Study on heavy-duty journal bearing lubrication based on Elasto-hydrodynamic lubrication method

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Abstract. Main bearing wear failure to become one of the main factors affecting the service-life of the engine, so it is necessary to form the effective analysis of the main bearing lubrication method, which is used in the design of main bearing lubrication and friction. In this paper, the model of substructure of V8 diesel engine is built based on flexible multi-body dynamics, the various nonlinear factors in the process of engine working process are considered, and the EHD of main bearings is analysed by the EXCITE software. On the basis of the V8 engine model, the bearing load and lubrication performances of different poly-condensation of main bearing are compared. The main bearing lubrication properties of the analysis results can be used for the engine lubrication system design and optimization.

1. Introduction
In the working process of journal bearing, the heavy load will cause large deformation of bearing bush and bearing block, which will significantly change the oil film thickness distribution, and then affect the lubrication performance of bearings. The results of elastic deformation of journal bearing show that the elastic deformation may reduce the radial clearance of main bearing by 20% \cite{1}. Therefore, the effect of deformation on lubrication must be taken into account for heavy-duty journal bearing simulation. It is of great significance to form the effective analysis of the journal bearing lubrication method, which can accurate obtain the lubrication and friction behavior of the journal bearing. The results can be used to optimize the main bearing structure and lubricating system. In this paper, the journal bearing behaviors of crankshaft for a V8 heavy-duty diesel engine is studied, based on the Elastohydrodynamic lubrication method (EHD). In order to realize effective engineering application, the computational efficiency need to be improved. The dynamic substructure condensation method is used to reduce the shaft and the bearing block, obtain the flexible body model of the crankshaft and the body, and reduce the computational solution scale.

2. Journal lubrication model

2.1. The oil film thickness equation
For typical journal bearing, and the wedge clearance effect and extrusion effect are two basic factors for hydrodynamic pressure formation in dynamic journal bearings \cite{2}. Its typical structure is shown in Figure 1. $R$ is the bush radius, $r$ is the journal radius, and $c = R - r$ is the radius gap. Under cycle loads $F$, there is an eccentricity $e$ and offset angle $\theta$ between the center of the journal $O_{j}$ and the center of the
bush $O_b$, and the center of the journal $O_1$ works in an eccentric position. On the extension line of the connecting line, there is maximum clearance $h_{\text{max}}=c+e$ at one end and minimum clearance $h_{\text{min}}=c-e$ at the other end. Along the direction of the journal rotation, the clearance is a convergent wedge from large to small in the half circle from the greatest oil film thickness to the smallest oil film thickness, which is the main geometric condition under which the lubricating oil film can generate pressure to withstand load. Oil film generally exists in a range of about 180 degrees from $h_{\text{max}}$ to $h_{\text{min}}$.

The hydrodynamic (HD) lubrication model of journal bearings regards the bearing block, bearing bush and journal as rigid bodies, regardless of the effect of bearing surface topography, thermal effect and oil supply characteristics. The calculation process is simple and the results can be obtained quickly. But for heavy duty diesel engines, the calculation results deviate greatly from the actual situation. The elasto-hydrodynamic (EHD) lubrication calculation further considers the elastic deformation of bearing bush and the bearing block, the oil supply characteristics and the cavitation effect, and the lubricating oil is considered as isothermal and isoviscous [3]. The EHD lubrication simulation result is closer to the actual working condition of diesel engine.

2.2. The extended reynolds equation

Assuming that the viscosity and density of the fluid are constant along the direction of oil film thickness, the cylindrical coordinate form of the two-dimensional generalized Reynolds equation can be expressed as follows [4]:

$$\frac{1}{r^2} \frac{\partial}{\partial \theta} \left( \frac{\rho h^3}{\eta} \frac{\partial \rho p}{\partial \theta} \right) + \frac{\partial}{\partial x} \left( \frac{\rho h^3}{\eta} \frac{\partial \rho p}{\partial x} \right) = 6V \frac{\partial (\rho h)}{r \partial \theta} + 12 \frac{\partial (\rho h)}{\partial t}$$

(1)

The following Reynolds boundary conditions are used in the solution process.

$$\left\{ \begin{array}{l}
  p(\theta, \pm B/2) = p(\theta, \pm B/2) = p_a \\
  p(\theta, x) = p_s \\
  \frac{\partial p}{\partial \theta} = 0
  \end{array} \right.$$  

(2)

