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Abstract: This paper deals with the experimental evaluation of contact pressure at the interface between an elastomeric rod seal for pneumatic cylinders and its metallic counterpart without interposing any intrusive measuring device. A new test bench, which is able to measure the radial force exerted by a rod seal displaced at constant velocity on a sensorized portion of a cylinder rod over time, was designed and manufactured. The seal was pressurised to reproduce actual working conditions. A data postprocessing methodology was developed for an indirect evaluation of contact pressure starting from the experimental data set of the radial force exerted by the seal on the rod. At first, the measured radial force signal was filtered and properly fitted obtaining a differentiable function; then, contact pressure distribution was computed as a function of radial force time derivative, seal velocity and rod diameter. Preliminary experimental results are presented.

Keywords: contact pressure, radial force measurement, seals, pneumatics

1. Introduction

The analysis of the elastomeric seal performance is a demanding task not only because of the variety of used materials, shapes and dimensions, but also because of the relevant influence of many operating parameters, such as applied loads, working pressure, lubricating conditions, temperature, and sliding velocity. In particular, the study of the contact characteristics at the rubber-metal interface is of great importance because both the sealing capability and the strength of the friction force depend on the contact pressure.

Much research has been addressed at the evaluation of contact pressure between a seal and its counterpart. Bignardi et al. [1] processed experimental data obtained on a pneumatic lip seal by a photoelastic reflection technique to compute analytically the contact pressure distribution at the seal-rod interface. Pinedo et al. [2] built an analytical tri-dimensional eccentricity model of a rod lip seal extrapolating contact pressure distribution from finite elements simulations and experimental measurements. Hermann and Dabish evaluated numerically the contact pressure distribution between a pneumatic rod seal and its countersurface [3]. Many research works reported the use of pressure sensitive films to determine experimentally contact pressure distribution over a friction couple, in various research fields. As an example, Lee et al. [4] employed film sensors to study the influence of interference fit on the contact characteristics between an hydraulic lip seal and a rotating shaft. The same kind of sensors was employed for static measurements of local contact pressure in pneumatic applications by Belforte et al. [5]-[7] and by Manuello Bertetto et al. [8]. To overcome the main drawbacks of pressure sensitive films, namely non-negligible thickness and laborious positioning [9], some other authors reported non-contact measuring techniques. A non-intrusive ultrasonic method was used by Marshall et al. [10] in a bolted connection. Prokop and Müller studied the contact pressure of PTFE rod seals by performing radial force measurements at certain axial positions step by step; contact pressure was indirectly obtained from this signal [11]. Debler et al. [12] developed a test bench to measure the radial force exerted over time by a rubber cuff seal mounted on a rod equipped by a strain gauge sensor of special design; also in this case, contact pressure was indirectly obtained. Tests were carried out under unpressurised working conditions. In the same paper both linear and non-linear models of the seal were developed; the material stiffness and model coefficients (two-parameter Mooney-Rivlin material) were adjusted for a good agreement with the radial force measurements.

This work is addressed to the experimental evaluation of contact pressure at the interface between an elastomeric rod seal for pneumatic cylinders and its metal counterpart without interposing any intrusive measuring device. Results were obtained using a specifically designed test bench able to detect the radial force exerted by the rod seal displaced at constant velocity on a sensorized portion of a cylinder rod over time [13]. With respect to [12], which is the main reference of our test set-up, the measurement technique does not require the use of a specifically designed sensor; in fact, commercial uniaxial load cells were used to measure the radial force. Furthermore, the developed test bench permits the seal pressurisation to reproduce actual working conditions. A custom made data postprocessing methodology was developed for an indirect evaluation of contact pressure starting from the experimental data set of the radial force exerted by the seal on the rod. At first, the measured radial force signal was filtered and properly fitted obtaining a differentiable function; then, contact pressure distribution was computed as a function of the time derivative of the radial force, of the seal velocity and of the rod diameter. Preliminary results obtained on a type of rod seal for pneumatic applications are presented.

