Research Article

Transient Response of a Simulated Aeroengine with a Fusing Structure during a Fan-Blade Out Event

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Received 29 May 2021; Revised 17 August 2021; Accepted 16 September 2021; Published 11 October 2021

Academic Editor: Juan Du

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A fan-blade out (FBO) event may cause complex vibrations in an aeroengine. A fusing structure protects the structural integrity of the whole aeroengine after an extreme accident, such as a blade-loss event. In this paper, we investigate not only the transient and steady responses of a simulated aeroengine model with a fusing structure after an FBO event but also the changes made to the model because of the fusing structure. Our simulated model of an aeroengine includes two rotors, bearings, and a casing. The results for the dynamic response of the simulated model show that the fusing structure can reduce the steady-state response and the impact load on the support bearings in the rotor system. The rubbing impact between the blades and casing was accounted for. A fast method for calculating the response of fused structures was developed, which may be useful when designing the stiffness of the fusing structure.

1. Introduction

Aeroengines have high safety and reliability requirements [1]. If there is an impact with an external object, the fan blades can fail and be ejected from the engine [2]. This type of aeroengine accident [3–5] is dangerous and called a fan-blade out (FBO). An FBO can induce a huge unbalanced load on the rotation components, which results in a transient response of the engine. Different countries have their own regulations for FBO events. The United States Federal Aviation Administration (FAA) [6] stipulates that in an FBO accident, the blade must lose more than 85% of its mass, a turbofan engine should be free of any noncontainment incidents or fires, and its integrity must be guaranteed over the duration of the longest flight. The aviation authorities in Europe [7] and China have similar standards.

In an FBO event, the lost blade may cause structural damage to the aeroengine, as it may crash into support structures. If the blade is not contained, the aircraft could be endangered by the damage caused [8, 9]. Moreover, an FBO event can also lead to the failure of the aeroengine due to the unbalanced response of the rotation system that

is excited by the loss blade. This imbalance increases the eccentricity of the rotation components, which may lead to rubbing [10–12], unbalanced vibration of the aeroengine rotors [13, 14], and destruction during deceleration [15].

Researchers have enhanced the ability of the casing to contain the lost mass and the vibration response of the rotor system in the transient and steady-state phases. Analyses of the response of an aeroengine suffering a fan-blade lost event are mainly based on experimental tests. However, these use very expensive equipment, which is destroyed by the test.

Thus, numerical simulations and models are increasingly used to predict the responses of aeroengines during an FBO event, including the transient response. Kalinowski et al. [16] established a model of a single-disk rotor suffering a load due to a lost blade. The rotation speed and excitation amplitude were found to be the main factors influencing the vibration response. Raffa and Vatta [17] established a single-disk rotor model using the transfer matrix method to simulate its motion and to understand the relation between an unbalanced load and the vibration response. Sinha [18] showed that a simulation model can clearly establish the dynamic characteristics of the rotor response caused

by sudden unbalanced loads. Li et al. [19] developed a dynamic finite element model with a rotor and blades. It accounted for damping and the stiffness of the support. The axis was a rotating beam element. Their results show that the stiffness of the supporting structure has an obvious influence on the system response. Rao et al. [20] studied the vibration of a rotor by two different methods and found that the finite element method was better for calculating the transient vibration of a rotating structure.

Accordingly, methods have been developed to analyze the response of a rotor system after an FBO event. However, in an aeroengine, there is coupling between the rotors and the casing, so that the rotor alone may not fully represent the dynamic characteristics of the entire system. Thus, it may be better to use a simulation model including the rotor and the casing [21, 22]. Chen [23] analyzed the vibrations of a simulated aeroengine with a finite element model. It took into account the nonlinearity of the support bearing and a squeeze film damper. The results were more accurate because it was a whole machine model. Bonello and Minh Hai [24] established a finite element model of an entire aeroengine using a beam element. However, further research on the sudden occurrence of an unbalanced load is still needed.

