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Elastohydrodynamic Film Formation in Multi-Element Hydraulic Rod Sealing Systems: Impact on Friction and Leakage

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Sealing systems play a critical role in hydraulic cylinders by enabling the pressure buildup required for proper system function and by preventing particles and water from entering the hydraulic circuit. The rod sealing system, in particular, is highly complex from a tribological perspective. It consists of several interacting components, including the ambient environment, hydraulic oil, housing, rod, wiper, guide band, and both primary and secondary seals. Together, these elements regulate system pressure and form the lubricating film on the rod surface.

Understanding how this lubricating film develops is essential for evaluating friction, wear, and ultimately the service life of hydraulic systems. A common method to simulate film formation is elastohydrodynamic lubrication (EHL) analysis. Although EHL simulations are well-established, only a few studies investigate the interactions of multiple sealing elements concurrently. This paper aims to provide new insights into hydraulic sealing systems by using EHL simulations to estimate lubricant film thickness and predict leakage and friction. The results indicate that the sealing contact that determines the final lubricant film height is not necessarily fixed within a sealing system but may change depending on operating conditions and the interaction of multiple sealing elements.

1 Introduction

Hydraulic rod sealing systems are essential for the reliable operation of hydraulic cylinders. They enable the functionally relevant pressure buildup, prevent hydraulic fluid from leaking to the environment and protect the system against the ingress of particles and moisture. Sealing systems have to provide low friction, sufficient lubrication and long service life under a wide range of operating conditions. Achieving this balance is challenging because sealing systems are highly complex tribological systems with interactions between the contact surfaces, seal geometry, lubricant material properties and operating conditions.

Especially relevant for assessing the performance of rod sealing systems is the lubricant film formed on the rod during the reciprocating motion. The film thickness that the sealing system leaves on the rod during the outstroke determines both friction and leakage. If more lubricant is transported out of the system during outstroke than can be returned during instroke, dynamic leakage may occur. A low film height on the rod causes high friction both during in- and outstroke. Consequently, the prediction of film formation is an important prerequisite for understanding and improving sealing system performance.

Elastohydrodynamic lubrication (EHL) simulations provide a useful approach for analyzing tribological contacts. They combine a calculation of the deformation of the contact partners with the hydrodynamic pressure build-up using the Reynolds equation. Up to now, many existing contributions focus on individual sealing elements. In real hydraulic cylinders, rod sealing systems typically consist of multiple interacting components, such as primary seal, secondary seal and wiper. The resulting film height and leakage behavior is therefore not determined by a single component alone but is the result of the interaction of the complete system.

This paper investigates the ability of an EHL simulation model to predict film formation, friction and leakage in multi-element hydraulic rod sealing systems. For this purpose, individual wipers as well as combinations of U-cup seals, wipers and tandem sealing arrangements are analyzed.

This paper aims to answer the following research questions: What predictions can be made with an EHL simulation model regarding the dynamic leakage of sealing components and sealing systems? How do operating conditions affect the interaction between different sealing elements?

2 Simulation Methodology

The simulation framework developed is a full mixed-friction elastohydrodynamic lubrication (EHL) model. The motivation for this is the inability of the previously used IHL formulation to capture asperity contact and therefore to predict friction forces reliably.

The proposed model combines three physical contributions:

- Hydrodynamic lubrication based on the Reynolds equation
- Contact mechanics of rough surfaces based on Persson theory
- Elastic deformation of the sealing contact

This combination enables the simultaneous prediction of film thickness, leakage and friction under mixed lubrication conditions typically encountered in rod seal applications.

The implementation follows a modular structure consisting of preprocessing, EHL simulation and postprocessing. The solver supports multiple contacts and both instroke and outstroke conditions.

Surface roughness is incorporated using a spectral representation based on the power spectral density (PSD) of the contacting surfaces. The PSDs of rod and seal are combined to obtain an equivalent surface description.

Contact mechanics is treated using Persson's theory for randomly rough elastic surfaces [1]. Based on this framework, the following quantities are computed:

- Pressure-dependent mean separation
- Real contact area
- Probability distribution of interfacial separations

These quantities are used to establish a pressure–separation relation and to derive homogenized flow factors for the hydrodynamic model.

