Analysis of Main Bearings Lubrication Characteristics for Diesel Engine

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Abstract. In order to study the main bearings lubrication characteristics for diesel engine under elastic deformation conditions, an elastic fluid dynamics simulation model of diesel engine main bearing is established. The law of the load, minimum oil film thickness and maximum oil film pressure of the seven main bearing for the diesel engine is studied. The results show that the minimum oil film thickness decreases gradually with increasing of the bearing clearance of the main bearing, but the maximum oil film pressure gradually increasing. The maximum oil film pressure and the minimum oil film thickness of the main bearing appears in the 3# and 5# main bearings.

1. Introduction
Main bearing is the key friction pair of the diesel engine, and its lubrication performance directly affects the reliability and economic performance of the diesel engine. The impact load on main bearing is increasing with the strengthening coefficient of diesel engines continuously improving, which puts forward higher requirements for lubrication characteristics of diesel main bearings [1, 2]. Therefore, in-depth study of the lubrication characteristics for diesel engine main bearings is great significance to improve its reliability and service life.

In the study, Cheng Peng analyzed the stress of the single main bearing and the big end bearing of the connecting rod, and analyzed the force and motion of the crank and connecting rod associated with it. Finally, the load of each bearing under the pressure of the cylinder was calculated [3]. On the basis of calculation of shaft orbit by Holland method, Yi Tailian analyzed the pressure distribution of bearing oil film by finite element method, and obtained the transient pressure distribution law of oil film [4]. Ma Xingguo established the oil film lubrication model of the main bearing based on Reynolds Equation. Then the joint simulation analysis was carried out on the crankshaft and the bearing system model, and the minimum oil film thickness was obtained. Finally, the oil film pressure distribution and the maximum oil film pressure under the characteristic conditions were analyzed [5].

Since the crankshaft and its support are a complex system, the crankshaft, as a continuous elastomer, is supported on the multi-elastic fulcrum. Therefore, in this paper, considering the elastic deformation of the crankshaft and bearing, the lubrication characteristics of a diesel engine are analyzed by establishing an elastic fluid dynamics simulation model of the main bearing in the paper.
2. Mathematical Model

2.1. Governing Equations

The Reynolds equation is the most basic equation in the theory of hydrodynamic lubrication, and which is the theoretical basis for studying the lubrication problem of the dynamically loaded sliding bearings.

Partially differentiated form of Reynolds equation:

\[
\frac{\partial}{\partial X} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial X} \right) + \frac{\partial}{\partial Z} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial Z} \right) = 6\omega \frac{\partial h}{\partial X} + 12 \frac{\partial h}{\partial t}
\]  

(1)

Using Holland method to solve the axis trajectory, the basic assumption is:

(1) The oil film pressure is formed by the rotary oil film pressure and the squeeze oil film pressure.

(2) Both types of pressure have their own boundary conditions, and do not affect each other.

For equation (1), it can be converted to:

\[
\frac{\partial}{\partial X} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial X} \right) + \frac{\partial}{\partial Z} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial Z} \right) = 6\omega \frac{\partial h}{\partial X}
\]

(2)

\[
\frac{\partial}{\partial X} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial X} \right) + \frac{\partial}{\partial Z} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial Z} \right) = 12 \frac{\partial h}{\partial t}
\]

(3)

2.2. Boundary conditions

In the calculation process, the oil is CD40, the working temperature is 70 °C; the radial clearance of the big end of the connecting rod is 0.20mm, the radial clearance of the main bearing is 0.52mm, the main parameters of the diesel engine are shown in Table 1. The pressure change in the cylinder is shown in Figure 1.

![Cylinder pressure](image)

Figure 1. Cylinder pressure
3. Results and Analysis
Figure 2 shows the variation of the crank pin load with the crank angle. The maximum load of the crank pin is 670095N. Figure 3 shows the load of the different main bearing with the crank angle, respectively. The maximum load of 3\textsuperscript{rd} main bearing is equal to 5\textsuperscript{th} main bearing, and both them highest. And the relationship between A and B is the same. The load changes of the seven main journals are symmetrical about the 4\textsuperscript{th} bearing, but the difference between the two is 360\textdegree angle.

