Damping Optimization of High Pressure Rotor Support Based on Harmonic Response Analysis

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Abstract. In order to reduce the vibration of aero-engine passing the critical speed, the vibration of aero-engine passing the critical speed is optimized through in-depth analysis of the supporting damping and dynamic characteristics of aero-engine high pressure rotor system. Firstly, the high pressure rotor of an aero-engine is modeled by SolidWorks 3D software. Then the dynamic characteristics of the high pressure rotor model were analyzed by using ANSYS finite element analysis software. Finally, the parametric design module of ANSYS is used to optimize the supporting damping of the model. The simulation results show that the parametric design of the supporting damping of the high pressure rotor can effectively reduce the vibration of the high pressure rotor passing the critical speed.

1. Introduction
The development trend of modern aero-engine is high speed, high thrust-to-weight ratio and low fuel consumption, which requires high economy as well as high performance. With the improvement of aviation engine speed, the working speed inevitably needs to pass the critical speed of the system. When the working speed passes the critical speed, there will be strong vibration, which may lead to instability of the rotor and even damage of the bearing. Therefore, the critical speed has become the basis and core of the research.

With the development of computer technology, ANSYS finite element analysis software not only fully considers the complex rotor model, but also has high calculation accuracy, so nowadays, modal analysis and harmonic response analysis using ANSYS software has become an effective means of rotor system dynamic characteristics analysis.

Zhang Weizheng et al. studied the impeller bearing dry gas seal system and obtained the steady-state unbalance response curve of the system by harmonic response analysis method[2]; Ren Zhengyi and others analyzed the electromagnetic bearing of flywheel energy storage system through modal analysis and harmonic response analysis, and determined the boundary conditions of the electromagnetic bearing support[3]; With the help of finite element analysis software, Xie Mengtao and others quickly determined the position of unbalance measuring point of rotor tester[4]; Liu Chengjie and others used ANSYS Workbench software to optimize the size and structure of the crank pin of beam pumping unit, which effectively increased the structural strength of the system[5]; Gang Haiming and others have optimized a certain type of high-speed gear box by finite element analysis method, which effectively improves the static and dynamic performance and design efficiency, and
reduces the weight of the box, which provides a reference for the design and optimization of other parts[6].

Based on the inspiration of similar ideas, the modal analysis and harmonic response analysis of high pressure rotor of aeroengine are carried out by finite element analysis software. According to the natural frequency and unbalance response, the support damping of high-pressure rotor is optimized to reduce the vibration of high-pressure rotor passing through the critical speed.

2. The establishment of the model and related theoretical analysis

2.1. Establishment of high pressure rotor model

In this paper, the aero-engine high-pressure rotor is taken as the research object, and a certain type of aero-engine is taken as the prototype. This type of aero-engine high-pressure rotor is composed of 6-stage high-pressure compressor and 1-stage high-pressure turbine. The rotor part is mainly composed of turntable blade and shaft. The high-pressure rotor model is established in SolidWorks 3D software. In order to facilitate the subsequent calculation, the process features such as chamfer, fillet and nut are ignored.

![Figure 1. High pressure rotor model.](image)

2.2. The concept of optimal damping

In the process of design and optimization of aeroengine rotor system, two key factors, dynamic load and unbalance response, should be considered. There are many factors that affect the dynamic load and unbalanced response of the support, such as the properties of the material, the position and size of the turntable, the characteristics of the support and the moment of inertia. However, the system structure, support scheme and performance parameters are often limited by the overall design and performance index, so it is difficult to make large changes. Therefore, it is an effective and easy method to control the vibration of the rotor system by selecting the appropriate support damping, so that the rotor system has good dynamic characteristics. In aeroengine, damper is often used to reduce vibration. In the practical complex rotor support system, it is difficult to determine the optimal damping, but in the qualitative analysis, it can be deduced by the following formula:

- \( X_k = F/K = F_0 \sin \omega t/K = X_{k0} \sin \omega t \)
- \( X_c = F/C = F_0 \sin \omega t/C \)
- \( \dot{X}_c = -F_0 \cos \omega t/(\omega C) = -X_{c0} \cos \omega t \)
- \( X = X_k + X_c = X_0 \sin(\omega t - \alpha) \)
- \( X_0 = (X_{k0}^2 + X_{c0}^2)^{1/2} \)
- \( \alpha = \arctan(X_{c0}/X_{k0}) \)
- \( X_k \) —— Displacement of supporting elastic element
- \( X_c \) —— Displacement of damping element

The optimal damping \( C_{opt} \) is the damping coefficient whose ratio of the energy absorbed by the damping in a period to the vibration energy of the system is the maximum. Assuming that the vibration mode of the rotor system remains unchanged, the amplitude ratio of each point of the rotor
system remains unchanged at a certain speed, so the conditions for \( e_c/E \) and \( e_c/X_0^2 \) to reach the maximum are the same.

