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An experimental study on the friction behaviour of aircraft hydraulic actuator elastomeric reciprocating seals

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This study is on reciprocating elastomeric seals for use in aircraft linear hydraulic actuator assemblies where reliability of operation is of critical importance.

Work has been carried out using an optical seal specific rig simulating the high pressure working conditions that a reciprocating seal encounters. The rig has provided friction measurements using transducers while simultaneously providing detailed 2-D contact images using a boroscope optical arrangement. The effect of pressure and stroking velocity on the friction of the seal is studied. Other effects such as complex flow patterns, leakage and wear in the seal contact are observed and discussed. It is found both through measurements and images that high pressure conditions encourage the development of thicker films and more effective lubrication takes place at higher pressures. The effect of speed is not significant for the range of speeds in the tests.

Particularly of concern are actuator start-up and slow-down cycles when surface adhesion effects lead to high friction. These conditions are present in real applications and the overall reliability of the seals is dictated by these cycles.

1. INTRODUCTION

Linear hydraulic actuators are employed in several engineering applications as a means of transferring large forces using pressurised fluid systems. The sealing of such high pressure systems is done using elastomeric seals. The seals under study here are specifically dynamic rectangular elastomeric seals, which experience motion against the actuator surface while maintaining a seal against pressure differentials. In conditions where safety is of paramount importance the safe and reliable operation of such seals is critical.

This study has been motivated by the aerospace industry where the seals are used in aircraft actuators. The actuators control several types of aircraft operation systems, such at the landing gears, wing flap controls, suspension systems and other aircraft utility systems. These seals demand very reliable and continuous operation through their entire lifetime through some of the most extreme physical conditions such as low temperatures experienced during flight and high temperatures for actuators near the vicinity of aircraft engines. Of particular concern is the need to ensure that there is no loss of sealing fluid into the atmosphere both while the aircraft is static on the runway and airborne. This requires that the seal has maximum reliability under a wide range of operating conditions.

An experimental rig, which simulates realistic geometric and operating conditions experienced by the seal, has been used. The rig provides friction data while at the same time valuable optical information on the seals operation is provided. The results and the significance of the findings are discussed in the following sections.

The study can be treated as a follow up to the initial preliminary work on the same project carried out using a different rig [1] and another analytical study on seals with a theoretical model already developed by Nikas [2].

1.1. Background and Previous Work

The design of rubber seals has been an empirical technique based on many years of experience. Seal design has improved through the decades thanks to better modelling techniques, manufacturing processes and practical experience but the basic design of the seal has remained virtually unchallenged since it's original conception. There have been several numerical studies of reciprocating seals under real operating conditions but relatively very little experimental work has been carried out on seals. A fuller understanding of seals is still unclear given the lack of experimental work using modern experimental techniques.

Some experimental work has been done on seals. To date the most exhaustive and complete work on seals has been carried out by White & Denny [3] shortly after World War II. This still remains the most elaborate and extensive experimental work on seals 50 years later. Kanters and Visscher [4], and Kaneta and co-workers [5-6] have carried out more recent experimental work on seals. The authors mentioned have presented friction data and a significant amount of data exists for elastomer lubrication and film thickness values. However this data is only for very simple loading and geometry cases of the seals. Real seals experience much higher loads and operate in confined spaces. Optical work on seals in their real geometries and loading conditions have only been known to be presented by Kassfeldt [7] and Lawrie and O' Donoghue [8].

The particular advantage of the rig in this study is to provide direct evidence of many effects that have only been stipulated. The rig is still under development so that it can be expanded to accommodate higher strokes and pressures. For this study however the reciprocating speeds and pressures were limited to a maximum of a tenth of the values seen in real actuators.

2. EXPERIMENTAL RIG ARRANGEMENT

The seal specific rig is modelled on a real linear hydraulic actuator with dimensions and clearances designed according to aerospace design specifications. A glass piston is used instead of a steel one. Figure 1 shows the layout of the rig.

