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Chapter 1

RESEARCH ON THE TRIBOLOGY OF HYDRAULIC RECIPROCATING SEALS

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ABSTRACT

Hydraulic seals are found in industrial applications involving linear or rotary motion, as for example in hydraulic actuators. They are usually made of a polymeric material (for example, elastomer or "rubber") or a combination of materials (composite seals, for example, elastomer and PTFE with glass fibres). Their shape varies from the typical rectangular cross-section with chamfered or rounded corners and the typical O-ring to hundreds of less conventional designs with complex geometries, although they all have the same basic function, which is the sealing of fluids, normally under relatively high pressure (typically up to 80 MPa) and with operating temperature ranging from subzero values (typically as low as –65 °C) to relatively high values of up to 200 °C, depending on application. Low-pressure applications are also met when seals are used as wipers, as for example in tandem seal arrangements.

Theoretical research on sealing involves concepts and methods from elastohydrodynamics, contact mechanics, thermoviscoelasticity, adhesion and surface topography, in order to achieve good agreement with experimental results and industrial experience, yet this is still quite difficult to achieve because of the mathematical and numerical complexity of the problem. Proof of such difficulty is the fact that after more than 60 years of research in this field, fundamental aspects of the problem are still being tackled, for example, elastohydrodynamics with surface roughness effects, whilst making simplifying assumptions about others, for example, treating seal mechanics in the frame of linear elasticity and ignoring frictionally-induced thermal effects.

The present chapter explores the progress and research trends in computational and experimental tribology of hydraulic, reciprocating, rod and piston seals. Topics include

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the solution of the elastohydrodynamic and contact mechanics problem of flexible polymeric and composite seals, modelling of seal extrusion and anti-extrusion rings, seal elasticity and its effect on sealing performance, modelling of tandem seals, rotary vane seals, transient effects in lubrication, as well as performance evaluation in terms of leakage, friction, extrusion and wear, followed by optimization. Experimental studies are also briefly discussed with a presentation of the difficulties in validating existing models and in producing realistic, reliable and consistent results. The review covers the period from the 1940s to 2008 and serves as a reference source for further study and development in this challenging field, from the original basic experimental rigs and archaic computers of mid 20th century to the sophisticated numerical methods and expensive experimental devices of the recent era.

1 INTRODUCTION

There are mainly two types of hydraulic reciprocating seals: rod and piston seals (Fig. 1). They are typically made of polymeric or plastic materials and most commonly by elastomers. Hydraulic seals are met in industrial, automotive and aviation applications involving linear and rotational motion, as for example in linear hydraulic actuators (Fig. 1) [1] and rotary vane actuators [2].

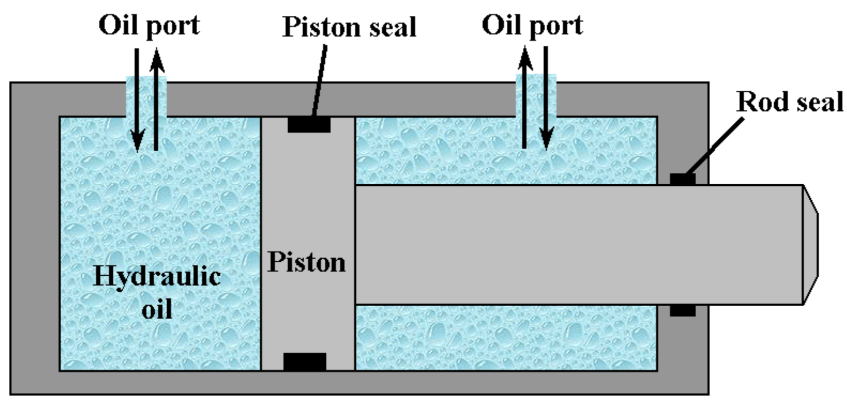
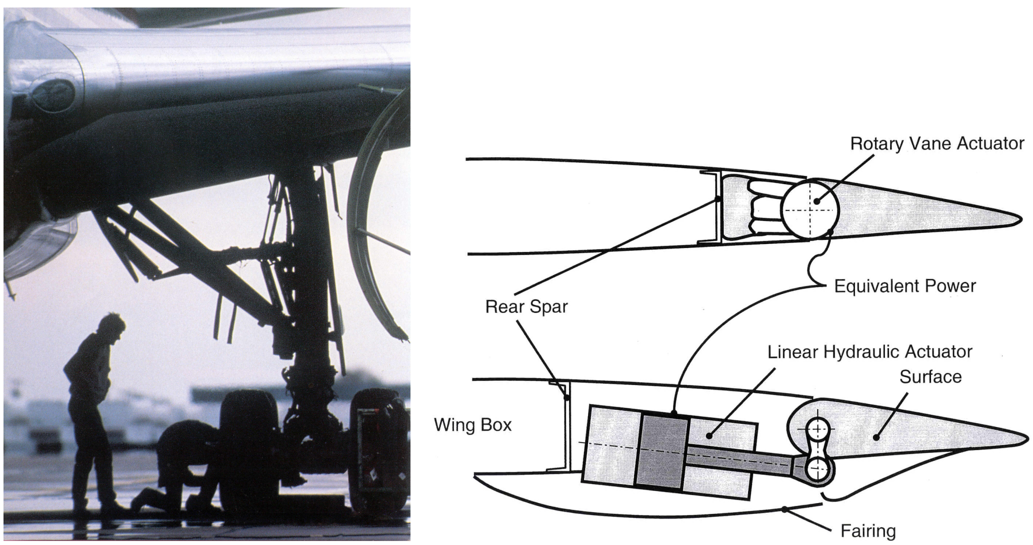


Figure 1. Typical hydraulic seals in a linear hydraulic actuator

These seals operate under dynamic conditions of variable sealed pressure, stroking velocity and operating temperature. Specifically, sealed pressures can reach 80 MPa (usually, they are lower than 50 MPa), stroking velocities can reach several metres per second (usually, they are lower than 0.5 m/s), and operating temperatures can be as low as –65 °C or as high as 200 °C (most often in aviation applications, as is explained later).

Depending on application, a number of seal shapes have evolved over the years based on experience and applied research, often at great financial cost. Seal designing was initially an empirical process through trial and error. In recent times, particularly in the 1990s and onwards, designs have become more sophisticated, based on modern computational tools such as Finite Element Analysis (FEA) software as well as on expensive experimental rigs to study sealing performance under controlled conditions in a laboratory in accordance with international standards. The shape of hydraulic seals varies from the typical seal of rectangular cross-section with chamfered or rounded corners and the typical O-ring to tens of less conventional designs with complex geometries, including composite seals and combinations of seals and other elements such as energizing springs and back-up rings.

Hydraulic seals are critical machine elements. A failure of such elements is associated with financial cost that greatly exceeds their low cost, often by hundreds to thousands of times. Even greater is the safety risk in applications such as in the aviation industry where, for example, hydraulic seals are used in linear hydraulic actuators [1, 3] and rotary vane actuators [2, 4] controlling aircraft landing gear (Fig. 2(a)) and wing flaps (Fig. 2(b)). A graphic example of the critical role of some “humble”-looking seals, although not related to reciprocating motion, is the tragic disaster of the NASA space shuttle Challenger in 1986; that was officially attributed to the failure of a static elastomeric O-ring, which was used to prevent hot gases from leaking through a joint during the propellant burn of the right rocket motor. Apart from financial cost and safety risk, the failure of hydraulic seals may also be responsible for environmental pollution from leaked fluids, particularly when the toxicity of some hydraulic fluids is taken into account. It is clear then that the understanding of sealing mechanisms is of paramount importance to seal designers and manufacturers.



(a) **(b)**

Figure 2. (a) Aircraft landing gear controlled by linear hydraulic actuator; (b) Aircraft wing control surfaces operated via classic linear hydraulic actuator and new rotary vane actuator (drawing courtesy of Smiths Aerospace, UK)

Hydraulic seals are elements of complex mechanical behaviour. Their inherent complexity stems from their material, which, generally, obeys nonlinear stress-strain laws. For example, elastomeric seals are nearly-incompressible, hyperelastic solids. Mechanical response to loading is, generally, viscoelastic or nonlinear, and suitable elasticity models should be used for their mechanical analysis, particularly when the maximum normal strain exceeds 10 to 15 per cent [5, 6]. Elastomer response to stress and strain changes significantly at temperatures close to the glass transition temperature (typically in the order of zero to –70 °C), where the material stiffens and behaves more like leather. Structural changes are normal at such temperatures and reversible, though they can be irreversible if the glass transition temperature is exceeded many times or for long periods of time. Moreover, the thermal expansion coefficient of elastomeric seals is high, typically between 10^{-4} and 3×10^{-4} K^{-1}. This means that their sealing performance is very much related to the operating temperature. In fact, their fundamental mechanical properties such as the moduli of elasticity and rigidity, the Poisson's ratio, the hardness and compressibility, all vary with temperature. Their flexibility is also causing leakage and friction variations during reciprocating motion, which depend on their geometry and operating conditions, and are difficult to be modelled with satisfactory precision. Chemical compatibility with hydraulic fluids is also of concern, as is fluid absorption and swelling in normal operation. Elastomer oxidation and ageing, even when seals are out of service, is yet another important factor to consider. This limits their storage life, typically to under 5 years. These and other factors, which are explained later in this chapter, mean that the computational modelling, performance analysis and application of hydraulic seals is a complicated topic.

Despite the importance of this topic in terms of the number of industrial applications, financial costs and safety risks, related studies in the scientific literature are rarely encountered. This is probably attributed to the difficulty in the computational modelling of hydraulic seals and the study of their tribological performance. However, there is no doubt that in-house studies and software from seal manufacturers exist, yet they are rarely published for obvious reasons of competitiveness secrecy. The author has first-hand experience with sealing research funded by some of the top seal manufacturers [3, 4] and can attest that even in recent times (1990-2000), seal analysis and evaluation is an empirical process to a significant extent. Nevertheless, modern trends and stiff competition dictate that precise computational modelling takes over the traditional expertise of chief engineers, and seal selection and evaluation changes from art to science.

The scientific evaluation of hydraulic seals involves the computation of leakage and friction, as well as the prediction of their wear rate in a given application with given operating conditions. As these seals, naturally, operate in lubricated conditions, the first step in the analysis is to solve the problem of lubrication, commonly known as elastohydrodynamic lubrication (EHL) [1]. The EHL theory originated mainly in the 1950s and 1960s, although there are a few earlier studies exploring the basics of EHL. Applying the elementary EHL theory, sealing research was given an impetus in the 1960s and 1970s when pioneering studies were published on the theoretical and experimental analysis of rectangular and toroidal hydraulic seals. However, lack of computers and robust numerical methods (such as FEA), as well as archaic experimental rigs hindered rapid progress. This is reflected on the leakage and friction results presented in many of those early studies, which are characterised by a degree of scattering.

It is surprising yet true that the pioneering work of White and Denny [7] at the end of World War II, which was an exhaustive, mainly experimental work on reciprocating seals, remains one of the definitive sources of reference. Following that original study, a number of remarkable studies were published mainly in the 1960s and 1970s, most notably references [8-15] on experimental sealing research and [16-32] on theoretical EHL modelling. Very useful reviews have been presented in [33-35] from some of the pioneers in this field.

This chapter explores some of the fundamental technological and engineering aspects in designing optimised hydraulic seals. The geometry, physical properties, mechanical behaviour and performance analysis of hydraulic seals are explored in view of making selective optimisations such as minimising leakage and friction in reciprocating motion. The foundation for the successful tackling of the EHL and contact mechanics problem is laid and some of the most notable solution methods to date are presented for historical and future reference. Topics covered include seal shapes, material properties, operating conditions, mathematical and computational modelling, experimental studies, as well as current research trends. Relatively satisfactory solution methods are presented for simple geometries such as rectangular and toroidal seals, which can be extended to more complex seal geometries and kinematical conditions. The author has been involved in fundamental sealing research and modelling of reciprocating seals for linear hydraulic and rotary vane actuators, having developed computational tools to analyse seal EHL and performance evaluation [1-6, 36-42], nonlinear seal mechanics and related effects [5, 6], transient EHL effects [38], tandem seals [40], extrusion effects [37], back-up rings [39], composite seals [42], etc. These topics are discussed in the following sections in an attempt to explain the basic sealing mechanism of hydraulic seals, providing a source of reference to seal designers, engineers and academic researchers.

2 Hydraulic Seal Materials And Performance Issues

The most common materials used for hydraulic seals are elastomers and thermoplastics [43]. Material selection is based on the intended use of the seal. Most of the simpler shaped seals such as rectangular, toroidal and U-cups are made of some kind of polymeric material, usually elastomer. Seals of less conventional design on demanding applications (as for example for high sealed pressures or extreme temperatures) may utilize composite materials such as bronze-filled polytetrafluoroethylene (PTFE) as in coaxial seals [43] or PTFE with glass fibres and bonded with elastomers as in rotary vane seals [2, 4, 42].

The said materials are viscoelastic or viscoplastic. This means that they are significantly and nonlinearly affected by changes in their stress and strain state, as well as by changes in temperature. They are also susceptible to chemical degradation in reaction to incompatible hydraulic fluids or contaminants, oxidation and ageing. Therefore, they play a vital role in sealing performance and should be matched with the projected application, that is, with the service environment and operating conditions. Some fundamental theoretical studies on the effects of sealing elastomers on the performance of reciprocating seals can be found in [3] and [44], which deal with rectangular and toroidal seals, respectively. In those and similar theoretical studies, material properties for the simulations are typically obtained from time-consuming experiments designed to measure the modulus of elasticity, Poisson’s ratio, thermal expansion coefficient, and, generally, stress-strain and relaxation curves at various temperatures. The data are then used as input to suitable material models, for example viscoelastic such as the generalized Maxwell model [44] or nonlinearly elastic such as the Mooney-Rivlin model [3, 5, 6], which are incorporated in numerical models such as in FEA software.

2.1 Elastomers for hydraulic seals, their benefits and deficiencies

Elastomers are suitable for hydraulic seals because of their flexibility. They have relatively low elastic and shear moduli, which means that they can accommodate large deformation, both tension and compression, as well as shear deformation, without permanent deflection or fracture. Additionally, they are nearly incompressible with Poisson's ratio very close to 0.5, normally greater than 0.490 (a typical value used in numerical simulations is 0.499); this is characteristic behaviour of liquids or metals in perfectly plastic deformation. Their flexibility and incompressibility mean that they can be accommodated in different housings or for different initial interference (pre-loading) and conform to space restrictions or adapt to temperature and sealed-pressure changes whilst maintaining their sealing ability. Moreover, as nearly incompressible materials, they offer seals the ability to transfer the pressure exerted by a hydraulic fluid onto the sealing surface without changing their volume and, thus, perform dynamic sealing, which is proportional to the sealed pressure.

However, elastomers are disadvantaged by several performance limitations. Their main problem is the dependency of their mechanical properties to temperature, strain and strain rate, as well as time (viscoelasticity, relaxation and ageing). Rubber, which is a particular form of elastomer and used extensively in hydraulic seals, is a compound of many macromolecules (see section 1.5 in [45]). Macromolecules are long molecular chains of three types: linear, branched and crosslinked. Linear chains move easily in relation to each other and this explains the softening of rubber with heating and hardening with cooling. Crosslinked chains do not move freely in relation to each other, which explains the resistance of rubber to flow when heated.

