Dynamic Characteristic Analysis on the Rod Fastening Rotor Bearing System

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Abstract. Modern large-scale gas turbine’s rotor is the main form rod fastening rotor. The contact friction between wheels, the pre-tightening force of the rod and bearing system have an effect on rotor dynamics. According to the structure characteristics of rod fastening rotor, contacting model is established. In the rotor modal, contacting friction, the pre-tightening force of rod and bearing system is included to get natural frequency and the mode. Through the analysis of rotor, with increasing the pre-tightening force, the overall stiffness and natural frequency has increased. The natural frequency is increase fast when the pre-tightening force is small. With the increase of the pre-tightening force, speed of the natural frequency is slow. Bearing system with changes in speed, its oil film dynamic coefficients varieties also have some impact on the inherent characteristics of the rotor.

1. Introduction
Modern large-scale gas turbine rotor is given priority to with rod fastening rotor, and the fault appear more in the rotor of gas turbine [1]. Not only in the gas turbine but also large power equipment system. Most of the faults in the rotating machinery are due to the vibration of the rotor [2]. In order to avoid failure, study to the inherent characteristics of rotor become one of the important content of rotor dynamics. Rotor is mainly composed of uniformly distributed along the circumferential tension rod bolts through the effect of preload will disc and shaft into one combination [3-4]. Because of the structure characteristic of the rotor, it has many other advantages not the rotor, at the same time also brings many difficulties to the rotor dynamic characteristics research. Mainly in: Contact coupling exists between the individual and the roulette wheel and the rod. The rotor structure is different from before, so bring a lot of inconvenience in modeling and calculation of the rotor. For a long time, rotor of the tie rod is studied to the overall model. Calculated as the analytic method, the transfer matrix method and finite element method based on beam element method and so on. The model has a considerable degree of simplification. This will lead to the results of the calculation model and the actual have larger error [5]. As the literature [6~7] put forward mechanical model of the rotor rod, and through dynamic substructure method for theoretical calculation and correlation analysis of its lateral vibration. The literature [8] considers contact of the rotor effect on stress and cracks, however, did not involve contact effect on the stiffness. Study the effects of different roughness contacts face on the critical speed of the rotor rod by through experiments [9~10], but the impact of the rotor preload does not take into account.

The constant improvement of the general finite element software ANSYS analysis level can create
high-fidelity models and consider the effect of friction between the contact wheels. For the above situation, the modal analysis of rotor with pre-tightening force is put forward.

2. The rod rotor model

2.1. Rotor Model
Because of the complex structure of the gas turbine rod fastening rotor, exactly the same three-dimensional structure model is established and calculated is difficult to achieve. Here the use of a simplified rod fastening rotor model, and use the analysis software ANSYS to study rod fastening rotor, get natural frequency and mode. The contact friction between wheel and the pre-tightening force of the rod and bearing system on influence of the rod rotor is researched.

The rotor model as shown in Figure 1, the rod fastening rotor has 8 wheels, and the wheel has a contact interface between coupling wheels. The contact friction between wheels is equivalent to a set of torsional spring. The physical parameters are shown in table 1.

![Figure 1. The rod fastening rotor model.](image)

| Physical parameters | L1/m | D1/m | D2/m | D3/m | H/m |
|---------------------|------|------|------|------|-----|
| Value               | 3.0  | 0.3  | 0.7  | 1.0  | 0.1 |

2.2. ANSYS Model
In the ANSYS, analysis mainly includes three parts: preprocessing module, analysis module solving, and post-processing module. Model of rod fastening rotor is established with a "top-down" approach in the preprocessing module. Type SOLID185 unit is used, which has eight nodes, and along the X, Y, Z directions with three translational degrees of freedom at each node. Rod fastening rotor is different from the overall rotor in the structure, mainly in the following two aspects: (1) rod fastening rotor, constituted by a plurality of wheels, has not been a continuous whole, contact each other between wheel interfaces; (2) contact stiffness of the wheel surfaces determined by pre-tightening force and contact force.