A high impact load on the bearings, such as those near the fan, can damage the support structure. Thus, fusing structures [25, 26] were established to reduce the high impact load. During an FBO event, the fusing structure can reduce the reaction force to ensure that the aeroengine is not further damaged during its deceleration. Similar structures are fusible supporting structures [27] and variable-stiffness fusing structures [28]. Many scholars have focused on a simple rotation mode, rather than the structure of an actual event. Although there are several 3D finite element models for simulating the response of the fusing structure, the method is too time-consuming and not suitable for estimating the distribution of the support stiffness during design.

We used a finite element model of an aeroengine to analyze the coupling within the whole system, its transient response, and the reaction force due to the sudden imbalance caused by an FBO event. A fusing structure was added to the supporting structure to reduce the impact load. Moreover, the influence of the rub impact between the fan blades and the casing was also considered. We have developed a fast method to simulate the transient response of the whole system, including a fusing structure, during an FBO accident. This can provide support for estimating the distribution of the support stiffness in the design stage.

2. Finite Element Modeling of a Simulated Engine System

Figure 1 is a model of a common type of aeroengine. The support structure of the low-pressure rotation has a 0-2-1 support form, whereas the high-pressure rotation system has a 1-0-1 support form. An intermediate bearing, which is the main reason for the coupling between the high- and low-pressure rotors, was placed between the rotors.

The simulated aeroengine model includes rotating disks, bearings, the spin axle, and the casing. The length of the casing was 2 m, and its thickness was 25 mm. Tables 1 and 2 give the dimensions of the shafts in the low- and high-pressure rotors, respectively. Table 3 lists the parameters of the rotating disks.

The model has beam elements, which can represent the gyroscopic moment, rotational inertia, and shear deformation. The model does not include lateral bending vibration, torsional vibration, or axial vibration of the rotation system. The stiffness of the support bearings is kept constant during the motion as they are considered to be a spring unit.

2.1. Response of the Simulated Aeroengine System under a Sudden Unbalanced Load. The finite element model of the low-pressure rotor is shown in Figure 2. The rotor system consists of disks and a spin axle. The equation of motion for a rigid disk is

$$\begin{cases} [M]\ddot{q}_y + [G]\dot{q}_z = Q_y, \\ [M]\ddot{q}_z - [G]\dot{q}_y = Q_z, \end{cases} \quad (1)$$

where Q_y and Q_z are generalized forces, and q_y and q_z are generalized displacements. The equation of motion for an elastic shaft is

$$\begin{cases} [M_T + M_R]\ddot{q}_y + [G]\dot{q}_z + [K]q_y = Q_y, \\ [M_T + M_R]\ddot{q}_z - [G]\dot{q}_y + [K]q_z = Q_z, \end{cases} \quad (2)$$

where $[M_T]$ is the translational inertia matrix, $[M_R]$ is the rotational inertia matrix, $[G]$ is the rotation matrix, and $[K]$ is the stiffness matrix.

The inertia matrix $[M]$ can be represented as follows:

$$= \begin{bmatrix} [m_{11}]^{(1)} + [M_d]^{(1)} & [m_{12}]^{(1)} & & & \\ [m_{21}]^{(1)} & [m_{22}]^{(1)} + [m_{11}]^{(2)} & [m_{12}]^{(2)} & \cdots & \\ & [m_{21}]^{(2)} & [m_{22}]^{(2)} + [m_{11}]^{(3)} & \ddots & \\ & \vdots & & \ddots & \\ \cdots & [m_{22}]^{(16)} + [m_{11}]^{(17)} & [m_{12}]^{(17)} & & \\ & [m_{21}]^{(17)} & [m_{22}]^{(17)} & & \end{bmatrix}. \quad (3)$$

FIGURE 1: Schematic of engine system model.

TABLE 1: Dimensions of shafts in the low-pressure rotor.