The elastic (Young's) modulus of elastomers generally decreases with temperature. For reference, the elastic modulus of a typical elastomer used for rod seals [5] is equal to 341 MPa at –54 °C, 8.9 MPa at 23 °C and 9.5 MPa at 135 °C, based on a compression test at maximum normal strain of ±10 per cent; this represents two orders of magnitude change in the elastic modulus with temperature, which is very significant in applications where the temperature has large variations, such as in linear hydraulic actuators in aircraft (Fig. 2(a)). Thus, elastomeric seals stiffen at lower temperatures. This type of response becomes nonlinear near the glass transition temperature, which, for typical hydraulic elastomeric seals used in the aviation industry [3], is about –45 to –70 °C. Structural changes ensue as a result, which are, generally, reversible, although they can be irreversible to some extend, depending on the duration and degree of the material exposure to such harsh conditions. Obviously, such effects are of paramount importance in designing seals that will remain leak-tight for the projected range of operating temperature and until the end of their anticipated service life.

The thermal expansion coefficient of elastomers is quite high, typically between 10^{-4} and 3×10^{-4} K^{-1} [43]. This means that seal dimensions change significantly with temperature. Therefore, the contact pressure of a sealing contact arising from interference fit of the seal is also significantly affected. This can cause failure of sealing at low temperatures from temporary loss of contact pressure, depending on the sealed pressure, as calculated in [4]. Such effects are obviously taken into account in seal modelling and performance evaluation, as for example in references [1, 4-6, 38, 39, 41] and in similar studies in the literature or in-house evaluations from seal manufacturers.

In conjunction with the effects of temperature on mechanical properties, elastomers exhibit nonlinear response to strain and strain rate. Mechanical response to strain even changes for repeated loading [43]. Figure 3 shows the stress-strain curves for a typical elastomer used for hydraulic seals [3, 5, 6], obtained from a standard test at three temperatures.

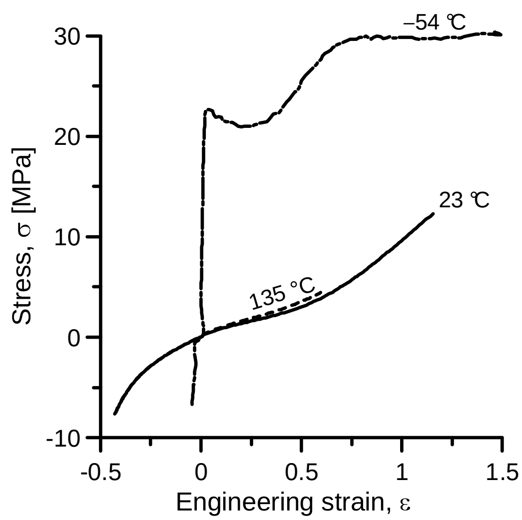


Figure 3. Stress-strain curves of typical elastomer for hydraulic seals [3, 5, 6] (results obtained from Trelleborg Sealing Solutions in England). The glass transition temperature of this elastomer is –47 °C

The elastomer responds rather linearly for very low load and stiffens when the load is increased beyond a limit. The results are clearly very different at subzero temperatures than at room or higher temperature; they are also different between tension and compression. In removing the load at the same rate as the original rate of application, the elastomer, generally, does not follow the same stress-strain path. The rate of load application is also influential, owing to the viscoelastic nature of elastomers. Further complications surface in repeated loading, with elastomers exhibiting hysteretic behaviour, that is, successive stress-strain curves appear displaced. Mechanical response is also affected by whether a previously attained strain level is exceeded or not in successive loadings, particularly so when such effects take place at low temperatures and near the glass transition temperature. These effects are obviously of great concern when designing elastomeric seals that work in a broad range of temperature and pressure for any number of operating cycles, as for example in hydraulic actuators used in aircraft landing gear and wing control surfaces (Fig. 2).

Other important deficiencies of elastomers include:

(a) sensitivity to oxidation, which is accelerated at higher temperature and limits their storage life;
(b) chemical attack from incompatible hydraulic fluids;
(c) swelling from fluid uptake, which obviously changes seal dimensions and affects sealing performance owing to change of the sealing contact pressure;
(d) ageing, which manifests as hardening, embrittlement and eventual loss of seal material;

(e) relaxation and adhesion as for example in seals remaining stationary for long periods of time, which then exhibit high friction during start-up of motion and stick-slip phenomena, manifesting as vibration.

Because of their detailed deficiencies, elastomers are often replaced by other materials in hydraulic seals, mainly plastics and composite materials, as explained in the next section.

2.2 Plastics and composites for hydraulic seals, their benefits and deficiencies

The use of plastics in sealing originated several decades ago and saw an impetus in the 1970s [46]. Progress in the development of new materials continues and the reason for this is the need to overcome elastomer deficiencies (see section 2.1), improve performance and extend the service life of seals. Among the main benefits of some plastics and composites used in sealing are the increased resistance to wear, lower friction, avoidance of stick-slip and relaxation phenomena, wider operating temperature range with more consistent performance, and higher resistance to extrusion during reciprocating motion. PTFE was among the first materials explored [47, 48] but other materials have also found applications. For example, ultra-high-molecular-weight polyethylene (UHMWPE), polyurethanes and other low-stiffness materials. Composites are also in use, for example, bronze-filled PTFE as in coaxial seals [43], PTFE with glass fibres and bonded with elastomers as in (reciprocating) rotary vane seals [2, 4, 42], and PTFE with other fillers such as stainless steel and graphite. The selection of these materials depends on the requirements in a particular application and should match the operating conditions; for example, UHMWPE cannot be used at temperatures higher than about 80 °C [46]. Some PTFE compounds on the other hand can be used at much higher temperatures because of their fillers, which provide added strength.

The tribology of polymers and plastics used in reciprocating sealing applications involves the study of their lubrication with hydraulic oils, wet and dry friction in sliding contacts, abrasive wear when in contact with metallic surfaces, erosive wear when lubricated with particle-containing fluids, etc. The state-of-the-art of polymer tribology (at least in 1998) can be found in an exhaustive review compiled by Zhang [49]; readers are also advised to refer to the excellent chapter on this topic in book [50] – chapter 16. PTFE in various compounds is the most widely used plastic material in reciprocating sealing, therefore it is discussed in more detail next.

PTFE is a thermoplastic material best known for its low friction properties caused by its surface porosity and low surface energy – see page 170 in [43]. In lubricated contacts such as in hydraulic sealing, its surface porosity allows for lubricant storage and subsequent reduction of friction, perhaps similarly to laser textured surfaces. As a result, it is particularly suited to reciprocating seals, which, otherwise, suffer from stick-slip and high friction immediately after long periods of inactivity. However, extensive periods of sliding result in polishing the PTFE contact surface, which, in turn, causes substantial rise of the friction coefficient and accelerated wear of the PTFE. Thus, sliding of PTFE on hard metallic surfaces – such as piston rods in the case of rod seals – under dry or boundary-lubricated conditions, causes excessive wear of the PTFE. The latter is owed to the PTFE undergoing delamination (see Fig. 16.2 in [50]), transferring a thin polymeric layer to its sliding counterface in chunks.

This can be reduced by reinforcing the PTFE with fillers such as glass fibres to produce a composite material of higher strength. However, the harder composites may increase the abrasive wear of a counterface such as a piston rod in the case of rod seals.

Owing to surface porosity and low effective area of contact, PTFE hydraulic seals are favourable towards lubricating film formation in their sealing contact, which results in higher leakage rate. This can be reduced by increasing the contact pressure, yet this negates the benefit of PTFE having a low friction coefficient and can in fact increase the friction force.

PTFE has other important properties, which matter in hydraulic seal applications. PTFE seals have a higher operating temperature limit, for example over 250 °C [46], although that may not be very useful in many applications (for example, linear hydraulic actuators for aircraft landing gear have an operating temperature that is, typically, lower than 140 °C [1, 3]). Moreover, PTFE has a very high resistance to ageing [48]. Its thermal expansion coefficient is in parity with other thermoplastics and in the order of 2×10^{-4} K^{-1}, which is close to that of typical engineering elastomers for hydraulic seals. The latter encourages matting of PTFE with elastomeric materials in composite seals [42], combining the benefits of both materials, as is discussed later. However, great attention to detail is advisable because of the many conflicting properties of PTFE. In general, this is a material that requires a lot of thought and analysis before it is used in hydraulic seal applications. This is perhaps more emphatically realised when considering that there is no such one "PTFE" material but, actually, different forms with different properties. For example, the Poisson's ratio of a PTFE compound can be as high as 0.46 or as (negatively) large as –12 (the latter in the so-called expanded PTFE). Moreover, the behaviour of PTFE in compression is different than in tension, which is easily observed in uniaxial stress-strain tests [51]. This means that its modulus of elasticity, yield point and work-hardening behaviour are different. Moreover, the mechanical properties of PTFE vary with time and temperature. They are also affected by fabrication methods [51]. Details on the mechanical properties of various forms of PTFE, including composites such as those with glass fibres, can be found in [51-54].

3 Design And Application Of Hydraulic Seals

The performance of seals designed for reciprocating motion in high-pressure hydraulic systems plays a critical role in the efficiency and safe operation of such systems. Hydraulic seals are typically designed to service hydraulic equipment for a few million operating cycles. The financial cost of a potential failure can be very high considering the loss of productivity and man-hours consumed to fix problems. Moreover, safety risks involved in applications such as in the aviation industry (Fig. 2) are very high. Therefore, the technology of hydraulic seals sees continuous progress towards optimised geometries, designs and material combinations for improved performance, even tailor-made for specific applications.

Depending on application, some important requirements in selecting the best available seal are as follows.

(a) Low leakage rate.
(b) Low friction and, as a result, low power loss and high efficiency.
(c) Resistance to wear and long service life with consistent performance.

(d) Resistance to gap extrusion (usually in high-pressure applications).
(e) Resistance to low and high temperatures (for example, in the aerospace industry).
(f) Chemical compatibility with sealed fluids.
(g) Ease of installation.
(h) Low cost.

The previous requirements are usually met in combinations and some may be conflicting, as for example requirements (a) and (b) (low leakage and low friction), in which case a compromise must be accepted. For a given application, the main parameters used for seal selection in conjunction with the listed requirements include the size, maximum sealed pressure, maximum stroking velocity (speed) and acceleration, the range of operating temperature and the chemical properties of the sealed fluid.

Based on the requirements and functional parameters used in seal selection, a large number of seal geometries and designs has appeared and more are being developed to meet specific demands. A large amount of specialised information can be found in product catalogues of seal manufacturers, with hundreds of seal designs. In the next two sub-sections, some important designs of rod and piston seal configurations are presented, as taken from one of the major seal manufacturers.

3.1 Examples of rod seal geometries and configurations

Rod seals are hydraulic seals used in equipment such as linear hydraulic actuators (Fig. 1). Polymeric, plastic and composite seals are suitable for this task, depending on application. Among the basic requirements of rod seals is the very low or zero leakage-per-cycle and low friction in dynamic conditions (reciprocating motion), zero leakage in static conditions (no motion), and ease of installation. These are among the most basic-functioning of all seals and have been under development for several decades. As a result, they have become highly specialised in their functionality and many geometries and designs have evolved.

Although the rectangular rounded seal [1, 3] and the O-ring are the most basic of all rod seals, they are not very efficient in many applications; for example, O-rings suffer from high leakage and are best suited to static sealing. A compilation of some characteristic geometries and designs is presented in Fig. 4, collected from a catalogue of one of the major seal manufacturers [55]. This is only a small selection and does not include some arrangements which are more complex. Many more seal shapes and combinations exist and details are provided in product catalogues of seal manufacturers. Rectangular seals and stand-alone O-rings are not included as they are very simple and well-known.

For the sake of understanding some of the functionality, limitations and benefits of rod seal designs, let us discuss some of the properties of the seals shown in Fig. 4. Although the description is mainly based on [55], the main characteristics of the seals are typical.

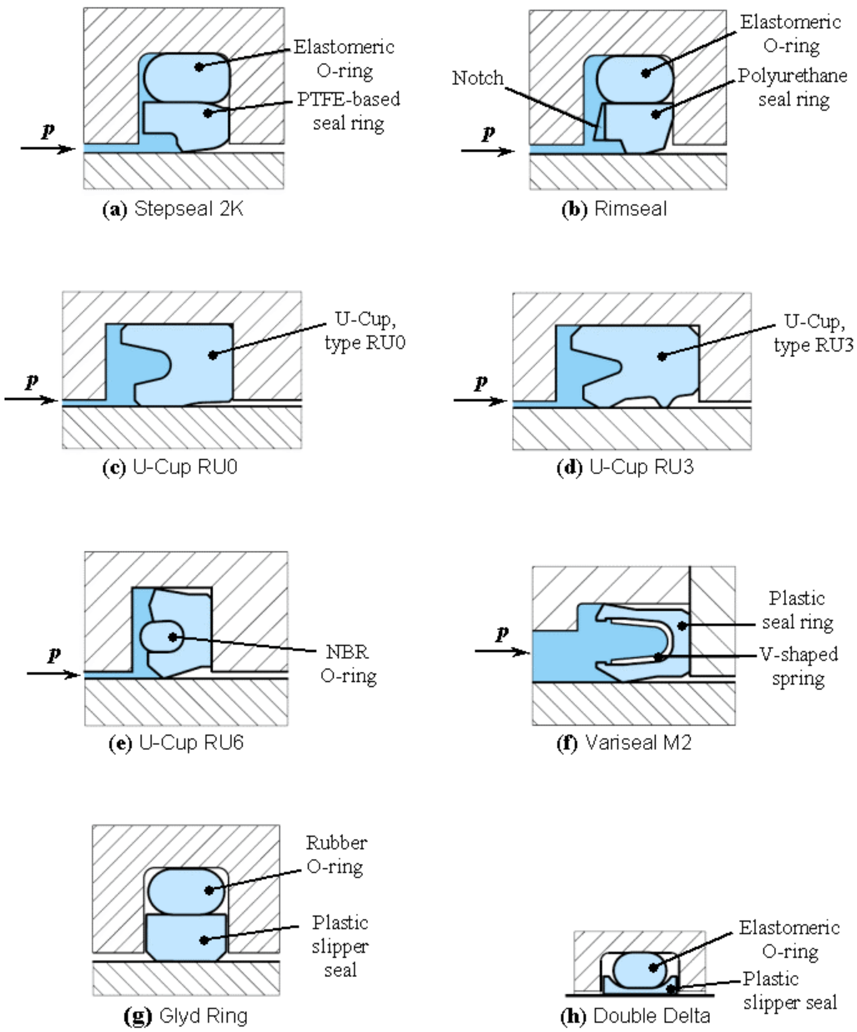


Figure 4. Examples of rod seals (compiled from catalogue [55])

(a) **Stepseal 2K**. This consists of a single-acting seal element made of PTFE-based or polyurethane material, which is energised by an elastomeric O-ring upon installation to provide pre-loading. The combination of the two elements gives this seal great flexibility, which allows it to be easily installed. The flexibility of the O-ring compensates hardware tolerances and movement. The plastic material of the seal ring allows for low friction, low abrasive wear, low gap extrusion, and eliminates stick-slip. The geometry of the seal ring allows for steep rise of the contact pressure on the left side during outstrokes (motion from left to right in the figure), which reduces film thickness under the seal; on the other hand, there is a smoother rise of the contact pressure on the right side during instrokes, which facilitates hydrodynamic film development under the seal to bring any leaked fluid back in. This pressure mechanism results in very low to zero leakage per cycle, as is explained in later sections of this chapter. This seal is suitable for a very wide range of operating conditions, namely temperatures between –45 and +200 °C, sealed pressures up to 80 MPa and maximum stroking velocity of 15 m/s with reciprocating frequency up to 5 Hz. It is often found in tandem seal arrangements as the primary seal.

