Simulation and Experiment study on lubrication and fatigue properties of white alloy journal bearings

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Abstract. In this paper, the elastohydrodynamic (EHD) simulation by AVL EXCITE is performed to analyze the lubrication performance including oil film thickness, oil film pressure, asperity contact pressure, and friction power losses. The journal bearing fatigue experiments were conducted on the fatigue test rig (Glacier Vandervel, made in the UK). The shell temperature was measured in real-time and the fatigue limit was finally determined. The simulation results combined with experimental results were discussed to comprehensively explain the failure mechanisms. It can be found that as the lubrication condition degrades, the asperity contact aggravates, and the shell temperature is increased sharply. This causes bearing performance degradation, poor bearing load carrying capacity, and severe surface damage, which finally lead to bearing failure.

Keywords: EHD; AVL EXCITE; Journal bearing; Lubrication performance; Fault mechanism

1. Introduction
Journal bearings are widely used due to their stability, reliability, and noiselessness, which are one of the critical elements in rotary machines. During operation, long-term high cyclic load, poor lubrication performance, and high shell temperature may become the main causes of bearing faults. It is necessary to combine the simulation and the experiment to comprehensively observe the lubrication condition and operation state to analyze the fault mechanisms to avoid as much as possible severe disasters.

Generally, the shaft and the journal bearing are both elastic bodies, and the surface of the shaft and the bearing will produce elastic deformation when subjected to force, which will affect the lubrication characteristics of the journal bearing. Therefore, the elastohydrodynamic (EHD) simulation analysis has been developed to evaluate the lubrication performance by many related studies. D.E. Sander et al. [1] and Florian König et al. [2] used the EHD simulation to evaluate the lubrication condition of journal bearing considering the lubricating oil properties, the elastic deformation, and surface topography parameters of the bearing and the shaft. C. Priestner et al. [3] and D.E. Sander et al. [4] evaluated the lubrication performance and wear behavior under different cyclic loads by EHD simulation. H. Allmaier et al. [5] used the EHD analysis to investigate the influence of the shell temperature on the lubrication condition. Besides, the authors [6] also applied the EHD analysis to study the relationship between the lubrication condition and friction behavior.
However, although most studies are widely proposed for the journal bearing lubrication performance evaluation, few researchers have verified the simulation results through experiments to comprehensively discuss the internal damage mechanism and failure causes. In this paper, the EHD analysis is firstly achieved to evaluate the journal bearing lubrication condition. Then the fatigue experiment is conducted to observe the surface damage phenomena. Based on the simulation and experiment results, the damage mechanisms of journal bearings are discussed to provide technical support for the actual application of failure analysis.

2. Theory

2.1. Averaged Reynolds equation

The full oil film interaction is solved by the averaged Reynolds equation, which considers the influence of roughness amplitudes. The specific formula is as follows [7]:

$$\frac{\partial}{\partial x} \left( h^T \frac{\partial P}{\partial x} + \frac{u_x + u_r}{2} h^T \right) + \frac{\partial}{\partial y} \left( h^T \frac{\partial P}{\partial y} \right) + \frac{\partial}{\partial t} h^T = 0$$

(1)

where \( h^T \) denotes the real clearance considering roughness heights of journal and bearing. \( P \) is the hydrodynamic pressure and \( x, y \) represents the circumferential and axial direction respectively. \( H \) indicates the oil viscosity.

2.2. Lubricating oil properties

The Vogel equation [8] is applied to describe the relationship between the oil viscosity and oil temperature in the simulation model:

$$\eta(T) = Ae^{B/T + C}$$

(2)

where \( A=0.198 \text{ mPa} \cdot \text{s} \), \( B=736.69 \text{ °C} \), \( C=88.9 \text{ °C} \) for oil Shell Rimula R3 Multi 10W-30 in this paper. \( T \) is the oil temperature.

