Stress field and contact force analysis in a low friction seal
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Abstract

The behaviour of a low friction elastomeric seal for piston in pneumatic cylinders is investigated using a numerical model which takes the seal’s geometrical assembly conditions and operating pressure in cylinder chambers into account. The model assumes plane strain because of the high ratio of average seal diameter to radial extension. Seal material was characterized with a special numerical/experimental procedure. The study made it possible to determine contact pressures between seal and cylinder chamber in the absence of tangential adhesion or friction forces.

1 Introduction

Pneumatic cylinder operation depends basically on the behaviour of the seals on the moving piston/rod assembly. In this context, the behaviour of the piston seals is particularly important: the need to provide double sealing (i.e. towards the cylinder’s front chamber and rear chamber) and the size of the piston mean that the friction forces of these seals predominate over those of the rod. Moreover, the friction forces depend on the normal forces exchanged between seals and barrel, and hence on the distribution of contact pressures. In general, the tighter the seal fits in the barrel, the higher the friction forces will be though sealing will also be better.

It is thus necessary to find an effective trade-off between the need to reduce friction forces and the need for good sealing.
Low friction seals capable of performing effectively with low resisting forces have recently attracted considerable interest. Such seals permit the regular movements needed in pneumatic servosystems and positioners, and waste less energy as a result of friction. Precisely because low friction seals are relatively recent developments, very little information is available in the literature. Much more extensive investigations have been made for sliding seals, hydraulic types in particular [1], [2] and for rotary shaft seals [3], for which both stress states and contact conditions have been analyzed. Consequently, the authors began an investigation of the behaviour of low friction pneumatic cylinder seals with the objective of determining the normal pressures exchanged, which are essential for sealing purposes, and the friction forces in conditions similar to the effective lubrication conditions. This paper discusses the results of the initial stage of this investigation, in which contact pressures were calculated. To this end, a finite element calculation code was employed, using experimental data to characterize seal material. Naturally, the operating pressures in cylinder chambers were taken into account, as they are a determining factor as regards the forces exchanged with the barrel. The presence of friction forces, on the other hand, was not considered for the moment, so the study provides a first approximation with piston stationary and zero adhesion forces, or piston moving with negligible friction forces. Calculation can also be used for seal life predictions, as it is possible to identify the more highly stressed areas where the higher pressures are concentrated.

2 The low friction seal

The seal is shown in Figure 1. It is a double-acting lobed seal designed for single installation on the piston, as it can withstand pressure acting in both directions. It consists of NBR elastomer and is normally used with a greased cylinder, making it suitable for operation with unlubricated compressed air.

Figure 1: The seal. Figure 2: Flow diagram for determining the equivalent elastic modulus.
Seal geometrical and mechanical characterization was performed using a profilometer for dimensional measurements and a dynamometer test machine to determine elastic modulus. Using an equivalent modulus of elasticity $E$ made it possible to perform the study with a linear field analysis calculation code, which is justified by the presence of small strains. This equivalent modulus of elasticity was determined using a test method based on direct comparison of the results of parallel experimental and numerical tests, and by converging on the exact value through successive approximations.

The flow diagram for the procedure followed is shown in Figure 2. The seal is first subjected to an axial crush test, after which experimental axial force $F_e$ is measured as a function of displacement of seal nodes at contact surfaces. A first tentative value $E'$ of the elastic modulus for use in the numerical model of the seal is assumed at a given crush force value. As the rubber effective stress-strain characteristic ($\sigma-\varepsilon$) is not linear, modulus $E$ in reality depends on the value of $\varepsilon$. With the assumptions indicated above, however, the value of a single equivalent elastic module corresponding to current strains was identified iteratively.

A tentative value can be obtained from formula (1):

$$E' = 2.31 \frac{H}{100 - H}$$

where $E'$ is expressed in (MPa) and $H$ in degrees (IRHD). This expression is valid for small strains. For the seal under test, whose hardness is $H=75$ (IRHD), it gives a value for $E'$ of 6.93 (MPa), which is very close to the values indicated in the literature [4], [5], [6], [7], [8]. With this tentative value $E'$, the numerical model is used to calculate the resultant of the nodal reaction forces on contact surfaces $F_{fem}$ corresponding to the crush force. This force is then compared with the force determined experimentally ($F_e$) on the dynamometer test machine. If $F_{fem}$ is less than or greater than $F_e$, the value of $E$ is respectively increased or reduced until the two force values are equal. This procedure was repeated for several values of the crush force $F$. These values were selected on the basis of seal strain under actual operating conditions, with seal installed in its seat and in the presence of supply fluid. These seal operating conditions were evaluated with a model using increasingly close approximations of the modulus of elasticity. Though laborious, this procedure was able to reach convergence with a relatively small number of iterations, in part because the initial value of $E$ was selected with care. The equivalent modulus of elasticity thus calculated was 6.0 MPa, while a Poisson’s ratio of 0.49 was assumed.
4 Numerical model

A numerical model was prepared in order to study contact forces between seal and barrel under actual operating conditions. The model thus envisages the seal as being installed in its actual seat in the piston, contacting the barrel externally, and subject to the pressure differential in the cylinder chambers. Element discretization was performed on the basis of the material’s isotropy homogeneity and geometric axial symmetry. These characteristics, associated with a low ratio of overall radial dimensions to mean seal radius and with small displacements on the part of the points of a seal section, make it possible to adopt a plane strain numerical model [10], [11]. In this case the stress values for the axially symmetric case tend to approach those for plane strain, as the variation in the length of circumferential fibers is negligible [12].