2. Indirect measurement of contact pressure

In order to contribute to a better understanding of the adopted measuring set-up, Fig. 1 shows a scheme of the seal assembled on the test bench. The rod seal, which is loaded by compressed air pressure on the left side, is displaced at constant velocity over a fixed cylinder rod (1), whose final portion is split into two sectors: the first one (1') is an extension of the rod body, the second one (2) is free to move in y-direction and is connected to two load cells, which are not shown in this scheme. These sensors perform the measurement of the overall radial force oriented in the negative y-direction and exerted by the seal under test on the rod portion (2). To this aim, a gap (3) is provided between the rod sectors, which permits a relative motion without friction. During the test, the seal initially slides over the full rod (Fig. 1a); then, it gradually engages the sensorized portion of the rod (Fig. 1b); finally, it completely covers the sensorized rod tip (Fig. 1c).

Figure 2 highlights that the radial force measured at a certain instant of time is the resultant of the axisymmetric contact pressure distribution over the corresponding contact area between the seal and the sensorized rod. While the contact area between the seal and the sensorized rod tip increases, the sensors detect an increasing radial force until a maximum value is reached.

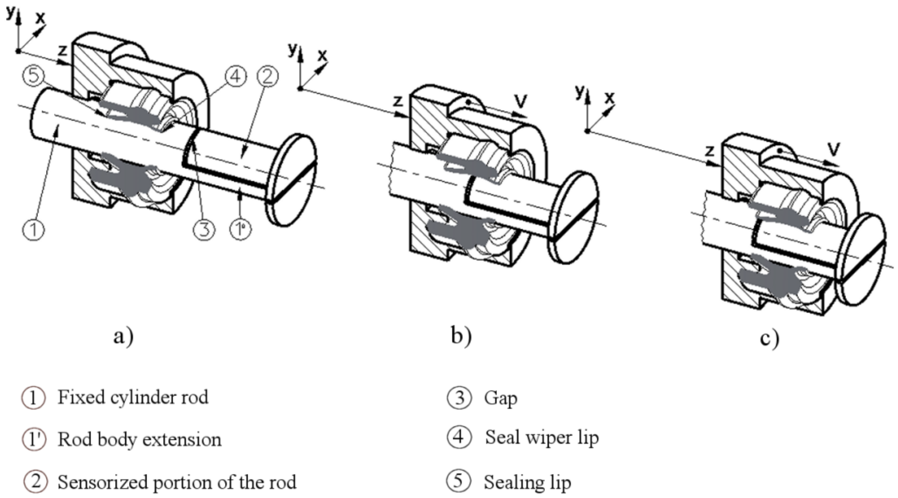

Figure 1: dynamic measurement of the seal/rod radial force.

Since the seal motion occurs at constant velocity V, the contact pressure distribution along the z coordinate can be obtained as a function of the force F time derivative, the rod diameter d and the seal velocity V:

$$p(z) = \frac{1}{d} \cdot \frac{dF}{dz} \cdot \frac{dt}{dt} = \frac{1}{d} \cdot \frac{dF}{dt} \cdot \frac{1}{V} \qquad (1)$$

Details on the analytical procedure can be found in [13].

This method of measurement reproduces actual working conditions without introducing any additional deformations of the seal; this characteristic represents a great advantage with respect to other experimental methods, as those making use of pressure sensitive films.

