PARTICULARITIES OF RUBBER LIP SEALS USED FOR PNEUMATIC LINEAR DRIVES

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ABSTRACT

The paper presents some particular aspects of reciprocating rubber lip seals for pneumatic linear drives. This study is focused on the following tribological aspects: friction variation under linear velocity, determination of the rod temperature, kinetic and dynamic friction coefficients. It was presented the graphical evolution of rod velocity and acceleration and the inertia of mass forces that followed. This study ends with the presentation of materials used for reciprocating rod and rubber lip seal.

KEYWORDS: Reciprocating lip seal, pneumatic drives, polymeric materials, elastomeric materials.

1. INTRODUCTION

While most previous theoretical studies assume that full film lubrication exists between the seal lip and the shaft, recent simulations [1] showed that mixed lubrication generally occurs, in agreement with previous experimental evidence.

These studies, on seals with single [1] and double lips [1], also showed that the seal surface roughness plays an important role in determining whether or not a seal will leak. For a given seal design and operating conditions, there is a critical value of seal roughness below which the seal will not leak. That critical roughness is smaller for a seal with a double lip than for one with a single lip, indicating a less favorable sealing characteristic.

The secondary lip of a double lip seal is intended to act as a second line of defense in the case of a defect in the primary lip, as well as a bearing. It also allows for retention of lubricant in the inter-lips region in between periods of operation.

However, it has been found [1] that the secondary lip can interfere with the operation of the primary lip and reduce its sealing effectiveness, due to a very strong coupling between the actions of the two lips. An alternate means of obtaining a second line of defense is to use a tandem seal arrangement, as schematically shown in figure 1 [1]. The operation of such a seal is the subject of the present study.

The analysis of a single reciprocating rod seal of a reciprocating rod seal with a double lip, comprises an interconnecting analysis of steady state fluid mechanics, deformation mechanics, and contact mechanics analyses, with an iterative computational procedure. This analysis assumes that the stroke length is significantly larger than the seal width.
Figure 2 [1] shows the sealing zone of one of the seals. The seal surface is considered rough, while the rod surface is treated as smooth.

Fig. 2. Sealing zone.

The results of this study indicate that the typical tandem hydraulic rod seal operates with mixed lubrication, such as the previously studied single lip and double lip seals. Its behavior is very dependent on the seal surface roughness and is characterized by a critical roughness below which there is zero net leakage. This critical roughness is higher than that of a comparable double lip seal and almost as high as that of a comparable single lip seal. The friction force produced by the tandem seal on the rod is somewhat higher than that produced by the single lip seal but lower than that produced by the double lip seal. Also, here is a difference between the frictional force and velocity on instroke and outstroke, figure 3 [1], during outstroke, frictional force is much higher.

Fig. 3. Friction force versus roughness, single lip seal.

According to the above presented aspects, the following characteristics have been found to promote zero or reduced leakage:

- small seal surface roughness
- small lubricating film thickness
- thicker film during instroke than during outstroke
- cavitation in film during outstroke
- structural decoupling of multiple lips

The basic assembly of a cylindrical rectangular seal for a reciprocating piston rod of a hydraulic actuator is shown in figure 4 [2].

The average contact pressure is found to be very close to the sealed pressure, a phenomenon which is explained by the incompressibility of the seal rubber material and the boundary constraints of the assembly.

Fig. 4. Basic assembly of the seal.

A complete set of parameters are taken into account; these include the temperature-dependent mechanical properties of all bodies involved, the basic geometry of the actuator assembly, thermoelastic deflections of all bodies, the corner radius of the seal, surface roughness effects, using both a linear and a non-linear stress-strain model for the seal.

