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Wear modeling and sensitivity analysis for seals in hydraulic cylinders.

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Abstract.

Fluid power systems are widely used in demanding and harsh environments such as marine and offshore installations, where service, maintenance, and repair costs are extremely high. This includes usage in offshore wind turbines, which impose stringent and rigorous requirements on system lifetime, reliability, and availability. As a result, accurately predicting these complex systems' lifetime, degradation patterns, and optimal service intervals is becoming increasingly important for ensuring operational efficiency and minimizing levelized cost of energy. However, the persistent absence of reliable and validated methods to accurately predict component lifetimes under realistic and variable operating conditions means that such systems are typically designed, operated, and maintained based on historical failure data, empirical observations, and conservative safety assumptions.

Among the most critical and failure-prone components in these systems are hydraulic cylinders, particularly the hydraulic seals, which are highly prone to wear and degradation over time. Previous studies have effectively combined Finite Element Analysis (FEA) with advanced numerical simulation techniques to model and predict seal wear performance under a wide variety of operating conditions. Extending on these established approaches, this study develops and applies a comprehensive sensitivity analysis examining the correlation and interaction between key influencing factors such as ambient temperature, pressure, and duty cycle, to predict seal wear.

This research incorporates wear simulation modeling in which Ansys is used to model the contact and deformation behavior, and a numerical framework is developed in MATLAB to predict fluid and asperity contact pressures, lubricant film thickness, and wear. This approach integrates detailed surface contact modeling, material properties, and realistic boundary conditions. Through a series of carefully designed simulations, this study attempts to predict the wear behavior of hydraulic seals over time, offering valuable insights that can be used for improving component performance and enhancing the overall reliability and longevity of fluid power systems in demanding operational environments.

Keywords: 1 Sensitivity analysis, 2 Wear Prediction, 3 Finite element analysis, 4 Hydraulic cylinders, 5 Numerical simulations.

1. INTRODUCTION

Reciprocating hydraulic seals serve as auxiliary components in hydraulic systems, yet they are also among the most critical parts because they prevent sealed fluid from leaking. If the seal operating mechanism is not well understood, it may be used incorrectly, potentially leading to seal failure and unpredictable consequences. White and Denny were the pioneers in analyzing the behavior and sealing efficiency of reciprocating seals between 1944 and 1947 [1] [2]. A further wave of experimental and theoretical analysis has been carried out since then.

According to experimental findings, reciprocating movement of the piston rod induces hydrodynamic forces. Two analytical methods are commonly used to study fluid film inverse hydrodynamic lubrication. The inverse hydrodynamic lubrication (IHL) approach simplifies the model by assuming full-film lubrication and exceeding the impact of fluid dynamic pressure on seal deformation, thereby limiting its effectiveness [3]. The elasto-hydrodynamic lubrication (EHL) method, however, incorporates a balance between static contact pressure and fluid pressure, which leads to a more realistic depiction of lubrication film behavior in a sealing context [4].

Among the type of seals that have been employed to analyse the lubrication characteristics, we can find O-ring seal, V- cup seals, and step seals. Previous studies on these kinds of seals have revealed that the lubrication in this interface is in the mixed lubrication state [5].

As a result of the interference fit between the seal and the inner wall of the cylinder, both surfaces initially possess fine asperities formed during the shaping and machining process. The reciprocating motion of the piston rod introduces frictional forces that not only smooth the cylinder wall but also cause migration and wear of the seal's surface profile. This wear leads to changes in the geometry of the sealing lip, which in turn, can significantly influence the integrity and performance of the sealing system [6] [7].

