International Journal of Applied Engineering Research
ISSN 0973-4562 Volume 10, Number 9 (2015) pp. 23479-23512

http://www.ripublication.com

A Review On Tribological Characteristics Of Reciprocating Hydraulic Seals

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Abstract

Reciprocating hydraulic seals are one of the critical machine elements to prevent leakage of hydraulic fluid for improved performance of the systems in variety of applications. Numerous researchers have conducted various studies on reciprocating hydraulic seals over several decades. However, most of the initial research works were experimental and hence time consuming. The existence of high performance computers facilitated significant progress in the seal design in recent years. In order to understand the limitations of experimental approach and assumptions and approximations in numerical analysis, a detailed literature review on the seal research has been carried out. This review article was focused on the study of parameters affecting seal performance in terms of leakage, friction, wear and extrusion of seal. This paper also provides an insight into sealing mechanisms; various hydraulic seal materials, material models used and critical challenges existing in the field of reciprocating hydraulic seals. The lacuna identified in the literature data has been brought out and need for further investigation is highlighted.

Keywords: Reciprocating seal, leakage, friction, wear, elastomers, seal extrusion

1. INTRODUCTION

Hydraulic seals are used to seal the opening between various components in the elements having rotating, reciprocating or static contact. Hydraulic seal is a relatively soft non-metallic ring captured in a groove or fixed in a combination of rings, forming a seals assembly to block or separate fluid in static, reciprocating or rotating motion application. Hydraulic seals are vital in machinery. Their use is critical in providing a way for fluid power to be converted to linear motion.All seal applications are classified based on their relative motions. Static and dynamic seals are two types of hydraulic seals. If there is no relative motion between the interacting surfaces then the hydraulic seal used is called as static seal. Under conditions involving reciprocating, rotary motion between the interacting surfaces the seal used is called as dynamic seal. These seals can be further classified as rotary seal and reciprocating seal. If there is rotating motion between the interacting surfaces then the seals used are called as rotary seals. This type of seal is illustrated as shown in Fig. 1. Rotary seals may be used for rotating shafts used in pumps, motors, etc. The commonly used configurations of these seals are M-seal and Labyrinth seal. These seals are used in conditions involving a reciprocating piston and rod as shown in Fig. 2. Reciprocating seals are the topic of study in this review work and are explained in detail.

Reciprocating hydraulic seals are one of the important machine elements for the prevention of leakage of hydraulic fluid for improved performance of the system. These are used in a wide range of applications consisting of industrial, automobile, aerospace, defence, etc. They are usually made elastomers, plastics, polyurethanes, metals, as well as composites. Seals operate under dynamic condition of variable speed (up to 15 m/s), pressure and temperature. The operating temperature can be as low as 203 K or as high as 523 K [1]. The pressure in some defence applications like in the demining equipment's the hydraulic linear actuators; driving the mechanisms experiences shock / blast load during operation and can experience peak pressure of upto 120 MPawhich is about 3-4 times of the normal working pressure.Hydraulic seals are commonly used in hydraulic linear and rotary actuators and many of the hydraulic components. The hydraulic linear actuators use both static and dynamic seals. The reciprocating dynamic seals are used as rod seals and piston seals. The rod sealing system are one directional and do not permit fluid to leak out from within the actuator. The piston sealing system are normally bidirectional and do not permit fluid to leak from one chamber to another.

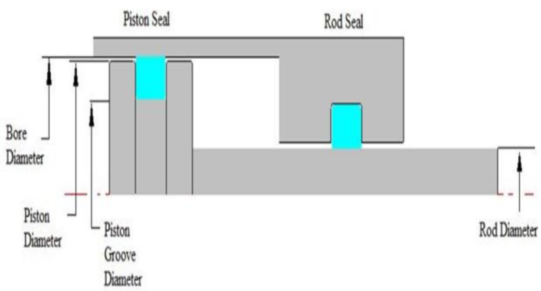


Fig. 1. Illustration of a reciprocating seals

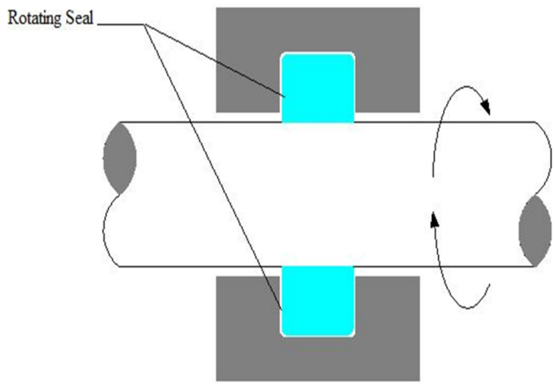


Fig. 2. Illustration of a rotating seal

The reciprocating dynamic seals are also used in shock absorber and similar applications wherein the strokes/seal travel are much smaller and the moving components are well guided. In such system the mechanical clearance between the stationary and moving part are much smaller. The dynamic seals, in such cases works with very less annular gap and the sealing system design are well established. For larger bore actuators with long stroke and high dynamic/shock /transverse loads, there is a need for large clearances.Large clearance for such cases is required to avoid metal to metal rubbing between piston and bore, also avoid contact between rod and gland so as to avoid damage to bore/rod surfaces. Seal extrusion leading to seal failure under such cases needs to be studied under extreme operating conditions and subjected to shock/blast loads.

A large number of variables have a significant effect on the performance of seals, which increases the complexity of the seal behaviour. Hence, the reciprocating hydraulic seal is a rather ignored machine element though it has a very important role in wide range of applications. Although there are difficulties in the evaluation of sealing performance, hydraulic seals are found in variety of critical applications with machinery which are 100 to million times costlier than the seals.

The critical importance of the seal can be quantified by the failure of a static O-ring due to the cold temperature freezing causing the catastrophe of NASA space shuttle “Challenger”, 1986 [2]. Therefore, the precise design and evaluation of reciprocating hydraulic seals is of supreme importance to avoid such costly mistakes. Failure of seals may also lead to huge financial loss and when the toxicity of some hydraulic fluids is taken into account leaked fluids may cause environmental contamination. These examples of the importance of seals and seal design places seal research and development in proper global perspective from both the seal manufacturers and society’s point of view. The increasing needs of society and end users have increased the importance of seal research and have contributed to a better understanding of the sealing and lubrication mechanism and the parameters which influence the performance of reciprocating seal.

The first well documented methodological research in the field of reciprocating hydraulic seals was linked to the development project of hydraulic actuation and control made to the air force during the 2nd world war [3]. A large number of experiments were conducted in this study hence, even after 70 years of its publication this remains the state of the art. Nevertheless, serious research on reciprocating seals dates back to at least 1964 [4, 5]. Many studies since then have been performed, however, with the advent of high performance computers there has been a significant progress in the seal design. Hence a detailed review study on the seal research is necessary.

The seal contact is the specific contact between the seal and the opposing surface. There are two type of seals contact seals and non-contact seals. Contact seals are used in most applications because non-contact seals normally show an inadmissible leakage. Such seals are usually made of elastomers. Generally they have a cross section of which the largest radial dimension exceeds the seal housing and the rod or cylinder. The seals are pre-stressed at assembly. When the seal is pressurized there is an automatic increase in pre stress due to the seal geometry and the material behaviour. Such seals are called “self-acting” seals. During static conditions there is less leakage in such seals as the maximum contact pressure between the seal and the housing wall of the inner and outer seal diameter always exceeds the seal pressure so that the sealing behaviour is improved. Non-contact seals cannot control the leakage due to the presence of clearance; this is the major difficulty of the non-contact seal. Hence, it is necessary to recognize the correlation between leakage and hydraulic power before selecting the type of seal. Non-contact seals are mostly utilised for sealing of dynamic system such as pumps and motors. The disadvantage of non-contact seal is due to the presence of a clearance, and therefore cannot entirely prevent the leakage, so it is really important to recognize the influence law of the leakage and hydraulic power.

