Lubrication characteristics study of main bearings on crankshaft balance rate

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Abstract. Unbalanced inertia force and moment of crankshaft generated by cylinder pressure have a significant effect on the vibration characteristics of diesel engine, but the study on the lubrication characteristics of crankshaft by balance rate needs to be studied. Based on Greenwood-Tripp asperity contact theory and average Reynolds equation, an elastohydrodynamic (EHD) lubrication model of the main bearing for a V-12 diesel engine is developed, and the influence of different balance rates on lubrication characteristics is analyzed. On this basis, a factorial experiment of ten influencing factors including the balance rate is designed, and the factors that significantly affected the lubrication characteristics of the main bearing are screened out by factor effect diagram, statistical analysis and residual probability diagram. The results show that the crankshaft with different balance rate has a great influence on the lubrication performance. With the increase of balance rate, the minimum oil film thickness first increases and then decreases, and the maximum oil film pressure first decreases and then increases. The balance rate, bearing width, bearing clearance, bush repair and oil viscosity have significant effects on the performance of main bearing, among which the balance rate of crankshaft has the greatest effect on the minimum oil film thickness, the maximum rough contact pressure and the average friction power loss.

1. Introduction

The future development direction of diesel engine is higher power, higher speed, smaller size, low energy consumption and environmentally friendly as possible[1]. As one of the key parts of the diesel engine, the crankshaft-bearing system bears great unbalanced inertia force and moment, so the friction loss caused by it is very significant. The counterweights are placed on the opposite side of each crank, and the centrifugal inertia force is generated to balance the crankshaft rotating inertia forces and internal bending moment. Therefore, it is necessary to study the effect of crankshaft balance rate on lubrication problem of main bearings, especially in the stage of intensified design of high-power diesel engine.

Stanley and Taraza[2] studied the maximum and average main bearing loads of two symmetric in-line engines using a rigid crankshaft model, and estimated the present of ideal counterweight mass that will result in the acceptable maximum bearing load. Yasin and Gunay[3] studied the effect of counterweight mass and position on main bearing load and crankshaft bending stress of an in-line six-cylinder engine using rigid, beam, and 3D solid crankshaft model, without considering the effect of balance rate on the lubrication characteristics of main bearing. To date, most of the previous works[4-]
[10] focuses on the analysis of influencing factors of main bearing lubrication, such as bearing geometry, surface roughness effect, lubricant properties. The above analysis does not consider the effect of crankshaft balance rate on lubrication problem of main bearing, nor does it consider the factor effect of the crankshaft balance rate, bearing width, bearing clearance and other parameters on all responses.

Herein, counterweight positions and masses of a V-12 diesel engine crankshaft system are studied. Then, we establish an elastohydrodynamic (EHD) lubrication model of the main bearing which includes the surface roughness and elastic deformation of bearing and shaft, the effect of balance rate on lubrication performance of main bearing is studied. The fractional factorial design of resolution IV is used to identify the most influential factors on impacting responses, which including minimum oil film thickness, maximum oil film pressure, peak asperity contact pressure, and friction power loss. The analyses can provide the reference for the intensified design of 12-cylinder high-power diesel engine.

2. Preliminary

2.1. Balance rate
The balancing rate for a given crankshaft can be written as

$$ k = \frac{2 \cdot U_{r,w,i}}{(U_{q,i} + m_{b,i} \cdot r_t) \cdot \cos \theta_i} $$

(1)

where $k$ is the balancing rate of the internal couple, $U_{r,w,i}$ is the static unbalance of each counterweight, $m_{b,i}$ is about 2/3 of the mass of the connecting rod, $U_{q,i}$ is the static unbalance of each crank throw, $r_t$ is crank radius, $\theta_i$ is the offset angle of counterweight mass.