Two models with thin shell triangular and quadrangular elements were created to evaluate convergence of results; these models have three and four nodes (linear model) and six and eight nodes (parabolic model) respectively.

5 Model validation

Together with photoelastic techniques, the use of numerical methods for studying seal contact forces is an effective analysis tool, but calls for experimental verification of results. The model presented here was validated by comparing the results obtained from a series of experimental tests performed using a suitable setup with the results of numerical simulation reproducing test conditions.

The test setup (Figure 3a) consisted of an aluminum piston with two circumferential seats having the same dimensions as the seal housing groove, and two aluminum jaws of the same diameter as the test cylinders. Four pieces of the seal, whose size is such that contact pressure distribution can be
considered constant, are installed in the seats on the piston. The latter is clamped between the two jaws and subjected to a strain corresponding to a load. The test was purely static, and was performed using a dynamometer test machine for the position control.

The model validation procedure was as follows. The compression force $F$ with which the two jaws clamp the seal specimens on the piston is measured for an arbitrary displacement $s$ on the part of the test machine’s moving crossmember. The experimental force $F_e$ thus obtained is then compared with the force calculated by numerical simulation on the two models, linear $F_{feml}$ and parabolic $F_{femp}$, where the boundary conditions are represented by the displacement of the force-displacement measuring machine’s crossmember. As shown in Figure 3b, results indicated that the best approximation of experimental data was obtained with the model provide with parabolic interpolation shape function. This model was thus used for numerical simulation.

6 Seal analysis in operating conditions

The study made it possible to evaluate seal behaviour while varying supply pressure from 2 to 8 bar. For numerical simulation, the seal was considered to be installed in normal geometrical conditions as regards both seat dimensions and cylinder barrel position. It was also assumed that pressure acts on one of the seal’s two side surfaces, while the other is forced against the seat. Sealing is achieved by contact between the seal’s outside diameter and the cylinder barrel, and between its inside diameter and the piston seat. As seal geometry is such that the only axis of symmetry is the diametric axis, the surface subject to the greatest deformation is on the outside radius. On the inside radius, the seal’s thicker geometry and larger surface area give it greater stiffness.

As an example of the results obtained, Figure 4a shows the deformed structure (solid line) and original structure (dash line) with the seal subjected to 4 bar pressure. As will be noted, maximum displacement occurs at the nodes under pressure load, while the lowest displacement values are found at the nodes in contact with the seat. Around the area of contact with the cylinder barrel, moreover, nodes belonging to the loaded edge tend to move towards the inside of the element, while those on the unloaded edge tend to move outwards. This latter condition is significant as regards sealing: it is clear that contact with the seat and with the barrel is greatest on the surface opposite to that on which the load acts.

The areas under the greatest stress are, of course, those which contact the side surface of the seat: in particular, the presence of geometrical discontinuities along the seal profile make this area the one in which the highest stress concentration occurs for all examined operating pressures.
Contact pressures between the seal surface and the cylinder barrel are of considerable importance, not only for seal tightness and friction forces, but also as regards seal wear. In fact, while a high pressure peak near contact is desirable in ensuring effective sealing, from the standpoint of durability a high contact pressure can lead to rapid seal wear and deterioration.

Figures 4b shows contact pressures between seal surface and cylinder barrel at a 4 bar supply pressure with no tangential adhesion or friction forces. Examination of contact pressure distribution made it possible to identify the contact area and evaluate the effective sealing surface as supply pressure varies. In particular, it was found that contact tends to move as operating pressure increases towards the edge which is not loaded by pressure as a result of rotation and translation on the part of the seal section.

7 Kinematic behaviour of the seal

Kinematic behaviour of the seal is illustrated in Figure 5, which shows that the seal section behaves essentially as an articulated system consisting of two rigid bodies hinged at the center of the section. When operating pressure is applied, the contact lobes are subject to a motion similar to rotation without sliding against the barrel at top and the seat at bottom. The central area, which acts as a virtual hinge, permits this motion inasmuch as its form makes it more yielding than the lobes. Because of this mechanism, deformation of the section does not result in an increase in seal-barrel contact pressure, even though seal tightness is maintained.
8 Conclusions

A method for predicting the contact pressures between low friction seals and the cylinder barrel was developed which takes the seal’s actual experimental behaviour into account.

Results indicated good sealing efficiency with a relatively small contact surface. In addition, the pressure peak on the barrel contact surface is not excessively high, thus making it possible to achieve a good compromise between sealing efficiency and seal durability.

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Symbols

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\begin{align*}
\sigma & \quad \text{Stress (kPa)} \\
\varepsilon & \quad \text{Strain} \\
l & \quad \text{Nodal displacement (mm)} \\
F & \quad \text{Crush force (N)} \\
F_{\text{fem}} & \quad \text{Force determined by the numerical model (N)} \\
F_{\text{feml}} & \quad \text{Force determined by the numerical model with linear elements (N)} \\
F_{\text{femp}} & \quad \text{Force determined by the numerical model with parabolic elements (N)} \\
F_e & \quad \text{Force determined by experimental test (N)} \\
H & \quad \text{Hardness (IRHD)} \\
E & \quad \text{Modulus of elasticity (MPa)} \\
E' & \quad \text{First tentative modulus of elasticity (MPa)} \\
E'' & \quad \text{Second tentative modulus of elasticity (MPa)} \\
p_c & \quad \text{Contact pressure (MPa)} \\
s & \quad \text{Test machine crush (mm)} \\
d & \quad \text{Distance on the contact (mm)}
\end{align*}
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References


5. ASTM American Society for the testing of materials D1415-68.


