Serbian Tribology Society

SERBIATRIB '25

19th International Conference on Tribology

Faculty of Engineering University of Kragujevac

Kragujevac, Serbia, 14 – 16 May 2025

Research paper DOI:10.24874/ST.25.186

LIFETIME PREDICTION MODEL OF RECIPROCATING SEAL CONSIDERING VARIABLE SPEED PROFILE

Yunhao ZHANG[1], Chao ZHANG[1,2]*, Shaoping WANG[1,2], Rentong CHEN[1,3], Jiashan GAO[1,4]

[1] School of Automation Science and Electrical Engineering, Beijing 100191, P.R. China

[2] Ningbo Institute of Technology, Beihang University, Ningbo 315800, P.R. China

[3] State Key Laboratory of Mechanical Transmission for Advanced Equipment, Chongqing University, Chongqing 400044, P.R. China

[4] Energy Department, Politecnico di Milano, Milano 20156, Italy

*Corresponding author: cz@buaa.edu.cn

Abstract: *Reciprocating seals serve as critical components in hydraulic actuator systems, where accurate service life prediction is essential for establishing preventive maintenance schedules. In this paper, we develop a life prediction model for reciprocating seals under the variable speed profile. First, the failure mechanism model is established using a multi-field coupling method. To investigate the effect of actuator speed on seal lifespan, the life prediction model combines the Archard wear model with specific sealing pressure distribution to compare two distinct speed profiles. Finally, a case study validates the proposed model. Results demonstrate that seal wear life varies significantly when accounting for speed variations, indicating that speed change constitutes a crucial factor in life prediction. The numerical results show close agreement with ANSYS simulation data, confirming the model's validity and rationality.*

Keywords: *Reciprocating Seals; Multi-field coupling; Archard Model; Variable Speed Profile; Life Prediction*

1. INTRODUCTION

Reciprocating seals constitute a critical sealing mechanism in hydraulic actuators, engineered to maintain fluid containment integrity under dynamic pressure cycles. These components are ubiquitously deployed in mobile hydraulic systems across high-demand sectors, including aerospace flight control actuators, marine steering gear, and heavy-duty construction equipment. Failure of reciprocating seals can directly affect the service life of the equipment, and it may also lead to economic losses or even safety accidents [1]. Consequently, the development of a prognostic framework integrating failure progression dynamics becomes imperative for precision life-cycle management.

The failure mode of the reciprocating seal is primarily caused by the wear, which subsequently leads to the alteration of the sealing performance. Lengiewicz et al. [2] introduced a technique based on the finite element method to simulate the wear deformation process. Salant et al. [3] introduced numerical simulation techniques in their research to investigate the wear evolution patterns of seals and estimate their service life. Békési et al. [4] conducted wear degradation test of the reciprocating seal, obtained surface wear information using 3D white light technology, and simulated the wear process using the finite element method to derive the contour changes of the seal lip under different reciprocating cycles. Zhao et al. [5] extracted parameters that can reflect the degradation of seal performance through the sensor, and based on the fitting results, established a reliability prediction model, namely R-N curve, to describe the deterioration process of the performance of the hydraulic reciprocating seal, so as to predict its life. Ramachandran et al. [6] collected the friction data of the reciprocating seal through the experimental device, and built the life prediction model based on the hybrid method of particle swarm optimization and support vector machine, using the penalty factor and kernel parameters. However, previous studies made few contributions on the wear mechanism under variable speed profile. In addition, the contour varies with the change of actuating speed and service time, which will also result in the change of oil film state. Since speed has a nonlinear relationship with sealing performances [7], its handling significantly affects wear simulation and life prediction.

To address the problems above, we propose a life prediction model for reciprocating seal based on the modified mixed lubrication theory and Archard model under speed profile. The main contributions of this paper can be summarized as follows:

1) To describe the oil film state accurately for reciprocating seal, we propose a modified multi-field coupling model, including fluid dynamics model, surface rough contact model, sealing microscopic deformation model, and sealing interface thermal analysis model, which is greatly used to explain the oil film state.
2) To improve the accuracy of life prediction for reciprocating seal under speed profile, variable speed profiles is considered.

