Research on Vibration Characteristics of Propulsion Shafting Under Unbalanced-misaligned Coupling Fault

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Abstract. Vibration characteristics under unbalanced-misaligned coupling fault of propeller shaft were studied. Firstly, the rigid-flexible mixing model of propeller shaft system test bench is established by using Solidworks and ADAMS. Then, the shafting vibration characteristics in time and frequency domain are obtained by simulation analysis under normal, unbalanced, misaligned and unbalanced-misaligned coupling faults. Finally, the influence of rotational speed on unbalanced-misaligned coupling fault shaft system is analysed. The simulation study shows that the vibration spectrum of shafting under unbalanced-misaligned coupling fault is mainly the superposition of the spectrum under each single fault, and the vibration caused by each fault is different in different locations. The increase of speed will aggravate the vibration of shafting.

1. Introduction
At present, there are a lot of in-depth studies on unbalanced and misaligned faults both at home and abroad. The research direction can be divided into two main parts: research content and research method. In terms of research content, a large number of studies are aimed at single faults. For example: Zhenyu Wu, Jingjun Lou, Driot N, Weijie Guo, etc. have studied the changes in rotor vibration characteristics caused by different types of rotors under the action of misalignment faults [1-4]; Guangfu Bin et al. studied the influence of residual unbalance on the vibration characteristics of three-support shafting [5]; Chenghong Xing et al. studied the application of unbalanced faults in diagnosis [6]. There is also a small amount of research on coupling failures. For example: Fuze Xu studied the difference between the vibration characteristics of a simple single-disk rotor and a single-fault rotor under the coupling of unbalance-misalignment faults [7]. In terms of research methods, it can be divided into three types: theoretical research, experimental research and simulation research. Xuanchen Lin et al. derived the mathematical model of shaft whirl vibration under misalignment fault [8]; Liufang Shentu et al. used Lagrange equation of the second kind to derive the dynamic differential equation of the system in the case of angular misalignment and rotor mass imbalance [9]; Qingyun Dong et al. studied the vibration characteristics of the double-disk rotor misalignment-rubbing fault by using simulation and experiment methods [10]; Lu Jie et al. simulated the vibration of the rotor under the conditions of parallel misalignment, angle misalignment and comprehensive misalignment by simulation software ADAMS, which proved that it is feasible to study the rotor fault using simulation software [11].

Based on the above research, it can be found that the research objects of scholars are relatively simple rotor models. While the propeller part of the propeller shaft system runs in water, due to the influence of non-uniform wake flow field and other factors [12] [13], the propeller excitation force will inevitably
occur, which will make the vibration of the shaft system more complex. Therefore, based on the dynamic model of the propeller shafting, this paper also considers the vibration response laws and characteristics of the shafting under the combined action of the propeller excitation force, unbalance and misalignment.

2. Fault Mechanism of Shaft System

Propellers adhere to a large number of aquatic organisms or sporadic entanglement of fishing nets and other debris during long-term operation, resulting in serious unbalanced. The unbalanced force is an alternating force with constant size and constant direction. The direction of centrifugal force varies by one cycle per revolution of the shaft system. When the height of the bearing changes, misalignment occurs at the rigid coupling connecting the stern shaft and the thrust shaft. The axial alternating force changes once for every rotation of a rotating shaft, and the frequency is the same as that of the rotating frequency. The radial alternating force changes twice, and its frequency is twice that of the rotating frequency. But in reality, the shafting is more in the comprehensive state under the joint action of these two faults. The schematic diagram is shown in Figure 1 below:

The meaning of the symbol is as follows, $O_1$-the geometric center of the stern front bearing; $O_2$-the geometric center of the aft stern bearing; $O$'-propeller mass center; $e$-propeller mass eccentricity; $\Delta \alpha$ - misalignment drift angle; $\Delta y$ - asymmetric parallel quantity; $m_1, m_2, m_3$ They are the equivalent mass of the stern shaft at the front bearing, the aft bearing and the propeller. The stern shaft can be regarded as a massless elastic shaft. $m_{b1}, m_{b2}$ - the mass of the stern front bearing and the stern rear bearing respectively; $x_1, x_2, x_3, y_1, y_2, y_3, z_1, z_2, z_3$ - they are the transverse, vertical and axial displacements of the stern shaft at the fore stern bearing, the stern rear bearing and the propeller; $x_{b1}, x_{b2}, y_{b1}, y_{b2}$ - they are the transverse and vertical displacements of the stern front bearing and the stern rear bearing respectively; $k, c, c_b$ - they are stern shaft stiffness, damping at propeller and damping at bearing; $k_1, k_2$ - they are the oil film stiffness between the stern shaft and the front bearing of the stern shaft and the rear bearing of the stern shaft; $c_1, c_2$ - they are the oil film damping between the stern shaft and the front bearing of the stern shaft and the rear bearing of the stern shaft; $k_{b1}, k_{b2}$ - they are the elastic support stiffness between the stern shaft and the front bearing of stern shaft and the rear bearing of stern shaft and the base respectively; $c_{b1}, c_{b2}$ - they are the support damping between the front bearing of the stern shaft and the rear bearing of the stern shaft and the base respectively; $F_x, F_y$ - they are the lateral and vertical components of the misalignment force; $F_y(t)$ - exciting force of propeller; $\omega$ - angular speed of stern shaft rotation.

Figure 1. Unbalanced-misaligned coupling fault dynamics model
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\begin{align*}
\mathbf{m}_1 \ddot{x}_1 + k(x_1 - x_2) + c_\theta x_1 = F_x \\
\mathbf{m}_1 \ddot{y}_1 + k(y_1 - y_2) + c_\theta y_1 = F_y \\
\mathbf{m}_2 \ddot{x}_1 + k(x_1 - x_2) + c_\theta x_1 = 0 \\
\mathbf{m}_2 \ddot{y}_1 + k(y_1 - y_2) + c_\theta y_1 = -m_1 g + F_y \\
\mathbf{m}_3 \ddot{x}_3 + k(x_3 - x_2) + c_\theta x_3 = m_3 e \omega^2 \cos(\omega t) \\
\mathbf{m}_3 \ddot{y}_3 + k(y_3 - y_2) + c_\theta y_3 = -m_3 g + m_3 e \omega^2 \sin(\omega t)
\end{align*}
\]

3. Establishment of simulation model
With reference to the composition of a certain type of ship's propulsion shafting structure, a propeller propulsion shafting test bench model was established. As shown in Figure 2, the shafting part of the test bench is mainly composed of 1-elastic coupling, 2-intermediate bearing, 3-thrust bearing, 4-rigid coupling, 5-stern front bearing, 6-stern rear bearing, 7-propeller, 8-stern shaft, 9-thrust shaft and 10-intermediate shaft.

4. Analysis of simulation results
4.1. Transient response analysis
Figure 3 shows the radial transient acceleration responses of the measuring points on the fore stern bearing pedestal (left figure) and middle bearing pedestal (right figure) of the shafting under the conditions of normal, unbalanced fault, misalignment fault and unbalanced-misalignment coupling fault respectively. The above four cases are all carried out at a speed of 600rpm (10Hz), the simulation time is set to 10s, and the number of simulation steps is set to 4096 steps. Among them, the unbalanced amount of the unbalance fault is 10g.mm, the axis parallel displacement of the misalignment fault is 0.003mm, and the axis angular displacement is 0.001°. It can be found from Figure 3 that when the shaft is in four different conditions, its vibration response shows obvious periodic characteristics, and the vibration amplitude when there is a fault is larger than when the shaft is running normally. The vibration amplitude at the front bearing pedestal at the stern is two orders of magnitude larger than that at the middle bearing pedestal due to the relatively close distance to where the fault occurred. It can also be seen from Figure 3 that the vibration characteristics of the two bearing pedestals are quite different,
mainly caused by the different effects of the propeller's excitation force, unbalance, and misalignment at the two bearing pedestals.