- \( e_c = \pi \omega c X_{c0}^2 \)
- \( e_c/X_0^2 = \pi \omega c X_{c0}^2/X_0^2 \)
- \( e_c = \frac{\pi k f}{f^{2+1}} \)
- \( f = \omega c \)

- \( E \) — Vibration energy of the system
- \( e_c \) — The energy absorbed by damping in a period

According to the maximum condition, \( f = 1 \):

\[ C_{opt} = \frac{K}{\omega} \]

The concept of optimal damping and its calculation formula provide a theoretical basis for the optimal design of rotor damping. Providing enough damping can effectively reduce the vibration of the whole system. But it is not the larger the damping value will have better vibration reduction effect.

2.3. Theoretical basis of modal analysis

When considering the influence of gyro moment on rotor system in Workbench, the rotor dynamic equation of multiple degree of freedom is

\[ [M][\ddot{\phi}] + ([C] + [C_{cor}])\dot{\phi} + ([K] - [K_{spin}])\phi = [F(t)] \]

- \( M \) is the mass matrix; \( C \) is the damping matrix; \( K \) is the stiffness matrix, and \( F \) is the force vector matrix.

Natural frequency and mode shape are one of the inherent properties of mechanical structure. It has nothing to do with the force, and modal analysis is a kind of linear analysis, therefore, damping and external excitation are not considered in modal analysis, so the above formula can be simplified as:

\[ [M][\ddot{\phi}] + ([K] - [K_{spin}])\phi = 0 \]

The free vibration of an elastic body can always be decomposed into a series of simple harmonic vibrations, that is to say, it can be assumed to be a harmonic response motion, the equation is simplified as:

\[ \left( ([K] - [K_{spin}]) - \omega_i^2[M] \right)\phi = 0 \]

\( \omega_i^2 \) is the eigenvalue, \( \omega_i \) is the natural frequency corresponding to the mode shape, \( \{\phi_i\} \) is the eigenvector. So the characteristic equation of the free vibration equation is:

\[ \left| ([K] - [K_{spin}]) - \omega_i^2[M] \right| = 0 \]

N unequal eigenvalues can be obtained, \( \omega_1^2, \omega_2^2, \ldots, \omega_N^2. \) After square root, the natural frequencies of each order can be obtained.

2.4. Theoretical basis of harmonic response analysis

The purpose of harmonic response analysis is to calculate the response curve of the structure at several frequencies, so that the designer can predict the sustained dynamic of the structure and test whether the design can overcome the harmful effects such as resonance. Its essence is to solve the forced vibration equation of structure, the equation of forced vibration is:

\[ [M][\ddot{u}] + [C][\dot{u}] + [K][u] = [F(t)] \]

\[ F(t) = F_0 \sin \omega t \]

3. Dynamic analysis of high pressure rotor

3.1. Modal analysis

In order to obtain the critical speed and the corresponding natural frequency of the rotor system, the modal analysis of high pressure rotor model was carried out by using the Workbench. The HP rotor
model established in SolidWorks is imported into workbench with setting the physical parameters of each part and making meshes, as shown in the figure:

![Figure 2. Meshing.](image)

Unlike other non rotating mechanical structures, when calculating the critical speed of rotor system, it is necessary to consider the influence of gyroscopic effect on the critical speed, therefore, the Coriolis effect is set to “on” in modal analysis. Through the workbench modal analysis, the critical speed and the corresponding natural frequency can be divided into positive precession and negative precession, and usually, only the frequency of Synchronous Forward whirl is considered as the natural frequency in actual operation. The first three modes obtained by modal analysis are shown in the figure.

As the modal design requires that the idle speed is below the first critical speed, and the cruise speed is above the first critical speed and below the second critical speed, so the amplitude of rotor speed near the street is considered as the subsequent optimization objective.

![a. First mode shape.](image)

![b. Second mode shape.](image)
According to the modal analysis, the Campbell diagram can also be extracted as shown in the figure. The abscissa represents the speed and the ordinate represents the frequency.

The straight line from the origin represents the exciting force frequency, and the horizontal line in the figure represents the natural frequency. Resonance occurs when the two intersect. The abscissa and ordinate of the intersection point are the critical speed and natural frequency. The values are shown in the table.

