VDMA
Fluid Power Association

22nd ISC

International Sealing Conference

Stuttgart, Germany
October 01 - 02, 2024

Sealing Technology –
Challenges accepted!

UNI STUTTGART

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ISBN 978-3-8163-0768-6

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Pneumatic Seals: A Review of Experimental Measurement and Theoretical Modeling of Sealing Friction

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Research on friction behavior in pneumatic sealing contacts has been ongoing since the late twentieth century, predominantly through experimental characterization of friction forces for different relative velocities and operating pressures. Simulative modeling of these contacts has utilized both lumped and distributed parameter models. This paper presents an overview of the most important findings in research and discusses consensus and discrepancies among various studies in the field of reciprocating pneumatic sealing friction. It was found that there are a lot of contradictions in the experimental results especially about the influence of the relative velocity on the friction force, which can possibly be attributed to different grease film heights.

1 Introduction

The correct functioning of seals in pneumatic components is essential for safe operation. Pneumatic seals prevent the leakage of compressed air or the ingress of unwanted particles into the pneumatic system. Leakage in particular can lead to a sharp drop in the efficiency of pneumatic systems or undesired behavior. Due to the low viscosity of air, even the smallest gaps lead to high leakages. O-rings and other elastomer components or even adhesives are used for static sealing. Dynamic seals are usually reciprocating and are used as piston or rod seals in cylinders or pneumatic valves [1].

While reciprocating seals are also used in hydraulic systems, the requirements for reciprocating seals in pneumatics differ significantly. In addition to the lower operating pressures, there is a significant difference in the type of lubrication as considerably lower quantities of lubricant are used in pneumatics compared to hydraulics. In pneumatic systems, there is often an initial lubrication that is not renewed during the service life of the component. This results not only in the requirement for better dry-running behavior of pneumatic seals, but also in particular the requirement for a lower wiping effect of the seal. As a result, the geometries of pneumatic seals are less sharp-edged than hydraulic seals in the contact area and have a flatter design [2].

A typical reciprocating pneumatic sealing system consists of the actual seal, an installation groove, the counter surface and the lubricating grease in the sealing contact. During operation, there is a relative movement between the counter surface and the installation groove. Typically, the seal is made of soft material like an elastomer. The strong focus on soft seals in industrial applications is also reflected in the available literature. In recent years, far more publications have appeared that deal with the behavior of conventional soft-sealed pneumatic contacts than with alternative concepts like using air bearings [3–5], air-lubricated seals [6] or ultrasonic friction reduction [7]. Therefore, these special concepts are not discussed in this publication.

Studies on the friction behavior of pneumatic sealing contacts have been conducted since the end of the twentieth century. A large part of this work is concerned with the purely experimental characterization and measurement of friction forces. Usually, the main focus of the research lies on influence of the relative velocity and the operating pressure and less frequently also on the manufacturing tolerances or the materials. There are also some studies that deal with the simulative modeling of pneumatic sealing contacts. Both distributed and concentrated parametric simulation models have been used for that.

The aim of this paper is to give an overview of the state of research of experiments and theory of the friction in reciprocating pneumatic sealing contacts. For that, the most important results of selected literature sources from the past 30 years are briefly summarized compared. Both consistent conclusions and contradictions between the results of different authors are discussed.

2 Experimental characterization of pneumatic sealing contacts

The aim of most experimental investigations is to measure the friction force in sealing contacts. The sealing systems under consideration are usually cylinders or spool valves. Although the two components fulfil different purposes and consist of different materials, the sealing systems are nevertheless comparable in terms of operating pressures and ambient media.

A common comparison criterion for tribological systems is the coefficient of friction (COF) μ, i.e. the ratio of friction force to normal force. However, the COF is not provided in most publications on pneumatic seals, as the normal force in the sealing contact is usually unknown. The normal force in most tests is caused by compression of the sealing material due to an interference fit and/or the applied operating pressures. Since therefore the COF cannot be calculated in most publications, it cannot be used as a criterion for comparing the literature sources. As the pneumatic systems differ in type and also in size and number of seals, a comparison of the measured values for the friction forces also appears to be difficult to compare. Instead, the focus in this section is placed on the qualitative influence of individual parameters such as applied pressure difference, lubricating grease and velocity. The qualitative parameter influences measured in the literature are compared below.

