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Application of Elasto-HydroDynamic (EHD) Model to Predict Erosive Wear Failure in Conrod Bearing

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A B S T R A C T

Erosive wear damage is common damage in the bearing shell of engines which causes a change in bearing profile and affects the oil film pressure and durability of bearing shell. The objective of the present paper is to present an appropriate algorithm for prediction and failure analysis of wear in BE bearing of engines using the Elasto-HydroDynamic (EHD) model. The mentioned model incorporating a mass-conserving algorithm is utilized to compute the lubrication characteristics of bearing, such as minimum oil film thickness and maximum oil film pressure. In EHD analysis, bearing housing is modeled by the finite element method to consider the bearing deformation. To estimate the wear volume, a code was written in MATLAB® software which modifies the bearing profile and surface roughness during the analysis. A modified Archard model is used to model the lubricated sliding wear of rough contacting surface. Change in bearing surface roughness due to wear is modeled by the Abbot curve. Finally wear damage progression of BE bearing during engine operation is calculated and the results are thoroughly discussed. The numerical simulation results confirm that the wear rate at the initial stage of engine running is significant. It is concluded that wear adapts the bearing geometry in proper condition and improves the contact problem at the edges of bearing.

1. Introduction

Wear in engine parts is an important and determinant factor for the life of moving and abrasive parts. Wear calculation is very important in determining the functional life of bearing as well as evaluating the correct design of its surrounding parts. Engine designers are progressively using more advanced simulation techniques to reduce design time and costs and at the same time to improve the accuracy of design and reduce the number of validation tests required. In recent decade, the demand to gain higher specific power has caused to impose higher bearing load in diesel engines. On the other hand, due to competition in market, engine parts should be durable enough and low damages should occur in critical parts. Wear is a usual damage in engine bearing shells, therefore predicting the damage, introducing a method to prevent the damage and improving the design are indispensable. Connecting rod of diesel engine is under high firing and inertial loading. In recent decade, design limitations for decreasing of connecting rod weight causes to have more flexible and lighter housing especially for the Big Eye (BE) bearing. Bearing deflection has significant effect on lubrication performance and therefore can cause the erosive wear damage. Thus the deflection of bearing and its housing should be considered in the design process of housing.

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Wear has been recognized as the phenomenon of material removal from a surface due to interaction with a mating surface. Almost, durability and reliability of all machines are affected by wear [1]. Cranktrain of diesel engine operates under high load condition which leads to form thin oil film thickness between bearing and pin. Under such conditions, lubrication regime shifts from fully hydrodynamic to mixed lubrication and bearing becomes prone to erosive wear damage. Fig 1 shows the erosive wear damage in BE bearing of a connecting rod. Minimum oil film thickness and asperity contact pressure are the two factors that have been used to determine the probable erosive wear damage in the bearing during the design process. However, there are many other factors involved in the erosive wear damage of bearing especially in running-in. Thus, it is necessary to study the wear during the progression of damage.

Figure 1: Erosive wear damage in bearing [2]

Ushijima et al. [3] and Okamoto et al. [4-5] used a wear model which was similar to Archard [6] model, however, just the reaction force from asperity contacts was included in the calculation. The bearing wear increased at the edges of the bearing length and the crack also was observed near the edges. Wang et al. [7] proposed a simple system for prediction of dynamic wear in run-in contacts under partial EHD conditions, that taken into account the considerations of lubrication effects. They involved the real contact area of asperity contact pressure and effect of surface roughness in their proposed model. However, the proposed model was based on line contact condition, and therefore, should be developed for 3D contact condition. The simulated results showed that the variation of wear volume and the change of average roughness can be described by a second order polynomial.

Chamani and Karimaei [8] performed lubrication analysis of oil film in main and BE bearings of a diesel engine using the elasto-hydrodynamic analysis. They showed that bearing deformation affect the bearing failure such as wear and fretting fatigue. Afterward, Karimaei and Chamani [9] carried out parametric study of lubrication analysis of oil film in a BE bearing of a heavy duty diesel engine using the elasto-hydrodynamic analysis and investigated the effects of different parameters. The results showed that oil temperature, rotational speed of the engine, bearing clearance and flexibility of the connecting rod big eye have considerable effect on lubrication characteristics. Karimaei et al. [10] predicted the cavitation failure in connecting rod BE bearing of a diesel engine using the elasto-hydrodynamic analysis. Finally, Chamani et al. [11] analyzed the Thermo-Elasto-HydroDynamic (TEHD) model of the oil film lubrication in BE bearing of a diesel engine and thus also considered the influence of temperature. The results showed that TEHD method estimates higher maximum oil film pressure and lower minimum oil film thicknesses.

