Dynamic Analysis and Experimental Research on the Cantilever High Speed Rotor of Turbofan Engine

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Abstract. Taking a simulated low-pressure rotor of a turbofan engine as the research object, the dynamic characteristic calculation model of the simulated low-pressure rotor with hollow shaft and solid shaft was established respectively by using the finite element method. The first three order critical speeds and mode shapes of the rotor were calculated systematically. The dynamic balance test of the simulated low-pressure rotor and the research on the damping effect of the squeeze film damper were carried out on the high-speed rotating test rig, and the dynamic characteristic test was completed in the full speed range. The results show that the finite element results agree well with the experimental results. The rationality of the rotor's hollow shaft structure and rotor dynamic design was verified. The research provides reference and technical support for structural design, dynamic design and experimental research of real low-pressure rotor in engine.

1. Introduction
Rotor is an important rotating part of turbine machineries such as motors, pumps, compressors, gas turbines and aero-engines. It is of great significance to study rotor dynamics. Many scholars like Zhou [1], He[2], Zhang[3], Tai[4], Du[5] and other researcher[6-9] have studied the dynamics of turbo machinery rotor and achieved remarkable results. Compared with the rotor of other turbomachinery, the rotor of aeroengine was characteristic of high speed, complex structure and bad working condition. Establishing a calculation model which can reflect the actual situation of aero-engine rotor was the basis of rotor analysis. At present, the finite element method was widely used to establish the calculation model of rotor. Deng [6,7], Zhu [8], Liu [9], Li [10], Miao [11], Mo [12] and other scholars used the finite element method to carry out the dynamic analysis of the aeroengine rotor, and achieved considerable results. The dynamic characteristics of the rotor were affected by many factors, in the structural design stage of the rotor, it was necessary to comprehensively consider many factors, such as the supporting layout, supporting mode, mass distribution and critical speed. Therefore, it was often necessary to design the rotor with various structural forms, and calculate and analyze the dynamic characteristics of the rotor with different structural forms. Through comparative analysis, select one of the more reasonable scheme of rotor structure. Deng [13,14] studied the law of critical speed of high-speed flexible rotor changed with support stiffness and the influence of support layout on the critical speed of a low-pressure rotor, which provided a theoretical basis for the design and adjustment of the critical speed of the rotor. Nie [15,16] analyzed the influence of tie bars on the dynamic characteristics of high-speed flexible rotor, and studied the influence of support stiffness and...
disk mass on critical speeds of that rotor. However, whether the rotor structure was reasonable need to be verified by test. In the process of test, it was often unable to operate safely to the rated working speed due to excessive imbalance, and high-speed dynamic balance test was often required. Deng [17,18,19] conducted high-speed dynamic balance research on power turbine rotor and low-pressure rotor of aeroengine, which developed high-speed dynamic balance technology of flexible rotor.

In this paper, a simulated low-pressure rotor of a turbofan engine was taken as the research object to carry out dynamic analysis and experimental research. Through calculation analysis and test, the rationality of hollow structure and critical speed distribution were verified. Aiming at the actual complex structure of the simulated low-pressure rotor, high-speed dynamic balance test was carried out, and the effect was good. Besides, the vibration reduction effect of squeeze film damper was verified, and the comparative analysis of calculation results and test results were made. The results show that the finite element model well reflected the real dynamic characteristics of the rotor, and the test verified that the structure and dynamic design of the simulated low-pressure rotor were reasonable, which provided a reference for the dynamic design, structural design and test of the real low-pressure rotor in engine.

2. Rotor Structure
The structural sketches of the simulated low-pressure rotor with hollow shaft and solid shaft are shown in figure 1 and figure 2 respectively. Except for the different structure of the low-pressure shaft, the other structures of the rotors were coincident. The whole rotor was mainly composed of simulated fan disk, simulated turbine disk, low-pressure shaft and other main parts. The cantilever of simulated fan disk was adopted. No. 1 bearing was ball bearing, No. 2 and No. 5 bearing were roller bearing. The rotor is a typical high-speed flexible rotor.

3. Finite Element Analysis Model
The finite element calculation models of simulated low-pressure rotor with hollow shaft and solid shaft are shown in figure 3 and figure 4 respectively. In the model, beam element was used to simulate the main body of the rotor, bearing element was used to simulate the support of the rotor, five hundred and fifty-seven beam elements, three bearing elements and fourteen rigid connection elements were used to simulate the low-pressure rotor with hollow shaft, four hundred and eighty-five beam elements, three bearing elements and fourteen rigid connection elements were used to simulate the low-pressure rotor with solid shaft.