2.1 Influence of pressure

Many sources do not clearly state which pressures are acting on the sides of the seal. Instead, usually only the operating pressure of the pneumatic system is given, so that an exact quantitative comparison is often not possible.

When measuring the friction force of the piston seal of a pneumatic cylinder, Qian et al [8] observed that for lip seals mounted in pairs on the piston, only the total pressure in both chambers, but not the differential pressure between the chambers, influences the friction force of the seal pair. However, if O-rings are used as piston seals instead of lip seals, the friction depends on both the total pressure and the differential pressure.

Azzi et al [9] measured the friction of piston and rod seals of pneumatic cylinders at different velocities and pressures for an O-ring, a U-ring and an X-ring piston seal. For all three seals, the friction force increases with increasing operating pressure. However, the friction force of the U-ring seal increases more steeply than that of the other two seals investigated. Whether this behavior can be attributed to the investigated geometries cannot be concluded from the investigations, as all three investigated geometries were manufactured from different materials.

The investigations by Papatheodorou [10] also show an increase in friction force with increasing operating pressure. The experiments by Tadic et al [11] investigated how the friction force changes when the chamber on the piston rod side is pressurized with ambient pressure while there is a pressure below ambient in the chamber on the piston side. Likewise, Belforte et al [12] found that an increased pressure difference at the seal also leads to an increased friction force.

Most of the sources mentioned thus show that the seal friction increases with increasing differential pressure across the seal.

2.2 Grease

Papatheodorou [10] determined the friction force of selected rod and piston seals for 15 different greases in new condition and after an unspecified “short endurance test”. The friction force for certain operating points and sealing materials differed between the grease with the highest and the lowest friction force by more than a factor two.

Further studies on the influence of the lubricating grease were carried out by Heipl [13], who selected three different greases with different thickeners and base oils. The friction when using the greases was compared for an O-ring seal and a lip seal as well as two accelerations. When using the softer grease, the friction force for an acceleration of $40\ \mathrm{\mu m/s^2}$ was up to $200\ \%$ higher than when using the other greases. For a higher acceleration of $100\ \mathrm{mm/s^2}$ and higher velocities, the friction force of the harder greases remained largely constant. The friction force of the softer grease, on the other hand, dropped significantly at high accelerations and velocities and approached the friction force of the other greases. At higher acceleration, however, the friction force for the lip ring differed significantly for all three greases tested. In particular, the two harder greases showed an increase in friction force with increasing velocity by a factor of more than 1.5. The friction force when using the softer grease was up to a factor 3 below the friction force of the other two greases from a velocity of around $30\ \mathrm{mm}/s$ and higher.

The results discussed by Heipl [13] thus show that the friction that occurs when using a lubricating grease depends massively on acceleration, velocity and seal geometry. The influence of different greases on the friction force can therefore only be compared with precise knowledge of all relevant boundary conditions. In addition, the comparison of greases is also hindered by the fact that it was not investigated whether and under which operating conditions how much lubricant was removed. Therefore, it cannot be concluded with absolute certainty whether the observed phenomena are solely related to the behavior of the lubricant in sealing contact. It is also possible that the lubricant film height was set to different levels in different test arrangements. A comprehensive comparison of the operating conditions would also have to include the lubricant film height.

2.3 Grease film height

Overall, there are only a few publications that systematically take into account the influence of film height on friction force in pneumatic sealing contacts. This is probably due to the fact that the film height can only be measured or specifically adjusted with great effort. One of the few publications that deals with the measurement of film height is by Pichon et al [14]. They determined the lubricating film height for an O-ring of a pneumatic valve in its initial state and after friction force measurements had been carried out. To determine the film height, they weighed the running surface with a precision balance. For one test, the lubricating film height was measured after each stroke. Already within the first five strokes, there was a significant decrease in the lubricant film height from about $1.2\ \mu m$ to approx. $0.2\ \mu m$.