Shaft number	Length (mm)	Internal diameter (mm)	External diameter (mm)
L1	300	70	80
L2	300	70	80
L3	100	35	40
L4	200	35	40
L5	600	35	40
L6	100	35	40
L7	220	35	40
L8	180	35	40

TABLE 2: Dimensions of shafts in the high-pressure rotor.

Shaft number	Length (mm)	Internal diameter (mm)	External diameter (mm)
L4	200	70	80
L5	600	70	80
L6	100	70	80

The boundary conditions also need to be considered. The support bearing introduces supporting stiffness and damping into the finite element model. Linear spring ele-

ments with a stiffness of 5×10^7 N/m and a damping of 250 Ns/m were used to simulate the support bearings.

Newmark-beta method is a common and effective simulation method, which is often used to calculate structural dynamics issues. This method has good accuracy and stability. When the beta value is greater than or equal to 0.25, the Newmark-beta method is unconditionally stable. The Newmark-beta method was used to analyze the rotor system response by the following formulas:

$$\begin{aligned}
 K_e &= K + a_0 M + a_1 G, \\
 F_e &= F + (a_0 q_{n-1} + a_2 \dot{q}_{n-1} + a_3 \ddot{q}_{n-1})M + (a_1 q_{n-1} + a_4 \dot{q}_{n-1} + a_5 \ddot{q}_{n-1})G, \\
 q_n &= \frac{K_e}{F_e}, \\
 \ddot{q}_n &= a_0 (q_n - q_{n-1}) - a_2 \dot{q}_{n-1} - a_3 \ddot{q}_{n-1}, \\
 \dot{q}_n &= \dot{q}_{n-1} + a_6 \dot{q}_{n-1} + a_7 \ddot{q}_n
 \end{aligned} \tag{4}$$

in which

$$\begin{aligned}
 a_0 &= \frac{1}{\beta \Delta t^2}, a_1 = \frac{\alpha}{\beta \Delta t}, a_2 = \frac{1}{\beta \Delta t}, a_3 = \frac{1}{2\beta} - 1, \\
 a_4 &= \frac{\alpha}{\beta} - 1, a_5 = \frac{\Delta t}{2} \left(\frac{\alpha}{\beta} - 2 \right), a_6 = \Delta t (1 - \alpha), a_7 = \Delta t \alpha \\
 \alpha &= \frac{1}{2}, \beta = \frac{1}{4}.
 \end{aligned} \tag{5}$$

TABLE 3: Parameters of rotating disks.

Rotating disks	Mass (kg)	Diameter of moment of inertia ($\text{kg}\cdot\text{m}^2$)	Polar moment of inertia ($\text{kg}\cdot\text{m}^2$)
Low-pressure compressor	120	4	8
High-pressure compressor	30	1	2
High-pressure turbine	30	1	2
Low-pressure turbine	60	2	4

FIGURE 2: Finite element model of the low-pressure rotor.

To verify the analysis method, an FBO event was simulated for a system with only a low-pressure rotor under conditions similar to those described in the literature [29]. R_u is the impact coefficient. It is used to describe the multiple relations between the responses of the compressor disk in the transient and steady-state phases.

We get the simulation results through calculation. In the comparison with the literature [29], we compare our results with both the simulation results and the experimental results. In comparison with the simulation results, the peak transient response was 64.2 mm, the amplitude of the steady response was 30.1 mm, and R_u was 2.13. Peng et al. [29] carried out blade-loss simulation results for the same structure, finding that the peak of the transient response was 63.5 mm, the amplitude of the steady response was 27.8 mm, and R_u was 2.28. Compared with the results in literature, the errors of the responses of the compressor disk in the transient and steady-state phases were 1.1% and 8.27%, respectively. The impact coefficient error was 6.58%.

In comparison with the blade-loss tests, the peak transient response was $838.79 \mu\text{m}$, the amplitude of the steady response was $334.10 \mu\text{m}$, and R_u was 2.5. Peng et al. [29] carried out blade-loss tests for the same structure, finding that the peak of the transient response was $780.24 \mu\text{m}$, the amplitude of the steady response was $360.98 \mu\text{m}$, and R_u was 2.16. Compared with the results in literature, the errors of the responses of the compressor disk in the transient and steady-state phases were 8.02% and 7.4%, respectively. The impact coefficient error was 15.7%.