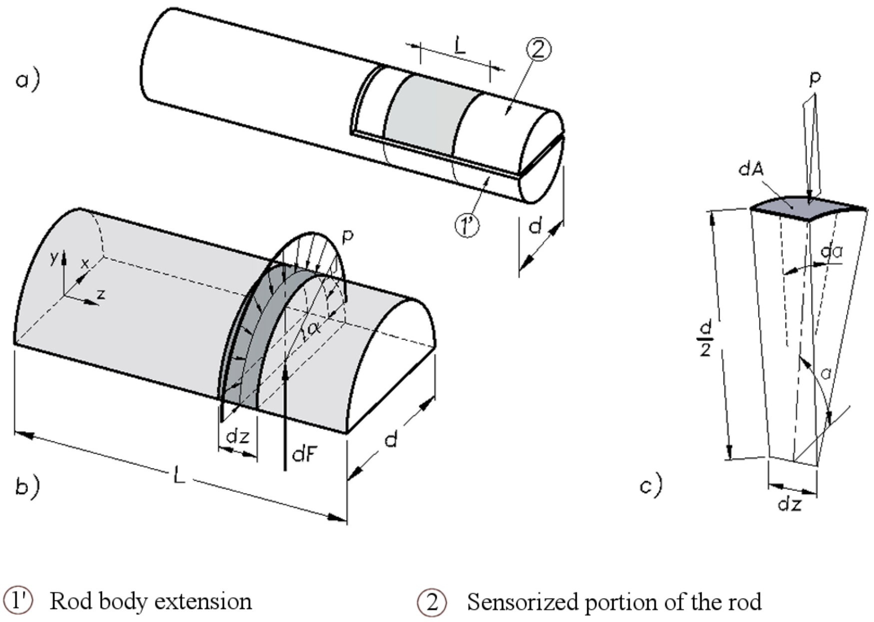

Figure 2: a) sensorized portion of the rod (the full contact area between the seal and the sensorized rod is in grey); b) axisymmetric contact pressure distribution; c) elementary portion of contact area.

3. Test bench

A test bench for the indirect measurement of contact pressure at the seal-rod interface was designed and manufactured, on the base of the solution proposed in [13].

A sketch of the measuring portion and a photo of the manufactured test bench are shown in Fig. 3 a) and b), respectively. In particular, it is possible to identify: the seal holder (1), which permits to pressurize the sealing lip of the seal; a pneumostatic bearing (2), used to obtain a smooth motion even at low velocity; the fixed parts (3) and (3’) of the measuring rod and the sensorized rod portion (4); the couple of load cells (5), which measure the radial force signal over time; precision screws (6), which were used to adjust the gap h_1 between the two portions of the rod correctly (see the detail in Fig.3a); the pair of pneumatic bearings (7) and the pneumatic pad (8), which permit to correctly align and centre the sensorized rod portion (4) with respect to the rod extension (3'). In particular, the pair of pneumatic bearings (7) has also the aim to balance the friction force exerted in z-direction and any torque produced about the x or the z axis; the pneumatic pad (8) balances any torque about the y axis. The choice of pneumatic guide bearings, namely low friction bearings, ensures that the seal radial force F exerted in y-direction is correctly measured by the load cells; any additional friction would misrepresent the measurement. More details on the test bench characteristics can be found in [13].

In this test bench, the generation of the measuring signal is obtained by displacing the sensorized portion of the rod in the negative y-direction. Since this displacement causes a modification of the shape and of the diameter of the rod, it has to minimized. To this aim, load cells FGP FN 3030 (accuracy +-0.1% F.S) with a full scale of 1000 N, definitely higher than that required, were chosen to ensure a high stiffness; thanks to the parallel assembling configuration, the equivalent stiffness of the measuring device is of $4.2 \cdot 10^7$ N/m. Both the load cells were calibrated out of the test bench. After having mounted them in the working configuration, a known weight was hanged on the sensorized portion of the rod, to further verify that the sum of the force measured by each cell was equal to the loading weight.

The radial force signal was recorded by a NI USB-6229 data acquisition system at a frequency of 1 kHz.