In addition, Pichon et al [14] found in their experiments that increasing the initial lubricant film height at the start of the measurements lead to a higher resulting film height after the measurements. When increasing the initial film height beyond values of $0.6\ \mu m$ and higher, only negligible changes in the resulting film height occured after the friction force measurements. With sufficiently high initial lubrication, the resulting lubricating film was up to $0.4\ \mu m$ for tests without pressure load on the seal. In tests with a pressure load of $0.8\ MPa$ on the seal, film heights of up to $0.2\ \mu m$ were measured.

When comparing the friction force for the unlubricated and lubricated condition, Pichon et al [14] found that the friction force in the unlubricated condition is up to a factor of 10 higher than in the lubricated condition. These results are consistent with the investigations by Heipl [13], in which the friction force can also be reduced by a factor of 5 to 20 by lubrication.

Overall, a strong influence of the lubricant film height on the friction force can therefore be observed. With decreasing lubricant film height above a certain limit value, the measured friction force drops only slightly. There is no source that specifies both the roughness of the contact partners and the measured film heights.

2.4 Velocity

In contrast to the influence of pressure already discussed, where the experimental observations agree that an increase in the applied pressure causes an increase in the friction force, there are qualitatively very different measurement results for the influence of velocity.

For example, Raparelli et al [15] observed that friction increases with velocity. In a simple tribometer test, the coefficient of friction was first examined for three different lubrication conditions (dry, boundary lubrication and fluid lubrication) for velocities in a range of 20 to $250\ \mathrm{mm}/s$. A degressive increase in the coefficient of friction with velocity was found for all three lubrication states. Nevertheless, these tests clearly show that there is an increase in friction force with increasing velocity for the NBR samples tested. This was also determined by Papatheodorou [10] for 14 of the 15 lubricating greases investigated. There, the friction force also increased with the velocity in the investigated velocity range between 5 and $100\ \mathrm{mm}/s$. For one of the greases tested, however, the friction force decreased with increasing velocity. These two observations were made both in the new state and after the aforementioned unspecified “short endurance test”. The grease for which the friction force decreased with increasing velocity had the lowest base oil viscosity.

This contrasts with the results of Pichon et al [14]. They only considered the two velocities $10\ \mathrm{mm}/s$ and $100\ \mathrm{mm}/s$, but found no significant influence of velocity on the friction force.

Another deviating observation comes from Tadic et al [11], who found that the friction force initially increases with an increase in velocity from 0 to $9.3\ \mathrm{mm}/s$. For higher velocities up to the investigated maximum velocity of $236\ \mathrm{mm}/s$, however, the friction force decreases continuously.

In addition, the work of Belforte et al [12] should be mentioned, in whose investigations classical Stribeck behavior was observed. An initial decrease in friction for velocities up to $100\ \mathrm{mm}/s$ is followed by an increase in friction, which persists up to the maximum velocity of $600\ \mathrm{mm}/s$ investigated. This corresponds qualitatively with the results of Nepp and Kröger [16], who also determined Stribeck behavior for two rod seals made of FKM or NBR with a transition from mixed to fluid friction (lift-off velocity) at about $10\ \mathrm{mm}/s$. The investigations by Heipl [13] also show Stribeck behavior with a degressive increase in friction after the lift-off speed. The lift-off was typically at velocities of around $5\ \mathrm{mm}/s$ in the unpressurized state. For one of the seal geometries examined, the lift-off velocity in the unpressurized state was up to $30\ \mathrm{mm}/s$. At an operating pressure of $6\ \mathrm{bar}$, the lift-off shifted to higher velocities. Up to the maximum investigated velocity of $100\ \mathrm{mm}/s$, no clear lift-off could be identified for one of the seals investigated. The lift-off velocity therefore depends heavily on the seal geometry used and the operating pressure present. The gradient of the friction after lift-off also depends heavily on the operating pressure and seal geometry. The experimental results from Wangenheim [17] also show Stribeck behavior for the friction force.

Consequently, it can be stated that no clear tendency regarding the influence of velocity on friction can be identified from the experimental observations. Assuming that all experiments were carried out and recorded correctly, it can therefore be assumed that complex interactions between the parameters of material, geometry or lubricant are responsible for the velocity dependence of the friction force, which cannot be clearly classified according to the current state of the art.

