Research Article

Vibration Characteristics and Simulation Verification of the Dual-Rotor System for Aeroengines with Rub-Impact Coupling Faults

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To study the rub-impact fault between the dynamic and static parts of the rotor system of aeroengines, the dual-rotor system of a typical aeroengine is introduced and taken as the research object. The analytical kinetic model is established based on the Lagrange equation considering the structural characteristics of the dual-rotor system, the coupling effect of the intermediate bearing, and the rub-impact fault between the high-pressure turbine disc and the casing. The dynamic characteristics of the dual-rotor system under the rub-impact fault are analyzed, and the change rule of the rub-impact shape is obtained. The vibration coupling and transfer among the high-pressure rotor and the low-pressure rotor are revealed. The influence of the unbalanced position and the speed of high and low rotors on the vibration response of the dual rotor is obtained. The sensitivity of the vibration response of the dual rotor at different test points to rub-impact stiffness, clearance, and friction coefficient is compared. The simulation model is established based on the rigid-flexible coupling multibody dynamic simulation platform. The analytical results and simulation results are compared, which have a good consistency. The theoretical research can deepen the understanding of the nature and law of aeroengine rotor operation, expose the possible faults and design defects, greatly improve the development efficiency and quality, reduce repeated physical tests, reduce the development risk and cost, and accelerate the development process. This study can provide a theoretical basis for the monitoring and diagnosis of engine rub-impact faults and provide theoretical and practical reference for the establishment of the vibration fault test and analysis method system.

1. Introduction

As the ‘heart’ of an aircraft, the aeroengine is the decisive factor of aircraft safety, reliability, and operational performance. With the increasing requirements of high speed and the high thrust-weight ratio of aeroengines, the clearance between the rotor and stator decreases, which increases the possibility of a rub-impact fault. The rub-impact fault has become one of the most common faults of the dual-rotor system in aeroengines. It is one of the causes of the complex nonlinear vibration of the dual-rotor system and an important reason for the instability of the rotor system. The rub-impact fault will not only lead to the increase of the rotor-stator clearance, bearing wear, and blade cracks but also may cause the rotor system motion instability and make

the casing deform greatly, and the excessive vibration will seriously affect the working efficiency of the engine. The vibration of the unit will increase dramatically, which will affect the service life of the unit. In serious cases, it will cause permanent bending of the rotor and even cause fatal accidents. It is difficult to find out the cause of abnormal vibration before the machine stops for overhaul. Therefore, it is of great significance to explore the vibration characteristics of the rotor rub-impact, especially the early rub-impact characteristics, and detect these characteristics for avoiding rub-impact faults and secondary faults.

Many theoretical research studies on the aeroengine rotor system have been carried out by scholars at home and abroad. Representative results include the long-term research achievements and contributions of Professor

Yushu Chen and Huabiao Zhang [1–3] of Harbin Institute of Technology, Litang Yan [4] of Beijing University of Aeronautics and Astronautics, Mingfu Liao [5–7] of Northwest Polytechnic University, Qingkai Han [8–10] of Dalian University of Technology, Guihuo Luo [11], and Guo Chen [12] of Nanjing University of Aeronautics and Astronautics. Wang [13] took a dual rotor as the research object, established a dynamic model considering the friction between the rotor and the casing, studied the nonlinear problem of the rub-impact rotor, and used the nonlinear rotor dynamics theory to analyze the model of the rub-impact rotor system and its chaotic motion evolution law. Liu et al. [14] established the dynamic equation of an aeroengine dual-rotor system with a rub-impact fault by using the Lagrange equation, simulated the rub-impact fault through numerical calculation, and proposed the effectiveness and reliability of the method of combining spectrum analysis and wavelet transform to diagnose the rub-impact fault of the rotor and stator. Considering the elastic deformation, contact penetration, and elastic damping support during the collision between the casing and the disc, Wang et al. [15] established the dynamic model of the dynamic and static rub-impact of the dual-rotor system by applying Hertz contact theory and Coulomb model and solved the dynamic response of the dual-rotor system under the rub-impact fault. The dual-rotor system testbed is established to carry out the dynamic and static rub-impact test, and it is proposed that the double-frequency and combined frequency components of the two excitation sources can be taken as the typical characteristics of the aeroengine dynamic and static rub-impact fault. Zhang et al. [16] took the dual-rotor system of an aeroengine as the research object, established the coupled bending-torsion dynamic equation with rub-impact force by considering the force of the intermediate bearing, and calculated the vibration response of the dual-rotor under the rub-impact fault by using numerical methods. The frequency spectrum and bifurcation characteristics of the bending and torsional vibration are analyzed, and experimental research on the bending and torsional vibration is carried out based on the rotor test bench. The results show that the bending and torsional vibration have a similar characteristic frequency, but the torsional vibration characteristic frequency is more obvious, and the torsional vibration signal can be better used for rub-impact fault diagnosis. Ouyang and Yang [17] established the dynamic model of the dual-rotor system of the aeroengine without the intermediate bearing by using a lumped-mass method. The rotor rub-impact fault was simulated, and the vibration response of the rotor system under different rub-impact conditions was obtained. Zhang [18] established the dynamic model of the aeroengine dual-rotor-casing coupling system, solved the critical speed and vibration mode of the system, discussed the influence of support parameters and speed ratio on the critical speed and vibration mode, calculated the rub-impact response of the dual-rotor-casing coupling system, and studied the influence of different rub-impact degrees and unbalance on the dual-rotor rub-impact fault.