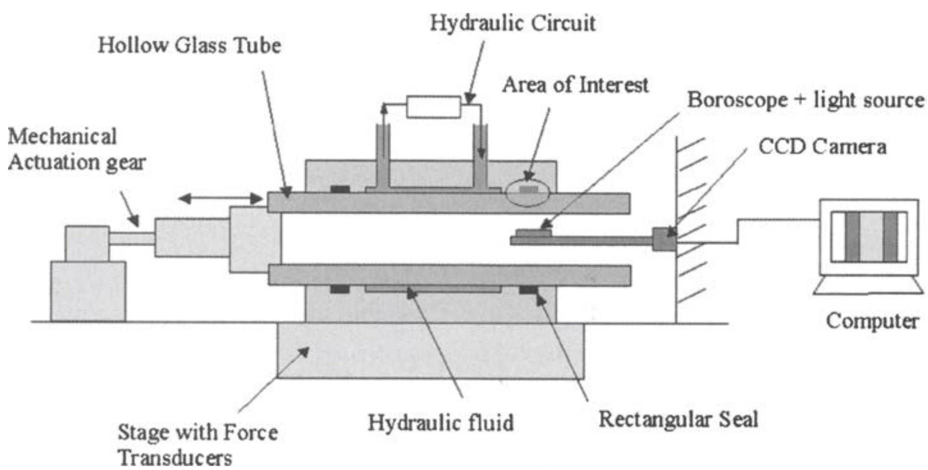

Figure 1 : General layout of seal specific rig

In a real actuator fluid pressure causes the motion of the piston. In this rig however the fluid pressure does not move the glass piston. Instead an external set of gears and motors is attached to the piston and drives the glass piston. This simulates the actuation cycles of a real actuator while simulating the high fluid pressure a seal experiences in a sliding contact. The seal to glass interfacial contact is viewed using a rigid boroscope that is placed inside the hollow glass piston. The boroscope has its own light source and live video and TV equipment is used to observe the seal while the machine reciprocates the glass piston.

Although currently no film thickness measurement techniques have been developed on the rig, the 2-dimensional images give enough information to provide direct evidence of many of the effects that will be discussed. The optical part of the rig is still under development to eventually include a system of measuring film thickness and be able to give detailed 3-D representations of the seal contact zone.

The main section of the rig is placed on a stage which has piezoelectric transducers attached to it at the bottom. The transducers measure the combined frictional resistance both the seals present when the glass shaft is moved. The transducers are connected to a PC via a charge amplifier and an analogue to digital converter.

The load on the seal due to the pressure is a combination of the initial seal squeeze and the pressure acting on the seal due to the fluid. This can be calculated using the Youngs Modulus of the seal. When a seal is assembled and functioning it is always in compression against the surface it is sealing, this state of compression is commonly known as "seal energisation" in industry. A seal in it's unassembled and free state is unenergised. For any sealing to occur the seal must always be energised.

2.1. Test Procedure

The experiments were carried out under room temperature and pressure conditions. The rig was pressurised using a hand pump and accumulator to pressures of up to 250psi. Some of the test parameters for the experiments are shown in table 1.

SEAL	
Seal Type (PSS Seals)	Rectangular
Modulus of elasticity	6.86 MPa
Thickness (along reciprocating direction)	3.45 mm
Depth	3.25 mm
Inner diameter	40 mm
Roughness - unworn seal (R_a)	1.48 μm
OIL PROPERTIES	
DEF STAN 91-48 Mineral oil	
Viscosity at 40 °C	13 mm^2/s
Viscostiy at 100 °C	4 mm^2/s
TEST PARAMETERS	
Max. reciprocating frequency	2 Hz
Max. reciprocating frequency	0.5 Hz
Ambient Temperature	22 °C
Stroke Length	10.5 mm

Table 1 : Test parameters for experiments

For various pressures and reciprocating speeds the friction was measured. Images were also taken of the seal in various states as direct evidence of effects that are discussed.

3. RESULTS AND DISCUSSION

Clear images showed all regions in and around the seals. Figure 2 shows the seal and it's surroundings under very low system pressure (<50psi).