To simulate the dynamic vibration response of the entire aeroengine after an FBO event, it is necessary to develop a dynamic finite element model with a casing and mounting structure. The model considers the effect of the missing blade and the reaction force on the support bearing. It has low- and high-pressure rotors, bearings, a casing, and a support structure. The high-pressure shaft has 17 elements and 18 nodes, as shown in Figure 3.

The casing can be regarded as a stationary shaft, so the method used is like that for a rotation shaft. The finite element model for the whole engine can be obtained by combining the finite element models of the major components, as shown in Figure 4.

The boundary conditions need to be considered. Linear spring elements with a stiffness of $5 \times 10^7 \text{ N/m}$ and a damping of 250 Ns/m were used for the support bearings. The intermediate bearing was simulated as described in the literature [30]. The rotation speeds of the low- and high-pressure rotors were set to 4040 and 6060 rev/min, respectively. The mass center of the lost mass was 0.5 m away from the rotating axis of the rotor, and the lost mass was 3 kg. The FBO event would happen since the system begins to rotate for 2 seconds.

The vibration response is shown in Figure 5. After the FBO event, the amplitude of the response increased sharply. When the amplitude reached a peak, the response fell and entered a steady state. The motion trace of the low-pressure compressor disk is elliptical. The aeroengine suffered a huge impact load when the FBO happened, causing a strong dynamic response, which makes it difficult to ensure the integrity of the whole system.

In the transient stage, the peak of the vibration response is much higher than the amplitude of the response in the steady state. Because of the coupling of the two rotors and the casing, it is necessary to consider the whole simulated model system as the target. Moreover, the dynamic characteristics of the simulated aeroengine model are more complex than those of the rotor alone. Figure 5(c) shows that the frequency component of the vibration is much more complex than that of the rotor system, due to the coupling between the rotor and the casing. The frequency component shows the dynamic characteristics of the simulated aeroengine model. Figure 6 shows the vibration response for the compressor and turbine disks.

The vibration response at the high-pressure turbine disk is lower than that at the low-pressure compressor disk, due to the coupling between the high- and low-pressure rotors. Because of the coupling, the vibration response may lead to a failure between the casings and the turbine blades. Table 4 shows the dynamic response.

After an FBO event, the support bearings suffered high reaction forces, which climbed quickly at first and then decreased to a steady state. Figure 7 shows the reaction forces at different support bearings. The amplitudes of the reaction forces at the different bearings are presented in Table 5.

FIGURE 3: Finite element model of the high-pressure rotor.

FIGURE 4: Finite element model of the simulated engine system.

FIGURE 5: Response of the low-pressure compressor disk in the simulated aeroengine system.

Bearing 1 is the rear fan support bearing. It has a larger vibration load than the other bearings. Since the rear fan support bearing is likely to fail, resulting in a severe accident, the safety requirements for the support bearings as well as the supporting structure need to be stricter.

2.2. Load Path Analysis of Simulated Aeroengine System under a Sudden Unbalanced Load. Owing to the coupling because of the complex structure of the engine, a blade-loss

accident at the fan results in a dynamic response by the high-pressure rotor system. The transfer path for the vibration load in the simulation of the entire engine is complex. By simultaneously interpreting the frequency components of the vibration response at different feature points, the characteristics of the vibration loads on different transfer paths can be observed.

Figure 8 shows that there are four load transfer paths in the whole engine. Path 1 goes from the blade-loss point to

FIGURE 6: Vibration response of the simulated engine system.

TABLE 4: Displacement response of the simulated engine system.

Position	Peak transient response (mm)	Amplitude of steady response (mm)
Low-pressure compressor disk	63.9	33.6
Low-pressure turbine disk	20.7	4.8
High-pressure compressor disk	10.4	2.3
High-pressure turbine disk	44.6	8.6

FIGURE 7: Reaction force of the simulated engine system.