(b) **Rimseal**. This consists of a single-acting, polyurethane seal element, which is energised by an elastomeric O-ring upon installation to provide pre-loading. It is often found as the secondary seal in tandem seal arrangements where the primary seal is the stepseal describes previously in (a). The chamfer this seal has at its low-pressure side (right) enhances the development of a hydrodynamic film during instrokes, thus reducing the leakage per cycle. Another chamfer on its high-pressure side (left) allow a notch (pictured) to come into contact with the flank of the groove at higher sealed pressures. Overall, the geometry of the seal ring can be optimised to minimise leakage in reciprocating motion by allowing for an optimum distribution of contact pressure of the seal ring during operation. This seal is suitable for operating temperatures between –45 and +100 °C (depending on the O-ring material), sealed pressures up to 25 MPa as an individual seal or up to 60 MPa as the secondary seal in tandem seal arrangements, and maximum stroking velocity of 5 m/s.

(c) **U-Cup RU0**. This is a single-acting, single-lip (asymmetric), compact seal made of polyurethane. Owing to its flexibility, it can easily accommodate deflections of the piston rod and adapt to changes of the sealed pressure. However, at low stroking velocities, it may suffer from stick-slip motion. This seal is suitable for mineral-based hydraulic fluids with operating temperatures between –35 and +110 °C, sealed pressures up to 40 MPa (depending on the rod-gland radial clearance) and maximum stroking velocity of 0.5 m/s.

(d) **U-Cup RU3**. This seal has similar functionality and properties as seal (c). Its most obvious difference from the previously described seal is the additional, small rear lip, which reduces leakage and prevents the entry of contaminants from the air-side of the seal (right). The small amount of lubricant inevitably trapped between the two seal lips keeps the seal lubricated and prevents dry running. As a result, the stick-slip tendency associated with the RU0 U-cup design (c) is reduced.

(e) **U-Cup RU6**. This U-cup seal has similar properties and technical data as other U-cup seals discussed previously (RU0 and RU3). Due to its geometry and integrated rubber O-ring, which gives it flexibility and provides a pre-loading upon installation, it has excellent sealing performance, regardless of sealed pressure level. The short sealing lip reduces friction in comparison with other U-cup designs.

(f) **Variseal M2**. This is a single-acting, plastic U-cup seal. The U-cup is made of a PTFE-based or polyurethane material and is energised by a spring, as shown in Fig. 4. The asymmetric profile of this seal with optimised angle of the lip offers low friction and long service life. The spring offers the necessary initial pre-loading of the seal to avoid leakage in low system pressure, whereas at higher sealed pressure, the seal works automatically as all other rod seals by transferring the sealed pressure to the sealing contact with the piston rod. Due to its materials and construction, this type of seal is suitable for a wide range of operating conditions, namely temperature between –70 and +260 °C, stroking velocity up to 15 m/s, and sealed pressure up to 40 MPa in static loading.

(g) **Glyd Ring**. This is a double-acting, plastic-faced seal, comprising a PTFE or polyurethane based slipper ring, energised by a rubber O-ring. This type of seal has been in service for several decades as it is reliable, effective, and has low friction, high wear resistance and virtually no stick-slip problems. The O-ring provides the pre-loading upon installation and at very low sealed pressures. As the sealed pressure is increased, the O-ring is squeezed by the sealed fluid and pushes the slipper ring against the rod surface to prevent leakage. As a matter of fact, the slipper seal may have lateral notches to allow fluid to enter the seal housing quickly and pressurise the O-ring when there is abrupt rise of the sealed pressure. This seal can be used in a broad range of operating conditions, namely temperatures between –45 and +200 °C, sealed pressures up to 80 MPa and maximum stroking velocity of 15 m/s with reciprocating frequency up to 5 Hz.

(h) **Double Delta**. This is a double-acting, plastic-faced seal, comprising a polyurethane-based slipper seal, energised by an elastomeric O-ring. It is designed to expand under sealed pressure and typically installed in existing O-ring grooves as an improvement to O-rings. The mechanism of operation is the same as that for the Glyd ring (g), namely preloading by the initial interference of the O-ring in the seal housing and subsequent pressurisation of the O-ring as the sealed pressure is increased, with the O-ring then energising the slipper and pushing it against the piston rod. The benefits of this arrangement are the same as for the Glyd ring but the maximum sealed pressure is much lower, namely 35 MPa.

3.2 Examples of piston seal geometries and configurations

Similarly to rod seals, piston seals are used in equipment such as linear hydraulic actuators (Fig. 1) and are made of polymeric, plastic and composite materials, depending on application. They have been under continuous development for several decades and many complex shapes have evolved from their study, based on industrial experience and scientific analysis. A compilation of some geometries and designs is presented in Fig. 5, collected from a catalogue of one of the major seal manufacturers [56]. This selection does not include some more complex designs. Many more seal shapes and combinations exist and interested readers can find detailed information in product catalogues of seal manufacturers.

(a) **Glyd Ring T**. This is a double-acting, plastic-faced seal, comprising a PTFE or polyurethane based slipper ring, energised by an elastomeric O-ring. The functionality and technical data of this seal are the same as for the simpler glyd ring used on piston rods (see section 3.1 – case (g)) and the aforementioned seals are interchangeable. This one though has inclined profile flanks on the seal ring, which, in conjunction with the edge angle (see chamfer) helps the seal tilt away from the sealed pressure side. Improved sealing is then achieved by the steep pressure rise at the edge of the seal on the sealed-pressure side.

(b) **AQ-Seal 5**. This is a double-acting, rubber energised, plastic-faced seal. It comprises a PTFE-based seal ring, energised by two elastomeric O-rings and hosting a quad-ring seal. The O-rings provide the necessary preloading for very low sealed pressures by pushing the two other elements of this seal. As the sealed pressure is increased, fluid entering the seal housing compresses the O-rings, which in turn push the other seal elements against the sealed surface. This design combines the benefits of a slipper seal and an elastomeric seal offering very good sealing, low friction and no stick-slip effects. It is suitable for operating temperatures between –30 and +200 °C, sealed pressure up to 60 MPa and speeds up to 3 m/s.

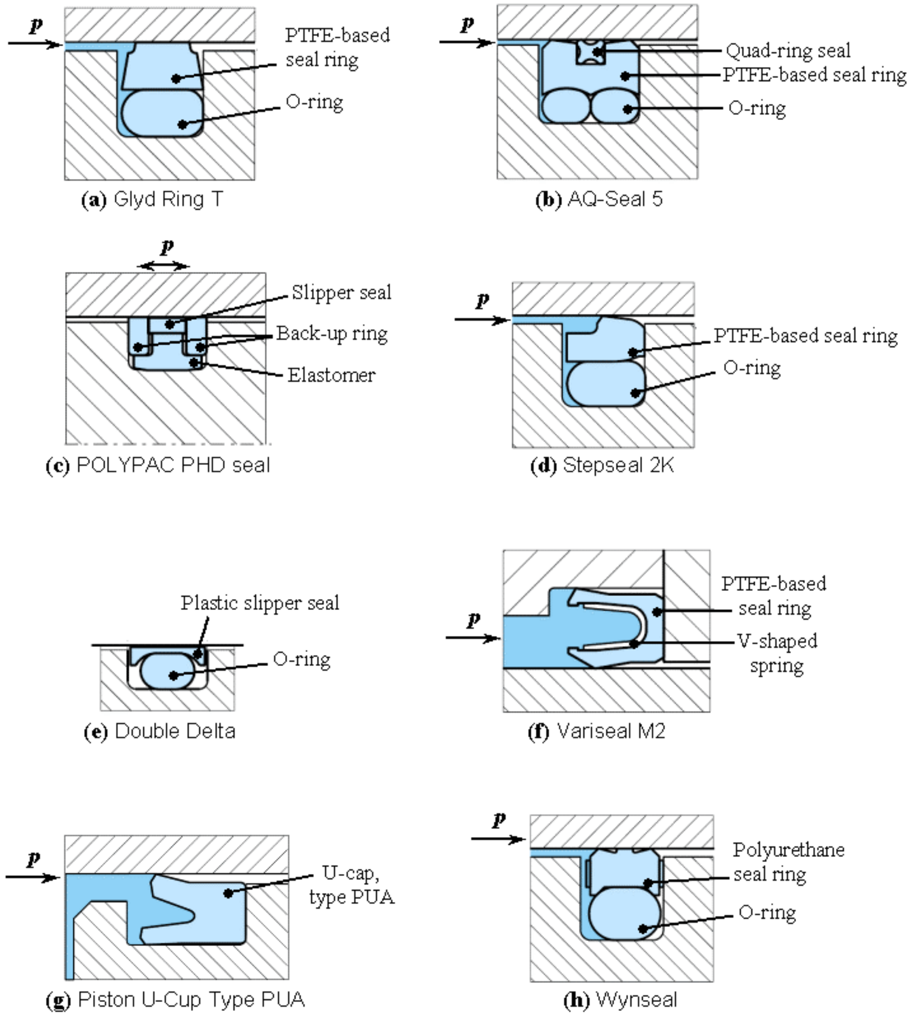


Figure 5. Examples of piston seals (compiled from catalogue [56])

(c) **POLYPAC PHD**. This is a double-acting seal for heavy duty, high-pressure applications. It comprises a PTFE-based slipper seal, which is energised by an elastomer and supported by a back-up ring on each side. The sealing mechanism is the same as in previously described seals and based on the preload offered by the elastomer in addition to its compression by the sealed fluid at higher sealed pressures, which is transferred to the slipper seal to achieve automatic sealing. Some of the main benefits of this seal include its high resistance to wear and extrusion (as a result of the back-up rings), low friction, no stick-slip effects and a long service life. It is suitable for operating temperatures between –45 and +135 °C, sealed pressures up to 40 MPa and sliding speeds up to 1.5 m/s.

(d) **Stepseal 2K**. This seal can also be used as a rod seal. It is described in section 3.1 – case (a).

(e) **Double Delta**. This seal can also be used as a rod seal. It is described in section 3.1 – case (h).

(f) **Variseal M2**. This seal can also be used as a rod seal. It is described in section 3.1 – case (f).

(g) **Piston U-Cup, Type PUA**. This is a single-acting seal with single, asymmetric lip. It is made of polyurethane. Its functionality and technical data are both similar to those of the U-Cup RU0 rod seal – see case (c) in section 3.1.

(h) **Wynseal**. This is a double-acting seal, comprising a polyurethane seal ring, energised by a rubber O-ring. The O-ring offers the necessary preloading for sealing at very low sealed pressures. As the sealed pressure is increased, the hydraulic fluid entering the seal housing compresses the O-ring, which in turn transfers pressure to the seal ring. The two upper seal edges act as primary seal for pressures from both directions (right and left), whereas the central back-up increases the sealing effect (this can be realised by looking into the similarities with a common twin-lipped U-cup seal – see for example case (d) in section 3.1). The seal is suitable for sealed pressures up to 40 MPa, operating temperatures between –35 and +110 °C, and speeds of up to 0.5 m/s.

4 Experimental Studies On Reciprocating Seals

4.1 A selection of some important early experimental studies

Systematic experimentation on hydraulic seal performance commenced in the 1940s. The pioneering work of White and Denny [7] between September 1944 and December 1946 was drafted in the United Kingdom by members of the (then) Ministry of Aircraft Production, the Royal Aircraft Establishment, and Imperial College of Science and Technology. In the era of World War II, the increased demands for improved performance and reliability of hydraulic seals used on aircraft dictated an understanding of their fundamental performance mechanisms. Time was of the essence and progress was swift. The exhaustive work of Denny under the general direction of Professor White produced a large amount of experimental data and remains valuable even today. The work of White and Denny [7] put science into the design of hydraulic seals, which, up to that time, was rather empirical. Their extensive report describes many experiments on flexible packings, including rectangular, toroidal and U-section seals. The experiments on various polymeric materials and with various sealed fluids dealt with measuring the friction force, friction coefficient and leakage rates at different sealed pressures and operating temperatures, the mechanisms of seal extrusion and how this could be eliminated, the mechanisms of seal failure from abrasion and extrusion or fracture, the effects of material hardness and the initial seal interference on the results, and similar topics of importance.

In lack of sophisticated experimental devices at the time, White and Denny had to resort to ingenuity to complete some of their tests with satisfactory precision, as for example in measuring the mass of leaked fluid. They demonstrated the effect of the seal material hardness in reducing the abrasive wear and extrusion (which led to cutting of a corner) of the seal. They proposed seal and housing arrangements, including anti-extrusion rings, to minimise or eliminate seal damage from extrusion. They managed to measure the distribution of the contact pressure at the sealing interface and located the most strained zone of a seal. They demonstrated the proportionality of the friction force on the contact area at the sealing interface and showed the effect of seal hardness on the frictional force. During their experiments with various fluids and at various speeds, and by demonstrating the effect of those on the measured friction force, they established the transition from the partial or mixed lubrication regime (i.e. with a significant degree of roughness asperity interactions at the sealing interface) to the hydrodynamic lubrication regime, or vice versa. Subsequently, they experimented with surface-finish effects, not only in relation to the frictional force but also to the static friction and stiction observed when an elastomer has remained at rest for some time and is relaxed.

Having collected many results from their parametric study, it became clear that elastomeric seals, owing to their incompressibility, achieve automatic sealing under dynamic conditions of variable sealed pressure by readily transferring the sealed pressure to the sealing interface, provided that they have been given an initial interference (pre-loading). White and Denny then embarked on a theoretical analysis based on the Reynolds (lubrication) equation to explain and back-up some of their experimental findings. Without a doubt, their work was ahead of its time and set the foundation for subsequent studies in the 1960s and 1970s to build on, taking advantage of modern hardware offering greater precision.

Nearly twenty years after the work of White and Denny [7], the first notable experimental studies began appearing in the literature, mainly in the international fluid sealing conferences organised by the British Hydromechanics Research Association. In 1964, three notable experimental studies were presented at the second international fluid sealing conference in England from some of the pioneers in this field.

Cnops [57] devised a rather simple experimental rig to measure the friction of elastomeric, cup, piston seals. The piston was loaded by a spring in a hydraulic cylinder and the volume of oil in the cylinder, which was brake fluid, was varied harmonically. The piston displacement, average pressure, and reciprocating velocity were all varied. The experiments demonstrated creep and relaxation effects typical of elastomeric materials. They also showed stiction effects and a subsequent development of a fluid film under the seal at increasing speeds, which gradually collapsed when the motion was slowed down or stopped. These effects are well-known today and explainable via the theory of hydrodynamic lubrication.