Further work by Belforte et al [18] provides an indication of a relevant influencing factor on the velocity dependency. In the studies mentioned, it was observed that the friction force depends strongly on whether lubricated or dry conditions are present. On the one hand, this affects the amount of friction force, which is reduced by up to a quarter, and on the other hand the velocity dependency. The friction forces of the investigated seals in dry running increased by more than $100\ \%$ with an increase in velocity from 0 to $100\ \mathrm{mm/s}$, whereas the increase in friction force at the same velocity range for seals with lubrication is less than $20\ \%$.

The results presented in [18] therefore suggest that the influence of the velocity is not only quantitatively but also qualitatively significantly influenced by the lubrication condition. A comparison of the various literature data is therefore not very meaningful without precise knowledge of the lubrication condition. However, very few sources provide precise information on the lubricant used. In addition, most sources did not record the thickness of the lubricant film applied. Furthermore, no information was given as to how it was ensured that an even and comparable lubricant film was applied for all tests. The comparability of different test series and thus of different sources, especially from different authors, is therefore severely limited. However, the sparse documentation of the lubrication conditions provides a further explanation for the different velocity dependence of the friction force. If different lubrication conditions were present in different publications, this provides an explanation for the deviations in the observed behavior.

2.5 Other investigations

For the sake of completeness, studies investigating other effects than the those discussed in the sections above shall also be mentioned here. For reasons of space, however, they will not be discussed in detail here. These other studies include the investigation of geometric tolerances [19], the development of new sealing geometries [12, 18, 20, 21] and the measurement of the contact pressure distribution in the sealing contact by a film sensor [22–24] or a force sensor [25]. Investigations of causes of failure were conducted by Chen et al [26].

2.6 Conclusion on the experimental characterization

From the comparison of the studies presented, it is clear that there is not yet a comprehensive understanding of the mechanisms of pneumatic seal friction. There is a broad consensus that the friction force increases when the pressure in the sealing contact is increased, no matter whether the pressure increase is caused by higher pressures or tighter fits. In contrast, no clear statement can be derived from the experiments conducted regarding the influence of velocity. However, the investigations carried out by Belforte et al [18] suggest that the influence of velocity depends massively on the lubricant film height.

The influence of the surface structure of the seal and counter surface has not yet been systematically investigated. Furthermore, although the friction forces were measured with different lubricants, the lubricants were not comprehensively characterized in terms of their material properties. Consequently, it is not possible to deduce from the results which lubricant properties influence the friction behavior of pneumatic sealing contacts.

The research results presented show that most investigations carried out in the past have so far only dealt with the measurement of the friction force in steady states. A characterization of transient phenomena was presented in Bauer et al [27] and for the measurement of breakaway friction, as for example by Pham and Twiefel [7].

3 Theoretical modelling of pneumatic sealing contacts

After the experimental studies on pneumatic sealing friction were discussed in the previous section, a brief overview of all the methods used to date for modeling pneumatic sealing contacts is provided here. The section is divided into the groups of lumped and distributed parameter simulation models.

3.1 Lumped parameter models

All simulation and calculation models that describe the friction in pneumatic components exclusively with the aid of algebraic equations and ordinary differential equations are to be subsumed under lumped parameter modeling. These models offer the advantage of a comparatively short calculation time, but often use empirical parameters that cannot be determined directly from geometric and material properties. The calculation of friction is often only one part of a larger simulation model that describes an entire component or system, for example. It is possible to model the friction of each individual sealing contact separately, as well as to describe the combined friction force of several friction contacts of a component simultaneously.

One of the most frequently used models in literature is the LuGre model by Canudas de Wit et al [28] and Olsson [29], which models friction using elastic bristles in order to determine the friction force not only in the steady state but also before the start of steady-state sliding ("presliding displacement"). Accordingly, the friction force is defined as a function of the deformation of the bristles and the relative velocity. The parameters of the model cannot be determined directly from geometry, material or lubricant properties, but are usually fitted with the help of measured friction forces. The parameterization of the LuGre model for a pneumatic cylinder was investigated by Carneiro and de Almeida [30], among others, who compared two methods for determining the static parameters for the friction force of a pneumatic cylinder.