Figure 1. Structural Sketch of the Simulated Rotor with Hollow Shaft.

Figure 2. Structural Sketch of the Simulated Rotor with Solid Shaft.

Figure 3. Finite Element Model of the Simulated Low-pressure Rotor with Hollow Shaft.

Figure 4. Finite Element Model of the Simulated Low-pressure Rotor with Solid Shaft.
4. Calculation Results of Dynamic Characteristics

4.1. Supporting Stiffness
The rotor is supported by squirrel cage elastic support structure, as shown in figure 5. The actual stiffness of each elastic support is shown in table 1. Under the stiffness of each elastic support, the first three order critical speeds and mode shapes of the simulated low-pressure rotor with hollow shaft and solid shaft were calculated respectively.

![Figure 5. Structural Sketch of the Elastic Support.](image)

Table 1. Actual Stiffness of Each Elastic Support.

| Supporting Number | No.1 | No.2 | No.5 |
|-------------------|------|------|------|
| Stiffness(E+7N/m) | 1.21 | 2.30 | 0.44 |

4.2. Comparative Analysis of Critical Speed
The calculation results and margins of the first three order critical speeds of the simulated low-pressure rotor with hollow shaft and solid shaft are shown in table 2. The critical speed margin related to rated working speed was defined as: Critical Speed Margin = ((Rated Working Speed - Critical Speed) / Rated Working Speed) *100%.

Table 2. The First Three Order Critical Speeds and Margins.

| Stage                      | First       | Second      | Third       |
|----------------------------|-------------|-------------|-------------|
| Rotor with Hollow Shaft    | Critical Speed(rpm) | 7831 | 15237 | 38322 |
|                            | Margin(%)   | >70         | >40         | >30         |
| Rotor with Solid Shaft     | Critical Speed(rpm) | 7780 | 15649 | 28178 |
|                            | Margin(%)   | >70         | >40         | <5          |

The simulated low-pressure rotor with hollow shaft has two critical speeds within the rated working speed range, and the simulated low-pressure rotor with solid shaft has three critical speeds within the rated working speed range. The first two order critical speeds of the simulated low-pressure rotor with hollow shaft and the solid shaft were not very different. It is mainly because the other factors affecting the critical speed and the structure of the two types of rotors were the same except for the low-pressure shaft. However, the critical speed of the third stage was quite different, indicating that the structure of the low-pressure shaft has no effect on the first two order critical speed of the rotor, but it has a great influence on the third stage critical speed. The first three order critical speed margins of simulated low-pressure rotor with hollow shaft were all greater than 30%, which meet the requirements of critical speed design criteria. The first two order critical speed margins of simulated low-pressure rotor with solid shaft were more than 40%, but the third stage critical speed margin was less than 5%, which does not meet the requirements of critical speed design criteria. From the perspective of critical speed margin, the simulated low-pressure rotor with hollow shaft should be selected.
4.3. Comparative Analysis of Mode Shape

The calculation results of the first three order mode shapes of the simulated low-pressure rotor with hollow shaft and solid shaft are shown in table 3. It can be seen from table 3 that the first three order mode shapes of the simulated low-pressure rotor with hollow shaft and solid shaft were all bending mode shapes, and the simulated low-pressure rotor was a typical high-speed flexible rotor; the first three order mode shapes of the simulated low-pressure rotor with hollow shaft and solid shaft were basically the same, and the structural form of the low-pressure shaft has no effect on the first three order mode shapes of the rotor.

Table 3. The First Three Order Mode Shape of The Rotor.

| Mode | First | Second | Third |
|------|-------|--------|-------|
| Hollow Shaft | ![Image](image1.png) | ![Image](image2.png) | ![Image](image3.png) |
| Solid Shaft | ![Image](image4.png) | ![Image](image5.png) | ![Image](image6.png) |

The first three order mode shapes of the simulated low-pressure rotor with hollow shaft and solid shaft were all bending mode shapes, and the simulated low-pressure rotor was a typical high-speed flexible rotor; the first three order mode shapes of the simulated low-pressure rotor with hollow shaft and solid shaft were basically the same, and the structural form of the low-pressure shaft has no effect on the first three order mode shapes of the rotor.