Hydraulic seals are critical machine elements. Although these are relatively small, inexpensive and simple components compared to the complex system of which they are a part of, their sealing performance may be of critical importance considering the reliability and the safety of the overall system and an important indicator of quality [6, 7]. The reciprocatory motion of the piston rod with respect to a reciprocating seal in a linear actuator is classified as instroke and outstroke. An outstroke is defined as a motion during which the sliding velocity, from a seals point of view is directed from the high pressure side to the low pressure side of the seal. Normally, during an outstroke, fluid is transferred out of the sealed space into the open space. A motion in the opposite direction is called as instroke. The leakage in outstroke and instroke is different. The net leakage is given by the difference between out and in leakage. Hence a small friction in the directions of motion and a zero net leakage is desirable. For zero net leakage per cycle, the rod seal must operate such that the fluid transported during instroke is more than the fluid transported during outstroke. It was observed that, frictional force during instroke is more than outstroke and decreased with rod speed[8]. The importance of tribological behaviour of seals dependsmostly on the magnitude of the losses (friction, leakage and wear) on the operating parameters on one hand and on the demands of the application on the other.

Friction will occur in the seal contact during relative motion. However, the sealed fluid functions as the lubricant for hydraulic seal thus reducing the friction and preventing a big loss of force and excessive wear of seal which would have led to a premature failure. Hence, separate lubrication for hydraulic seal is not needed. When a seal is effective at preventing leakage, the friction force at the sealing interface tends to be especially high, due to the proximity of the sealing surfaces. Since friction can be large and likewise tends to vary significantly with sliding conditions, it may determine the dynamics of the hydraulic system of which it is a component. Therefore, it is suitable to model reciprocating seal friction behaviour simply, based as much as possible on physical process and parameters.

The dynamics of sealing consists of various physical phenomena taking place at several length scales. The mounting interference, sealed pressure, macroscale contact mechanics at the seal-rod and seal-housing interfaces, as well as the viscoelasticity of the seal are the factors which influence the macroscale geometric configuration. Small local deformations caused due to the effect of fluid pressure in the sealing zone, as well as the mechanics of contact between the microscopic asperities and the rigid rod surface are the factors which influence the microscopic configuration of sealing edge. The fluid pressure in the film depends on hydrostatic effects, hydrodynamic effects, the film thickness distribution, and on the viscosity of the fluid. The fluid viscosity itself is influenced by the pressure and temperature of the fluid. The temperature of the fluid may vary due to change in ambient conditions, heat generated due to friction from the shearing action of microscopic asperities, and from the fluid shear, as well as from the distribution of this heat to the fluid, seal, and the rod. If the fluid pressure drops below the cavitation pressure, cavitation takes place, which in turn affects the fluid film thickness and flow rates. Friction forces arising due to the asperity contact and from fluid viscous shear can change both the macroscale and microscale geometrical configurations of the seal and hence, affect the fluid film thickness and contact pressure distributions. The viscoelastic material properties of the seal are one of the major factors affecting the dynamic seal configuration and thus, the overall sealing mechanism.

The fluid mechanics of the lubricating film is governed by the Reynolds Eq. and is coupled with the elastic deformation mechanics of the sealing element. The coupling is handled in one of the two ways, the direct method or the inverse method. In the direct method [9-12], iteration is applied to maintain both the hydrodynamic pressure distribution and the film thickness distribution. In the inverse method [13,14], the hydrodynamic pressure distribution is assumed to be known and equal to the static contact pressure distribution. These models are based on two assumptions (i) full film lubrication and (ii) perfectly smooth sealing surfaces. It was found as early as in 1973 that there are major troubles associated with the basic assumptions of the above mentioned models [15]. It was discovered that over a wide range of condition mixed lubrication takes place [16]. While most previous theoretical studies assume that full film lubrication occurs between the seal lip and the shaft, mixed lubrication is considered in recent simulations [17, 18] in correspondence to the experimental evidence [19]. A schematic of typical hydraulic rod seal is shown in Fig. 3. The area where the elastomeric seal is observed to be touching the rod is named as the sealing zone; it is where the sealing action takes place. Figure4 shows how the sealing zone has been interpreted by earlier models; the surfaces of the rod and seal are assumed perfectly smooth and completely split by a continuous film of hydraulic fluid, i.e. full film lubrication.

Since the 1990s and with the advent of sophisticated computers, the study of hydraulic seals has become simple as the use of CAE tools has increased, replacing the time consuming and expensive empirical methods. The use of CAE tools such as FEA allows engineers to design sealing components more effectively and minimizing the risk of undesired prototype features resulting in the necessity of redesigning and repetitive testing.

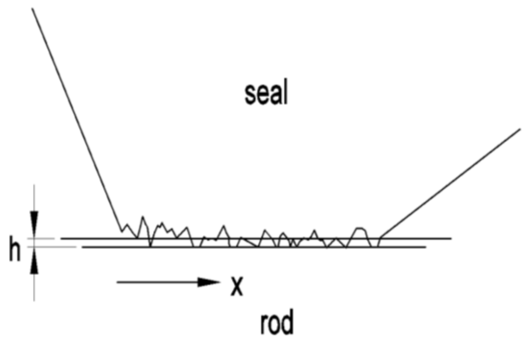


Fig. 3. Hydraulic seal zone [20]

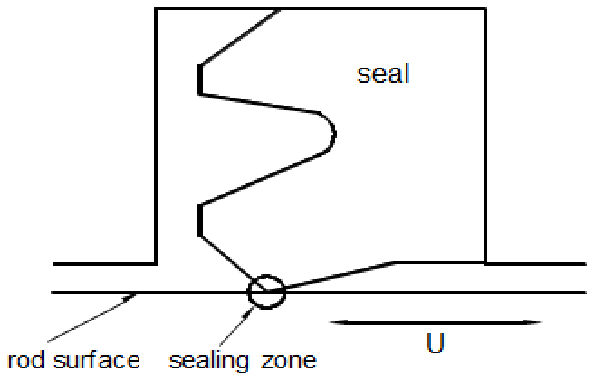


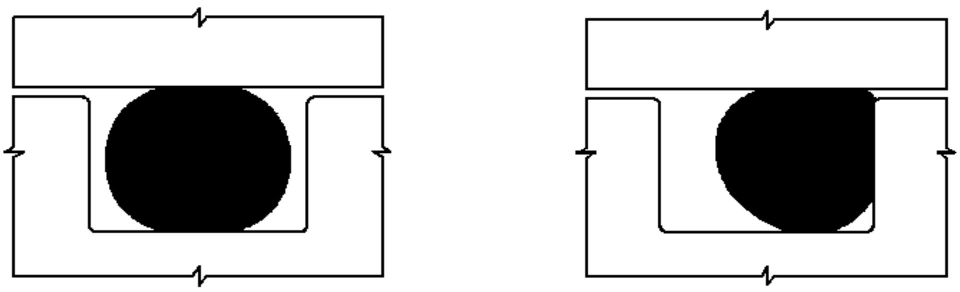
Fig. 4. Typical hydraulic seal [20]

However, seal designers have no analytical tools, beyond FE structural analysis [21-24] with which to predict the behaviour of potential seal designs or interpret test results. Increased demands on performance, operating conditions, reliability and environmental safety of a new or improved seal design within a shorter time to market from the end users need up to the new design has increased the need to link practical to theoretical knowledge. This report covers topics including seal mechanism, seal materials, factors involving the working of reciprocating seals.

2. SEALING MECHANISM

Seals are compressed between two mating surfaces and are retained in a seal gland. The initial compression provides sealing necessary for a successful sealing. As the pressure is increased across the seal, the seal is forced to flow to the lower pressure side of the gland (Fig. 5). The sealing movement causes the increase in force of seal contact and it gains greater area. At the designed pressure limit of the seal, the seal just begins to squeeze into the gap.

