AN EHD STUDY OF A CONNECTING ROD BIG END BEARING INCLUDING ELASTICITY AND INERTIA EFFECTS OF THE BEARING STRUCTURE

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SUMMARY
Journal bearings of modern internal combustion engines often operate in the mixed lubrication regime. In order to have a better understanding of the tribological behaviour of these bearings and hence improve bearing design, a more rigorous elastohydrodynamic lubrication analysis is required. In the present work, an EHD simulation has been conducted for a big end connecting rod (conrod) bearing. The bearing is of 390.5 mm in diameter in a MB430 large diesel engine manufactured by Mirrlees Blackstone. Effects of the elasticity of the bearing structure and the pressure-dependence of oil viscosity on bearing performance characteristics have been examined. It is shown that an EHD calculation is necessary for this bearing in order to capture its detailed behaviour and to establish a realistic film pressure distribution as a boundary condition for stress analysis of the rod, and subsequent fatigue and wear evaluation.

Keywords: elastohydrodynamics (EHD), lubrication, bearing, internal combustion engine, connecting rod

1 INTRODUCTION
Connecting rod (conrod) and crankshaft (main) bearings are important components of an internal combustion (IC) engine. Their tribological performance has a significant impact on the mechanical efficiency and durability of an engine. For satisfactory operation of these bearings, the minimum oil film thickness (MOFT) should be above a recommended level, and the peak (maximum) oil film pressure (POFP) should be below a critical value for reduced wear and long fatigue life. The bearings of modern IC engines often operate in mixed lubrication regime. Therefore, the design of these bearings calls for a more rigorous lubrication analysis.

An EHD lubrication analysis plays an important role in the design of dynamically loaded engine bearings as it can offer more realistic predictions of the bearing performance characteristics such as the film thickness, pressures, power losses and flow rates. Two major features of an EHD model are (1) inclusion of the effect of elastic deformation of the bearing shell/structure, and (2) consideration of the piezo-viscous effect of lubricants. Many researchers and engineers have developed various sophisticated EHD models for dynamically loaded journal bearings [1–4].

In this study, a big end conrod bearing has been analysed using the bearing lubrication module of AVL EXCITE software package, which considers effects of both elasticity and inertia of the bearing structure. The bearing was from a MB430 medium speed diesel engine manufactured by Mirrlees Blackstone for power generation. The bearing performance characteristics were obtained for both rigid and elastic cases. Results show a significant influence of the elasticity on the bearing behaviour.

2 THEORY
The Reynolds equation used in this study is written as follows,

\[
\frac{\partial}{\partial x} \left[ \frac{\theta h^3}{12\eta} \frac{\partial p}{\partial x} \right] + \frac{\partial}{\partial z} \left[ \frac{\theta h^3}{12\eta} \frac{\partial p}{\partial z} \right] = \left( u_t + u_z \right) \frac{\partial (\theta h)}{\partial x} + \frac{\partial (\theta h)}{\partial t}
\]

(1)
The coordinate system used is illustrated in Figure 1. Equation (1) includes a mass conserving cavitation model, which is reflected by the additional variable,
clearance fill ratio \( \theta \). If \( \theta = 1 \), the equation becomes the ordinary Reynolds equation.

For an EHD analysis, the effect of elastic displacements of the bearing surface has to be included. The film gap including this effect is written as,

\[
h(\beta, z) = C_R - e_x \cos \beta - e_y \sin \beta + \delta(\beta, z)
\]

(2)

where, \( \delta \) is the radial deformation of the bearing surface, which can be obtained from the nodal displacements of the bearing surface along the x and y axes in the Cartesian coordinate system as depicted in Figure 1.

The nodal displacements of the bearing surface are determined by solving the equations of motion for the condensed bearing structure (discretized elastic bodies). These equations can be written in the vector form as follows,

\[
M_B \ddot{u}_B + D_B \dot{u}_B + K_B u_B = F
\]

(3)

where, \( M_B \) is the reduced mass matrix of the bearing structure, \( \ddot{u}_B \) is the acceleration matrix of the bearing surface, \( D_B \) is the damping matrix of the bearing, \( \dot{u}_B \) is the velocity matrix of the bearing surface, \( K_B \) is the reduced stiffness matrix of the bearing structure, \( u_B \) is the displacement matrix, and \( F \) is the matrix of nodal forces.

The oil viscosity is determined by Barus’ equation,

\[
\eta = \eta_0 e^{\alpha p}
\]

(4)

where, \( \eta_0 \) is viscosity at ambient pressure, \( \alpha \) is pressure-viscosity coefficient.

The finite difference (FD) scheme has been used to discretize the Reynolds equation. The Jakobsson-Floberg and Olsson boundary conditions have been implemented to determine the film reformation and cavitation boundaries.

3 STIFFNESS AND MASS MATRICES

It is known from equation (3) that prior to conducting the EHD simulation of a bearing, the stiffness and mass matrices have to be extracted from the bearing structure. This was done using a finite element (FE) analysis. 8 node hexahedral elements were used to mesh both the bearing bush and the big end structure. This has been proven by the authors to be necessary in order to obtain more accurate EHD results.

Figure 2 shows the FE mesh of the bearing structure. It can be seen that there are 48 and 8 elements along the circumferential and axial directions respectively. They are equally spaced. The nodes of the bush and the big end of the rod are equivalenced at the interface.

The stiffness and mass matrices were extracted using the standard condensation algorithms, which are available in both ABAQUS and NASTRAN. For the present case, ABAQUS was used to carry out the condensation analysis. The stiffness and mass of the bearing structure were reduced to the master nodes (48 \( \times \) 9) on the bearing surface in the Cartesian coordinate system as can be seen in Figure 2. The material properties for both the rod and bush were included in the FE analysis.