Liu [19] established the dynamic model of the dual-rotor system with a rub-impact fault and studied the vibration response of the dual-rotor system under the rub-impact fault. Compared with the unbalanced dynamic characteristics of the dual-rotor system, he found that the rub-impact fault makes the rotor system vibration response appear the combined frequency of rotation frequency, which makes the rotor system vibration appear nonlinear. The influence of the parameters such as the rub-impact clearance on the dynamic characteristics of the dual-rotor system was also studied, and the severity of the rub-impact fault can be judged according to it.

The dual-rotor system of aeroengines has a special structure. For example, the low-pressure rotor is supported by three fulcrums, and the high-pressure rotor and the low-pressure rotor are connected by an intermediate bearing. Although there have been some results on the dynamics of the dual-rotor system, the characteristics of dynamic coupling and vibration transmission of the dual-rotor system for aeroengines have not been studied in detail. According to the engineering observation, the sensitivity of the low-pressure rotor and the high-pressure rotor to the rub-impact is different, and the sensitivity of the rub-impact at different locations is analyzed in [20]. In this paper, the vibration sensitivity of different positions of the rotor with a rub-impact fault is studied by using the method of analysis and multibody dynamics simulation verification. The influence characteristics of rub-impact stiffness, clearance, and friction coefficient on the vibration response of the dual rotor are compared.

2. Establishment of the Dynamic Model for the Dual-Rotor System

Taking a typical aeroengine shown in Figure 1 in [20] as a prototype, a simplified model of the dual-rotor system as shown in Figure 2 is obtained according to the principle of similarity design. The main parts of the main structure of the dual-rotor system are the fan rotor, the low-pressure turbine rotor, the high-pressure rotor, turntable, support, and coupling. The low-pressure rotor is composed of a fan disc and one turbine disc. The total length of the low-pressure rotor is about 1261 mm. It adopts the supporting form of three fulcrums, namely, a 1-1-1 support form. The front fulcrum of the fan and the rear fulcrum of the low-pressure turbine are elastic supports. The low-pressure turbine shaft is slender, and the fan rotor and the low-pressure turbine rotor are connected by a sleeve gear coupling which can transmit axial force and torque. The high-pressure rotor is composed of the compressor and one turbine, in which the support span is 580 mm. The high-pressure compressor and the high-pressure turbine are rigidly connected through the hollow shaft, and the 1-0-1 two-point support mode is adopted. The front support of the high-pressure compressor is the elastic support, and the rear support is the intermediate bearing. The stiffness of each fulcrum is shown in Table 1. In order to simplify the structure, reduce the quality of the rotor system, and reduce the dynamic influence between the high- and low-pressure rotors, the rotor system

FIGURE 1: Structural sketch of a dual-rotor system for a typical aeroengine.

FIGURE 2: The diagram of the simplified dual-rotor model.

TABLE 1: Stiffness of the supports.

K_{b1}	K_{b2}	K_{b3}	K_{b4}	K_{b5}
$9 \times 10^6 \text{ N/m}$	$2.5 \times 10^8 \text{ N/m}$	$3 \times 10^6 \text{ N/m}$	$2.5 \times 10^8 \text{ N/m}$	$3 \times 10^6 \text{ N/m}$

adopts the intermediate bearing design with the high-pressure rotor rear fulcrum as close as possible to the low-pressure turbine rear fulcrum. The high- and low-pressure rotors are coupled through the intermediate bearing. The rotor transmits the load through the fulcrum of another rotor. Therefore, the dynamic characteristics of the dual-rotor system are not only affected by the high- and low-pressure rotors but also have some unique vibration characteristics.

3. Dynamic Model of the Dual Rotor with the Rub-Impact Fault

The analytical equations of multisupport motion of the dual-rotor system are derived utilizing the Lagrange method, and the governing differential equations of the dual-rotor system considering the axial and bending vibration but not torsional vibration are obtained. In the model, the four discs, the fan

shaft, and the high-pressure shaft are considered to be rigid, while the slender shaft of the low-pressure turbine is considered as elastic. The support of the rotor system is considered to be elastic, and the speed of the rotating shaft is considered to be constant. The rubbing fault often occurs between the high-pressure rotor and the casing [21]. In this study, only the rub-impact between the high-pressure turbine disc and the stator casing is considered.