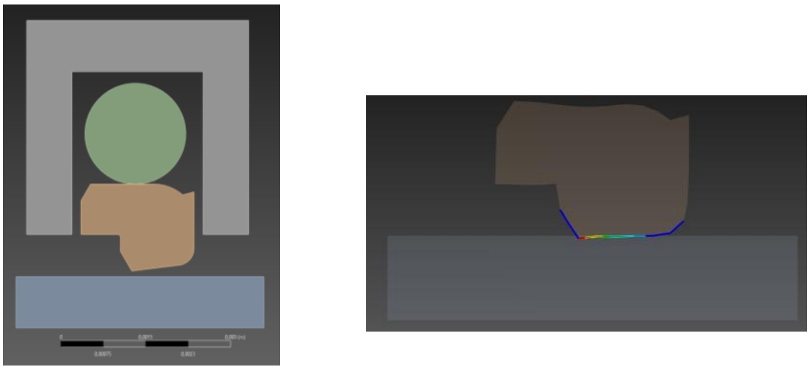
The most common kind of seal failure is wear, so it is crucial to accurately predict seal wear characteristics. For the prediction of wear in hydraulic seals, both experimental and simulation methods have been employed. Among the empirical work that has been analysed, we can find the studies of Ridgway et al. [8], which analysed the wear behaviour of the seal ring in slurry-free environment and reached the conclusion that non-uniform packing compression is not the sole factor influencing the lifetime and reliability of the gland seal. Yang et al. [9] analysed the friction behaviour of seals containing different PolyTetraFluoroEthylene (PTFE) content. Moreover, Lee et al. [10] suggested an experimental method for the accelerated wear of fluororubber (FKM) seals by introducing 0 – to 1 microns alumina particles mixed with the lubricant; nonetheless the outcome of this experiment indicated that the wear performance was similar to normal working conditions. Pan Deng et al. [11] evaluated the impact of different pressures to analyse their effect on the wear rate of PTFE piston seals. The outcome of this experiment indicated that the wear performance was similar to normal working conditions. Liu et al. [12] investigated how varying pressure levels influence the wear rate of PTFE piston seals. The outcome of these experiments indicate that wear was more pronounced at higher pressures. In addition to this, Gao et al. [13] analysed the wear behaviour of different coating seals. They found that the highest possible temperature and the increment of temperature related to the friction surface affect the wear resistance and coating durability. Furthermore, the relationship between leakage rate and skewness surface topography was explored by Ruhmani et al. [14]. Even though the tribological characteristics of a seal can be explored through experimental means, this entails manpower and resources. As a result, it has been explored to use numerical simulations to predict the wear behaviour. One of the first analytical models to predict wear on seals was implemented by Yang et al. [15]. In addition to this, Beksi et al. [16] implemented a birth-death technique on an FEM model to better study the seal wear in reciprocating seals. Moreover, other studies like the ones presented by Frolich et al. and Li et al. [17] [18] proposed methods to assess the effects of factors like friction and heat generation on wear. Nonetheless, these methods primarily examine the friction behaviour associated with the seals large–scale deformation under external compression, while neglecting the impact of microscopic factors at the lubrication interface on seal wear. The macro deformation of the seal generates heat, which in turn reduces the fluid's viscosity. Thus, the macro deformation mechanics are closely linked with the micro-hydrodynamics of the fluid film. Moreover, studies performed by Liu et al. [19] and Zhang et al. [20] employed A multiscale wear simulation approach for rotary lip seals operating under mixed lubrication conditions, considering the microscopic characteristics of the interacting surfaces. The results of these studies showed that decreasing the axial asperity density and increasing the circumferential asperity density can effectively reduce seal wear.

Even though other studies have modelled wear in hydraulic seals using various approaches, including the modelling of micro-elastohydrodynamic effects under different working conditions, no sensitivity analysis based on this approach has been conducted for hydraulic seals to examine how ambient temperature, sealing pressure, and speed during outstroke affect the wear. The purpose of this study is to describe wear using a micro-elasto hydrodynamic model that incorporates the effects of lubrication in a reciprocating seal. This is achieved by calculating the parameters present at the contact interface to account for lubrication effects and to improve the wear prediction using Archard's wear equation.

2. SEAL DESCRIPTION

This study aims to analyse the contact between a step-seal made of an O-ring and a Turcon Ring as shown in Figure 1, Typically, the O-ring is manufactured from nitrile rubber, while the Turcon ring is made from a composite material based on polytetrafluoroethylene (PTFE). When the step seal is mounted on the rod, the sealing edge forms a steep contact pressure gradient on the high-pressure side and a gentler gradient on the low-pressure side. This distribution of pressure reduces fluid adhesion to the rod during the outstroke and helps return any remaining fluid film on the rod into the system.

