Finite Element Analysis of O-ring Performance Reinforced by a Metallic Core

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Abstract

This work presents a comparative analysis between a homogeneous O-ring and an another composed of the union of two materials. Two axisymmetric finite element models developed in this article using the ANSYS software study the seals behavior during their deformations. The results of the numerical model are compared with those of the analytical approach based on Hertz's contact theory. The introduction of a metal core inside the elastomer O-ring can improve not only the seal's resistance but also the maximum value of the contact pressure.

Keywords: O-ring, reinforced O-ring, stress, analytical model, finite element model, contact pressure.

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1 Introduction

The O-ring is one of the most popular seals used in pneumatic and hydraulic systems in either static or dynamic applications because of its low-cost price and ease of installation. The proper functioning of an O-ring sealing system is due to the initial seal compression, which generates contact pressures that can prevent leaks.

Since the O-ring is used in a wide range of applications, it is subject to many types of defects that affect the life and reliability of assemblies with this type of seal differently.

The most common O-ring failure mode is the abrasion occurred from repetitive contact between the O-ring surface and the housing resulting in excessive friction between the two. Improper lubrication or surface finish of the metalwork can aggravate the risk, as can let abrasive contaminants into the sealing system.

Another common failure is due to compression set. Twisted O-rings with spiral marks or surface cuts are defects that can occur due to the variable compression of the seal cross-section in dynamic applications. In static applications the O-ring has usually been twisted when mounted in its groove. Explosive decompression is another failure in which pressurized gases penetrate into the sealant material. The seal surface may be blistered, cracked, marked with deep splits or completely ruptured in the worst examples.

The main objective of the research studies carried out in this field is the improvement of the characteristics of the O-rings to avoid these defects and others during normal use. Several researchers have investigated the O-rings sealing performance. Dragoni et al. [1] developed a theoretical model to describe the mechanical behavior of an elastomer O-ring installed in a rectangular groove. They dealt with the influence of the groove width variation and the friction coefficient on the seal mechanical behaviour. Karaszkiewic [2] proposed equations to determine the geometric distortion and contact forces of an O-ring mounted in a groove. Some of these equations are verified against the experimental results of other researchers. The proposed equations may be useful in engineering practice for solving O-ring operating problems. Diany et al. [3, 4] have shown that the equations of the classical Hertz contact theory remain valid for the study of O-ring relaxation during the first day of installation without a groove.

They have also shown that the installation of the seal in a rectangular groove reduces the value of the crushing necessary to create a contact pressure necessary to seal the assemblies. In another study [5], they proposed an analytical approach, based on the results of the Brazilian test of compression on a disk, to determine the stress distribution in the cross-section of the O-ring. The results of this analytical approach have been compared to those of finite element analysis. Zhang et al. [6] have shown that an O-ring can be replaced by a D-ring in static applications. They also demonstrated that in dynamic reciprocating installations, the D-joint can last longer than an O-ring when the coefficient of friction is low. Di Wu et al. [7] proposed an analytical model for calculating stress distributions of an O-ring inserted into a groove and undergoing both a clamping force and a fluid pressure. This model expresses the relationship between stress distribution, Young's modulus and Poisson's ratio. The effectiveness of the proposed model was validated by comparing its results with those of finite element analysis.

When choosing the material of the O-ring, many factors must be taken into account: the temperature, the pressure ranges of service, the fluid confined by the sealing system and the value of the coefficient of friction. To improve the performance of the seal and best meet the effects of these factors, this work proposes to use composite materials to build such O-ring.

These materials successfully replace conventional metals in many applications. Resistance, lightness, high specific stiffness, affordability and ease of manufacture are just some of the advantages of composites over traditional metals. The large scale introduction of composite materials in engineering applications requires a thorough knowledge of the characteristics of these materials. Many studies have been conducted to develop equations describing their behaviour. Chamis and Sendeckyj [8] presented a synthesis of the different methods and models used to compute the properties of composites. Halpin et al. [9] developed empirical generalized equations to determine the properties of composites.