The rest of this paper is organized as follows. Section 2 introduces the failure mechanism and modified multi-field model. In Section 3, the calculation method of wear and the prediction method of reciprocating seal life are given. In Section 4, a simulation study is conducted to show the effectiveness of the numerical model we proposed and compared the effect of reciprocating speed profile on life span. Section 5 concludes the whole paper.

2. FAILURE MECHANISM MODEL

The reciprocating sealing system consists of a piston rod, seal, groove, and medium, as shown in Figure 1. The piston rod performs reciprocating motion [8].

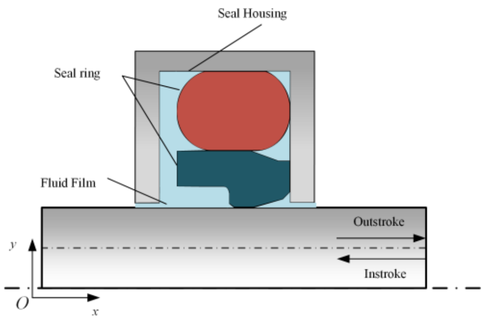

Figure 1. Working diagram of sealing ring

In order to analyse the sealing mechanism of the hydraulic rod seal, a modified multi-field coupling model is established in this paper, and its main analysis framework is shown in Figure 2.

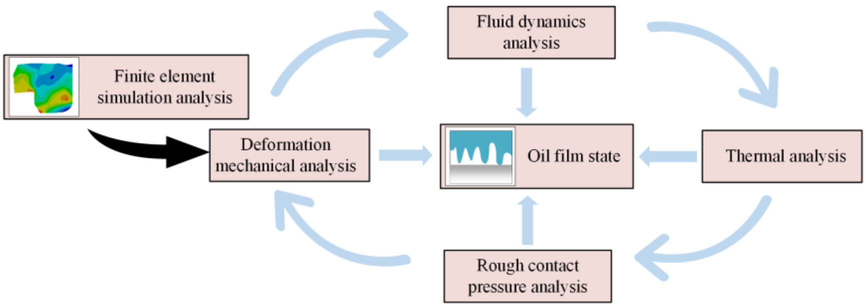

Figure 2. Multi-field coupling scheme

Fluid mechanics model for oil film is conducted based on Reynold equation. Considering the axial symmetry of the rod seal and the influence of cavitation effect and surface roughness on the fluid oil film distribution, the governing equation of the oil film can be simplified as:

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

where h represents the oil film thickness, μ represents the oil film viscosity, σ is the seal surface roughness, h_t is the average oil film thickness, ϕ_x is the pressure flow factor, $\phi_{s.c.x}$ is the shear flow factor. F denotes the cavitation factor, and Φ denotes the general variable.

According to the Greenwood-Williamson contact model, the probability p_c of contact of the micro-convex body can be expressed as:

$$p_c(z>h)=\int_h^{\infty} f(z)\mathrm{d}z \quad (2)$$

where $f(z)$ is the probability density function of the height distribution, z is the height of the micro-convex body. The total contact area A of the micro-convex body on the contact surface can be expressed as:

$$A=N\pi R\int_h^{\infty}(z-h)f(z)\mathrm{d}z \quad (3)$$

According to Hertz contact theory, the total rough contact load is calculated as:

$$W=\frac{4}{3}N\tilde{E}R^2\int_h^{\infty}(z-h)^{\frac{3}{2}}f(z)\mathrm{d}z \quad (4)$$

where N is the number of micro-convex bodies, R is the radius of micro-convex bodies, $\tilde{E}$ is the equivalent elastic modulus. The contact surface stress can be obtained by dividing the total load by the contact area.

