Non-linear friction in reciprocating hydraulic rod seals: simulation and measurement

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Abstract. Non-linear seal friction can impede the performance of hydraulic actuation systems designed for high precision positioning with favourable dynamic response. Methods for predicting seal friction are required to help develop sealing systems for this type of application. Recent simulation techniques have claimed progress, although have yet to be validated experimentally. A conventional reciprocating rod seal is analysed using established elastohydrodynamic theory and the mixed lubrication Greenwood-Williamson-average Reynolds model. A test rig was used to assess the accuracy of the simulation results for both instroke and outstroke. Inverse hydrodynamic theory is shown to predict a $U^{0.5}$ power law between rod speed and friction. Comparison with experimental data shows the theory to be qualitatively inaccurate and to predict friction levels an order of magnitude lower than those measured. It was not possible to model the regions very close to the inlet and outlet due to the high pressure gradients at the edges of the contact. The mixed lubrication model produces friction levels within the correct order of magnitude, although incorrectly predicts higher friction during instroke than outstroke. Previous experiments have reported higher friction during instroke than outstroke for rectangular seals, suggesting that the mixed lubrication model used could possibly be suitable for symmetric seals, although not for seal tribology in general.

Nomenclature

- $E$: Seal elastic modulus (Pa)
- $h$: Film thickness (m)
- $h_a$: Film thickness at critical point (m)
- $h_m$: Film thickness integration constant (m)
- $I$: Influence coefficient
- $L$: Contact length (m)
- $N$: Asperity density ($m^{-2}$)
- $p_a$: Gauge pressure (Pa)
- $p_c$: Asperity contact pressure (Pa)
- $R$: Asperity radius (m)
- $U$: Rod speed ($m/s$)
- $x$: Axial coordinate (m)
- $\alpha$: Pressure-viscosity coefficient (Pa$^{-1}$)
- $H$: Dimensionless film thickness
- $H_s$: Dimensionless static film thickness
- $H_T$: Dimensionless truncated film thickness
- $P_c$: Dimensionless asperity contact pressure
- $P_f$: Dimensionless fluid pressure $p/p_a$
- $P_t$: Dimensionless total pressure $P_f+P_c$
- $\dot{x}$: Dimensionless axial coordinate $x/L$
- $\dot{\sigma}$: Dimensionless pressure-viscosity coefficient
- $\dot{\zeta}$: Dimensionless roughness $\sigma_N^{2/3}R^{1/3}$
- $\eta_0UL/(p_c\sigma^3)$: Dimensionless rod speed

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1. Introduction

Seal friction characteristics are important in hydraulic actuation systems designed to combine high precision positioning with favourable dynamic performance. Hydraulic seals typically have non-linear friction characteristics that can impose negative damping on the system. This non-linearity combined with slip-stick effects can impede how precisely an actuation system can follow changes in demand position. Accurate knowledge of seal friction behaviour can determine the precision and bandwidth limitations imposed on hydraulic systems by the sealing components. Simulation techniques for predicting seal friction can also be used to develop new seal concepts that allow improved precision and dynamic performance.

Previous investigations have had debatable success in obtaining agreement between simulations and experimental data. The most common approach has been to use elastohydrodynamic theory to predict the lubricant film thickness and calculate friction from the fluid shear stress. Several studies [1-4] have used the method of inverse solution of the Reynolds equation. An alternative approach [5] combining asperity contact with elastohydrodynamic theory was recently adapted for reciprocating seals in [6]. This mixed lubrication approach has yet to be definitively verified against experimental data.

Several studies [7-9] have measured seal friction experimentally. It is generally accepted that significant amounts of data scatter exist between different seals due to uncertainty in the manufactured geometry [10]. There is also relatively poor repeatability for successive cycles with a particular seal. The ISO 7986:1997 standard was recently established in attempt to reduce results variation between different studies. A definitive experimental study of seal friction to this standard was carried out by Parker Hannifin [11]. A few experimental studies [8, 12] have compared instroke friction to outstroke. Having the ability to measure friction in each direction of stroke would assist in validating simulations that predict direction-dependent friction characteristics.

The current study employs existing tribology models to predict the frictional behaviour of a standard U-cup type rod seal. An established method of the inverse solution of the Reynolds equation has been used. An alternative set of friction predictions has been obtained with the Greenwood-Williamson-Average Reynolds mixed lubrication model. A test rig has been produced and the seal friction forces measured for a range of operating conditions.

