Effect of coupling weight of marine diesel engine on crankshaft vibration and lubrication

Limin Wu1,2, Mei Li2, Xinmin Yang3,3, Liting Li2, Gang Liang2 and Xinqi Qiao2

1Shanghai Jiaotong University, Shanghai 200240, China
2Shanghai Marine Diesel Engine Research Institute, Shanghai 201108, China
3E-mail: 18800292785@163.com

Abstract. Journal misalignment caused by coupling weight result in rotary unbalance. To study the unbalance effect, based on AVL EXCITE PU software, coupled multi-body dynamics and lubrication model of a marine diesel engine had been built, and validated by engine experimental data. Based on the simulation model, vibration and bearing lubrication of crankshaft with different rear-end configuration had been discussed. The results show that the unbalance caused by rear end coupling weight of crankshaft have effect on the 1st and 2nd main bearing near flywheel, and have little effect on other main bearings. It also changes the pressure distribution of upper and lower shell, and brings edge thermal load increase. Meanwhile, it has more effect on the heat transfer to shell than to journal.

1. Introduction

With the development of light weight design and the high-power output of diesel engine, the crankshaft is designed by a smaller strength safety margin to adapt the demand. Journal nearby flywheel is more liable to bend if heavier flywheel and coupling is assembled. Unbalanced force and moment caused by bending journal result in the worse bearing lubrication, vibration and noise[1].

Lambar [3] analyzed that journal misalignment with very small angle can result in the thinner oil film on bearing border, causing border contact and collision. Sun Jun et al. [4] verified the journal misalignment had remarkable influence on bearing frictional and wearing performance. Bou-Said [5] and San Andres [6] analyzed characteristic of dynamic and static pressure considering journal misalignment. Han Qingkai et al. [7] analyzed the dynamic characteristics of oil-film under the conditions of mass unbalance based on a rotor-bearing numeral model, which provided reference for fault diagnosis of rotor-bearing system. Bin Guangfu et al. [8] analyzed the influence of the rotor residual unbalance on the vibration characteristics of the shaft system with three supports. Cui Yahui et al. [9] studied the influence law of stream turbine shaft vibration response with different unbalanced weight and positions by simulation method. Xia Yebao et al. [10] considered a detailed cantilever branch structure on which turbine disks of aero-engine were mounted instead of simplified mass in the simulation model. By this method, more accurate model of unbalanced response had been obtained.

In summary, the scholars had done much work on the influence law of unbalanced shaft on vibration and lubrication respectively in rotary machine field. However, there is little research on crankshaft unbalance of diesel engine except counterweight balance, especially the high elastic coupling mounted on the flange of flywheel, which has an effect on the crankshaft and main bearing performance.
In this paper, the coupled multi-body dynamics and lubrication model is used to simulate the unbalanced force and moment caused by the bending journal subjected to the rear-end gravity. The rationality of the simulation model had been verified by comparing with experimental results. Based on the model, vibration and bearing lubrication of crankshaft with different rear-end configuration had been discussed in detail.

2. Basic theory
Considering the elastic deformation of main journal and shell, the dynamic equation [11] can be written as:

\[
[M] \ddot{x}_B + [C] \dot{x}_B + [K] \ddot{x}_B = f(t)_{oil} \tag{1}
\]

\[
[m] \ddot{x} = f(t)_{oil} + f(t)_{ext} \tag{2}
\]

Where: \([M]\) \([K]\) \([C]\) is the mass matrix, stiffness matrix, damping matrix; \(x_B\) is the elastic displacement, \(t\) is time, \(x\) is the elastic displacement of journal, \(f(t)_{oil}\) is the force produced by oil film pressure, \(f(t)_{ext}\) is the main bearing load act on journal.

Gravity of rear-end act on the shaft bends bearing’s journal along the vertical direction. Figure 1 shows journal misalignment caused by gravity of rear weight. Taken the journal center as reference coordinate system, the equation [11] of the unbalanced force can be written as:

\[
F_y = me \omega^2 \cos \omega t \tag{3}
\]

\[
F_z = me \omega^2 \sin \omega t \tag{4}
\]

Figure 1. Journal misalignment.

Where \(F_y\) is the projected unbalanced force on Y axis, \(F_z\) the projected unbalanced force on Z axis; \(e\) is eccentricity; \(m\) is eccentric mass, \(\omega\) is rotational speed.

Bearing lubrication is analysed by Reynold equation [12], as shown in Equation (5). The left end of the equation indicates the changes in the lubricating film pressure on the lubricated surface with the coordinates \(x\) and \(y\), and the right end indicates various effects that produce the lubricating film, such as dynamic pressure effect, stretching effect.

\[
\frac{\partial}{\partial x} \left( \frac{\rho h^3}{\eta} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{\rho h^3}{\eta} \frac{\partial p}{\partial y} \right) = 6(U_0 + U_h) \frac{\partial (\rho h)}{\partial x} + 6(V_0 + V_h) \frac{\partial (\rho h)}{\partial y} + 6\rho h \frac{\partial (U_0 + U_h)}{\partial x} \\
+ 6\rho h \frac{\partial (V_0 + V_h)}{\partial y} + 12 \frac{\partial (\rho h)}{\partial t} - 12\rho (U_h \frac{\partial h}{\partial x} + V_h \frac{\partial h}{\partial y}) \tag{5}
\]

Where: \(p\) is pressure; \(\eta\) is viscosity, \(h\) is clearance height, \(U\) and \(V\) is axial and circumferential speed respectively, \(t\) is time.