Additionally, correction factors for stress and displacement based on Yang and Müser [2] are applied to improve the accuracy of the elastic contact description.

The fluid flow is described by a one-dimensional stationary Reynolds equation including roughness flow factors:

$$\frac{\partial}{\partial x}\left(\frac{h^3\phi_\mathrm{p}}{12\eta}\frac{\partial p}{\partial x}\right)-\frac{v}{2}(1+\phi_\mathrm{s})\frac{\partial h}{\partial x}=0 \quad (1)$$

where h is the local film thickness, p the fluid pressure, v the sliding velocity and η the lubricant viscosity.

Cavitation and starvation are modeled using a filling factor θ, defined as:

$$\theta=\begin{cases}1 & \text{for } p>0\\ <1 & \text{for cavitated regions}\end{cases} \quad (2)$$

leading to the modified Reynolds equation:

$$\frac{\partial}{\partial x}\left(\alpha\frac{\partial p}{\partial x}\right)-\theta\beta\frac{\partial h}{\partial x}=0 \quad (3)$$

where α and β include flow-factor corrections.

Elastic deformation of the sealing surface is modeled using a linear elastic half-space formulation:

$$h(x)=h_{FE}(x)+h_0-\frac{2x_0}{\pi E}\sum_j K_{ij}\left(p_j-p_{FE,j}\right) \quad (4)$$

where K_{ij} are influence coefficients depending on spatial distance and E is the effective modulus.

The solid contact pressure p_sol is obtained from the pressure–separation relation derived from Persson contact mechanics. The total contact load is thus composed of:

$$p_\mathrm{total}=p_\mathrm{fluid}+p_\mathrm{sol} \quad (5)$$

This formulation enables a consistent mixed-lubrication description.

The governing equations are discretized using a finite volume approach and a variable grid spacing is used to better resolve steep gradients in pressure and film thickness.

The coupled system of equations for pressure, film thickness and filling factor is solved using a monolithic Newton-type approach. The Jacobian includes:

- Pressure derivatives of the Reynolds equation
- Film thickness dependence via elastic deformation
- Cavitation constraints

Cavitation is treated using a penalized Fischer–Burmeister formulation [3], which allows a smooth transition between full film and cavitated regions.

To improve robustness, the system is scaled prior to solution, and adaptive relaxation is applied.

Finally, friction is calculated as the sum of fluid and asperity contributions:

$$\tau = \tau_{\text{fluid}} + \tau_{\text{solid}} \quad (6)$$

The fluid shear stress includes viscous and pressure-gradient terms corrected by flow factors, while the solid contribution is proportional to the local contact pressure and friction coefficient.

3 Investigated Sealing Systems

Figure 1 shows a typical rod sealing system within a hydraulic cylinder. It consists of the metal parts, rod, rod end and cylinder tube, the medium, the guide bands which prevent contact between rod and rod end, the wiper, inner and outer seal. The inner and outer seal, also referred to as primary and secondary seal, have the task to seal the pressure so that outside of the outer seal only ambient pressure occurs. The wiper controls the remaining lubricant film on the rod and prevents the ingress of particles and water.

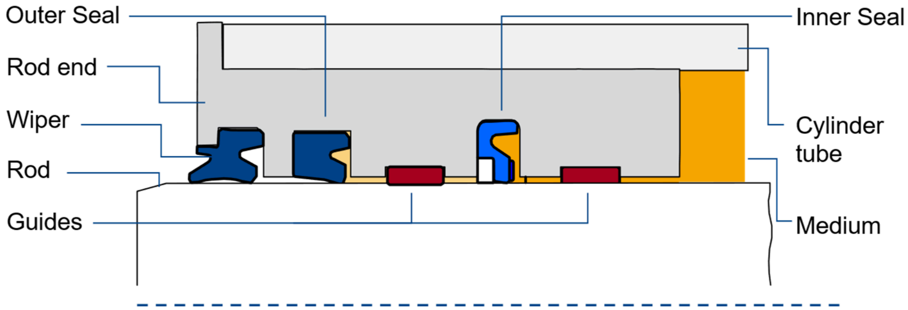

Figure 1: Typical hydraulic rod sealing system.

