Study on modal dynamic balance method of bearing rotor system

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Abstract. In order to make dynamic balance for bearing rotor system with unbalanced vibration, a modal balancing method for bearing rotor system is proposed. The vibration of the rotor is decomposed into multi-order components according to vibration mode, the unbalanced components are balanced step by step, and the correction amount of each unbalance is solved by establishing a modal matrix equation. Through simulation analysis, it can be seen that under different conditions of speed and unbalance, the amplitude of the rotor after dynamic balance is greatly reduced, and the average vibration is reduced by more than 70%. This method can suppress vibration effectively for bearing rotor system and improve running performance of high speed bearing rotor system.

1. Introduction

Bearing rotor system is the core component that directly affects the operation of the high-end rotating machinery. The unbalanced force and unbalanced force couple existing on the high speed unbalance bearing rotor will produce vibration displacement. The shaft will produce deflection deformation, and even resonance will cause mechanical damage when the rotor is close to the critical speed. The natural frequency of lateral vibration is multi-order, and its corresponding critical speed is also multi-order, so there will be multi-order vibration mode unbalance.

The core of the modal balancing method lies in the ability to balance the unbalanced components of the main vibration modes of the rotor step by step. In recent years, scholars at home and abroad have also conducted a lot of research on the method of rotor modal dynamic balance control. Mendes proposed balancing method using parametric excitation is presented along with simulations to illustrate its potential, which avoided the need for spinning the rotor above its first critical speed[1]. Jia proposed a high speed flexible rotor without test weight modal dynamic balance method, combined with the measured vibration information, the size and phase of the balance weight were calculated[2]. Chen proposed an automatic balancing control algorithm based on adaptive particle swarm optimization[3]. Rafael used coordinate transformation methods to identify the ideal modal parameters that should be used in the modal balance process, and can ensure the optimal extraction position of the modal parameters by exciting the direction tracking of a single vibration mode[4]. Deepthikumar avoided the multiple test runs required for the modal balance of the flexible rotor and introduced the concept of using the eccentric polynomial function "norm" to quantify the distribution unbalance, which was verified on the experimental rotor with both bending and unbalance[5]. Battiaio proposed a novel modal interface reduction method combined with a layer of Jenkins contact elements to effectively predict the effect of friction on the nonlinear forced response of the stator blade disk[6]. Max proposed the Mode Shape Weighted Optimization Method for evaluating residual modal
unbalances. This method allows considering all vibration amplitudes for all bearings and all resonance and service speeds at the same time[7].

The modal balancing method can effectively suppress the vibration of different types of rotor systems and effectively balance the flexible rotor systems. How to further improve the dynamic balance vibration suppression level of bearing rotor system is an important problem to be solved. This paper proposes a modal dynamic balancing method for bearing rotor system to effectively suppress multi-order unbalanced vibration of bearing rotor system.

2. Modal balance method

According to the dynamic theory, the unbalance of the rotor includes static unbalance, couple unbalance, and dynamic unbalance can be divided into multiple unbalanced components according to the main vibration mode, and each component corresponds to a main vibration mode of the rotor. The modal balance method can balance the unbalanced components separately. Each component does not affect each other. The balanced rotor is balanced at all speeds including the critical speed, and there will be no resonance phenomenon due to the speed reaching the critical speed. Figure1 is a schematic diagram of the bearing rotor force.

Assume that the unbalanced force on the rotor is $F$, the vibration differential equation of the rotor is:

$$M\ddot{x}(z) + C\dot{x}(z) + Kx(z) = F(z)$$

(1)

In the formula, $M$ is the mass matrix, $C$ is the damping matrix, $K$ is the stiffness matrix, $x(z)$ is the vibration vector of the rotor. When $F(z)=0$, and without considering the rotor damping, then equation (1) becomes:

$$M\ddot{x}(z) + Kx(z) = 0$$

(2)

The above formula has a natural exponential solution to $x(z) = Xe^{iz}$, $X$ is the amplitude and direction of vibration. Substitute $x(z)$ back to the above formula and simplify to get:

$$(Ms^2 + K)X = 0$$

(3)

If the above formula has a solution, then $|Ms^2 + K| = 0$, this formula is the characteristic equation of the rotor. The eigenvalue $\lambda_i$ and the eigenfunction $\xi_i$ of the eigenequation are solved. The eigenvalue $\lambda_i$ corresponds to the natural frequency of the free rotation of the rotor, and the eigenfunction $\xi_i$ is the main mode of vibration of each order. The vibration modes of each order are orthogonal to each other, including:
In the formula, \( N_i \) is the modal quality of the \( i \)-th mode, \( m \) is the mass of the rotor. The unbalanced force \( F(z) \) of the rotor is:

\[
F(z) = \Omega^2 \sum_{i=1}^{\infty} c_im(z)\xi_i(z)
\]

