Tribological analysis of the plain bearings in an external gear pump

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Abstract. Tribological loads in the plain bearings of external gear pumps are influenced by system-specific parameters (e.g. elasticity of the components, micro-geometry and manufacturing tolerances) and the operating point (rotation speed, operating pressure and temperature of the pressure medium). Due to the pressure build-up in the tooth chambers, fluctuating radial forces are acting on the gear shafts and lead to complex deformations of the bearing bushings and the housing. These deformations have an influence on the tribological loads in the plain bearings. To decrease tribological loads and failure rates, it is possible to modify the plain bearings in terms of their micro-geometry (e.g. clearance and contours) or to take constructive measures on the overall system (e.g. on the bearing bushings). Therefore, it is necessary to understand the influences of external loads, elastic environment and micro-geometry on the tribological loads in the plain bearings. This paper describes the first steps to understand these interactions by building up a simulation model of a specific external gear pump using the MBS/EHD-tool FIRST developed by IST [1]. The focus is to evaluate the required modeling depth regarding the consideration of an elastic housing.

1. Motivation and Objective
External gear pumps are one of the most robust pump types for hydraulics in mobile machinery. Increasing demands for lower emissions and higher efficiency requires the continuous development of these pumps.

The most important components of these pumps are the bearing bushings. Based on their elastohydrodynamic characteristics in the mixed lubrication regime, these bearings define significant frictional losses and the width of important sealing gaps that limit the hydraulic pump efficiency. The detailed design of gear pump bushings did already mature over decades and thus further design optimization is not an easy task at all. To exploit additional potential, it is essential to develop a deeper understanding of the overall system. This means that the influences of external loads, elastic environment and micro-geometry on the tribological loads in the radial and axial plain bearings have to be well known.

Endurance tests cause high costs and are only useful to develop a deep system understanding to a limited extent, because they do not provide an insight into the effects in the pump. A simulation model in contrast makes it possible to investigate various influences on the tribological loads in the plain bearings at comparatively low costs.

The objective of this paper is to introduce the approach for the build-up of a simulation model which helps to investigate the stated influences qualitatively for a specific external gear pump manufactured
by Bosch Rexroth. Previous work did not consider an elastic housing [2, 3]. Therefore, the focus is to investigate the influence of an elastic housing on the tribological loads in the plain bearings. Based on these investigations it is possible to evaluate the need for an elastic housing implemented in the model.

2. Loads in the analysed external gear pump
The investigations were made for an external gear pump manufactured by Bosch Rexroth with involute gear profile and single flank sealing. It is a pump with axial clearance compensation. This means that the axial sealing gap is regulated by the operating pressure. The investigated pump has a displacement of 16 cm³ per rotation. A maximum permanent operating pressure of 250 bar and an intermittent, maximum 20 s lasting operating pressure of 280 bar is permitted. The permitted rotation speed depends on the viscosity of the pressure medium and on the operating pressure. It is between 500 rpm and 3000 rpm [4]. A commonly used pressure medium (e.g. in endurance tests) is the Shell Tellus MX46 [5].

External gear pumps with axial clearance compensation according to Figure 1 mainly consist of two gear shafts, aluminium bearing bushings and an aluminium housing. One of the two gear shafts (the driving gear shaft) is directly connected to a power unit. The gear shafts are each supported by two radial and two axial plain bearings. The tooth chambers of the two meshing gears transport the fluid under increasing pressure (internal pressure field) from the suction side to the discharge side. Due to the pressure build-up in the tooth chambers, fluctuating radial forces are acting on the gear shafts [6]. These radial forces lead to the radial sealing between the teeth and the housing, see Figure 1 a). Even at high operating pressures, a sufficient axial sealing of the tooth chambers between the gear front side and the bearing bushings must be ensured. Therefore, the bearing bushings are pressurized with the operating pressure (external pressure compensation field). Due to this external pressure compensation field the bearing bushings are pressed against the gear wheels, see figure 1 b). Because of the pulsation of the operating pressure, the resulting force of the external pressure compensation field pulsates too. The area of the external pressure compensation field is sealed against the suction side by a sealing ring.