Figure 2 depicts the sealing systems chosen for the investigations presented in this paper. In the left column, two kinds of wipers are presented. The first one is a single lip wiper with just one contact to the rod. The second wiper is a double lip wiper with two contacts, one at the inner lip and a second one at the outer lip. In the middle column, the sealing systems consist of a U-cup seal facing the oil side and one wiper. Finally, the last system consists of two identical two-piece seals with a rubber O-ring as a preload element and a profile ring.

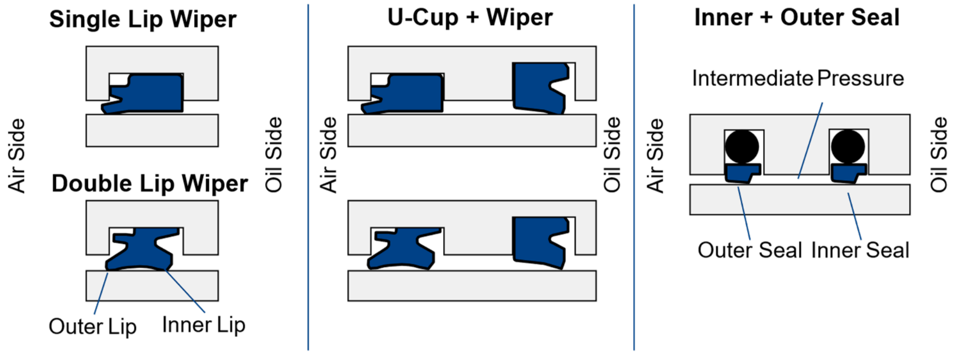

Figure 2: Sealing systems investigated in this paper.

For parametrizing the model, the following values were chosen:

- Viscosity of the oil: $0.006\ \mathrm{Pa \cdot s}$
- Rod diameter: $50\ \mathrm{mm}$
- Counter surface: surface scan of a hard chrome plated steel rod
- Sealing surface: surface scan of a polyurethane seal
- Material parameters of sealing elements:
    - U-cup, wipers and profile element: Polyurethane (hyperelastic)
    - Preload element: elastomer (hyperelastic)

As for the boundary conditions of the pressure, no static fluid pressure is applied to the wipers. For the simulations with a pressurized U-cup, the pressure always acts on the oil side. Behind the U-cup, the static fluid pressure is set to zero. For the sealing system with inner and outer seal, the full pressure is acting on the inner seal. For the outer seal, it is assumed that an intermediate pressure is acting on the seal.

For the boundary conditions of the film height, two different conditions are investigated as presented in Figure 3. For the outstroke, flooded conditions are assumed at the oil side. This means that the fluid film height is equal to the local gap height. If the EHL predicts a reduction of the film height after one of the sealing contacts, the following sealing contact receives only the film height that the previous sealing contact passed to it. The film height $h_{\mathrm{fluid,out}}$ during the outstroke is the resulting film height after passing all sealing contacts. Its value is used as an input for the starved instroke. Here, only the fluid which was transported out during the outstroke is given to the sealing system.

If one of the sealing contacts cannot return the entire film height it receives as an input into the system, flooded conditions occur at this sealing contact.

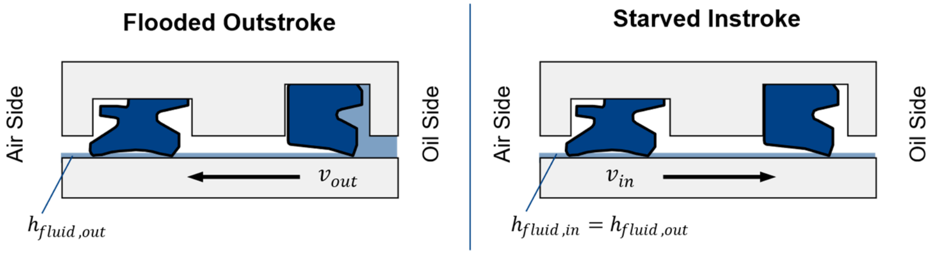

Figure 3: Depiction of the boundary conditions for in- and outstroke.

4 Simulation Results

In this section, the results of the simulations with different sealing elements and their combinations are presented. In the first subsection, a single lip and a double lip wiper are compared regarding pressure distribution and film heights.