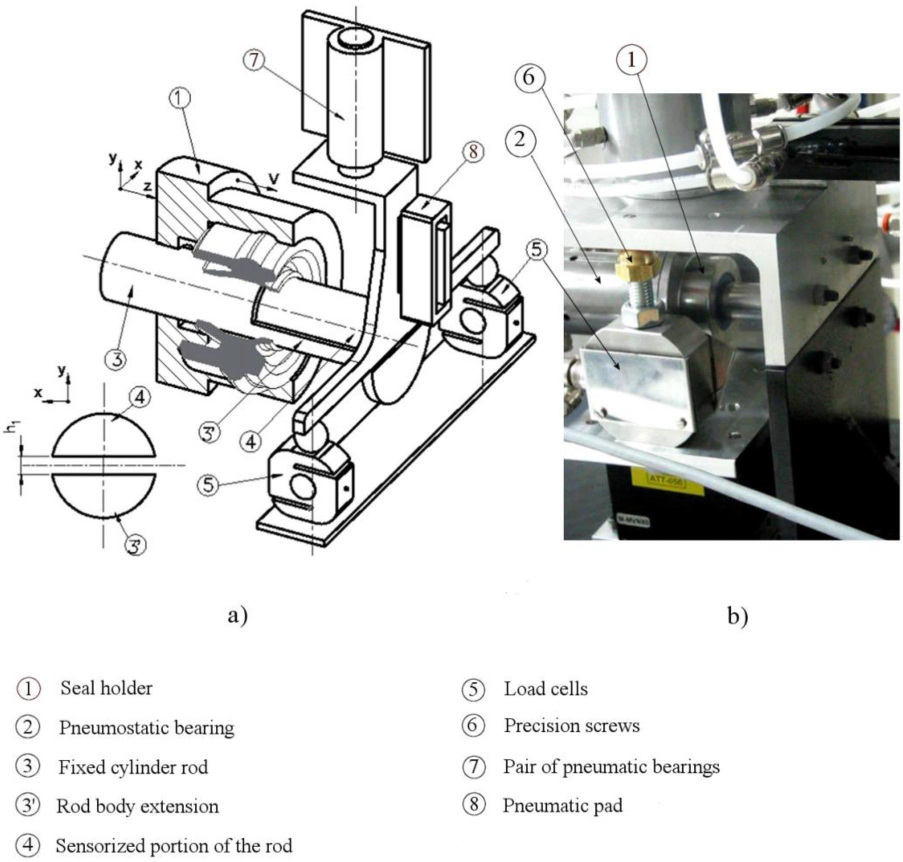

Figure 3: sketch a) and photo b) of the measuring portion of the test bench.

4. Test results

An elastomeric double lip seal for a pneumatic cylinder with a rod diameter d=20 mm was chosen to perform experiments. Nevertheless, the experimental test bench and the computational methodology described are general and can be applied to pneumatic rod seals with geometries and materials other than those described in this study.

Fig. 4 shows a cross-section of the seal, provided by a wiper lip (4) and a sealing lip (5). The main dimensions of the seal are: L= 10.7 mm; h= 5 mm; inner diameter of the sealing lip equal to 19.37 mm; inner diameter of the wiper lip equal to 19.73 mm. The seal is made from thermoplastic polyurethane (TPU), which can be considered incompressible, isotropic and hyperelastic; hardness ≈ 90 IRHD.

The rod has a diameter d =20 mm (machining tolerances: +0 -0.033mm), as for a commercial rod for pneumatic cylinders.

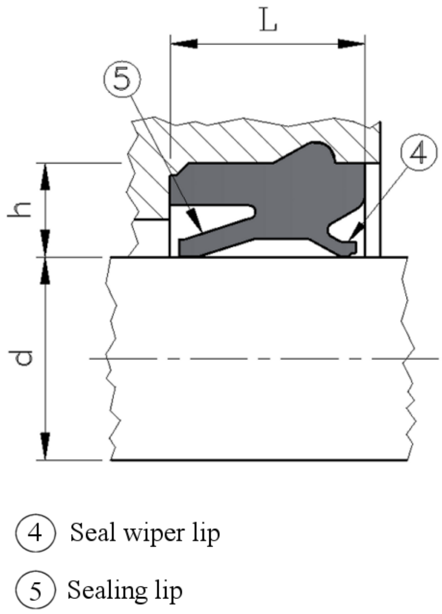

Figure 4: cross-section of the seal under test.

