Crank Bearing Design Based on 3-D Elasto-hydrodynamic Lubrication Theory

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The demand for high efficient diesel engines makes ample design of crank shaft bearings difficult. Deformation due to bearing load is in millimeters, while bearing oil film thickness is less than several microns. This paper details the application of 3-D elasto-hydrodynamic lubrication theory to crank shaft bearings for 4-stroke diesel engines. The theory includes bearing deformation and oil film history in a bearing gap.

1. Introduction

Recently, the output of internal combustion engines has been increased and their weight has been reduced. As a result of this, bearings are used under severer operating conditions, and deformation of housings of connecting rod big-end bearings and main bearings have significant influence upon the bearing characteristics. To solve this problem, Mitsubishi Heavy Industries, Ltd. (hereinafter referred to as MHI) has developed a bearing characteristic prediction method applying the elasto-hydrodynamic lubrication (EHL) theory for these dynamically loaded bearings, and used the method to design and evaluate MHI’s large-bore diesel engines.

The EHL analysis coupling oil film pressure with elastic deformation of bearing surface enables to consider the influence of local bearing surface deformation and evaluate the quantitative bearing performance with remarkably improved accuracy of bearing characteristic prediction as compared to the conventional analysis assuming bearings to be rigid bodies.

In addition, in these years, MHI has introduced the EHL analysis considering oil film history that couples oil film pressure with bearing deformation while tracing the oil filling ratio in bearing clearance according to the time history to improve the evaluation accuracy.

This report introduces examples of application of this technique to connecting rod big-end bearings and main bearings of MHI’s 4-stroke diesel engines.

2. Theory

2.1 Basic equations

Fig. 1 shows the system of coordinates used in this report.

Reynolds equation that controls the oil film pressure distribution $p$ is expressed as eq. (1).

$$\nabla e^{-\sigma p\nabla p} - \frac{12}{\pi} \left( \frac{L}{2} \frac{\partial (\phi h)}{\partial \theta} + \frac{\partial (\phi h)}{\partial \theta} \right) = 0$$

When the equation (1) and the following equations of force equilibrium are combined into a simultaneous system of equations with respect to time $t$, information on shaft center locus and oil film thickness can be obtained.

$$\int_{0}^{\theta} p(-\cos \theta) dQ - W_x = 0$$

$$\int_{0}^{\theta} p(-\sin \theta) dQ - W_y = 0$$

The oil film thickness $h$ is expressed by eq. (4) in consideration of the initial geometrical clearance, misalignment of shaft, and elastic deformation.

$$h = c_r + (\epsilon_x + \alpha Z) \cos \theta + (\epsilon_y + \alpha Z) \sin \theta + L p$$

where,
- $\sigma$ : Viscosity pressure coefficient
- $\epsilon_x$ : Misalignment around X axis
- $\epsilon_y$ : Misalignment around Y axis
- $c_r$ : Bearing radial clearance
- $e_x$ : Eccentricity in X direction
- $e_y$ : Eccentricity in Y direction
- $h$ : Oil film thickness
- $L$ : Compliance
- $N$ : Engine speed
- $\Omega$ : Area
- $p$ : Oil film pressure

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2.2 Analyzing technique

2.2.1 EHL analysis considering oil film history

We developed an EHL analysis considering the movement of oil in the bearing clearance based on the concept of oil film history proposed by Jones(2). MHI’s conventional EHL analysis calculated the pressure distribution on the assumption that the whole bearing surface is covered with lube oil, replaced the negative pressure region obtained as the result of the calculation with the ambient pressure and regarded the boundary of the region as the oil film rupture boundary. Under this boundary condition, flow continuity on the oil film rupture boundary cannot be filled. On the other hand, the EHL analysis considering oil film history follows the oil filling ratio with time so that the flow continuity is filled.

For bearings that receive fluctuating load, such as engine bearings, the bearing area where positive oil film pressure can develop is restricted, and, therefore, the EHL analysis may show higher pressure than that obtained by the conventional EHL analysis. This was verified by actual measurement on the main bearings of MHI’s large-bore marine diesel engines. The EHL analysis considering oil film history is used for design and evaluation as a useful analysis tool.

2.2.2 Calculating method

The oil film pressure $p$ and shaft eccentricities $e_x$ and $e_y$ can be obtained by the simultaneous equations (1) to (4). Since the equation (1) is nonlinear with respect to the oil film pressure, we used Newton-Raphson method to determine them. We used the finite element method (FEM) (Galerkin method) for numerical calculation, four-point isoparametric elements as elements and Skyline method for numerical solution of the linear equation system. To determine the oil film rupture boundary, we improved and used the technique of Kumar, et al. (3) that applies the oil film history algorithm to the finite element method. Fig. 2 shows the flow of the calculation.

3. Case study

3.1 Connecting rod big-end bearing

Using the connecting rod big-end bearing of an S3L engine, we compared the rigid body analysis and the EHL analysis, and compared the EHL analysis considering oil film history and the conventional EHL analysis.

Moreover, we evaluated the influence of the shape of the crank pin on the bearing characteristics of the connecting rod big-end bearing.