2. RECIPROCATING SEALS FOR PNEUMATIC DRIVES

Historically it was common to use light duty hydraulic cylinder seals for pneumatic applications [7]. It is now normal practice to use seals specifically designed for pneumatic applications, for several reasons:

- Pneumatic systems now have to work with minimal lubrication or totally dry which places very different criteria on seal selection.
- Pressures are much lower, the maximum for pneumatic normally being 10 bar, which means that much lighter duty seal designs can be used.
- The lubricants used can be very different to those in contact with hydraulic seals. It is typical to use grease during installation, with the type of grease being dependent on the application.
- The seal material must be resistant to the oxidizing effects of compressed air which may also be at higher temperatures than those experienced by hydraulic systems.
- A small leakage of compressed air is not a major problem, and may be preferable to high friction that will cause premature wear.
- The lower pressure with consequent lower stresses and high manufacturing volumes provide the potential for alternative materials of construction and manufacturing processes.
- The seal needs to retain the lubricant in the contact zone rather than exclude it. This is in direct contrast to the requirements for an oil hydraulic seal.
For these reasons, pneumatic seals have now evolved as a very distinct class of seals with very different characteristics to hydraulic cylinder seals. The lubricant available for a seal in pneumatic systems [7] will consist of:

- Small amounts of carryover, from the system compressor.
- Minute amounts from a system lubricator.
- Grease pre-applied to the seals when assembled.

The increasing restrictions on the exhaust of any form of oil mist aerosol mean that in most modern systems the only available lubricant is that applied to the seals during assembly. This is especially so in applications concerned with food, pharmaceuticals and other products where cleanliness is paramount. In such applications, the air is rigorously dried and cleaned before it is used and exhausted into the atmosphere. It is therefore important that any lubricant applied during assembly is retained in the area of the seal and encouraged to lubricate the seal lip and other potential wearing parts of the seal.

The basic contact zone design for pneumatic seals is therefore quite different to a hydraulic seal, [4]. The seals are designed to have a much lower contact force and so the lip design will be a more flexible and reduced section of material. In addition to the lighter loading, the lip design is intended to encourage any lubricant present under the seal. The pressure side of the seal will have either a radius or a shallow chamfer to encourage entrainment. An example is shown in figure 5 [3].

These are typically a light construction of U-seal. As the seal does not have to withstand a very high pressure loading, it is also quite common to integrate the wiper seal with the pressure seal. In this case it is important to ensure that the wiper does not exclude any lubricant on the return stroke. The seal and the wiper will therefore both have a chamfered or radius profile to ensure retention of lubricant under the seal, figure 6 [3].

The relatively light construction of pneumatic cylinders means that in most cases a one piece seal and wiper can be used. Separate wipers are also available. In both cases, these are of different design and construction to hydraulic excluders. They are designed for low friction and also with a rounded or chamfered profile to encourage return of any lubricant. They will not provide the same degree of exclusion performance that may be expected from a hydraulic cylinder excluder, but there is likely to be a very low level of compressed air leakage during operation that will assist with removal of fine contaminant from the seal contact zone.

2.1. Tribological Aspects

The sliding of two solid bodies relative to one another is a linear unbalanced process where the kinetic energy of the uniform motion is transferred into energy of irregular microscopic motion, i.e. into heat. This dissipative process leads to friction, called dry friction, which differs from viscous friction. Since more than 200 years the phenomenological laws of dry friction (Coulomb’s laws) are well-known and are well established in applied physics [6]. They state that the friction force is given by a material parameter (friction coefficient) depending on the normal force. The coefficient of static friction (i.e., the force necessary to start sliding) is larger or equal to the coefficient of the kinetic friction (i.e., the force necessary for sliding at a constant velocity).

One cannot expect that Coulomb’s law can be derived as rigorously as the laws of viscous friction. The reason is that the latter one can be deduced from the well-established theory of near-equilibrium thermodynamics whereas dry friction works mostly far from thermal equilibrium. In fact, Coulomb’s laws oversimplify a very complex behavior which involves elasto-mechanical, plastic, and chemical processes operating on different length and time scales (for a general overview of the interdisciplinary aspects of dry friction [6]). Experimental deviations of Coulomb’s law are often found. For example, the kinetic friction depends on the sliding velocity. A common pattern is that it first decreases, then goes through a minimum and finally increases [6]. In the case of a thin lubrication layer one often finds the contrary: The kinetic friction first increases, then goes through a maximum and finally decreases [6].

In the last ten years, the interest in dry friction has been aroused again due to the possibility of
reproducible friction experiments on the mesoscopic and nanoscopic scale (nanotribology).

Typical investigations on static friction report on experimental measurements under different surface conditions and, in some cases, propose friction models, which are usually applicable to a specific interface. For example, Hwang and Zum Gahr evaluated the effect of surface roughness and lubrication on the transitional behavior from static to kinetic friction, and Etsion et al. [6] performed an experimental investigation using a sphere on flat configuration and measured the decrease of the static friction coefficient with increasing normal loads.

