HIGH FREQUENCY MODAL TEST AND DYNAMIC PERFORMANCE EVALUATION OF TURBINE ROTOR BLADES

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Abstract

In order to evaluate the dynamic characteristics of turbine blades in simulated centrifugal force, the vibration characteristics of blades were studied by means of numerical analysis and modal test. In the experiment, the frequency response curve contained the peak, generated by the inner structure of the blade. In this paper, the energy of peak value of the vibration relatively is determined to estimate the peak value, generated by the vibration of the blade. In this paper, the test results are in good agreement with the numerical calculation, and the frequency error is 1.8%, which is of great reference value to the engineering development of high speed turbine blades.

1 Title of Section (e.g. General Introduction)

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The papers will be in A4 size with margins top 30 *mm*, left and right 17.5 *mm*, bottom 22 mm. A column separation of 10 *mm* is used. The text should be Times or Times Roman with font size 12 point.

1 Introduction

Turbine rotor blade is one of the key parts of the aviation engine. Its work is not only subjected to high temperature, high pressure and large centrifugal force, but also is affected by the periodic fluctuation of the aerodynamic force. When the frequency of the excitation force is very close to the natural frequency of the blade, the blade will be resonant, causing the blade to bear a large dynamic stress [1], which causes the failure of the high cycle fatigue of the blade, with catastrophic consequences. In order to improve the safety and reliability of the air engines and the operational life of the blades, in the development, production and use of the aviation engine, it is necessary to obtain the kinetic properties of the blades in conjunction with the numerical and experimental methods[2], and to obtain the vibration stresses of the actual work state, to test the reliability and validity of the design analysis.

Because of the special structure and complicated internal structure of the turbine blades, the aero-engine has the characteristics of high frequency and small quality. In the modal test of turbine blades, setting the boundary conditions, the distribution of the excitation points and response points and selection, the size of the exciting force and the analysis of the related parameters Settings, and so on, all brought certain difficulty to test; In addition, the use of different test methods and analysis will also affect the final result, appear even wrong results, therefore, the rotor of low frequency in the modal test is often difficult to use conventional test method.

Wang and Cheng[3] developed a kind of only using the free vibration of the structure of information to identify more order modal parameters of the method, which using the discrete time sequence data ingeniously converts the generalized eigenvalue problem to general eigenvalue problem, and verified this method by using a numerical example of band noise. Chen[4] used compressor blades as the research object, by means of unloading load, to ensure that the measured free vibration contained the components of high frequency modality, the first 8 order mode damping ratio within 4300Hz was identified. Because of the need to recognize multi-order mode frequency, it is necessary to gradually eliminate the influence of the recognized mode from the original data, and the operation is more complicated.

In the above literature, the precondition is that the data identified has clearly contained the modal information concerned. However, the high frequency mode response is generally not easily aroused in the laboratory environment. In order to stimulate the high frequency mode of the system, this paper adopts the method for combination of numerical simulation and modal test to analyze the dynamic characteristics of an aero-engine turbine rotor blade, which is of great reference value for the engineering development of high speed turbine blades.

2. Numerical analysis of blade dynamic characteristics

To investigate the dynamic characteristics of the engine when engine working, the five adjacent blades are installed in a test device that can be used to simulate the centrifugal force, the five mortise of the test device is designed to be the same as the five mortise of the wheel, finally, the test device is fixed.

2.1 calculation model

The mechanical property parameters of the blades are given at normal temperature, the elastic modulus is 131.5GPa, the density is 8780kg/m3, and poisson's ratio is 0.344. The length of blade span is 50mm approximately, span-chord ratio is 2:1 approximately. the fir-tree mortises is installed in tenons. the fir-tree mortises is fixed in tenons by the centrifugal force. Therefore, it is a non-linear problem of elastic contact. While, the contact surface of tenon and mortise is unchanged when working, when calculating the the natural frequency of the blade, the tenon surface that is in contact with the mortise groove can be completely fixed, and the degree of freedom is zero, so that the nonlinear problem is transformed into a linear problem.

The purpose of the simulation calculation is to find the first order natural frequency and mode for the blade mounted in the test device. The test device and five blades were converted into the calculation model, and the leaf parts of the four blades on both sides were cut off, and the tenon and mortise surfaces of the blade were completely fixed. The intermediate blade model of the simulated calculation was shown in figure 1. Using structured hexahedron and pentahedral mesh to divide gray parts, the tetrahedral mesh was used to divide green parts. The two parts use Tie connections on the interface.

