

Effects of Journal Roundness Phase and Amplitude on Lubrication of Engine Bearings with Consideration of Straightness

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ABSTRACT

Manufacturing tolerances are inevitable in nature. For the bearings used in internal combustion engines, the manufacturing tolerances of roundness, which is of the micron scale, can be very close to the bearing radial clearance, and as a result the roundness could affect the lubrication of the bearings and thus affecting the friction loss of the engine. However, there is insufficient understanding of this mechanism. This study aims to find out the effects of the amplitude and the phase of journal roundness in the shape of ellipse on the lubrication of engine bearings. The elastohydrodynamic (EHD) theory is applied to model the bearing since the EHD model takes account of the elastic deformation of the journal and the bearing shell. The analysis of the DOE results shows the existence of roundness can be beneficial to the lubrication in some cases. The roundness phase determines whether the existence of roundness will lead to an increase or decrease of friction loss, and the roundness amplitude determines the variation range of the friction power loss. The coupling effects of roundness and straightness are also studied. It is shown that coupling does exist between the two geometrical tolerances and needs to be considered in tolerance design.

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INTRODUCTION

The rotary motion joints of internal combustion engine are working under bad conditions characterized by large variations of dynamic loads and high peak loads, and high requirements are desired for components of the engine. Hydrodynamic journal bearings, with simple structures, low costs, and smooth operation at high speeds, are widely used in automotive engines to reduce the friction power loss. Most studies of dynamic performances of hydrodynamic journal bearing in literature did not take into account of manufacturing tolerances, which are inevitable during the manufacturing process and do affect the working performance of bearings. Unreasonable tolerance will lead to either a severe decline in quality or a substantial increase in the manufacturing and measuring costs. Therefore, it is necessary to study the effect of manufacturing tolerance on the working performance of bearings to provide guidance on designing the tolerances reasonably.

Some research have shown that the tolerances have an obvious effect on bearings lubrication. XU et al theoretically analyzed how manufacturing tolerances of the bearing width and the diameters of both the bearing shell and the journal affect the working performance of bearings based on Taguchi method [1]. In the studies of Wei et al, the effects of the friction power loss brought by the dimensional tolerances of the dynamic viscosity, bearing width, bearing diameter and journal diameter were analyzed [2]. Ogrodnik et al. found that the tolerance of a single dimension may not demonstrate a noteworthy effect on the system performance in isolation, but when it is coupled with other tolerances the effect can be significant, and the

2^k factor design was applied to study the coupling effects of different manufacturing tolerances on the bearings system [3]. Zhou et al discussed the relationship between the dimensional tolerances and friction power loss of the engine system and determined the optimal tolerance based on a Multi-Disciplinary optimization approach [4].

However, those studies only focus on the dimensional tolerances, and there is insufficient understanding of how geometric tolerances of the journal may affect the bearing lubrication, especially the roundness coupled with straightness. The sizes of roundness and straightness tolerances are of the micron scale, which is the same with that of the bearings clearance, so it is large enough to generate remarkable effects on working performance of bearings. In the work of Dehart et al, the relationships between the roundness amplitude, the number of lobes and bearing wear are analyzed [5]. Mehenny et al discussed the effects of the number and the amplitudes of circumferential lobes on the operating parameters such as maximum lubricant pressure and minimum oil film thickness, but the elastic deformation of the bearing shell is neglected [6]. Torii et al found that lubrication of a connecting rod journal bearing is significantly influenced by the elastic deformation and shapes of a connecting rod bearing [7], so the elastic deformation of the journal and shell should be considered when analyzing the effect of roundness on bearing lubrication. Zeng et al presented a dynamics model of hydrodynamic bearing rotor system where roundness is considered, and effects of the eccentricity on friction power loss was studied [8]. Xu, et al theoretically discussed the effect of journal out-of-roundness on the dynamic performance of rotating machines supported by oil film bearings [9].

Although the above-mentioned studies analyzed the effects of the number and the amplitudes of circumferential lobes on bearing lubrication with consideration of elastic deformation of the bearing shell and journal, the influence of the roundness phase (the angle of the roundness geometric shape along the journal circumferential direction) is ignored. As the load condition of the bearings is changing all the time during the 4-stroke cycle of an engine, the lubrication condition of the bearings with different roundness phases also changes accordingly. In addition, all of the above mentioned researches are under the assumption that the axial direction of the journal bearing is ideally straight (that is, the straightness amplitude is zero), and so the effect caused by the existence of straightness is neglected. This work aims to illustrate the effects of the roundness phase and amplitude on the bearing lubrication with the consideration of straightness. First, impacts of the roundness phase and amplitude on lubrication are analyzed, respectively. Then the coupling effects of roundness and straightness on friction loss of the bearing are analyzed and presented. It is shown that the roundness will increase or decrease the friction loss which is determined by the roundness phase, and the variation range of friction loss is determined by the roundness amplitude. It is also concluded that the coupling of roundness and straightness will increase the variation of friction loss.