The fixed coordinate system of the system is oxy , and the dynamic equation of the whole system is as follows:

$$M\ddot{\mathbf{q}} + \mathbf{C}\dot{\mathbf{q}} + \mathbf{G}\dot{\mathbf{q}} + \mathbf{K}\mathbf{q} = \mathbf{F}, \quad (1)$$

where $\mathbf{M}$, $\mathbf{C}$, $\mathbf{G}$, $\mathbf{K}$, and $\mathbf{F}$ are the mass matrix, damping matrix, gyro force matrix, stiffness matrix, and generalized force matrix, respectively. $\mathbf{q}$ is the generalized coordinate vector. Among them,

$$\mathbf{q} = \{ y_1 \ z_1 \ \theta_{y1} \ \theta_{z1} \ y_2 \ z_2 \ \theta_{y2} \ \theta_{z2} \ y_3 \ z_3 \ \theta_{y3} \ \theta_{z3} \ y_4 \ z_4 \ \theta_{y4} \ \theta_{z4} \ }, \quad (2)$$

$$\mathbf{F} = \{ F_{1y} \ F_{1z} \ 0 \ 0 \ F_{2y} \ F_{2z} \ 0 \ 0 \ F_{3y} \ F_{3z} \ 0 \ 0 \ F_{4y} + F_{Ry} \ F_{4z} + F_{Rz} \ 0 \ 0 \ },$$

where F_{iy} , F_{iz} ($i = 1 \sim 4$) are the component of unbalanced excitation force in the y -direction and z -direction by four

discs and F_{Ry} , F_{Rz} are the component of rub-impact force in the y -direction and z -direction.

4. Unbalanced Model

The unbalance of the dual-rotor system mainly includes the fan disc unbalance, the low-pressure turbine disc unbalance, the high-pressure compressor disc unbalance, and the high-pressure turbine disc unbalance. The components of the centrifugal force produced by the unbalanced mass of each disc in the y -direction and z -direction are as follows:

$$\begin{cases} F_{iy} = m_{ei}\Omega_i^2 \cos(\Omega_i t + \phi_i) \\ F_{iz} = m_{ei}\Omega_i^2 \sin(\Omega_i t + \phi_i) \end{cases}, \quad i = 1 - 4, \quad (3)$$

where m_{ei} is the unbalanced mass, and the unit is $Kg \cdot m$. ϕ_i is the initial phase angle of the disc centroid, and the unit is rad. Ω_i is the rotational angular velocity of the low- and high-pressure rotors, and the unit is rad/s.

5. Rub-Impact Model

The schematic diagram of the local rub-impact between the rotor and stator is shown in Figure 3. F_y and F_z stand for the normal force and tangential force exerted on the disc by the rubbing plate, respectively. φ denotes the angle of the rubbing plate normal direction and the x -axis, and e is the moving distance of the rotor axis. The deformation of the rubbing plate caused by the rub-impact between the disc and rubbing plate is supposed to be linear. k_c is the radial stiffness of the rubbing plate. f_r and δ are the friction coefficient at the rubbing point and the clearance between the rubbing plate and the disc, respectively. Function H means that when $(z - \delta)$ is less than 0, it is equal to 0, and otherwise, it is equal to 1. The concrete expression is as follows:

$$\begin{cases} F_{Rz} = -k_r(z - \delta)H \\ F_{Ry} = F_{Rz}f_r = -k_r(z - \delta)Hf_r \end{cases} \quad (4)$$

Rayleigh damping is adopted in the damping model, and the structure is as follows:

$$C = \alpha M + \beta K. \quad (5)$$

The Rayleigh damping matrix is a linear combination of the mass matrix and the stiffness matrix, where α and β are damping coefficients, which are related to the modal damping ratio ξ of the structure. The calculation formula is as follows:

$$\begin{aligned} \alpha &= 2\omega_1\omega_2 \frac{(\xi_1\omega_2 - \xi_2\omega_1)}{\omega_2^2 - \omega_1^2}, \\ \beta &= 2 \frac{(\xi_2\omega_2 - \xi_1\omega_1)}{(\omega_2^2 - \omega_1^2)}. \end{aligned} \quad (6)$$

After derivation, other matrices can be easily obtained.

6. Inherent Characteristics of the Dual-Rotor System

According to matrices M , G , and K in formula (1), the modal frequency and modal shape of the rotor can be obtained. The displacement of the rotor system is written as follows:

$$\mathbf{q} = \mathbf{q}_0 e^{i\omega_n t}, \quad (7)$$

where ω_n is the whirl frequency of the rotor system; then, equation (3) can be written as

$$-\omega_n^2 \mathbf{M} \mathbf{q}_0 + i\omega_n \Omega \mathbf{G} \mathbf{q}_0 + \mathbf{K} \mathbf{q}_0 = 0. \quad (8)$$

When calculating the critical speed, the whirl frequency and rotation frequency of the rotor are equal; then, the above formula can be written as

$$(\mathbf{M} - i\mathbf{G})^{-1} \mathbf{K} \mathbf{q}_0 = \omega_n^2 \mathbf{q}_0. \quad (9)$$

ω_n^2 is the eigenvalue of matrix $(\mathbf{M} - i\mathbf{G})^{-1} \mathbf{K}$. The eigenvalue and eigenvector of matrix $(\mathbf{M} - i\mathbf{G})^{-1} \mathbf{K}$ are obtained by MATLAB programming. The square and eigenvector of the eigenvalue are the modal frequency and mode shape of the rotor system. The natural frequencies and vibration modes of the dual-rotor system are shown in Figure 4.