In the thermal analysis, the change of sealing surface temperature affects thickness distribution of lubricating oil film. The viscosity-pressure temperature equation is used to calculate the viscosity of the lubricating film in the sealing area [8]:

$$\mu=\mu_0\exp\left\{\begin{matrix}(\ln\mu_0+9.67)\times \\ \left[\left(1+5.1\times10^{-9}p_f\right)^{s_1} \\ \left(\dfrac{T-138}{T_e-138}\right)^{-s_2}-1\right]\end{matrix}\right\} \quad (5)$$

where μ_0 is the oil viscosity under the conditions of air pressure and ambient temperature T_e, T_e is the ambient temperature, T is the oil film temperature, s_1 and s_2 are dimensionless parameters, and p_f is the oil pressure.

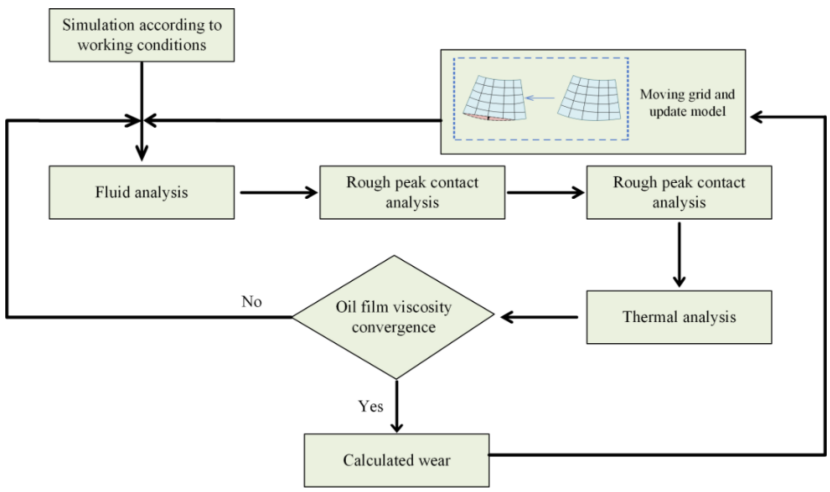

Figure 3. Simulation and calculation flow of sealed multi-field coupled model

Due to the strong coupling relationship between fluid mechanics, rough contact mechanics, sealing microscopic deformation mechanics and thermal analysis in the mixed lubrication model, a cyclic iterative method was adopted in this paper to solve the model wear, as shown in Figure 3. Through finite element analysis, the length of sealing contact area is calculated and the microscopic deformation coefficient matrix is obtained. Then Reynolds equation was used to calculate the oil film pressure distribution and rough contact model was used to calculate the rough contact pressure distribution. The microscopic deformation model is used to calculate the oil film variation, the thermal model is used to calculate the oil film temperature distribution, and the fluid viscosity in the lubrication area is calculated combined with the fluid pressure. The iteration cycle is terminated when the fluid viscosity converges.

3. LIFE PREDICTION MODEL

Speed is the key factor affecting the sealing pressure of the contact surface. This study is to consider the effect of speed change on the life of reciprocating seal.

For reciprocating seals, two speed profiles of steady state speed and segmented speed are defined here. As shown in Figure 4.

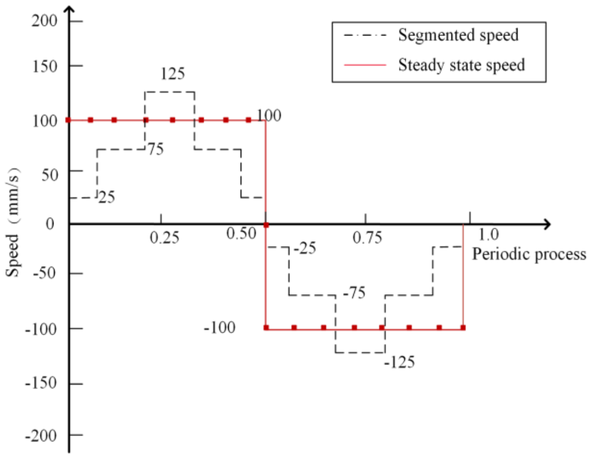

Figure 4. Unit period velocity diagram

Steady state speed: Refers to the uniform speed in the process of a single stroke inside and outside, that is, the change of speed is ignored. Using the same speed profile simulation calculation, the wear amount is calculated according to the contact pressure information of each node. This is the treatment method commonly used in current research.