During dynamic sealing there is a relative motion between the shaft and the seal surface. The thick line through the contact areas between shaft and seal in Fig. 6 denotes the film thickness during relative motion. The area between the sealing lips and the dust lip can be filled with grease or fluid while the film thickness in the visible area at the air side of the dust lip must be minimal. During instroke all fluid that was sent through the contact zone must be transferred back into the shock absorber. In soft highly deformed contacts the entry and exit regions are only a small fraction of the total contact zone.This contact under lubricated conditions can be classified into three regions. The overall behaviour of the contact is controlled by a short entrainment zone, in which the film thickness is formed this region exists at the leading edge.The oil from film thickness formed at the leading edge is then dragged through the central region at a speed close to the instantaneous entrainment velocity, with only few modifications in the process fluid emerges at the final region (i.e. trailing end).At the trailing end is the exit zone where the oil from oil film thickness undergoes a short lived reduction as it passes through an exit restriction before leaving the contact.



a. Before application of pressure b. Application of pressure

Fig. 5. Sealing Mechanism [25]

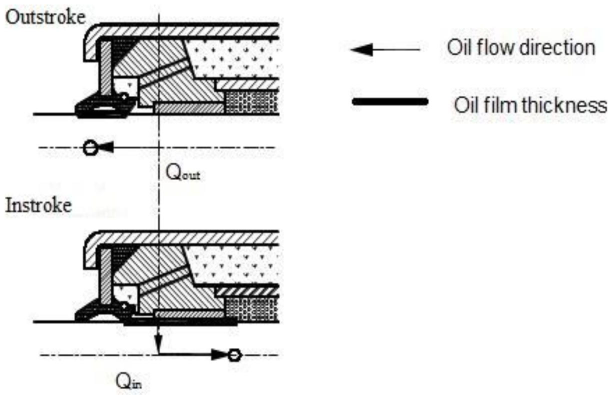


Fig. 6 Definition of instroke and outstroke of a reciprocating seal

3. PARAMETERS AFFECTING THE SEAL PERFORMANCE

The major parameters which affect the performance of seal are leakage, friction, wear and extrusion.

LEAKAGE

Leakage is the loss of fluid across shaft or housing and seal. Thus the main task of the designer is to produce a seal, which has zero leakage. In recent years, researchers are giving high attention on the elasto-hydrodynamic behaviour of seals to minimize the leakage problem with seals and improve the overall performance and efficiency of seals. Leakage of oil can occur under different circumstances and two types of leakage may occur which are identified as static leakage and dynamic leakage.

- **Static leakage**

It is termed as the loss of fluid in places where no relative movement between the seal and the shaft or housing takes place. This type of leakage can occur in the contact areas between the rod and the seal during standstill of the rod.

- **Dynamic leakage**

It is defined as the loss of fluid during the relative movement between the rod and seal through the area where relative movement takes place. The primary cause of leak is the pressure difference between the fluid and the ambient pressure, resulting in a pressure penetration of fluid between rod and seal when fluid pressure exceeds the contact force per unit area. Some other cause of leak can be the existence of asperities having microscopic channels in the contact region between the rod and seal resulting in a fluid flow due to the pressure differential over the channel combined with capillary suction effects which allow fluid to extend the contact area resulting in leakage.

The surface roughness can result in asperities extending through the surface of the lip to make dry contact with the extremely thin fluid film producing high wear. The surface finish of the shaft at the nominal contact strip must be very high.

Scratches or other surface irregularities can result in local thickening and breakdown of the fluid film and consequently cause leakage from the seal. On the other hand, too smooth a finish may exhibit wetting and retention of the fluid film on the shaft [25].

➢ NUMERICAL METHODS TO CALCULATE LEAKAGE

Over the years, many mathematical models have been produced in an effort to hold the film thickness and pressure distribution to predict the leakage for hydraulic seals. To account for the effects of roughness on the pressure, Patir and Cheng [26] derived the Reynolds equation for the average pressure by using flow factors. A Using a flow continuity equation Reynolds equation in general form can be derived [27] as shown in Eq. (1).

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial y}\right) = 6\,\mathrm{U}\,\frac{\partial h}{\partial x} + 12\frac{\partial h}{\partial t} \qquad (1)$$

Mathematical flow simulation of a rough surface with a Gaussian distribution which simulates various types of microscopic channels was used to obtain the above factors. The average-flow Reynolds equation, the bulk deformation elasticity equation, the force-compliance relationship for the asperity contacts and the temperature equation for heat generation were solved simultaneously and the results were given in their study. Elasticity equation and Reynolds equation were solved simultaneously by Field and Nau [9] using numerical techniques. The results showed a strong influence of rod speed on the leakage when full film lubrication and surface finish of high order was assumed. Kanters et al. [28] and Müller and Nau [13] assumed a hydrostatic pressure distribution and equated it to the static contact pressure distribution. They developed a method for solving the Reynolds equation for the film thickness distribution, yielding an expression used to obtain an estimate for the leakage of the seal during instroke and outstroke of the rod. Nikas [29, 30], Nikas and Sayles [31] used in their study a mathematical model developed by them that considers mixed lubrication and surface roughness effects. But their study was restricted to rectangular seals. From the results of the numerical model it was concluded that the sealed pressure and the striking velocity vary the performance of leakage rate, film thickness and friction force.

Lebeck [32] presented a complete mixed lubrication model for mechanical face seals with plain faces. For the purposes of design and analysis of plain-face contacting mechanical face seals, this model is necessary to predict the seal leakage and the friction power as the function of the facial profile and the operating conditions. Kasem [33] introduced a mathematical analysis of leakage rate for the selection of elastomeric sealing materials. The leakage is influenced by many factors, such as the seal material, temperature, pressure, seal compression and seal geometry. He investigated leakage rate for six different elastomeric materials by using the visco-elastic property of the materials. Non-linear stress-strain behaviour, including hyperelasticity, viscoelasticity and viscoplasticity property is presented by materials having strains exceeding a certain bound, hence making material modelling a complicated task. The most popular phenomenological models for rubber [34-36] are hyperelastic material models such as Ogden model [37-39], Neo-Hookean model [40] and Mooney-Rivlin model [41, 42]. However, other methods like Arruda-Boyce

Model [43] and Sussman-Bathe [44] model can also be used for material modelling.

Yank et al. [45] numerically investigated the seal leakage of U-cup seal and seals under the actuator conditions of low pressure on the outstroke and high pressure on the instroke. It was found that thick lubrication film was formed during outstroke. Salant et al. [46] proposed a numerical model to reveal the underlying physics of the sealing behaviour, however the outcomes were not validated by the experimental measurement. Hörl et al. [47] subsequently investigated the sealing performance of the simple seal structure in terms of the leakage measurement, pumping rate measurement and film thickness measurement on the rod surface experimentally. Nikas and Sayles systematically considered the most popular non-linear (Mooney-Rivlin) model in analysing reciprocating seals at temperature between 219 K to 408K [30, 31, 32, 48, 49] in their various research works. Johannesson[50] calculated oil film thickness, oil leakage and friction forces in seals using numerical methods. The study also included the effect of cavitation.

The outcomes from these mathematical models were useful, but experimental results would show issues involved in their growth.

➢ EXPERIMENTAL EVALUATION OF LEAKAGE

Measurement of the net-leakage provides the basis in practice to judge the performance of a seal. Separate measurement of the out or in-leakage may be desirable when studying the tribological process in the seal contact. Different methods have been used to measure the leakage:

- Removing oil from the rod and weighing it
- Measurement of the oil flow
- Electrical methods

▪ *Removing oil from the rod and weighing it*

Sampling the oil, omitted from the rod, has been used very frequently to measure the net-leakage, e.g. by Nikas [1], Iwanami and Tickamore [51], Müller [52, 53], Hirano and Kaneta [54], Field and Nau [55], Field [56], Schrader [57], Messner [58] and Kaneta [59]. The method can be very accurate, averaging the leakage over many strokes.White and Denny [3] tried to take out the leaked oil by absorption with e.g. blotting paper to find the out-leakage. They claimed without proof that only one molecular layer of fluid could not be taken away. Tests performed in this study indicated that a fluid film of about 0.25 µm thicknesses could not be removed in this way. The method was also used by Lang [60].

▪ ***Measurement of the oil flow***

The net-leakage can also be found by measuring the flow of oil, necessary to hold constant pressure in the sealed space. This was done by Field and Nau [55], Field [56], Lindgren [61] and Karaszkiewicz [62]. Field and Nau [55] and Lindgren [61] measured only the average net-leakage of two seals. They reported nothing on the influence of leakage at other parts of the assembly, such as the oil supply pump. Karaskiewicz [62] found out that their influence on other parts of assembly is low. He also examined the influence of pressure variation and temperature. The overall accuracy was calculated and it was found to be around 3 mm^3 in leaked volume. The influence of leakage caused by the piston movement was not studied. Obssen [63] measured the out leakage of unpressurised O-ring seals by determining the amount of fluid, needed to restore the fluid level in an open, vertical housing. Prokop [64] measured the in and outstroke volume fluid flows across the seal by measuring the difference of the fluid volumes transported between the air side and the pressurized fluid side of a seal during instroke and outstroke.