The model parameters of the simplified dual-rotor system are shown in Tables 2–4, in which four equivalent discs are shown in Table 2, the equivalent parameters of the rotating shaft are shown in Table 3, and the distance parameters are shown in Table 4.

It can be seen from Figure 4 that the first and second modes of the low-pressure rotor are mainly the first bending mode of the low-pressure rotor turbine section, the first mode of the high-pressure rotor is the translation mode, and the second mode is the pitching vibration.

7. Analysis Results

7.1. Method of Calculation. In this paper, the Newmark- β method is used to solve the vibration response of the dual-rotor system under the rub-impact fault. Newmark- β method is an integral dynamic numerical analysis method. Different from the analytical method and numerical integration method, it gives the complex calculation task to the computer through simple programming, which is convenient and can achieve good accuracy. The Newmark- β integration method is a kind of step-by-step integration method. It can set the time interval between every two calculation substeps and calculate the response result of the next calculation substep according to the previous calculation results. As long as the parameters are set reasonably, good convergence can be achieved.

FIGURE 3: The dynamic model of fixed-point rubbing. (a) Schematic diagram of rubbing between the casing and blisk. (b) Mechanical model of fixed-point rub-impact.

FIGURE 4: Vibration modes' diagram of the dual-rotor system. (a) First-order vibration (58.8 Hz). (b) Second-order vibration (75.4 Hz).

TABLE 2: Parameters of four equivalent discs.

Disc	h (mm)	D (mm)	d (mm)	m (Kg)	J (Kg·m ²)
D_1	18	330	35	14.0	$J_p = 0.164, J_d = 0.101$
D_2	10	316	48	11.3	$J_p = 0.059, J_d = 0.060$
D_3	35	305	73	20	$J_p = 0.230, J_d = 0.120$
D_4	14	304	62	10	$J_p = 0.102, J_d = 0.050$

TABLE 3: Parameters of the equivalent rotation shaft.

Shaft number	D (mm)	d (mm)	L (mm)
The shaft of the low-pressure compressor	30	0	381
The shaft of the low-pressure turbine	30	0	923
The shaft of the high-pressure rotor	65	40	680

TABLE 4: Dimension parameters of the simplified dual-rotor system.

Parameter name	OD_1	B_2D_2	B_3D_3	D_3B_4	D_4B_4	C_1C_2	D_1C_1	OB_3	B_4B_5	B_2B_5
Numerical value (mm)	127	737	116	459	69	70	138	1186	581	814

FIGURE 5: The arrangement of vibration measuring points of the dual-rotor system.

TABLE 5: Parameter setting of the rub-impact model.

Parameter	I	II	III	IV
Rub-impact stiffness	1×10^6 N/m	7×10^6 N/m	3×10^7 N/m	0
Friction coefficient	0.4	0.4	0.6	0
Rub-impact clearance	1×10^{-7} m	1×10^{-6} m	1×10^{-7} m	0
LPC unbalance	500 g mm	500 g mm	500 g mm	500 g mm
HPC unbalance	500 g mm	500 g mm	500 g mm	500 g mm

To study the vibration response of the dual-rotor system under the rub-impact fault, four measuring points 1–4 are selected on the low-pressure rotor, among which No. 1 measuring point is 1# fulcrum, No. 2 measuring point is the center of mass of the LPC (the fan disc), No. 3 measuring point is the center of mass of the LPT (the low-pressure turbine disc), and No. 4 measuring point is 6# fulcrum. Four measuring points 5–8 are selected on the outer rotor, among which No. 5 measuring point is 4# fulcrum, No. 6 measuring point is the center of mass of the HPC (the high-pressure compressor), No. 7 measuring point is the center of mass of the HPT (the high-pressure turbine disc), and No. 8 measuring point is 5# intermediary support. The arrangement of vibration measuring points of the dual-rotor system is shown in Figure 5.

7.2. Vibration Characteristics of the Dual Rotor with HPT Rubbing

7.2.1. Influence of Rub-Impact Stiffness on the Vibration Response. Based on the analysis method, the vibration response of the dual-rotor system under the rub-impact fault is analyzed by the Newmark- β method. Speed setting: low-pressure speed $N1 = 170$ Hz; high-pressure speed $N2 = 221.67$ Hz. As shown in Table 5, the vibration characteristics of the dual-rotor system under four different parameter settings are analyzed, and the vibration time domain, frequency spectrum, and axis orbit of eight measuring points are obtained. The vibration response of the dual-rotor system under three different rub-impact parameter settings is shown in Figures 6–8.

Based on the analysis of the unbalance vibration characteristics of the dual-rotor system in [22], it can be seen that only high- and low-pressure rotating frequencies $N1$ and $N2$ appear in the unbalanced vibration response of the dual-rotor system. In this paper, the vibration trajectories of the low-pressure and high-pressure rotors are obtained as shown in Figure 9, under condition IV in Table 5.