The experimental study of Das et al. [12] is to investigate the effects of process parameters on performance characteristics (surface roughness, machining force and flank wear) in hard turning of AISI 4340 steel. Combined effects of cutting parameter on performance outputs were explored employing the analysis of variance. Results showed that, feed rate has statistical significance on surface roughness and the machining force is influenced principally by the feed rate and depth of cut whereas, cutting speed is the most significant factor for flank wear followed by cutting speed. Generally, vehicle engine mounts are used by three types of targets (motor bearing weight, motor vibration absorption, motor overloading, acceleration or braking). Hosseini and Marzbanrad [13] described a concise functional overview of the engine mount in automobiles. Afterwards, the structure and the basic functional of hydraulic engine mount were described as the most common engine mount categories. Finally, advantages and disadvantages

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of various types engine mounts with capability of use in the vehicle were compared with each other. Heidari [14] proposed a novel intelligent method based on wavelet packet transform and multiple classifier fusion which was applied to the fault diagnosis of gears and bearings in the gearbox. The diagnosis results show that the proposed method is able to reliably identify the different fault categories which include both single fault and compound faults, which has a better classification performance compared to any one of the individual classifiers.

In the present paper, an attempt has been made to provide and introduce a practical method to calculate the wear level in connecting rod bearings, which is one of the most critical abrasive parts in the engine. Since the design of bearing housing affects the deformation of bearing shell, therefore it affects the amount of local clearance and wear in the bearing. Thus, using the introduced method, the design of bearing surrounding parts can be evaluated. It is also possible to assess whether the wear occurred in the bearing is progressive, or whether the abrasive surfaces will be compatible with each other after a period of operation and consequently the wear stops. In current study, in order to calculate the wear volume of BE bearing, a model based on the Archard wear model is used.

2. Simulation

In the present paper, BE bearing analysis of a V-16 diesel engine with medium speed turbocharge is conducted. This engine generates 1700 kW power at the rated speed of 2000 rpm. The lube oil SAE grade is 15W40. In the following, the models applied to the assessment of erosive wear damage is presented.

2.1. Elasto-HydroDynamic Model (EHD)

In this paper, the bearing analysis was carried out using EHD model. AVL\Excite® software is a powerful tool for bearing analysis which was employed for EHD analysis of connecting rod BE bearing. EHD model solves the Reynolds equation (1) [15] in the bearing surface. It contains a mass-conserving cavitation model, which is reflected by an additional variable named clearance fill ratio θ. Reynold´s equation is solved for pressure p in the lubrication region and for fill ratio θ in the cavitation region. Fill ratio is defined as the fraction of volume filled with oil to the total volume [16]. For $\theta = 1$, the equation becomes the ordinary Reynold's equation. It means that the gap is fully filled with oil. $\theta = 0$ indicates a completely empty gap.

$$\frac{\partial}{\partial x}\left(\frac{\theta h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{\theta h^3}{12\eta}\frac{\partial p}{\partial z}\right) = \left(\frac{u_1+u_2}{2}\right)\frac{\partial(\theta h)}{\partial x} + \frac{\partial(\theta h)}{\partial t} \tag{1}$$

The effect of elastic displacements of the bearing surface has to be included. The oil film thickness h is expressed by Eq. (2). The initial geometrical clearance, misalignment of shaft, and elastic deformation are considered in this equation.

$$h(\beta,z) = C_R - \left(e_x + \alpha_y Z\right)cos\beta - (e_y + \alpha_x Z)sin\beta + \delta(\beta,z) \tag{2}$$

Radial deformation of the bearing surface obtains from the nodal displacements of the bearing surface along the radial and circumferential axes. The nodal displacements of the bearing surface are determined by solving the equations of motion for the condensed bearing structure. Barus' equation is applied to define the oil viscosity:

$$\eta = \eta_0 e^{\alpha p} \tag{3}$$

where, η_0 is the viscosity at the ambient pressure and α is the pressure viscosity coefficient.

