

A Study on Wear Progress of Plain Bearing under Mixed Lubrication Condition

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ABSTRACT

Recently, automotive engines have been operating under harsh conditions of high-power, low viscosity oil and increase of start-stop (e.g. idling stop). In plain bearing used within engine, as oil film thickness decreases, the frequency of direct contacts on the sliding surfaces between the shaft and the bearing are gradually increasing. In fact, the plain bearings for engines would tend to be used under mixed lubrication and the contacts of the surface roughness asperities sometimes occur between the shaft and the bearing. As a result, the bearing wear on the sliding surfaces is accelerated by the contacts of the roughness asperities. In order to predict the bearing performance exactly, it is very important to understand the change progress of the geometric shape of sliding surfaces caused by the wear.

Therefore, in this study, to establish the theoretical analysis method of the wear progress of the bearing under mixed lubrication conditions, the following procedures were conducted on a static load:

- The bearing performance analyses were carried out, based on the mixed lubrication theory which incorporates both the EHL theory with consideration of the surface roughness and the contact theory of the roughness asperities.
- By the generated contact pressures between the roughness asperities, the wear depth in the bearing surface were calculated, and the wear progress of the bearing sliding surface was analyzed from calculated results.
- The mixed lubrication analysis involved iteration of above two steps with consideration of the wear progress on the bearing surface.
- Theoretical analysis results of the wear progress were verified by experiments on the rig tests.

INTRODUCTION

In recent years, automotive engines are developing with the customer's requirements for high performance, high power, low viscosity oil for low friction loss, and increase of start-stop for low fuel consumption (e.g. idling stop). Under these circumstances, plain bearings used within engines have been operating under harsh conditions, and as oil film thickness decreases, the frequency of direct contacts on the sliding surfaces between the shaft and the bearing are gradually increasing. The lubrication condition which mingles the hydrodynamic lubrication and the direct contact of roughness asperities is defined as mixed lubrication condition. Moreover, the wear of bearing is progressed by an interaction of the roughness asperities on the bearing and shaft [1]. Owing to the running condition of bearings under mixed lubrication, the bearing wear on the sliding surfaces is accelerated by the direct contacts, and numerous problems related to the bearing wear are newly occurring.

The change of the shape of bearing surface due to the wear causes the change progresses of the bearing clearance in the bearing. This leads the changes of the oil film formation and direct contacts between the shaft and the bearing. The oil film thickness, the oil film pressure and the contact pressure of roughness asperities change along with the progress of the bearing wear. Therefore, in order to predict the bearing performance exactly, it is very important to understand the change progress of the geometric shape of sliding surfaces caused by the wear. Focusing on the correlation between bearing wear and bearing performance, it is necessary to establish the theoretical analysis method which shows their behavior.

There are a few papers [2,3,4,5,6,7] about the theoretical analysis models of the wear progress of the bearing under mixed lubrication conditions. In terms of theoretical analyses,

a rigid analysis based on the hydrodynamic lubrication theory is widely used. But its results of the oil film thickness are constant distribution in the axial direction. A rigid analysis could not be used for explaining the causes that lead to the occurrence of bearing wear in the axial direction. By an application of the elastohydrodynamic lubrication (EHL) theory in which elastic deformation is taken into consideration, the theoretical analysis of the correlation between the oil film thickness distribution by EHL and the bearing wear was enabled. The theoretical models of [2], [3] and [4] can predict the wear from the EHL analysis results that do not consider with the direct contacts of the roughness asperities, and that the wear prediction models were verified by experiments on the rig tests. In the theoretical analysis under mixed lubrication condition, the direct contacts of the roughness asperities between the shaft and the bearing should be considered in order to predict the bearing wear progress and the bearing performance exactly. Hence [6] and [7] showed the wear prediction models which combined the EHL theory with consideration of the surface roughness, the contacts of the roughness asperities, and the wear theory. However, these theoretical models have been not verified by the experiments.