![Figure 1](image)

**Figure 1.** a) Radial forces and radial sealing b) Axial forces and axial sealing. [4]

3. Approach
As already stated, a simulation model in contrast to endurance tests offers the opportunity to develop a deeper system understanding. The focus of this paper is the investigation of the influence of an elastic housing. The MBS/EHD-tool FIRST provides a coupling between multi body dynamics and elastohydrodynamics and it allows to consider elastic components. Therefore it is a suitable tool to carry out this investigation.

The simulation model was built-up stepwise with increasing modeling depth and complexity. One advantage of this stepwise model build-up is the fact that the influence of different modeling approaches on the simulation results can be evaluated.
The following subsection gives an explanation of this stepwise model build-up. In further subsections the modeling approach of model stage 4 is described, because model stage 4 was used for the investigation related to this paper.

3.1. Stepwise model build-up

The simulation model was built-up stepwise using the MBS/EHD-tool FIRST. Figure 2 shows this stepwise model build-up with an increasing modeling depth in four model stages. In model stage 1 the simulation model contains the driven gear shaft only. The driven gear shaft is supported in radial direction by two bearing bushings. All components are modeled as rigid or elastic. The radial forces from the tooth chamber pressures and from the meshing of the gears are applied to the gear shaft. Model stage 1 just considers radial forces and the radial bearings. In model stage 2 the internal and the external pressure compensation field was added. Therefore, modeling stage 2 also considers the axial bearings. In model stage 3 the driving gear shaft with its radial and axial bearings was added to the model. Model stage 4 contains additionally the housing. To evaluate the influence of an elastic housing, the housing is modeled as rigid and alternatively as elastic, while all other components are modeled as elastic. To reduce the complexity of the simulation model in this initial approach, both housing covers as well as their bolt connection were not modeled. The forces from the bolt connection between the housing and the covers are thereby applied to the housing. The effect of the housing covers (e.g. a stiffening effect) can be analyzed in subsequent investigations.

![Figure 2](image_url)  
**Figure 2.** Stepwise model build-up in four stages.

3.2. Forces on the gear shafts

The radial forces on the gear shafts were determined with HYGESim [2, 3]. These radial forces are dependent on the rotation angle of the gear shafts and result from a superposition of two forces, see figure 3. One component results from pressure in the tooth chambers. The other component results from the meshing of the gears. The tooth contact is not in the focus of the investigations. Hence the meshing forces are applied to both gear shafts separately and the tooth contact is not modeled.

![Figure 3](image_url)  
**Figure 3.** Radial forces acting on the gear shafts, qualitative.
The forces from the meshing and from the tooth chamber pressures add up in y-direction on the driven gear shaft. Consequently, the driven gear shaft takes a higher resulting radial force than the driving gear shaft.

3.3. **Forces on the bearing bushings**

Due to the bolting of the covers against the housing, the sealing of the external pressure compensation field is preloaded. The sealing ring and the covers are not modeled. The preload force is applied to the bearing bushing in the contact area of the sealing ring, see **figure 4 a**). The operating pressure is applied to the area of the external pressure compensation field, see **figure 4 b**). As a simplification, the external pressure compensation field is thereby assumed to be constant (no pulsation). The pulsation of the internal pressure field due to the pressure build-up in the tooth chambers is considered, see subsection 3.5. The contact area of the sealing ring belongs to the area of the external pressure compensation field because the sealing ring transfers the pressure force to the bushing. Areas of the bearing bushings on the gear side which are not part of the modeled axial bearing surface are pressurized with the operating pressure too, see **figure 4 c**).

**Figure 4.** a) Contact area of the sealing ring b) Area of the external pressure compensation field c) Area with operating pressure on the gear side.

3.4. **Forces on the housing**

The operating pressure is applied to the marked surfaces of the housing, see **figure 5 a**). Investigations have shown that contact stresses between the covers and the housing are greatest in the areas around the boreholes [7]. Therefore, the forces of the housing bolt connection are largely transferred from the covers to the housing around the boreholes. These forces were applied to the marked areas in **figure 5 b**).