2.3. Asperity contact interaction

Asperity contact interaction exists when asperity summits of two nominal flats rub each other. The Greenwood and Tripp model is used to evaluate the behavior of asperity contact. The asperity contact pressure is calculated as [9]:

$$P_a = KE_F S_{a/2} (H_s)$$

(3)

where the elastic factor \( K \) is set to 0.003. \( F_{a/2}(H_s) \) stands for the form function and \( E_F^* \) is the composite elastic modulus, as listed in Eq. (4) and (5):

$$F_{a/2}(H_s) = \begin{cases} 4.4086 \times 10^{-3} (4 - H_s)^{6.644} & H_s < 4 \\ 0 & H_s \geq 4 \end{cases}$$

(4)

$$E_F^* = \frac{1}{\frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2}}$$

(5)

where \( H_s \) is the non-dimensional summit clearance, \( v_1 \) and \( v_2 \) are Poisson ratios of bearing material and shaft material respectively, \( E_1 \) and \( E_2 \) are Young moduli of bearing material and shaft material respectively.

2.4. Lubrication conditions

Generally, the film thickness ratio is designed as a criterion for the determination of the lubrication condition. It is calculated as follows [10]:

$$\lambda = \frac{h_{min}}{\sqrt{\sigma_1 + \sigma_2}}$$

(6)
where $h_{\text{min}}$ represents the minimum oil film thickness. $\sigma_1$ and $\sigma_2$ are surface roughness of bearing and journal respectively. According to the range of the film thickness ratio, journal bearing lubrication regimes are divided into three kinds: boundary lubrication ($\lambda \leq 1$), mixed lubrication ($1 < \lambda < 3$), hydrodynamic lubrication ($\lambda \geq 3$).

3. Simulation

Generally, the bearing and the shaft are fully separated by the oil film at the hydrodynamic lubrication. Due to the increase of the external load or the oil starvation, the oil film may be broken eventually resulting in the direct contact between the bearing and the shaft. The asperity contact pressure will play a dominant role in supporting the bearing operation. The lubrication condition gradually transforms from hydrodynamic lubrication to mixed lubrication or boundary lubrication. Based on the principle, in AVL EXCITE, the interaction in the simulation model including total normal pressure, tangential stress, and energy dissipation are divided into two types: full oil film interaction and asperity contact interaction. The simulation model comprises the test connecting rod and the test bearing. The model information about the geometry, mass, stiffness, and degree of freedom is condensed to the FE condensation model. The simulation model specimen is presented in Figure 1. The simulation parameters are listed in Table 1. The test loads for the maximum external load of 20MPa, 27MPa, 34MPa are applied in the simulation model, as shown in Figure 2(a). The simulation results are listed in Table 2.

### Table 1. Simulation parameters.

| Parameter | Value |
|-----------|-------|
| Bearing width (mm) | 29.5 |
| Bearing wall thickness (mm) | 1.825 |
| Bearing clearance (mm) | 0.04 |
| Bearing material | ZSnSb11Cu6 |
| Elastic modulus of bearing material (GPa) | 47.93 |
| Poisson ratio of bearing material | 0.2847 |
| Bearing surface roughness (μm) | 0.4 |
| Shaft diameter (mm) | 52.7 |
| Shaft material | 42CrMo |
| Elastic modulus of shaft material (GPa) | 212 |
| Poisson ratio of shaft material | 0.28 |
| Shaft surface roughness (μm) | 0.2 |
| Shaft Speed (rpm) | 3000 |
| Maximum external load (MPa) | 20/27/34 |
| Lubricant grade | Shell Rimula R3 Multi 10W-30 |
| Lubricating Oil supply pressure (MPa) | 0.6 |
| Lubricating Oil supply temperature (℃) | 70 |

### Table 2. Simulation results.

| Parameter | Maximum external load (MPa) |
|-----------|-----------------------------|
|           | 20MPa | 27MPa | 34MPa |
| MOFT (μm) | 1.87 | 1.24 | 0.94 |
| POFP (MPa) | 58.2 | 78 | 93.8 |
| PASP (MPa) | 0 | 54.7 | 103 |
| Total friction power losses (W) | 170 | 228 | 403 |
| Film thickness ratio | 4.156 | 2.756 | 2.089 |
| Lubrication condition | Hydrodynamic lubrication | Mixed lubrication | Mixed lubrication |
Figure 1. The simulation model.