Comparing the vibration results in Figures 6–9 of each measuring point with or without rub-impact, the following can be seen:

- (1) Rub-impact fault makes the vibration response of the dual-rotor system have nonlinear characteristics. The dynamic response of the rotor also contains the combined and division components of the speed frequency, and the occurrence of the rub-impact fault makes the axis track no longer a regular circle.
- (2) It can be seen from the time-domain diagram that the occurrence of the rub-impact fault increases the amplitude of the rotational frequency of the dual-rotor system.
- (3) Compared with Figures 7–9, it can be seen that, with the increase of rub-impact stiffness, the amplitude of rub-impact spectrum components increases, and the component of the spectrum becomes more complex. In the case of the slight rub-impact, there are no other components in the vibration spectrum of each measuring point except the rotational frequency and $N2-N1$. With the increase of rub-impact degree, $(2N2-N1)$, $1/2N2$, $2N1$, $N1 + N2$, $2N2$, etc., appear in the frequency components.

FIGURE 6: Continued.

FIGURE 6: The time-domain diagram, spectrum, and axle center orbit of the test point under the Case I.

Under the condition of no rub-impact and three different rub-impact degrees, the amplitude of speed frequency in each measuring point of the dual-rotor system is compared and analyzed, as shown in Figure 10. It can be seen that the vibration amplitude of every test point with the rub-impact fault is greater than that with an unbalanced condition, and with the increase of the rub-impact degree, the vibration amplitude of each measuring point increases. The

No. 5 (4# fulcrum) and No. 6 (HPC) measuring points on the high-pressure rotor are the least affected by the high-pressure turbine disc rub-impact and are mainly affected by the high-pressure compressor disc unbalance. Especially, the position of the No. 6 measuring point is the unbalanced position of the high-pressure rotor, and its axis orbit is relatively regular. Compared with Figures 10(a) and 10(b), it can be seen that the vibration of the low-pressure turbine

FIGURE 7: Continued.

FIGURE 7: The time-domain diagram, spectrum, and axle center orbit of the test point under Case II.

disc position is least affected by the high-pressure turbine disc rubbing. The transfer law of speed frequency of the dual-rotor vibration on high- and low-pressure rotors remains unchanged. The vibration magnitude ordering of $N1$ was $8 > 4 > 7 > 3 > 5 > 1 > 2 > 6$, and that of $N2$ was $5 > 6 > 4 > 8 > 3 > 1 > 2 > 7$.

7.2.2. Influence of Rub-Impact Parameters on the Vibration Response of the Dual-Rotor System. Rub-impact stiffness, rub-impact clearance, and friction coefficient are three

important parameters in the rub-impact fault model. In this section, the influence of the three typical rub-impact parameters on the vibration response of the dual-rotor system is studied with three cases shown in Table 6. The vibration amplitude of fundamental frequency ($N1$ and $N2$) and 2 times of the high-pressure rotational frequency ($2N2$) are obtained in three cases. The influence of different rub-impact stiffness, rub-impact clearances, and rub-impact friction coefficients is compared. The representative results are shown in Figures 11–13 which

(a)

FIGURE 8: Continued.

FIGURE 8: The time-domain diagram, spectrum, and axle center orbit of the test point under Case III.

are the amplitudes of N_1 , N_2 , and $2N_2$ of each measuring point with the change of the rub-impact parameters.

From Figure 11, we can see that under the three different parameter settings, the vibration amplitude of N_1 at each measuring point has a small change, among which the variation of measuring points 2 and 6 is the smallest.

From Figure 12, we can see that under the three different parameter settings, the vibration amplitude of N_2 at each measuring point changes little, among which the variation of

measuring points 5 and 6 of the high-pressure rotor is the largest.

It can be seen from Figure 13 that the amplitude of $2N_2$ at each measuring point changes slightly under the setting of I and III rubbing parameters, while the amplitude of $2N_2$ at each measuring point increases under the setting of the rubbing parameter in case II.

From Figures 11–13, it can be seen that the amplitude changes of N_1 and N_2 are less affected by friction coefficient

FIGURE 9: Axis orbit diagram of the high- and low-pressure rotors. (a) The low-pressure rotor axis orbit diagram. (b) The high-pressure rotor axis orbit diagram.

and rub-impact clearance and mainly affected by the unbalance of high and low rotors. The influence of friction coefficient on the amplitude of $2N2$ is greater than that of rub-impact clearance.

Among the three rub-impact parameters, the rub-impact stiffness has the greatest influence on the vibration response of each measuring point of the dual-rotor system, while the friction coefficient and rub-impact clearance have little influence on the vibration response of each measuring point of the dual-rotor system.