4 ANALYSIS

For the lubrication analysis, a uniform FD mesh was used, which had 25 x 145 nodes in the axial and circumferential directions respectively. The converged results were obtained in two engine cycles.

Three calculations were performed: (1) EHD with a pressure-dependence oil viscosity (PDV), (2) EHD with a constant viscosity (CV) of 0.0256 Pa\( \cdot \)s, and (3) rigid hydrodynamic analysis (RHD) with the CV. Table 1 lists the input data for these calculations.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Journal diameter</td>
<td>390.5 (mm)</td>
</tr>
<tr>
<td>Bearing width</td>
<td>148 (mm)</td>
</tr>
<tr>
<td>Crank throw</td>
<td>240 (mm)</td>
</tr>
<tr>
<td>Rod length</td>
<td>1134 (mm)</td>
</tr>
<tr>
<td>Diametric clearance</td>
<td>0.5165 (mm)</td>
</tr>
<tr>
<td>Engine speed</td>
<td>600 (r/min)</td>
</tr>
<tr>
<td>Oil type</td>
<td>SAE 40</td>
</tr>
<tr>
<td>Viscosity at ambient pressure</td>
<td>0.0256 (Pa s)</td>
</tr>
<tr>
<td>and 75 ( ^\circ )C (measured)</td>
<td></td>
</tr>
<tr>
<td>Viscosity-pressure coefficient</td>
<td>0.0216 (MPa(^{-1}))</td>
</tr>
<tr>
<td>Oil supply pressure</td>
<td>0.5 (MPa)</td>
</tr>
<tr>
<td>Width of central oil groove</td>
<td>26 (mm)</td>
</tr>
<tr>
<td>Extent of oil groove</td>
<td>67.5(^{\circ}) to 292.5(^{\circ})</td>
</tr>
<tr>
<td>Diameter of oil holes in journal</td>
<td>40 (mm)</td>
</tr>
<tr>
<td>Positions of the oil holes</td>
<td>@ -90(^{\circ}) and 90(^{\circ})</td>
</tr>
</tbody>
</table>

Table 1: Engine data and bearing sizes
5 RESULTS AND DISCUSSION

5.1 Effect of Bearing Elasticity

For the following figures the converged results are plotted (i.e., the second engine cycle); TDC firing corresponds to 1080° crank angle.

Figures 3 and 4 show comparison of the MOFT and POFP variations over an engine cycle for the elastic and rigid cases with the constant oil viscosity. It can be seen that generally, for the bearing studied the rigid hydrodynamic (RHD) analysis predicts larger MOFT than the EHD solution using the constant viscosity (CV), but it gives significantly higher POFP because of no elastic compliance. Some detailed film shape and pressure profiles are presented in section 5.3.

Figure 5 compares the oil flow rates over the engine cycle for the EHD and RHD cases. It can be noted that the oil side leakage rates for the EHD case are higher than those for the RHD case. The cyclic average of the oil flows from the EHD solution is about 17 per cent higher than that from the RHD solution. This arises since the angular extent of the film pressures is larger for the elastic case than that for the rigid case.

Figure 6 presents the predicted frictional losses over the engine cycle from the two solutions. The cyclic average of the frictional losses from the EHD solution is about 12 per cent higher than that from the RHD solution. This is due to the wrapping effect of the bearing around the journal for the elastic case.

5.2 Effect of Pressure-Dependence of Lubricant Viscosity

This section examines what effect the inclusion of the pressure dependence of oil viscosity may have on the predicted MOFT and POFP of the big end bearing using an EHD solution.

Figure 7 shows comparison of the MOFT predicted using the Barus viscosity relation and the constant oil viscosity. It is noticed that generally the use of pressure-dependent viscosity can increase the MOFT values over the engine cycle.

Figure 8 compares the POFP values over the engine cycle for the two cases. Again, when the Barus relation is used, a larger POFP is predicted over most of the engine cycle. The absolute maximum value for the PDV case is about 30 per cent higher than that for the CV case.
5.3 Film and Pressure Profiles

In order to take a closer look at the influence of the bearing elasticity on the predicted bearing behaviour, an inverted three-dimensional film shape and a pressure profile at a crank angle of 990° (270°), where the absolute MOFT occurs (see Figure 7), are presented in Figure 9 and 10 for the PDV case. It can be seen from Figure 9 that the smallest film thickness is localised and happens at the edges of the bearing in the rod half. (The 0 degree of bearing angle is in the rod (x) axis.) This correlates well with the wear marks from engine tests [5]. Whereas, the RHD analysis can only predict the uniform film across the bearing width. This suggests that the use of the RHD film pressure distribution and film shape cannot determine an accurate stress distribution and wear pattern in the bearing.

It is clear from Figure 10 that the positive pressure region is consistent with the region where the film thickness is small. It is noteworthy that the positive pressure region is only a small fraction of the bearing area. This occurs since the bearing structure is much stiffer in the rod half and there is no oil groove in this region of the bush.

6 CONCLUSIONS

The EHD simulations have been conducted for a big end bearing of large diesel engine for power generation using AVL EXCITE. From this study, the following conclusion may be drawn:

- The elasticity of the bearing structure has a significant influence on the predicted performance characteristics of the big end bearing, and this effect should be considered in the bearing design.
- The inclusion of the pressure-dependence of oil viscosity appears to have a favourable effect on the predicted MOFT. However, the authors believe that the thermal effect should also be considered in the conjunction with the pressure influence on the oil viscosity.
- EXCITE has proved to be an effective and robust tool for the EHD and RHD analysis of dynamically loaded journal bearings.

7 REFERENCES