In all analysis models mentioned above, the circumferential roughness (see Fig.1 and 2) which is typical surface roughness of the engine bearings has been not considered. The verification of the performance of theoretical analysis model comparing with experiments is very important to use results of the wear prediction by the theoretical analyses for the optimized design of the bearing. Therefore, in this study, to establish the theoretical analysis method of the wear progress of the bearing under mixed lubrication condition, the plain bearing which has the circumferential roughness in the aluminum alloy which is the typical material of bearing is targeted, and the following procedures were conducted on a static load:

- The bearing performance analyses were carried out, based on the mixed lubrication theory which incorporates both the EHL theory with consideration of the surface roughness and the contact theory of the roughness asperities.
- By the generated contact pressures between the roughness asperities, the wear depth in the bearing surface were calculated, and the wear progress of the bearing surface was analyzed from calculated results.
- The mixed lubrication analysis involved iteration of above two steps with consideration of the wear progress on the bearing surface.
- Theoretical analysis results of the wear progress were verified by experiments on the rig tests.

MODEL OF BEARING SLIDING SURFACE

Fig. 1 shows the coordinate system of the bearing. θ is the bearing angle, and z is the axial direction. The direction of the bearing angle θ and the direction of the shaft rotation are same.

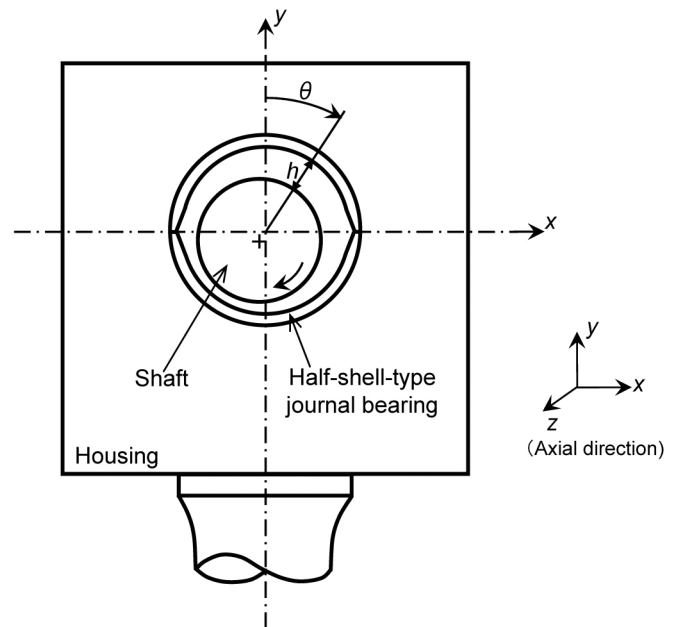
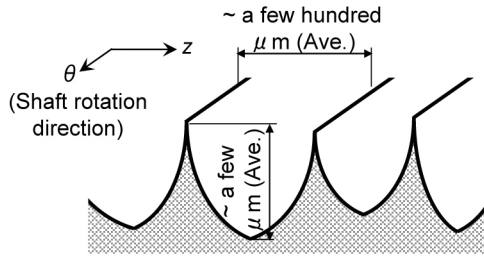
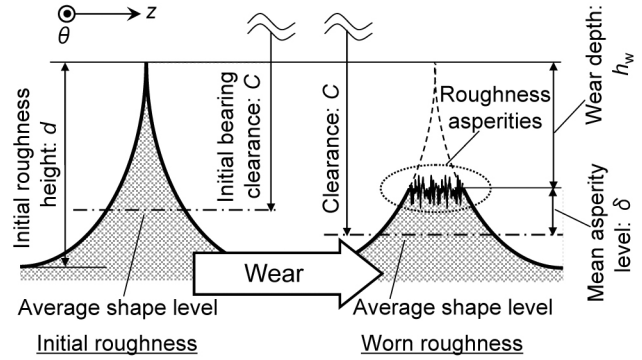


Fig.1. Coordinate system

Fig. 2(a) shows the circumferential roughness model of the bearing sliding surface with consideration of the bearing on rig test. The circumferential roughness on the bearing is typical surface roughness, and the orientations of the circumferential roughness correspond to the direction of the shaft rotation. Fig. 2(b) shows the wear model of the circumferential roughness. Three assumptions apply to this model. 1) As the shaft is extremely harder than bearing material, the wear of shaft is ignored. 2) The bearing circumferential roughness is truncated by the direct contacts of the roughness asperities between the shaft and the bearing, and then the local roughness of worn surface is transferred from the shaft roughness (roughness asperities). 3) The direct contacts between the bearing and the shaft occur only at the local area between the roughness asperities of the bearing and the roughness asperities of the shaft.