2. Methodology

2.1. Inverse hydrodynamic method

Most previous simulations of reciprocating seal tribology have used elastohydrodynamic theory. In this approach the opposing surfaces are assumed to be completely separated by a film of lubricant and the only source of friction is fluid shear stress. It is further assumed that the additional pressure required to deform the seal and create a channel for the fluid film is small compared with the pressure obtained when there is no fluid under the seal. This simplification is usually made due to the low elastic modulus of the seal material and low film thicknesses relative to the total seal depth.

To obtain the pressure distribution a finite element model (FEM) was produced for the geometry of a Parker B3 1624 rod seal (figure 1) using ANSYS 11. The fluid film between the rod and seal was not modelled at this stage. A uniform pressure of 80 bar was applied to the faces on the sealed fluid side of the seal and plane strain conditions were assumed. The pressure distribution obtained is shown in figure 2.
In the absence of a fluid film, the contact pressure distribution is obtained over the seal geometry from FEM. 80 bar sealed pressure.

Inverse hydrodynamic theory was used to calculate the film thickness from the pressure distribution. Fluid pressure and film thickness across the seal were assumed to be governed by the one-dimensional Reynolds equation with an exponential relationship between fluid pressure and viscosity:

\[
\frac{d}{dx}\left(\frac{h^3}{\eta_0} e^{-ap} \frac{dp}{dx}\right) = 6U \frac{dh}{dx}
\]  

A single integration produces the standard integrated form of the Reynolds equation:

\[
e^{-ap} \frac{dp}{dx} = 6U\eta_0 \left(\frac{h - h_m}{h^3}\right)
\]  

To obtain a value for \(h_m\) equation (1) is rearranged to give

\[
h^3 \frac{d}{dx} \left(\frac{1}{\eta_0} e^{-ap} \frac{dp}{dx}\right) - \frac{dh}{dx} \left(6U - \frac{3h^2}{\eta_0} e^{-ap} \frac{dp}{dx}\right) = 0
\]  

As the pressure distribution is known, a location where

\[
\frac{d}{dx} \left(\frac{1}{\eta_0} e^{-ap} \frac{dp}{dx}\right) = 0
\]  

can be identified. This point was approximately 0.09mm from the seal inlet on figure 2. At this point the local film thickness can be calculated from

**Figure 1.** Finite element discretisation of the seal in order to obtain pressure distribution in the absence of a fluid film

**Figure 2.** Contact pressure distribution obtained over the seal geometry from FEM. 80 bar sealed pressure.
Using equation (5) to substitute for the pressure gradient in equation (2) allows the constant of integration in (2) to be evaluated as $h_m = \frac{2}{3} h_a$. Equation (2) is then used to calculate the film thickness across the seal. The shear stresses exerted on the rod by the fluid are then calculated. Solutions were obtained for a range of rod speeds between $0.1 \times 10^{-3}$ m/s and 0.5 m/s (figures 4 and 5.)

### 2.2. Greenwood-Williamson-average Reynolds method

During mixed lubrication there are significant contributions to overall friction from both asperity contact and fluid viscosity. The total pressure at any point is normally assumed to be the sum of the nominal fluid and asperity contact pressures. In one approach [5] the asperity contact pressure distribution was determined from the surface mean separation using the Greenwood-Williamson (GW) contact model [13] and the fluid pressure from the solution of the Reynolds equation. This physical concept was adapted to reciprocating seals in [6] with a different numerical technique used to solve the coupled asperity and fluid equations.

For a particular film thickness distribution the fluid pressure was calculated from the average Reynolds equation using the finite difference method. A dimensionless form of the Reynolds equation was used with Patir-Cheng semi-empirical flow factors [14, 15] to account for the effects of surface roughness on flow:

$$\frac{d}{d\hat{x}} \left( \phi \alpha \frac{d}{d\hat{x}} H^3 e^{-\alpha\hat{x}} d\hat{P} \right) = 6 \zeta \left( \frac{dH_T}{d\hat{x}} + \frac{d\phi_d}{d\hat{x}} \right)$$  (6)

A Reynolds condition cavitation model was used where a pressure gradient of zero is forced at the beginning of the cavitated region.