Table 1. Formatting sections, subsections and subsubsections.

| Mode | Critical speed | Natural frequency |
|------|----------------|-------------------|
| 1    | 3659.5 rpm     | 60.992 Hz         |
| 2    | 14593 rpm      | 275.5 Hz          |

In the actual modal design process, the idle speed is required to be below the first critical speed, and the cruise speed is above the first critical speed and below the second critical speed. Therefore, the amplitude of rotor at the first critical speed is considered as the objective of subsequent optimization.

3.2. Harmonic response analysis
The input of harmonic response analysis is harmonic load whose magnitude and frequency are known and it is used to determine the steady-state response of the rotor system under the sinusoidal (simple harmonic) load varying with time. Using workbench to analyze harmonic response needs to know the distribution and frequency of external load.
According to the mechanical equation, the formula of centrifugal force caused by mass eccentricity is as follow:

- \( F = me\omega^2 \)

\( F \) is the centrifugal force; \( M \) is the eccentric mass; \( e \) is the eccentricity and \( W \) is the angular velocity of rotation. Without considering the effect of the axial force, \( F \) is decomposed into two components in X and Y directions. Let the harmonic forces in X and Y directions of the radial plane be:

- \[
\begin{align*}
F_x &= me\omega^2 \sin \omega t \\
F_y &= me\omega^2 \cos \omega t
\end{align*}
\]

Only forced vibration when the external excitation is harmonic force is considered here, although this is the simplest case and the working process of the rotor system is more complex in practice, the dynamic loads received by the aero-engine are generally periodic excitation, which can always be expressed as the series of simple harmonic functions by Fourier decomposition. Therefore, it is general to regard external load as harmonic force.

According to the results of the modal analysis, the frequency of the external load is set as 0 ~ 300Hz, covering the natural frequencies corresponding to the first and second critical speeds. The amplitude of front and rear supports and high pressure turbine are selected as the output, and the unbalance response curve obtained by harmonic response analysis is shown in the figure.
According to the results of harmonic response analysis, the frequencies corresponding to the peak of amplitude are 60Hz and 275hz, just corresponding to the first and second natural frequencies obtained by modal analysis. It can be seen that the amplitude reaches the maximum near the natural frequency corresponding to the first critical speed, where a strong resonance occurs. Subsequently, the amplitude of these three places is taken as the optimization objective to optimize the support damping of the high-pressure rotor.

4. Optimization of support damping

The vibration of aero-engine rotor system passing through critical speed is a key problem in anti vibration design. In order to solve this problem, in addition to shorten the time of rotor passing through critical speed, the unbalance sensitivity of rotor system can be reduced by adjusting the support damping, so as to reduce the external force.

The method of selecting multiple optimization variables and optimization objectives is called multi-objective optimization, which using the parametric design function of workbench to carry out the multi objective optimization design of high pressure rotor model. The damping of front and rear supports is selected as the optimization variable. The damping range is 0 ~ 4000 n.s/m, and the initial value is 2000 n.s/m. The amplitudes of the front and rear supports and the high pressure turbine are taken as the optimization objectives.

In general, the influence of the support damping of the rotor system on the modal and critical speed is very small and can be ignored, but it has a great influence on the response of the rotor system. The sensitivity of the front and rear support damping to the unbalance response is based on the parametric design results, as shown in the figure.
Figure 6. Sensitivity of damping of front and rear supports to response.

As shown in the figure, red is the sensitivity of the front support damping to the response, and blue is the rear support damping. The three sets of data correspond to the amplitudes of three positions. It can be seen that the front support damping has little effect on the response, only about 10%, and the rear support damping can reach more than 80%.

The three optimization schemes of parametric design are shown in the table:

Table 2. Optimization scheme.

| Scheme  | Front support damping | Rear support damping |
|---------|-----------------------|----------------------|
| Scheme 1| 3000                  | 3700                 |
| Scheme 2| 3800                  | 2450                 |
| Scheme 3| 2600                  | 1700                 |

By comparing the harmonic response analysis results of the rotor system before and after optimization, it can be seen that changing the damping of the front and rear supports at the same time can reduce the optimization objective.

5. Conclusion

in view of the influence of supporting damping on the vibration of rotor system passing through critical speed, this paper carried out modal analysis, harmonic response analysis and parametric design of high pressure rotor model are carried out through the Workbench, and drew the following conclusions:

(1) According to the modal analysis and harmonic response analysis of the high-pressure rotor, when the working speed reaches near the critical speed, the vibration of the rotor system will come to a peak due to resonance, which is not conducive to the normal operation of the system.

(2) According to the results of simultaneous optimization and adjustment of the front and rear support damping, the influence of the front and rear support damping on the response of the rotor system through the critical speed is significantly different, and the influence of the rear support damping is much greater than that of the front support damping.

(3) At the same time, the optimal design of support damping is helpful to reduce the response of rotor system through critical speed.
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