According to the literature, seals are made from one or more materials. Seals bonded with either PTFE for normal temperatures or metallic sealing surfaces for high pressures and temperatures are available in many variants [10]. The manufacturers' catalogs provide application areas and installation instructions for these products. For elastomer O-rings, one of the methods used to circumvent these high pressure and temperature problems is to change the O-ring material to PTFE or to use metallic O-rings made of stainless steel or copper, but with unsatisfactory results [11].

To improve the performance of assemblies with O-rings, this work presents a comparative analysis of the behaviour of a homogeneous O-ring and an O-ring composed of the union of two materials.

2 Modelling the Studied O-rings

Currently, the O-rings used for this investigation are made of homogeneous isotropic and hyperplastic materials. This study proposes to compare the behaviour of a conventional O-ring with a new two-part seal. The new gasket is composed of a metal core and an elastomeric envelope. Figure 1 shows the cross-section of the two types of seal.

The equivalent Young's modulus of the reinforced seal, E_r, by the metal core can be calculated using the Halpin-Tsai equations presented in numerous works on the mechanical behaviour of composite materials [12–15]. Indeed, the Young's modulus of the compound seal is:

$$E_r = \frac{1 + \xi\eta\frac{D_1}{D_2}}{1 - \eta\frac{D_1}{D_2}} E_e \tag{1}$$

With

$$\eta = \frac{\frac{E_a}{E_e} - 1}{\frac{E_a}{E_e} + \xi} \tag{2}$$

Where E_r represents the equivalent module of a seal composed of the union of two materials. E_a and E_e represent respectively Young's modulus of the metalcore and elastomer casing. D_1 is the diameter of the cross-section of the core and D_2 is the diameter of the cross-section of the casing. The term ξ is an empirical factor used to make the Equation (1) consistent with experimental data. It measures the reinforcement by the composite material which depends on the geometry of the metal core, the geometry of the elastomeric envelope and the loading conditions. Eugenio Giner et al. [16] performed a finite element analysis concluding that a value of this empirical factor ξ of 1.5 is a better estimate than the value of 2 proposed by Halpin-Tsai [9] in this work $\xi = 1.5$ is used.

3 Conventional Analytical Theory

Hertz contact theory proposes to determine the contact pressure between two surfaces of two bodies in contact, as a function of the applied normal force and the mechanical and geometrical characteristics of the bodies. Based on this classical theory, Lindley [17, 18] developed a simple Equation (3) expressing the compression force, F, as a function of the initial crushing of the O-ring, C:

$$F = \pi D D_2 E\left(1.25C^{\frac{3}{2}} + 50C^6\right) \tag{3}$$

The value of the contact width, b, and the maximum contact pressure, p_0, are given by Equations (4) and (5). The distribution of the contact pressure as a function of the radial position on the seal is given by Equation (6).

$$b = D_2\sqrt{\frac{6}{\pi}\left(1.25C^{\frac{3}{2}} + 50C^6\right)} \tag{4}$$

$$p_0 = 4E_c\sqrt{\frac{\left(1.25C^{\frac{3}{2}} + 50C^6\right)}{6\pi}} \tag{5}$$

$$p(x) = p_0\sqrt{1 - \left(\frac{2x}{b}\right)^2} \tag{6}$$

This analytical model reflects the relationship between the distribution of contact pressure and the mechanical and geometrical properties of the O-ring. Therefore, these equations are valid for comparing the performance of the O-ring reinforced with a metal core and a conventional seal. The two seals have the same overall dimensions, D and D_2, and they have an elastic behavior characterized by the equivalent elasticity model, E. This modulus is calculated for the classical seal, E_c, with the Equation (7).

$$E_c = 4(1 + \nu)(C_1 + C_2) \tag{7}$$

With C_1 and C_2 are the Mooney-Rivlin coefficients.

For the reinforced seal the equivalent elastic modulus is computed by relation (1).

4 Finite Element Models

The greatest difficulty in modeling elastomers is the nonlinear stress-strain behavior [19]. Thus, the laws of mechanical behavior of an O-ring must be formulated within the framework of a large deformation model [20]. One of the models frequently encountered to describe the mechanical behavior of elastomers is the Mooney-Rivlin hyperelastic model. This one allows to describe correctly the behavior of elastomers [21]. In this work the Mooney Rivlin model with two parameters is used.