In a more advanced study, Lawrie and O’Donoghue [9] constructed an apparatus to examine the friction and lubrication of piston seals made of natural rubber and used in automotive clutch and brake master cylinders. They used a hydraulic cylinder with commercial brake fluid pressurised by a pump. The piston velocity and friction forces were measured by displacement transducers. The existence of rubber-metal contact was established by using conducting rubber seals and measuring the sealing contact resistance; zero resistance indicated full contact and infinite resistance indicated no contact or, in other words, full-film separation between the seal and its counterface. A multi-channel recorder enabled simultaneous reading of pressure, frictional force, speed and contact resistance for a complete operating cycle. Several tests were performed this way and useful results were obtained, showing how the seal performance varies during a cycle.

In an equally important study, Müller [8] presented an experimental analysis of elastomeric O-rings and quad (X) rings in reciprocating motion. His emphasis was on understanding hydrodynamic film formation at the sealing contact and the transition from the boundary to the hydrodynamic lubrication regime. He established the effects of fluid viscosity, stroking velocity, interference pressure and seal dimensions on the leakage and friction of (mainly) O-rings. He also discussed the differences in hydrodynamic film formation between outstrokes and instrokes of the piston rod, and verified the thinness of the typical lubricating film at the sealing contact.

In 1969, Dowson and Swales [11], following on the path of Müller [8] and taking advantage of the emerging EHL theory by the first author (Dowson), presented their experimental findings at the 4th international conference on fluid sealing. They used a rotating disc machine to test a cylindrical rubber block under conditions simulating the action of a reciprocating seal undergoing an infinitely long stroke. Capacitance techniques were used to measure film thickness and a piezo-electric transducer measured the sealing contact pressure. Their results were compared with calculations based on the EHL theory and the agreement was reasonable. Among some of their important findings was the realization of the fundamental sealing mechanism in reciprocating seals, namely the difference in film thickness at the sealing interface between the extending and the retracting stroke. They also confirmed that the film thickness increases with speed and decreases with applied pressure. Nevertheless, the present author would advise caution to the reader to not generalise such results, the reasons explained later in this chapter.

Another important contribution from that era, presented at the same conference in 1969, was that of Aston et al. [58]. They described three apparatuses and a series of experiments to measure the sealing force of rubber seals (Viton and fluoro-silicone compound) at temperatures up to 200 °C. Thus, they showed the change of the sealing force with temperature, following expansion and contraction of the rubber specimens. More importantly, they demonstrated the physical and chemical relaxation of rubber leading to a reduction of the sealing force in time and studied the recovery rate of the material. This is of great importance in elastomeric seals given the periods of inactivity they undergo under static compression at very low or very high temperatures (for example in the aviation industry), as well as when considering the effects of rubber ageing. A physical explanation of these phenomena was known at the time and provided by the network theory of rubber – see for example reference [59].

In 1971, Nau [14] presented his results on a series of experiments to measure friction of reciprocating rectangular rubber seals in lubricated conditions over a wide range of speeds and pressures. The goal was to understand stick-slip phenomena and the relation of friction to speed. This was already established and explained theoretically: as speed is increased from zero, friction rises, peaks and then falls. The speed at which the friction peak is observed depends on temperature and on the viscoelastic properties of rubber [60]. He found good correlation with experimental data on dry rubber friction and speculated on the nature of rubber seal friction, which he believed to be related to boundary lubrication phenomena. In the case of rubber friction on rough surfaces, it is worth noting that the friction mechanism involves hysteretic losses in the rubber, which may cause a secondary peak in a friction-speed diagram [14]. In any case, the friction of polymers on hard surfaces is a complex phenomenon involving many parameters [61].

Continuing on the topic of reciprocating-seal friction, Field and Nau [10, 35] produced a variety of experimental results in the 1970s on the pressure distribution, film thickness, friction and leakage of rectangular rubber seals. In measuring film thickness, they used optical interferometry and electrical transducers. Those results have similarities with the experimental results of White and Denny [7] but, in this author’s opinion, they were often masked by some apparent inconsistencies. The inconsistencies – as for example in the form of wavy experimental curves in performance diagrams – could be attributed to the limited availability of high-precision instrumentation at that time. It may be surprising though that such inconsistencies are highlighted in a study [62] published much later, in 1988. The results reported in [62] revealed a significant degree of scatter in experimental results on reciprocating rubber seal performance. The surprising fact is that the said scatter refers to differences in results obtained from seven laboratories in different countries and for tests performed under strictly controlled conditions. A possible explanation was the lack of standardised methods for the tests or difficulty in conforming with the test specifications.

In 1975, taking advantage of their experience from previous studies, Field and Nau [15] presented a remarkable experimental study on the effects of design parameters on the performance of reciprocating rubber seals. Among other things, they studied the effects of seal hardness, interference (initial strain or pre-loading), back-up clearance and seal edge geometry. They confirmed the now well-known fact that the development of a hydrodynamic film at the sealing contact depends on the stroking length and its ratio to the contact width of the seal: if the stroking length is greater than two times the contact width, a full hydrodynamic film can be developed and leakage (per stroke) takes place. If the said length is less than two times the contact width, the development of a hydrodynamic film is incomplete. (This can be understood by visualizing the fluid transportation under the seal at the average speed of the two counterfaces, which is equal to half the stroking velocity because one of the counterfaces is stationary.) The observations of Field and Nau led to plotting two characteristic diagrams showing the film thickness and friction versus the position of measurement through the stroke – see Fig. 6.

Similar results had been derived in an earlier study (1971) by Hirano and Kaneta [13]. Apart from the criticality of the stroking-length-to-contact-width ratio in establishing a full elastohydrodynamic film, Hirano and Kaneta discussed the starting friction of seals that have remained at rest for some time. That friction is considerably higher prior to full hydrodynamic film development and gives the characteristic stiction during the start-up of motion. Those effects had been discussed much earlier in the literature, as for example by Denny [63] in 1959.

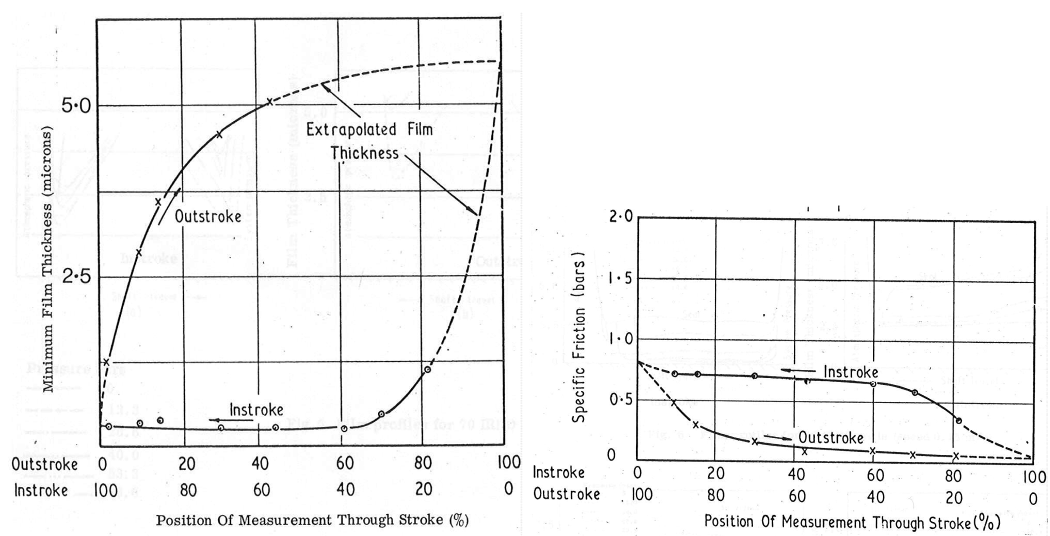


Figure 6. Variation of minimum film thickness and frictional force through the stroke of a reciprocating rectangular rubber seal (from Field and Nau [15])

4.2 Other experimental studies, progress and research trends

Apart from the pioneering experimental studies discussed in the previous sub-section, a number of other studies can be found in the literature, although the volume of published work in this field is substantially smaller than that of other machine elements such as bearings and gears. A selection of those is presented next. The selection excludes a few studies (mostly doctoral theses and reports) published in languages other than English. A detailed discussion of these and some other studies up to the early 1990s can be found in Kanters' thesis [64] and related publication [65], as well as in a follow-up thesis by Visscher [66].

Kambayashi and Ishiwata [67] studied the contact width and sealing force of reciprocating seals. Static contact pressure distributions have been measured by several researchers [20, 68-75]. A number of techniques and apparatuses have been used in those studies. For example, strain gauges, inductive transducers for measuring displacements, piezo-electric force transducers [10, 11, 76], and photo-elastic techniques [77]. Film thickness measurements in sealing contacts have also been performed by several researchers. The methods used involved inductive transducers for measuring seal surface displacements, optical methods involving optical interference and fluorescence techniques [78-80], electrical capacitance methods [10, 11, 15, 35, 71, 76], and electrical resistance methods [9, 72, 81]. Static and dynamic extrusion of elastomeric seals was studied by Reddy and Nau [82].

Leakage measurements have been performed with several methods. The most basic of those involves the removal and weighing of the leaked oil from piston rods [7, 8, 10, 13, 76, 83, 84], which can lead to accurate results if executed with care to make sure no oil layer remains on the rod. Another method used for measuring leakage is by maintaining a constant sealed pressure and measuring the oil flow needed to achieve this [10, 76, 85, 86]. Electrical methods have also been used, e.g. in [67], consisting of measuring the electrical capacitance of leaked oil film layers with one or two electrodes.

Friction measurements of reciprocating polymeric seals have been performed by most of the researchers involved in experimental sealing research. The pioneering, fundamental studies discussed in the previous sub-section are a good starting point. Many different apparatuses and test rigs have been used over the years, which makes discussion on their details difficult and of little point. Interested readers can find information on those methods (up to the early 1990s) in [64-66].

The early experimental studies up to and including the 1970s, as discussed in the previous sub-section, provided valuable information on the behaviour of reciprocating polymeric seals and set up the basic methodologies to measure seal performance. Furthermore, experimental techniques developed or improved by the pioneers in sealing research offered a foundation for later studies to build upon and improve the methods and apparatuses to achieve better precision. Most of the important parameters affecting the performance of hydraulic seals had been established in the first 30 years of research. What remained to be done was to extend the range of operating conditions in experimental studies and to apply emerging, high-precision techniques, focusing on interfacial phenomena of the micro-scale. For example, using a camera and video to record the lubrication of the sealing interface in real time. In this respect, Schrader [87] in the late 1970s (as reported by Kanters [64]) was probably among the first to use a high-speed camera to photograph the contact of a seal sliding on a glass cylinder. A few years later, Kawahara et al. [72] published results using the same method.

In more recent times, Kanzaki et al. [88] used optical interferometry in the sealing contact to study oil film behaviour. Interferometric methods to study fluid film thickness and profile have been reported since at least the 1960s in publications dealing with the contact of polymers, steel and glass, as well as between rubber and glass. In the latter case, which is of importance for the subject of this chapter, the work of Blok and Koens [78], presented in 1965, provided a solution to the problem of poor reflection of rubber surfaces – which is mainly owed to their roughness – by covering the rubber surface with a thin sheet of smooth plastic, aluminised on its outer surface. Details on this method with application to rubber lubrication were published quite early in, for example, [89].

In recent years, Kaneta and co-workers [12] used a mono-chromatic optical interferometry technique to directly observe oil films formed between band-shaped nitrile rubber specimens with "D" or lip-shaped cross section on sinusoidally reciprocating glass. There was no fluid pressure gradient in that system and that gave low contact forces. The rubber had to be specially moulded to be optically smooth, which means that its original roughness was lost. Of particular interest in their study was the measurement of film profiles in dynamic conditions, the average film thickness and the friction variation through a stroke. The differences between pumping and motoring strokes were clear to see as their graphs were for one complete cycle. Publication [12] was rather a modern version of an earlier study [13] of some of the same authors and reached similar conclusions regarding the critical stroke-length-to-contact-width ratio for the development of a stable hydrodynamic film, the importance of the contact pressure gradient on film formation and seal leakage, as well as the importance of the so-called "duty parameter". The latter is a dimensionless quantity, which is useful in interpreting friction data with Stribeck-like curves. It is defined as the product of the oil dynamic viscosity with speed and divided by the product of the average contact pressure with the seal contact width. Kaneta et al. [12] confirmed that the friction characteristics of a seal are controlled by a critical duty parameter. Specifically, if the duty parameter is greater than the critical value, the friction coefficient increases with the duty parameter, whereas when the duty parameter is lower than the critical value, the friction coefficient increases with decreasing duty parameter and friction force maxima appear near the ends of the stroke. Other researchers had used this parameter at least since the 1960s, as for example Müller [8].

With advances on imaging technologies, sealing research is lately focused on phenomena taking place at the sealing interface for a better understanding of seal behaviour. In this respect, the role of surface roughness is examined in view of minimising leakage, friction and wear [90]. Direct observation of a sealing interface during operation under realistic conditions is, naturally, the best approach. A collaborative research project between a University and some major seal manufacturers in England explored this avenue [3, 91]. The 3-year project, which was sponsored by the British Department of Trade and Industry through the Civil Aircraft Research and Demonstration programme, involved experimental and theoretical work on reciprocating elastomeric seals used in linear hydraulic actuators for the control of aircraft landing gear (Fig. 2(a)). The experimental work involved – among other things – the development of a rig for measuring seal friction and monitoring the sealing interface in real time with a microscope.

Figure 7 shows a schematic of the original rig [36, 91]. A rectangular seal is clamped on a vice and a glass plate in contact with the seal is reciprocated on top of it. The contact load on the seal is varied by a weight attached at the end of the slider assembly and two force transducers measure the force exerted on the block holding the seal. The transducers are aligned perpendicularly to each other and can give the friction force variation in directions perpendicular and parallel to the direction of the reciprocation. The rig is equipped with a microscope, light source and video recording equipment. A data logging device and a computer are used to calculate the friction once the gain of the system is calculated after a calibration with a known load.

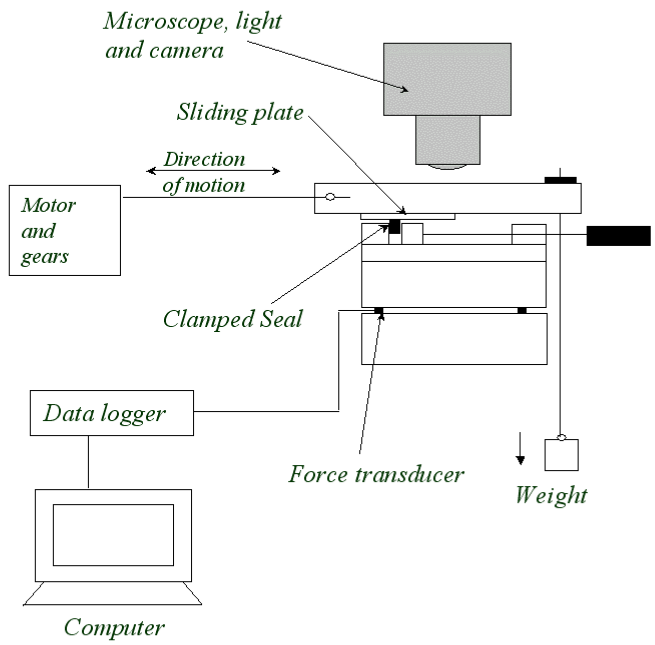


Figure 7. Reciprocating rig for monitoring the sealing interface and performance testing [36, 91]

The stroke length, stroke speed and the load on the rectangular seal can be varied. Moreover, roughness effects can be studied by changing the seal and/or slider with others of different roughness profiles (for example, by substituting a glass plate with a steel plate) and results can be obtained under both static and dynamic conditions, as well as both dry and lubricated conditions with various liquids. The poor reflectivity of rubber can be overcome as was done in [91] by applying a gold sputtering process at high temperature to coat the seals with a 200 nm coating of gold consisting of four 50 nm layers.