However, due to the abstracted modelling of the contact by bristles, the LuGre model is not or only to a very limited extent possible to optimize or redesign sealing systems. As the model is more descriptive than explanatory, it is suitable for describing the behavior of existing friction contacts as part of a larger system model. For example, a modified version of the LuGre model by Valdiero et al [31] was used to model a pneumatic servo cylinder.

Another example of the use of a lumped parameter model for a pneumatic servo cylinder comes from Soleymani et al [32]. They used a modified version of the generalized Maxwell slip model to model the friction of a pneumatic servo cylinder. They used this modeling for the position control of the servo cylinder.

Mazza and Belforte [33] took a different approach to the friction models described so far. They developed a friction model for pneumatic lip seals that is not based on the abstracted description of the contact, but instead describes an abstracted form of the macroscopic geometry. For this purpose, the sealing lip is modeled as a rigid body supported by a torsion spring, which is pressed onto the seal mating surface by the applied pneumatic pressure. In this case, the friction in the contact is modeled by a constant coefficient of friction. Unlike the friction models presented so far, this model makes it possible in principle to estimate the friction force of modified or newly developed lip seals or to investigate the influence of the operating pressure.

In contrast to the aforementioned lumped parameter approaches, the approach used by Wangenheim [17] to model pneumatic sealing friction is based on the surface topography. The approach is based on the hysteresis friction model proposed by Lindner [34], where the coefficient of friction is calculated using a spring damper element which slides over a line scan of the rigid counter surface. With his calculations, Wangenheim was able to show that less than 1 % of the dissipated energy dissipated due to friction in a pneumatic rod seal goes into the seal itself, which means that the temperature change of the seal can be neglected for friction modeling of typical pneumatic sealing systems. The temperature calculations were validated using a thermal camera. Furthermore, since his model considers the surface topography, he was able to predict and optimize the friction behavior of a pneumatic sealing contact by changing the surface topography.

3.2 Distributed parameter models

In contrast to the lumped-parameter models considered in the previous section, distributed parameter models are based on partial differential equations. As a result, these models can also take into account input parameters such as the geometries of the seal and counter surface. A disadvantage of these models is that their solution is significantly more time-consuming and computationally expensive than the solution of lumped parameter systems.

For pneumatic sealing contacts, by far the most common approach to distributed parameter modeling is structural simulation using commercial finite element programs (FEM programs) such as Abaqus/Standard or Ansys Mechanical.

To calculate friction, contact pressure and/or wear, a model of the seal geometry has to be created first. Hyperelastic material models such as the Mooney-Rivlin model are often used as material models, for example in the work of Debler [35], Belforte et al [12, 18] or Zhang et al [36]. Following the meshing and parameterization of the material model, various loads and boundary conditions are imposed on the model, such as pressures or the installation situation.

A decisive point in the modeling of friction is the choice of contact model. In addition to the Coulomb friction model, many commercial FEM programs also offer the option of implementing their own friction models. Nevertheless, the Coulomb friction model has been used most frequently in the past in distributed parametric modelling due to its simple parameterization. For friction contacts in pneumatic sealing systems, values of 0.3 up to 0.6 for dry contacts and values of 0.05 up to 0.2 for lubricated contacts were usually selected [12, 18, 21–24].

In addition to selecting a constant coefficient of friction, it is also possible to specify a function for the coefficient of friction that defines it e.g. as a function of the velocity. For example, Raparelli et al [37] used a velocity dependent coefficient of friction determined experimentally on a tribometer. They determined the coefficient of friction for three lubrication conditions, which they described as dry, boundary lubrication and fluid friction. The values of the coefficient increased degressively with velocity in the considered velocity range of 20 to $250\ \mathrm{mm/s}$, in the dry state from 0.55 to 0.75 and in boundary lubrication from 0.2 up to 0.35. For fluid lubrication, the coefficient of friction was below 0.05 for all speeds. Raparelli et al carried out simulations with all three lubrication conditions. He showed that the test results best matched the simulation results with boundary lubrication.