Schrader [57] used a process in which a non-ferrous spring pressed an iron pin, against the seal surface. The displacement of the pin was measured by an inductive transducer. However this experimental procedure has disadvantages that, additional concentrated mechanical load acts on the relatively soft seal, causing a discontinuity of the lubricating process at the measuring point.

- ***Electric method***

Field and Nau[15], [55], [65], and Austin et al.[66] used rubber seals filled with carbon black to measure the electric capacity over the lubricant film. The specific resistance used for rubber was 2.78 Ωm [66], which is very low compared to the specific resistance of oil used (10^7 Ωm). In the measurements of Field and Nau [55] the errors vary about 3% at low film thickness (0.25 µm) to about 40% at film thickness of (5 µm), which is attributed to parasitive capacities in the electric circuit [15]. They reported that the pressure above 8 MPa was estimated to be approximately 30% lower than the errors at lower pressures. This could be because of the dependence of pressure on the seal resistance. Wernecke [67] conducted experiment to find the electric resistance of the lubricant film. The electric conductivity of the rubbers would be sufficient according to [68]. According to [69], however, it would not. It is observed, that for the same material(rubber) different concept have been used in both publications. Field andNau [55], reported about the influence of the seal resistance on the measurements. Thorough analysis of the applied method is essential as measuring the film thickness by an electric meter is not as straight forward as it might appear. The accuracy of the performance measurements is not considered to be very good. The electric resistance of the seal material must be depressed; hence the applicability of the method to commercial seals is very poor.

Other methods such as fluorescence method was applied by Smart and Ford [69] and Ford andFoord [70]. The effect of this method is that electro-magnetic radiation of a certain frequency is transformed by oil particles to the radiation of a lower frequency. Continuous measurement of out leakage is possible.

FRICTION

Friction will occur in the seal contact during relative motion. Normally, the seal contact must be lubricated to reduce the friction and so prevent a large loss of power and excessive wear of the seal, leading to a premature failure.

➢ NUMERICAL METHODS TO CALCULATE FRICTION

It is important to study the frictional behaviour of non-linear, viscoelastic material components sliding on rough surfaces in a number of applications, including seals, wiper blades, and tires. Previous studies done by researchers dealt with the improvement of tribological properties of sliding pairs and decreasing the friction. Theoretical work on reciprocating seals is focused on solving the contact mechanics and lubrication problems. As discussed in theoretical calculation of leakage, the pressure and film thickness distributions in the contact are obtained; the friction force is computed from the results obtained from the numerical methods used for leakage calculation. A mathematical model of elastohydrodynamic contact coupled to a multiscale surface texture model was developed by Demirci, et al. [71] that allows tracking the scale effects of surface features and their iterations on friction performance and lubricant flow under hydrodynamic lubrication conditions.Applied pressure and seal installation forces are the factors which influence the friction of dynamic seals used in hydraulic systems. Hydraulic fluid types, hardware temperature, sealing surface finishes, seal material, and duration of seal contact are the other factors that should be considered as well. The iteration of all the variables affecting the seal operation have to be considered, but at the same time the contributions of these variables to the calculation of friction force with the numerical analysis is very complex, due to the simple reason that the above factors overlap and act cumulatively, so it is difficult to establish an exact or accurate statement regarding the expected friction level [72].

Bhaumik et al. [73] investigated the contact mechanics phenomenon of the reciprocating hydraulic seals by measuring friction force at varying rod speeds and contact pressure. They calculated the contact pressure at the seal/rod interface using FE model and employed IHL model to evaluate the friction force. Numerical methods showed that at constant oil pressure the frictional force decreased with increase in rod speed, it was also observed that at higher contact pressure the stored energy in the seal increased and hence sealing performance was improved. These numerical results were in good agreement with the experimental results.

A radical departure from the existing sealing theories was first presented by Kuzma [74], by developing a concept based on tangential deformation of the seal surface due to viscous shear forces. Wassink et al. [75] modelled seal friction under constant speed sliding as sum of three physically based components: 1. Viscous shear loss in the lubricant, 2. Hysteresis losses due to roughness imposed deformation of the seal material, and 3. Hysteresis losses due to deformation caused by varying intermolecular forces at the sliding interface. It was observed that simulation results matched the experimental values.

➢ EXPERIMENTAL MEASUREMENTS OF FRICTION

Most of the experimental works consider the measurement of friction force exerted on the rod by the reciprocating seals. Various test rigs have been designed to measure the seal friction from the beginning of seal research. The measurement of frictional force is not difficult and can be accessed directly by using force transducer itself [76-78]. There mostly lie two problems. The suspension of housing or rod, which may introduce non-negligible contributions to total friction; different techniques were utilized to minimize or evaluate that part. Second is the separation of the measurement for the outstroke and instroke friction. Some experimental studies on rectangular seals conducted by Field and Nau [65] have shown that the frictional levels have significant direction dependence even though this phenomenon has not been investigated extensively.

The housing was reciprocated instead of the rod in the test rig of Hirano and Kaneta [54] and of Kawahara et al. [79]. For both the rod and the housing suspension,linear motion roller bearings were used. A test rigarrangement was used by Nau [80] as shown in Fig. 7.Test rigs following different concepts have been applied to determine separate out and instroke friction. The sum friction of a rod seal and a clearance is measured using the concept as shown in Fig. 8.

Such measurements were performed by Cheyney et al. [81]. A housing in arrangement with the clearance at the upper side was used by Kawahara et al. [19]. The friction of the clearance was considered negligible by partly filling the housing with fluid and pressurized air. For friction measurement, the housing was supported by two leaf springs and supplied with strain gauge.

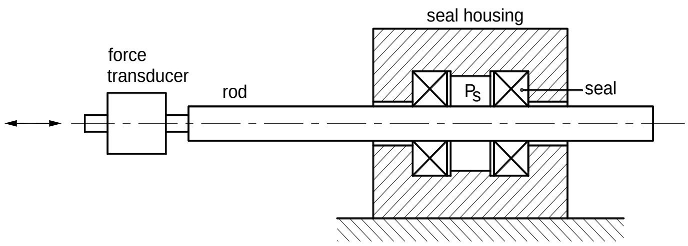


Fig. 7. Basic concept of friction measurement on two rod seal [75]

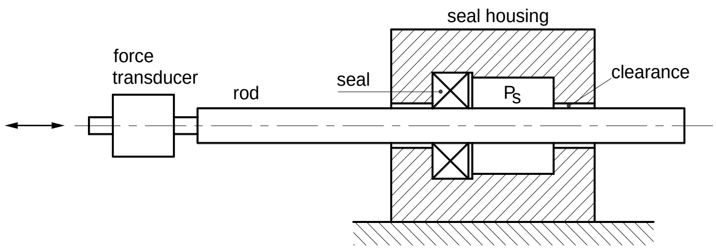


Fig. 8. Test rig to determine friction of single rod seal [81]

The concept of Fig. 9 in which the rod ends in a closed housing was among others used by Lawrie and O’Donoghue [5] and by Kambayashi and Ishiwata [4]. Lawrie and O’Donoghue [5] suspended the rod by roller bearings and the housing by a guide block. The force transducer was attached to the housing. A major disadvantage of the concept as shown in Fig. 9 is that during reciprocating motion the pressure is not constant. This was eliminated by Müller [52] and Sick [82] using a second rod, reciprocating 180 degrees out of phase. Müller [52] guided the rod motion with a hydrostatic bearing, the friction of which was calculated by an empirical formula. Sick [82] used roller bearings, but their friction did not enter into the measured force. Another disadvantage of the concept of Fig. 9 is the force on the rod and the housing caused by the fluid pressure. This force participates in the total power measured and will generally be a lot larger than the seal friction. Hence, very accurate measurement of the fluid pressure is needed. As illustrated in Fig. 10 the concept of a divided rod and the parts connected by a force transducer, was used by Gawrys and Kolleck [83].