The displayed gap heights are normalized according to the outstroke film height of a single U-cup seal at $v_{\text{out}} = 0.3\ \text{m/s}$.

4.1 Wipers

The results of the film height and pressure of the single lip wiper are shown in Figure 4. The locally averaged distance between seal and countersurface is shown as the curve h_{gap}, the local film height is shown as a surface underneath. The local fluid pressure p_{fluid} and solid contact pressure p_{sol} are depicted in the same plot.

The first plot shows the outstroke with $v_{\text{out}} = 0.3\ \text{m/s}$, where the rod moves to the left side. During this simulation, the right side is defined as fully flooded by the boundary condition. The film height after leaving the contact gets reduced to about 5.1.

For the second simulation with the single lip wiper, this film height was applied as a boundary condition on the left side for simulating the instroke at $v_{\text{in}} = 0.3\ \text{m/s}$. The purpose of this is to check whether the wiper can fully return the fluid film of the outstroke back into the system during the instroke.

Despite the starved input, the film height on the left side (i.e. on the air side of the system) increases to fully flooded conditions. This is because the wiper cannot fully return the fluid film that was transported out during the outstroke into the system. The maximum intake of the wiper is the resulting film height on the right side during the instroke, which is 2.1 and thus less than the film height that was transported out during the outstroke.

The results indicate that the simulated wiper geometry would cause leakage in fully flooded conditions with the simulated velocities, since the outstroke film height is higher than the instroke film height.

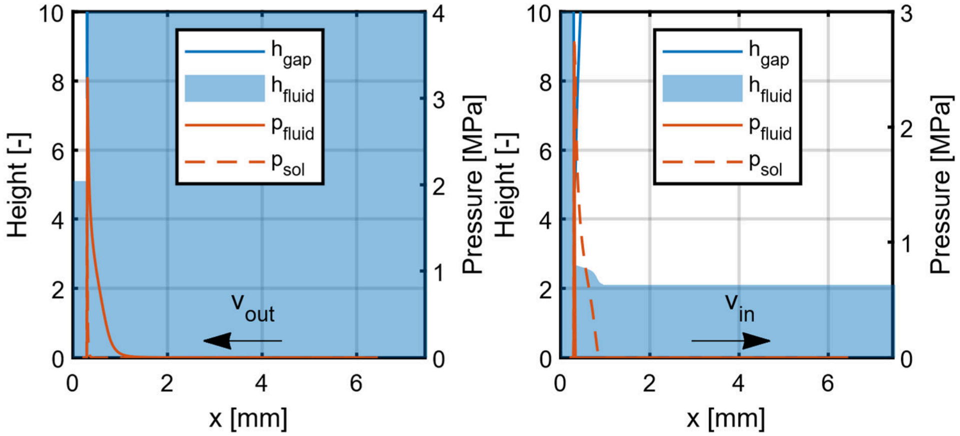

Figure 4: Film height and pressure for single lip wiper (left: outstroke, right: instroke).

For the second wiper simulation, a double lip wiper geometry was used. Like for the simulation of the single lip wiper, a fully flooded outstroke was simulated first. The resulting film height was used to model a starved instroke. For the velocities, again $v_{\text{out}} = v_{\text{in}} = 0.3\ \text{m}/s$ were chosen. The results are shown in Figure 5.

The contact stress of the inner sealing lip (right side) is considerably higher than the contact stress of the outer lip. The pressure gradient at the inner sealing lip is also considerably higher than at the outer sealing lip. For the fluid film height, the inner sealing lip causes a reduction of the fully flooded film height to 0.95 which is also the film height after the outer lip. The outer sealing lip is not lubricated by the fluid film in this case, which can be seen by the fluid pressure of zero at the outer lip.

It should be noted that there is a change of film height visible underneath the outer lip even though there is no interaction between the outer lip and the fluid. This is because of the shear flow factor in the Reynolds equation which changes according to the gap height as seen in equation (1).