Tests were carried out driving the seal holder at a constant velocity of 1 mm/s. The four seal samples were tested after a short running-in phase, equal for all samples. The data recording was started manually during the stroke of the seal over the intact rod.

4.1 Dynamic seal/rod radial force

Fig. 5 shows four experimental curves, representing the measured seal/rod radial force over time, obtained on four samples of the lip seal under study. These preliminary tests were carried out without supplying the seal holder, denoted as (1) in Figure 3, by compressed air. Nevertheless, because of the vent of the pneumostatic bearing (2), a pressure load of 0.4 bar (gauge pressure) is applied on the seal. As shown in the magnified detail, a stair-step shape of the force curve over time ascribable to a stick-slip phenomenon was detected; nevertheless, these short-term fluctuations do not have any influence on the long-term trend of the radial force signal. The postprocessing analysis was applied to the curve shown in Fig. 6, which plots the average data gathered during the four experiments, preliminary aligned in time. A maximum standard deviation of 0.95 N was calculated.

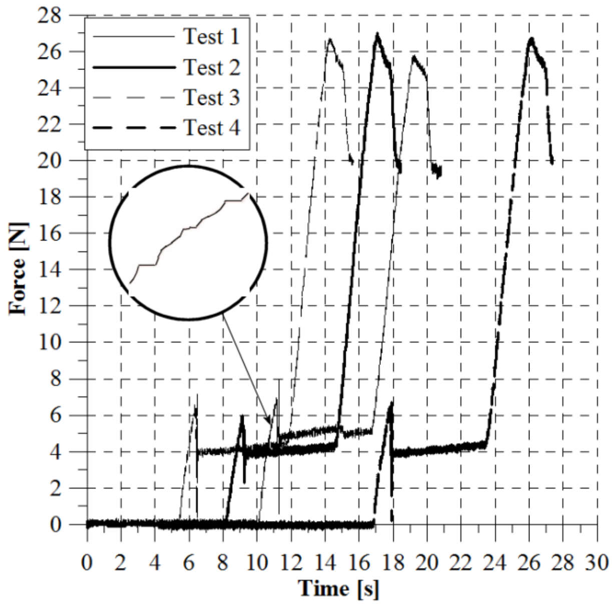

Figure 5: radial force trend over time and detail of the signal.

As shown, while the seal is sliding over the intact rod, the force detected by the sensor is equal to zero; then, when the wiper lip engages the sensorized portion of the rod, the radial force increases and subsequently remains almost stable. The pick of force detected at about 11 seconds could be explained considering that, even if the gap between the intact rod and its sensorized portion is very narrow, the rod wiper could partially intrude inside it. All over the crossing phase, the measured radial force could have been a little higher than that obtainable without any gap, because of the partial lip binding. At the end of the crossing phase, the wiper lip could have relaxed, retaking the correct position. For this reason, at the end of the crossing phase an abrupt drop of the measured force occurs. Later, a slight increase of force was detected, because the portion of the seal between the two lips is touching lightly the rod. When also the sealing lip passes over the sensorized rod tip, an abrupt increase of force is detected. Later, the force signal falls down instead of remaining stable at a constant value. It is due to the fact that, when the sealing lip passes over the gap provided between the rod sectors (gap (3) in Fig. 1), the compressed air which loads the seal is quickly exhausted.