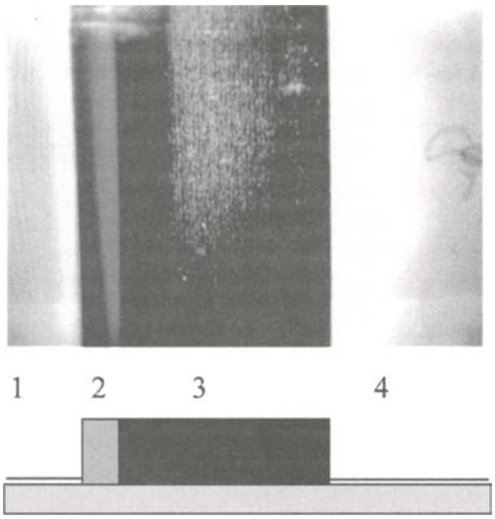

Figure 2 : Seal contact details

The photographic image shows the inner contacting face of the seal as it touches the glass tube. The boroscope looks into the tube from the inside as shown in the figure 1 schematic. The seal is held in place by the gland bearing groove, the bearing having a pale colour. The diagram below the image is a cross section of the system showing the glass tube at the bottom and the seal on top of it. An explanation of the zones in figure 3 as one goes from left to right in the image is as follows:

1. High Pressure fluid trapped by seal. The fluid appears transparent since the fluid is too shallow (0.11mm) to show it's deep red colour.
2. Fluid channel. This is a clearance between the gland bearing edge and the seal. The depth of the hydraulic fluid is 3.21mm which is enough to show the red color of the fluid.
3. Seal (width = 3.45 mm).
4. Low Pressure region i.e atmosphere.

The objective of the seal is to prevent significant flow of liquid to the atmospheric region. In the case of this seal, the fluid must be prevented from leaking from the high pressure left hand side of the image via the seal into the right hand side into the atmosphere.

It is quite interesting to note how the image shows different regions of the seal and how well the seal is "wetted". The lighter regions from the left centre to the right of the seal are regions where there is no lubricant present between the seal and the glass tube. From earlier image work it has already been found that a dry rubber/glass contact resembles the lighter regions mentioned [1]. The darker regions look "wetted" since the refractive index between wet rubber to glass and dry rubber to glass is very different. The presence of the lighter regions is therefore a very good indicator of the lack of any film in the contact. The image in figure 2 also shows how air is trapped by the seal in the contact despite the entire system being purged with hydraulic oil.

3.1. Friction Vs Pressure

Friction seemed to be higher than expected particularly during assembly of the tube into the casing. The rough rubber conforms completely to the smooth glass giving a total area of contact much higher than what would be expected if a rougher steel piston was used. The use of a smooth glass tube instead of a representative rough steel piston is the only limitation of the rig. Glass is necessary for optical data and it's use justifiable to study the overall mechanisms and problems involved in sealing.

The output of the friction data from the rig is in the form of a regular periodic variation of friction force with time for a given set of conditions, i.e a fixed pressure and speed. Note that the friction is a combination of friction from both the seals, so while one seal experiences pumping motion, the other experiences motoring. One such friction trace is presented in figure 3.

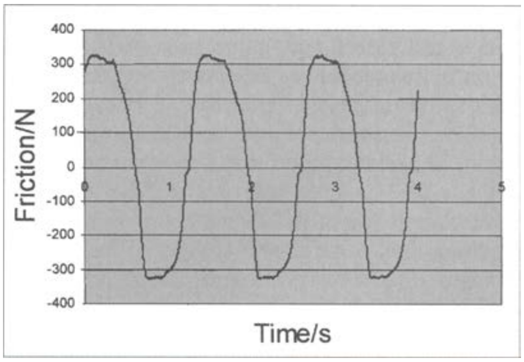

Figure 3 : Friction trace at 100psi pressure

The trace is in the form of a periodically varying force in response to the input velocity, which is sinusoidal. The output friction trace has some valuable information on the behaviour of the seal during the stroke. Before looking at the friction trace cycle it is important to define some friction terminology used in industry.