For the instroke no pressure occurs at the outer lip. The film during instroke causes only pressure at the inner sealing lip. After the inner sealing lip, the film height remains at the same value as in the boundary condition. This means that the model predicts no dynamic leakage at these operating conditions for the investigated double lip wiper. Here, the outer sealing lip acts primarily as a protection against dust ingress, while the inner sealing lip controls the film height remaining on the rod.

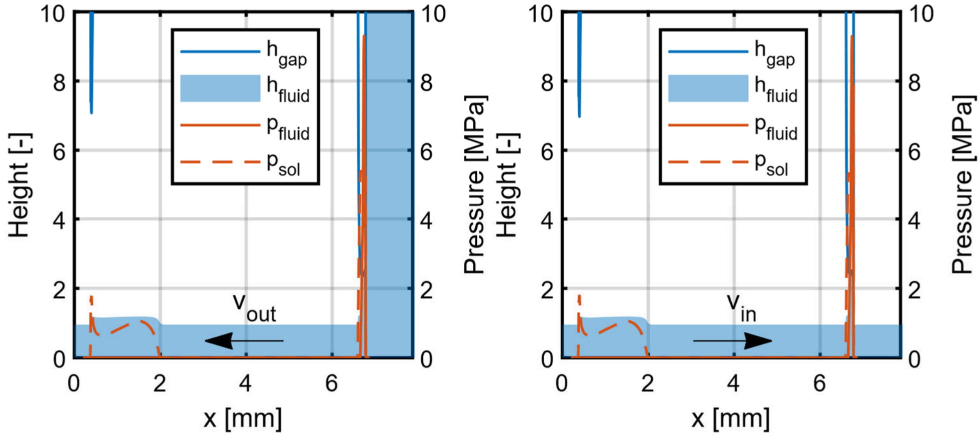

Figure 5: Film height and pressure for double lip wiper (left: outstroke, right: instroke).

4.2 U-cup Seal and Wiper

After investigating the wipers individually, the combination of a wiper seal with a U-cup seal is investigated. For that, each wiper is combined with a U-cup seal pressurized at $10\ \mathrm{bar}$.

For the combination of single lip wiper and U-cup seal, the plots are shown in Figure 6 for in- and outstroke with $v_{\mathrm{out}} = v_{\mathrm{in}} = 0.3\ \mathrm{m/s}$. Like in the previous subsection, the outstroke was simulated with fully flooded conditions on the oil side. For the instroke, starved lubrication with the film height that was transported outside was assumed.

The resulting film height during outstroke is determined by the U-cup seal. In comparison to just using the single lip wiper as shown in Figure 4, the resulting film height after the outstroke is significantly smaller so that the system can fully return the film into the system during instroke. The system with U-cup and single lip wiper thus acts similarly to the double lip wiper, where the inner contact with the higher contact pressure gradient determines the film height and the outer lip acts as a barrier against ingress of dust in these operating conditions.

Therefore, the sealing system becomes dynamically tight, because it can completely return the film of the outstroke back into the system during instroke. This indicates that a sealing system always must be assessed as the combination of its components rather than just element by element. While the wiper itself might cause leakage as a single part in fully flooded conditions, in combination with the U-cup seal the system becomes dynamically tight.

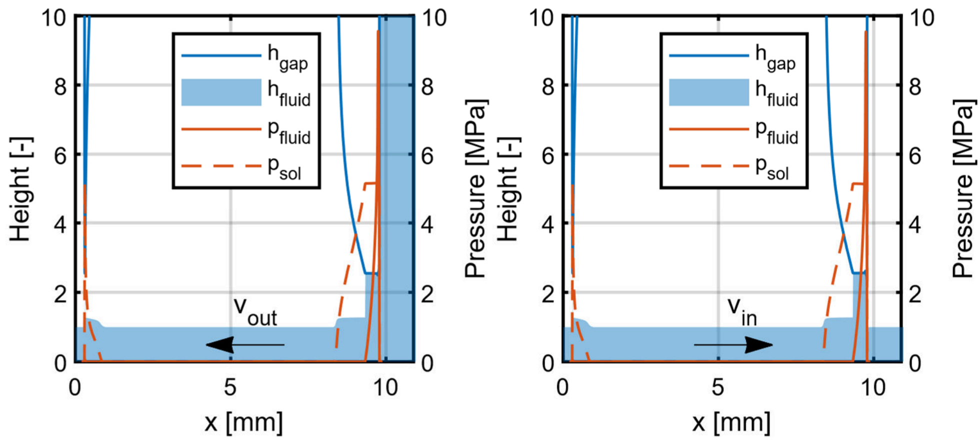

Figure 6: Film height and pressure for combination of single lip wiper and U-cup seal at an outstroke and instroke velocity of 0.3 m/s *(left: outstroke, right: instroke).*

