Numerical Simulation of Radial and Axial Compressed Elastomeric O-Ring Relaxation

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Abstract

The elastomeric O-ring gaskets are often used in pressurized hydraulic and pneumatic equipments to ensure their sealing. The quality of the O-ring is measured by achieving the desired tightness level. The satisfaction level of practical operation depends on the consistency of long-terms mechanical properties located during the useful life and on the conditions of assembly installation and the O-ring location. Indeed, the gasket is pressed on flat faces or housed in specially arranged grooves for which the dimensions influence greatly the assembly behavior. In this article, an axisymmetric finite element model is proposed to simulate the O-ring relaxation behaviour during the few first days of its installation in both the unrestrained and restrained radial and axial loading cases. The contact stress profiles and the peak contact stresses are determined versus the time relaxation in order to specify the working conditions thresholds.

Index terms— O-ring, contact pressure, relaxation, FEA, Radial and axial loading.

1 Nomenclature

F total compression load (N) e initial O-ring axial displacement (mm) d the O-ring cross-section diameter (mm) D the O-ring mean diameter (mm) C the ratio e/d R the axial compression ratio b the contact width between the gasket and plat (mm) x radial position compared to the vertical axis of the Oring cross-section (mm) p o maximum contact pressure value or peak contact stress (MPa) E relax relaxation modulus (MPa) E j elastic modulus for gasket (MPa) ? j coefficient ? j relaxation time (s) ? j viscosity (MPa.s)

2 Introduction

The elastomeric O-ring gaskets are widely used in hydraulic and pneumatic equipments to ensure the sealing of the shaft, the pistons and the lids. The correct operation of the O-ring is conditioned, on the one hand, by the maximum value of the contact pressure created during the O-ring compression and on the other hand by the preservation in operating stage of a minimal threshold value below which the sealing of the joint is blamed. Therefore, the evaluation of the maximum value of contact pressure evolution in time has a primary importance to ensure the correct O-ring function during its nominal useful lifetime.

There are two O-ring types, static and dynamic seals. The first seals are classified as either axial or radial squeeze. For the second ones three main classifications are considered, reciprocating, rotary and oscillating seals. The O-ring seals are placed in grooves specially arranged to block their possible displacement and to assure the maximum contact pressure giving the maximum performance. The dimensions of these grooves are provided by the seals manufacturers in tables which are selected according to the cross section O-ring diameter. The origin formulas giving the groove dimensions are not known and the theoretical approach used by the manufacturers is not published.

Several teams were interested in O-ring assembly used in various industrial services. The published works on this subject propose the same analytical approach based in all cases on the Hertzian classical theory [1]. Others studies have more experimental studies using various assemblies to characterize the O-ring itself in traction and compression loads and to model his real behaviour. In the third shutter, finite elements models are developed to numerically simulate assemblies with the O-ring.

George and al. [2] used a finite elements model to study the behaviour of the O-ring compressed between two plates. The gasket characteristics were introduced into the program according to parameter defining the total deformation energy or by using the Neo-Hookean model. The results of this analysis were compared with those of several experimental studies and analytical approaches based on the Hertzian theory. Dragoni et al. [3] propose an approximate model to study the O-ring behaviour placed in rectangular grove.

The work of Green and al. [4] reviews the majority of used O-rings configurations. Finite elements models were developed considering hyperelasticity behaviour. The results of these models were confronted with those of empirical studies. New relations expressing the maximum contact pressure and the width of contact were proposed. Rapareilli and al. [5] present a validation of the experimental results by a numerical model which regarded the joint as an almost incompressible elastic material. The effects of the fluid pressure as well as the friction effect between the gasket and the shaft are studied.

In an experimental study [6], the authors tried to determine the influence of the fluid pressure on the contact pressure, which ensures of sealing as well as the ageing deterioration of the joint. Kim and al. [7,8] tried to find an approximate solution for the mechanical behaviour of the O-ring joints in several configurations. The influence of the friction coefficient is highlighted. An experimental study was carried out to find more realistic elastic modulus values for elastomeric O-ring. They compared their results with those obtained in experiments and by the finite element analysis. They found that the values given by the Lindley [9,10] to calculate the compressive force are similar to those determined by the finite elements model. The O-ring relaxation was treated by the reference [11] where the degradation is caused by oxidation or nuclear irradiation. The authors describe several improvements to the methods used in there previous studies.