Hydraulic actuator designers, such as at SAAS-C (acknowledged at the end) usually distinguish between two types of friction encountered by the seal.

"Breakout friction" is encountered at the beginning (and end) of the cycle when the piston has to provide enough force to start the seal moving against the actuator surface. It is thought that during these conditions adhesive forces between the seal and metal interface have to be overcome and consequently a high start up force is needed. The friction in these conditions must therefore be in the dry/boundary regime then changing to mixed.

"Dynamic friction" is when the seal is in motion having already passed the breakout stage. Enough speed between the surfaces allows the development of a thick enough film to provide significant lubrication thereby lowering the friction.

As a guide to order of magnitude, the designers normally expect the breakout friction to be twice that of the dynamic friction. Breakout friction is encountered for slow speeds and small stroke actuators. Small strokes are encountered when the stroke is smaller than the width of the seal, such as in conditions where the piston may experience a fretting motion. Dynamic friction is more prevalent in a few cases in aeroplane actuators when piston speeds and stroke lengths are highest, for example in the landing gear of the aircraft.

The tests carried out were for relatively large strokes (10.5mm stroke against a 3.45mm wide seal). The friction trace does not show any evidence of significant drop in friction within one cycle or two types of friction in one cycle. This suggests that the friction is always in the breakout zone. This is expected since the glass piston is very smooth and the speeds and pressures encountered are lower than for the "dynamic friction" actuators. Unlike a steel piston surface the glass does not have any roughness valleys where fluid could be trapped and used for lubrication.

The maximum friction in the cycle encountered always remains the same and repeats periodically. To test the effect of varying the fluid system pressure on the system, a trace for each input pressure was recorded. The maximum friction from each trace was then plotted against the input pressure in figure 4.

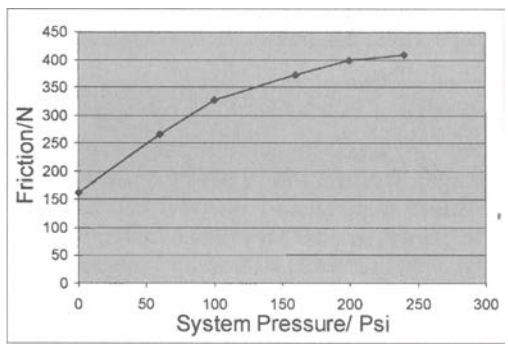

Figure 4 : Max friction for varying input pressures

Figure 4 shows that increasing the system pressure has a direct effect on increasing the friction. The curve does however trail off towards higher values. This suggests that friction does not increase linearly with the system pressure. The effect of the pressure becomes less dominant at higher values. Two effects are obvious from what should happen with increasing pressure on the seal. Firstly, increasing the pressure increases the normal load on the seal and hence the friction. Secondly, a pressure gradient encourages the formation of a film in the contact, which leads to a reduction in overall friction.

The friction is reduced since the easiest way for the fluid to enter the low pressure region is through the glass tube which entrains the fluid into the contact. Increasing the pressure therefore seems to encourage the entrainment of even more fluid and ultimately the formation of a more substantial film in the interfacial contact. An image taken of the seal during reciprocating motion at 100psi in figure 5 shows this effect clearly.

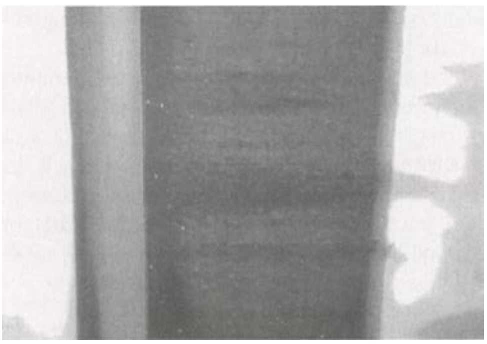

Figure 5 : Image of seal during motion

Some important findings are also apparent from comparing figures 2 and 5. Note that figures 2 and 5 are taken using slightly different optical set ups due to clarity reasons. An obvious finding is that at low or zero pressures such as in figure 2 there is hardly any film in the contact. This is because the seal already has an initial interference fit with the piston and the seal's natural tendency is to force fluid out of the contact. This was indeed observed to be the case and leaving the system under 0 pressure dwelling conditions gave rise to a virtually dry contact again. This effect has also been reported by Lawrie and O' Donoghue [8] who carried out some visualisation work and also attempted to measure film thickness.