2.2. Boundary Lubrication Model

When lubrication regime of engine bearings shifts from fully hydrodynamic to mixed lubrication, the bearing clearance drops to an extremely small level and the surface asperities of the shaft and bearing start interacting with each

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other. Figure 2 depicts the contact between two sliding surface asperities in the boundary lubricating condition.

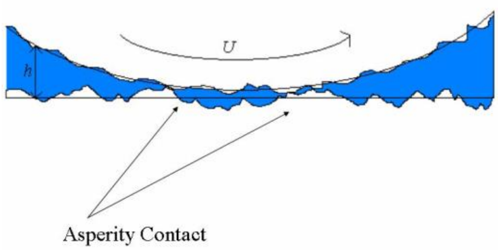

Figure 2: Boundary lubrication between two rough surfaces

AVL/Excite® software applies the model of Greenwood and Tripp [17] to model the asperity interaction of two rough surfaces in the mixed-lubrication regime [15]. This model considers the contact between two nominally flat surfaces with a Gaussian distribution of asperity height and a fixed radius of asperity curvature. The nominal pressure generated by the asperities at contact is expressed as follows:

$$P_a = KE^*F(\frac{h}{\sigma_s}) \quad (4)$$

where K denotes elastic factor, E^* composite elastic modulus and F is a factor which is function of $\Delta = h/\sigma_s$. h denotes the nominal clearance between the contacting surfaces and σ_s is standard deviation of the summits of two surface asperity and is called composite surface asperity height. Elastic factor describes the surface topography which can be experimentally determined. Table 1 shows the surface roughness of shaft. Elastic factor K is calculated in as new (unworn) and worn condition of bearing.

Table 1: Surface roughness of bearing and shaft

Items	shaft roughness (μm)	bearing roughness (μm)
R_a	0.14	0.57
R_z	1.54	1.3
R_{max}	2.15	1.33
R_t	2.47	1.12

2.3. FE Model Reduction

The nodal displacements of the bearing surface are ascertained by solving the equations of motion for the condensed bearing structure. Thus, the mass and stiffness matrixes of the condensed bearing and connecting rod should be extracted using sub-structuring method. to decrease the analysis time, only that piece of bearing housing which has effect on deflection and deformation of bearing is modeled using FE by Abaqus® software. Linear tetrahedral elements were used for FE meshing of connecting rod. Due to steep pressure gradient and probability of wear at the edges of BE bearing, the FE mesh was refined near the bearing edges. FE mesh in circumferential direction was uniformly produced.

In AVL\Excite® software, for the connecting rod model, nonlinear joint type (NONL) was employed in Small Eye (SE) bearing and EHD joint type is used in BE bearing. To consider the effect of crankshaft flexibility on the oil film performance of BE bearing, a model consisting crankshaft, rigid crankcase and connecting rods was created. Figure 3 shows the condensed model of conrod.

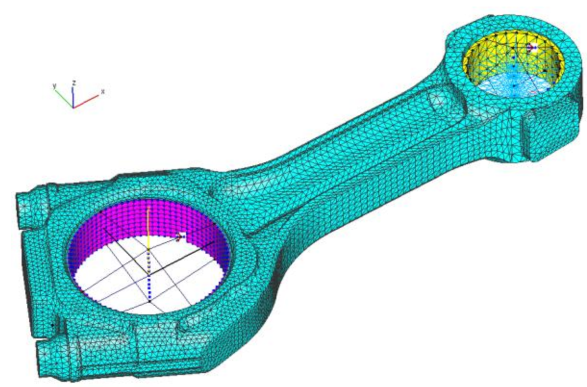

Figure 3: Condensed model conrod in AVL\Excite®

2.4. Wear Assessment Model

There are some wear models based on the fatigue wear of bearing material. Sastry et al [18] proposed a fatigue wear model for the wear of lubricated surfaces and considered the plastic

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strain at asperities as a main reason of low cycle fatigue of asperities. Zhu et al [19] classified the sliding wear models and concluded that Eq. (9) can be used as the general form of sliding wear:

$$\frac{dW}{dt} = k\frac{p^a u^b}{H^c} \quad (5)$$

where dW/dt is wear rate, p contact pressure, u relative sliding velocity, H material hardness, k wear coefficient and a, b and c are constants. Archard's model is abundantly referenced because of its simplicity and wide application in different practical cases [19]. In this study, Archard's model is used to predict wear in BE bearing material.