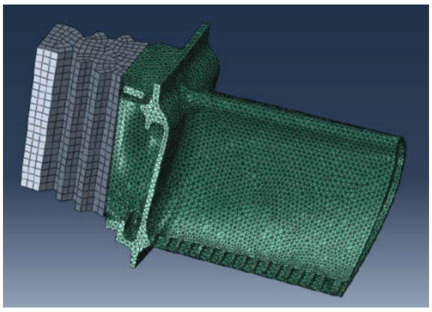


FIG. 1 simulation model

2.2 calculation results

After the completion of the solution, the local mode of the test device and the blade is received, natural frequency of the first order mode is 3071.5Hz,the first-order mode shape is shown in FIG. 2.

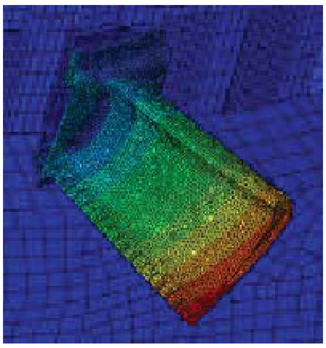


FIG. 2 first-order mode shape

3. High frequency modal test of turbine blades

3.1 Excitation scheme and measurement scheme

In order to provoke all the blade mode within 10kHz, the required excitation scheme should satisfy the following orders: The vibration source can still produce enough power in the high frequency, the excitation force distribution is not orthogonal to the mode shape. In order to meet two requirements on above, the hammer strikes the blade[5], the contact time between the hammer and the blade surface is short, so the excitation signal is close to the Dirac pulse, the base band is 10kHz.

In order to obtain the blade mode shape, the initial design adopts the test method of the fixed response point moving tapping point (MISO) ,in order to minimize the chance that the order mode can't be excited because of the excitation point selected on a certain order mode node.

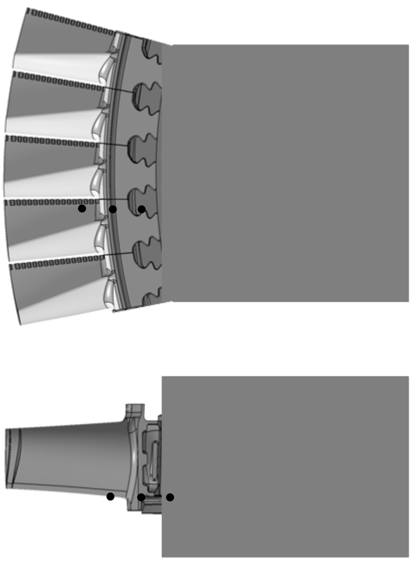
3.2 Determination of blade installation position, excitation point and test point.

In the modal test, the test piece mounting point shall be selected in a position of a small displacement response. Therefore, in order to determine the relative magnitude of displacement response of a certain degree of freedom under excitation, the average driving degree of freedom displacement(ADDOFD) of the j degree of freedom shall be the minimum, in order to determine the mounting point of the blade in accordance with the ADDOFD's minimum degree of freedom; In addition, following the actual conditions of the blades, the tenons of the blades are attached to the ground through the test device.

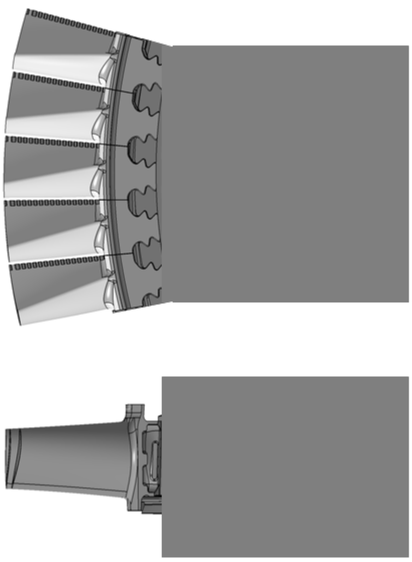
The excitation point should not be too close to the node or the line, and to avoid the higher value of the average drive degree of freedom velocity (ADDOFV), in order to avoid the impact of the double click in the test, the preliminary excitation point is three points on both long sides uniformly.

The installation of acceleration sensor will influence the modal test because of the additional quality, the quantity and quality of acceleration sensor should be minimized. The preliminary test point is larger than the average drive degree of freedom acceleration (ADDOFA).

The test piece installed was shown in Fig. 3. The black dot in (a) is six excitation points, and the black dot in (b) is a measuring point.