Where $\theta_1$ is the inlet position of the bearing; $\theta_2$ is the oil position; $p_a$ is environmental pressure; $p_s$ is for oil supply pressure and $V$ is the flow velocity. Assuming that the fluid is an incompressible flow, $\rho$ is a constant, which simplifies the equation to:

$$\frac{1}{r^2} \frac{\partial}{\partial \theta} \left( \frac{h^3}{\eta} \frac{\partial h p}{\partial \theta} \right) + \frac{\partial}{\partial x} \left( \frac{h^3}{\eta} \frac{\partial h p}{\partial x} \right) = 6V \frac{\partial (h)}{r \partial \theta} + 12 \frac{\partial (h)}{\partial t}$$

(3)

The distribution of oil film thickness is considered as the result of the combination of the rotational motion and the radial extrusion motion between the journal and the bearing. In the above equation, $V = r \Omega$, and $\Omega$ is the effective angular velocity of moving bearing, which can be obtained by the following formula:

$$\Omega = \omega_b + \omega_j - 2\dot{\epsilon}$$

(4)

Where $\omega_b$ is the bearing angular velocity, $\omega_j$ is the journal angular velocity, $\dot{\epsilon}$ is the change rate of eccentric angle. For the engine main bearing, $\omega_b = 0$, equation (5) can be simplified as:

$$\Omega = \omega_j - 2\dot{\epsilon}$$

(5)

As the thickness of the oil film can be expressed as $h = c \left(1 + \varepsilon \cos \theta \right)$, and eccentricity is $\varepsilon = e/c$, equation (1) can be expressed as:

$$\frac{1}{r^2} \frac{\partial}{\partial \theta} \left[ c^3 \left(1 + \varepsilon \cos \theta \right)^3 \frac{\partial \rho p}{\partial \theta} \right] + \frac{\partial}{\partial x} \left[ c^3 \left(1 + \varepsilon \cos \theta \right)^3 \frac{\partial \rho p}{\partial x} \right] = 6\Omega \eta c \sin \theta + 12 \eta \dot{\epsilon} c \cos \theta$$

(6)

And $\dot{\epsilon}$ is the variation of eccentricity with time, which reflects the velocity of the rotation of the journal.

If the oil filling rate $\gamma$ is considered, the generalized Reynolds equation of journal bearing is extended as follow:
\[
\frac{\partial}{\partial y} \left( \frac{1}{12\eta} \gamma h^3 \frac{\partial p}{\partial y} \right) + \frac{\partial}{\partial x} \left( \frac{1}{12\eta} \gamma h^3 \frac{\partial p}{\partial x} \right) = \gamma \left( \frac{V_i + V_z}{2} \right) \frac{\partial (h)}{\partial y} + \left( \frac{V_i + V_z}{2} \right) h \frac{\partial \gamma}{\partial y} + \frac{\partial (\gamma h)}{\partial t}
\]  

(7)

In the working process, the bush is fixed, and the shaft neck is rotated, \( V_i = 0 \), \( V_z = V \), the equation can be simplified as:

\[
\frac{\partial}{\partial y} \left( \frac{1}{12\eta} \gamma h^3 \frac{\partial p}{\partial y} \right) + \frac{\partial}{\partial x} \left( \frac{1}{12\eta} \gamma h^3 \frac{\partial p}{\partial x} \right) = \gamma \frac{V}{2} \frac{\partial (h)}{\partial y} + \frac{V}{2} h \frac{\partial \gamma}{\partial y} + \frac{\partial (\gamma h)}{\partial t}
\]  

(8)

The equation (6) is solved by finite differential method, with JFO mass conservation boundary conditions, including axial boundary conditions, circumferential boundary conditions, boundary conditions of oil supply regions and cavitation boundary conditions.

3. The establishment of dynamic model of the journal

3.1. Substructure condensation method

The modal synthesis method is a typical dynamic substructure analysis method, the method by the modal transformation, will represent the structure of the sports in physical coordinate equation expressed in modal coordinates transformation, because of the mode number of the degrees of freedom is much less than the physical degrees of freedom, which can make the structure motion equation was simplified [5].

3.2. The condensation dynamic model of the engine

Firstly, the finite element model of the crankshaft and body was built based on the geometry, and the principal degree of freedom node of the above model was defined, then the dynamic substructure condensation model of the crankshaft, including the quality matrix, stiffness matrix and geometry of the condensation, was obtained by finite element method After importing sub-structure condensation result to the software of AVL EXCITE [6], the condensation dynamic model of the whole engine is established as shown in Figure 2.