In this study, axisymmetric finite element models are proposed to simulate the O-ring relaxation behaviour during the few first days of its installation in both the unrestrained and restrained radial and axial loading cases. Figure **??** presents the studied cases. The contact stress profiles and the peak contact stresses are determined versus the time relaxation in order to specify the working conditions thresholds. The effect of the temporal variation of the longitudinal elasticity modulus as well as the influence of the axial compression ratio will be analyzed.

3 II.

4 O-ring Mechanical Characteristics

Most of the previous work dedicated to study the O-ring gasket behaviour use the same analytical model based on the Hertzian pressure contact theory. By adopting this classical theory, Lindley [9,10] developed a simple approximate formula, relation (2.1), expressing the compressive force, F, according to the ratio of initial compressed displacement by the crosssection O-ring diameter, .

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The same theory allowed finding out the contact width, b, and the maximum value of the contact stress po, according to the formulas (2.2) and (2.3).

(2.3)

The contact pressure distribution according to the radial position on the gasket is given by the equation (2.4).

(2.4) These formulas do not utilize the mechanical characteristics of the components in contact with the seal. Only the O-ring elastic modulus, E, are used. Practically, the most used O-ring material has a hyperelastic or viscoelastic behavior. For the relaxation studies, the viscoelastic behavior is the best choice to take into account the effect of hyperelasticity and time variation of mechanical properties.

In previous study [12], it was confirmed that the same equations remain valid for the time evolution study of the O-ring behaviour but using a variable elastic modulus according to time, called relaxation modulus E relax . The viscoelastic behaviour of the gasket is given by the modified Maxwell model [13], presented in figure **??**. The relaxation modulus is defined by: (2.5)

5 With (2.6)

The relaxation modulus in Eq. (2.5) becomes:

(2.7)

The initial elasticity modulus, E 0 , and the coefficients ? J , called Prony series coefficients, are deduced from the experimental data of the reference [14]. The time variation of this relaxation modulus is presented in figure 3.

The relaxation study consists to evaluate the variation of the contact stress versus time, when an initial displacement, e, characterized by a compression ratio R, given by the equation (2.8), is imposed to the gasket. For each axial compression ratio, R, the variation of the contact pressure distribution as well as the change of the contact surface width are determined with the relations (2.2), (2.3) and (2.4). (2.2)2 0 2 1) (? ? ? ? ? ? ?

= b x p x p ? ? ? + = j t j relax j e E E t E ?) (j j j E ? ? = , ? = E E j j ? and ? + = ? j j E E E 0 ? ? ? ? ? ? ? ? ? ? ? = ? ? j t j relax j e E t E) 1 (1) (0 ? ? 2. 100 200. e R C d = × = III.

6 Finite Element Models

The study of the O-ring relaxation, in the four cases presented in Fig. ??, when it is compressed by the application of a constant displacement, consists in following the time evolution of the contact pressure and contact width. To achieve this goal, an axisymmetric finite elements model of each assembly was produced using ANSYS software [15] as showed for RAL case in figure ??.

Since the problem is axisymmetric and the median horizontal or vertical plane, respectively for axial or radial loading, cutting the O-ring in two equivalent parts is a symmetry plane, the joint is modeled by a halfdisc with four node's 2D planes elements. The O-ring material is regarded as viscoelastic characterized by the Prony coefficients. The rigid component is modeled by rigid elements for which the displacements are blocked in all directions. The geometric and mechanic characteristics of the O-ring joint are summarized in table ??. In order to check the influence of the O-ring rigidity two initial Young modulus values are considered. The mesh refinement is optimized to have the convergence while using less computer memory capacity.

The value of the imposed displacement on the free seal surface is calculated by the axial compression ratio, R, which varied between 5 and 35 % compared to the O-ring cross-section diameter. Thereafter, the contact pressure distribution is recorded according time.

IV.

7 Results and Discussions

The work presented in this paper is the continuation of an previous work [12] in which it was concluded that the classical theory of Hertzian contact, developed initially for steady operation, remains valid for the relaxation and it is in good agreement with the used finite element model. Consequently, we will just use the finite element analysis to compare the case where the joint is free to move in the direction perpendicular to that of the applied compressive stress and the case where this movement is blocked by placing the seal in a rectangular groove.