2.5. Implementation of Wear

Erosive wear damage causes a change in surface roughness of bearing and increases the local radial clearance between bearing and shaft. It leads to adjust in the oil film performance, oil film pressure distribution and oil film thickness. Thus, along with wear progression, a new EHD analysis should be performed to investigate the updated oil film performance. Initially, an EHD analysis of as new bearing is performed considering the asperity contact properties of as new bearing. Bearing analysis results in each node are imported in the wear code, which was written in MATLAB® software. This calculates the wear of bearing and updates the radial clearance in each node. The bearing area curve represents the distribution of bearing surface roughness, therefore, it is used as the key parameter for surface roughness evolution during the wear progression. Figure 4 depicts the bearing surface roughness in as new bearing and worn bearing conditions.

The calculated worn shape of bearing is mapped on the nodes of bearing surface. The wear amount is added to the diametric clearance between bearing and shaft. Then, a new EHD analysis is carried out and this process is repeated until wear rate decreases to a small value. Wear rate at initial stage of engine operation is high and then decreases due to adjust in bearing surface geometry and oil film characteristics, too. Thus, the time interval to calculate the wear amount at the initial stage is small and then will increase.

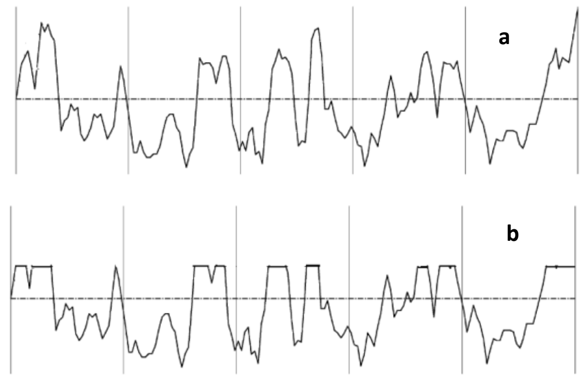

Figure 4: Bearing surface roughness; a) As new bearing, b) Worn bearing

3. Results and Discussions

Minimum oil film thickness usually cannot indicate the potential bearing wear level. But, this characteristic is good guidance to design the bearing and investigate its failure. Figure 5 shows the graph of Minimum Oil Film Thickness (MOFT) and Maximum Oil Film Pressure (MOFP) of BE bearing of connecting rod versus crank angle in one engine operating cycle. MOFT has occurred at the vicinity of MOFP crank angle position. The value of MOFT is about 0.2 μm which is lower than surface roughness, therefore, the boundary lubrication regime and asperity contact pressure are expected at this location.

Figure 6 depicts the peak asperity contact pressure and its angular position in BE bearing at each crank angle.

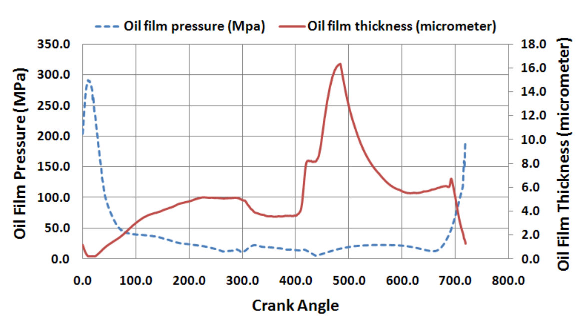

Figure 5: Maximum oil film pressure and minimum oil film thickness in BE bearing of connecting rod

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Distribution of asperity contact pressure in BE bearing is shown in Figure 7. BE bearing structure significantly deforms due to firing load and the minimum oil film thickness takes place at the vicinity of maximum firing load. The deformation of BE bearing structure forms locally concave shape in the axial direction. The oil film is thick enough at the center line and gradually becomes thinner toward the edges of bearing. Asperity contact pressure happens at the edges of bearing.