Oil film thickness is analyzed by Equation (6) with elastic-hydrodynamic model (EHD) [13].

\[
h(\theta) = h_{min}(\theta) + \Delta h(\theta) + \delta h(\theta) + \sigma h(\theta) \tag{6}
\]
In the formula: $\Delta h(\theta)$ represents the difference between the thickness of the oil film at each point of the journal and $h_{\text{min}}(\theta)$; $\delta h(\theta)$ represents the thickness of the oil film due to roughness, $\sigma h(\theta)$ is the elastic displacement function related to the pressure distribution.

The rough surface contact model can be calculated by the following formula [11], proposed by Greenwood and Tripp (this model assumes that the surface height is Gaussian and the surface topography is isotropic):

$$
\rho_c(h) = \left(\frac{16\sqrt{2}}{15}\right)\pi(\eta\chi\sigma)^2E\sqrt{\frac{\sigma}{\chi}}F_s\left(\frac{h}{\sigma}\right)
$$

(7)

The rough peak contact area is:

$$
A_p(h) = \pi^2(\eta\chi\sigma)^2F_s\left(\frac{h}{\sigma}\right)
$$

(8)

Where: $\eta$ is the peak density of rough surface, $\chi$ is the rough radius of curvature, $E^*$ is the comprehensive elastic modulus,

$$
F_n = \int_{h/\sigma}^{\infty} (s-h/\sigma)\varphi^*(s)ds \quad n = 5/2, 2
$$

(9)

Where: $\varphi^*(s)$ is the probability density function of the height distribution of the micro-convex body, based on the assumption that the surface roughness height distribution is Gaussian Distribution, the above formula can be solved by (10) and (11).

$$
F_{s1}\left(\frac{h}{\sigma}\right) = \begin{cases} 
2.134\times10^{-4}\left\{3.804\ln\left(4-\frac{h}{\sigma}\right)+1.34[\ln(4-\frac{h}{\sigma})]^3\right\} & 0 < \frac{h}{\sigma} \\
1.705\times10^{-4}\left\{4.504\ln\left(4-\frac{h}{\sigma}\right)+1.37[\ln(4-\frac{h}{\sigma})]^3\right\} & \frac{h}{\sigma} \leq 3.5
\end{cases}
$$

(10)

$$
F_{c}\left(\frac{h}{\sigma}\right) = \begin{cases} 
8.8123\times10^{-7}\left(4-\frac{h}{\sigma}\right)^{1.15} & 0 < \frac{h}{\sigma} \\
1.12\times10^{-4}(4-\frac{h}{\sigma})^{-1.247} & 3.5 < \frac{h}{\sigma} \leq 4
\end{cases}
$$

(11)

Figure 2. Multibody dynamics model.
3. Multi-body dynamic model of diesel engine
Based on a coupled multi-body dynamics and elastic-hydrodynamic, the model is showed in Figure 2. Crankshaft and engine, shown in Figure 3, were built by using finite element model. The crankshaft center had been taken as global coordinate system. The gravity of high-elastic coupling was simulated by the local reference coordinate system. The equivalent model of high elastic coupling model had been built according to the actual mass and stiffness distribution.

![Figure 3. Crankshaft model and flexible-body model.](image)

The main bearings model had been built by finite difference method, which is calculated by modified Reynolds equation. The surface roughness and elastic modulus of shell and journal had been considered. Oil supply boundary is shown in Figure 4. The 0° position of the shell is defined as the upper shell center.

![Figure 4. Main bearing oil supply boundary.](image)

4. Dynamics simulation model validation
The 16V diesel engine bench had been equipped with a vibration test system and a data acquisition system. As showed in Figure 5, two measuring points at both ends of crankshaft were used to measure torsional vibration, and another measuring point was used to measure transverse vibration. Eddy current sensors had been used to measure the relative shaft vibration.
The simulated and measured natural frequencies of the crankshaft are shown in Table 1. The maximum calculation error is 5%. The simulation model of crankshaft stiffness and mass distribution is reliable enough for dynamic response calculation.

### Table 1. Calculation data of natural frequency of crankshaft.

| Test /Hz | 52.1 | 52.4 | 83.9 | 93.9 | 134.9 | 156.1 |
|----------|------|------|------|------|-------|-------|
| Calculation /Hz | 53.8 | 53.7 | 82.3 | 88.1 | 134.2 | 156.3 |
| Error /% | 3 | 3 | 1.2 | 5 | 0.6 | 0.06 |

The crankshaft vibration comparison in frequency domain between test and simulation results have shown in Figure 6. The main resonance frequency of torsion vibration by test is 9Hz, compared with 9.1Hz obtained by simulation method. It is the half of baseband frequency of crankshaft speed. The torsional amplitudes of the test and simulation are 0.25° and 0.22° respectively at half of baseband frequency, and the simulation accuracy is 88.2%. It can be seen that the half baseband frequency has an effect on torsional vibration.