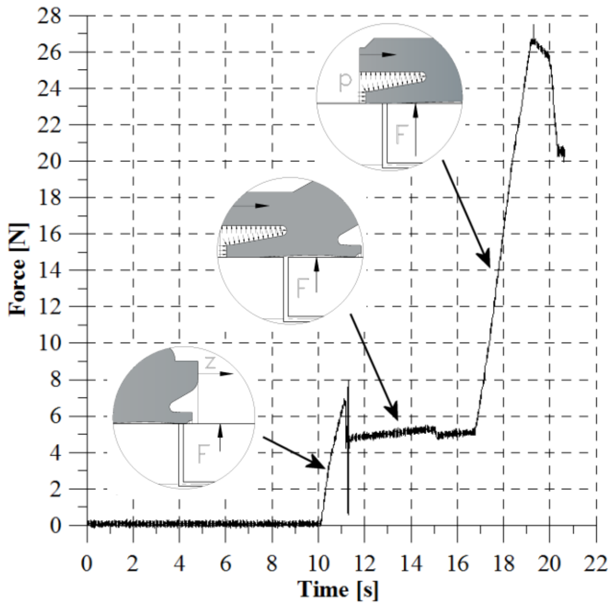

Figure 6: average experimental curve.

It can be noted that a certain noise and some spikes affect the signal given by the force sensor. Since, according to the methodology described in section 2, the time derivative of this signal is calculated to obtain the seal/rod contact pressure distribution, the radial force signal was firstly smoothed. A moving average filter, typically used to smooth out short-term fluctuations and highlight longer-term trends (low pass filter), was applied. Then the curve was splitted in three parts and a data fitting was performed using polynomial functions.The degree of the polynomial functions, which ensure the best fitting of each part of the experimental curve, was found by checking the coefficient of determination R^2. In particular, a fifth degree polynomial function was used for fitting the first and the third part with a R^2 of 0.997 and 0.999, respectively. A linear interpolation was used for the second part. There were some discontinuities among the fitted curves, involving few milliseconds, corresponding to the zones where the first derivative varied too much. Therefore, having calculated the first time derivative of each fitting function, the continuity among them was finally imposed. It must be remarked that the abrupt drop of the measured force occurring at about 11 s corresponds mathematically to a negative sign of the force first derivative for few milliseconds; correspondently, a negative contact pressure is calculated. Since a negative contact pressure is a nonsense, this part was removed.

Fig. 7 shows the three fitted curves, together with their first time derivative, connected one to the other.

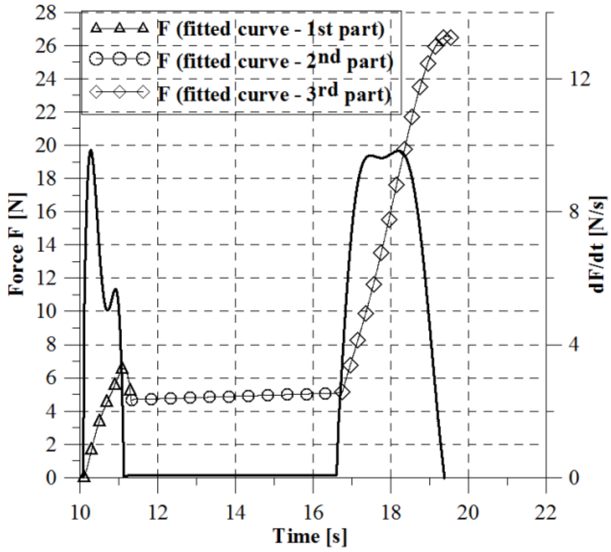

Figure 7: fitting of experimental data and their first time derivative dF/dt.

4.2 Contact pressure distribution

Fig. 8 shows the contact pressure distribution between the double lip seal and the rod, as obtained applying eqn. (3), i.e. dividing the time derivative signal by the product of the sliding velocity V and the rod diameter d. Since both V and d remain constant during the test, their product simply acts as a scale factor between the trend of force time derivative versus time and the trend of contact pressure distribution versus the z coordinate, i.e. along the seal/rod contact surface. The relationship between the time and the z coordinate is simply given by $z=V\cdot t$.

With reference to Figure 8, it must be remarked that Part I and Part III refer to the wiper lip/rod contact surface and to the sealing lip/rod contact surface, respectively. It can be noted that the radial force exerted by the sealing lip, higher than that exerted by the wiper lip because of the effect of the compressed air load, results in a contact pressure distribution characterized by the same level of that obtained on the wiper lip, but spread over a larger contact area.