(a) Schematic view of circumferential roughness



(b) Wear model of circumferential roughness

Fig.2. Circumferential roughness model of bearing surface

ANALYSIS METHOD

The mixed lubrication theory which incorporates both the EHL theory with consideration of the surface roughness and the contact theory of the roughness asperities can be described as follows. The oil film characteristics under a static load are expressed by Christensen's modified Reynolds equation (1) [8], which is the basis equation for hydrodynamic lubrication with consideration of the circumferential roughness.

$$\frac{\partial}{\partial \theta} \left[\frac{\partial p_{\theta,z,i}}{\partial \theta} E(h_{m\theta,z,i}^3) \right] + \frac{\partial}{\partial z} \left[\frac{\partial p_{\theta,z,i}}{\partial z} \frac{1}{E(h_{m\theta,z,i}^3)} \right] = 6\mu_{\theta,z,i} \Delta u_{\theta,z,i} R \frac{\partial}{\partial \theta} E(h_{m\theta,z,i}) \quad (1)$$

where h : Oil film thickness ($=E(h_m)$), h_m : Oil film thickness including the random components of surface roughness, $E(h_m)$: Expected value of h_m , p : Oil film pressure, R : Shaft radius, Δu : Relative velocity between two surfaces, μ : Viscosity, and subscript i : Time step of natural number. The oil film pressure p at the edge of bearing is zero, without involving any misalignment between the shaft and the bearing. Because the analysis model is axially symmetric, a half model in the axial (z) direction is applied. It has an axial pressure gradient of dp/dz at the center of the bearing is zero.

The lubricant is used as a Newtonian fluid, and the temperature of the oil film is assumed to be constant. The pressure dependence of viscosity is considered through the Roelands law.

$$\mu_{\theta,z,i} = \mu_0 \exp \left[\frac{\alpha p_0}{B} \left(\left(1 + \frac{p_{\theta,z,i}}{p_0} \right)^B - 1 \right) \right] \quad (2)$$

where

$$B = \alpha p_0 / \ln(\mu_0 / \mu_{\text{inf}}) ,$$

μ : Viscosity, μ_0 : Viscosity at ambient pressure, μ_{inf} : Viscosity at infinite pressure, α : Coefficient of piezo-viscosity, and p_0 : Roelands constant pressure.

The shaft is assumed as a rigid body, and the elastic deformation is obtained from the compliance matrix of the bearing surface and added to the oil film thickness. This oil film thickness can be expressed by the following equation.

$$h_{\theta,z,i} = C_{\theta,z,i-1} (1 - \varepsilon_{xi} \cos \theta - \varepsilon_{yi} \sin \theta) + L p_{\theta,z,i} \quad (3)$$

where C : Radial bearing clearance, ε : Eccentricity ratio, and L : Compliance matrix.

The contact pressures of the sliding surface are assumed to be generated at the local area between roughness asperities of the shaft and roughness asperities of the bearing. The Greenwood-Tripp theory [9] is used to calculate the contact pressures.

$$p_{c\theta,z,i} = K_{Ac\theta,z} E' F_{5/2} \left(\frac{h_{\theta,z,i} - \delta_{\theta,z,i-1}}{\sigma_{\theta,z}} \right) \quad (4)$$

where

$$K_{Ac\theta,z} = \frac{16\sqrt{2}\pi}{15} (\eta_{\theta,z} \beta_{\theta,z} \sigma_{\theta,z})^2 \sqrt{\frac{\sigma_{\theta,z}}{\beta_{\theta,z}}} , \quad (5)$$

$$E' = \left(\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right)^{-1}, \quad (6)$$

p_c : Asperity contact pressure, K_{Ac} : Coefficient of asperity contact, η : Surface density of asperities, β : Curvature of asperities, σ : Standard deviation of asperity height, δ : Mean asperity level, E' : Composite elastic modulus, E : Young's modulus, ν : Poisson's ratio, subscript 1: Shaft, and subscript 2: Bearing.