The GW contact model was used to calculate the nominal asperity contact pressure distribution from the film thickness. GW contact theory is based on the analytical solution of contact with parabolic asperities averaged over a Gaussian distribution of asperity heights. For a particular mean film thickness the nominal asperity contact pressure is given by

$$p_c = \frac{4}{3} \left( \frac{1}{1 - \nu^2} \right) E\hat{\sigma} \frac{1}{\sqrt{2\pi}} \int_{-\infty}^{\infty} (z - H)^{3/2} e^{-z^2/2} dz$$  (7)

The additional pressure required to open the fluid channel by deflecting the seal was included in the analysis. These additional pressures were much lower than the pressures in the undeformed case (figure 2) and the final solution was relatively insensitive to these pressure additions. The relationship between pressure and seal deformation was linearised as a compliance matrix with the deflection at a particular node given by

$$H_i = H_s + \sum_{k=1}^{n} (I)_{ik} (P_i - P_{sc})_k$$  (8)

To obtain the compliance matrix of influence coefficients the stiffness matrix was first obtained from the FEM. Each node along the surface was deflected in sequence while the remaining surface nodes
were constrained in position. The stiffness matrix produced was inverted numerically to give the compliance matrix.

In the overall solver the film thickness is iterated using the successive over-relaxation method until a film thickness profile is reached where the sum of the fluid and asperity contact pressures is equal to the sum of the undeformed pressure and the additional pressure from deflecting the seal. The final converged film thickness distribution produces fluid and asperity pressures that are compatible with the film thickness given by the undeformed and deformation pressures in equation (8). The undeformed pressure distribution was obtained from FEA and was similar to that shown in figure 2. It was found necessary to remove the curvature at the corners of the seal geometry to produce a pressure distribution that would converge with the GW-average Reynolds iteration. Allowing local pressure peaks near the inlet and outlet of the pressure distribution creates local contractions in the film thickness distribution that appear to inhibit the stability of the GW-average Reynolds coupling.

2.3. Experimental friction measurement

An experimental test rig was produced to measure the friction of a Parker B3 1624 rod seal during instroke and outstroke. Complete adherence to the ISO 7986 standard for rod seal friction measurement would have required particularly long testing programmes that were not felt to be appropriate for the current project. Additionally, the relevant ISO standard does not allow for direction-dependent friction measurement which would be useful for validating current simulation techniques. Figure 3 indicates the arrangement of the components in the test rig.

![Figure 3. Test rig diagram used for measurement of rod seal friction](image)

The linear actuator was used to extend and retract the rod through the seal housing at a constant velocity. Friction force was measured from the load cell connected in series between the rod and actuator. Hydraulic fluid was supplied to the seal housing at a pressure set by a relief valve. Data were collected during instroke and outstroke for a range of constant speeds between $0.1 \times 10^{-3} \text{ m/s}$ and $0.3 \text{ m/s}$ and sealed pressures between 0 bar and 80 bar. An effective stroke length of 250 mm was used for most of the testing except at low speeds where a maximum stroke time of 15 s was set. A minimum stroke length of 5 mm was used for the lowest rod speeds.

Some post-processing of the load cell readings was required to correct for the force exerted on the end face of the rod by the pressurized fluid. The rod was retracted to a fixed position and steady state load cell readings taken for a range of sealed pressures. For all measured friction data the forces from the fluid on the rod face were interpolated from the fluid pressure measured in the housing.
3. Results

![Graph showing force vs. rod speed](image1)

**Figure 4.** Outstroke friction levels from inverse EHL solutions of Reynolds equation over contact pressure distribution at 80 bar sealed pressure.

![Graph showing force vs. rod speed](image2)

**Figure 5.** Instroke friction levels from inverse EHL solutions of Reynolds equation over contact pressure distribution at 80 bar sealed pressure.

Note close similarity between instroke and outstroke friction predictions from inverse EHL theory.