7.3. Influence of Rotating Speed on the Vibration Response of the Dual-Rotor System. To study the influence of rotating speed on the vibration of the dual-rotor system under the rub-impact fault, in this section, without changing the unbalance of the rotor system, the vibration response of the dual-rotor system with the low-pressure speed $N1 = 50$ Hz, high-pressure speed $N2 = 60$ Hz, and rub-impact stiffness 7×10^6 N/m is studied. The vibration time domain and frequency spectrum axis trajectory of each measuring point are obtained, as shown in Figure 13.

In view of the influence of rotating speed on the vibration characteristics of the double-rotor system under the rub-impact fault, the vibration response of the double-rotor system with low-pressure speed, high-pressure speed $N2 = 60$ Hz, and rub-impact stiffness 7×10^6 N/m is studied without changing the rotor system unbalance. The vibration time domain and frequency spectrum of each measuring point are obtained, as shown in Figure 14.

By comparing the results of Figure 14 with Figure 8, it is found that the vibration response characteristics of the dual-rotor system caused by the rub-impact fault are not completely identical at two speeds. The frequency components $N1$, $N2$, and $N2-N1$ appear in the vibration spectrum in both speed conditions. In Figure 13, in addition to the above frequency components, there are $2N2-N1$, $2N2$, and $3N2$ in the spectrum of each measuring point. Under the case of rotating speed $N1 = 50$ Hz and $N2 = 60$ Hz, high-voltage frequency conversion $N2$ is dominant in the vibration spectrum of each measuring point, and there is no obvious

peak in the time-domain diagram. The orbit of the shaft center is enveloped ellipse shape, which is more regular than the results in Figure 8.

8. Simulation Verification

8.1. Establishment of a Rigid-Flexible Coupling Multibody Dynamic Model for the Dual-Rotor System. ADAMS (automatic dynamic analysis of mechanical systems) is a professional product with rich industry application experience. It is the only dynamic simulation software verified by a large number of practical projects [23–28]. The dynamic modeling, analysis, and vibration prediction of the aeroengine rotor system based on rigid-flexible coupling multibody dynamic system simulation technology can meet the requirements of characteristics' analysis of the hierarchical system and rigid-flexible coupling structure and vibration prediction of the rotor system in the process of new rotor design. In particular, the simulation analysis of typical vibration problems such as unbalance and rub-impact of the rotor system can be realized effectively.

The 'impact' is used to calculate the rub-impact force in ADAMS, which is based on the impact function to calculate the contact force between the two objects. The contact force is composed of two parts: the first part is the elastic force due to the mutual penetration between the two objects, and the other part is the damping force produced by relative speed.

The rub-impact mechanics expression is

$$F = \begin{cases} 0, & x \geq \delta_0 \\ k(\delta_0 - x)^e - NC_{\max} \text{step}(x, \delta_0 - d, 1, \delta_0, 0) & x \ll \delta_0 \end{cases} \quad (10)$$

Among them, k is the stiffness coefficient, δ_0 is the initial distance between the two contact objects, x is the actual distance between the two contact objects in the rubbing process, $C_{\max}$ is the maximum damping coefficient used to characterize the collision energy loss, N is the rotating shaft speed (high- and low-pressure rotor speeds are $N2$ and $N1$, respectively), e is the rub-impact index used to represent the nonlinear degree of the material, and d is the penetration depth.

FIGURE 10: The amplitude of vibration frequencies $N1$ and $N2$ at each measuring point of the dual rotor is affected by rub-impact. (a) $N1$. (b) $N2$.

TABLE 6: Parameter setting of the rub-impact model.

Parameter	I	II	III
Rub-impact stiffness	3×10^6 N/m	3×10^6 N/m	3×10^6 N/m
Friction coefficient	0.4	0.4	0.6
Rubbing clearance	1×10^{-7} m	1×10^{-6} m	1×10^{-7} m
LPC unbalance	500 g mm	500 g mm	500 g mm
HPC unbalance	500 g mm	500 g mm	500 g mm

FIGURE 11: The vibration amplitude of $N1$ with different rub-impact parameters.

The low-pressure turbine shaft is considered as a flexible shaft, and the elastic support is simulated by a squirrel-cage elastic structure, and other parts are assumed to be rigid. When modeling the dual-rotor system, in order to truly

reflect the connection state between various parts of the model, the parts originally fixed by bolts are connected with fixed pairs. The bearing is defined by stiffness and damping, simulated by the “bushing” element, and the coupling is

FIGURE 12: The vibration amplitude of $N2$ with different rub-impact parameters. FIGURE 13: The vibration amplitude of $2N2$ with different rub-impact parameters.

simplified as a short shaft structure. In order to ensure the steady increase of the speed, the step function is usually used to set the speed.

The dual-rotor system unbalance is simulated by adding a mass block on the low-pressure compressor disc and the high-pressure compressor disc. A rub-impact block is set at a certain clearance from the high-pressure turbine disc (HPT). The fixed pair is used to define the rub-impact block as the stator. The contact adjustment clearance and rubbing parameters are set between the high-pressure turbine disc and the rubbing block to simulate different rubbing conditions. Finally, the rigid-flexible coupling multibody dynamic model of the dual-rotor system with rub-impact is shown in

Figure 15. Low-pressure speed $N1 = 50$ Hz and high-pressure speed $N2 = 60$ Hz. The rigid-flexible coupling simulation model of the dual-rotor system is shown in Figure 15.