After the application of an initial load on unrestrained O-ring upper surface (UAR case), an initial displacement is taking place and will be kept fixe over the relaxation time. Figure ?? presents the variation of the contact pressure between the O-ring and the fixed component when the compression ratio is equal to 5, 15 and 30% for various relaxation times. When the relaxation time increases the contact pressure decreases and the relaxation rate can be calculated for each applied compression ratio. In addition, the contact area is larger when the compression ratio is greater. On the other hand, whatever the relaxation time for the same compression ratio, the contact width remains the same. Thus, when the compression ratio increases from 5% to 15% and to 30%, the contact width, successively, is evaluated to 20%, 38% and 62% of the seal cross section diameter.

For the restrained axial loading case, Fig. ?? shows the contact pressure distribution for R=20% and for two relaxation time, 8 and 24 hours. The same observations, as for UAL case, remain valid but the contact pressure values decreases. The limitation of the radial displacement by the groove creates a pressure contact distribution along the contact surface side of the groove. For each relaxation time, the curve is symmetric about the seal section center. This shows that the influence of the radial position is negligible on the symmetry of contact pressure distribution even in the groove presence.

In order to perceive the importance of the groove on the contact pressure, Fig. ?? compares the two cases of axial loading for different initial elastic modulus. The contact pressure ratio is represented as a function of the relative radial position. For the same compression ratio, the stress value in RAL case is larger than in the URL case. The axial contact area is almost identical in the two cases. So it is clear that the primary advantage of placing the seal in a groove is to increase the contact pressure which ensures more sealing with the same compression ratio. However, the chosen groove dimensions are not optimized to provide better performance. In aerospace applications, the most used standard is SAE AS5857A [16] that provide standardized gland (groove) design criteria and dimensions for elastomeric seal glands for static applications.

To know the extent of the pressure contact, the contact width is shown in Fig. 8 as a function of compression ratio. It is evident that when the O-ring is more compressed the contact area is larger. The analytical values of the contact width given by Eq. (2.2) are closer to those given by the finite element analysis in the URL case. Indeed, the used analytical model does not take into account the presence of seal-groove lateral contact.

For the piston rod and in the static state, the Oring is solicited radially in the perpendicular axis direction. This configuration is represented by the RRL and URL cases in Fig. ??. To illustrate the effect of compression ratio on the initial contact pressure distribution, Fig. ?? compares these two cases and highlights the creation of the contact pressure at the lateral groove-seal contact surface. At the radial contact, the pressure is greater when the compression ratio increases or when the seal is placed in the groove. For the axial contact, the contact pressure reaches almost 50% of the maximum value recorded at the radial contact.

Figure ??0 presents the distribution of the contact pressure at contact interface between the time periods and for the same compression ratio, the contact width is the same.

Figures ??1 and 12 compare the reductions in maximum pressure contact due to the relaxation at different compression ratio for URL and RRL cases. For the same compression ratio, the variation of this stress is more significant at the first hours and it stabilised after. Indeed, for both cases and whatever the compression ratio, the ratio of the maximum contact pressure by the initial elastic modulus decreases with a rate of 56% from its initial value when the relaxation time is 12 hours while it is about 57% after 72 hours.

V.

8 Conclusion

This study shows that the installation of the seal in a groove reduces the compression ratio needed to create a contact pressure threshold provided for sealing the assembly. On the other hand, the first few hours after installing are critical and must be controlled in order to ensure the proper O-ring functioning. The geometric configuration of the groove used in our model is idealized which requires to verify the results obtained when the geometrical defects are introduced to the model. An experimental study is expected to confirm these results and characterize the mechanical behavior of industrial O-rings.

9 O-ring

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Figure 1: (2. 8)

[1] [2] [3] [4]

[1] July © 2012 Global Journals Inc. (US) Numerical simulation of radial and axial compressed elastomeric O-ring relaxation

[2] (A) © 2012 Global Journals Inc. (US) 2012 July groove and the seal for different time periods. The stress decreases rapidly versus time for all axial position except at the contact area. It is to be noted that for all

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[4] (A) © 2012 Global Journals Inc. (US) 2012 July

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Figure 2: Figure 3 :

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Figure 3: Figure 4 :

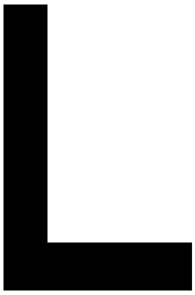

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Figure 4: Figure 5 : 6 :Figure 7 :Figure 8 :

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Figure 5: Figure 9 :Figure 10 :

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Figure 6: Figure 11 :Figure 12 :

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