**Figure 5.** a) Surfaces of the housing with operating pressure b) Surfaces of the housing with forces from the bolt load.
The contact and the fit tolerance between the bearing bushings and the housing are considered. Furthermore, the flow of the pressure medium into the gap between bearing bushings and housing as well as hydrodynamic effects are considered. The bearing bushings and the housing are coupled by a calculation mesh on which the Reynolds equation is solved.

**Figure 6 a)** shows the contact surfaces on the bearing bushings and the housing. The radial forces from the pressure build-up in the tooth chambers are acting on the gear shafts in the direction of the suction side. The bearing bushings take these radial forces and consequently are pressed against the housing on the suction side. The size of the resulting contact area was evaluated in previous investigations at Bosch Rexroth. The pressure medium cannot flow into this contact area. Furthermore, there is suction pressure ($p=0$, overpressure above atmospheric pressure) on the suction side. Therefore, the contact area on the calculation mesh is bordered by a pressure boundary condition with $p=0$, see **figure 6 b**. On the discharge side a gap between the housing and the bearing bushings arises (gap area). The pressure medium flows into this gap under operating pressure. Therefore, the gap area is bordered by a pressure boundary condition with the operating pressure. To allow hydrodynamic effects (e.g. squeeze effect), no pressure boundary conditions were set inside both areas.

**Figure 6.** a) Surfaces on the housing and the bearing bushing b) Pressure boundary conditions on the calculation mesh.

### 3.5. Bearing simulation

The radial plain bearings consist of the bearing pins of the gear shafts and the bearing sleeves, see **figure 7 a**. The solution of the Reynolds equation is carried out on a cylindrical calculation mesh. The pressure medium can flow into and out of the radial bearing due to grooves under atmospheric pressure. Therefore, a pressure boundary condition with $p=0$ is applied to the nodes of both edges of the calculation mesh.

The axial plain bearings consist of the gear front side and the axial sliding surface of the bearing bushings, see **figure 7 b**. The pressures in the tooth chambers are dependent on the rotation angle $\vartheta$ of the gear shafts and were determined with HYGESim [2, 3]. The tooth chamber pressures are applied to the calculation mesh of the axial plain bearing in form of pressure boundary conditions. The node groups representing the tooth chambers are not variable. Additionally, all nodes of a certain tooth chamber receive the same pressure boundary condition at a certain rotation angle $\vartheta$. Consequently, the sealing in the tooth contact cannot be considered.
3.6. Assumptions and Simplifications

For the calculation of the hydrodynamic pressure build-up, the Reynolds equation is used. The viscosity of the pressure medium is assumed to be pressure independent. The viscosity of the pressure medium results from a temperature assumed to be constant in time and location independent. Furthermore, it is assumed that the pressure medium is free of particles and that there is no cavitation. Pressure and shear flow factors are not considered due to missing surface roughness measurements.

The rigid body contact is modeled according to Greenwood and Tripp. The roughness of the surfaces was selected within the manufacturing tolerances. Contours of the plain bearings (e.g. chamfers on the tooth edges) are not considered. However, these contours can be implemented in the model in subsequent studies. The calculation of wear is not included in the model. The stated simplifications were made to reduce the complexity of the model in this initial approach. However, FIRST provides a huge amount of functions to overcome these simplifications if necessary.

4. Results

The focus of this paper is to evaluate the need for an elastic housing implemented in the model. Therefore, model stage 4 I (rigid housing, rest elastic) and model stage 4 II (all components elastic) are compared regarding two chosen comparison parameters.