The results of the outstroke for the combination of double lip wiper and U-cup seal are shown in Figure 7 for $v_{\mathrm{out}} = 0.5\ \mathrm{m/s}$ and $v_{\mathrm{out}} = 0.1\ \mathrm{m/s}$, respectively. The system is dynamically tight during the instroke for both velocities (not depicted here).

For this combination of U-cup and wiper, a different behavior is observed depending on the velocity. For the high velocity of $0.5\ \mathrm{m/s}$, the film height is dominated by the U-cup seal just as with the previous sealing system. Both lips of the wiper do not reduce the film height, even though under the inner sealing lip of the wiper hydrodynamic pressure occurs.

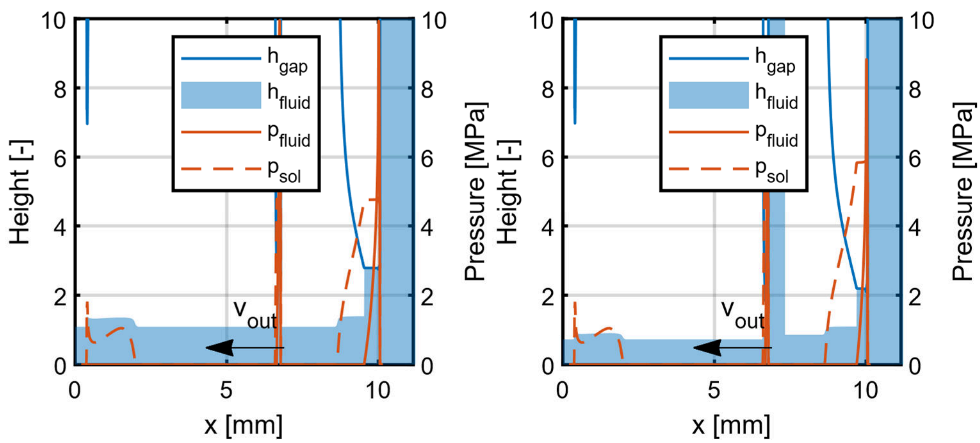

Figure 7: Film height and pressure for combination of double lip wiper and U-cup seal at an outstroke velocity of $0.5\ m/s$ (left) and $0.1\ m/s$ (right).

For the lower outstroke velocity of $0.1\ \mathrm{m/s}$, the film height after the U-cup is 0.86 and higher than the film height the inner lip of the wiper can pass through. This causes flooded conditions before the inner lip of the wiper as the fluid gathers in front of the lip. After the inner lip of the wiper, the film height is reduced to 0.73 which is lower than the film height after leaving the U-cup seal.

These results indicate the interaction of the elements within a sealing system. According to the EHL simulation, the film height is not necessarily determined by just one part for all operating conditions. Instead, the part which determines the film height can vary depending on the operating conditions, if the sensitivity of film height with respect to the velocity is different for each sealing element.

4.3 Primary and Secondary Seal

Finally, the combination of two identical sealing elements acting as a tandem system is investigated. For the sake of simplicity, two identical sealing elements, each consisting of an O-ring as a preload element and a profile ring, are used.

The inner seal is pressurized with $200\ \mathrm{bar}$, the outer seal is pressurized with an intermediate pressure of $20\ \mathrm{bar}$. The combination of starved lubrication between the seals and an assumed intermediate pressure is a simplification and may not fully represent the real behavior. Since the main purpose of this plot is to qualitatively compare film-height transport, this effect is not considered further in this contribution.

Figure 8 (left) shows the results of the outstroke with $v_{\mathrm{out}} = v_{\mathrm{in}} = 0.3\ \mathrm{m/s}$. The pressurization causes the contact pressure and its gradient to be higher at the inner seal than at the outer seal. Therefore, just like for the double lip wiper, the inner sealing contact determines the film height, while the outer sealing contact is lubricated but does not reduce the film height any further. The sealing system is dynamically tight and can fully return the film height into the system during instroke.