Table 1. The Main Parameter of Diesel Engine

| piston stroke  | 380mm | Connecting rod mass  | 104.5kg |
|----------------|-------|----------------------|---------|
| firing order   | 1-5-3-6-2-4 | big end bearing width | 120mm   |
| piston diameter | 300mm | main journal diameter | 275mm   |
| piston mass    | 70.45kg | Piston pin diameter  | 120mm   |
| piston pin diameter | 220mm | main bearing width   | 115mm   |

Table 2. The Results of Maximum Load $F_{\text{max}}$

| $F_{\text{max}}$(N) | $F_{\text{max}}$(N) |
|---------------------|---------------------|
| 1\textsuperscript{st} MB | 2.74×10\textsuperscript{5} | 5\textsuperscript{th} MB | 8.66×10\textsuperscript{5} |
| 2\textsuperscript{nd} MB | 5.49×10\textsuperscript{5} | 6\textsuperscript{th} MB | 5.49×10\textsuperscript{5} |
| 3\textsuperscript{rd} MB | 8.66×10\textsuperscript{5} | 7\textsuperscript{th} MB | 2.74×10\textsuperscript{5} |
| 4\textsuperscript{th} MB | 5.95×10\textsuperscript{5} | | |

Table 3. $H_{\text{min}}$ and $P_{\text{max}}$ under Different Radial Clearance Conditions

| C=0.4mm | C=0.52mm | C=0.65mm |
|---------|---------|---------|
| $h_{\text{min}}$/\textmu m | $P_{\text{max}}$/MPa | $h_{\text{min}}$/\textmu m | $P_{\text{max}}$/MPa | $h_{\text{min}}$/\textmu m | $P_{\text{max}}$/MPa |
| 1\textsuperscript{st} MB | 5.52 | 73.4 | 5.26 | 94.3 | 4.93 | 110.6 |
| 2\textsuperscript{nd} MB | 3.79 | 130.7 | 3.04 | 170.1 | 2.40 | 211.4 |
| 3\textsuperscript{rd} MB | 2.95 | 241.6 | 2.59 | 281.0 | 2.15 | 337.2 |
| 4\textsuperscript{th} MB | 3.17 | 154.3 | 2.15 | 223.7 | 2.12 | 242.1 |
| 5\textsuperscript{th} MB | 2.94 | 242.3 | 2.60 | 282.3 | 2.15 | 337.1 |
| 6\textsuperscript{th} MB | 3.79 | 130.5 | 3.03 | 170.5 | 2.40 | 210.2 |
| 7\textsuperscript{th} MB | 5.72 | 73.4 | 5.26 | 92.6 | 4.92 | 110.6 |
Table 2 shows that the maximum load ($F_{\text{max}}$) of each main bearing, and 595056N in the 4th main bearing (MB). It can be seen that the maximum load is symmetrically distributed about the 4th main bearing in Table 2.

Table 3 shows that the minimum oil film thickness ($h_{\text{min}}$) is decreasing with the increase of bearing clearance ($C$), while the maximum oil film pressure ($p_{\text{max}}$) is just opposite.

The maximum oil film pressure of 3rd main bearing is 337.2MPa when Clearance between bush sleeves is 0.65mm, while 5th main bearing is 337.1MPa. The maximum oil film pressure of 3rd main bearing is 241.6MPa when Clearance between bush sleeves is 0.4mm, while 5th main bearing is 242.3MPa. The maximum oil film pressure of 3rd main bearing is 281.0MPa when Clearance between bush sleeves is 0.52mm, while 5th main bearing is 282.3MPa. Considering the error caused by the calculation, so it can be seen that the maximum oil film pressure is symmetrically distributed about the 4th main bearing in Table 3.

The minimum oil film thickness of 3rd main bearing is 2.15μm when Clearance between bush sleeves is 0.65mm, while 5th main bearing is also 2.15μm. The minimum oil film thickness of 3rd main bearing is 2.95μm when Clearance between bush sleeves is 0.4mm, while 5th main bearing is 2.94μm. The minimum oil film thickness of 3rd main bearing is 2.59μm when Clearance between bush sleeves is 0.52mm, while 5th main bearing is 2.60μm. Considering the error caused by the calculation, so it can be seen that the minimum oil film thickness is symmetrically distributed about the 4th main bearing in Table 3.

Therefore, based on the above analysis, it can be considered that the mechanical properties of the bearing are symmetrical about the 4th bearing.

4. Conclusion

In this paper, the simulation model of the lubrication characteristics of the diesel engine main bearing is established. The simulation analysis was carried out to obtain the minimum oil film thickness and the maximum oil film pressure of the main bearing. The major conclusions are summarized as follows:

1) The load changes of the seven main journals are symmetrical about the 4th bearing, but the difference between the two is 360° angle.

2) The minimum oil film thickness is decreasing with the increase of bearing clearance, while the maximum oil film pressure is just opposite.

3) The maximum oil film pressure and minimum oil film thickness of the main bearing in the diesel engine are present in the 3rd and 5th main bearings, but not the 4th main bearing.

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