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Figure 1. The journal bearing model  
Figure 2. Condensation dynamic model of the engine.

Then the system parameters are defined in the POWER UNIT module, including the body element and the connection, the mechanical coupling relation, the engine structure parameters, and the system connection model of the engine is established as shown in Figure 2.

The engine cylinder number arrangement is defined as shown in Figure 2, and the ignition sequence of the engine is 8-4-5-7-3-6-2-1. The dynamics model of engine is driven by the explosion pressure curve of the cylinder. The working conditions of the engine at 1600 rpm are selected for analysis, and the burst pressure curve in this paper is obtained through the actual engine cylinder pressure measurement experiment, which is shown in Figure 3. The burst pressure curve of each cylinder is the same, but the crankshaft angle of each cylinder is 90 degrees different. Table 1 is the bearing specific parameters required for EHD lubrication calculation.
Figure 3. Burst Pressure Curve at 1600 rpm.

Figure 4. The journal bearing model

Table 1. Parameters of journal bearing EHD model

| Parameter                                      | Value         | Parameter                      | Value         |
|-----------------------------------------------|---------------|--------------------------------|---------------|
| Main bearing width                            | 30mm          | Oil type                       | SAE5W-30      |
| Main bearing diameter                         | 108mm         | Oil supply pressure            | 4.6 bar       |
| Radial clearance                              | 0.025mm       | Cavitation pressure            | 0.6 bar       |
| Modulus of elasticity                         | 0.008         | The coefficient of friction    | 0.05          |
| The root-mean-square root of the peak surface of crankshaft | 0.4          | The average height of the main shaft neck | 0.3            |
| The mean-square root of the peak roughness of the bearing surface | 0.8          | The average peak point height of the bearing | 0.5            |
| Oil temperature                               | 100 °C        | Oil groove position            | 315°  ~135°   |

The circumferential condensation joint of each bearing bush should correspond strictly to the axial condensation joint of the main journal. The number of circumferential condensation nodes of bearings will affect the total condensation point, and then affect the speed of bearing lubrication simulation calculation. Comparing with actual calculation experiment, 40 nodes of main bearing condensation per week are better in accuracy of result and economy of time. The journal bearing model in EXCITE as shown in Figure 4.

4. The result of EHD lubrication simulation

Results of bearing EHD can be obtained by simulating the whole engine model, such as main bearing load, maximum oil film pressure, minimum oil film thickness etc. Figure 5 shows the maximum oil film pressure and minimum oil film thickness of five main bearings in a working cycle.

(1) Maximum oil film pressure of 1st bearing

(2) Minimum oil film thickness of 1st bearing

(3) Maximum oil film pressure of 2nd bearing

(4) Minimum oil film thickness of 2nd bearing
Figure 5. Results of main bearing simulation based on EHD model at 1600 rpm

It shows that the maximum oil film pressure corresponds to the minimum oil film thickness. The calculation results show that the maximum oil film pressure of the third main bearings is the largest and the minimum oil film thickness is the smallest. Compared with the results of bearing load calculation, the fourth main bearing has the largest load, but the maximum oil film pressure is not the largest of the five main bearings, and the minimum oil film thickness is not the smallest. Thus, the greatest load bearing, lubrication performance is not the worst, bearing lubrication is also affected by the elastic deformation of bearings and other factors, accurate bearing lubrication performance results must be obtained through EHD simulation analysis.

According to the above analysis, the lubrication performance of the second, third and fourth main bearings is poor at 1600 rpm, and the third main bearing is the worst. Therefore, the average oil film pressure, the average lubricant filling rate and the misalignment of the main journal of the third main bearing in one cycle are calculated. It can be seen from the figure 6 that when the oil film pressure is large, the filling ratio of lubricating oil is small, and the misalignment of the main journal is serious.
5. Conclusions

For heavy-duty journal bearing, the structural flexibility of the shaft and the bearing block has an important influence on the dynamic characteristics of bearing behaviour. The following conclusions can be obtained through the above analysis:

- For the V8 engine, the lubrication performance of the second, third and fourth main bearings is relatively poor, especially the third main bearings.
- It is necessary to simulate the bearing lubrication performance through EHD method, because bearing lubrication is affected by the elastic deformation of bearing.
- The lubrication performance of bearings is also affected by oil film pressure, average filling rate of oil, misalignment of main bearing and etc. It is necessary to optimize the design process of bearings.

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