The reciprocating rod step seal consists of an elastomeric O-ring, commonly manufactured from nitrile rubber (NBR), and a sealing ring made of a PTFE-based composite that is in direct contact with the rod surface. The O-ring provides elastic preload and accommodates installation and pressure-induced deformations, while the PTFE sealing ring is responsible for sealing performance due to its low friction coefficient and good wear resistance. This combination of materials enables the seal to operate reliably under mixed lubrication conditions, where both fluid film lubrication and asperity contact occur during reciprocating motion.



a) b)

Figure 1. a) Ansys FEA model of the Step-seal including the O-ring, the Turcon ring and the Housing, with the direction of the stroke, where the left side is pressurised and the right side is ambient pressure. b) Sealed pressure contact distribution from Ansys FEA model, red indicates the highest contact pressure.

3. MODEL OF MIXED LUBRICATION

3.1. Numerical Simulation

This study aims to characterize wear through a micro-elastohydrodynamic model that accounts for the influence of lubrication in reciprocating seals. The model calculates key parameters related to the lubricated contact, intending to determine the asperity contact pressure for use in the Archard's wear equation. On a microscale level, the lubrication model displays robust coupling among fluid mechanics, asperity contact mechanics, deformation mechanics, and thermo-mechanics effects. Therefore, an iterative computational procedure has been built and employed to manage the coupling relationship. These relationships are described by a set of governing equations (including the Reynolds equation), which are solved using the finite element method.

The employed multiscale wear simulation model integrates a microscale mixed Thermo-Elasto-Hydrodynamic (TEHD) lubrication model with a macroscale wear model to simulate the wear behaviour of reciprocating lip seals under mixed lubrication conditions. The microscale model is designed to resolve critical tribological parameters such as lubrication film thickness, hydrodynamic pressure, and asperity contact pressure. The governing equations form a coupled multiscale framework in which each equation represents a dominant physical mechanism governing reciprocating seal wear under mixed lubrication. The Reynolds equation describes the hydrodynamic behavior of the lubricant film, while asperity contact is modeled using the Greenwood–Williamson formulation, which represents solid–solid interactions through a statistical description of surface asperities. The model links microscale contact, hydrodynamic pressures, and thermal relations account for frictional heating and its influence on lubricant viscosity. The equations are implemented through an iterative numerical scheme in which hydrodynamic and asperity contact pressures are solved simultaneously and jointly determine the film thickness. Frictional heat generation updates the lubricant viscosity and feeds back into the pressure solution until convergence is achieved at each time step. After convergence, the asperity contact pressure is incorporated into a modified Archard wear model to compute material loss. As a result, the model behavior emerges from the coupled evolution of lubrication, contact, thermal, and wear mechanisms rather than from any single equation in isolation.

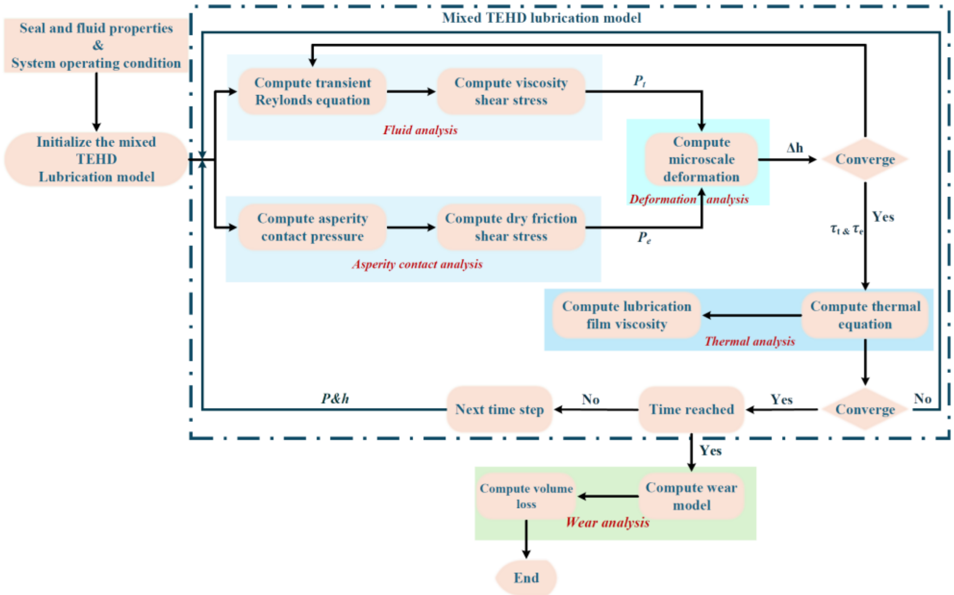


Figure 2. Schematic of the computational procedure TEHD: Thermo-elasto-hydrodynamic.