When the seal is dynamic however, the contact image looks much more like figure 5 with the seal fully lubricated. It is interesting to note the darker thin trail near the centre of the image. The glass tube is moving from the left of the image to the right in this case (i.e pumping stroke). The darker region most probably represents a thin strip in the seal that has a thicker film across it. Local contact conditions must have encouraged this thicker film region. This effect shows that film formation in the seal is rather complex and that the film does not develop homogenously thought the entire seal. Resulting flow patterns into and out of the seal are complex too since the seal and contact pressure is not perfectly even.

The calculation of the friction coefficient proved to be difficult due to the non linear stress/strain behaviour of rubber and very small variations in the dimensions of the seal/glass/housing. A better technique is needed to calculate the friction coefficient accurately for the rig. To get an idea of the order of magnitude involved approximate friction coefficients were calculated to fall between 0.1 and 0.4. The trend being that the friction coefficient drops and steadies towards the higher pressure values as expected from the pressure/friction trace. Friction coefficients were measured previously as part of this seals study earlier [1]. The values are fairly viable since dry rubber/glass gives a friction coefficient of 0.4 representing one extreme case i.e dry friction. Wet rubber/glass gives a friction coefficient of 0.12 for this lubrication regime.

White and Denny [3] too found that the coefficient of friction drops sharply with increasing the system pressure and then steadies at higher pressures. They did not explain the reason for this effect however. From the experiments and the visual observations it is obvious that the higher pressure helps the formation of a thicker film since the friction coefficient is a good indicator of the effectiveness of lubrication.

3.2. Friction Vs Speed

The reciprocating speed of the piston was changed too. The results of varying the speed and it's effect on the friction are presented in figure 6.

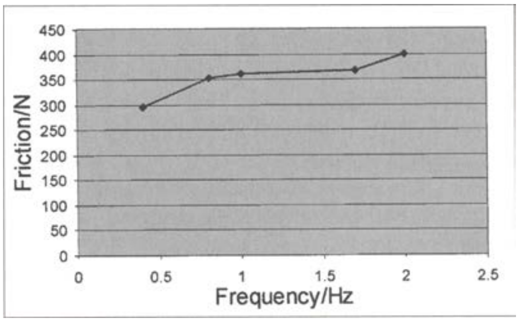

Figure 6 : Friction variation with speed

In a lubricated contact one would normally expect a trend that is explained by the Stribeck curve. The Stribeck curve normally indicates that at low speeds one expects high friction because of boundary or mixed lubrication. The friction falls with the formation of a film as the speed increases.

For the 100psi pressure involved in this test it can only be stipulated that a boundary film has already been formed by the high pressure even at the lowest reciprocating speed. Increasing the speed increases the friction for full hydrodynamic films in the Stribeck curve. It is difficult to conclude from this variation in figure 6 that the film present is indeed a hydrodynamic one due to the reasons mentioned previously, namely the high friction and complex nature of flow of fluid in the contact.

It seems that from both pressure and speed friction traces that the formation of a full hydrodynamic film where the two surfaces are completely separated and only viscous friction is encountered is difficult to achieve in practice. It is apparent from the works of White and Denny [3] and Kanters and Visccher [4] amongst others that during most conditions the seals encounter boundary/mixed lubrication. This is even more the case in these experiments since adhesive forces at the seal interface result in very conforming surface contacts due to the smooth glass surface involved. These effects are very important and have been observed practically in the "breakout friction" stage by the actuator companies.