![Figure 6. Torsion vibration in frequency domain.](image)

(a) Test  
(b) Simulation

The comparison of vibration in time domain between test and simulation are shown in Figure 7 and Figure 8. The periodicity of the results is consistent. Although the phase and amplitude have deviation, the calculation accuracy of vibration amplitude is 88% and acceptable. The simulation model ignored the thin cover, valve excited force and the burning difference of cylinders etc., which affect crankshaft vibration directly.
5. Rear-end coupling weight optimization

5.1. Basic parameters
The two kinds of rear-end coupling configuration parameters are shown in Table 2. Unbalanced mass of crankshaft system had been measured respectively.

| Configuration | Value | unit |
|---------------|-------|------|
| A             | Coupling and flywheel weight | 1.7 | t |
|               | Coupling Stiffness           | 0.67 | MN/rad |
|               | Coupling Inertia             | 142 | kgm² |
|               | Unbalanced mass of crankshaft with coupling A | 2 | kgm |
| B             | Coupling and flywheel weight | 1.3 | t |
|               | Coupling Stiffness           | 1  | MNm/rad |
|               | Coupling Inertia             | 110 | kgm² |
|               | Unbalanced mass of crankshaft with coupling B | 1.1 | kgm |
The natural frequency of crankshaft system with configuration A and B had been calculated. The natural frequency difference of the two-shaft system is shown in the Table 3. The torsion natural frequency of the two-crankshaft system has little difference, and the maximum natural frequency difference is 2%.

|                         | Crankshaft with configuration A /Hz | Crankshaft with configuration B /Hz | Discrepancy /% |
|-------------------------|------------------------------------|------------------------------------|----------------|
| Transverse 1\textsuperscript{st} order | 12.82                              | 12.57                              | 1.9            |
| Transverse 2\textsuperscript{nd} order | 15.02                              | 14.72                              | 2.0            |
| Transverse 2\textsuperscript{nd} order | 36.96                              | 36.75                              | 0.6            |
| Torsional vibration 1\textsuperscript{st} order | 39.7                               | 39.67                              | 0.1            |

5.2. Transverse and torsional vibration

The effects of different configurations on the dynamic response of the crankshaft system and the working characteristics of the main bearing oil film were studied. As shown in Figure 9, after rear end weight is reduced, the combined force in the horizontal direction of the first main bearing at the flywheel end is reduced by 5%, and the maximum value of the combined force in the vertical direction is similar, but the maximum inertia moment is reduced by 16%.

The inertia moment affects transverse vibration mostly. Because of the less inertia moment, the transverse vibration amplitude of crankshaft with configuration B decreases, shown in Figure 10(a), 10(b). In addition, the light weight of the flywheel and the coupling may relieve eccentricity of the rear journal. It can be seen that the difference in torsional vibration amplitude is small, as shown in Figure 10(c). This is because the coupling has a uniform mass distribution in the circumferential direction.
5.3. Bearing lubrication

Comparison of the peak total pressure and asperity contact pressure of main bearings are shown in Figure 11. The rear end weight has a great impact on the first and the second main bearings, while has little impact on other main bearings. It can be seen from Figure 12, the pressure in middle area of the lower shell is increased, and the upper shell remarkably reduce, which is mainly due to the lower inertia force and moment of configuration B.

The unbalance caused by the coupling weight has a great impact on the average thermal load, which is showed in Figure 13. The unbalance of configuration A is larger than that of configuration B by 0.9 kgm. In particular, the edge thermal load of MB1 and MB2 increases obviously, but the edge thermal load of MB8 declines slightly. This is because unbalance mass brings the intensified journal eccentric rotation, while the journal of MB8 which is far away from rear end is hardly subject to the unbalance.
Figure 12. Peak total pressure distribution.

Figure 13. Average thermal load.

Figure 14. Heat transfer of MB1.
As shown in Figure 14, compared with configuration A, the MB1 average heat transfer amount to journal of crankshaft with configuration B reduced by 2.8%, while the average heat transfer amount of the bearing shell is reduced by 15%. The rear-end configuration affect heat transfer to shell more obviously than to journal.

6. Conclusions
The unbalance caused by journal misalignment because of rear-end weight had been considered in the lubrication simulation model of crankshaft and coupling system. It provided good results which had been verified by experiment. The resonance frequencies are consistent, and the vibration amplitudes accuracy reaches 88.2%. It is reliable to evaluate an unbalanced crankshaft performance.

The rear weight should be controlled to improve vibration and lubrication. The unbalanced excited force caused by rear weight of shafting brings the increase of the transverse force of bearing and inertia moment, which intensified the transverse vibration. The unbalance caused by rear weight of shafting has effect on the 1st and 2nd main bearing near flywheel, and has little effect on other main bearings. It also changes the pressure distribution of upper and lower shell, and brings edge thermal load increase. Meanwhile, it has more effect on the heat transfer to shell than to journal.

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