The equilibrium between the external load that acts upon the bearing and the reaction force are as follows:

$$\begin{aligned} W_{xi} - \int_A (p_{\theta,z,i} + p_{c\theta,z,i}) \sin \theta dA &= 0 \\ W_{yi} - \int_A (p_{\theta,z,i} + p_{c\theta,z,i}) \cos \theta dA &= 0 \end{aligned} \quad (7)$$

where W : Bearing load, A : Surface area at bearing, $\int_A (p_{\theta,z,i} + p_{c\theta,z,i}) \sin \theta dA$ and $\int_A (p_{\theta,z,i} + p_{c\theta,z,i}) \cos \theta dA$: Reaction force of the oil film and roughness asperities. As the equilibrium between the reaction force and the external load, and the simultaneous equations between the reaction force and the eccentricity at the nodes on the bearing surface are nonlinear, the Newton-Raphson method is adopted for the numerical analysis.

The wear of shaft is ignored, and the bearing wear caused by the direct contact between the shaft and the bearing can be described as follows. The Archard equation [1] is used to calculate the bearing wear:

$$\Delta h_{w\theta,z,i} = \frac{k}{H} p_{c\theta,z,i} \cdot \Delta u_{\theta,z,i} \cdot \Delta t_i \quad (8)$$

where Δt : Time span, Δh_w : The amount of change in Δt of wear depth, k : Wear coefficient, and H : Plastic flow pressure of bearing alloy. The wear depth of bearing in a certain time can be as follows:

$$h_{w\theta,z,i} = \sum_{n=1}^i \Delta h_{w\theta,z,n} \quad \text{at} \quad t_i = \sum_{n=1}^i \Delta t_n \quad (9)$$

where h_w : Wear depth of bearing, and t : Time.

The oil film thickness h and the mean asperity level δ are set up on the basis of the average shape level of Fig. 2(b). The change of the average shape level caused by the increase of the wear depth h_w is taken into consideration as the increase

of the bearing clearance C as the change of the geometric shape of the bearing surface. The bearing clearance C and the mean asperity level δ are calculated using the wear depth h_w and the assumptions of the worn shape of circumferential roughness.

Fig. 3 shows a flowchart for the calculation of the wear progress of the bearing surface under mixed lubrication condition. The initial values of bearing parameters such as the bearing clearance C and mean asperity level δ at time step $i=1$ are given. The oil film pressure $p_{\theta,z,i}$ and contact pressure $p_{c\theta,z,i}$ are calculated by using equation (1) and (4). The calculation is iterated to satisfy the force equilibrium condition of equation (7). When the equation (7) is satisfied, the amount of change in Δt_i of wear depth $\Delta h_{w\theta,z,i}$ are determined from equation (8). The wear depth $h_{w\theta,z,i}$ and time t_i are also determined from equation (9). After that, the maximum $p_{c\theta,z,i}$ and t_i are checked. If the maximum $p_{c\theta,z,i}$ does not equals to zero and t_i is not reached to the assigned time, the $C_{\theta,z,i}$ and $\delta_{\theta,z,i}$ (geometric shape of bearing surface after the wear) are calculated by using $h_{w\theta,z,i}$. The i is added from $i=i$ to $i=i+1$. The next calculation is done considering with worn surface shape. The wear calculation involves the iteration of above processes.

EXPERIMENT METHOD

Fig. 4 shows the structure of the rig test machine which is used for the tests of the bearing performances under static load condition [10, 11]. This test machine is able to generate a static load by the hydraulic oil. A static load is applied to the direction of the bearing angle $\theta=180$ degrees on the test bearing and is held by four supporting bearings. The distances between their supporting points are short, thus minimizing the influence of the load in bending the shaft. In addition, its housing stiffness is high and the test bearing is not subjected to an extreme edge loading. Theoretical analysis results of the wear progress were verified by experiments on the rig tests.

Fig. 5 shows the appearance of the bearing surface after rig test. The wear was mainly observed near the edge of bearing. The wear depth h_w of the bearing surface has been defined to be the change in the shape of the surface roughness before and after the test. As shown in Fig. 6, the amount of wear depth h_w was measured as the deference of the surface roughness between before and after rig tests at the axial direction, every 10 degrees in the circumferential direction within a range of -40 to $+40$ degrees relative to the load direction of the bearing angle $\theta=180$ degrees.