The rest of the paper is organized as follows: Section 2 presents the background of this work. In Section 3, the individual effect of roundness phase and amplitude on friction loss is studied. The coupling effect of the roundness and straightness is given in Section 4, followed by the conclusion in Section 5.

BACKGROUND

In this section, the background information of this work is presented. First the bearing model based on elastohydrodynamic (EHD) theory is introduced briefly, and then the roundness model defined in terms of roundness phase and the roundness amplitude is presented.

Bearing Model

The elastic deformation of the bearing shell is of a micron scale, which is comparable with the amplitude of roundness, thus it is necessary to take account of elastic deformation when establishing the bearing model for a more accurate analysis of how tolerances may affect the friction power loss. Therefore, AVL Excite PowerUnit, which is proved to be able to provide accurate results for the engine bearings [10], is applied to build the bearing model based on the elastohydrodynamic (EHD) theory which takes the elastic deformation into consideration.

The differential equation applied in the EHD model describes the pressure variation of the oil film between the shaft and the bearing shell, as shown in Eq. (1) [11]:

$$\frac{\partial}{\partial x} \left(\frac{\rho h^3 \theta_r}{\eta} \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial z} \left(\frac{\rho h^3 \theta_r}{\eta} \frac{\partial P}{\partial z} \right) = 6 \frac{\partial}{\partial x} (v_t h \rho \theta_r) + 12 \rho \frac{\partial (h \theta_r)}{\partial t} \quad (1)$$

where P is the oil film pressure, h is the modified thickness of oil film, ρ is the oil density, η is the dynamic viscosity of the oil, v_t is the tangential velocity, and t is time. The effects of fill ratio, represented by θ_r , on bearing lubrication are taken into account as well.

The modified oil film thickness h concerning the elastic deformation, roughness, and roundness error is computed by the following equation:

$$h(\theta) = h_o(\theta) + h_e(\theta) + h_R(\theta) + h_z(\theta) + h_s(\theta) \quad (2)$$

where $h_o(\theta)$ is the oil film thickness with the assumption of rigid journals and a rigid bearing shell, $h_e(\theta)$ is the thickness variation caused by the elastic deformation of the journals and bearing shell, $h_R(\theta)$ is the variation resulted from the roughness of the journals and bearing shell, $h_z(\theta)$ is the variation due to the existence of roundness, and $h_s(\theta)$ is the variation caused by the existence of straightness.

The boundary condition of JFO (Jakobsson, Floberg, Olsson) theory is adopted to ensure mass conservation at the boundary of the oil film. The crankshaft and connecting rod in this model are taken as flexible bodies, both of which are meshed into hexahedral units by finite element software (Hypermesh).

The calculated oil film thickness is taken into the extended Reynolds equation (Eq. (1)), which is then solved via the finite difference method with the bearing surface divided by hydrodynamic (HD) mesh [12]. The pressure variation across the entire bearing surface computed from Reynolds equation is utilized as the load condition for the finite element (FE) mesh of the connecting-rod. Then the elastic deformation throughout the journal surface under that load condition will be calculated and plugged into Eq. (2) to get the oil film thickness. After that the newly obtained oil film thickness will replace the old one in the Reynolds equation, and the pressure variation across the bearing surface will again be calculated. The HD mesh and FE mesh will be computed repeatedly until the pressure and fill ratio converge, that is, the differences of both values between the last two consecutive iterations are less than 0.1%.

Roundness Model

Most profiles of crankpins are similar to ellipse in geometry [7,8], and roundness tolerance in shape of an ellipse is studied in this work to investigate the effects of roundness tolerance on bearing lubrication. The geometry of an ellipse roundness tolerance is shown in Figure 1.

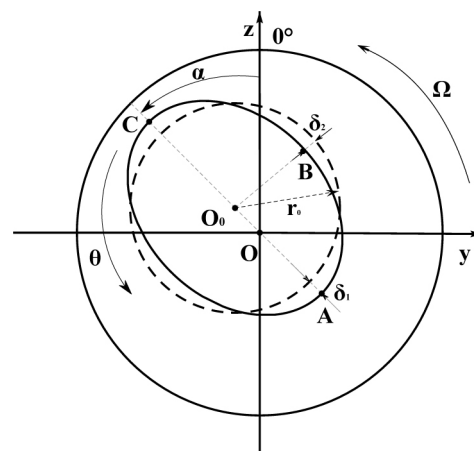


Figure 1. Definition of ellipse roundness



The roundness phase is defined as the angle between O_0C and z axis when the piston is at TDC during exhaust stroke, and it is denoted by α as shown in [Figure 1](#). z axis in [Figure 1](#) is along the direction of the piston cylinder liner, O is the center of the bearing, O_0 is the center of the journal, and Ω is the angle that journal has rotated. AC in [Figure 1](#) is the major axis of the ellipse, and O_0B is the semi-minor axis of the ellipse. The amplitude of the ellipse roundness is defined as $\delta = \delta_1 + \delta_2$ where δ_1 is the distance between the major axis endpoint A to the ideal circular profile and δ_2 is the distance between the minor axis endpoint B and the ideal circular profile. Then the ellipse can be described by