Figure 2. The simulation results: (a): the test load; (b): the peak oil film pressure (POFP); (c): the minimum oil film thickness (MOFT); (d): the peak asperity contact pressure (PASP).

Figure 2(b) shows the plot of the peak oil film pressure (POFP). With the increase of the maximum external load, the POFP improves from 58.2MPa to 93.8MPa to enhance the carrying load capacity. Simultaneously, the minimum oil film thickness (MOFT) declines correspondingly in Figure 2(c). According to Eq. (7), the MOFT of 1.4μm is the basis to judge whether the asperity contact occurs. For 20MPa, the MOFT of 1.82μm is higher than the threshold, thus there is no asperity contact pressure. The test bearing runs in the hydrodynamic lubrication and has good performance. The total pressure is entirely provided by the hydrodynamic pressure. The film thickness ratio is calculated as 4.156, which satisfies the demand of the hydrodynamic lubrication condition. Nevertheless, the MOFT is 1.24μm and 0.94μm respectively for 27MPa and 34MPa, which are lower than the threshold. The oil film is broken and asperity contact between the shaft and the bearing occurs. The bearing operating condition transforms from hydrodynamic lubrication to mixed lubrication. The total pressure consists of the hydrodynamic pressure and the asperity contact pressure. It can be verified that the film thickness ratio is 2.756 and 2.089 respectively for 27MPa and 34MPa. The bearing lubrication condition belongs to the mixed lubrication.

Figure 3. Comparison of the friction power losses for three load cases.

Figure 3 demonstrates the friction power losses condition under three external loads. Total friction power losses are classified into two aspects: hydrodynamic power losses generated from hydrodynamic pressure and power losses because of asperity contact. For 20MPa, no asperity contact pressure exists, hence asperity friction power loss is zero and total friction power losses only include hydrodynamic power losses, with a value of 170W. For 27MPa, due to the occurrence of asperity contact pressure, 52W asperity friction power losses generate, which account for 22.8% of total power losses. With the asperity contact pressure increasing, power losses generated from asperity contact increase to 246W, which are 61% of total power losses for 34MPa. Thus it can be found that the appearance of the asperity contact improves the asperity contact power loss ratio. Friction power
losses usually dissipate in the term of heat, which causes the local high temperature on the rubbing area of the bearing surface and the deterioration of material performance.

The contour of the PASP distribution at different crank angles for three load cases is investigated in Figure 4. Three crank angle 120°, 160°, 200° are chosen to describe the variation on the asperity contact distribution. For 27MPa, the slight asperity contact remains at one edge at crank angle 120°. At crank angle 160°, the asperity contact exists at both edges. At crank angle 200°, there appears the mild asperity contact at one edge and in the central region of the bearing. For 34MPa, the PASP is increased. Besides, the asperity contact area spreads from one side to both sides at crank angle 120°, from the bearing edges to the central area at crank angle 200°. Generally, these areas may inevitably suffer to wear. Serious wear will endanger directly the long term normal operation of bearings.

Figure 4. The contour of the PASP distribution at different crank angles for three load cases.

4. Experiment
The bearing specimens are obtained by centrifugal casting, which were made of the bimetal structure containing a ZSnSb11Cu6 alloy substrate layer with a thickness of 0.6mm and the 10#steel back. The chemical compositions of bearing specimens are listed in Table 3. To verify the simulation results, the fatigue tests were conducted on the journal bearing fatigue test-rig at Glacier Vandervall, as sketched in Figure 5. The load-increasing method is applied to measure the bearing fatigue limit. Firstly, the test bearing is run for 30 minutes without load. Secondly, the initial load is set to 20MPa, the test-rig continues to run for 20h (equivalent to 3.6×10^6 load cycles), if no damage on the bearing surface is observed, the initial load will be increased by 7MPa. The test is repeated until the occurrence of bearing surface damage or the test-rig load limit. The bearing fatigue limit is determined as the previous load or the test-rig load limit. During the test, the shell temperature is measured in real-time by the temperature sensor.