Results from friction tests for various reciprocating speeds and with various slider roughness profiles are shown in Fig. 8. Images of seal contacts are shown in Fig. 9 for dry and lubricated (with oil) conditions. Many interesting and useful results are obtained with this configuration such as results on stick-slip phenomena, seal running-in and wear, cavitation phenomena at the edges of the seal after long running times (for example, after 30 minutes or more) and film depletion from the reciprocating motion of the assembly, effects from contamination or wear particles in the sealing contact, hydrodynamic film development and local collapse based on stroking length and speed, etc.

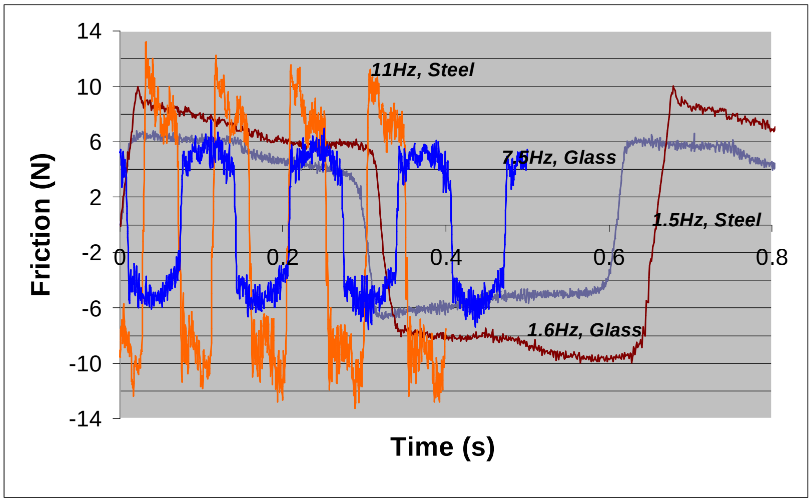
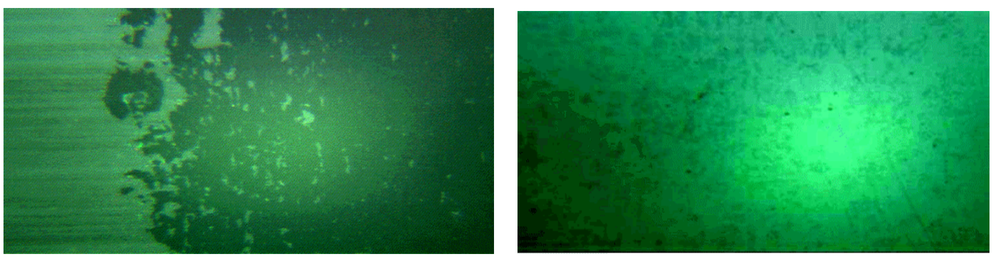


Figure 8. Friction measurements at various reciprocating frequencies and glass-plate roughness from the rig pictured in Fig. 7 (from [36, 91])



(a) Dry conditions (O-ring, 1 mm cross section). **(b)** Wet conditions (dark spots represent contact).
Figure 9. Images of rubber seals in contact with glass plate [36, 91] (from the rig in Fig. 7)

As in previous studies [13, 15], it has been verified that the development of a hydrodynamic film is linked to the stroking length in relation to the seal contact width. When a full hydrodynamic film has not been developed and there are areas of boundary lubrication in the contact, friction rises significantly. This is accentuated in rough contacts. Given that elastomeric seals are normally quite rough with typical average roughness in the order of 1.5 μm [1, 3], it is the roughness of the seal counterface (for example, the piston rod or cylinder bore surface) that matters and can make some difference if prepared before installation. However, as shown experimentally in [36, 91] (and theoretically in, for example, [3]), reducing the average roughness beyond a certain limit offers negligible benefit in reducing friction. Thus, expensive super-finishing of, for example, piston rods, that is, opting for an average roughness of less than about 0.05-0.10 μm in order to reduce friction, is not really necessary.

Advancing the realism of seal testing, an improved rig was built [91, 92] in collaboration with some major seal manufacturers in England. The seals, dimensions and clearances used were typical of linear hydraulic actuators to aerospace design specifications. A schematic of the rig is shown in Fig. 10. A hollow, transparent, high-strength tube is attached to a motor and gear mechanism transferring reciprocating motion. A steel casing hosts gland elastomeric seals and a hydraulic circuit with pump supplies oil under pressure between the casing and the tube. A boroscope with its own light source is placed under one of the seals and feeds its signal to a CCD camera, video recording equipment and computer. Sealed pressures are restricted to about 7 MPa for safety reasons.

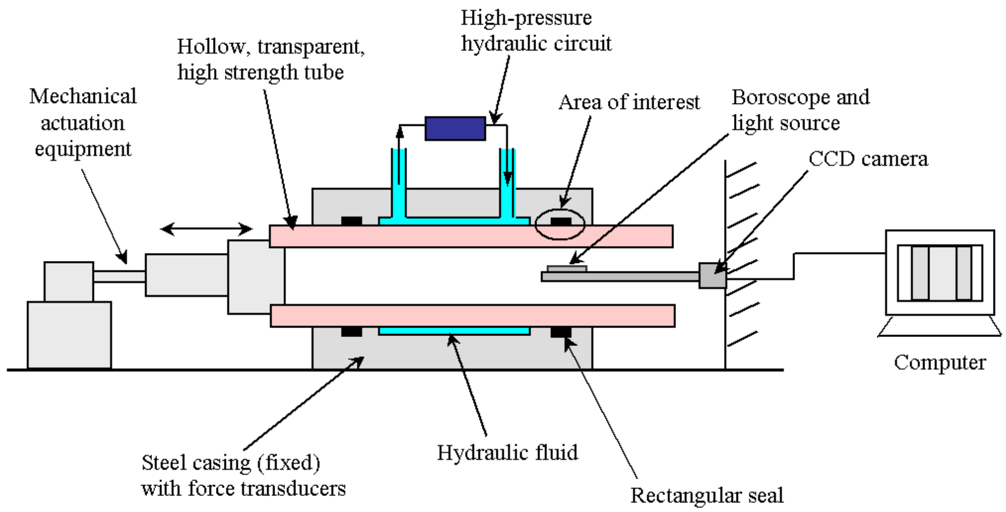
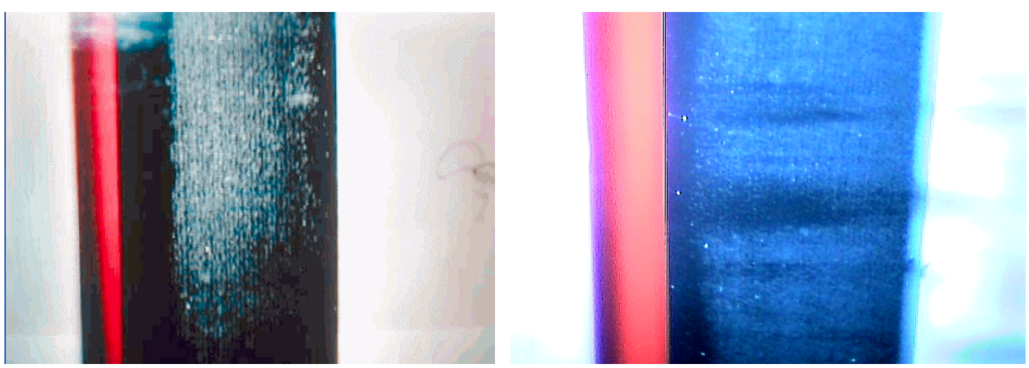


Figure 10. A seal specific rig with glass tube arrangement and boroscope [91, 92]

With a still boroscope, unaffected by vibrations in the system, images captured are clear. Figure 11 shows images from the rig at nearly zero sealed pressure (left image) and at 0.7 MPa sealed pressure (right image). The more fuzzy image on the right is owed to the development of a fluid film under the seal, which varies in thickness as is realised by the different colour shades. This is not the case on the left image of Fig. 11 where the fluid pressure is nearly zero and the film under the seal has partially collapsed.



(a) **(b)**

Figure 11. Sealing contact (sealed fluid on the left side of each image) from the rig in Fig. 10 [91].
(a) Nearly zero sealed pressure (thin film); (b) 0.7 MPa sealed pressure (thick, non-uniform film).

These images can be magnified and stored for later examination. Many results are collected with this arrangement [91, 92], including results on leakage, friction, stick-slip, effects of roughness and sealed pressure, real-time monitoring of the sealing interface and observation of development of a fluid film with roughness asperities deforming dynamically, study of cavitation effects as air bubbles enter the contact, study of the effect of debris particles and how they affect seal leakage, etc.

Seal wear is also studied by measuring the seal roughness before and after the test. Results reported in [91, 92] show a smoothening of the seal surface (running-in), despite sliding on a very smooth counterface (glass tube). Specifically, the average roughness was reduced from 1.8 to 1.1 μm and skewness was reduced from 1.07 to 0.23 μm. Wear appears to take place mostly during the beginning and ending of strokes, which is explained by the thinness of the lubricating film owed to the absence of motion at those intervals, for example, at the reversal of motion. Abrasive wear of the seal is also affected by the presence of contamination particles harder than the seal material. Debris particles are also responsible for increased leakage because they can create micro-channels on the counterfaces for the highly pressurized sealed fluid to escape. The latter was also discovered quite early by White and Denny [7] by creating small grooves onto the surface of a seal. The wear is not confined to the seal though. As the seal is normally much softer than its counterface (for example, a steel surface), debris particles are trapped and, embedded in the seal, they can scratch the harder counterface during the sliding motion.

Research trends on reciprocating seals include the continuous improvement of existing methods and development of novel techniques that are either more accurate or could provide real-time analysis. The latter could be used for condition monitoring. For example, lately, there is interest in ultrasonic methods to measure the film thickness in machine elements, including mechanical seals [93]. Whether these methods are developed to the point they offer clear advantages over existing methods remains to be seen but preliminary results are promising.

5 Theoretical Studies On Reciprocating Seals

Theoretical work on reciprocating seals is focused on solving the contact mechanics and lubrication problems in order to calculate seal leakage, friction and wear. This task has met formidable obstacles since first undertaken more than 60 years ago and a truly accurate solution has yet to be demonstrated. Despite the mathematical equations of the problem being well posed, their numerical solution is far from easy or straightforward. The challenges met can be summarized as follows.

(a) Polymeric seals are objects of complex mechanical behaviour. As explained in section 2, their mechanical properties vary significantly with temperature and they have large thermal expansion coefficient. Elastomeric seals are nearly incompressible, hyperelastic solids, with nonlinear response to stress or strain, profoundly different response near their glass transition temperature, and exhibiting significant relaxation and creep effects. They age, even when not in use, owing to oxidation, and their mechanical performance deteriorates accordingly. They may suffer from swelling from hydraulic fluid absorption, chemically react with incompatible hydraulic fluids, and wear quickly when rubbed on relatively rough, hard surfaces. Similar problems are met in thermoplastic or composite seals such as those made of PTFE (see section 2.2).

(b) The mechanics of polymeric seals is a difficult topic. Suitable models should be used to properly account for their thermo-viscoelastic or thermo-viscoplastic nature. These are models of nonlinear mechanics such as the Mooney-Rivlin model and mechanics of finite deformations, the latter explained by the fact that the maximum normal strain is normally around 10 per cent and sometimes exceeds even 20 per cent.

(c) The contact mechanics of polymeric seals is also a difficult topic. Given the complex shape such seals have, calculating the contact pressure distribution is impossible to do analytically, except for the simplest geometries such as rectangular (and even then, a compromise in precision at the edges of the seal should be accepted). Therefore, advanced numerical methods such as FEA should be used. Moreover, the typical polymeric seal surface is rough. A typical average roughness of elastomeric seals is in the order of 1.5 μm [3]. From a contact mechanics perspective, the proper modelling and accounting of surface roughness is vital in studying interfacial phenomena, which may be crucial in a performance analysis such as in calculating leakage and friction. Modelling highly deformable surface roughness asperities in any contact mechanics model is a difficult task.

(d) The lubrication problem of polymeric seals belongs to the category of "soft EHL". This is among the most computationally demanding elastohydrodynamic problems, even more so given that it involves transient operations of variable speed and sealed pressure. The reason for this difficulty is the high sensitivity of the numerical solution algorithm to small errors in the calculation of the film thickness. If these are not prevented or corrected, the numerical solution becomes unstable very quickly. Moreover, in reciprocating motion, the reversal of motion at the end of strokes posses great numerical difficulties in terms of properly accounting for the local inlet conditions to solve the EHL problem.

(e) The contact mechanics and lubrication problems are coupled. For flexible reciprocating seals, the shear stress at the sealing interface alters the pressure distribution, which, in turn, alters the development of the hydrodynamic film at the inlet zone of the seal, which determines the film thickness at the sealing contact and shear stress. In other words, a computational loop should be used to correctly resolve the contact pressure and film thickness coupling. The difficulty in this case comes from the flexibility of the seal and the fact that the contact pressure distribution is impossible to calculate analytically with high precision. This means that resorting to complex and time-consuming FEA is the only way possible yet this has to be repeated in every iteration until convergence is achieved at a given time step. In a transient analysis with hundreds of steps or more, solution time becomes excessively long or plainly unacceptable. This is the reason why, to the best of the author's knowledge, the coupled problem remains unsolved in the literature to date (2008).

The previously listed theoretical obstacles are the major ones. Details are provided in the next sections to help readers understand the specific problems, solution methods, and the potential for future research.

5.1 Phenomenological models of polymeric seal materials

Polymeric materials for reciprocating seals are either elastomeric (rubber compounds) or plastic (for example, PTFE). Composite materials are also used, for example, particle-reinforced rubbers and PTFE with glass fibres. The mechanics of thermoplastics such as

PTFE or polyurethane, as well as that of composite materials is a complex and specialised topic – see for example [51]. Interested readers can find information on this topic in the volumes accompanying most of commercial FEA software. The mechanics of elastomeric materials on the other hand is a much older topic and of greater interest as most hydraulic seals are elastomeric. Thus, the discussion is confined to the mechanics of elastomeric seals in this section.

Elastomeric materials are typically rubber compounds, that is, substances for which vulcanised natural rubber is the prototype. (Synthetic rubbers are also produced with sulphur or other additives.) They are also referred to as "rubber-like materials" [59]. Typically, these are hyperelastic and nearly incompressible. The hyperelasticity is exhibited in sustaining large strains without fracture and recovering to initial dimensions when stresses or strains are removed, without appreciable hysteresis at temperatures above the glass transition temperature. According to the statistical-molecular or network theory of rubber elasticity [94], they consist of very long molecular chains, which are folded and kinked [43]. The chains are chemically cross-linked, forming a three-dimensional network. Free space exists between chains, which varies transiently in volume and location. As atoms in said chains are thermally agitated, they assume a variety of statistically determined conformations [95]. This dynamic or transient chain motion is slowed down when the material is cooled and eventually ceases when the material is close to its glass transition temperature (usually between zero and –70 °C, depending on the particular elastomer). This explains the extensibility of elastomers at temperatures well above the glass transition temperature. It also explains their rigidity at subzero temperatures, especially near the glass transition temperature where the material behaves like brittle glass with significantly altered crystallinity.