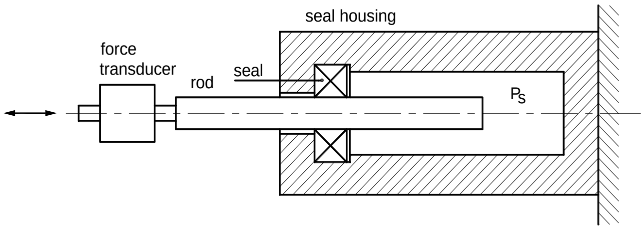


Fig. 9. Use of closed housing to determine friction of a single seal [5]

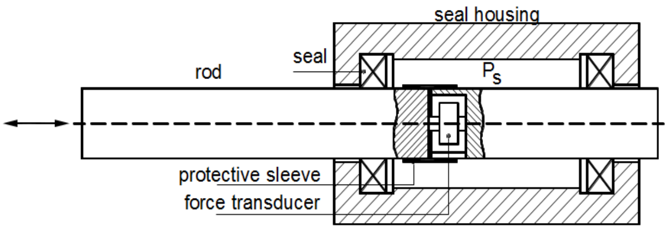


Fig. 10. Use of two rod parts connected by a force transducer to determine the friction of a single seal [77]

A protective sleeve covered the rod partition to prevent the fluid pressure to act on the ends of the rod parts and on the force transducer; it was found out experimentally that the influence of measured friction was close to 3%. The rod was guided in the housing by brass bushes adjacent to the seals; hence, the influence of friction in the measurement was also included. The concept of Fig. 10 was also used by Messner [58] for studying polytetrafluoroethylene rod seals. In order to study the friction characteristics of the seal-subject to the different design and working conditions friction characterization tests are performed. On the basis of these test results it is possible to compare the friction behaviour of different seals working under various conditions within the operating ranges of the trials.

The friction characteristic tests are carried out for a certain number of cycles or length of the path and the following two curve series are obtained:

i. The friction force-working pressure diagram Fig. 11 were used first and applied for a long time by customers and designers as the major source of information to compare different types of seals and assess their friction behaviour [84]. If the operating pressure is higher than the designed sealing gap then the backup edge of the seal penetrates into the sealing gap and some particles of the seal material are broken off. These particles damage the seal back up and may cause sudden friction force increment while remaining in the sealing gap during reciprocating motion as shown by the dots in Fig. 12.

ii. The friction force reciprocating speed curves (Fig. 13) at a fixed operating pressure show the change of the friction force with the change of the alternating speed. These curves give references on the recommended and not recommended speed ranges and on the optimum alternating speed ranges for application purposes [85-87].

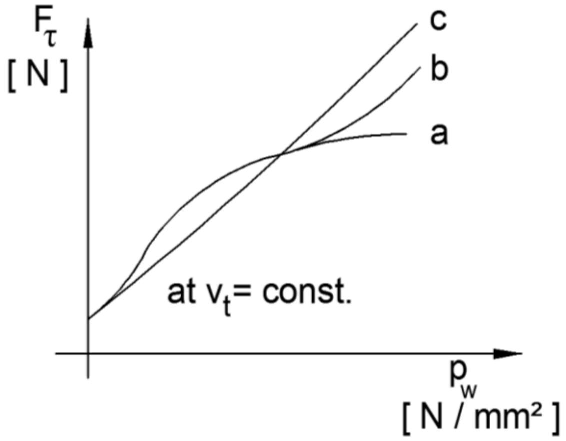


Fig. 11. Friction characteristic curve at constant speed [78]

For U-cup type seals Kawahara [19] constructed stribeck curves for sealed pressure of up to 50 bar. They showed that friction initially falls before reaching a trough after which it increases for increasing rod speeds of over several orders of magnitude. It was also seen that with increasing sealed pressure the friction coefficient decreases.

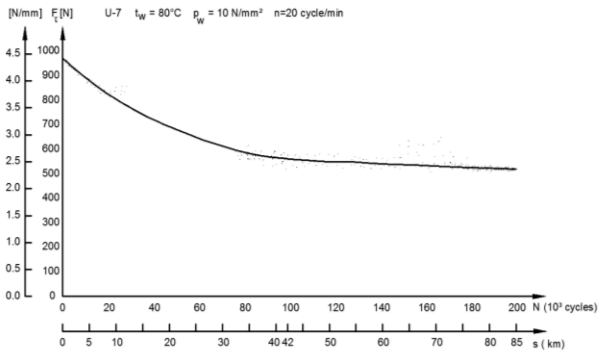


Fig. 12. Friction force curve of endurance test [78]

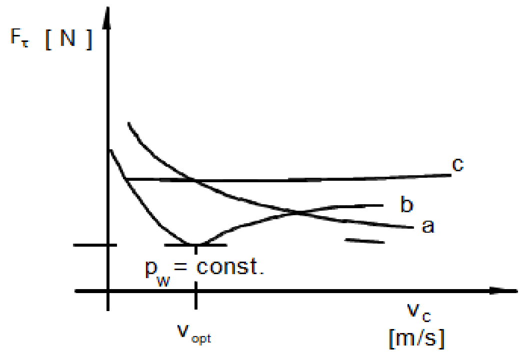


Fig. 13. Friction characteristic curve at constant pressure [78]

This is demonstrated experimentally by various seal manufacturers [88]. Kaneta et al. [89] investigated friction in low pressure fluid sealing through experiments with lubricated wedged and 'D' shaped polymeric pads against a sliding glass surface. A study by Rana et al. [90] concluded seal friction to be independent of rod speed for speed of less than 100 mm/s for smooth rods as a result a fluid film being unable to work at these low sliding velocities. Nau [91] has suggested that use of rod with a surface roughness height below a critical value results in additional friction and wear, possibly by reducing the propensity for lubricant pockets to be formed between the contact surfaces. It is also possible that, at lower asperity heights, the hysteresis effect of friction from the delayed elastic recovery of the material becomes less significant. Papatheodorou's [92] study used ISO 7986 standard for a range of sliding speeds and did not report a region of significantly increasing friction. This may be because the seal is known to experience changes in friction level at higher velocities.

In defence based applications the seal may be exposed to a pressure if up to 120 MPa. Bhaumik et al. [93] developed a test rig to experimentally study the contact mechanics phenomenon of seal which involves high pressure of the orders of up to 120MPa. The test rig was designed according to ISO 7986. They also worked out a test rig as shown in Fig. 14 to measure leakage and friction of a single rod seal wherein the performance of the seal can be measured separately during instroke and outstroke.Yoshimura et al. [94] used interferometer to measure the oil film thickness in rubber piston seal. Monochromatic interferometer as shown in Fig. 15 was used to measure the variation in the oil film profile if the oil film formed on the contact area is sufficiently thick. While, white light interferometer shown in Fig. 16 was used to measure the mean oil film thickness if the oil film is even and thin. It was observed that during instroke and outstroke the oil film thickness remained the same, this may be because the oil leaked during outstroke was recovered during the instroke.

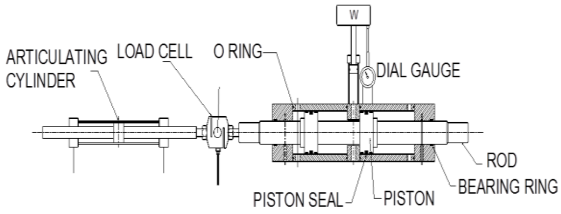


Fig. 14. Rod seal leakage test [93]

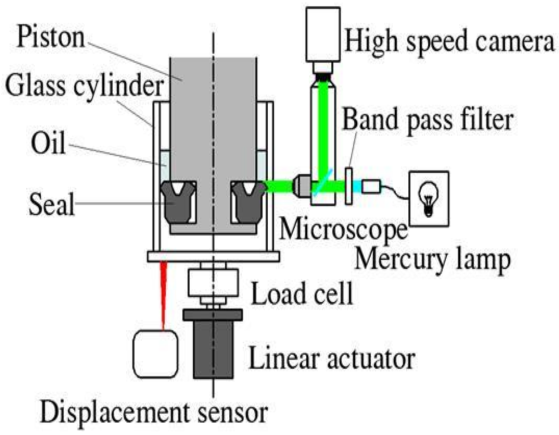


Fig. 15. Test setup for monochromatic interferometry [94]

Grosch [95] connected friction in the dry sliding of rubber pads for glass, steel and abrasive plates with viscoelastic deformation in the rubber. It was mentioned that the friction of rubber in smooth surfaces reached a maximum at a certain speed, falling off at a higher velocity.