Segmented speed: Considering the changing process of speed, different speed points within a single stroke are taken to represent the velocity state.

Archard model is used to calculate the amount of wear, in this model, the volume of wear is defined as a function of contact pressure, wear coefficient, material hardness, and sliding distance. As shown in formula (6) [9]

$$\Delta V = K \frac{F_N}{H} L \qquad (6)$$

where ΔV represents the volume of wear, K is the wear coefficient, F_N is the normal load, it is affected by many factors, and the method of obtaining it is shown in Figure 3. H is the material hardness, L is the sliding distance (or wear path). With the increase of the amount of wear, the contact surface stress is getting smaller and smaller, and when it is less than the oil difference on both sides, it is regarded as leakage and seal failure.

4. CASE STUDY

4.1 Parameter settings

The size of the seal ring used in the research is shown in Figure 5.

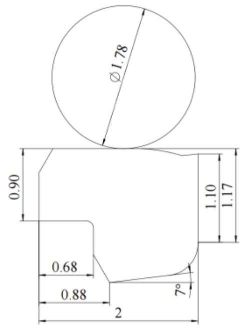

Figure 5. Sealing ring size (mm)

The oil difference between the two sides of the sealing ring is 21Mpa, so the failure threshold is 21Mpa. The reciprocating speed of 100mm/s was simulated, and the wear was calculated by obtaining the contact surface stress. The Mooney-Rivlin constitutive model of the hyper elastomer is as follows:

$$W_a = C_{10}\left(I_1 - 3\right) + C_{01}\left(I_2 - 3\right) \qquad (8)$$

where the values of C_{10} and C_{01} are shown in Table 1.

Table 1. Operating condition information

Parameter	Value	Unit
O-ring Mooney-Rivlin coefficients C_{10}, C_{01}	1.87、0.47	Mpa
PTFE seal ring elastic modulus, E_{seal}	380	Mpa
PTFE seal ring Poisson's ratio, v_{seal}	0.40	
oil pressure difference	21	Mpa
PTFE Wear coefficient, K	1.718×10^{-6}	
PTFE Material hardness, H	380	N/mm^2

4.2 Simulation and comparison of two situations

Figure 6 demonstrates the temporal stress evolution under steady-state speed conditions. During the initial state, the maximum equivalent stress reaches approximately 44.997 MPa, with the stress reduction rate during the first 2000-hour period exceeding that of subsequent phases. As shown in Figure 6(h), after 3500 hours of wear progression, the maximum lip stress decreases to 23.555 MPa, approximating half of the initial value. The compensatory behavior of the O-ring toward the wear ring ensures consistent maintenance of substantial contact width at the primary sealing surface.

Figure 7 illustrates the temporal evolution of equivalent stress on the contact surface under segmented speed conditions. The wear calculation methodology employs a time-averaging approach across the operating cycle: The total duration is distributed proportionally to individual speed points, after which segment-specific wear quantification is performed. This process involves determining asperity contact pressure through macroscopic force equilibrium analysis at each segmented speed point, calculating nodal wear depths based on corresponding duration intervals, and subsequently accumulating wear contributions from all velocity points. The resulting wear depth distribution serves as input for lip geometry updates, where the worn profile from the current cycle is iteratively applied to predict the subsequent cycle's contour.