The thermal mechanics, fluid mechanics, and asperity contact mechanics, and deformation mechanics are recalculated until the lubrication film thickness, and the average temperature achieve convergence which has been explained in the following sections. Afterwards, the asperity contact pressure is used in the Archard's modified equation to calculate wear.

3.2. Mixed TEHD Lubrication Model

The multiscale wear simulation model used here incorporates a microscale mixed Thermo-Elasto-Hydrodynamic (TEHD) approach with a macroscale wear model to simulate the wear behavior of reciprocating lip seals under mixed lubrication conditions. The microscale model is designed to resolve critical tribological parameters such as lubrication film thickness, hydrodynamic pressure, and asperity contact pressure, employing the version of the Reynolds equation that considers cavitation given by Payvar et al. [21]:

$$\frac{\partial}{\partial x}\left(\phi_{xx}\frac{h^3}{\mu}\frac{\partial(F\Phi)}{\partial x}\right) 6u\frac{\partial}{\partial x}\{[1+(1-F)\Phi](h_T+\sigma\phi_{s.c.x})\} + 12\frac{\partial}{\partial t}\{[1+(1-F)\Phi](h_T+\sigma\phi_{s.c.x})\} \quad (1)$$

Where, h is the film thickness, μ is the fluid viscosity, σ is the root mean square roughness of the seal, and u is the rod speed. ϕ_{xx} is the pressure flow factor, related to the fluid obstruction due to surface roughness, and $\phi_{s.c.x}$ is the shear flow factor. This represents the extra flow transport resulting from surface sliding. F is the cavitation index and Φ is the fluid pressure or the density of the cavitation region.

In the full film region: Φ > 0, F = 1, $p_f = F\Phi,\ \rho = \rho_f$ (2)

In the cavitation region: Φ < 0, F = 0, $p_f = 0,\ \rho = \rho_f(1 + \Phi)$ (3)

When solving the Reynolds equation, it is essential to define both the inlet and outlet boundary conditions. During the outstroke and instroke phases, the fluid pressure corresponds to the sealed pressure on the fluid side and the ambient pressure on the air side, respectively.

At fluid side: $\phi\ (x = 0) = p_s$ (4)

At air side: $\phi\ (x = L_x) = p_e$ (5)

where L_x is defined as the sealing region, p_s corresponds to the pressure within the sealed area, and p_e refers to the surrounding ambient pressure.

As described by Ran et al. [22] the film thickness can be expressed as:

$$h = h_{st} + \Delta h \quad (6)$$

where h_{st} denotes the static film thickness, and Δh represents the elastic deformation of the sealing lip in the normal direction, which can be determined using the influence coefficient method. Ran et al. [22] mentioned that the mean truncated film thickness is expressed as follows:

$$h_T = \int_{-h}^{\infty} (h + \delta) f(\delta) d\delta \quad (7)$$

Where δ represents the roughness amplitude of the seal surface measured relative to the mean level. $f(\delta)$ denotes the probability density function of δ. When the asperity heights of the seal surface follow a Gaussian distribution, the mean truncated film thickness can be expressed as follows:

$$h_T = \frac{h}{2} + \frac{h}{2} erf\left(\frac{h}{\sqrt{2}}\right) + \frac{1}{\sqrt{2\pi}} e^{-\frac{h^2}{2}} \quad (8)$$

Where erf represent the error function

The shear stress resulting from the fluid viscosity can then be determined by the following relationship, as described by [23]:

$$\tau_f = \frac{-\mu u}{h}(\phi_f - \phi_{fss}) - \phi_{fpp}\frac{h}{2}\frac{dp_f}{dx} \quad (9)$$

where ϕ_f, ϕ_{fss} and ϕ_{fpp} are shear stress flow factors.

Asperity contact analysis

When the elastic seal is compressed against the rod, contact occurs at multiple microscopic asperities within the nominal contact area. The distance or separation between the seal and rod surfaces greatly affects key sealing characteristics, including leakage and friction.

