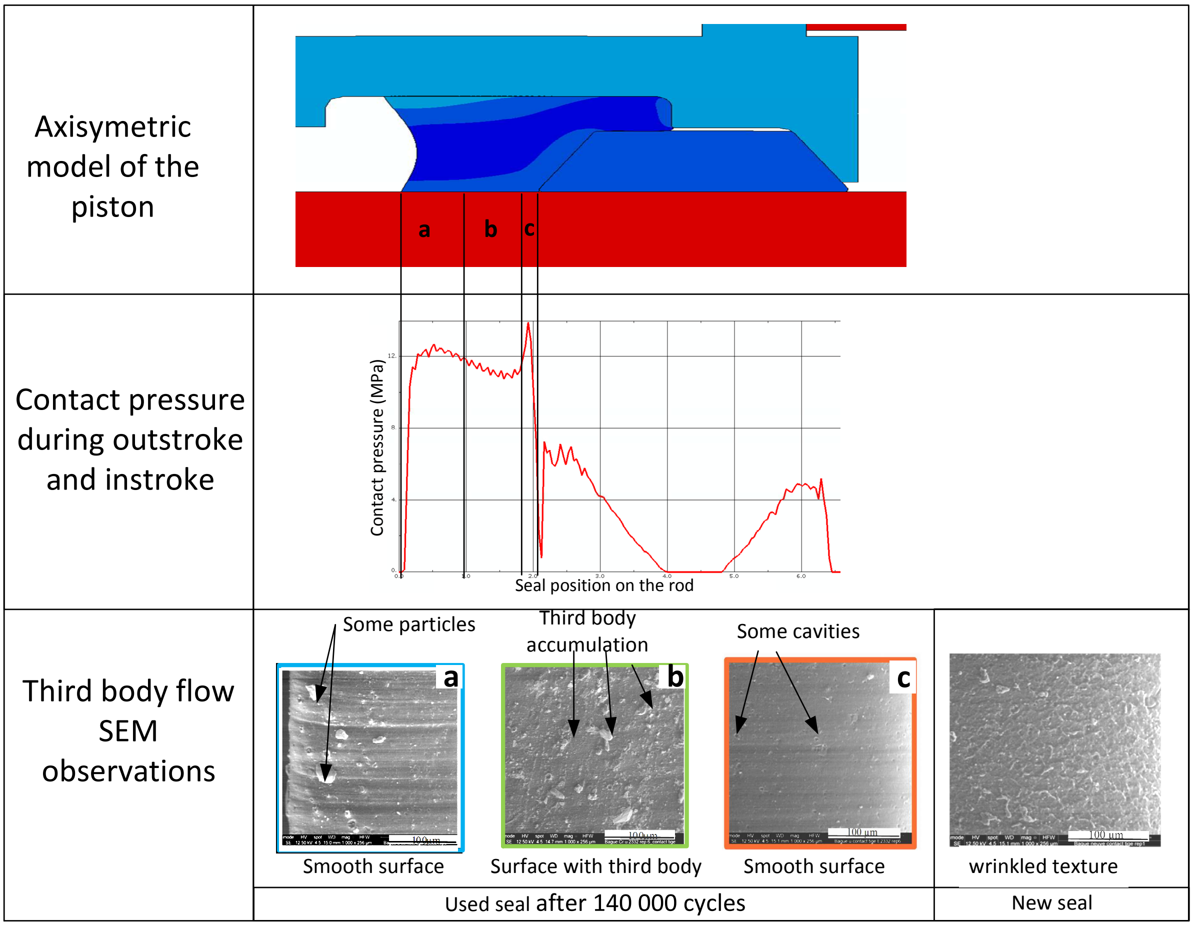

Figure 11 : Experimental validation of the numerical design

The texture of used samples is compare with the texture of a new seal. During the operating phase, wrinkles observed at the surface of a new seal are detached under the effect of the contact pressure and become the third body at the rod/seal interface. Then, the contact pressure peaks observed in simulations on both extremities of the rod/seal interface are consistent with the smooth texture observed in pictures a. and c. of SEM observations (Fig. 11). These locations can be associated with larger degradation of the interface materials, triggering third body generation. In contrast, the low contact pressure area at the inner zone of the seal explains the particles accumulation observed in picture b. (Fig. 11). In fact, it is difficult for a particle to cross a contact pressure peak. Thus, both peaks play the role of a third body barrier especially as those peaks exist during all the operating phases.
It appears that the numerical model provides results which are consistent with the experimental observations. It provides some understanding of the wear process at the rod-seal interface, and demonstrates that the seal design is correct.

5 Conclusion

This study demonstrates that a methodology based on a contact pressure criterion can help for a quick design of reciprocating sealing systems. It confirms the fact that the rod/seal contact pressure can be considered as the main parameter affecting the sealing performance. It is important to note that the seal design is not sufficient to achieve the reliability of a reciprocating system. It is also important to consider manufacturing technology, and the assembly of the sealing system.
The next step of this work is the study of the mechanism effect on the contact pressure through a 3D FE model, in order to investigate the effect of a possible coaxiality default on the contact pressure between the rod and the seal.

6 References

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