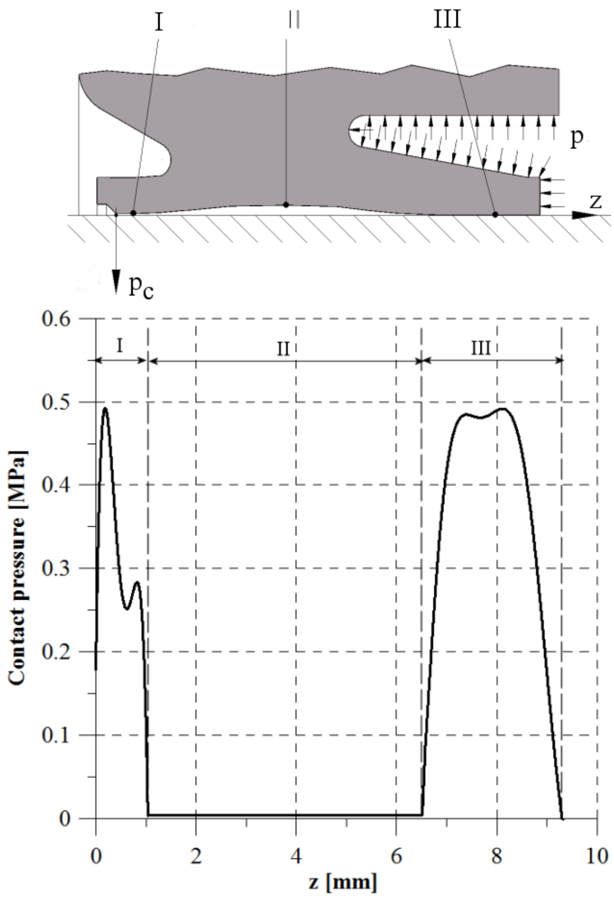

Figure 8: seal/rod contact pressure distribution.

A static numerical model of a TPU rod seal for pneumatic applications was presented by Hermann and Dabisch [3]. A direct comparison cannot be made because the modelling conditions are not completely specified in that paper and because results on contact pressure are presented in percentages; nevertheless, the contact area width and the shape of the contact pressure distribution shown in [3] at the sealing lip are well corresponding to those we found experimentally.

It is important to point out that the seal/rod friction conditions highly affect the contact pressure distribution: the relative sliding motion between the seal and the rod is taken into account in this paper.

5. Conclusions

Results of experimental tests aimed at the evaluation of contact pressure at the interface between an elastomeric rod seal for pneumatic cylinders and its metallic counterpart are presented. Tests were carried out using a specifically designed test bench able to detect the radial force exerted by the rod seal displaced at constant velocity on a sensorized portion of a cylinder rod over time. Pressure load can be applied on the seal to reproduce actual working conditions. No intrusive measuring device was interposed between the tribological couple, which is the main advantage of the presented method. On the other hand, the test bench assembling requires a special care and parts alignment must be done accurately. Certainly, the displacement of the sensorized portion of the rod needed to generate the radial force signal could affect the contact pressure measurement. Nevertheless, the choice of high stiffness load cells in a parallel assembly ensures a maximum displacement of less than 1 μm at the highest value of measured force. On the other hand, the possibility of a small intrusion of the wiper lip into the gap between the fixed cylinder rod and its sensorized portion could be a limit of the proposed test bench. Experimental results, which gave the radial force trend over time, were postprocessed to indirectly calculate contact pressure distribution. In particular, the radial force signal was filtered and splitted in parts, which were fitted by polynomial functions. In this way, the radial force signal could be differentiated and contact pressure distribution was computed as a function of the radial force time derivative, the seal velocity and the rod diameter. Preliminary results obtained on a type of rod lip seal for pneumatic actuators are presented.

The experimental test bench and the computational methodology described are general and can be applied to pneumatic rod seals with geometries and materials other than that described in this study. Future work would be addressed to the evaluation of the influence of different working conditions and to the development of a numerical model to compare results.

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