Table 1 shows the specifications for the bearing, and Fig. 3 shows the load vector that affects the bearing and the 3D FEM model used to calculate the compliance.

| Table 1  Dimensions of S3L engine connecting rod big-end bearing |
|---------------------------------+-----------------|
| Bearing diameter                | 48 mm           |
| Bearing width                   | 21 mm           |
| Radial clearance                | 0.025 mm        |
| Rod length                      | 145 mm          |
| Stroke                          | 78.5 mm         |
| Engine speed                    | 3,600 rpm       |
| Lube oil viscosity              | 10 cP           |

Fig. 2 Flow of calculation
3.1.1 Comparison of analyzing techniques

Fig. 4(a) shows the change in the maximum oil film pressure with time. At almost all timing points, the oil film pressure determined by the rigid body analysis is higher than that obtained by the EHL analysis. This is apparently indicated, for example, at the explosion timing at a crank angle of about 10° where the load is relatively large. The maximum oil film pressure determined by the EHL analysis is 133 (MPa), while that determined by the rigid body analysis is 180 (MPa). Figs. 4(c) and (d) show the oil film thickness distribution and the oil film pressure distribution in the bearing center part at a crank angle of 10°. In consideration of the elastic deformation, the oil film thickness obtained by the EHL analysis is larger as compared to that obtained by the rigid body analysis and almost uniform in the pressure region. Compared with the rigid body analysis, the EHL analysis gives pressure distribution wider in the circumferential direction and shows lower maximum oil film pressure.

As is evident from the shaft center locus shown in Fig. 3 Conditions of calculation for bearing Change in bearing load and 3D solid model for calculation of deformation are shown.

(a) Bearing load  (b) FEM model of big-end

Fig. 3 Conditions of calculation for bearing

(a) Change in max. oil film pressure with time

(b) Shaft center locus

Fig. 4 Results of analysis of characteristics of big-end bearing

When the elastic deformation is taken into consideration, the calculated oil film pressure decreases, and the calculated oil film thickness increases. When the oil film history is taken into consideration, the bearing area where positive oil film pressure can develop is restricted, and the pressure peak becomes higher.
Fig. 4(b), the EHL analysis indicates that the shaft center is remarkably eccentric over the bearing clearance.

In Fig. 4(a), the maximum oil film pressure at a crank angle of about 250° determined by the EHL analysis considering oil film history is different from that determined by the conventional EHL analysis. This crank angle timing corresponds to the point of shift of load from the upper metal to the lower metal, where the shaft moves toward the side having an oil film rupture.

Fig. 4(e) shows the oil film pressure distribution and oil filling ratio at a crank angle of 250°. As is evident from this figure, the analysis considering oil film history shows that the bearing area where positive oil film pressure can develop is restricted owing to the shortage of oil, and gives higher oil film pressure as compared to that given by the conventional EHL analysis.

3.1.2 Influence of crank pin shape on bearing characteristics

We evaluated the influence of the crank pin shape on the characteristics of connecting rod big-end bearings. We examined three crank pin shapes shown in Fig. 5, i.e. (1) straight, (2) barrel shape, and (3) hourglass shape. Fig. 6 shows the results of analysis of the maximum oil film pressure and minimum oil film thickness. It is revealed that the straight crank pin (1) gives higher oil film thickness and is suitable for the operating conditions of the bearings.

3.2 Main bearing

This analyzing technique is applicable not only to big-end bearings, but also to main bearings and small-end bearings. An example of analysis of the No.4 main bearing of an S6R engine is shown below. Table 2 shows the specifications for the bearing, and Fig. 7 shows an FEM model of the engine frame including the bearing load and main bearing.

The time history of maximum oil film pressure shown in Fig. 8(a) indicates that the rigid body analysis estimates the pressure higher as compared to the EHL analysis. As is evident from the pressure distribution at a crank angle of 245° shown in Fig. 8(b), the EHL analysis considering oil film history restricts the bearing area where positive oil film pressure can develop owing to the shortage of lube oil and gives higher oil film pressure than that obtained by the conventional EHL analysis. Accordingly, it is considered that the analysis of main bearings shows a similar tendency to the analysis of big-end bearings.

4. Conclusion

The EHL analysis and EHL analysis considering oil film history are introduced as the most advanced techniques to improve the reliability of engine crank shaft systems.

Table 2 Dimensions of S6R engine No. 4 main bearing

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bearing diameter</td>
<td>140 mm</td>
</tr>
<tr>
<td>Bearing width</td>
<td>53 mm</td>
</tr>
<tr>
<td>Radial clearance</td>
<td>0.07 mm</td>
</tr>
<tr>
<td>Engine speed</td>
<td>1 800 rpm</td>
</tr>
<tr>
<td>Lube oil viscosity</td>
<td>10 cP</td>
</tr>
<tr>
<td>Oil groove</td>
<td>Horizontal and lateral, 45°, 100 mm wide</td>
</tr>
</tbody>
</table>
To design bearings for development of lighter engines that have higher power, we will ensure higher reliability using the above-mentioned evaluation techniques and enhance the convenience of the techniques linking them with three-dimensional CAD for systematization. Part of this development was conducted in cooperation with the Truck & Bus Research & Development Center, Mitsubishi Motors Inc.

Fig. 8 Results of analysis of main bearing characteristics

The pressure peak increases when the oil film history is taken into consideration.

References
(1) Furuno, K. et al., Improvement of Reliability for UEC Marine Diesel Engines, Mitsubishi Juko Giho Vol.34 No.4 (1997) p.256