For investigating the influence of different intermediate pressures, the simulation was repeated with an intermediate pressure of $10\ \mathrm{bar}$ acting on the outer seal. Since the inner seal determines the film height during outstroke, the resulting film height was unchanged and the results are thus not depicted in this paper.

For investigating the influence of the film height on the friction force, the simulation was conducted five times with different instroke velocities between 0.1 and $0.5\ \mathrm{m/s}$, with a constant outstroke velocity of $0.3\ \mathrm{m/s}$. Additionally, the simulation was repeated with five outstroke velocities between 0.1 and $0.5\ \mathrm{m/s}$, conversely with a constant instroke velocity of $0.3\ \mathrm{m/s}$. The results of the friction forces during in- and outstroke are depicted in Figure 8 (right). The variation of the friction force during outstroke with different instroke velocities was not depicted. All friction forces are normalized by the outstroke friction force at $0.3\ \mathrm{m/s}$.

The results show that the instroke friction force changes by less than 5% over the investigated range of instroke velocities. In contrast, the outstroke velocity has a significantly larger influence on the instroke friction force. This indicates that the instroke friction is mainly determined by the lubricant film remaining on the rod after the outstroke. Higher outstroke velocities leave a thicker lubricant film on the rod, resulting in improved lubrication and therefore lower friction during the subsequent instroke.

Furthermore, the instroke and outstroke friction forces show an almost identical dependence on outstroke velocity. This suggests that the lubrication conditions during both stroke directions are largely controlled by the lubricant film remaining on the rod after the outstroke. As a result, changes in outstroke velocity affect both instroke and outstroke friction in a similar manner.

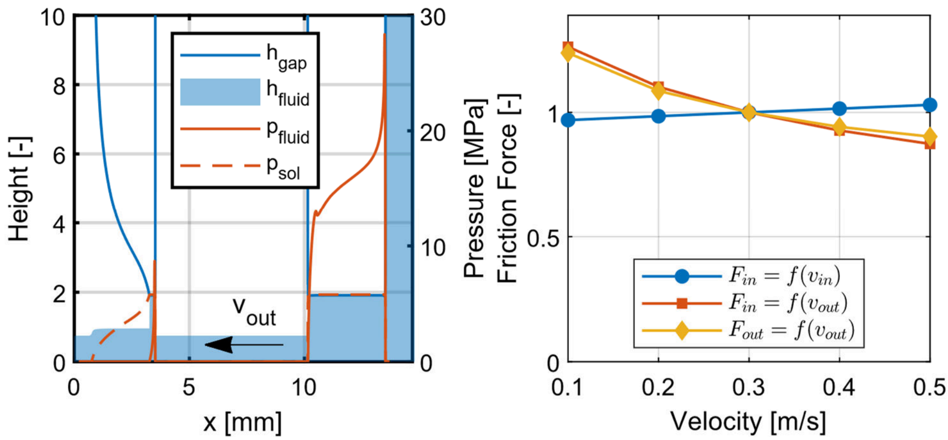

Figure 8: Film height and pressure for two seals during outstroke at an outstroke velocity of $0.3\ m/s$ *(left). Normalized friction force as a function of instroke and outstroke velocity (right).*

5 Discussion

5.1 Comparison with Experimental Results from Literature

To validate the tendency predicted by the simulation model, measured film heights for the sealing systems of U-cup and wiper from Kaiser et al [4]/ are used. They found that the combination of double lip wiper and U-cup seal was dynamically tight for all investigated operating conditions. However, for the combination of single lip wiper and U-cup seal they found leakage occurring when the outstroke velocity was at least twice as high as the instroke velocity. This behavior could not be found in the simulations made for this paper, as leakage did not occur for the said velocity ratio.

However, despite the disagreement for the combination of U-cup and wiper, the shown behavior was found for the single wipers. One possible explanation is that the current parametrization overestimates the film reduction of the U-cup in comparison to the experiment.

For investigating this discrepancy further, additional experiments and parametrizations of the simulation model, e.g. with different surfaces are planned.