To verify the correctness of the model and the simulation accuracy, the first two natural frequencies of the dual-rotor system are obtained by ADAMS in Figure 16. And the results are compared with the analytical results based on the dynamic equation. The results show that the first two modes are consistent. The first and second modes of the low-pressure rotor are mainly the first-order bending of the low-pressure turbine shaft. The first mode of the high-pressure rotor is translational, and the second is the pitching mode. The first natural frequency difference ratio is 5%, and the

(a)

FIGURE 14: Continued.

FIGURE 14: The time-domain diagram, spectrum, and axle center orbit of the dual-rotor system.

second natural frequency difference ratio is 4.5%. The rationality of the simulation model is verified.

8.2. Parameter Setting of the Simulation. Based on the multibody dynamics model of the dual-rotor system with the rub-impact fault in Figure 15, the vibration response of the dual-rotor system is simulated with five different rub-impact states as shown in Table 7.

8.3. Simulation Results

8.3.1. Influence of Different Rubbing Stiffness on the Vibration. As shown in Table 4, the vibration of the dual-rotor system under three different impact stiffness (1–3) is simulated and analyzed, and the vibration time domain, frequency spectrum, and high- and low-pressure rotor center orbit of each measuring point under three conditions are obtained, as shown in Figures 17–19.

FIGURE 15: The multibody dynamic model of the dual-rotor system based on ADAMS.

FIGURE 16: The first two modes and natural frequencies of the dual-rotor system based on ADAMS. (a) The first mode shape (53 Hz). (b) The second mode shape (79 Hz).

TABLE 7: Parameter setting of the simulation model based on ADAMS.

Parameter	Case 1	Case 2	Case 3	Case 4	Case 5
Rub-impact stiffness	1×10^6 N/m	1×10^7 N/m	1×10^8 N/m	1×10^7 N/m	1×10^7 N/m
Rub-impact damping	100 Ns/m	1000 Ns/m	10000 Ns/m	1000 Ns/m	1000 Ns/m
Force exponent	1.5	1.5	1.5	1.5	1.5
Penetration depth	0.1 mm	0.1 mm	0.1 mm	0.1 mm	0.1 mm
Rubbing clearance	1×10^{-7} m	1×10^{-7} m	1×10^{-6} m	1×10^{-7} m	1×10^{-7} m
LPC unbalance	500 g mm	500 g mm	500 g mm	1000 g mm	500 g mm
HPC unbalance	500 g mm	500 g mm	500 g mm	500 g mm	1000 g mm

Analyzing the results shown in Figures 17–19, through the vibration time-domain diagram of each measuring point of the dual-rotor system, it is found that, with the increase of the rub-impact stiffness, the vibration amplitude of each measuring point increases. Through the frequency spectrum, it can be seen that, with the increase of the rub-impact stiffness, the vibration spectrum components of each measuring point become more complex, and there are multiple combination and division frequency components, such as $(2N_2-N_1)/2$, $2N_2-N_1$, $(4N_1+N_2)/2$, and $(2N_1-N_2)/2$, and N_2 and $2N_1-N_2$ are the main frequencies. Through the analysis of the axis trajectory diagram, it is found that, as the degree of rub-impact increases, the axis trajectory of each measuring point becomes messy, no longer a regular circle or ellipse, and sharp corners appear in the rubbing direction. The

simulation results are consistent with the theoretical analysis results.

8.3.2. Influence of Unbalance of the Dual-Rotor System on the Rub-Impact Vibration. The rub-impact vibration characteristics of the dual-rotor system with 4 and 5 parameters in Table 4 are analyzed, and the influence of unbalance on the rub-impact vibration is studied. The vibration response of the dual-rotor system is obtained, as shown in Figures 20 and 21, respectively.

Comparing Figure 20 with Figure 21, it can be found that the influence of the fan disc unbalance on the rub-impact response of the dual-rotor system is greater than that of the high-pressure compressor disc unbalance. The increase of fan disc unbalance has a great influence on the vibration

FIGURE 17: The vibration of the dual-rotor system with slight rubbing (case 1). (a) Time-domain diagram of each measuring point. (b) Spectrum of each measuring point. (c) Axis orbit of the low-pressure rotor. (d) Axis orbit of the high-pressure rotor.

FIGURE 18: Continued.

FIGURE 18: The vibration of the dual-rotor system with less severe rubbing (case 2). (a) Time-domain diagram of each measuring point. (b) Spectrum of each measuring point. (c) Axis orbit of the low-pressure rotor. (d) Axis orbit of the high-pressure rotor.

FIGURE 19: The vibration of the dual-rotor system with severe rubbing (case 3). (a) Time-domain diagram of each measuring point. (b) Spectrum of each measuring point. (c) Axis orbit of the low-pressure rotor. (d) Axis orbit of the high-pressure rotor.

response of the low-pressure rotor, but has little influence on the vibration response of the high-pressure rotor.