5. Dynamic Characteristics Test and Result Analysis

The dynamic characteristic test of the simulated low-pressure rotor was carried out on a high-speed rotating test rig. The schematic diagram of the installation and test of the rotor is shown in figure 6. The displacement sensors were arranged at positions ①, ②, and ③ as shown in figure 6 to measure the deflection of the rotor. The displacement sensors \( D_1 \) and \( D_2 \) were arranged at vertical direction of position ① and ③ respectively, and \( D_1 \) and \( D_2 \) were arranged at vertical and horizontal direction of position ② respectively. Four vibration acceleration sensors were arranged on two rigid supports to measure the vibration accelerations. In Figure 6, "⊥" indicates vertical direction, "=" indicates horizontal direction, \( A_1-A_4 \) indicates acceleration sensor, \( D_1-D_4 \) indicates vibration displacement sensor. The installation photo of the rotor on the tester is shown in figure 7.

Figure 6. Schematic Diagram of the Installation and Test of the Simulated Low-pressure Rotor.  
Figure 7. Photo of the Rotor on the Tester.
5.1. Test Result
According to the test results, the curves of rotor deflection change with rotation speed measured by four displacement sensors are shown in figure 8. The critical speeds measured by each sensor and the corresponding deflection value are also identified.

![Figure 8. Curves of Rotor Deflection Change with Relive Speed Measured by Four Sensors.](image)

5.2. Result Analysis
The analysis of the critical speed calculated results and the experimental results (average value) were obtained, as shown in table 4. The calculation error was defined as: \( \text{Calculation Error} = \left( \frac{\text{Calculation Result} - \text{Test Result}}{\text{Test Result}} \right) \times 100\% \).

| Stage | First | Second |
|-------|-------|--------|
| Calculation Result(rpm) | 7831  | 15237  |
| Test Result(rpm)         | 7524  | 15375  |
| Error (%)                | 4.08  | 0.90   |
| Margin of Test Result (%)| 74.06 | 46.98  |

The calculated errors of the first two order critical speeds are all less than 5%, the calculated results are in good agreement with the test results. The rotor has obvious resonance peak at the critical speed, indicating that the rotor has been bending deformation, it is also consistent with the calculated vibration mode shapes. From the test results, the margin of the first two order critical speed of the rotor are all greater than 20%, which meets the critical speed margin requirement [20].

6. Experimental Study on High Speed Dynamic Balance
It can be seen from figure 8 that the rotor can smoothly cross the first two order critical speeds and run to the rated working speed, but the deflection at the rated working speed is large, and the deflections measured by \( \text{D}_1 \), \( \text{D}_3 \) and \( \text{D}_4 \) sensors are not less than 82\( \mu \)m, high-speed dynamic balance for the rotor is required to reduce the rotor deflection, in order to running safely and stably at the rated working speed.

6.1. Balance Difficulties
The simulated low-pressure rotor working across the two-stage bending critical speed, it is a typical high-speed flexible rotor and it is very sensitive to the unbalance, so it is very difficult to carry out the high-speed dynamic balance test on it, which is mainly reflected in the following aspects.
(1) The ratio of length to diameter of low-pressure shaft is greater than 20. It is a high-speed flexible rotor with long diameter ratio slender flexible shaft, and the additional imbalance caused by vibration mode is very large.

(2) The low-pressure shaft adopts hollow thin-walled structure, and the coaxially of the inner and outer walls is difficult to guarantee, which inevitably leads to a certain initial imbalance of the rotor.

(3) The balance surface on the rotor is very limited, with only two balance surfaces and two balance bosses (No.1 boss and No.2 boss as shown in figure 1) on the low-pressure shaft.

(4) The length of the two balance bosses is only 20 mm and the height is only 1.5 mm. Not only the amount of material removed by grinding is very limited, but also the low-pressure shaft cannot be damaged in any form during grinding, so it is difficult to operate.

6.2. Balance Process
The influence coefficient method was used to balancing, and the rated working speed with a balancing speed of 89.66% was selected. The measuring surface was near the No.2 boss. The sensor was D3, and the balance face was No.1 boss. The high-speed dynamic balance process is shown in table 5.

