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Starting and stopping behavior of worn journal bearings

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ABSTRACT

Metal-metal contact occurs during the starting and stopping of a shaft in a lubricated journal bearing which causes continuous wear and limits the bearing lifetime. To study the starting and stopping behavior of worn journal bearings, a validated elasto-hydrodynamic simulation approach is applied. A gradually developing wear scar is predicted which is caused by repeated starting of the shaft. Wear depth, geometrical extent of the wear scar, shaft movement, contact pressure distribution and the formation of the lubrication wedge with increasing shaft speed are discussed. Furthermore, friction moment during a start-stop cycle are calculated and compared to measurement results.

1. Introduction

When the shaft rests in a journal bearing, it is completely supported by metal-metal contact. By applying a torque to accelerate the shaft, static friction has to be overcome before the shaft starts to rotate. This maximum torque is also called breakaway torque. After a short sliding, a hydrodynamic pressure establishes and lifts the shaft. The bearing operates in mixed lubrication regime until the contacting surfaces are completely separated. This behavior of journal bearings was experimentally investigated by Mokhtar et al. [1]. The authors analyzed the movement of the journal during starting and stopping on a journal bearing test-rig. During the starting, a hydrodynamic film was rapidly formed and after reaching a final running speed, the journal found its steady state operating condition. Before the shaft was completely separated from the bearing shell, a sliding motion in circumferential direction was observed with little or no initial rolling. The authors also analyzed the stopping behavior of the shaft. The shaft operated in hydrodynamic lubrication regime until the shaft stopped to rotate. After stopping the shaft followed a squeeze film trajectory to a resting position.

In a second publication, Mokhtar et al. [2] investigated the wear behavior of journal bearings exposed to repeated starting and stopping of the shaft. The authors found that the surface roughness of the bearing shell got smoother especially during the first 1000 cycles. The final surface roughness of the bearing shell approached the roughness of the hardened shaft. They also analyzed the circumferential extent of the wear scar. The location of the wear scar was shifted in the direction of shaft rotation. The authors observed that no significant wear took place during stopping of the shaft.

Bouyer and Fillon [3] measured the friction torque during start-up for different specific pressures, various radial clearance and different bearing length. The authors found a linear relation of the maximum torque in the beginning (breakaway torque) with specific pressure. The friction torque further increased with increasing roughness and was also influenced by temperature and oil feeding. The influence of lubricant additives on journal bearing performance during start-up and shut-down was experimentally analyzed by Durak et al. [4]. The authors evaluated the average friction during several start-stop cycles and presented wear characteristics for differently formulated oils.

An analytical model to investigate friction during the transient start-up of hydrodynamic journal bearings was presented by Harnoy [5]. At low start-up acceleration, the results showed extensive slip-stick behavior. At higher acceleration, slip-stick disappeared. The author further concluded that it is possible to decrease wear by applying high start-up acceleration. Monmousseau and Fillon [6] studied the transient thermo-elasto-hydrodynamic behavior of a tilting-pad journal bearing during start-up. The authors' main objective was to estimate operation conditions which ensure a safe running without seizure during the start-up. The influence of parameters like bearing clearance, feeding temperature and acceleration was analyzed. In a recent numerical study, Cui et al. [7] investigated the transient behavior of journal bearings during start-up. The contact time and bearing locus was evaluated and compared with the results obtained by Mokhtar et al. [1]. Based on the validated model, the authors discussed the influence of relative clearance and acceleration time on bearing behavior. Liu et al. [8] recently studied the start-up behavior of crankshaft main bearings. Minimum oil film thickness, hydrodynamic pressure and friction were evaluated during cranking, run-up and the transition to idle. Metal-metal contact

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was present during the early stage of start-up which has worn the bearing. With increasing oil temperature, more serious wear could be expected.

Journal bearings which are exposed to repeated start-ups show a wear scar at the highly loaded bearing region [2]. Accordingly, a cumulative procedure to calculate a wear scar during start-up and coast-down was presented by Chun and Khonsari [9]. The procedure considered rigid and perfectly aligned components. The authors showed that most of the wear occurred in a very short period after the shaft started to rotate and before the shaft came to rest.

The influence of a worn bearing geometry on journal bearing behavior has been of interest in several publications. They have in common that the bearing operation considers a steady shaft rotation. Hashimoto et al. [10,11] created a worn bearing geometry with a maximum wear depth in load direction. The authors analyzed the pressure distribution, eccentricity ratio and attitude angle under steady state and dynamic condition. Further, Fillon and Bouyer [12] analyzed the thermal behavior of a worn journal bearing. Similar to Hashimoto et al. a worn geometry was assumed. The authors concluded that wear defect could improve the thermo-hydrodynamic performance of a bearing. In particular, a lower maximum temperature was identified with increasing wear defect. A friction analysis was performed by Nikolakopoulos and Papadopoulos [13] in worn journal bearings. The authors concluded that the friction coefficient increases with increasing wear depth. Additionally, the misalignment of the journal was studied. An increase of misalignment leaded to higher power loss.