One of this comparison parameters is the relative minimum gap width \( h_{\text{min,rel}} \) in the radial and axial plain bearings. It is the minimum gap width \( h_{\text{min}} \) relative to the critical gap width \( h_{\text{crit}} \), see equation (1). The critical gap width is the gap width from which mixed friction occurs. The critical gap width is dependent on the roughness of the sliding surfaces. In FIRST, the gap width is defined as the distance between the base lines of the roughness profiles. When these base lines overlap, the gap width becomes negative. Therefore a negative gap width does not necessarily imply a penetration of the bodies.

\[
 h_{\text{min,rel}} = \frac{h_{\text{min}}}{h_{\text{crit}}} \cdot 100\%
\]  

The minimum gap width is directly related to wear in the bearing. Therefore it is a suitable parameter to investigate the tribological load in the bearing. However, the minimum gap width can occur at just one node of the calculation mesh. Consequently, the minimum gap width can be highly sensitive to variations on the model. Therefore, the resulting contact force in the plain bearings is chosen as a second comparison parameter.
The resulting contact force is the contact pressure integrated over the bearing surface. It is the proportion of the external force on the bearing that the bearing takes due to rigid body contact. Because the resulting contact force is an integral quantity, it is not as sensitive as the minimum gap width.

The investigations were carried out for the basic conditions in Table 1. As already mentioned, the driven gear shaft takes a higher resulting radial force than the driving gear shaft. Therefore, the simulation results are shown for the driven gear shaft. In the investigated operating point, a radial force up to 13.5 kN acts on the driven gear shaft. The axial bearing takes an axial force of around 7 kN.

| Table 1. Basic conditions |
|---------------------------|
| operating point           | 3000 rpm, 280 bar |
| viscosity of the pressure medium | 25 mPa·s (at 50 °C) |
| radial bearing clearance  | 4 % |

4.1. Results for the radial plain bearing

Figure 8 compares model stage 4 I and 4 II regarding the relative minimum gap width and the resulting contact force in a radial plain bearing of the driven gear shaft for one full rotation. For both model stages the relative minimum gap width is less than 100 %, see Figure 8 a). Consequently, mixed friction occurs for both model stages. Furthermore it is visible that the housing elasticity has no significant influence on the relative minimum gap width in the radial plain bearing. The same applies for the resulting contact force, see Figure 8 b). The deflection of the gear shafts shows a greater influence on the tribological load in the radial plain bearings than the deformation of the housing. As already mentioned, the calculation of wear is not included in the model. Due to a high roughness of the coating of the bearing sleeves in new condition, this simplification affects the simulation results. However, the simulation results allow to compare the effects in model stage 4 I and 4 II qualitatively.

Figure 8. Comparison of model stage 4 I and 4 II for a radial plain bearing of the driven gear shaft a) relative minimum gap width b) resulting contact force
4.2. Results for the axial plain bearing

Figure 9 shows that in contrast to the radial plain bearing the housing elasticity has a significant influence on the relative minimum gap width and the resulting contact force in the axial plain bearing. For both model stages mixed friction occurs in the axial plain bearing (the relative minimum gap width is less than 100 %), see figure 9 a). In model stage 4 II a smaller relative minimum gap width occurs, because the deformation of the elastic housing causes a different displacement and deformation of the bearing bushings compared to a rigid housing. According to figure 9 b) the resulting contact force in the axial plain bearing increases under consideration of an elastic housing.

The relative minimum gap width and the resulting contact force oscillate with the frequency of the gear meshing, because over one full rotation all twelve teeth of the gear pass the area of the minimum gap width. Figure 9 a) and b) shows a phase shift between model stage 4 I and 4 II. This occurs because the angular location of the minimum gap width changes.

![Figure 9](image)

**Figure 9.** Comparison of model stage 4 I and 4 II for an axial plain bearing of the driven gear shaft a) relative minimum gap width b) resulting contact force

It is not apparent from figure 9 a) where the minimum gap width in the axial plain bearing occurs. For this reason figure 10 shows the relative gap width in an axial plain bearing of the driven gear shaft for both model stages. The relative gap width $h_{rel}$ is defined relative to the critical gap width $h_{crit}$, see equation (2).

$$ h_{rel} = \frac{h}{h_{crit}} \cdot 100\% $$  \hspace{1cm} (2) 

The grey areas in the color-plots in figure 10 have a relative gap width greater than 100 % showing full fluid lubrication. With a rigid housing there is mixed friction on the discharge side and on the suction side with the minimum gap width appearing on the suction side. Under consideration of the elastic housing the areas of mixed friction mainly appear on the suction side. In addition the minimum gap width on the suction side is smaller in case of the elastic housing.
Figure 10. Comparison of the relative gap width in model stage 4 I and 4 II for an axial plain bearing of the driven gear shaft at a rotation angle of 0°.