Table 5. High Speed Dynamic Balance Process.

| Results Measured by D3 Sensor | Process                                      |
|-------------------------------|----------------------------------------------|
| Deflection(μm) | Phase(°) | Initial State                                      |
| 86     | 102     | After Adding the Fitting Weight at 180 ° Position of No. 1 boss |
| 103    | 104     | According to the Calculation of the Test Instrument, It is Necessary to Add 2.21 Times of the Fitting Weight at the 353 ° Position of No.1 boss, and Actually Move the Original Fitting Weight to the 353 ° Position |
| 70.5   | 80      | After Adding 2 Times of Original Weight at 353 ° Position of No.1 Boss |
| 26.5   | 117     | After Grinding the Material at the Corresponding Position of No.1 Boss |
| 33.5   | 359     |                                                        |

As shown in table 5, the rotor deflection of the D3 sensor was reduced from 86μm to 33.5μm after high speed dynamic balance, and a good balance effect was achieved.

6.3. Balance Effect
After high-speed dynamic balance the deflection of the rotor measured by four displacement sensors at rated working speed were reduced, and the effect of high-speed dynamic balance at rated working speed is shown in table 6.

Table 6. High Speed Dynamic Balance Effect at Rated Working Speed.

| State            | D1   | D2   | D3   | D4   |
|------------------|------|------|------|------|
| Before Balancing(E-6m) | 84.5 | 30   | 82.5 | 83   |
| After Balancing(E-6m)  | 23   | 26   | 10   | 16   |
| Effect (%)        | 72.78 | 13.33 | 87.88 | 80.72 |

As can be seen from table 6, the dynamic deflection of the four deflection measuring points at the rated working speed has decreased by 72.78%, 13.33%, 87.88% and 80.72% respectively, and the balance effects are very satisfactory.
7. Experimental Study on High Speed Dynamic Balance

In order to verify the damping effect of squeeze oil film damper, two structures were set at No. 2 bearing: one with squeeze oil film and the other without squeeze oil film. And verification test on simulated low-pressure rotor with hollow shaft was carried out. The oil film parameters are shown in table 7, and the schematic diagram is shown in figure 6.

**Table 7. Oil Film Parameter.**

| Parameter | Oil Film Length (mm) | Oil Film Thickness (mm) |
|-----------|---------------------|------------------------|
| Value     | 16                  | 0.16                   |

**Figure 9. Schematic Diagram of Squeeze Film Damper.**

The test results of the rotor with squeeze oil film damper are shown in Figure 10.

**Figure 10. Curves of Rotor Deflection Change with Relive Speed Measured by Four Sensors.**

The deflection measured by each vibration displacement sensor at critical speeds is shown in table 8.

**Table 8. Comparison of Deflection Values at Critical Speed.**

| Sensor               | First Order Critical Speed | Second Order Critical Speed |
|----------------------|---------------------------|----------------------------|
|                      | D₁ | D₂ | D₃ | D₄ | D₁ | D₂ | D₃ | D₄ |
| Without Squeeze Film | 62 | 30 | 53 | 60 | 80 | 48 | 95 | 83 |
| With Squeeze Film    | 47 | 21 | 49 | 54 | 62 | 42 | 89 | 68 |
| Decrement (%)        | 24.19 | 30.00 | 7.55 | 10.00 | 22.50 | 12.50 | 6.32 | 18.07 |
As shown in table 8, the deflection values of all measured points at critical speeds are significantly reduced, and the reduction ranges are not less than 6.32%, indicating that the squeeze film damper has a good effect of vibration reduction in the full speed range.

8. Conclusion
In this paper, the first three order critical speeds and mode shapes of simulated low-pressure rotor with solid shaft and hollow shaft were calculated and compared. The high-speed dynamic balance test of the rotor was carried out. The balance effect was good and the vibration reduction effect of the squeeze film damper was verified. The research results can be used directly to the real rotor in engine. The main conclusions are as follows.

1. The first three order mode shapes of the simulated low-pressure rotor with hollow shaft and solid shaft are basically the same, and they are all bending modes. The structure of the low-pressure shaft has no substantial influence on the first two critical speeds of the simulated low-pressure rotor, but has a great influence on the third critical speed, and has no substantial influence on the first three order mode shapes.

2. Low-pressure shaft should adopt hollow shaft structure to making the rotor critical speed meet the design criteria.

3. The critical speed calculation error is less than 5%, which reflects the real dynamic characteristics of the rotor very well.

4. The balance method can achieve high speed dynamic balance of cantilever flexible rotor, and the balance effect is good.

5. Squeeze film damper has significant effect on vibration reduction of slender shaft high-speed flexible rotor.

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