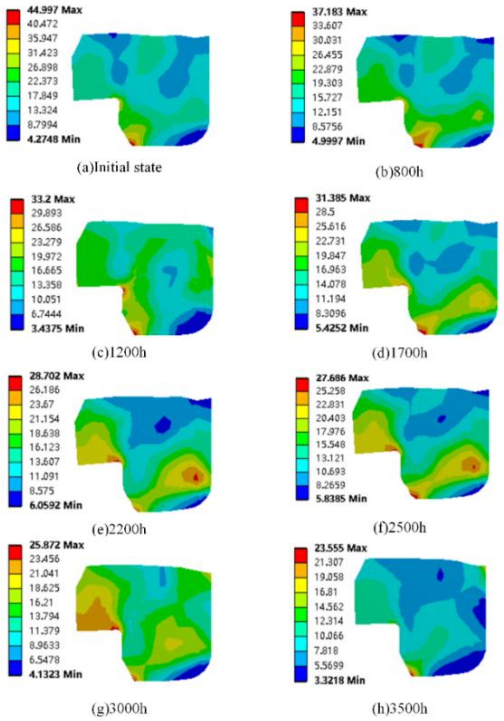

Figure 6. Variation of equivalent stress with time under steady state speed

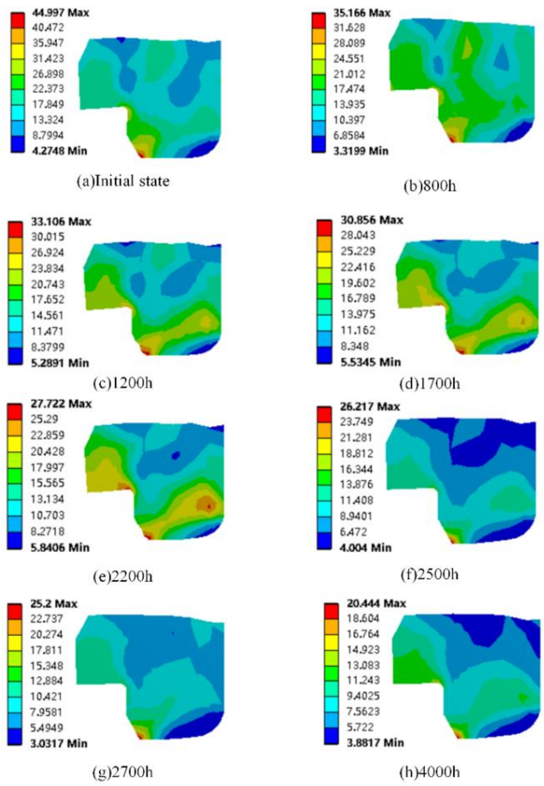

Figure 7. Variation of equivalent stress with time under segmented speed

Figure 8 shows the evolution of lip profile with wear time during 0 ~ 3000 hours under steady state speed. In order to show the change trend more clearly, the vertical coordinates of the lip contour were enlarged. The results show that the part of the lip near the oil side has greater deformation and more serious wear due to contact. With the increase of wear time, the overall wear speed decreases, the seal lip gradually tends to be flat, and the wear rate of oil side angle is significantly higher than that of air side angle.

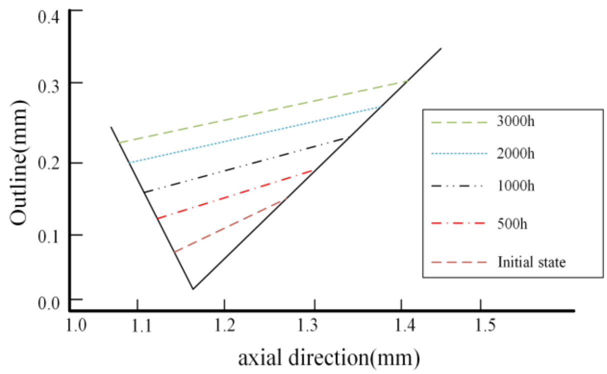

Figure 8. Contour wear degradation under steady state speed

The stress degradation curves under two different calculation methods are plotted on the same graph, as shown in Figure 9.