In order to study and compare the mechanical behavior of the O-ring reinforced by a metallic core to a classical O-ring, two axisymmetric finite element models, presented in Figure 1, were developed using the Ansys

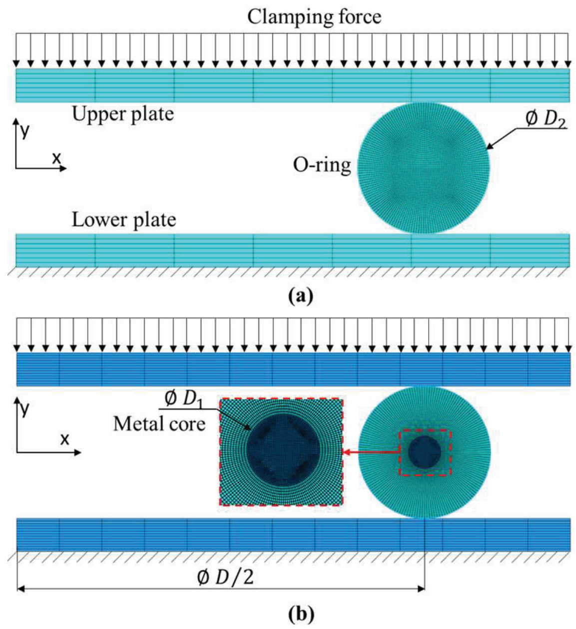

Figure 1 Installation of (a) the classic O-ring, (b) the O-ring reinforced by a metal core.

software [22]. In both cases, the O-ring is modeled by a disk with four nodes 2D plane elements (PLANE182) compressed between two rigid plates. Displacements of the lower plate were canceled in all directions. The upper plate whose displacements are cancelled along the x axis, is loaded along the y axis by a uniformly distributed clamping force. The analysis types is static analysis with geometric nonlinearity (NLGEOM, ON and SOLCONTROL, ON). Contact elements, CONTA171 and TARGE169, are used to simulate the reaction between the elements that are in contact. The contact elements are constructed using 2D 2-Node Surface-To-surface contact approach. The interface between the elastomer and the metal core in the finite element model is merged by using NUMMRG command. A frictional interaction is defined for the contact pairs and the friction coefficient of 0.2 is applied (via the MP command). The contact formulation used is the “Lagrange multiplier on contact-normal and penalty-on-tangent” (KEYOPT (2) = 3 on CONTA171). The geometric and mechanical characteristics of the two seals are summarized in Table 1.

The value of the clamping force imposed on the upper surface of the upper plate varies between 100 N and 300 N.

Table 1 Mechanical and geometrical characteristics of the seals

$D(mm)$	16.35
$D_1(mm)$	$0.25D_2$
$D_2(mm)$	2.65
$E_c = E_e(MPa)$	13.80
$E_a(MPa)$	210000
$E_r(MPa)$	16.099
C_1	2.334
C_2	−0.034

5 Results and Discussions

Preventing leaks in pressurized assemblies with O-rings requires control of the installation and operating conditions that affect the contact pressure distributions at the seal-structure contact surfaces. The threshold values of the contact pressure depend on the geometrical characteristics of the various components, the seal material, the clamping load and the operating conditions in fluid pressure and temperature.

The proposed analytical model calculates the maximum contact pressure as a function of the clamping force initially imposed on the upper plate. The finite element model is used to follow the progression of the deformation and to determine the stress concentration zones in both assemblies cases, either with a conventional O-ring or with an O-ring reinforced by a metal core.

The purpose of introducing the metal core into the elastomer seal is to improve the behavior of the seal. Figure 2 shows the deformation of the joint in the two configurations under the effect of a clamping force of 300 N. The axial and radial deformations of the conventional seal, Figure 2(a), are greater than those of the reinforced seal, Figure 3(b). This is due to the presence of the metal core in the second seal and which increases the resistance to compressive deformations.