2.2. Reynolds equation
Figure 1 shows the coordinate system and the parameters of the main bearing model. Assuming that the lubricant is a laminar Newtonian fluid and is incompressible, the pressure in the direction of the film thickness is constant. The modified Reynolds equation is employed with the surface roughness and the oil fill factor

$$ \frac{\partial}{\partial x} \left( \bar{\theta} \cdot \phi_x \cdot \frac{h^3}{12 \eta} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \bar{\theta} \cdot \phi_y \cdot \frac{h^3}{12 \eta} \frac{\partial p}{\partial y} \right) = \frac{u_1 + u_2}{2} \frac{\partial (\bar{\theta} \cdot \bar{h}_r)}{\partial x} + \frac{u_1 - u_2}{2} \frac{\partial (\bar{\theta} \cdot \sigma \cdot \phi_r)}{\partial x} + \frac{\partial (\bar{\theta} \cdot \bar{h}_r)}{\partial t} $$

(2)

where $h$ is the nominal film thickness. $\bar{h}_r$ is the average film thickness between both rough surface. $p$ is the oil film pressure. $u_1$ and $u_2$ are the velocity in the circumferential direction of the journal and the bearing ($u_2 = 0$). $\sigma = \sigma_x^2 + \sigma_y^2$ is the composite rms roughness, $\eta$ is the lubricant viscosity. $\phi_x$ and $\phi_y$ are the pressure flow factors of the $x$ and $y$ direction, respectively. $\phi_r$ is the shear flow factor. $\bar{\theta}$ is the filling factor, $erf \left( \cdots \right)$ is the error function. $e$ is the eccentric ratio. $c$ is the radial clearance, and $t$ is time.

Figure 1. Model and coordinate of main bearing
2.3. Film thickness equation
The film thickness can be written as

\[ d = h + h_e + h_j + h_d \]  

(3)

where \( h_d \) is the local film thickness, \( h \) is the nominal film thickness, \( h_e \) and \( h_j \) are the elastic deformation of the journal and bearing due to the asperity pressure and oil film pressure. Because the hardness of the journal material is obviously higher than that of the bearing, here \( h_j \) is ignored \[12\]. \( h_d \) is the oil film thickness caused by roughness.

2.4. Friction power loss
When the friction pairs are under the partial lubrication regime, the friction force can be attributed to the viscous shear of the fluid and the asperity contact of the surfaces. The friction power loss can be expressed as follows

\[ P_{\text{frc total}} = P_{\text{frc oil}} + P_{\text{frc asp}} \]  

(4)

\[ P_{\text{frc asp}} = \int \int_S \left( \tau_0 + \mu_0 \cdot p \right) dx dy \cdot \omega R \]  

(5)

\[ P_{\text{frc oil}} = \int \int_S \left[ \eta (u_i - u_j) (\phi_i + \phi_j) + \phi_i h \frac{\partial p}{\partial x} + \left( \frac{\sigma}{\sigma} \right)^2 \left( \phi_i h - h_i \right) \frac{\partial p}{\partial x} - 2\eta (u_i - u_j) \phi_i \right] dx dy \cdot \omega R \]  

(6)

where \( \phi_i, \phi_j \) and \( \phi_d \) are the shear flow factors. These three terms can be obtained based on the work of Patir and Cheng[11]. \( \mu_0 \) is the boundary friction coefficient. \( \tau_0 \) is the Eyring shear stress of the lubricant[13]. \( \omega \) is angular velocity of the journal. \( R \) is the journal radius.

3. Model of crankshaft system
The studied engine is a V-12 diesel one, with a peak power at 2600 rpm. The engine general specifications are given in table 1.

| Parameters                        | Value       |
|-----------------------------------|-------------|
| Firing order                     | 1-12-9-4-5-8-11-2-3-10-7-6 |
| Balance rate (%)                 | 70          |
| Bearing width (mm)               | 44          |
| Bearing diameter (mm)            | 140         |
| Radial clearance (μm)            | 56          |
| Surface roughness of shell / journal (μm) | 0.4 / 0.8 |
| Crank radius (mm)                | 80          |
| Flywheel mass (kg)               | 86.69       |
| Flywheel moment of inertia (kg•m²) | 2.01       |