TABLE 5: Reaction force of support bearings.

Position	Peak transient response ($\times 10^5$ N)	Amplitude of steady-state response ($\times 10^5$ N)
1	12.23	4.8
2	10.58	4.4
3	4.5	0.9
4	3.5	0.6

FIGURE 8: Transfer paths for the vibration load in the simulated engine.

FIGURE 9: Vibration response frequency analysis at characteristic points.

FIGURE 10: Response frequency analysis at key points in path 1.

FIGURE 11: Response frequency analysis at key points in path 2.

bearing 1, the casing, and the front suspension point. Path 2 is from the fan-loss point to support bearing 1, support bearing 2, the casing, and the front suspension point. Path 3 goes from the high-pressure turbine to bearing 3, the casing, and the rear suspension point. Finally, path 4 is from the high-pressure turbine to the intermediate bearing, bearing 4, the casing, and the rear suspension point.

To analyze the response and transmission of the excitation in the whole machine, key nodes in the model were selected, as shown in Figure 9: node 1 (fan disk), nodes 4 and 40 (bearing 1), nodes 7 (bearing 2) and 43, nodes 15 and 36 (intermediate bearing), nodes 18 and 54 (bearing

4), nodes 19 and 45 (bearing 3), node 42 (front suspension point), and node 52 (rear suspension point).

The components of the response frequency at different key points included the low-pressure shaft frequency, the natural frequency of the high-pressure shaft, and the rotation speed. The response frequency components were similar at each key node, but in significantly different proportions, which shows that there were different load transfer paths in the system. In paths 1 and 2, the main frequency components were the first-order natural frequency and the rotation frequency of the low-pressure rotor, which hardly reflect the impact of the high-pressure rotor (Figures 10 and 11, respectively).

FIGURE 12: Response frequency analysis at key points in path 3.

Figures 12 and 13 shows that for paths 3 and 4, the response frequency components clearly reflect the impact of the high-pressure shaft. The vibration load generated by the excitation of the high-pressure shaft is mainly transmitted to the rear suspension point through the intermediate bearing and bearing 4.

From Figure 14, it can be seen that the two fulcrums bear loads from different components, which were obviously in different paths.

2.3. Fusing Structure

2.3.1. Fusing Structure Form. In fan-blade loss accidents, the support bearing at the rear of the fan bears a huge impact load, which may cause the supporting structure to break. A fusing structure is a safety control structure. It has a lower support stiffness at the design location. These structures can effectively protect an aeroengine from the huge transient impact load due to an FBO event. Figure 15 shows a variable-stiffness fusing structure. The fusing structure, mainly acting on bearing 1, decreases the support stiffness when an FBO accident occurs. In normal operation, an external support shell provides greater support stiffness for bearing 1. In an FBO accident, the fusing structure fails at the design point, and the external supporting shell structure loses its function. Bearing 1 is then supported only by an inner shell, which has a lower stiffness. As the support stiff-

ness is actively reduced by the weak point, the impact load due to the FBO event is lower. Thus, the stiffness of the support structure changes if there is a large vibration load. The design point, which has a low stiffness, breaks under such a large load.

2.3.2. Response of Simulated Aeroengine with a Variable-Stiffness Fusing Structure. The rotation speeds of the low- and high-pressure rotors were set to 4040 and 6060 rev/min, respectively. The center of mass of the lost mass was 0.5 m away from the rotating axis of the rotor, and the lost mass was 3 kg. The support stiffness decreased from 5×10^7 N/m to 2×10^7 N/m when the support reaction force of bearing 1 reached 6.5×10^5 N. Figure 16 shows the vibration response of the compressor disk of a low-pressure rotor system.