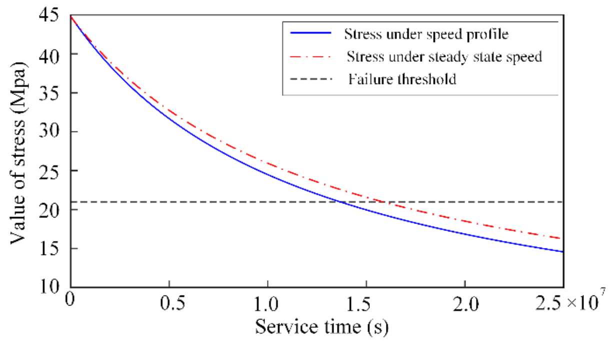

Figure 9. Stress degradation curves in both cases

It can be found that the initial state stress of the two cases is consistent. With the increase of time, the stress at steady state speed is greater than that at segmented speed. When the stress value drops to 21Mpa, the sealing ring fails. The life of the steady state speed is about 4400 hours, and the life of the segmented speed is about 3800 hours, there is a certain error. Under the two different calculation methods, the life difference is 600 hours. The main reason is that the oil film temperature changes due to the different speed profile, which affects the calculation of the wear amount. Therefore, when predicting the wear life of the sealing ring, the speed change of the actuator is an important factor to be considered in simulation.

5. CONCLUSION AND FUTURE WORK

In this paper, we developed a simulation method for predicting the wear profile life of reciprocating seals. Based on the modified multi-field model, we identified the failure mechanism of the reciprocating seal ring. To account for the effect of speed on the lifespan of reciprocating seals, we employed two speed profiles to compare lifespan predictions. The results show that the lifespan under segmented speed profiles is approximately 600 hours shorter than that under steady-state velocity conditions, which may be attributed to differences in oil film temperature. These findings suggest that even with identical surface strokes, speed variations remain a critical consideration.

For future work, the treatment of speed variations in wear life simulations could be enhanced by incorporating experimentally measured speed profiles. By selecting representative speed points from characteristic curves and refining their fitting to actual speed variations, the simulation accuracy may be improved to better approximate real-world values.

ACKNOWLEDGEMENT

The authors would like to thank the financial support from the National Natural Science Foundation of China (Grant nos. 52375036, U2233212), the Postdoctoral Fellowship Program of CPSF (Grant no. GZC20242158), the program of the China Scholarship Council (Grant no. 202406020147), Open Fund of State Key Laboratory of Mechanical Transmission for Advanced Equipment (Grant no. SKLMT-MSKFKT-202421) and the Fundamental Research Funds for the Central Universities.

ORCID iDs

Yunhao ZHANG 0009-0001-8826-5825
Chao ZHANG 0000-0001-9054-5132
Shaoping WANG 0000-0002-8102-3436
Rentong CHEN 0000-0002-3264-7189
Jiashan GAO 0009-0009-3346-0410

REFERENCES

[1] Ishiwata H, Kambayashi H. A study of oil seal for reciprocating motion. In: Proceedings of the second BHRA international conference on fluid sealing, vol. B3, 1964.

[2] LENGIEWICZ J, STUPKIEWICZ S. Continuum framework for finite element modelling of finite wear[J]. Computer Methods in Applied Mechanics and Engineering, 2012, 205: 178-188.

[3] SALANT R F, FLAHERTY A L. Elastohydrodynamic analysis of reverse pumping in rotary lip seals with microasperities[J]. Journal of Tribology, 1995, 117(1): 53-59.

[4] BEKESI N, VARADI K, FELHOS D. Wear simulation of a reciprocating seal[J]. Journal of Tribology, 2011,.133(3): 031601-1-031601-6.

[5] Xiuxu Zhao, Emmanuel Appiah, et al. Degradation process analysis and reliability prediction modeling of hydraulic reciprocating seal based on monitoring data. Engineering Failure Analysis, 140 (2022) 106565.

[6] Madhumitha Ramachandran, Jon Keegan. Hybrid PSO-SVM Based Method for Degradation Process Prediction of Reciprocating Seal. Annual Conference of the PHM Society, 11(1), September 2019, DOI: 10.36001/phmconf.2019.v11i1.852.

[7] YANG B, SALANT R F. Elastohydrodynamic lubrication simulation of O-ring and U-cup hydraulic seals[J]. Proceedings of the Institution of Mechanical Engineers Part J: Journal of Engineering Tribology, 2011, 225(7): 603-610.

[8] Field GJ, Nau BS. A theoretical study of the elastohydrodynamic lubrication of reciprocating rubber seals. ASLE Transactions 1974;18:48-54.

[9] Yang Y, Hughes WF. An elastohydrodynamic analysis of preloaded sliding seals. ASLE Transactions 1983;27:197-202.