There are many papers and phenomenological models on rubber thermoelasticity in the literature. Some classical, detailed studies are [59, 95-97]. Treloar's classic paper [95] in 1976 and Ogden's review [98] in 1986 give a good introduction to phenomenological models at the time of their publication and are still useful today. A thorough presentation can be found in a book published in 2000 by Holzapfel [99]. For reciprocating hydraulic seals specifically, a satisfactory constitutive model is the so-called Mooney-Rivlin model.

The Mooney-Rivlin model is based on the pioneering work of Mooney [100] and Rivlin [101] on finite elasticity in the 1940s. Its derivation is based on the elastic strain energy per unit volume, W, which is a function of the three principal stretches of deformation or extension ratios [98], that is, $W = W(\lambda_1, \lambda_2, \lambda_3)$, where $\lambda_i \equiv l_i / L_i$ (i = 1, 2, 3), l_i and L_i being deformed and reference length, respectively. It is assumed that the mechanical properties of rubber-like solids can be represented in terms of the energy function. Assuming isotropic solids and isothermal conditions, the energy must be independent of the coordinate system used (isotropy). Thus, it can be expressed in terms of the three strain invariants $I_1 = \lambda_1^2 + \lambda_2^2 + \lambda_3^2$, $I_2 = (\lambda_1\lambda_2)^2 + (\lambda_2\lambda_3)^2 + (\lambda_3\lambda_1)^2$ and $I_3 = (\lambda_1\lambda_2\lambda_3)^2$, that is, $W = W(I_1, I_2, I_3)$. For an incompressible solid such as an elastomeric seal (which is nearly incompressible with typical Poisson's ratio of 0.499), the volume is constant, which means that $\lambda_1\lambda_2\lambda_3 = 1$, hence $I_3 = 1$. Many explicit forms of function W have been reported in the literature [98] but the Mooney-Rivlin function has been extensively used; it can be expressed as

$$W = \frac{c_1}{2}(I_1 - 3) + \frac{c_2}{2}(I_2 - 3) \tag{1}$$

where coefficients c_1 and c_2 are obtained experimentally at the temperature of interest. According to [95], this is the most general first-order expression in I_1 and I_2.

Given the strain energy function W, the Cauchy (true) principal stresses σ_i can be calculated from (see Eq. (6.69) in [99])

$$\sigma_i = \lambda_i \frac{\partial W}{\partial \lambda_i} - p_c \quad (i = 1, 2, 3) \tag{2}$$

where p_c is a hydrostatic pressure, which can be determined from equilibrium equations and boundary conditions. Using the Mooney-Rivlin function (Eq. (1)), Eq. (2) yields

$$\sigma_i = \lambda_i^2 \left[c_1 + c_2\left(\lambda_j^2 + \lambda_k^2\right)\right] - p_c \quad (i, j, k = 1, 2, 3 \text{ and } i \neq j \neq k \neq i) \tag{3}$$

Coefficients c_1 and c_2 in Eq. (3) are calculated from stress-strain data in uniaxial tension or compression of the rubber-like material such as in Fig. 3. In uniaxial tension or compression, assuming that “1” is the load direction, $\sigma_2 = \sigma_3 = 0$ and $\lambda_2 = \lambda_3$. Setting $\sigma_1 \equiv \sigma$ and $\lambda_1 \equiv \lambda$, pressure p_c is calculated from Eq. (3) using either $\sigma_2 = 0$ or $\sigma_3 = 0$ and utilizing the incompressibility constraint $\lambda_1\lambda_2\lambda_3 = 1$. The result is $p_c = \left[c_1 + c_2\left(\lambda^2 + 1/\lambda\right)\right]/\lambda$. Then, Eq. (3) yields

$$\sigma = \lambda\left(c_1 + \frac{c_2}{\lambda}\right)\left(\lambda - \frac{1}{\lambda^2}\right) \tag{4}$$

The engineering (or nominal) stress, σ_{eng}, is equal to σ/λ. Thus, using Eq. (4),

$$\frac{\sigma_{\text{eng}}}{\lambda - \lambda^{-2}} = c_1 + \frac{1}{\lambda}c_2 \tag{5}$$

Equation (5) represents a straight line. Using engineering test data (σ_{eng}, λ) and plotting them as “reduced pressure” $\sigma_{\text{eng}}/\left(\lambda - \lambda^{-2}\right)$ versus the inverse of stretch, $1/\lambda$, coefficients c_1 and c_2 are easily calculated. If a stress-strain curve is not available and only the modulus of elasticity, E, is known, then, according to reference [102] (page 7-33), a reasonable approximation to use is $c_1 \cong 4c_2$ and $3(c_1 + c_2) \cong E$, resulting in $c_1 \cong 4E/15$ and $c_2 \cong E/15$.

According to [43], the Mooney-Rivlin model has been used for strains up to 200 per cent. This means that it is adequate for the problem of reciprocating hydraulic seals, where the maximum normal strain rarely exceeds 25 per cent and is more typically less than 15 per cent. Owing to its simplicity and satisfactory precision, the Mooney-Rivlin model has been adopted in many studies in the literature. Though it only requires calculating two constants, it can be further simplified to what is known as the neo-Hookean model. The strain energy function for the latter is $W = c(I_1 - 3)$ (c being a constant). The neo-Hookean model may be simpler and easier to apply, however, it is much less accurate than the Mooney-Rivlin model at higher strains. Therefore, if precision is a priority, the Mooney-Rivlin or a better model should be used.

Other phenomenological models have also been developed. According to Holzapfel [99], the (more complex) Ogden model [103] "*excellently replicates the finite strain behaviour of rubber-like materials*". However, it should be remembered that the differences between the various models become significant only at larger values of strain, which are normally not met in hydraulic reciprocating seals. Moreover, these models work under isothermal conditions. Some approximations in engineering calculations can be done when there is temperature variation, as demonstrated by Nikas and Sayles [5, 6] in the case of reciprocating rod seals for linear hydraulic actuators (Fig. 2(a)). Models have also been developed for compressible materials [99], though they are more complex and probably not justified in the case of reciprocating seals because of the relatively low strains involved.

Further complications in the phenomenological modelling of rubber-like materials comes from inelastic effects such as hysteresis, frequency-dependent response, strain-stiffening at large stretch, scission of molecular cross-links at high temperature leading to time-dependent softening or permanent set, and stress-softening, the latter known as the Mullins effect. (The Mullins effect is observed in cyclic loading during the first and successive cycles at given strain when the stress drops, hence the term "stress-softening". This effect is important in engineering elastomeric parts. As it is related to fatigue, it clearly has implications in the life expectancy of elastomeric parts.) The aforementioned effects are more evident in rubbers hardened with fillers or particle-reinforced elastomers and there are several models developed in recent years to estimate elastomer behaviour in such cases – see for example [99, 104-108].

For the purposes of calculating the performance of hydraulic reciprocating seals, even the classic Hookean (linear elasticity) model with allowance of thermal effects may be adequate if the maximum normal strain does not exceed about 10 per cent [5, 6]. For strains larger than about 10 per cent and up to 15 per cent, Nikas and Sayles [5, 6] reported that a more advanced model should be used. Comparing the Hookean and the Mooney-Rivlin model predictions on rod seal leakage, hydrodynamic friction and extrusion predictions at temperatures of –54, +23 and +135 °C, and sealed pressure of up to 35 MPa, Nikas and Sayles [5, 6] reported a maximum difference in leakage of about 15 per cent at the highest temperature and high sealed pressure (of 25 MPa), although the differences are usually between zero and 5 per cent.

5.2 Contact mechanics of hydraulic reciprocating seals

The contact mechanics of hydraulic reciprocating seals involves the calculation of the contact pressure and tangential traction at a sealing contact. The latter is about calculating the shear stress or friction in the contact. The contact mechanics may also involve calculating the stress field in the body of a seal in order to locate zones of stress concentration and to link those with fatigue modelling or, simply, as a guide to make improvements in design in order to eliminate stress concentrations. Finally, the contact mechanics may also involve calculating the overall deformation and change of shape of the seal in dynamic conditions in order to establish potential performance issues such as extrusion, which may be of concern.

The aforementioned computational tasks are all complicated because the solid mechanics of polymeric seals is highly nonlinear and the typical boundary conditions in reciprocating sealing are complex and transient. Precise analytical solutions are not feasible. Analytical solutions can only be applied on the simplest of seal geometries such as rectangular and only approximately. Such analytical solutions on the static contact pressure assuming plane-strain conditions have been presented by Hooke et al. [29, 109] in 1966, Johannesson [20] in 1979, and Dragoni and Strozzi [110] in 1988 on O-ring rubber seals, Field and Nau [17] in 1975 on rectangular rubber seals, Strozzi [73] in 1986 on rectangular rounded seals, Johannesson and Kassfeldt [111] in 1989 on seals of arbitrary cross-section, and more recently by Nikas [1-6, 37-42] on rectangular rounded elastomeric seals and rotary vane seals, including composite (PTFE-elastomer) seals [2, 4, 42]. The contact pressure in most of those studies was calculated from the amount of surface overlap or interference and contact friction was usually neglected.

For complex seal geometries and boundary conditions, for example those pictured in figures 4 and 5, numerical solution is the only viable option. This means that FEA should be used whenever possible, though care should be taken to use finite elements formulated for incompressible materials to avoid meaningless results. FEA has been applied mainly since the 1970s in the study of hydraulic seals – see for example references [32, 64, 73, 75, 85, 112-121]. A bibliographic review of FEA of rubber-like materials, covering the period 1976-1997, can be found in reference [122]. Figure 12 shows an example of FEA analysis of a rubber O-ring, pressed between two frictionless plates.

5.3 Elastohydrodynamics and performance of hydraulic reciprocating seals

Hydraulic reciprocating seals typically operate with a thin lubricating film in their sealing contact (see for example figures 9 and 11). The film separates the seal from its counterface, as for example in the simplified schematic in Fig. 13, depicting a rod seal in a linear hydraulic actuator. The fluid film thickness varies from a few nanometres to a few micrometres. This form of lubrication is characterised as elastohydrodynamic (EHL). The fluid film is developed when lubricant enters the sealing contact by viscous shear and/or at high sealed pressure. The thickness of this film depends on many factors such as the stroking velocity, contact pressure distribution, surface roughness, lubricant viscosity and density at operating conditions (pressure, temperature), inlet geometry, degree of lubricant starvation, etc. As the main seal performance variables, namely the leakage rate and friction, both depend on the thickness of the sealing contact fluid film, the precise calculation of the film thickness is of major importance.

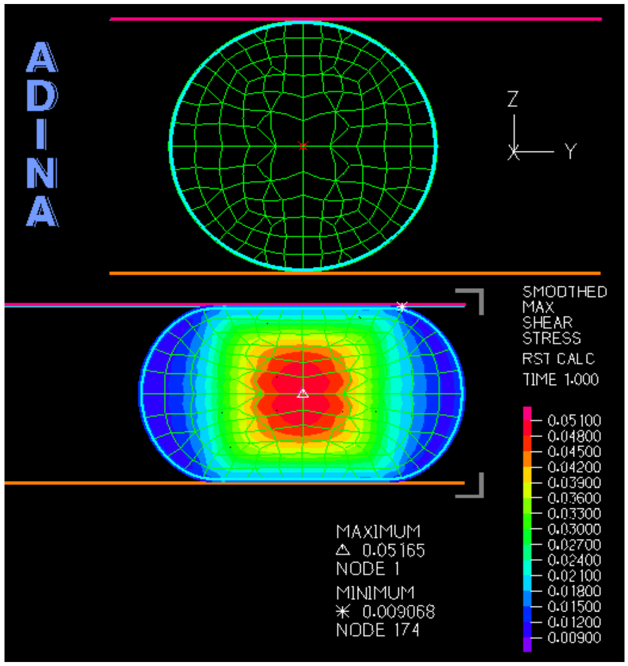


Figure 12. FEA example of a rubber O-ring pressed between two rigid, frictionless plates; original and deformed mesh shown with maximum shear stress distribution

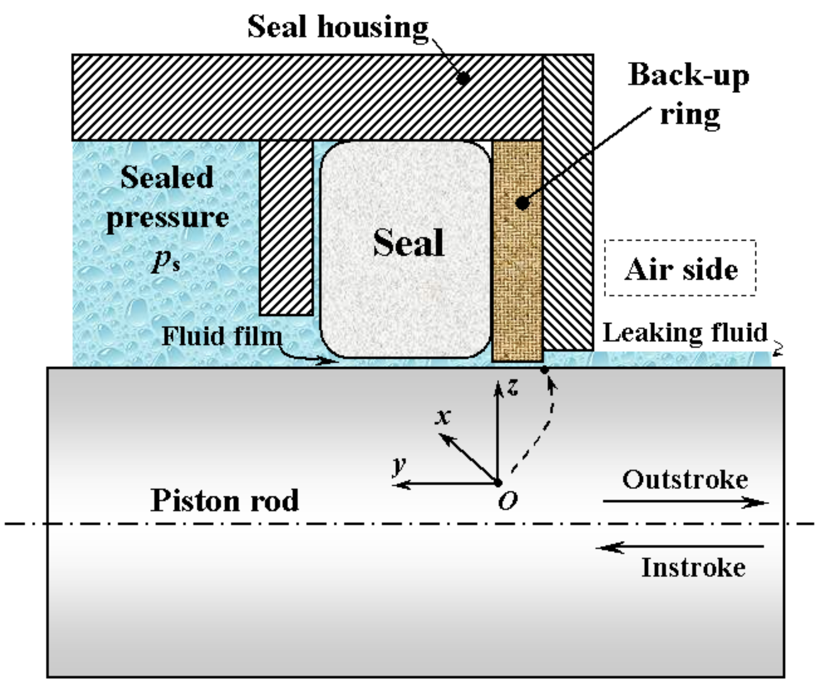


Figure 13. Schematic of rectangular elastomeric seal with back-up ring in a linear hydraulic actuator (only the upper half of the seal, ring and housing shown)

Calculating the film thickness distribution in the sealing contact is far from straightforward and simple. The calculation is based on a number of simplifying assumptions.

For example, fluid inertia is neglected because the film is very thin; frictional heating of the contact and fluid can be neglected when the stroking velocity is low, as for example in typical hydraulic actuators controlling aircraft landing gear; surface roughness effects can be neglected as a first approximation or because the low-stiffness seal roughness asperities are flattened under pressure. With such reasoning, several simplifications can be made prior to writing down a suitable equation to calculate the film thickness distribution. The aforementioned equation is derived from the Navier-Stokes equations of Fluid Mechanics after several simplifications and is the well-known Reynolds' equation [123] in a form that suits the particular application with assumptions accepted.