A typical friction transition in contacting solid bodies under (non-lubricated) dry conditions and constant normal load is depicted as static friction coefficient being higher than kinetic friction coefficient, as shown schematically in figure 7 [6]. The tribological tests are sliding friction tests between a steel ball and a disk of one of the studied rubbers, under dry conditions.

![Fig. 7. Schematic of a typical friction coefficient, showing static and kinetic friction.](image)

As shown in figure 8 [6], cyclic tests are carried out by rotating of the plane under the sliding ball loaded with a constant load of 5.5 N. The tangential load and thus the friction coefficient are measured by strain gauges. Indeed, during friction experiments, parameters like the sliding speed and the test temperature can be controlled. A thermocouple placed on the top of the ball records the temperature increase due to the heat dissipated at the contact interface versus the time or the number of revolutions.

![Fig. 8. Cyclic test ball-plane rig a) schematic ball-plane contact and thermocouple localization; b) ball-plane tribometer.](image)

However, some ageing experiments have been made on both rubbers during 7 days at 100ºC either with air. In term of hardness, tensile strengths or young moduli, the behaviour differences between two types of rubbers are not strongly affected by the ageing process.

The most critical aspect of the contact between the lip and the matting surface consists in dry conditions, therefore most of experiments were carried out under dry conditions, in order to focus on the relationship between the viscoelasticity and the tribological behaviours of the two types of rubbers.

The wear difference between rubbers might be explained by a competition between wear and thermo-oxidation processes. In one case, this competition leads to damage; while in the other case, a thin protecting surface against wear is formed. For the first rubber, the fact that internal friction is lower compared to second rubber leads mainly to the wear process. Grosch showed that the variations of the friction coefficient are similar, and that the abrasivity is inversely proportional to the friction coefficient [6].

To improve rubber surface properties, viscoelastic properties and not especially elastic properties have to be taken into account. This means that to obtain a low wear rate, it would be better to be very close to the main relaxation temperature, which is also frequency dependent.

Comparing surfaces coded smooth I and rough II in figure 9, interfaces at the same value of roughness clearly shows that the smoother interface has a much higher friction coefficient (around 1.0) compared to the rougher interface, which has a value of around 0.1 [6]. Also, in both cases, the friction coefficient is about constant, within the range of external loads shown, except at very small loads, suggesting that within a limited range of external load, the constant Coulomb friction assumption may be valid, particularly for low adhesion energy values.

![Fig. 9. Modeling of the static friction coefficient using KE elastic-plastic model.](image)

The above static friction model results assume that the contact interface is under dry conditions, independent of dwell time and temperature, and independent of the interface and tribosystem
dynamics, such as the rate of application of the velocity or acceleration in initiating sliding. As in practical applications, some of these effects cannot be avoided; some of them will be considered in the static friction experiments reported in this paper. Specifically, experiments in the presence of a trace of lubricant, wear debris, displacement rate dependence, and dwell time will be reported.

2.2. Dynamic Aspects

In case of dry friction, sliding velocity at the ends of the race is minimal while the acceleration value is maximum. Sliding velocity reaches the maximum value halfway down when the acceleration is minimal. Figure 10 presents the velocity and acceleration variation, along stroke length for a linear drive's rod.

![Fig.10. Sliding velocity and acceleration variation in reciprocating sliding motion.](image)

The inertial forces that arise, due to motion's frequency, are variable; therefore the friction forces' regime is also variable, depending on sliding velocity.

2.3. MATERIALS

The seals in pneumatic cylinders [7] are expected to work with minimal lubrication and low friction while achieving a long life, for example 2 x 10^6 cycles. The materials are therefore usually specifically formulated for pneumatic applications. Nitrile elastomers are still used regularly but there is also increasing use of polyurethane and polyester material. Pneumatic systems are also often used in temperature and hazardous areas where it may not be considered safe to use hydraulic systems. It may then be necessary to use a higher temperature resistant material such as fluorocarbon [3].