The present study focuses on the behavior of the journal bearing during starting and stopping of the shaft. Therefore, a mixed elasto-hydrodynamic simulation model is presented which considers metal-metal contact as well as the elastic deformation of bearing and shaft. The isothermal approach has extensively been validated by the authors for dynamically and statically loaded journal bearings [14–19].

The key objective is to describe the transient behavior of the journal bearing and the friction losses during a complete start-stop cycle. After an initial resting of the shaft, the shaft starts to rotate and operates in boundary, mixed or hydrodynamic lubrication regime. The calculated friction losses during one start-stop cycle are compared to results obtained from the journal bearing test-rig.

A further objective is to calculate a wear scar by performing a wear analysis which allows an iterative adaption of the bearing shell. A wear load is calculated during the start-up and a worn geometry is calculated by employing to Archards wear equation [20]. The simulation approach was previously published and validated for dynamic loads in Sander et al. [17].

With the worn bearing geometry, the hydrodynamic bearing behavior is analyzed during the start stop cycle. Furthermore, metal-metal contact pressure and contact area as well as friction losses are discussed.

2. Simulation method

The applied simulation method was previously discussed in several publications by the authors. A comprehensive summary is for instance given in Ref. [21]. For completeness of the present publication, a brief overview is recapped here.

The oil film in the journal bearing is calculated according to the modified Reynolds equation introduced by Patir and Cheng [22,23] to take rough surfaces into account. To consider cavitation in the low-pressure region, the mass conserving cavitation model based on the Jakobsson-Floberg-Olsson (JFO) approach is implemented [24]. The Reynolds equation is therefore given as:

$$-\frac{\partial}{\partial x}\left(\phi_x\theta\frac{h^3\rho}{12\bar{\eta}}\frac{\partial p}{\partial x}\right)-\frac{\partial}{\partial y}\left(\phi_y\theta\frac{h^3\rho}{12\bar{\eta}}\frac{\partial p}{\partial y}\right)+\frac{\partial}{\partial x}\left(\theta h\rho\frac{u_1+u_2}{2}\right)+\frac{\partial}{\partial x}\left(\phi_s\theta\rho\frac{u_1+u_2}{2}\sigma_s\right)+\frac{\partial}{\partial t}(\theta h\rho)=0, \quad (1)$$

where x is the circumferential and y the axial direction. p and h represent the spatially varying hydrodynamic pressure and oil film thickness. u_1 and u_2 denote the sliding speeds of the facing surfaces. The pressure flow factors ϕ_x, ϕ_y and the shear flow factor ϕ_s consider rough surfaces. The oil viscosity $\bar{\eta}$ is also a function of x and y and is dependent on the temperature, pressure and shear rate. Θ represents the fill ratio for the mass conserving cavitation model. In the cavitation region, the hydrodynamic pressure p becomes the cavitation pressure and the fill ratio is below 1. In the lubricated region (high pressure region) where p is above cavitation pressure, the fill ratio becomes 1.

Metal-metal contact occurs when the fluid film cannot separate the connecting bodies any more. Therefore, Greenwood and Tripp [25] proposed a basic contact model which specifies the metal-metal contact pressure p_a dependent on a dimensionless clearance parameter H_s:

$$p_a = KE^*F_{\frac{5}{2}}(H_s), \quad (2)$$

E^* is the composite elastic modulus and K is the elastic factor which depends on surface roughness, asperity radius and asperity density … is a form factor given by Greenwood and Tripp [25]. Surface scans of bearing shell and shaft are analyzed and the parameters for the contact model are derived (for more details see Refs. [17,19]). The obtained parameters for the contact model are given in Table 1.

2.1. Friction losses in mixed lubrication

The moment which is caused by friction losses can be calculated by integrating the hydrodynamic shear stress τ_h and the asperity shear stress τ_a over the bearing surface (A):

$$M_{Friction} = r\iint_A (\tau_h + \tau_a)dxdy, \quad (3)$$

The hydrodynamic shear stress is calculated for rough surfaces according to Patir and Cheng [22,23] who introduced the shear stress factors ϕ_f, ϕ_{fs} and ϕ_{fp}. The shear stress is then given by:

$$\tau_h = \bar{\eta}\cdot\frac{u_1-u_2}{h}(\phi_f \pm \phi_{fs}) \pm \left(\phi_{fp}\frac{h}{2}\cdot\frac{\partial p}{\partial x}\right), \quad (4)$$

where + and − refer to the shell surface and the journal surface, respectively. The shear stress caused by asperity contact is calculated by multiplying the asperity contact pressure with a boundary friction coefficient μ_{Bound}:

$$\tau_a = \mu_{Bound}\cdot p_a. \quad (5)$$

In this study, a constant μ_{Bound} of 0.02 is defined which represents a reasonable value for a lubricated contact with fully formulated engine oils containing friction modifying additives [14].

Table 1
Input parameter for contact pair.

Surface	Shell		Shaft
σ [μm]	0.25		0.13
δ [μm]	0.36		0.22
Γ [−]	1.5		4
K [−]		0.001	
E^* [GPa]		53.3	
μ_{Bound} [−]		0.02	
C [−]	10^{-6}		
H [MPa]	1470		