Figure 3. Time domain analysis

4.2. Characteristic frequency analysis

The amplitude-frequency characteristic analysis of vibration signal is one of the most common methods for mechanical fault diagnosis. It is simple and effective. The amplitude-frequency characteristics of vibration signals under four different conditions are analyzed.

It can be seen from the frequency domain diagram in Figure 4 (a) under normal operation of the shaft system that the rotating frequency (10Hz), blade frequency (30Hz) and double blade frequency (60Hz) are displayed at both bearing pedestal, of which the amplitude of double blade frequency is the largest.

It can be seen from the frequency domain diagram of Figure 4 (b) in case of unbalanced fault, the rotating frequency, blade frequency and double blade frequency are shown at both bearing pedestal, and the amplitude of the rotating frequency is obviously increased compared with the normal situation, especially at the front bearing of stern shaft. Among them, the front bearing pedestal of the stern shaft is mainly at one time of the rotating frequency, and the middle bearing pedestal is mainly at double blade frequency.

As can be seen from the frequency domain diagram of Figure 4 (c) in the case of misalignment fault, compared with the above two conditions, two times of the rotational frequency appear at both bearing pedestal. The front bearing pedestal of the stern shaft is mainly twice the rotating frequency, and the middle bearing pedestal is still dominated by double blade frequency, but the amplitude of double rotating frequency is equivalent to that of blade frequency.

From Figure 4 (d) frequency domain diagram of unbalanced-misalignment coupling fault, it can be seen that the vibration spectrum of coupling fault is mainly the superposition of each single fault spectrum, indicating that the shafting can be regarded as a linear system. Secondly, compared with single fault, the characteristic frequency and amplitude of coupling fault are more. The front bearing pedestal of stern shaft is mainly rotating frequency and double rotating frequency, and the middle bearing
pedestal is mainly blade frequency and double blade frequency. It shows that the sensitivity of different positions on the shafting to various faults is different.

![Frequency domain analysis](image)

(a) Normal  
(b) Unbalanced fault  
(c) Misalignment fault  
(d) Unbalanced-misalignment coupling fault

**Figure 4.** Frequency domain analysis

4.3. The influence of speed on the vibration of coupling fault shaft

According to the above analysis, the vibration characteristics of unbalance-misalignment coupling faults are closely related to the fault types. In order to further study the vibration characteristics of coupled faults, it is necessary to conduct a quantitative analysis of the faults. Among them, the speed has a great influence on the vibration characteristics of the shafting, so the three-dimensional waterfall diagram method is used to study the influence of the speed on the unbalance-misalignment coupling fault shafting. The advantage of the three-dimensional waterfall chart is that you can intuitively observe the changes in the amplitude of each frequency multiplier with the speed.

It can be seen from Figure 5 that from 0rpm to 1200rpm, the vibration response frequency of the front bearing pedestal of the stern shaft is always dominated by the rotational frequency, and the amplitude of the rotational frequency is the most sensitive to the change of the rotating speed; for the middle bearing block, the amplitude of each frequency doubling is significantly smaller than that of the front bearing pedestal of the stern shaft in numerical value, and the vibration response frequency is mainly at double blade frequency, and other components appear between the speed of 600 rpm and 1200 rpm.
5. Conclusion

In this paper, firstly, the mechanism of unbalance and misalignment is analyzed, and then the rigid flexible hybrid model of propeller propulsion shafting experimental platform is established by using SolidWorks and Adams. The transient dynamic analysis of the model under unbalance, misalignment and coupling conditions and the influence factors of coupling fault shafting vibration are carried out, and the simulation results are obtained, the conclusions are as follows:

(1) The vibration spectrum of the shafting system under the action of unbalanced-misalignment coupling fault is mainly manifested as the superposition of the frequency spectrum under each single fault, and the vibration performance caused by different single faults is different from different measuring points;

(2) When a certain fault value is increased, the corresponding fault frequency amplitude caused by the fault will increase obviously; the speed has a greater impact on vibration, and the unbalanced fault is the most sensitive to the change of speed, as the speed increases, The frequency amplitude of the fault caused by this fault increases significantly faster than the frequency amplitude of other faults.
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