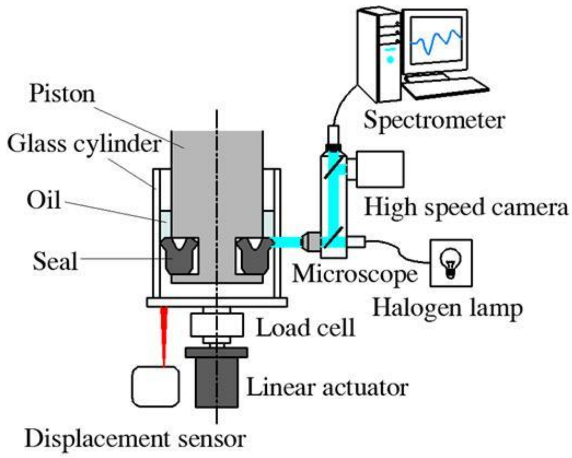


Fig. 16. Test setup for white light interferometry [94]

Gujrati and Ludema [96] tried to account for both viscous and viscoelastic losses due to bulk deformation in the event of a ball sliding on a rubber surface, finding qualitative, but not quantitative agreement between analysis and experiment. Additional important details of rod seal operation have been uncovered by a transient numerical model which follows the history of seal behaviour [97].

WEAR

Another tribological factor which affects the performance of hydraulic seal is wear. Wear is a process of gradual removal of a material from surfaces of solids subjected to contact and relative motion. In hydraulic systems, the seals are well lubricated in most cases, so those parts show minimal wear and they can function for years without any sign of damage. As mentioned by Hoffman [98] that for PTFE seal the friction moments at dry running is almost the same as that at oiled running. Hence most of the measurements were made in dry conditions. However, in some special cases, the rubber seals work under mixed or boundary lubrication conditions. Hence wear modelling becomes necessary.

Mathematical wear simulation techniques were developed in the last ten years and have been extensively used by many researchers. The finite element method approach is the most successful and popular approach as compared to other numerical wear simulation methods such as the boundary element method [99] or the discrete element method [100], since it is a general method for mechanical stress calculation.There are several theories for modelling wear; there are even some which take the fracture or the fatigue properties into account [101]. The first trials on theoretical and numerical calculations of wear profiles have been carried out by Galin and Goryacheva [102].The earliest contributions to the wear constitutive equation were made by Holm [103]. Holm developed a model to relate the volume of the material removed by wear in the sliding distance and the true area of contact. Archard [104] formulated the wear equation (shown in Eq. (2)) in which, the volume of the material removed per unit time is directly proportional to the sliding velocity, the normal pressure and the dimensionless wear coefficient and inversely proportional to the hardness of the surface being worn away.

$$\dot{w} = \frac{K_w}{H} \cdot P_n \cdot V \qquad (2)$$

The wear equation developed by Archard [104] is the most frequently used one. A method of modelling wear, in which the elements are much smaller than the elements in the FE model, in this method the contact nodes are moved according to the nodal wear increments. This was proposed by Pödra and Andersson [105]. Kónya and Váradi [106] improved the method of Pödra and Andersson so as to consider heat generation and time dependent material properties during the wear simulation. Despite the widespread use of the Pödra-Andersson method, the wear simulation is highly limited; only the top layer of elements can be worn.

This limitation does not affect the usability of the method in case of relatively hard materials such as metals. However in the case of rubber the wear takes place in micrometres the wear needed for a rubber part to malfunction is much greater than those of metal parts; therefore the wear of the top layer is insufficient in rubber applications. Hence the material is to be modelled to simulate wear regardless of element type Békési et al. [107] used three different wear simulation techniques for different purposes. The first method was used for modelling relatively small amount of wear. The second model was used to model wear even bigger than the elements in the model by global remeshing. The third method deactivates the damaged elements, thus the effect of the surface rupture of the elastomeric part can be modelled. Sui et al. [108] established a curve based on the change of contact pressure distribution. The area of contact increased, while the maximum values of the contact pressure decreased. This phenomenon can be seen in Fig. 17 for the contact pressure distribution across the lip. The wear behaviour of sealing systems was predicted by a test rig with pin on disc apparatus [108]. Békési et.al [107] performed experimental tests on seal in which seal was cut and a section of it was straightened and fixed on the test bench. The aluminum rod was then pressed and rubbed against the surface of the seal. The worn surface of the seal was inspected using white light profilometer.

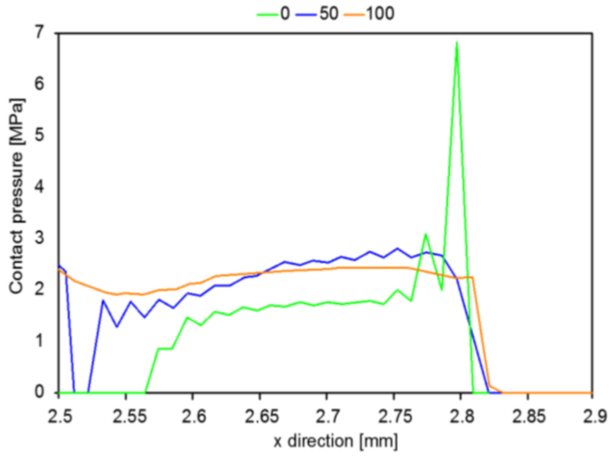
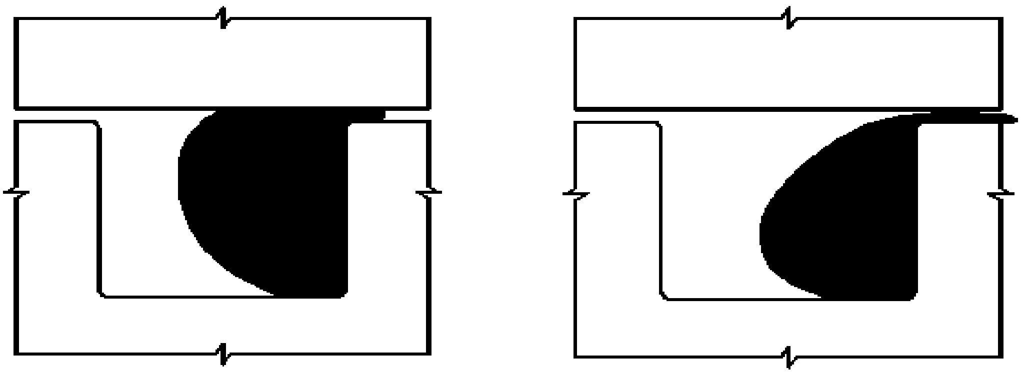


Fig. 17. Change in contact area and contact pressure at the lip over wear process (After 0, 50, 100 simulation cycles) [105]

EXTRUSION

One of the major problems associated with reciprocating elastomeric seals is extrusion that occurs at the low pressure side of the cylinder where the seal is squeezed into the clearance between the housing and the piston rod. Normally if the pressure is exceeded beyond the limit, the seal will fail by extruding into the gap as shown in the Fig. 18. However, depending on the seal corner geometry and contact friction extrusion of seal may take place at zero sealed pressure.



a) Seal extruding b) Seal failure

Fig. 18. Schematic of seal extrusion [25]

The result could be local damage of the seal in the form of a cut or abrasion [88] which can affect the sealing performance and may lead to a reduction of the seal service life. In some cases, under high sealed pressure or contact friction the seal can suffer from roll deformation [110]. Nikas [111] formulated algebraic equations to describe the shape and contact pressure of the extruded part of the seal with the rod. In order to minimize the risk of damage, a study was presented about the effects of various operating parameters. Back-up rings are used to prevent seal extrusion for high pressure static and dynamic applications.

4. SEAL MATERIALS

The materials used to manufacture seals cover the entire range of engineering materials. Elastomers, Plastics, Metals and composites can be used as seal materials.

ELASTOMERS

Elastomers are widely used and potential applications cover the full range of static and dynamic seals. Some major advantages of using rubber seal over other seals are as follows:

Elastomers have relatively large deflections for small stress values [112]. Therefore it is possible to use large strains both during installation and to provide seal interference. Also elastomeric seals can accommodate a relatively wide range of tolerance and misalignment without high contact stress. Figure19 provides an example of stress/strain properties of elastomers compared with both plastics and metals.