The asperity contacts P_c between the seal surface and the cylinder wall can be described by the Greenwood Williamson (G–W) surface contact model [24].

$$P_c = \frac{4}{3}\frac{1}{(1-\nu_s^2)}\sigma^{3/2}R^{1/2}\eta\frac{1}{\sqrt{2\pi}}\int_H^\infty (z-H)^{3/2}e^{-z^2/2}\,dz \quad (10)$$

$$P_a = \int_d^\infty (z-d)\;\phi(z)\,dz \quad (11)$$

Where η is the seal asperity density, R is the asperity radius, and σ is the RMS roughness of the seal surface

E represents the equivalent Young's modulus, which can be expressed as

$$E = \frac{E_{seal}}{1-\upsilon_{seal}^2} \quad (12)$$

Where E_{seal} and υ_{seal} are the Young's elastic modulus and Poisson's ratio of the seal, respectively.

The dry friction shear stress τ_c is also described by the following formula found in [22]

$$\tau_c = -\frac{u}{|u|}fp_c \quad (13)$$

For this case, f is the dry friction coefficient.

Thermal analysis

Since viscosity, the key parameter in the Reynolds equation, depends on temperature, its effect on sealing performance must be considered. Friction in the sealing area raises the surface temperature, and the resulting heat is mainly dissipated through conduction and convection. Part of the heat is conducted through the seal and rod, while a smaller portion is carried away by the lubricant. Wang et al. [25] introduced a modified quasi-3D energy equation that accounts for surface roughness to analyse temperature distribution T in the fluid film. Because the lubricating layer is only a few microns thick, the heat removed by the lubricant is negligible. Salant et al. [26] proposed a simplified model assuming all heat transfers to the rod, given its higher thermal conductivity compared to the seal. Where T_e is the ambient temperature L_x is the length of the surface contact, ρ_r is the density of the rod, c_r is the specific heat capacity of the rod, k_r is the thermal conductivity of the rod.

$$T - T_e = 1.07\frac{QL_x}{2k_r}\left[\frac{\rho_r c_r |u| L_x}{2k_r}\right]^{-1/2} \quad (14)$$

$$Q = (\tau_f + \tau_c)u \quad (15)$$

Because the viscosity of the lubricating film changes with both pressure and temperature, a viscosity–pressure–temperature relationship is used to calculate the film viscosity in the sealing region, which is expressed as [25]:

$$\mu = \mu_0 exp\left\{(ln\mu_0 + 9.67) \times \left[(1 + 5.1 \times 10^{-9} p_f)^{\zeta} \times \left(\frac{T - 138}{T_e - 138}\right)^{-\psi} - 1\right]\right\} \quad (16)$$

where μ_0 represents the viscosity under the atmospheric pressure p_e and ambient Temperature T_e. The dimensionless constants ζ and ψ can be defined as:

$$\zeta = a/[5.1 \times 10^{-9}(ln\,\mu_0 + 9.67)] \quad (17)$$

$$\psi = b(T_e - 138)/(ln\,\mu_0 + 9.67) \quad (18)$$

where a denotes the Barus viscosity–pressure coefficient (Pa^{-1}) and b represents the Reynolds viscosity–temperature coefficient (K^{-1}).

Deformation analysis

For this approach, an offline FEA model is first adopted for analysing deformation mechanics. The first is the macroscopic deformation used to calculate sealed pressure distribution and contact length; the other is the microscopic parameters such as the asperity contact pressure and the fluid pressure employed for the wear calculation.

The static contact pressure, P_{st}, is assumed to be the combined effect of the hydrodynamic pressure and the asperity contact pressure, expressed as $P_{st} = P_f + P_c$ (19).

Because the lubricating film is relatively thin, the microscale deformation of the sealing lip is assumed to exhibit linear elastic behaviour. By dividing the contact length into N axial nodes, the normal deformation of the seal lip at the *i*-th node caused by the film, (the change in film thickness described in eq. (2) Δh) can be calculated by the following expression as described by [22]:

$$\Delta h_i = \sum_{j=1}^{N} C_{ij}\,(p_f + p_c - p_{st})_j \quad (20)$$

Where C_{ij} is the deformation coefficient matrix obtained from FEA.

Wear analysis

Multiple theoretical and experimental models in previous work have been done regarding material wear, such as abrasive wear and adhesive wear.