As fluid flow in the sealing contact under reciprocating motion is mainly one-dimensional, a suitable form of the Reynolds' equation is as follows [4]:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right) = 6V\frac{\partial(\rho h)}{\partial x} + 12\underbrace{\frac{\partial(\rho h)}{\partial t}}_{\text{Transient term}} \tag{6}$$

where p is local contact pressure, h is local film thickness, V is the sum of the tangential velocities of the cooperating surfaces of the contact, t stands for time, and ρ and η are the mass density and dynamic viscosity of the lubricating fluid, respectively, which are functions of temperature, pressure and shear rate [42], although the latter is neglected because it is weak in the low-speed, low pressure applications met in reciprocating sealing. For low accelerations/decelerations or, generally, as an approximation, the last "transient term" in Eq. (6) is omitted and Eq. (6) then yields

$$\frac{\mathrm{d}}{\mathrm{d}x}\left(\frac{\rho h^3}{\eta}\frac{\mathrm{d}p}{\mathrm{d}x} - 6V\rho h\right) = 0 \tag{7}$$

which, obviously, holds for steady-state conditions, that is, operation with constant speed and sealed pressure. Almost all studies in the literature of reciprocating seals deal with the one-dimensional form of the Reynolds equation and the vast majority assume that the sealing contact is perfectly smooth. If surface roughness effects are accounted for, micro-EHL effects among neighbouring roughness asperities play some role in performance. Therefore, a more general, two-dimensional form of the Reynolds equation can be used, as was done in [1, 3, 5, 38-40] for rectangular rod seals.

The Reynolds equation may be accompanied by kinematical and boundary conditions. For example: (a) the no-slip boundary condition, dictating that the fluid velocity on a boundary wall is equal to the wall's velocity (zero slip); (b) the inlet condition: $\mathrm{d}p/\mathrm{d}x = 0$ "away" from the contact in the inlet zone, etc. For a detailed presentation of these topics, please refer to the literature, for example [1-6, 16-32, 38-42].

Once the mathematical problem is well-posed, the Reynolds equation can be solved numerically for the film thickness if the contact pressure distribution is known, or for the contact pressure if the film thickness distribution is known, or for both, in conjunction with contact mechanics equations, as explained in the previous section. Given the highly nonlinear nature of this problem, a successful solution is obtained only by using a convergence loop in an algorithm involving the lubrication and contact mechanics equations. Several solution methods have been published in the literature over a period of decades, varying in realism, sophistication and precision. Among the earliest studies are those of White and Denny [7], who assumed a tapered film profile and parabolic pressure distribution in deriving the film thickness, and that of Müller [8], who also used a tapered film profile, different between outstrokes and instrokes, and measured contact pressure distributions.

If the pressure distribution is somehow known (either measured or empirically assumed), Reynolds' Eq. (7) can be easily inverted to a cubic algebraic equation and solved for the film thickness. This is the so-called inverse solution of the Reynolds equation, which is usually attributed to Blok [124]. For details of this method in general, see reference [123]. In the study of reciprocating hydraulic seals the method has been applied by most researchers [16, 18, 19, 21, 22, 24-27, 64, 76, 77, 125, 126]. Despite its relative simplicity, the method is not without problems because it involves the tricky part of calculating the roots of a cubic equation and doing so for many points in a sealing contact. This is not an easy task from a numerical perspective and erroneous, imaginary roots can destabilise the solution process. The difficulties and pitfalls associated with this method in sealing have been detailed by Ruskell [23]; an interesting discussion can also be found in reference [26].

A modification of the method that avoids the cubic equation and solves a first-order, ordinary differential equation instead was postulated in reference [41] and further applied in references [2, 4, 42] on rectangular rounded reciprocating seals, including composite and rotary vane seals. According to the latter development, the equation to solve is [41]

$$\frac{\mathrm{d}H}{\mathrm{d}x} = \frac{H^3 \dfrac{\mathrm{d}^2 q}{\mathrm{d}x^2}}{6V - 3H^2 \dfrac{\mathrm{d}q}{\mathrm{d}x}} \tag{8}$$

where $H \equiv \rho h$ and $\mathrm{d}q/\mathrm{d}x = (dp/dx)/(\eta\rho^2)$. In order to solve Eq. (8) for H, a boundary condition is needed, that is, H must be known at any one point in the sealing contact. Such a suitable point is the extremum point (say $x = x_\mathrm{m}$) of q, where $\mathrm{d}q(x_\mathrm{m})/\mathrm{d}x = 0$ and, by the definition of q, $\mathrm{d}p(x_\mathrm{m})/\mathrm{d}x = 0$. With a known pressure distribution, the extremum point is easily located. Then, following the analysis of [41], the required boundary condition, which is the value of H at the extremum point of q, is calculated from

$$H_\mathrm{m} = \frac{2}{3}\rho_\alpha \left| \frac{2V\eta_\alpha}{\dfrac{dp_\mathrm{in}\left(x_\mathrm{in}^{(\alpha)}\right)}{dx_\mathrm{in}}} \right|^{1/2} \tag{9}$$

where index "α" refers to the inflexion point of q, p_in is the EHL inlet film pressure, and $x_\mathrm{in}^{(\alpha)}$ is the distance between the inflexion point and the nearest edge of the "dry" contact zone.

As is realised from Eq. (9), in order to calculate the film thickness with the inverse hydrodynamic theory, it is necessary to locate the inflexion point of a curve related to the contact pressure distribution. The correct location of the latter is crucial in the precise computation of the boundary condition to solve the film thickness equation and small errors lead to irrelevant results in terms of seal leakage and friction. Moreover, the role of the pressure gradient at the inflexion point (see denominator in Eq. (9)) on sealing performance is now revealed: the higher the gradient, the thinner the average film in the contact. This was quite early established in the literature and comprises the “secret” of minimising leakage, though, unfortunately, this increases hydrodynamic friction because thinner films result in more viscous drag [2, 4]. Thus, the inlet geometry of the seal and resulting local pressure distribution in that critical area is of paramount importance, a fact taken advantage of in modern seal designs with optimised profiles to suit even dynamic operating conditions.

The modified inverse hydrodynamic theory expressed via equations (8) and (9) has also been used with the more general transient Reynolds Eq. (6) as in reference [2]. The results produced by this and similar techniques are realistic but most published studies are based on static contact pressure distributions, which are realistic only at very low speed and for well supported seals such as rectangular seals supported by back-up rings on both sides. In cases where seals are allowed to move in their housing, even by a small amount (for example, O-rings), the inlet contact pressure has a dynamic variation and so does sealing performance. This is explained by the friction in the sealing contact, which deforms the seal, changing the inlet geometry dynamically. The problem is clearly coupled and solutions based on de-coupling it are more or less inaccurate when applied to dynamic sealing conditions. Unfortunately, to the best of the author’s knowledge at the time of writing (2008), no published study has tackled the coupled problem. The reasons include computational complexity and time restrictions. Ideally, the problem must involve FEA for the contact mechanics of the seal combined with computational fluid dynamics or, equivalently, FEA with fluid-structure interactions. Such tasks require a high degree of sophistication and are time consuming, being rather unsuitable for parametric analyses involving hundreds of computer software executions.

Alternative methods have been applied by several researchers, though still not tackling the coupled problem in dynamic conditions. The said methods solve the Reynolds equation together with an elasticity equation for the seal, either simultaneously or serially in a number of iterations. Such “direct” methodologies were adopted in [1, 17, 76, 127]. In one of the earliest such studies, Field and Nau [17] in 1975 developed an elasticity equation from simple compression of a rectangular rod seal in smooth contact, including internal shear stresses in the material. After a lot of effort, they obtained results for outstrokes but failed to produce results for instrokes due to numerical instabilities. This is not surprising at all. The Reynolds equation is very sensitive to even very small errors in the film thickness (of sub-micrometre order of magnitude). When such errors are not eliminated in a numerical iteration scheme, they quickly destabilise the numerical procedure. This was reflected on the results [17] with wavy curves, which are indicative of numerical instability. Similar problems have been reported by Swales et al. [128].

Nikas [1, 3, 5, 38-40] also applied the direct solution method but extended this to the two-dimensional Reynolds equation [1] on rough contacts. He calculated a static contact pressure distribution for the rough contact with a columnar stress model but separated the pressure perturbations induced by roughness asperities in the numerical analysis, which are very weak in comparison with the bulk contact pressure, in order to achieve and accelerate convergence. Typical results from his theoretical analysis on film thickness are shown in Fig. 14 for the rough contact of a rod seal and a piston rod. His approach was based on the solution of the Reynolds equation with a Successive Overrelaxation (SOR) method but that also encountered numerical problems.

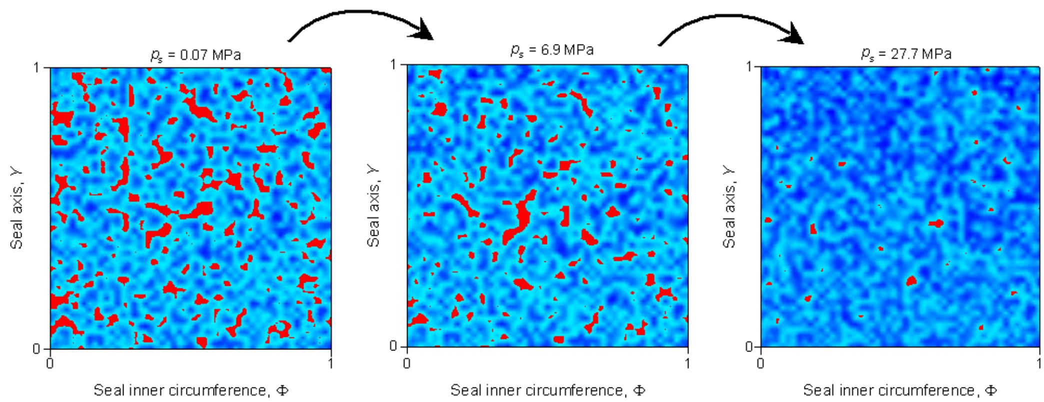


Figure 14. Film contour maps from the rough contact simulation of rod seal on piston rod. Lighter shaded areas indicate thinner film and red are areas of solid contact. Notice the partial film collapse at low sealed pressure p_s (left image)

A better yet much more complicated method was presented by Ruskell [32] in 1980. Ruskell's method overcame convergence problems by incorporating the Reynolds and seal elasticity equations into a single equation, which was solved numerically in a few iterations, usually up to six. His elasticity equation though was based on contact pressure distribution pre-calculated via FEA on a static, frictionless contact. This is of course not the real coupled EHL-mechanics problem but his method is still an efficient approach to a difficult problem. A similar approach was adopted by Prati and Strozzi [77]. A more sophisticated method, incorporating surface roughness and inter-asperity cavitation effects, was published in 2007 by Salant and co-workers [121, 129] for steady-state conditions of a reciprocating rod seal. The authors reported realistic results and emphasised the role of surface roughness in sealing performance.

Apart from the methods already described, methods developed for solving the EHL problem of soft solids in general have been published since the 1960s. The work of Hooke [130-132] is of particular interest to elastomeric reciprocating seals, as it deals with soft contacts under conditions of reciprocating motion. Though Hooke's approach is approximate in dealing with the EHL inlet and exit zones, it is mathematically sound and provides useful results on transient motion. Of great interest are his results on the issue of the reversal of motion, showing the expected film thinning at the edges of the contact, where most of seal wear takes place. Indeed, the precise numerical treatment of the reversal of motion in reciprocating seals is a complicated issue and remains challenging to date. A further analysis of this issue, though not confined to soft solids, was also presented by Hooke for line contacts using the one-dimensional Reynolds equation [133].

In order to avoid the numerical complexities in solving the transient Reynolds equation (Eq. (6)), Chang [134] proposed a simple and validated method, based on the solution of a first-order partial differential equation of one-dimensional wave propagation. This is essentially derived from Eq. (6) by setting the left-hand side term equal to zero and is sometimes referred to in the literature as the “reduced” Reynolds equation. The method provides a practical alternative to avoiding a full numerical solution, though it may be suitable only for well-supported seals, without significant flexing during the reversal of motion. Ikeuchi et al. [135] have also published a simplified method to solve the transient problem in the contact of compliant solids, including dynamic seals.

Further insight into the issue of transient lubrication of reciprocating seals was offered quite early by the theoretical simulations of Hirano and Kaneta [22, 24]. They studied the development and potential collapse of EHL films with pre-determined contact pressure distributions such as parabolic and Gaussian [22]. Their theoretical results confirmed their experimental results [13] on the criticality of the stroking-length-to-contact-width ratio in establishing a full elastohydrodynamic film, which must be at least 2. This is physically explained by considering the speed of wave propagation of the fluid from the contact inlet to the outlet, which is half the speed of the moving surface in the contact. Therefore, relatively short strokes result in partially collapsed films and increased wear of the seals. These results have been confirmed in several other studies. However, as Hirano and Kaneta pointed out [13], the film collapse (in the exact sense) is theoretical because it neglects the micro-EHL taking place between surface roughness asperities of the counterfaces in the contact, as for example the asperities of a rod seal and a piston rod.

Micro-EHL is indeed an efficient method of wear reduction and can be analysed on roughness scale, although its theoretical treatment and numerical analysis are both complex [136, 137]. In the author’s experience (e.g. [41]), a proper treatment of roughness in elastomeric contacts should involve aspects such as inter-asperity cavitation, viscoelastic effects to explain stick-slip micro-scale phenomena such as Schallamach waves [45], and asperity adhesive forces [138, 139]. The latter, related to attractive molecular forces at the sealing interface among closely engaged roughness asperities, affects friction during start-up of motion, when the material is relaxed and the fluid film at the sealing interface is partially collapsed. Moreover, as friction is related to wear, proper roughness modelling will be beneficial in predicting seal wear more accurately. However, the topic of abrasive wear of elastomers rubbed on hard surfaces is vast and cannot be adequately discussed within the confines of this chapter. Interested readers are directed to references [49, 140, 141] for an introduction to the subject and, particularly to Zhang’s book [45].

The real necessity of roughness modelling may be questionable considering that polymeric seals, when rubbed against hard, rough surfaces, can deposit a thin layer of polymer on the hard surface, filling up valleys between asperities of the hard surface [142, 143]. This is well known for PTFE for example and would effectively reduce the average roughness of the hard surface, giving some justification for a smooth-contact model. Furthermore, as already discussed in section 4.2, experimental results reported in the literature, for example [91, 92], show smoothening of the seal surface during running-in, even when sliding on very smooth counterface such as glass. Thus, it would appear that roughness modelling is useful as a theoretical improvement, with potentially more accurate friction predictions, but it is doubtful that it is absolutely critical to the seal designer. In well lubricated elastomeric contacts, the largest portion of friction comes from viscous shear and that can be realistically predicted with existing models of smooth EHL [41]. In contacts starved of lubricant, predictions are also realistic with a smooth-surface model [42], provided it is capable of calculating very small film thicknesses. Roughness modelling can thus be understood in terms of refining and fine-tuning an existing theoretical model.

5.4 Performance of hydraulic reciprocating seals and related issues

The main variables to consider when evaluating the performance of reciprocating seals are leakage, friction and extrusion. Leakage is a measure of how efficient sealing is. Friction is related to the resistance to motion and the associated power loss in the system. Finally, extrusion is related to the risk of damage to the seals from stress concentration and damage to their edges.