The schematic diagram of twelve-counterweight arrangement of the crankshaft is shown in figure 2. The properties of the counterweight at different balance rates are shown in table 2.
4. Results and discussion

4.1. Lubrication characteristics analysis

In general, the counterweights are designed for balance rates between 50% and 100% [3]. Therefore, the lubrication characteristics of the main bearing #7 are analyzed for different balance rates between 50% and 100% (as shown in figure 3-5).

| Balance rate | 50% | 70% | 90% |
|--------------|-----|-----|-----|
| Counterweight mass (kg) | 3.72 | 4.96 | 6.039 |
| Center of gravity position from crank rotation axis (mm) | 91.001 | 96.18 | 100.837 |
| Static unbalance of counterweight (kg mm) | 338.523 | 477.053 | 608.955 |

(a) Oil film thickness over an engine cycle

(b) Minimum and average oil film thickness

Figure 3. Effect of different balance rates on oil film thickness of main bearing #7

(a) Oil film pressure over an engine cycle

(b) Minimum and average oil film pressure

Figure 4. Effect of different balance rates on oil film pressure of main bearing #7

(a) Total friction power loss over an engine cycle

(b) Average friction power loss

Figure 5. Effect of different balance rates on friction power loss of main bearing #7
Figure 3(a)-5(a) shows that the variation of oil film thickness, oil film pressure, friction power loss for different balance rates. It can be seen that the oil film thickness increases with increasing balance rate (as shown in figure 3(a)), whereas the oil film thickness and friction power loss decrease with increasing balance rate (as shown in figure 4(a) and figure 5(a)) in most of the time in an engine cycle. In figure 4(a), as the balance rate increased, the oil film pressure spike amplitude increased, such as position A and position B. Because the pressure peak at position A and position B happens near the explosion pressure position of the cylinders 12 and 11, respectively. The reciprocating inertia force and the rotating inertia force component in the cylinder direction reach the maximum at the position of expositive pressure. Due to the direction of the pressure and inertia forces are opposite, the maximum bearing load increases with increasing balance rate, the increase of the oil film pressure spike amplitude.

Figure 3(b)-5(b) shows that the variation of minimum and average oil film thickness, maximum and average oil film pressure, average friction power loss with different balance rates. With the increase of the balance rate, the minimum oil film thickness increases, the 80% balance rate reaches the maximum, and then decreases rapidly (as shown in figure 3(b)), while the maximum oil film pressure has an opposite trend (as shown in figure 4(b)). This is because the lower balance rate will not be large enough to overcome the bending deformation of the crankshaft under explosion the pressure and inertia force. The mass and impact force of crankshaft increase when balance rate is large, which makes oil film thickness decrease.

The average film thickness increases significantly with increasing balance rate (as shown in figure 3(b)), whereas average film pressure and average friction power loss decreases with increasing balance rate (as shown in figure 4(b) and figure 5(b)). This is due to the cylinder pressure is relative less when not at piston top dead center, and the main bearing load is mainly affected by inertia force. With the balance rate increases, the inertia force that is overcome increases, hence the average bearing load decrease and the average oil film thickness increases.

4.2. Factorial experiment analysis

Factorial experiment design is important as a formal way of maximizing information gained while minimizing required resources, and explained and fully detailed in Ref.[14]. In this paper, considering the main effects and two-factor interactions, the design is utilized for a $2^{10-4}$ design of resolution IV, which would require 32 runs. It means that a total of $n = 10$ factors are studied. The ten factors and their range are shown in table 3. The bearing responses are Minimum oil film thickness (MOFT), Maximum oil film pressure (MOFP), Peak asperity contact pressure (PACP), Average friction power loss (AFPL). The results of 32 runs are shown in table 4.