Contact occurs on individual micro convex body during contact roulette and form of contact. This micro convex body in the surface is Gaussian distribution. The contact surface of wheels is pressed together by pre-tightening force. Friction can make consistent lateral displacement between the adjacent wheels, but not make equal angle, and there is a slight elastic deformation on the contact surfaces. In such cases, the behavior of different contact surfaces is simulated in ANSYS. Then control KEYOPT (12) = 4 is chosen.

The pre-tensile load divided into the following steps: (1) rod fastening bolt is meshed (shown in Figure 2); (2) drawing unit is inserted to form a stretching part (shown in Figure 3); (3) a force or displacement is applied on the drawing unit nodes, and then the calculation is carried out. At the end of the shaft, the COMBI214 element is used to simulate the sliding bearing.
2.3. Theoretical Analysis

Rotor Dynamics Equation:

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

(1)

Among them, \([M]\) is mass matrix, \([C]\) is damping matrix, \([K]\) is stiffness matrix, and \([F]\) is external force vector. If there is no external load, (1) is changed into (2).

\[
[M]{\ddot{u}} + [C]{\dot{u}} + [K]{u} = 0
\]  

(2)

Supposing, \(\{u\} = \{\varnothing\}_i e^{\lambda_i t}\), Among them, \(\lambda_i\) is the i-th Eigen values. Natural frequency of the output in ANSYS:

\[
\overline{\lambda_i} = \sigma_i \pm j\omega_i
\]  

(3)

Among them: \(\sigma_i\) is the real part of complex eigen values, \(\omega_i\) is the imaginary part of complex eigen values. \(\omega\) is steady angular frequency, \(\sigma\) is the stability of the rotor. If \(\sigma_i < 0\) indicates that the i-order system is stable, vibration displacement amplitude of the rotor system will be decreased; If \(\sigma_i > 0\), then the i-order system is unstable, Vibration displacement amplitude of the rotor system will be increased. If there is no damping, \(\sigma_i = 0\).

2.4. The results analysis

Different pre-tightening force is applied to each bolt (from 200N to 2000N), there is no stiffness and damping. The first to three order Eigen frequencies of the rotor are pick up. Force and constraints loads as shown in Figure 4. The relationship between each frequency and pre-tightening force is shown in Figure 5, 6, 7. The corresponding vibration mode is as shown in Figure 8, 9, 10.
From the figures it is found that the pre-tightening force has a certain influence on the natural frequency of the rotor; when the pre-tightening force increase, the natural frequency also increase. The increase speed of natural frequency is rapid with smaller force. The decrease speed of natural frequency is rapid with larger force. The reason is that under the action of a load, the earliest contact is the highest part of the disk surface, when the disc surface contact with each other. As the load continues to increase, the other parts are also contact each other. Each micro convex body began to contact. Elastic deformation occurs first. But with the increase of load, plastic deformation occurred (or in the elastic-plastic deformation). Because of the different height of the different micro convex body, different surface height of the micro convex body’s deformation is not the same in the same time. Some micro convex body with smaller height, in the case of large load did not contact. Although load continued to increase, but there will be no obviously change. So there will be a natural frequency rising faster, when the pre-tightening force increase.

When the rotational speed of the rotor (100~2500rad/s) is changed, the bearing’s stiffness and damping are also changed. The rules are shown in Figure 11, 12. The first Eigen frequency, the second Eigen frequency and the third Eigen frequency are also changed, shown in Figure 13, 14, 15.
3. Conclusion
A three-dimensional rod fastening rotor model is created in ANSYS, and finite element analysis is applied for the rod fastening rotor.

When the pre-tightening force is increased, the natural frequency of the rod fastening rotor increases also. When the pre-stressing force is small, the natural frequency increases rapidly. When the increase of pre-tightening force is larger, the growth rate of the natural frequency gradually slows.

With the increase of rotor speed, the bearing stiffness and damping coefficients corresponding decrease, which will cause the decrease of the natural frequency of the rod fastening rotor.

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