Figure 3 shows the distribution of the Von-Mises stresses in the cross-section of the two seals, under a clamping load of 300 N. It is clear that the two seals do not react in the same way and that the stresses in the seal with metal core are larger and mainly in the area corresponding to the location of the metal part. Indeed, the maximum stress in the conventional seal is 4.26 MPa, against that in the seal with metal core is about 23.26 MPa.

Figure 4 shows the effect of the presence of the metal core on the contact pressure distribution determined by finite elements analysis. For both

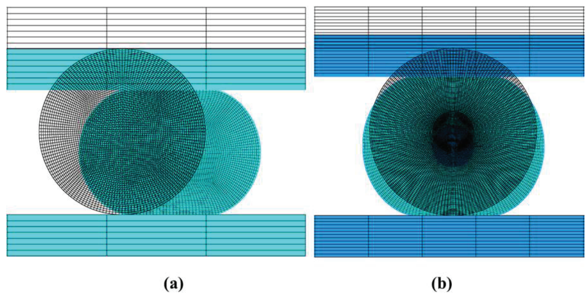

Figure 2 (a) Progression of the O-ring deformation. (b) Progression of deformation of the O-ring reinforced by a core (clamping force of 300 N).

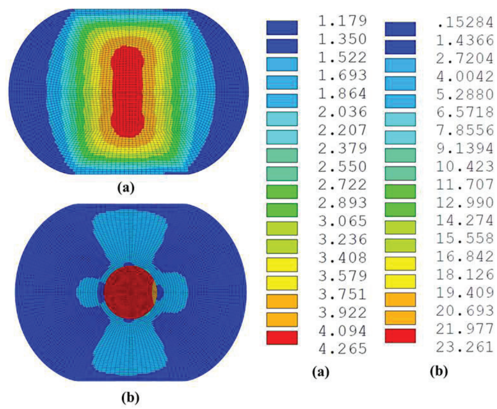

Figure 3 The distribution of Von-Mise stress obtained by the EF model in MPa. (a) O-ring. (b) reinforced O-ring.

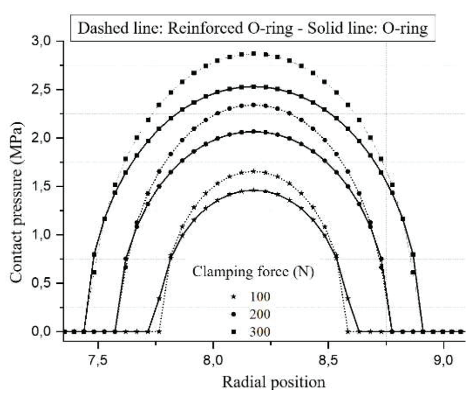

Figure 4 Variation of contact pressure of O-ring and reinforced O-ring by FE model.

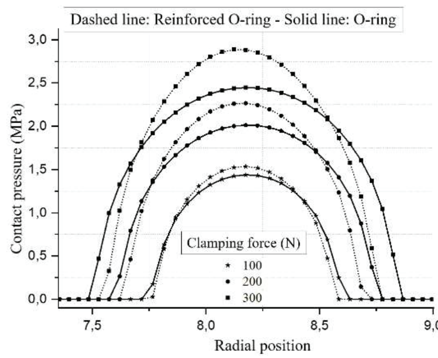

Figure 5 Variation of contact pressure of O-ring and reinforced O-ring by the Hertz contact theory.

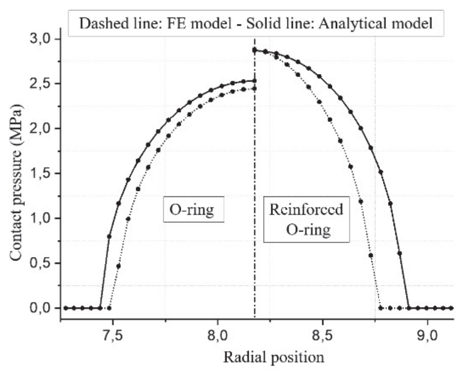

Figure 6 FE and Analytical models comparison at a clamping force of 300 N.

configurations, the contact pressure distribution is shaped like that of the conventional Hertz contact pressure. For the same clamping load, the contact width in the case of the conventional seal is slightly larger than that in the case of the reinforced seal. On the other hand, the maximum value of the contact pressure is much greater in the case of the reinforced seal. Indeed, for a clamping force of $300\ N$, the maximum contact pressure, in the case of the reinforced seal is 15% greater than that in the case of the conventional seal. This reinforces the tightness of the assembly provided with seal with metal core.