Leakage refers to the rate of mass of liquid passing through the sealing contact in the direction of reciprocation. As the fluid continuity equation (mass conservation) must be satisfied everywhere in the contact, leakage can be calculated at any one point. However, avoiding an arbitrary point selection, it is, generally, better to calculate the average leakage in the contact. Thus, in the case of the one-dimensional Reynolds equation (Eq. (6)), the mass leakage rate Q is [41]

$$Q = \frac{L}{W}\int_0^W \left(\frac{Vh}{2} - \frac{h^3}{12\eta}\frac{\mathrm{d}p}{\mathrm{d}x}\right)\rho\,\mathrm{d}x \tag{10}$$

where W is the contact width in the direction of reciprocation and L is the contact size in the transverse direction. In a transient analysis, some variables in Eq. (10) are, naturally, functions of time, for example, speed V and local film thickness h. In the case of the two-dimensional Reynolds equation, see for example [1, 38] for the analysis of rod seals.

The frictional force on a reciprocating seal consists of a hydrodynamic component and other components related to adhesive or molecular forces such as van der Waals forces between closely engaged roughness asperities of the counterfaces in a sealing contact. In practise, the hydrodynamic force, which is greatest in thick-film lubricating conditions, is, almost always, the only one calculated, especially when the model refers to smooth instead of rough EHL. The hydrodynamic frictional force on the seal, F, is calculated by integrating the local viscous shear stress on the seal surface in the sealing contact. In the case of the one-dimensional Reynolds equation (Eq. (6)), the hydrodynamic frictional force is [41]

$$F = L\int_0^W \left(\frac{\eta V}{h} - \frac{h}{2}\frac{\mathrm{d}p}{\mathrm{d}x}\right)\mathrm{d}x \tag{11}$$

In the case of the two-dimensional Reynolds equation, see for example [1, 38] for the analysis of rod seals. For other sources of friction, refer to [138, 139].

Apart from leakage and friction, reciprocating seals often suffer from extrusion damage. For example, in the case of rod seals in linear hydraulic actuators without back-up rings, the seal may get squeezed into the clearance between its housing and the piston rod as in Fig. 15. The localised strain in the form of the extruded part is a zone of stress concentration and is readily affected by the sealed pressure and friction on the sealing contact. This localised deformation can lead to a cut or abrasion of the seal after a number of operating cycles, causing a premature end to the service life of the seal.

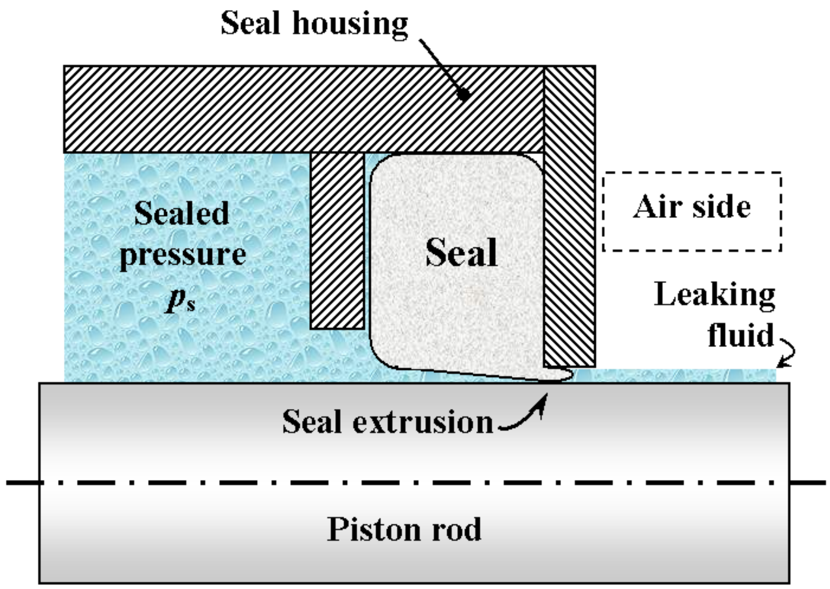


Figure 15. Extrusion of a rectangular, rounded, elastomeric, rod seal (only the upper half of the seal and housing shown)

The phenomenon was discussed quite early by White and Denny [7]. Nikas [37] analysed seal extrusion mathematically and produced algebraic equations giving the length, shape and contact pressure on the extruded part as a function of the operating conditions and corner geometry of the seal. It was thus demonstrated how to minimise extrusion. The results showed that extrusion is unavoidable for seals with sharp corners but it was not significantly influenced by the stroking velocity and the viscosity of the sealed fluid. Moreover, it was shown that extrusion rises with the sealed pressure and the rod-housing clearance; it is reduced with the seal width, the seal corner radius and the seal elastic modulus. As a result of those findings, it was suggested that the two most obvious solutions to minimise seal extrusion are (a) to use seals with rounded corners or chamfered ends and (b) to use anti-extrusion or back-up rings.

Back-up rings (Fig. 13) are devices resembling washers. Their main task is to support elastomeric seals in high-pressure hydraulic systems, preventing seal gap extrusion. Thus, they are also known as anti-extrusion rings. They are also used to prevent seal roll deformation [144], which can cause total seal failure. As they normally affect seal deformation during operation, they should be accounted for when modelling seal performance. This is even more crucial if the rings are in contact with the reciprocating surface (e.g. a piston rod) and, thus, perform some sealing action. A mathematical analysis of back-up rings for rod seals was developed by Nikas [39], calculating their effects on leakage and friction of the seals for operating temperatures between –54 and +135 °C, and sealed pressures between 1 and 35 MPa. It was found that the contact pressure and average surface roughness of a back-up ring can be optimised to minimise the leakage-per-cycle of the system. It was also found that there exist a critical sealed pressure over which a back-up ring can become a more effective sealing element than the seal it supports. The effect of operating parameters on those results was also explored and several other conclusions were postulated in what was a computationally complex analysis. However, it should be emphasized that such results cannot be generalised because they depend on many factors such as the exact geometry of the solids involved (seal and rings), their dimensions and the operating conditions.

Seal extrusion can also be reduced by using composite seals. For example, consider the seal shown in Fig. 16. It has a centrally placed elastomeric part bonded to two outer PTFE parts. The depicted seal is actually just the horizontal part of a goalpost-shaped vane seal developed for rotary vane actuators [2, 4] and operating in reciprocating motion. The PTFE (with glass fibres) gives the seal the needed rigidity at the edges to reduce extrusion and allows the corners to be sharp, minimising leakage. A theoretical analysis of this type of composite, rectangular, reciprocating seal was presented in [42] with an extensive parametric study of sealing performance and how it can be optimised in a broad range of temperatures. Results of leakage, friction and seal extrusion were presented on both starved and flooded contacts, and an optimum PTFE-to-seal volume ratio was calculated, based on given priorities, such that the composite seal outperforms the elastomeric seal of the same dimensions. This means that an elastomeric seal can be replaced by a composite seal of the same dimensions (thus fitting in existing housings without modifications), giving lower leakage, friction and extrusion.

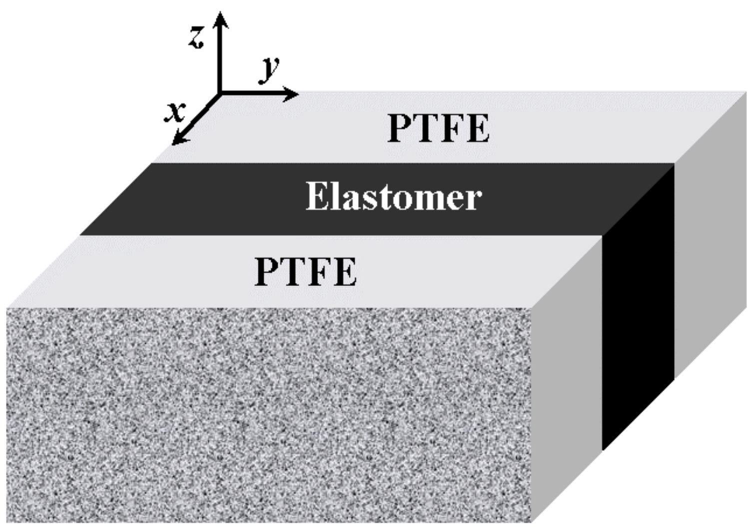


Figure 16. Example of composite seal [42]

Other means of improving sealing performance in a hydraulic system involve the fitting of two seals in a row. The seals in this arrangement are known as dual or tandem seals (Fig. 17) and their primary goal is to minimise the leakage-per-cycle. The primary seal, which is the closest of the two to the high-pressure chamber, performs most of the sealing. The secondary seal merely wipes the fluid that has leaked from the primary seal. Various seal combinations can be had in such an arrangement, based on design priorities.

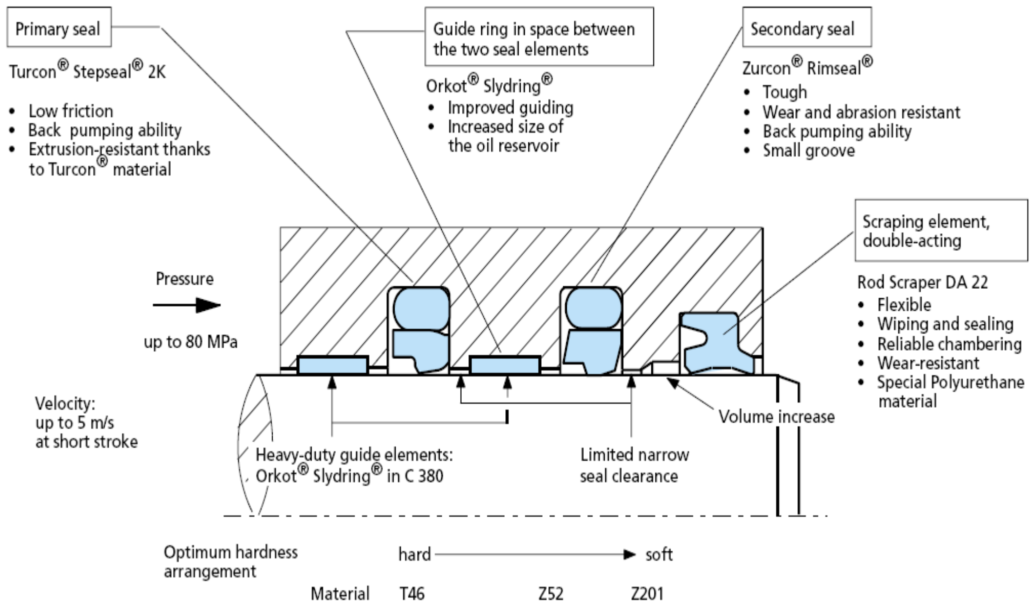


Figure 17. Tandem seal arrangement (courtesy of Trelleborg Sealing Solutions [55])

If properly designed, the tandem seal arrangement can reduce system leakage significantly. However, caution is advised to avoid designs promoting the development of an interseal pressure. The latter, mostly observed in identical-seal arrangements, is about the abrupt rise of the pressure at the interseal space to very high values, sometimes exceeding sealed pressure and causing damage to back-up rings and/or sealing failure. This phenomenon was quite early recognised and studied experimentally by Field and Nau in [145]. Nikas and Sayles [40] presented a mathematical analysis of identical tandem rod seals supported by back-up rings and studied their benefits in terms of leakage and friction, reporting on leakage reduction in the order of 50 to 70 per cent over a wide range of temperatures compared with a single-seal arrangement.

Furthermore, assuming that the interseal space is initially filled with air and using the van der Waals equation of state for that air, they analysed the evolution of the interseal pressure with the number of strokes. A typical result of their analysis is shown in Fig. 18 where it is clear that the interseal pressure rises abruptly after about 1700 strokes (the analysis was stopped as the pressure exceeded system pressure after 1 more stroke). Nevertheless, the critical number of strokes to avoid damage can be predicted (about 1600 strokes in the case of Fig. 18) and taken into account during operation to allow for servicing, for example, venting the interseal space. The critical number of strokes may be too large in some systems to pose any problem, as for example in hydraulic actuators controlling aircraft landing gear, where one thousand strokes take a rather long time.

It is also worth noticing in Fig. 17 the use of a scraping element (right end of picture). This is primarily used to prevent solid contaminants entering the system and damaging the seals and gland bearings but it may also perform some fluid sealing, particularly in single-seal arrangements (unlike Fig. 17). Therefore, its geometry plays a vital role in system performance [146].

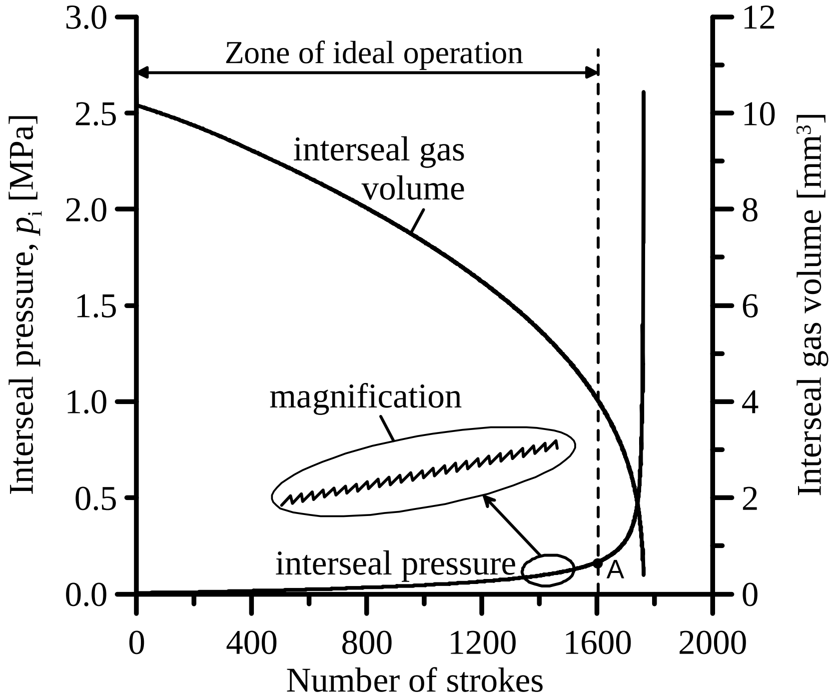


Figure 18. Variation of interseal pressure and gas volume with the number of strokes in a tandem seal arrangement of identical, rectangular, rounded elastomeric seals [40]

The presence of solid contaminants is detrimental not only to seal life in the long term, but also to system leakage in the short term. Debris particles can become embedded on a soft seal and scratch a hard piston rod. The scoring marks are, effectively, micro-channels that allow high-pressure liquid to escape. Unfortunately, it is not only relatively large debris that are capable of such damage. Remarkably, the wear of shafts and seals in tests performed by Tanoue et al. [147] in 1971 was significantly affected by sub-micrometre particles (< 0.25 μm) contained in used lubricating oils. Wear was proportional to particle concentration and it was reported that even for small particle concentrations, e.g. 0.2 per cent by weight, wear was significant.

6 CONCLUSION

Hydraulic reciprocating seals are critical elements of complex mechanical behaviour. Knowledge of their functioning is of paramount importance in hydraulic efficiency and safety. Evaluating the performance of reciprocating seals requires a combination of specialised mathematical tools from contact mechanics and tribology. This is a very difficult task and expert advice is required to achieve successful designs and reliable operation. As with other machine element applications, attention to detail is of great significance. This is often of little concern to the end-user who does not purchase the seals but the equipment fitted with the them. Nevertheless, basic knowledge of sealing mechanisms is advantageous or even sometimes necessary to avoid costly mistakes in high-risk applications and prevent catastrophes such as the destruction of space shuttle Challenger in 1986…

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