(5)

Among them \( c_i = c_i e^{ia_i} \), \( c_i \) represents the \( i \)-th mode component, \( a_i \) represents the azimuth of the component in the plane, \( \Omega \) is the speed of the rotor. Suppose \( \Phi(z) \) represents vibration deformation, then:

\[
\Phi(z) = \sum_{i=1}^{\infty} a_i \xi_i(z)
\]

(6)

The vibration curve of the rotor can be regarded as the superposition of the main vibration modes of each order generated according to the ratio of \( a_i \). According to formula (4):

\[
a_j = \frac{\Omega^2}{\lambda_j^2 - \Omega^2} c_j
\]

(7)

ake \( A_1(\Omega) = \Omega^2 / (\lambda_1^2 - \Omega^2) \), \( A_1(\Omega) \) is the dynamic method coefficient when the rotor is running, it can be seen that when the rotation speed \( \Omega \) of the rotor is close to a certain fixed frequency \( \lambda_i \) of the rotor, \( A_1(\Omega) \) tends to infinity, at this time the other modes can be ignored.

When the rotor reaches balance, the support reaction force of the bearing is zero, and can get:

\[
\begin{bmatrix}
F_A \\
F_B
\end{bmatrix} = \frac{\Omega^2}{l} \left[ \int_0^l m(z)\phi dz + \int_0^l zF(z) dz + \sum_{k=1}^{K} z_k Q_k \right] - \left[ \int_0^l (l-z)m(z)\phi dz + \int_0^l (l-z)F(z) dz + \sum_{k=1}^{K} (l-z_k) Q_k \right]
\]

(8)

In the formula, \( Q \) is the corrected mass of the rotor, \( F_A, F_B \) are the supporting reaction force of the bearing, \( \phi \) is the vibration of the rotor. The unbalance of the rotor and the correction quality have the following relationship:

\[
\begin{bmatrix}
\xi_1 Q_1 + \xi_2 Q_2 + \cdots + \xi_n Q_n = F_1(z) \\
\xi_2 Q_1 + \xi_2 Q_2 + \cdots + \xi_n Q_n = F_2(z) \\
\cdots \\
\xi_n Q_1 + \xi_n Q_2 + \cdots + \xi_n Q_n = F_m(z)
\end{bmatrix}
\]

(9)

3. Dynamic balance analysis

The unbalance is added to the front end of the rotor to simulate the unbalance. Three types of unbalance are added: \(5g\cdot\text{mm}, 10g\cdot\text{mm}, \) and \(15g\cdot\text{mm}, \) and the phase is \(0^\circ, 90^\circ \) and \(180^\circ, \) the unbalance is not added at the rear end, and the unbalance at the rear end is zero. The unbalance is known, and then the first two order vibration modes of the rotor are obtained by software analysis, then the
correction amount needed to be added to the balance correction surface can be calculated according to the balance conditions of the modal balance method, and the balance simulation of the modal balance method is performed on the rotor.

At the same rotation speed, the vibration amount of the rotor increases with the increase of the unbalance amount, and the amplitude of the rotor as shown in Figure 2 can be obtained. It can be seen from Figure 2 that the amplitude fluctuation of the rotor at 6000r/min and 14000r/min is large, because the rotor speed is very close to the critical speed. The amplitude after balancing the rotor using the modal balancing method is shown in Figure 3. The maximum amplitude before balancing occurs at a rotational speed of 14000r/min, an unbalanced amount of 15g•mm, the maximum is 32.78μm. The minimum amplitude appears when the rotation speed is 2000r/min and the unbalance is 5g•mm, the minimum is 16.78μm. The maximum balance efficiency occurs when the rotation speed is 2000r/min and the unbalance amount is 5g•mm. The maximum balance rate is 71.31%, and the average balance rate is 66.0%.

4. Conclusion
The maximum balancing efficiency occurs when the speed is 2000r/min, and the unbalance amount is 5g•mm, the maximum balance rate is 71.31%, and the average balance rate is 66.0%. Through analysis, it can be seen that the modal balancing method can effectively reduce the vibration of the rotor. It can accurately obtain the mass of the counterweight, and quickly and effectively balance the high-precision rotating machinery reducing the number of balancing and improving the balancing efficiency.
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