Table 3. The chemical compositions of bearing specimens.

| Material   | Chemical composition | Sn  | Pb   | Sb   | As  | Cu   | Fe    |
|------------|----------------------|-----|------|------|-----|------|-------|
| ZSnSb11Cu6 | Chemical composition |     |      |      |     |      |       |
| Value      |                      | Bal.| 0.35 | 10-12| 0.1 | 5.5-6.5 | 0.1    |
| 10#Steel   | Chemical composition | C  | Si   | Mn   | Cr  | Ni   | Cu    |
| Value      |                      | 0.07-0.13 | 0.17-0.37 | 0.35-0.65 | ≤0.15 | ≤0.30 | ≤0.25 |

Figure 5. Journal bearing fatigue test-rig at Glacier Vandervall.
5. Discussion

5.1. Surface damage condition
Figure 6 shows the test bearing surface damage conditions for three load cases. For 20MPa, no obvious damage is observed on the surface of the testing bearing, which represents that the testing bearing may work normally during this stage. Combined with the simulation results, the bearing runs in the hydrodynamic lubrication, good lubrication performance does not cause bearing fatigue failure. For 27MPa, it can be seen clearly that the fatigue wear exists at one edge of the bearing. According to the above simulation results, the bearing runs in the mixed lubrication, the presence of the asperity contact pressure marks that the rubbing occurs between the bearing and the shaft. Because the rubbing area locates at one edge, so slight damage is eventually formed. While for 34MPa, the bearing surface is seriously damaged. A lot of peelings and severe material adhesion appear on the surface of the bearing. Based on the simulation results, it can be found that the lubrication condition of the bearing is the mixed lubrication and the asperity contact pressure increases sharply in the rubbing region, which leads to the increase of asperity friction power losses. This causes high shell temperature in the rubbing region, which makes the material performance degenerate due to the low melting point of the ZSnSb11Cu6 alloy. Consequently, the material adhesion appears on the surface of the bearing. In addition, the comprehensive influence of cycle load and asperity contact pressure promotes the initiation and propagation of fatigue cracks and finally the formation of peelings. Based on the above analysis, the fatigue limit is determined as 27MPa.

Figure 6. The surface damage conditions of the testing bearing.

5.2. Shell temperature
The shell temperature is recorded after the testing bearing runs for 2.5 hours in Figure 7. The curves reflect that the temperature gradually rises with time and then stabilizes. And the temperature changes within 80℃—90℃ for 20MPa and 27MPa. The friction power loss dissipation in the form of heat is a consequence of the increase in shell temperature. Based on simulation results, no asperity contact power loss is generated for 20MPa, while only 52W asperity contact power losses are produced for 27MPa. Consequently, the shell temperature of 27MPa is higher than that of 20MPa. No severe adhesive damage appears on the bearing surface. However, the temperature increases sharply to 103℃ for 34MPa. The high shell temperature promotes the melting of ZSnSb11Cu6 alloy and the degradation of bearing performance, which is an important cause of bearing failure.

Figure 7. The shell temperature during the fatigue test.
6. Conclusion
In this paper, the lubrication performance of the journal bearing cover with a ZSnSb11Cu6 alloy is studied by AVL EXCITE. Combined with the simulation results, the fatigue tests are conducted to analyze the damage mechanisms of journal bearing. The main conclusions are as follows:

(1) For 20MPa, no damage is observed on the surface of the bearing. The simulation result indicates that the bearing runs in hydrodynamic lubrication. The experiment results are consistent with the simulation results.

(2) For 27MPa, the fatigue damage occurs on one side of the bearing, which means that the bearing lubrication regime transforms into the mixed lubrication and the asperity contact appears. This conclusion is also consistent with the simulation analysis.

(3) For 34MPa, the bearing still runs in mixed lubrication. Under the comprehensive effect of cycle load and asperity contact pressure, the material adhesion and fatigue damage exist on the surface of the bearing, eventually leading to bearing failure.

(4) As the asperity contact aggravates, the shell temperature rise caused by friction power losses increases significantly, which causes bearing performance degradation, poor bearing load carrying capacity, and bearing failure.

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