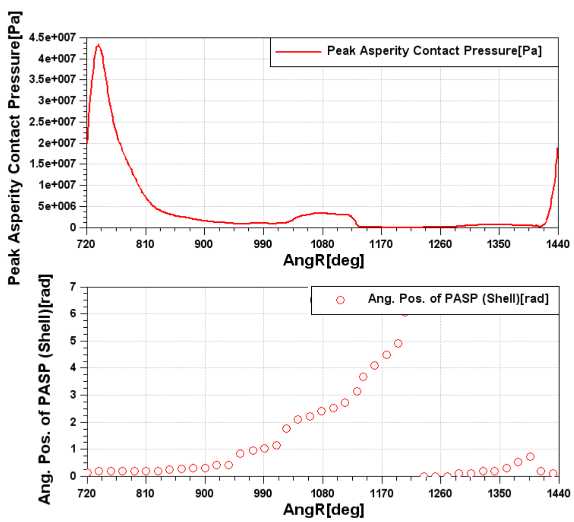

Figure 6: Asperity contact pressure and angular position in BE bearing

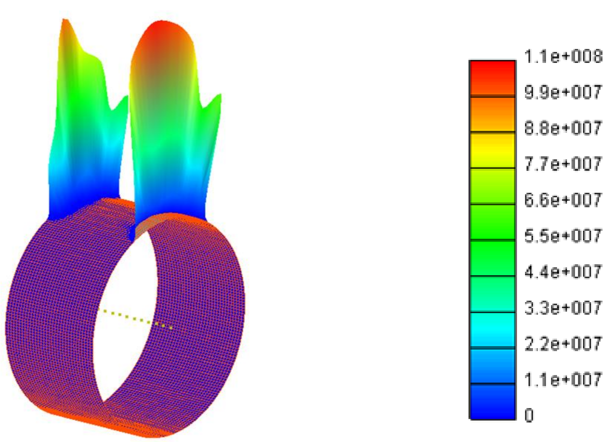

Figure 7: Distribution of asperity contact pressure in BE bearing

Figure 8 shows the distribution of total pressure of oil film at crank angle related to maximum oil film pressure. Total pressure is hydrodynamic pressure plus asperity contact pressure. Due to deformation of BE bearing, oil film pressure expands over the loaded area. Furthermore, asperity contact pressure, at the edges of bearing, is added to the hydrodynamic oil film pressure. Oil film distribution is in concave shape and the peak of oil film pressure as well as minimum of oil film thickness take place at near of the bearing edges. The oil film pressure at the edges sharply drops down to the ambient pressure.

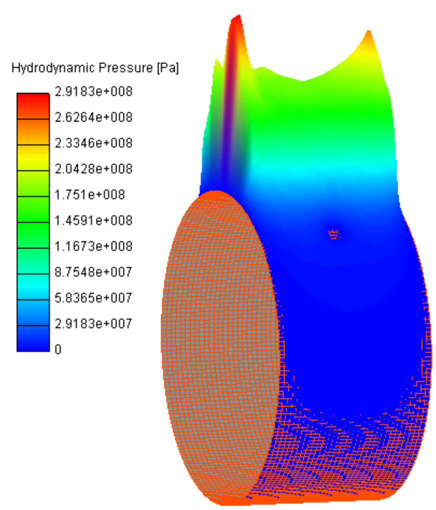

Figure 8: Distribution of total oil film pressure at maximum oil film pressure

Shaft surface roughness was the same in both of the cases. Figure 9.a shows the wear depth of BE bearing of connecting rod with mentioned surface roughness condition at initial stage of engine operation and Figure 9.b shows the progress of wear depth in running-in. Erosive wear damage is happened locally at the edges of bearing. By comparing the wear distribution and oil film thickness distribution and also considering locations of asperity contact, it is observed that erosive wear damage location corresponds to location of MOFT and asperity contact. On the other hand, erosive wear damage locations are coincident with minimum oil film thickness positions.

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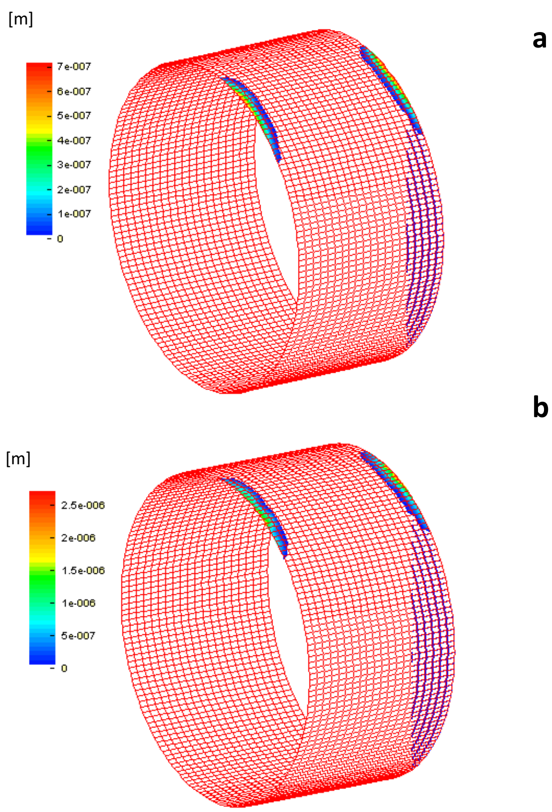