Elastomers are very widely used and potential applications cover the full range of static and dynamic seals. They are also a key component of other seal types. For instance the majority of mechanical seals contain several elastomer seals and these are often one of the limiting components in the fluid resistance and temperature range of the seal. There is quite often a severe lack of understanding of elastomer materials even among those who use them regularly.

Large deflections only require low stress: Elastomers have a very low modulus of elasticity (E) of typically 5 to 20MPa compared to 50 to 200 GPa for typical engineering metallic materials. This is combined with an elongation to break, which is normally well in excess of 100%. It is therefore possible to use relatively large strains both during installation and to provide seal interference. Elastomer seals will typically be designed to have an installed strain between 10% and 30%.

This large value of elastic strain will be achieved with a relatively low stress. Strains of 50–100% may be used during installation without adverse effects on most elastomer materials. (Note that some especially hard elastomers may have an elongation to break of less than 100%.) The seals can accommodate a relatively wide range of tolerances and misalignment without high contact stresses and will conform to the mating groove surface texture to provide a high integrity seal without plastic flow of either the seal or the counter-face.

It is a resilient material with low hysteresis: when used within the appropriate temperature range for the material grade elastomer, materials provide a high degree of resilience with low hysteresis. This provides the ability to respond rapidly to pressure changes, vibration plus conditions such as runout in dynamic applications.

They have relatively low creep: the tensile strength of the materials is relatively low, compared to most plastics and obviously metals, but when
straining within the range of material properties they do not suffer excessive creep. This is important for maintaining sealing stress for long periods of time and through high temperature cycles. This can be a particular advantage of elastomers when compared with plastic seals. Figure 11 shows that plastics will generally creep when strained beyond 5…30%, depending on the type of plastic and filler content [3].

Elastomers have a high Poisson’s ratio, very close to 0.5: This means that the materials are almost incompressible. The high Poisson’s ratio combined with the very low elastic and shear modulus combine to make a material that is both easily deformed and incompressible. The inherent resilience of the material ensures that when the distorting force, such as pressure or groove constraint, is removed, the seal will return to the original geometry.

In practice the degree of resilience and ability to restore completely to the original geometry will be dependent on the individual characteristics of the material. However, there are also some disadvantages associated with these materials that introduce design constraints and limit the applications to which they may be applied.

A large group of compounds comprise the silicone elastomers. The most common is Silicone MVQ. The main attribute of silicone elastomers is the wide temperature range. They can be used from below -50°C to in excess of 200°C. The major problem is the relatively poor tensile strength and tear resistance which in turn makes them susceptible to wear and also more prone to damage on assembly. Although the physical properties are relatively poor at room temperature compared to other elastomers used for seals, they have the benefit that the properties are less affected by temperature. Hence they retain some viable strength at high temperature and remain flexible to very low temperatures. They are generally inert with respect to the human body which makes the material popular for food and medical applications. Silicones are also widely used as sealants, also known as liquid silicone rubbers (LSR). These are either a single part, incorporating the curing agent, or a two part, which is mixed immediately prior to application. The curing process is triggered by atmospheric moisture and occurs at room temperature (hence these compounds are also known as RTV) so some introduction of moisture to the area may be necessary to achieve a sensible cure rate in conditions of low humidity.

Major applications of silicone are medical, food, automotive and aerospace. Care is often required due to the limited mechanical properties which can be further reduced by a higher swell than some other elastomers. The poor resistance to steam limits the food applications if high temperature sterilization is required. Silicone has limited use as a dynamic seal but is used for some automotive rotary shaft lip seals because of the superior low temperature flexibility.

5. CONCLUSIONS

Taking into account the above presented aspects, the following conclusions can be drawn:

- Reciprocating seals of pneumatic linear drives operates with some special particularities, the most important being the dry friction regime.
- The reciprocating motion leads to variable velocities and alternating the static friction with dynamic friction.
- In order to reduce the inertia forces, non-metallic rods are used, polymeric materials being preferred.
- Polymeric material rods present a particular thermal behaviour during their functioning in dry friction conditions.
- Elastomeric materials are sensitive to high temperatures, which can lead to changes in the stiffness of the lip.
- In pneumatic drives case, the leakage is not as dangerous as in other fluids case, having also a cooling effect on the seal system.

REFERENCES


Fig. 11. Stress/strain range for elastomers, plastics and metals