$$\begin{cases} y_r(\theta) = (r_0 - \delta_2) \sin(\theta - \Omega - \alpha) \\ z_r(\theta) = (r_0 + \delta_1) \cos(\theta - \Omega - \alpha) \end{cases} \quad (3)$$

where y_r and z_r are the axes along the minor axis and the major axis of the ellipse, respectively, and r_0 is the mean radius of the ellipse calculated by the least squares method. Then the variation of oil film thickness caused by the ellipse roundness can be computed from the following equation.

$$h_z(\theta) = r_0 - \sqrt{y_r^2(\theta) + z_r^2(\theta)} \quad (4)$$

ANALYSIS OF ROUNDNESS PHASE AND AMPLITUDE ON FRICTION LOSS AND LUBRICATION

The parameters of the bearing studied in this work are listed in [Table 1](#).

Table 1. Parameters of the bearing

Terms	Nominal sizes
Bearing/Journal diameter	38mm
Bearing shell width	16.2mm
Oil type	SAE 5W-10
Shaft rotational speed	6000rpm
Roundness Tolerance	5micron
Straightness Tolerance	5micron

As we all know the friction power loss is an important output performance of the bearing. In this section, the friction power loss, peak oil film pressure (POFP) and minimum oil film thickness (MOFT) of the bearing under different roundness phases and amplitudes are analyzed to show the effects of roundness on the friction loss and bearing lubrication. Single-factor analysis is performed to show how the phase and the amplitude of roundness could affect the lubrication of the bearing, respectively.

Effects of Roundness Phase on Bearing Lubrication

Results of the friction loss, POFP and MOFT of the bearing at different roundness phases with the same roundness amplitude (5 μm) are shown in [Figure 2](#). For a 4-stroke engine, from the intake stage to the exhaust stage within which the crank angle rotates 720 degrees, the load conditions of the bearing is changing all the time, and as a result the lubrication state of bearings varies with different roundness phases even with the same roundness amplitude. Since the ellipse roundness is axisymmetric, the roundness phase in this paper is set to vary from 0° to 180° . The dashed lines in [Figure 2](#) show the corresponding result of an ideal journal, that is, the cross section of the journal is ideally circular.

It can be concluded from [Figure 2](#) that the roundness phase of the journal has an obvious influence on the lubrication condition and friction loss of the bearing. With the increase of the roundness phase, the friction power loss also increases, and the maximum friction loss appears at the roundness phase of 67° . Then the power loss decreases until the minimum friction loss appears at the roundness phase of 157° . On the other hand, when compared with the ideal journal case, it is indicated that the roundness phase could increase the friction power loss, and it is also able to decrease the friction power loss of the bearing. So the existence of roundness will help to reduce the friction loss under some conditions.

The variation of POFP with respect to the roundness phase is contrary to that of the friction loss, as shown in [Figure 2\(b\)](#). This indicates that different roundness phases result in different oil pressure distributions and lubrication conditions. With the roundness amplitude of 5 μm , different roundness phases may lead to an increased or decreased POFP.

As for the MOFT, although in most cases the existence of roundness results in a reduced MOFT, but some roundness phases may lead to an increased MOFT with the variation range less than 0.1 mm. It is noted that the variation of MOFT is different with that of the friction loss and POFP in that it has two peaks. The position of MOFT with respect to roundness phase along with the MOFT are presented in [Figure 2\(c\)](#), which shows that roundness phase also changes the position where MOFT appears and the occurrence of the two peaks is resulted from this position change of MOFT.

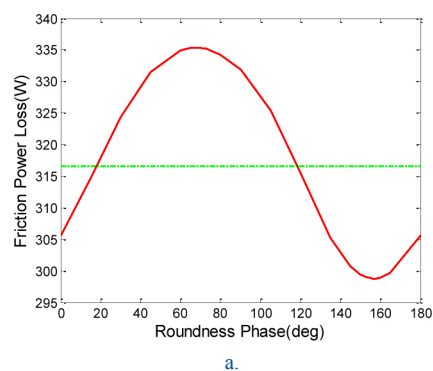


Figure 2. (a) Friction Loss, (b) POFP and MOFT, (c) MOFT position and corresponding values of the bearing at different roundness phases