The analytical model allows, using Equations (3) to (6), to determine the contact pressure distributions for both types of O-rings. Figure 5 gives these distributions which are similar to the finite element analysis one presented in Figure 4. To compare the two approaches, analytical and numerical,

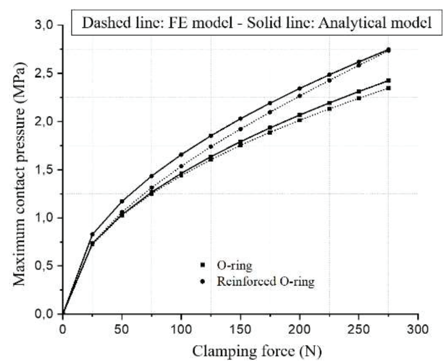

Figure 7 Variation of Maximum contact pressure of O-ring and reinforced O-ring against Clamping force.

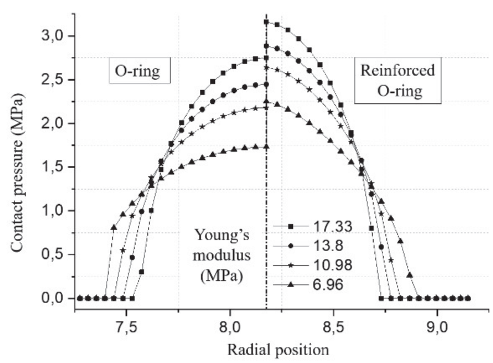

Figure 8 Variation of contact pressure of O-ring and reinforced O-ring for 4 Young's modulus values at a clamping force of 300 N.

Figure 6 shows the contact pressure distribution for a clamping force of 300 N of both models. For the maximum value of the contact pressure, the difference between the two approaches is about 3.4% in the installation with a conventional seal and about 0.5% in the case of the reinforced seal installation. On the other hand, for the contact width defined by the initial compression ratio, C. The comparison between the values of Equation (4) and the finite element analysis results indicates a maximum difference of 13% in the reinforced O-ring case and 6.5% in the conventional O-ring case. These differences are due to the weakness of the analytical model which involves only the O-ring outer diameter.

The performance of the O-ring is measured by its ability to prevent leakage by generating rather large contact pressures when clamping over a sufficiently wide contact area. Indeed, the value of the imposed clamping load conditions the values of the maximum contact pressure and the contact zone width. Figure 7 shows the variation of the maximum contact pressure, determined analytically and by EFA, for the two studied seal types, and as a function of the clamping load. For the same load, the maximum contact pressure generated when using a reinforced seal is greater than that generated by the conventional seal. It is also clear that this contact pressure increases when the clamping load is greater. The curves of this figure also allow predicting the maximum value of the fluid pressure that the seal can ensure its confinement inside the assembly, for a given clamping load and without taking account the interaction between the clamping and the application of the fluid pressure. To maximize the performance of the O-ring, it is important to choose the rigidity. Figure 8 shows the effect of the Young's modulus of the elastomer on the contact pressure distribution for both seals types. The results of the finite element analysis confirm that the pressure and the contact width depend on the elastic modulus value. When the stiffness of the seal is greater, the contact pressure is greater, and the contact width is small. Improving the rigidity of the O-ring by the use of a metal core increases the contact pressures even if the contact width becomes smaller.

6 Conclusion

This study has demonstrated that the presence of a metal core in an O-ring increases its compressive strength and improves the maximum value of the contact pressure which enhances the seal.

The FE model confirms the classic form of Hertz contact pressure. On the other hand, the analytical model remains weak since it only involves the outer diameter of the O-ring for the calculation of the contact width.

Disclosure Statement

No potential conflict of interest was reported by the authors.

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