![Graph showing friction vs. rod speed](image3)

**Figure 6.** Seal friction levels predicted by Greenwood-Williamson-average Reynolds theory

![Graph showing friction vs. rod speed](image4)

**Figure 7.** Seal friction levels measured from test rig during instroke and outstroke

4. Discussion

4.1. Inverse elastohydrodynamic solution

Figures 4 and 5 suggest there to be a $U^{0.5}$ power law relationship between rod speed and friction force. Previous studies such as [4] have predicted similar friction behaviour for rectangular seals when using...
a similar method. Inspection of equation (5) indicates the film thickness at the inflexion point in the reduced pressure distribution to be proportional to $U^{0.5}$. The fluid shear stress from the Couette term in the Reynolds equation is directly proportional to rod speed and also proportional to the reciprocal of film thickness. Therefore viscous friction is expected to be proportional to $U^{0.5}$ if the film thickness $h_a$ is indicative of the mean film thickness.

Comparing the simulation results with the experimental measurements taken (figure 7) clearly indicates any simple relationship between rod speed and friction will not be possible. The experiments also showed the measured friction levels to be at least an order of magnitude greater than predicted by elastohydrodynamic theory. It is a known problem (noted in [12]) that the high measured friction levels suggest boundary lubrication is dominant while other experimental techniques suggest a fluid film is present.

The undeformed pressure distribution obtained from FEA (figure 2) is in close agreement with previous investigations [7] that have experimentally measured contact pressures for a U-cup type seal. Peaks in the pressure distribution near the inlet and outlet have been known about since the 1960s [16]. These peaks are caused by Hertzian stresses between parabolic surfaces.

One of the problems of incorporating the near-inlet and outlet peaks into the model is that the gradient of the Hertzian pressure profile approaches infinity as the outlet or inlet is approached. It is not physically realistic to have very high pressure gradients within the fluid. In practice it would be expected for there to be inlet and outlet regions near where the seal begins to rapidly recede from the surface and the fluid pressure distribution does not match the undeformed case. It is not possible to calculate the film thickness in very close proximity to the inlet and outlet using the undeformed pressure distribution. For the current study it has been assumed that the fluid behaviour in the inlet and outlet regions does not significantly affect the film thickness outside these regions.

4.2. GW-average Reynolds solution

Figure 6 indicates there to be significantly higher friction levels during instroke than during outstroke. The higher friction levels occur due to fluid cavitation extending from the outlet where (in the absence of fluid pressure) high asperity contact pressures are required to create the required total pressures. Coulomb friction from asperity shear contributes far more than viscous fluid shear to total friction.

There is a reasonable experimental agreement for the outstroke friction predicted by the GW-average Reynolds simulation. Although there is the obvious problem that the high friction levels simulated during instroke do not appear to occur in practice. Inclusion of the GW contact model allows friction levels of the correct order of magnitude to be predicted compared with pure elastohydrodynamic theory.

4.3. Experimental data and other experimental studies

There is a reasonable agreement with a previous study of a similar seal [7] for the speed dependence of friction. It is significant that there appears to be little direction dependence on the friction levels. Previous experimental studies [8, 12] that have compared instroke and outstroke behaviour have been concerned with rectangular profile seals. These studies showed rectangular seals to have significantly higher friction levels during instroke. It is unclear why rectangular seals should have quite different friction characteristics between instroke and outstroke while U-cup seals appear not to. The fact that the GW-average Reynolds model predicts higher friction during instroke suggests the technique may produce accurate results if applied to rectangular seals. For the current line of research into developing new seal designs and concepts it is not particularly useful to have a modelling technique that is only accurate for a narrow range of seal geometries. It is possible that the GW-average Reynolds method does not describe what physically happens in rectangular seals and may produce reasonably accurate friction predictions by coincidence.
5. Conclusions

Use of inverse hydrodynamic theory predicts a $U^{0.5}$ power law relating rod speed to friction. This relationship was not observed in experimental data. Elastohydrodynamic modelling predicts friction levels an order of magnitude lower than are observed in practice. This indicates conventional lubrication approaches to be unsuitable for friction modelling in elastomeric seals. It is not possible to include the regions immediately near the inlet and outlet with the method used. The GW-average Reynolds model was found to predict much higher friction for instroke than outstroke due to fluid cavitation occurring near the outlet during instroke. In cavitated regions the asperity contact pressure is higher, which increases friction through additional asperity shear.

Rod seal friction measured from the test rig showed similar speed-dependent behaviour to previous publications. There was little difference in measured friction between instroke and outstroke for the U-cup seal. Previous studies of rectangular seals have reported higher friction during instroke. It is possible that the GW-average Reynolds model may predict accurate results for rectangular seals, although not for reciprocating seals in general.

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