After the FBO event, the fusing structure became activated since the dynamic response of the rotor system increased sharply. As before, the amplitude reached a peak and then fell to a steady state. The motion trace of the low-pressure rotor was elliptical. The steady-state response was lower with the fusing structure, which protected the aeroengine when it was wind milling. Results for effective and ineffective fusing structures are shown in Table 6. Figure 17 shows the displacement response at the compressor and turbine disks.

FIGURE 13: Response frequency analysis at key points in path 4.

When the FBO event happened, the reaction force changed as well as the displacement response. Figure 18 indicates that the reaction force at support bearing 1 was evidently lower while that of support bearing 2 increased, but it did not overload the support structure. The reaction forces of the simulated engine system with and without the fusing structure being activated are listed in Table 7.

Compared with an aeroengine that does not have the fusing structure, the reaction force at almost all rotor support bearings was lower when the fusing structure was activated. There was a significant decrease (45.3%) in the peak reaction force at bearing 1, protecting it from destruction.

2.4. Response Characteristics of a Rotor System with Continuous Rub Impact. When the blade is lost, the response of the rotor system caused by the unbalanced load causes the other blades to rub against the casing. A continuous rub impact is the most common form of flexible rotor rub impact. A rotor is then subject to a radial impact, which restricts the displacement of the rotor and changes the stiffness of the rotor system. This can be considered an additional bearing, which changes the dynamic characteristics of the rotor system and affects its response, as shown in Figure 19.

When the amplitude of the rotor exceeds the clearance and there is continuous rub impact, the additional stiffness

FIGURE 14: Response frequency analysis of suspension point.

FIGURE 15: Variable-stiffness fusing structure [31].

FIGURE 16: Response of rotor system with a variable-stiffness fusing structure.

TABLE 6: Response of low-pressure compressor disk with a variable-stiffness fusing structure.

Position	Peak transient response (mm)	Amplitude of steady response (mm)
Fusing structure not activated	63.9	33.6
Fusing structure activated	80.5	29.4
Relative deviation	25.90%	-12.50%

FIGURE 17: Displacement response of simulated aeroengine with a variable-stiffness fusing structure.

FIGURE 18: Reaction force of the rotor system with a variable-stiffness fusing structure.

of the rotor is K_p , and the motion equation of the system becomes

$$\ddot{\mathbf{M}}\mathbf{u} + (\mathbf{C}_p + \mathbf{G})\dot{\mathbf{u}} + [\mathbf{K}_p(t) + \mathbf{K}]\mathbf{u} = \mathbf{Q}. \quad (6)$$

The additional stiffness K_p is given by

$$K_p = k_c(1 + \text{sign}(\Omega\delta + \omega u)\mu)\left(1 - \frac{\delta}{u}\right), \quad (7)$$

where k_c is the contact stiffness, Ω is the rotor whirl angular velocity, ω is the angular velocity, μ is the friction coefficient

TABLE 7: Reaction force of the simulated engine system with a variable-stiffness fusing structure.

Bearing	Peak transient response	Amplitude of steady response
Without fusing structure	$12.2 \times 10^5 \text{ N}$	$4.8 \times 10^5 \text{ N}$
1 With fusing structure	$6.67 \times 10^5 \text{ N}$	$1.2 \times 10^5 \text{ N}$
Percentage change	-45.3%	-75%
Without fusing structure	$10.58 \times 10^5 \text{ N}$	$4.4 \times 10^5 \text{ N}$
2 With fusing structure	$6.49 \times 10^5 \text{ N}$	$2.34 \times 10^5 \text{ N}$
Percentage change	-38.7%	-46.8%
Without fusing structure	$4.5 \times 10^5 \text{ N}$	$0.9 \times 10^5 \text{ N}$
3 With fusing structure	$5.27 \times 10^5 \text{ N}$	$0.9 \times 10^5 \text{ N}$
Percentage change	17.1%	0%
Without fusing structure	$3.5 \times 10^5 \text{ N}$	$0.6 \times 10^5 \text{ N}$
4 With fusing structure	$2.96 \times 10^5 \text{ N}$	$0.37 \times 10^5 \text{ N}$
Percentage change	-15.4%	-38.3%

FIGURE 19: Model of a rotor with continuous rubbing [29].

of the rotor and stator, δ is the blade casing clearance, and u is the rotor response amplitude.