Figure 9: Progress of wear profile for BE bearing

The main loaded region in the bearing, which is under firing load, is subject to extensive wear at the both edges of bearing. Wear at the edges of bearing changes the bearing surface geometry and subsequently the surface roughness, therefore, the distribution of oil film pressure and oil film thickness are modified. Along with progress in erosive wear damage, the distribution of oil film pressure in axial direction adjusts. During the wear process, since the asperities peaks are removed from the surface, the surface roughness will certainly decrease. If the wear becomes more severe to the extent that exceeds the asperities, then the material will also be removed. Due to material removal from the bearing surface, the local clearance height increases and the surface roughness of bearing reduces. The surface roughness in bearings is 1.6 microns. When more than this amount is removed, it means that the material is removed from the surface of the bearing. As a results, the local clearance height increases. Results display that by progression of wear, the amount of oil film thickness increases and the asperity contact pressure decreases.

Figure 10 plots the variation of local hydrodynamic oil film pressure during the progression of bearing erosive wear damage. Along with progression of wear, hydrodynamic pressure reduces at the edges of bearing and hydrodynamic pressure forms in the convex shape. Figure 11 plots the variation of local clearance height during progression of bearing erosive wear damage. Wear tends to increase the clearance height and uniform the shape of bearing deformation. If the wear level is slight, it may not lead to fatigue crack initiation and therefore it can solve the contact problem at the edges of bearing.

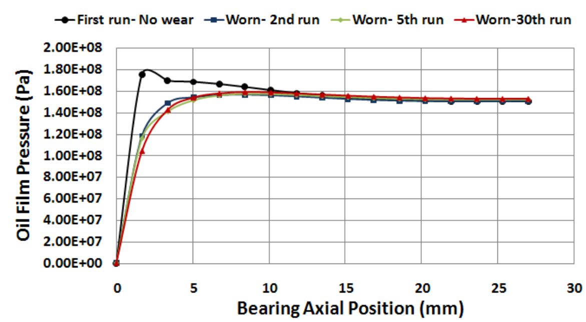

Figure 10: Variation of local hydrodynamic oil film pressure during wear progression of bearing

Figure 11: Variation of local clearance height during wear progression of bearing

4. Conclusions

High-speed diesel engines generally use un-grooved BE bearings and do not supply the piston cooling oil via a connecting rod drilling. In the present paper, an appropriate algorithm for prediction and failure analysis of wear in BE

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bearing of engines was presented using EHD model. The numerical simulation results confirm that the wear rate at the initial stage of engine running is significant. Wear at the edges of bearing changes the surface geometry and surface roughness, therefore, the distribution of oil film pressure and oil film thickness will modify. If the wear level is not significant, wear will not lead to fatigue crack initiation or bearing failure. In this condition, wear will adapt the bearing geometry in proper condition and improve the contact problem at the edges of bearing.

Declaration of Conflicting Interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

List of symbols

a, b and c	Constants of wear model
C_R	Bearing radial clearance
dW/dt	Wear rate
E	Elastic modulus
E^*	Composite elastic modulus
e_x	Eccentricity in X direction
e_y	Eccentricity in Y direction
h	Oil film thickness
K	Elastic factor
p	Oil film pressure
t	Time
u_1	Oil film velocity in X direction
u_2	Oil film velocity in Z direction
X	Vertical position of bearing nodes
Y	Horizontal position of bearing nodes
Z	Axial position of bearing nodes

Greek symbols

α	Pressure viscosity coefficient
α_x	Misalignment around X axis
α_y	Misalignment around Y axis
α_z	Misalignment around Z axis
β	Circumferential coordinate of bearing
δ	Radial deformation of the bearing surface
η_0	Viscosity at ambient pressure
η	Viscosity
θ	Fill ratio
σ_s	Composite surface asperity height (r.m.s)
v	Poisson ratio

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