Even for high speed actuators the speed has to build up from zero to a maximum value and the seal will always encounter these high friction conditions through the life of the seal. Long dwelling times such as when the aeroplane waits on the runway and the hydraulics are off will also mean that the seal pushes fluid away from the contact. To encourage a full film White and Denny [3] proposed to use very smooth surfaces, high viscosity oils and high speed conditions. These conditions are not always possible in real life applications.

It is apparent from these initial studies using the optical information and friction measurements that surface contact effects are very important, such effects are transient and change as the time of reciprocation processes. These effects can be expected to be prevalent through the seal's lifetime even when running in has taken place.

All these factors must be taken into account when the seals are being designed in order to maximise the all-important reliability of the seals through their expected lifetime. Models that exist based on full film lubrication [2] or flow theories must be used with these experimental findings in order to carry out a more complete study on seals.

3.3. Wear & Other effects

Given that the glass tube was extremely smooth, it wasn't intuitively expected to wear the seal as much as say a rough tube might. However wear particles were seen in the images taken even after a few stroking cycles. One such example is a visible fluid trail in figure 7.

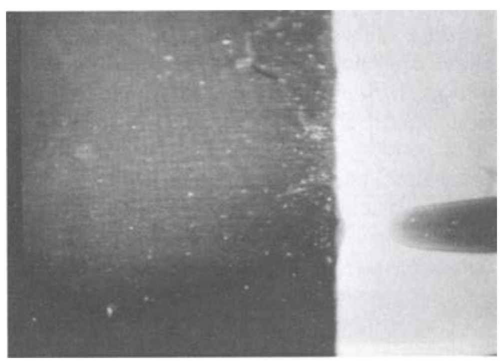

Figure 7 : Seal image and leakage trail

The image shows clearly that the fluid, which should normally appear as transparent, or a very pale shade of red has been contaminated by darker debris. The dark debris are from the seal and if one looks even closer into the image of the fluid trail, small black particles of wear can be observed.

Fluid globules were collected in the leaked atmospheric zone on the surface of the tube. This extra "reservoir" of fluid in the atmosphere could be entrained back into the contact zone encouraging fluid film formation from the atmospheric pressure side too. This interesting find was also reported by White and Denny [3] and they too concluded that fluid left on the actuator surface helps with overall lubrication. The image obtained in these experiments prove this finding effectively.

Wear also seems easy to instigate in these experiments, even with the smooth glass tube. It takes only the smallest amount of contamination to start seal wear. The majority of wear seems to take place during the start-up and slow-down of each reciprocating cycle. The only way to reduce wear is therefore to encourage the formation of a good mixed/hydrodynamic film as soon as possible in the stroke.

4. CONCLUSIONS

Experiments have been carried out using a novel optical rig that provides images of a gland seal in contact with a glass tube while simultaneously providing 2-D images of the contact itself.

When the actuator is left standing the seals natural tendency is to squeeze fluid out of the contact. Such almost dry contact conditions are encountered right when the actuator is started after a rest period. Seal friction in these cases can be expected to be maximum since surface adhesion forces have to be overcome to get relative motion.

The flow of lubricant and leakage in the seal contact zone is very complex. The conditions are not homogenous and thicker films and patches can occur at different regions in the seal. Moreover fluid that has already been entrained out of the contact can play a role in the overall lubrication process by helping motoring stroke lubrication. The overall lubrication is a result of many localised flows. The system is made more complex when wear particles are introduced.

From the friction results with varying pressure and speeds it is obvious that mixed/boundary lubrication is encountered for the range of pressures and speeds used. It has shown by other work that full film lubrication is possible using favourable conditions but the effects encountered in these experiments dominate the behaviour of the actuator seals particularly when the speeds concerned are low. Such low speeds are encountered when the actuator starts up and slows down within one cycle.

Lubrication is generally better at higher pressures so high pressure systems would give rise to a better film much better than low pressure conditions.

Wear is easy to instigate and the majority of it takes place during the relatively slow speeds encountered in these experiments. To reduce wear conditions encouraging good lubrication must be designed in to the overall actuator system.

5. ACKNOWLEDGEMENTS

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