A linear spring element with a stiffness of $5 \times 10^7 \text{ N/m}$ and 250 N s/m damping was used in the model. The speeds of the low- and high-pressure rotors were 4040 and 6060 rev/min, respectively. The center of mass loss was 0.5 m away from the rotor shaft, and the mass loss was 3 kg. The clearance of the compressor blade casing was 0.006, the impact stiffness was $2.5 \times 10^7 \text{ N/m}$, and the friction coefficient was 0.1. The vibration response is shown in Figure 20.

After the FBO event, the response amplitude increased sharply. The amplitude reached a peak and then decreased and entered a stable stage. The motion trajectory of the low-pressure compressor disk was elliptical, and the fan

FIGURE 20: Response of a low-pressure compressor disk with rub impact.

TABLE 8: Response of a low-pressure compressor disk with rub impact.

Position	Peak transient response (mm)	Amplitude of steady response (mm)
Rub impact	74.3	31.4
No rub impact	80.5	29.4
Relative deviation	7.7%	-6.8%

blade and the casing rubbed together continuously. In the transient phase, the maximum of the vibration response was much higher than in the steady phase. Thus, due to the coupling between the double rotor system and the casing, it is necessary to model the whole system. Table 8 compares the response with and without the continuous rub impact between the casing and blade. With the rub impact, the peak transient response of the low-pressure compressor disk when a blade is lost decreased from 80.5 to 74.3 mm.

The transient response of the fusing structure can be analyzed more accurately by considering the rub impact. The maximum response of the low-pressure compressor disk after a blade loss is lower due to the additional stiffness of the casing, and the negative effects of the fusing structure are reduced. Thus, when analyzing the system response after a blade is lost, it is necessary to establish a model of the whole machine, including the casing, and to consider the effects of the rubbing between the blade and the casing on the response.

3. Conclusions

The dynamic response of a simulated aeroengine suffering a sudden unbalanced load caused by an FBO event was calculated. The effect of the fusing structure and the change to the transmission path were also analyzed. Our conclusions are as follows:

- (1) A finite element model of an aeroengine was developed, mainly with beam elements. The model can accurately calculate the vibration response and reaction force of an aeroengine suffering from an unbalanced load

- (2) After a blade loss, the system suffered from a much larger impulse response in the transient stage than in the steady state. Bearing 1 may fail due to the large reaction force. Our analysis of the four transmission paths in the system revealed that they have different load components. The frequency components of the vibration response for paths 1 and 2 were mainly dominated by the low-pressure rotor system, whereas the dynamic characteristics of the high-pressure rotor were mainly reflected in paths 3 and 4
- (3) The fusing structure can lower the large reaction force of the transient response to protect the structure, because it fails when the reaction force reaches the design point, reducing the support stiffness of bearing 1. A fast method for calculating the response of the fusing structure was developed, which may be an effective tool for predicting the stiffness distribution during design
- (4) When a blade is lost, the response of the rotor system caused by the unbalanced load causes the rub impact between blades and the casing. The rub impact changes the response characteristics of the whole system. After the fusing structure is activated, then with the rub impact, the instantaneous peak response of the rotor after a blade loss was lower. Thus, to analyze the response of an aeroengine after a blade loss, it is necessary to model the whole system, including the casing. The effect of the rub impact cannot be ignored

Data Availability

All data, models, and codes that support the findings of this study are available from the corresponding author upon reasonable request.

Additional Points

Code Availability. The computer software that supports the findings of this study is available from the corresponding author upon reasonable request.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this paper.

Acknowledgments

This work was supported by the National Science and Technology Major Project (2017-IV-0006-0043) and by the Funding of Jiangsu Innovation Program for Graduate Education (Fundamental Research Funds for the Central Universities) (KYLX16_0395).

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