| Factor                  | Abbreviation | Unit | Low level (−) | High level (+) |
|-------------------------|--------------|------|---------------|----------------|
| Balance rate            | BAR          | %    | 50            | 90             |
| Bearing width           | BEW          | mm   | 44            | 48             |
| Radial clearance        | RAC          | μm   | 40            | 80             |
| Bearing crown height    | BEC          | μm   | 0             | 5              |
| Oil viscosity           | VIS          | Pa·s | 7             | 12             |
| Oil supply pressure     | OSP          | MPa  | 4             | 10             |
| Bore diameter           | BOD          | mm   | 3             | 7              |
| Oil groove width        | OGW          | mm   | 3             | 7              |
| Roughness of journal    | ROJ          | μm   | 0.4           | 0.8            |
| Roughness of bearing    | ROB          | μm   | 0.4           | 0.8            |
Table 4. Results of the runs

| Run | A | B | C | D | E | F | G | H | J | K | MOFT (µm) | MOFP (MPa) | PACP (MPa) | AFPL (W) |
|-----|---|---|---|---|---|---|---|---|---|---|-----------|------------|------------|----------|
| 1   | + | + | + | + | - | - | - | - | - | - | 1.361     | 211        | 60         | 3555     |
| 2   | + | - | - | + | + | + | - | - | - | + | 1.002     | 194        | 128        | 5465     |
| 3   | + | + | - | - | - | + | + | + | - | - | 1.048     | 188        | 117        | 4288     |
| 4   | - | - | - | - | + | - | - | - | + | + | 0.831     | 170        | 179        | 5516     |
| 5   | - | - | - | - | - | - | - | - | + | + | 0.775     | 197        | 200        | 5751     |
| 6   | - | + | + | - | + | + | + | + | - | - | 1.11      | 154        | 102        | 7092     |
| 7   | - | - | - | - | - | - | - | - | + | + | 0.76      | 236        | 205        | 4315     |
| 8   | + | + | - | - | - | + | + | + | + | - | 1.044     | 154        | 117        | 5740     |
| 9   | + | - | - | + | - | + | - | + | + | + | 1.067     | 203        | 112        | 4087     |
| 10  | - | + | - | - | - | + | + | + | + | - | 0.865     | 225        | 168        | 4659     |
| 11  | + | + | - | + | + | - | - | - | + | + | 1.358     | 157        | 60         | 5131     |
| 12  | - | - | - | - | - | + | + | + | + | - | 0.817     | 274        | 184        | 6192     |
| 13  | - | - | - | - | - | - | - | - | - | + | 0.782     | 247        | 197        | 5226     |
| 14  | + | - | - | - | - | - | - | - | - | - | 0.963     | 230        | 138        | 4243     |
| 15  | - | + | - | - | - | + | + | - | + | + | 0.953     | 192        | 141        | 4526     |
| 16  | + | - | + | - | + | - | + | + | - | + | 0.944     | 237        | 144        | 4494     |
| 17  | - | - | - | - | + | - | - | - | - | - | 0.836     | 223        | 178        | 5585     |
| 18  | + | - | + | - | - | + | + | + | - | + | 1.006     | 193        | 127        | 4080     |
| 19  | - | + | + | - | + | - | + | + | - | + | 0.921     | 170        | 150        | 5677     |
| 20  | + | - | + | - | - | - | - | + | - | + | 1.318     | 195        | 65         | 3882     |
| 21  | - | - | - | - | - | + | - | + | + | + | 0.815     | 254        | 185        | 6979     |
| 22  | - | - | + | + | + | + | - | - | - | + | 0.998     | 148        | 129        | 5524     |
| 23  | + | - | - | - | - | - | - | + | + | + | 1.000     | 237        | 129        | 5649     |
| 24  | + | - | + | - | + | - | + | + | + | + | 1.163     | 234        | 92         | 3219     |
| 25  | + | + | - | - | - | + | + | - | + | + | 0.895     | 243        | 158        | 3232     |
| 26  | + | + | - | + | - | + | + | + | + | + | 1.467     | 187        | 47         | 4418     |
| 27  | - | + | - | - | - | + | + | + | + | - | 0.877     | 170        | 164        | 7402     |
| 28  | - | - | + | + | - | - | - | - | + | + | 0.983     | 199        | 133        | 4054     |
| 29  | + | + | + | - | + | + | + | + | + | + | 1.009     | 194        | 126        | 4070     |
| 30  | - | - | - | - | + | - | + | + | - | + | 0.845     | 245        | 174        | 7693     |
| 31  | - | + | + | - | - | + | + | - | + | + | 1.042     | 150        | 118        | 5125     |
| 32  | + | - | + | - | - | - | - | - | - | - | 0.878     | 256        | 163        | 3706     |