Elastomers have high degree of resilience with low hysteresis. Tensile strength is less compared to plastics and metals but elastomers have excellent creep resistance. Elastomers have a very high Poisson’s ratio close to 0.5 hence these materials are nearly incompressible.

In spite of the advantages there are some disadvantages associated with these materials. They have a limited temperature range of 243 K to 373 K for general purpose materials. Chemical resistance is dependent on both elastomeric type and individual grades within a material type. As elastomers are soft and are easily torn away they require careful design and handling. Elastomers exhibit non-linear stress-strain curve. The stress strain curve for different elastomers is as shown in Fig. 20.

The use of appropriate elastomeric material is of fundamental importance to the life of a seal operating under a specific set of imposed service condition. The decision on which polymer to choose so that the seal can operate effectively under these conditions is usually decided by the compatibility of the polymer with the fluid it is intended to seal, and the upper temperature limit of the polymer's physical properties. The lower limit temperature is also of importance in aerospace applications. In order to modify existing physical properties and to reduce the cost of elastomeric compounds fillers are incorporated into the rubber. The proportions of the filler by weight and the type predominantly determine the wear characteristics and elastic modulus of the sealing material [113, 114].

Rubber is assumed to be isotropic in elastic behaviour and incompressible, according to Rivlin's phenomenological theory [44]. The deformation gradient can be expressed in terms of principal strain direction, the Cauchy-Green deformation tensor C is explained with its eigenvalues and the strain invariants, which are related in the following manner.Based on the continuum mechanics, a mathematical formulation is offered by this theory as shown in Eq. (3), (4), (5) to study the rubbery behaviour of materials.

$$I_1 = \lambda_1^2 + \lambda_2^2 + \lambda_3^2 \tag{3}$$

$$I_2 = \lambda_1^2\lambda_2^2 + \lambda_2^2\lambda_3^2 + \lambda_3^2\lambda_1^2 \tag{4}$$

$$I_3 = \lambda_1^2\lambda_2^2\lambda_3^2 \tag{5}$$

Hence, the total strain energy density function can be expressed in terms of the eigen values.

$$W = f(I_1, I_2, I_3) = f(\lambda_1, \lambda_2, \lambda_3) \tag{6}$$

The Eq. (6) can also be expressed as

$$W = \sum_{i+j+k=1}^{\infty} C_{ijk}(I_1 - 3)^i \cdot (I_2 - 3)^j \cdot (I_3 - 1)^k \tag{7}$$

For, incompressible material like rubber I_3= 1. Hence, the Eq. (7) is reduced in the form,

$$W = \sum_{i+j=1}^{\infty} C_{ij}(I_1 - 3)^i \cdot (I_2 - 3)^j \tag{8}$$

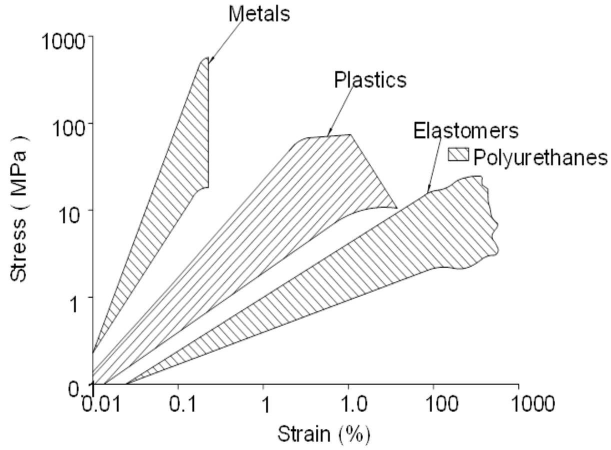
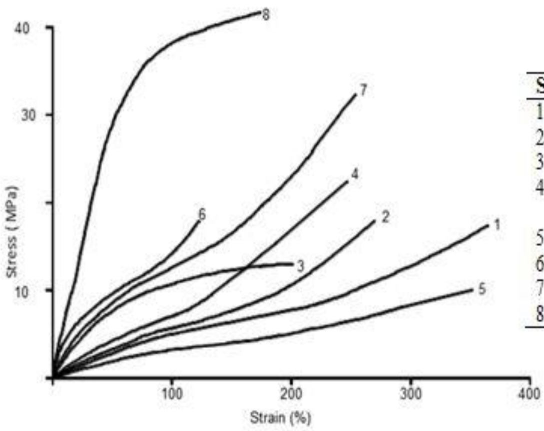


Fig. 19. Stress vs Strain for elastomers, plastic and metals [112]



S. No.	Material Type	Hardness (IRHD)
1	Nitrile	65
2	Nitrile	70
3	Fluorocarbon type A	75
4	Fluorocarbon terpolymer peroxide curve	70
5	Fluorosilicone	60
6	Fluorocarbon, type B	90
7	Hydrogenated nitrile	90
8	Polyurethane	95

Fig. 20. Stress vs Strain curves for elastomers at 296 K [112]

- ***Polynomial model***

Based on the first and second variant of the deviatoric Cauchy-Green tensor a mathematical formulation was presented by this model in the compressible form as shown in Eq. (9).

$$W = \sum_{i+j=1}^{N} C_{ij}(\overline{I_1} - 3)^i \cdot (\overline{I_2} - 3)^j + \sum_{i=1}^{N} \frac{1}{D_i}(J_{el} - 1)^{2i} \quad (9)$$

This model is used generally for modelling stress-strain behaviour of filled elastomers of up to five terms [115].A modified polynomial model was proposed by [116] which neglected the 2nd variant of the left Cauchy-Green tensor as given in Eq. (10),

$$W = \sum_{i=1}^{N} C_{i0}(\overline{I_1} - 3)^i + \sum_{i=1}^{N} \frac{1}{D_i}(J_{el} - 1)^{2i} \quad (10)$$

- ***Ogden model***

This model questioned the necessity of restriction of the form of the strain energy density function W to a second power function of the principal stretches, λ as embodied in the Rivlin's representation using the strain invariants. Ogden expanded the polynomial expression of the principal stretches $\lambda_1, \lambda_2, \lambda_3$and proposed the Ogden model [37] assuming separability of the strain energy density expression [117]. The strain energy can be expressed as in Eq. (11),

$$W(\lambda_1, \lambda_2, \lambda_3) = \sum_{i=1}^{N} \frac{2\mu_i}{\alpha_i^2}(\bar{\lambda}_1^{\alpha_i} + \bar{\lambda}_2^{\alpha_i} + \bar{\lambda}_3^{\alpha_i} - 3) + \sum_{i=1}^{N} \frac{1}{D_i}(J_{el} - 1)^{2i} \qquad (11)$$

This model works well for incompressible materials like rubber for entire stretching range except the limiting stretch region. Ogden model generally requires at least six parameters irrespective of any physical insight into the mechanics governing that state of deformation. In this model huge oscillations outside the experimental data range may be expected, similar to Rivlin model. Another major disadvantage in using this model is the problem of implementing this to 3-D models. Ogden energy function is computationally intensive than polynomial model and hence often used [115].