5.2 General Discussion

Before concluding the findings of this paper, it should be stated that the presented results should not be generalized. The assumptions made for creating the simulation exclude some potentially relevant effects in a sealing system, which are for example:

- Wear
- Compression set
- Effect of different surface structures
- Temperature-dependent material and viscosity
- Non-rotationally symmetric systems due to tolerances or perpendicular forces
- Imperfections in the surfaces
- Ingress of particles or water

This means, based on the presented results it is not possible to fully assess the function of the outer sealing lip of the double lip wiper as the outer sealing contact did not change the occurring film height. In practical applications, there might be operating conditions, where the outer lip affects the film height or has other effects such as preventing the ingress of particles.

One possible example for the function of the outer lip can already be derived from the presented results. With the higher transport capability of the outer lip, it is possible to create a kind of reservoir between the outer and the inner sealing contact, to "buffer" operating conditions which would otherwise cause leakage. If the inner contact would not be able to fully return the film height on the rod into the system on its own, the oil film would first gather between the inner and outer sealing contact before causing leakage to the environment. As long as the transported volume of fluid does not exceed the space in between the two seals, the system would be dynamically tight.

The height of the transported film is typically in the sub-micrometer range whereas the height of the space in between the seals is in the millimeter range. This indicates that the unfavorable operating conditions would have to last for a long time in order to cause leakage.

6 Summary and Conclusion

The presented results demonstrate that the EHL simulation model can reproduce different lubrication regimes observed in hydraulic rod sealing systems. These include dynamic leakage, as demonstrated for the comparison of single lip wiper, double lip wiper, and combined sealing systems, and changes in the film-height-determining sealing contact depending on operating conditions.

The model predicts that the sealing contact determining the final lubricant film height is not necessarily fixed within a sealing system. Depending on operating conditions, different sealing elements may control the transported film height as seen for the combination of U-cup and double lip wiper. For low velocities, the wiper determines the remaining film height whereas for high velocities, the film height is determined by the U-cup.

Furthermore, the results demonstrate that sealing performance cannot be assessed reliably from individual sealing elements alone. The interaction between multiple sealing contacts can fundamentally change the resulting film height and leakage behavior.

For the tandem sealing system, the simulations indicate that both instroke and outstroke friction are strongly influenced by the lubricant film remaining on the rod after outstroke. The results suggest that the outstroke conditions have a substantially stronger effect on friction than the instroke velocity itself, as the remaining lubricant film determines the lubrication conditions during both stroke directions.

Future work will focus on experimental investigations of leakage and friction in different sealing systems to further assess the predictive capability of the model and identify opportunities for refinement and extension.

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8 Nomenclature

Variable	***Description***	***Unit***
E	Effective modulus	$[\mathrm{Pa}]$
h, h_{gap}	Local gap height	$[\mathrm{m}]$
h_{fluid}	Local fluid film height	$[\mathrm{m}]$
$h_{\mathrm{fluid,in}}$	Fluid film height before starved instroke	$[\mathrm{m}]$
$h_{\mathrm{fluid,out}}$	Resulting fluid film height after outstroke	$[\mathrm{m}]$
K_{ij}	Influence coefficient of elastic half-space formulation	$[-]$
p, p_{fluid}	Fluid pressure	$[\mathrm{Pa}]$
p_{total}	Total contact load	$[\mathrm{Pa}]$
p_{sol}	Solid contact pressure	$[\mathrm{Pa}]$
$v, v_{\mathrm{in}}, v_{\mathrm{out}}$	Sliding velocity (for instroke / outstroke)	$\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$
x	Axial coordinate	$[\mathrm{m}]$
η	Viscosity	$[\mathrm{Pa} \cdot \mathrm{s}]$
θ	Filling factor for cavitation / starvation	$[-]$

Variable	*Description*	*Unit*
Φ_{p}	Pressure flow factor	$[-]$
Φ_{s}	Shear flow factor	$[-]$
τ	Total shear stress	$[\mathrm{Pa}]$
τ_{fluid}	Fluid shear stress	$[\mathrm{Pa}]$
τ_{solid}	Solid shear stress	$[\mathrm{Pa}]$

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