The main purpose of the current work is to determine the factors that have great effect on the performance of main bearing. The factor effects of all responses with a confidence interval of 95% are shown in figure 6-9. It can be concluded that five of the ten factors mentioned have great effect on all response: balance rate, bearing width. The radial clearance, The bearing crown height, The oil viscosity. These five factors are used to establish a regression model of reduction.
In order to verify the correctness of the reduction model, statistical verification of the model is required. The model statistics before and after reduction are shown in Table 5.

Table 5. Fit statistics for two models

|                | Full model |                   | reduced model |                   |
|----------------|------------|-------------------|---------------|-------------------|
| $R^2$          | 0.758      | 0.661             | 0.789         | 0.961             | 0.735             | 0.623             | 0.763             | 0.944             |
| $R^2_{adj}$    | 0.642      | 0.499             | 0.689         | 0.942             | 0.685             | 0.550             | 0.717             | 0.933             |
| $R^2_{pred}$   | 0.437      | 0.213             | 0.510         | 0.908             | 0.600             | 0.429             | 0.640             | 0.915             |

Table 5 shows that the reduced model $R^2$ is smaller than the full model. However, that the adjusted $R^2_{adj}$ for the reduced model is larger than the adjusted $R^2_{adj}$ for the full model except for average friction power loss (AFPL). Meanwhile, $R^2_{pred}$ has improved significantly. Obviously, the final model is obtained by removing the non-significant terms from the full model, which is more effective in predicting new data. The normal probability plot of the residuals for the reduced model containing five factors that have great effect on all responses are shown in figure 10-12.
The residuals are distributed near the normal line and between -4 and 4 studentized residuals, which means that the residuals are normally distributed and have no outliers[15]. It can be seen from figure 10-12 that the points on the residuals plots from all responses lies reasonably close to a normal line, lending support to above conclusion that the balance rate, bearing width, radial clearance, bearing crown height, and oil viscosity are significant effects on all responses.

5.Conclusions
Based on elastohydrodynamic lubrication and balance theory, the effect of different balance rates on lubrication characteristics of main bearing #7 are analyzed. Considering the unbalanced inertia force and moment of the crankshaft, it can be seen that the crankshaft with different balance rates has a greater influence on the lubrication performance. With the increase of the balance rate, the minimum oil film thickness increases first and then decreases, and the maximum oil film pressure decreases first and then increases. On the other hand, the average film thickness increases significantly with increasing balance rate, whereas average film pressure and average friction power loss decreases with increasing balance rate due to lower inertia force.

The 2^{10-5} fractional factorial design of resolution IV is utilized to calculate the factor effects of all responses. It can be observed that the most influential parameters among ten factors can be determined by the experimental design with only 32 runs, and these parameters include: balance rate, bearing width, radial clearance, bearing crown height, oil viscosity. Among them, the balance rate of the crankshaft has the greatest effect on the minimum oil film thickness, peak asperity contact pressure and average friction power loss. The reduced model is constructed using the most influential parameters and compared to the predictive power of the full model. It can be concluded that the predictive power of the reduced model is more effective.

The paper presents a novel analytical method for dynamic lubrication of bearings. When many factors are likely to be studied, the primary factors can be screened out with the smallest number of runs.
All the analysis can provide useful guidance for the efficiency to design the bearing.

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