Archard's wear model is one of the most commonly employed, and widely used for the calculation of wear on seals.

For this case, the wear volume loss is considered to be linear to the sliding distance, load, and the inverse of hardness of the softer material. The original equation is expressed as:

$$V = \frac{K}{H} WS \quad (21)$$

In this scenario V is the volume loss, K is the dimensionless Archard coefficient, W is the contact force, H is the surface hardness of the softer material, and S is the sliding distance.

The wear modulus then is

$$k = K/H \quad (22)$$

Then, the Archard's wear model may be written as.

$$V = kWS \quad (23)$$

When this is divided by the contact area A, the wear film thickness can be obtained by

$$h_W = kp_nS \quad (24)$$

in which case p_n is the normal contact pressure.

The lubricating conditions present in the rod seal are of mixed lubrication. For this micro hydrodynamic lubrication model a competition between the fluid and asperity contact is considered. If lubrication is not present in sealing zone, the normal contact pressure would be equal to the static pressure. Under mixed lubricating conditions, the sealing lip is supported by both the fluid dynamic pressure and asperity contact pressure. Wear in this case is generated through asperity contact; as a result, the wear model is redefined as

$$h_w = kp_cS \quad (25)$$

Where p_c is the asperity contact pressure obtained through the mixed lubrication model.

Table 1. Main simulation parameters

Parameter	Value	Parameter	Value
Ambient Temperature (T_e)	40°C	Compression rate of seal (ε)	8 %
Sealed Pressure (p_s)	0-20 MPa	Rod diameter (d)	25.4 mm
Ambient Pressure (p_e)	0.1 MPa	Initial sectional height of seal (l)	3.18 mm
Reference Viscosity (η_0)	0.04 Pa.s	Thermal conductivity rod (k_r)	25 $W \cdot m^{-1} \cdot K^{-1}$
Density of rod (ρ_r)	7700 $kg \cdot m^{-3}$	Viscosity–temperature coefficient (β)	3.18×10^{-2} K^{-1}
Stroke length (S)	200 mm	Root mean square roughness of seal (σ)	0.6 µm
Speed of rod (u)	20–200 mm/s	Specific heat capacity of rod (c_r)	460 $J \cdot kg^{-1} \cdot K^{-1}$
Dry friction coefficient (f)	0.25	Viscosity–pressure coefficient (α)	2×10^{-8} Pa^{-1}
Young's modulus of seal (E_{seal})	380 MPa	Poisson's ratio of seal (ν seal)	0.4

Wear coefficient (k)	1.2×10^{-5} mm³/Nm		

Simulation values reported in Table 1 are based upon [19] and [22].

4 RESULTS AND DISCUSSIONS

The following results are presented showing how the asperity contact pressure varies as function of the sealed pressure at different temperatures. Figure 3 displays the curves of asperity contact pressure at different sealed pressures and temperatures.

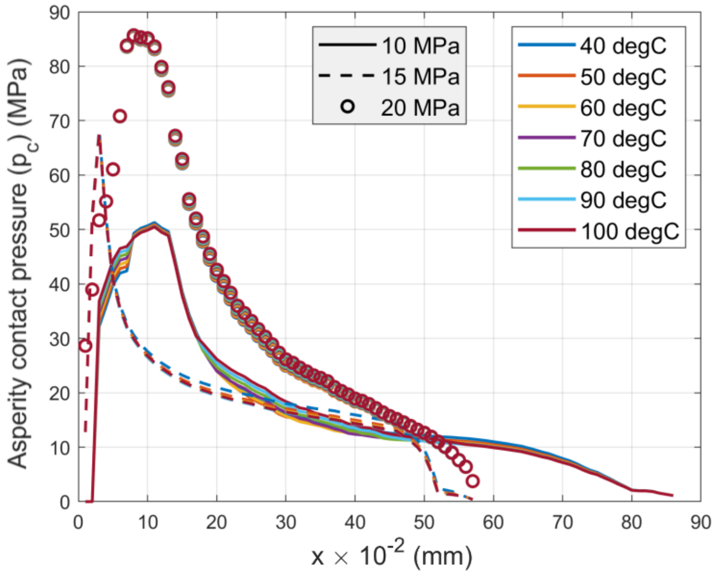


Figure 3. Curves of asperity contact pressures at various sealed pressures under varying ambient temperatures obtained from the computational procedure TEHD (thermal elasto-hydrodynamic).