Regardless of the material or friction model, distributed parameter simulations can be used for various applications. Since the modelling is dependent on material and geometry parameters, existing geometries can be analyzed and optimized. For example, Calvert et al [20] used an FEM model with a constant coefficient of friction to calculate the force required to move an X-ring seal in a spool valve. The analysis of the calculated deformation of the seal predicted a loss of the sealing effect in certain operating conditions. This assumption was confirmed experimentally. Based on the simulation results, the seal cross-section was optimized so that no more leaks occurred in the simulation. Subsequent experiments with the optimized geometry confirmed that the problem had been solved.

Further optimizations of the geometry of the seal and seal seat were carried out by Conte et al [12, 18, 21–23]. In these studies, the simulation model was not validated using the calculated friction forces but using the contact pressure distribution. There was good agreement between the calculated and measured contact pressure distribution. The aim of the optimizations was to ensure the highest possible maximum contact pressure with a simultaneously low normal force in order to achieve the highest possible sealing effect with a low friction force.

In addition to geometry optimization, distributed parameter simulations can also be used to investigate the influence of assembly and manufacturing tolerances. These investigations were carried out by Belforte et al [18] for the diameter of the counter surface of a valve seal. Lin et al [19] investigated at the influence of straightness and roundness of the running surface on the friction force of the piston seal of a pneumatic cylinder.

Debler [35] dealt with the calculation of wear for a pneumatic lip seal, the wear of which he calculated mathematically using an FEM model. The local amount of wear was assumed to be proportional to the local pressure. The local pressure was calculated as the weighted sum of the pressures for different load cases. A comparison of measurement and simulation shows that the measured wear profile could be predicted very well using the FEM-based wear model.

3.3 Conclusion on computational modelling

In summary, all of the presented lumped parameter models use strong abstractions. The models can be used well for descriptive modeling or control of systems, as shown for example by Valdiero et al [31] or Soleymani et al [32]. However, it is not possible to increase the understanding of the mechanisms of pneumatic sealing contacts with these models, as measured data of the friction force or empirical knowledge must be used for parameterization.

In order to gain a better understanding of the processes in the modelled sealing contact and to make optimizations, distributed parameter models are more suitable than lumped parameter models. The potential for the analysis and optimization of sealing contacts was shown, for example, in the investigations by Calvert et al [20]. Using an FEM calculation, they succeeded in correctly predicting the leakage of a specific sealing ring geometry and developing an optimized concept, the performance of which they subsequently confirmed experimentally.

Most optimizations based on distributed parameter models are mainly based on a consideration of stresses in the material or the contact pressure. Friction in the contact, which is determined, for example, by the material pairing, the surface structure and the properties and film height of the lubricant, has not yet been taken into account by the optimization. What all the sources mentioned have in common is that no detailed modeling of the coefficient of friction has been carried out. Instead, the friction was modeled using Coulomb friction with a constant coefficient of friction or with an experimentally measured characteristic diagram. Thus, no understanding of the processes in the actual contact zone can be gained. Consequently, the theoretical potential of distribution parametric simulations to evaluate and optimize different designs before carrying out experiments cannot be fully exploited.

4 Conclusion and outlook

The research into the state of the art shows that a large number of experimental studies have been carried out in recent decades. However, the results of the experimental investigations reveal some significantly different outcomes, even for basic qualitative relations such as the velocity dependence of friction. These differences can presumably be explained by the fact that many publications only provide incomplete information on the boundary conditions of the tests presented, such as material behavior, the nature of the surfaces and, in particular, the type and quantity of lubricant used. Consequently, the results are difficult to compare with each other and it is difficult to derive general findings or gain an understanding of the mechanisms in pneumatic sealing contacts. Accordingly, there is little literature on the theoretical modeling of friction in pneumatic sealing contacts. Most of the literature sources listed in these publications are more focused on empirically based description than on increasing the understanding of the mechanisms in pneumatic sealing contacts.

According to the presented literature review, the next steps for research on pneumatic sealing friction should focus on the grease. Both the relevant properties of the grease as well as the resulting grease film heights during operation need to be better understood before further research into other influencing factors is conducted. With precise experimental investigation and theoretical modelling, new system models and design paradigms for pneumatic sealing contacts to reduce friction and increase lifetime can be derived.

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https://doi.org/10.61319/1H3THQXY