(a) excitation points



(b) measuring point

Fig.3 installation drawing

3.3 data processing method

When using the force hammer to stimulate, the force signal and the response signal are the deterministic signals. After the Fourier transform, the frequency response function H(f) can be obtained. However, there is always noise interference in the actual measurement signal. In order to eliminate the influence of noise, the real frequency response function must be obtained [6]. Six groups of transfer rate data are available for this test.

The six groups of frequency response function data contain all modal information of the blade and the internal structure of the blade, there are many peaks in the graph, which are characterized by that the blade vibration is relatively prominent in these frequency bands. The frequency bands can be divided into two types:

1) In the first type of frequency band, the internal structure is more involved in the vibration, or the blade vibration occur at the same time;

2) In the second type of frequency band, only the blade vibrates, and the internal structure does not participate in vibration obviously.

Only the blade is resonant, and the frequency band with no resonance in the internal structure is likely to be the resonant frequency band dominated by blade vibration. the following method is used for discrimination:

Within the spectrum, if $H_b(f) > H_i(f)$, and the peak of the frequency response function of the blade is happened, there is only a ω_b, makes

$$\frac{dH_b}{d\omega}(\omega_b) = 0 \tag{1}$$

This means that there is a possibility of the modal of the blade vibration, and then the peak point of the frequency response function in the frequency band is continued to be research.

1) case 1, if the transfer function of the internal structure also has a peak value, and there is also a certain amount of ω_i, which is satisfied.

$$\frac{dH_i}{d\omega}(\omega_i) = 0 \qquad (2)$$

It is considered that the frequency band is the first frequency band.

2) case 2, if not satisfied case 1, and

$$\frac{\int_{\omega_s}^{\omega_e} H_b(\omega)d\omega}{\int_{\omega_s}^{\omega_e} H_i(\omega)d\omega} > 5 \qquad (3)$$

It is thought that only blade resonate and belong to the second frequency band.

3) case 3, if the case 1 is not satisfied, and the formula (3) is not satisfied, although the peak of the frequency function for the blade is occurred, and the peak of the transmission rate of the internal structure is not occurred, but the energy ratio in the frequency band is less than 5, the internal structure is still fairly involved in the vibration, and still judged to be the first frequency band.

3.4 test data and results.

The peak value of the transmission rate in each frequency response function is obtained, and the envelope operation is taken. All these peaks are set in one curve. The peak value of the curve includes not only the modal information of the blade, but also the modal information of the internal structure of the blade. According to (1) - (3) calculation, modal vibration mode is extracted at each peak, and the first order modal parameters of the blade are obtained by combining the above methods.The mode shape is shown in Fig.4.

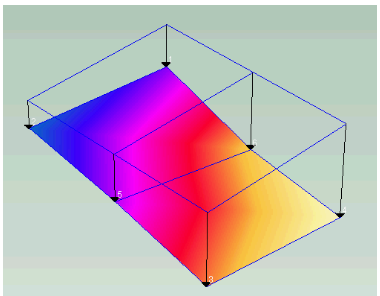


Fig.4 First-order modal vibration mode f1=3016.64Hz.

The test value of the first order bending mode is compared with the calculated value, and the first-order frequency error is 1.7%. The test results is lower than the calculating result. The reason is that the test device is connected with the ground through the large bolt, and the test device is fixed in the numerical calculation, so the boundary condition of the test device model causes the error.

4. Conclusion

In this paper, the dynamic characteristics of turbine blades are analyzed and evaluated by means of numerical analysis and experiment. The conclusions are as follows:

(1) In terms of mode, the numerical calculation can better capture the natural frequency and mode shape of the blade, the deviation from the test result is a certain deviation, the relative error of the frequency is less than 2%, therefore, in the actual analysis of the dynamic characteristics of the turbine, the model can adopt a single blade tenon and the mortise fixation, and cut off the blade body of the adjacent blade, the numerical calculation can be used to guide the design of the test, and the test result can be used to calibrate the numerical calculation model.

(2) In terms of the determination of the excitation point and measuring point, the excitation point shall avoid the point in which the value of ADDOFV is larger, and the test point determines a point where ADDOFA is larger.

(3) in terms of data processing, it is suggested to determine the vibration peak prevailing vibration peak of blade vibration with the energy relative size of vibration peak to eliminate this effect. The energy transfer rate is introduced, so that the energy transfer rate of the six groups includes all modes based on this method, which gives the first-order modal data of the turbine blades from the measured data.

Reference

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