Figure 3 displays the evolution of the asperity contact pressure along the sliding direction x for sealed pressures of 10, 15, and 20 MPa at ambient temperatures ranging from 40 °C to 100 °C. Generally, the asperity pressure displays a sharp initial peak near the contact entry, followed by a gradual decay towards the outlet region. This behaviour would reflect the localized concentration of contact stress at the starting edge, where the load is supported by asperities.

The decay of the asperity contact pressure follows a similar trend for all sealed pressures and temperatures, indicating that the contact mechanics remains dominated by surface topography rather than temperature-induced geometric changes. Compared with the lower sealed pressures, the overall asperity contact pressure magnitudes are higher as the sealed pressure increases, with maximum values of the asperity contact pressure approaching 65–75 MPa at a sealed pressure of 15 MPa and the asperity contact pressure would reach a maximum at 80–90 MPa at a sealed pressure of 20 MPa near the contact entrance.

Similar to the behaviour observed for each sealed pressure individually, all profiles nearly overlap through the entire contact domain, indicating that within this load range, the asperity deformation shows minor variation to ambient temperature variations. Nonetheless, when the ambient temperature rises, a slight increase in asperity contact pressure distribution can be seen.

Finally, the results indicate that the overall asperity contact pressure distributions augment when the sealed pressure increases. As the external load grows, the contact asperities experience augmented deformation, resulting in increased localized contact stresses at the edge of the contact zone.

4.1 Combined effect of sealed pressure, rod speed and ambient temperature.

The wear behaviour of the reciprocating hydraulic step rod seal and its lubrication performance are greatly influenced by the rod's operating outstroke speed. Therefore, the proposed method is used to analyse the seal wear and lubrication characteristics under various rod outstroke speeds, sealing pressures, and ambient temperatures. The results of this analysis may be seen in Figure 4. Showing the resulting wear as a function of sliding outstroke speed, temperature, and sealed pressure.

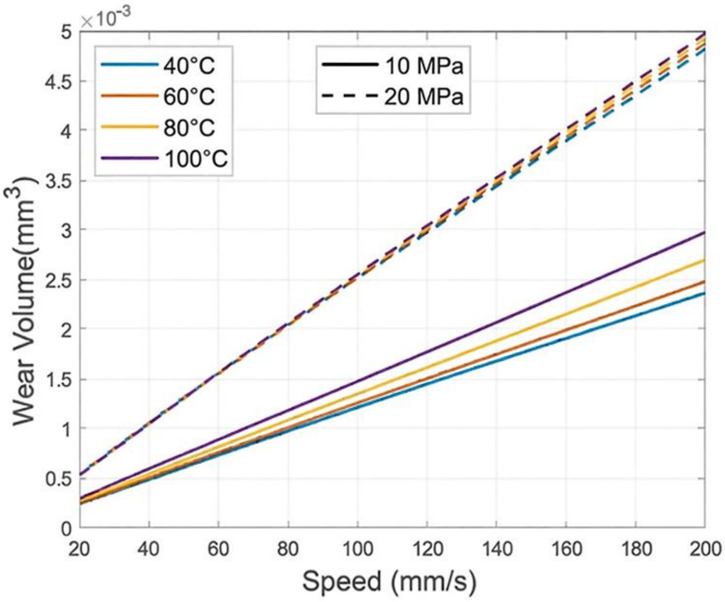


Figure 4. Wear volume per stroke of the rotary lip seal versus speed under different sealed pressures and ambient temperatures.

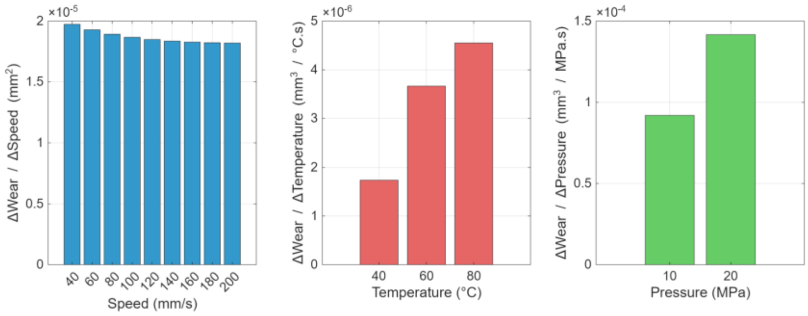


Figure 5. Sensitivity Analysis of Wear with each parameter.