The amplitude of frequency N2 in the spectrum of each measuring point of the low-pressure rotor increases with the increase of unbalance, and the impact of LPC unbalance is more obvious than that of HPC. It can be seen from the

analysis that rotor unbalance will aggravate the vibration of the dual-rotor system caused by the rub-impact fault, and the rub-impact fault should be avoided as far as possible in design, assembly, and processing.

The application of simulation technology in the aviation field can improve the vibration prediction and analysis

FIGURE 20: The influence of LPC unbalance on the vibration response of the dual-rotor system with the rub-impact fault (Case 4). (a) Time-domain and frequency spectrum of the low-pressure rotor at measuring points 1–4. (b) Time-domain and frequency spectrum of the high-pressure rotor at measuring points 5–8.

FIGURE 21: Continued.

FIGURE 21: The influence of HPC unbalance on the vibration response of the dual-rotor system with the rub-impact fault (Case 5). (a) Time-domain and frequency spectrum of the low-pressure rotor at measuring points 1–4. (b) Time-domain and frequency spectrum of the high-pressure rotor at measuring points 5–8.

technology of a complex aeroengine rotor system. It can improve the stability and vibration control ability of the aeroengine rotor system, as well as the ability of dynamic analysis and design of the aeroengine rotor system.

Comparing the vibration response of the dual-rotor system under the rub-impact fault in this paper with the results in [29–32], it can be seen that the double frequency and combined frequency of the high- and low-pressure rotors appear in the rub-impact vibration response of the dual-rotor system, such as $2N_2-N_1$, $2N_1-N_2$, N_2-N_1 , $2N_1$, $2N_2$, $N_1 N_2$, and N_2-N_1 . Moreover, with the difference in rotational speed, rub-impact stiffness, and rub-impact position, the performance of each component is obviously different. In addition, the dual-rotor system model applied in this paper also has the frequency division $N_2/2$ of the fundamental frequency and the frequency division components of the combined frequency, such as $(2N_2-N_1)/2$, $(2N_1-N_2)/2$, and $(4N_1 N_2)/2$.

9. Conclusions

The dynamic modeling, analysis, and vibration prediction of the aeroengine rotor system based on the rigid-flexible coupling multibody dynamic system technology can meet the requirements of characteristics' analysis of the hierarchical system and rigid-flexible coupling structure and vibration prediction of the rotor system in the process of new rotor design. In particular, the simulation analysis of typical vibration problems such as unbalance and rub-impact of the rotor system can be realized effectively. In this paper, the vibration of the dual rotor with the rub-impact fault is studied by using the method of analysis and multibody dynamics simulation verification, and conclusions are obtained as follows:

- (1) The low-pressure rotor rotation frequency (N_1) and the high-pressure rotor rotation frequency (N_2) are the only frequencies of the unbalanced response of the dual-rotor system. The axis orbit of each measuring point is a regular circle.
- (2) The rub-impact between the high-pressure turbine disc and the stator makes the vibration response of the dual-rotor system have nonlinear characteristics. The dynamic response of the rotor also contains the combined and division components of the speed frequency, and the occurrence of the rub-impact fault makes the axis track no longer a regular circle. The occurrence of the rub-impact fault increases the amplitude of the rotational frequency of the dual-rotor system.
- (3) With the increase of the rub-impact stiffness, the amplitude of the rub-impact spectrum components increases, and the component of the spectrum becomes more complex.
- (4) The transfer law of rotational frequencies (N_1 and N_2) of the dual-rotor vibration on high- and low-pressure rotors remains unchanged.
- (5) In the dual-rotor vibration caused by unbalance and rub-impact faults, the amplitude of N_1 decreases along the transmission paths of low-pressure rotor 4-3-1-2 and high-pressure rotor 8-7-5-6. The amplitude of N_2 decreases along the transmission paths of low-pressure rotor 4-3-1-2 and high-pressure rotor 5-6-8-7.
- (6) The vibration response characteristics of the dual-rotor system caused by the fault are not completely identical at two speeds. The frequency components N_1 , N_2 , and N_2-N_1 appear in the vibration spectrum in both speed conditions.
- (7) Among the three parameters of the rub-impact fault, the stiffness has the greatest influence on the vibration response of each measuring point of the dual-rotor system, while the friction coefficient and clearance have little influence on the vibration response of each measuring point of the dual-rotor system.
- (8) The influence of the fan disc unbalance on the rub-impact response of the dual-rotor system is greater

than that of the high-pressure compressor disc unbalance. The increase of fan disc unbalance has a great influence on the vibration response of the low-pressure rotor, but has little influence on the vibration response of the high-pressure rotor. The No. 5 (4# fulcrum) and No. 6 (HPC) measuring points on the high-pressure rotor are the least affected by the high-pressure turbine disc rub-impact and are mainly affected by the high-pressure compressor disc unbalance.

Data Availability

The data used to support the findings of this study are included within the article.

Conflicts of Interest

The authors declare no conflicts of interest.

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