According to figure 10 the elasticity of the housing leads to a shift of areas with mixed friction to the suction side and a reduction of the minimum gap width. Figure 11 compares the deformations of the bushings for both model stages. Due to the pressure forces acting from the discharge side to the suction side, the bushings are pressed towards the suction side.

In case of a rigid housing the bushings flatten on the suction side in the contact area towards the housing. This moves the minimum gap to the suction side. At the same time the bushings tilt towards the gear on the discharge side because of the external pressure field and a moment of tilt due to the radial force acting on the gear. This leads to a large area of mixed friction on the discharge side.

In case of an elastic housing the contact surface between bushing and housing tilts, see figure 11. The bushings partially follow this tilt, resulting in a further reduction of the gap width on the suction side with a simultaneous increase in gap width on the discharge side.

In addition the investigated operating point with a rotation speed of 3000 rpm reduces leakage flows from tooth chambers with high pressure to tooth chambers with low pressure. Consequently, the pressure build-up takes place at a later rotation angle. Therefore, tooth chambers on the suction side have low pressure and cannot support the axial bearing hydrostatically. The influence of the elastic housing at other operating points has not been investigated yet.
5. Summary
Electrification of mobile hydraulics and an increasing marked demand for higher operating pressures and rotation speeds make further development necessary. To ensure this, simulation-tools are intended to complement testing. This paper introduces an approach building up a simulation model of an external gear pump using the MBS/EHD-tool FIRST. The focus is to evaluate the required modeling depth regarding the consideration of an elastic housing. Therefore, the influence of the elastic housing on the tribological loads in the plain bearings was investigated for the driven gear shaft on the basis of the relative minimum gap width and the resulting contact force.

The investigation shows that the deformation of the housing has no significant influence on the tribological load in the radial plain bearing. However, a greater influence on the axial plain bearing was observed. The deformation of the elastic housing causes a different displacement and deformation of the bearing bushings compared to a rigid housing. This leads to a smaller relative minimum gap width and a higher resulting contact force and therefore a higher tribological load in the axial plain bearing.

A calculation of wear has not been included in the model yet. Therefore, it is not possible to make a statement regarding the influence of the elastic housing on the performance of the axial plain bearings. Assuming that the gap width is directly related to wear, the investigation indicates that the consideration of an elastic housing is essential to analyze the tribological load in the axial plain bearings. The stated effects are related to applications with high pressure and rotation speed. In low-pressure applications like lubrication they are negligible.

References
[1] IST: Simulationstechnische Auslegung, Optimierung und Schwachstellenanalyse thermo-elasto-hydrodynamisch gekoppelter Mehrkörpersysteme, URL: https://www.ist-aachen.de/18.11.2020
[2] Vacca A and Guidetti M 2011 Modelling and experimental validation of external spur gear machines for fluid power applications Simul. Model. Pract. Theory 19 2007–31
[3] Thiagarajan D, Vacca A and Watkins S 2019 On the lubrication performance of external gear pumps for aerospace fuel delivery applications Mech. Syst. Signal Process. 129 659–76
[4] Bosch Rexroth: External gear pump High Performance AZPF, URL: https://www.boschrexroth.com/en/xc/products/product-groups/mobile-hydraulics/pumps/external-gear-pumps/azpf 18.11.2020
[5] Shell: Tellus S2 MX 46, URL: https://www.shell-livedocs.com/data/published/de-AT/308ff126-e553-4cc6-b4be-d200538d7cf2.pdf 18.11.2020

[6] Ivantysyn J and Ivantysynova M 1993 Hydrostatische Pumpen und Motoren – Konstruktion und Berechnung (Würzburg: Vogel-Verlag) p 514

[7] Fiebig W and Heisel U 1994 Schwingungs- und Geräuschoptimierung von Hydropumpen durch Analyse des Schwingungsverhaltens von ihren Gehäusen 11. Aachener Fluidtechnisches Kolloquium 171-88