The set of line plots shown in Figure 4 displays a dependence of wear volume on both parameters with different behaviour observed for each pressure condition. At both pressures, the wear volume increases with rising outstroke speed and temperature. This effect is more pronounced at 20 MPa, where wear volumes are higher all over the full range of test conditions. This suggests that the increased contact pressure would intensify the wear process, mainly due to augmented asperity deformation, and slightly worsened lubrication conditions at higher ambient temperature. However, the change in wear for both the 10 MPa and 20 MPa sealed pressures is more strongly influenced by outstroke speed than by the increase in ambient temperature. For the different speed cases analysed in this study we can observe that wear increases with rod outstroke speed. Having a maximum value of wear at a rod speed of 200 mm/s, and for the distinct sealed pressures the asperity contact profile has a low variation when the ambient temperature is changed for all the analysed sealed pressure cases. It is observed that wear is slightly higher at an increased ambient temperature of 100 degrees Celsius for both sealed pressures, with a lower variation between temperatures for the 20 MPa sealed pressure case. In Figure 5 it is observed that both speed and pressure have more or less the same influence, this confirms that controlling pressure and speed is key to reducing wear in hydraulic seals.

5 CONCLUSIONS

In this paper, a sensitivity analysis regarding accelerating wear factors, including sealed pressure, rod outstroke speed, and ambient temperature, is presented based on a wear simulation model which is developed for a reciprocating hydraulic step-rod seal under mixed lubricating conditions. From the model, it is observed that with increased sealed pressure the resulting calculated asperity contact pressure increases. In addition, ambient temperature rising from 40°C to 100°C causes a modest increase in the peak asperity contact pressure, despite the low overall variation. It is observed that with an increase in sealed pressure, the value of wear also increases, as a higher sealed pressure results in higher asperity contact pressure and increased contact between asperities. Moreover, it is observed for all sealed pressure scenarios that the calculated wear increases in an arguably mostly linear manner with the outstroke speed. The increased effect that the augmentation of both sealed pressure and operating outstroke speed had on the calculated wear could be explained by the elevated contact between asperities and the worsening lubricating conditions. It is observed that the highest accelerating factor out of the ones analysed in this study is the sealed pressure, followed by operating outstroke rod speed, whereas the ambient temperature effect has a lower effect across all operating conditions.

In conclusion, this study presents a comprehensive investigation into the wear behavior of reciprocating hydraulic step-rod seals operating under mixed lubrication conditions, with a particular focus on the sensitivity of wear to key accelerating factors: sealed pressure, rod outstroke speed, and ambient temperature. Using a simulation model grounded in Archard's wear equation, the analysis reveals that sealed pressure is the most dominant contributor to wear progression. As sealed pressure increases, the asperity contact pressure rises significantly, leading to intensified surface interactions and greater material loss. Rod outstroke speed also plays a critical role, as higher speeds exacerbate dynamic contact conditions and reduce the effectiveness of lubrication, further accelerating wear. In addition, ambient temperature directly influences wear. This impact is attributed to the thermal exchange between the environment and the lubricated contact zone, with a direct effect on viscosity, confirming ambient temperature as a contributing factor in wear.

To complement the simulation-based insights, the study also emphasizes the importance of accelerated testing protocols for evaluating seal wear characteristics under controlled yet extreme operating conditions. These tests would simulate high-pressure, high-speed, and variable-temperature environments to induce wear over shortened timeframes, allowing for rapid assessment of seal durability and performance. Accelerated testing not only validates the simulation results but also provides empirical data that can be used to refine predictive models and improve material selection and seal design. Employing simulation approaches, this study offers a robust framework for understanding and managing wear in hydraulic sealing systems. The overarching vision is to enable predictive maintenance, reduce downtime, and enhance the reliability and operational lifespan of hydraulic components. Ultimately, controlling sealed pressure and rod speed emerges as the most effective strategy for minimizing wear. This dual approach simulation-driven analysis and accelerated testing would lay the foundation for smarter, more resilient hydraulic systems in demanding industrial applications.

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