- ***Mooney-Rivlin model***

Strain energy potential according to this model is presented as follows in Eq. (12),

$$W = \sum_{i,j=0}^{N} C_{ij}(\bar{I}_1 - 3)^i(\bar{I}_2 - 3)^i + \sum_{i=1}^{N} \frac{1}{D_i}(J_{el} - 1)^{2i} \qquad (12)$$

The first order for incompressible material is proposed as in Eq. (13),

$$W = C_{10}(\bar{I}_1 - 3) + C_{01}(\bar{I}_2 - 3) \qquad (13)$$

The Mooney-Rivlin model is obtained by setting N= 0 and α_1= 0 and α_2= -2 in the Ogden model [118]. Mooney-Rivlin model is represented as shown in Eq. (14),

$$W = \frac{\mu_1}{2}(\bar{\lambda}_1^{2} + \bar{\lambda}_2^{2} + \bar{\lambda}_3^{2} - 3) - \frac{\mu_1}{2}(\bar{\lambda}_1^{-2} + \bar{\lambda}_2^{-2} + \bar{\lambda}_3^{-2} - 3) \qquad (14)$$

Where, $C_{10} = \frac{\mu_1}{2}$ and $C_{01} = \frac{\mu_1}{2}$

- ***Neo-Hookean model***

If N=1 the Neo-Hookean equation Eq. (15) can be obtained from the reduced polynomial equation Eq. (10),

$$W = C_{10}(\bar{I}_1 - 3) + \frac{1}{D_1}(J_{el} - 1)^2 \qquad (15)$$

Timbrellet al. [119] found out that this model can be obtained in terms of the first deviatoric invariant. In the Ogden model if N=1, $\alpha_i = 2$ are substituted the Neo-Hookean model can be obtained [120] as shown in Eq. (16):

$$W = \frac{\mu_i}{2}(\bar{\lambda}_1^{\,2} + \bar{\lambda}_2^{\,2} + \bar{\lambda}_3^{\,2} - 3) = C_{10}\ (I_1\text{-}3) \tag{16}$$

For elastomeric materials this model is the simplest hyperelastic model when the material data is insufficient. One of the major advantages of this model is the statistical theory of rubber elasticity appearing at the strain energy function:

$$W = \frac{1}{2}\,\text{NKT}\ (I_1\text{-}3) \tag{17}$$

Achenbach and Duarte [121] found out at though the statistical and phenomenological theory come from different quarters; Eq. (15) and Eq. (17) are of the same form.

- ***Arruda-Boyce model***

The most successful statistical mechanics method so far for rubber is a non-Gaussian eight-chain molecular network model known as the Arruda-Boyce model [43]. The strain energy density function of this model is derived as follows for polynomial form

$$W = \mu \sum_{i=1}^{N} \left[\frac{C_i}{\lambda_m{}^{2i-2}}\left(I_1{}^{i} - 3^{i}\right)\right] \tag{18}$$

From the experiments conducted by Arruda and Boyce it was proved that this model is well suited for rubber materials such as silicon and neoprene with strains up to 300%. It was also seen that even if the test data are limited this model has no issues with curve-fitting [43, 122]. However, it was seen that in the small deformation range the values obtained from this model deviate from the experimental results. From experimental calculations it was found out that the fifth order approximation of the expression is more accurate.

- ***Sussman-Bathe model***

Sussman and Bathe [44] model is based on the assumption that the strain energy density function is a sum of separable strain energy density functions. In this model the true strain e is employed instead of the principal stretch to express the total strain energy density function. In this method cubic splines are used to fit the uniaxial stress-strain curve and thus stress could be expressed as a function of true strain.

$$w'(e) = \sum_{k=0}^{\infty} \left[\ \tau\left(\left(\frac{1}{4}\right)^{k} e\right) + \tau\left(\frac{-1}{2}\left(\frac{1}{4}\right)^{k} e\right)\ \right] \tag{19}$$

The total strain energy density function is obtained by integrating the first derivatives of the strain energy density function. This equation can be expressed as

$$W = \sum_{i=1}^{3} w'(e_i) \tag{20}$$

Uniform cubic splines are employed to calculate the values of the strain energy function instead of proposing an explicit analytical expression. No material constants need to be fitted for Sussman-Bathe model. This model can produce very accurate 3D simulation results given correct and enough experimental data. The accuracy of this model relies on the accuracy of experimental data as in the case of other material models. If only limited data is supplied, the model may become unstable and inaccurate hence test data have to be obtained over a sufficient large range of strain values. From the above mentioned material models Mooney-Rivlin model, Ogden and Neo-Hookean model are mostly used, though Sussman-Bathe model can fit both extension and compression experimental data perfectly while for other models there is a significant deviation from the experimental data especially from compression experimental data.

PLASTICS

Plastic materials can be used for seals, directly as the sealing component or as a part of the overall assembly such as anti-extrusion rings or bearings. The advantages associated with using plastic as seal material is the dry running which is advantageous for dry or marginally lubricated seals. They provide good wear resistance when compounded with appropriate filler. They have comparatively high strength hence provide good extrusion resistance. However, they do not provide inherent energisation like elastomers. They cause fitting problems as they cannot be stretched as in the case of elastomers. They undergo creep or plastic flow.

METALS

A wide range of metals can be used for the manufacturing of seals. They can be used as seal material itself, as in a metal seal, a major proportion of the seal as in mechanical seal or as a energizing spring of a polymer seal or the metal components of a semi-metallic gasket. Stainless steel and nickel alloys are widely used as a material of energizing spring.

COMPOSITES

With the growing possibilities of designing and producing material with complex structures the materials that respond to changing in use condition can be designed so that part or system performance is maintained or improved. Composite materials are designed to address in service requirements by combining more than one material into one part. These materials are a topic of study for many researchers [123].

5. GAPS IDENTIFIED IN THE EARLIER INVESTIGATIONS

Recently, there have been some exciting developments in the study of hydraulic seals that take into consideration the transient, non-linear fluid structure, also including mixed lubrication, surface roughness and cavitation effects. From the previous study it can be seen that after more than 80 years of research on hydraulic seals most of the issues affecting the performance of hydraulic seals have been identified and relatively understood [124] however,some important aspects have been neglected and assumptions are made without quantifying them. These aspects have to be accounted for so as to have a better understanding of sealing performance. There are many problems which still have to be addressed. A model with an agreement between leakage, friction and wear has to be formulated which still remains as a major problem. Some of the gaps identified in the literature are as follows:

- There is a need for optimum seal design through design of experiments to study interdependency of leakage, friction and wear satisfying the customer requirements.
- A life expectancy seal model is to be designed with the parameters affecting its performance under varying operating conditions.
- Though numerical modelling has been carried out by many researchers, the effect of material worn out on contact pressure and axial compression of the seal was not studied.
- Cold freezing of the seal may take place below the operating temperature of the seal affecting the sealing ability. On the other hand, at elevated temperatures, the seal may experience high friction due to boundary lubrication effects and accelerated wear. Though effect of temperature has been studied experimentally in the previous work [1], numerical model needs to be developed considering wide range of temperatures.
- Study of seal failure due to extrusion under shock load needs to be carried out withan aged seals i.e, especially seals exposed to elevated and sub-zero temperatures
- The pressures encountered in sealing systems subjected to shock loading conditions especially in applications such as defence and automobile equipments may range from 100-150MPa. Therefore, there is a need to investigate the seal performance and suggest improved sealing systems that can sustain high pressures.

6. CONCLUSION

The data available in the literature provides a basic understanding on the numerical and experimental studies on hydraulic seals. A thorough understanding of hydraulic seals considering transient, non-linear fluid structure including mixed lubrication, surface roughness and cavitation effects on the sealing performance in terms of leakage, wear, friction by a suitable FE model validated by experimental results is very much required.

Acknowledgement

The authors would like to extend thanks to Mr Rakesh, Mr Abhishek Kumar, Mr PushkrajBhandari, Mr CK Waghmareand Mr Manoj Bhujbal of RandDE (Engrs), Dighi, Pune, India for providing assistance during the course of work. One of the authors Shankar Bhaumik would like to thank Vice Chancellor, DIAT(DU), Pune for permitting him to do PhD.

Nomenclature

C_{ijk}	Material constant for describing shear behaviour
D_i	Material constant for compressibility
e	True strain
h	Film Thickness, mm
H	Hardness of the surface
I_1, I_2, I_3	The three invariants of the Green's deformation tensor.
$\overline{I_1}$, $\overline{I_2}$, $\overline{I_3}$	First, Second and Third invariants of deviatoric strain
J	The Jacobian determinant
J_{el}	The elastic volume ratio
K	Boltzmann's constant, J/K
K_w	Dimensionless wear coefficient
N	Number of network chains per unit volume
P	Pressure, MPa
P_n	Normal Load, N
T	The absolute temperature, K
U	The initial shear modulus, N/mm^2
μ	Lubrication pressure dependent dynamic viscosity, Pa-s
μ_i, α_i	Describe shear behaviour of material
V	Sliding velocity, mm/s
W	strain energy density function
$\dot{w}$	Wear rate (volume of material removed per unit time), mm^3/s
w'	First derivatives of the strain energy density function
τ	Stress, N/mm^2
λ_1, λ_2, λ_3	Principal extension ratios

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