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Experimental study on vibration suppression in a rotor system under base excitation using an integral squeeze film damper①

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Abstract

Base excitation is one of common excitations in rotor system. In order to study the dynamic characteristics of rotor systems under base excitation and the effect of integral squeeze film dampers (ISFDs) on their dynamic characteristics, a single-disk rotor test rig, where mass imbalance and base excitation could be applied, is developed. Experimental research on the rotor system response under sinusoidal base excitation conditions with different frequencies and excitation forces is performed and the effect of ISFD on the dynamic characteristics of the rotor is investigated. The experimental results demonstrate that when the sinusoidal base excitation frequency approaches the first critical speed of the rotor system or the natural frequency of the rotor system base, strong vibration occurs in the rotor, indicating that the base excitation of the two frequencies has a greater impact on rotor system response. In addition, with the increase of the base excitation force, the vibration of the rotor will be increased. ISFDs can significantly inhibit the vibration due to unbalanced forces and sinusoidal base excitation in rotor systems. To a certain extent, ISFDs can improve the effect of sinusoidal base excitation with most frequencies on rotor system response, and they have a good vibration reduction effect for sinusoidal base excitation with different excitation forces.

Key words: Jeffcott rotor, dynamic characteristics, base excitation, integral squeeze film damper (ISFD), vibration suppression

0 Introduction

Turbomachinery, such as turbines, pumps, and compressors, is installed in transportation systems, including warships, submarines, and space vehicles. In their life cycle, they are often subjected to various sudden impact forces, whose specific impact depends on the working conditions and the external environment. Such impacts can be transmitted directly to the rotor-bearing system of turbomachinery through the base or foundation, causing rotor vibration. In severe cases, this may result in direct collision between rotor and bearing, seals and other stators, causing serious damage. Therefore, it is necessary to consider the rotor-support-base system as a whole in order to study the dynamic characteristics of rotor systems under base excitation.

Although it is common for rotor systems to be subjected to inertial forces related to the fundamental motion, it seems that there is a lack of literature on suppressing the lateral vibration of rotors caused by the base motion. Lee et al.[1,2] proposed a transient response analysis method for rotor systems, and proved that the transient response of the rotor is sensitive to the duration of the impact. Sharma et al.[3,4] conducted numerical analysis to demonstrate that the electromagnetic actuator can significantly reduce the response amplitude of the rotor-shafting system under seismic excitation and improve its stability. Han et al.[5,6] explored the parameter instability of a flexible rotor-bearing system under time-periodic base angular motions. Wang et al.[7] investigated the response of a nonlinear bearing-rotor system under the interaction between unbalanced forces and base excitations. Duchemin et al.[8] conducted theoretical and experimental studies on the dynamic behavior of a flexible rotor system under base excitation (support motion). Song et al.[9] designed an active control system of a rotor-electromagnetic dynamic vibration absorber that was excited through the base. Under the combined action of the vibration ab-

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sorber and the base excitation, multi-frequency dynamic vibration absorption in the rotor system was achieved. Fawzi and Fred[10] studied the dynamic characteristics of rigid rotors under the combined action of fundamental excitation and mass imbalance, and provided theoretical guidance for the design of rotating systems.

A new type of damper with novel structure, named integral squeeze film damper (ISFD), appeared in the early 1990s and can provide excellent vibration control performance and effectively improve system stability. Compared to the traditional squeeze film damper, ISFD can effectively suppress rotor vibration, improve the stability of the rotor system, and achieve better vibration reduction performance[11-13]. Santiago and Andrés[14,15] investigated the unbalanced response characteristics of ISFD rotor systems with and without end seals, and revealed that the stiffness and damping of the ISFD had excellent linear characteristics under varying working conditions. Delgado et al. [16] studied the effect of vibration speed on the damping performance of ISFDs. The experiment proved that ISFDs can provide a higher damping coefficient, have better dynamic characteristics than traditional squeeze film dampers, and can effectively suppress the cavitation phenomenon of the oil film. Ertas et al. [17,18] investigated the effect of different end seal clearances on the dynamic coefficients of an ISFD. The hammering method is used to measure the variation trend of the damping coefficients of ISFD, which are found to be linearly correlated with the vibration velocity without cavitation. The experiments demonstrated that ISFDs can solve vibration problems by providing sufficient damping in turbomachinery, such as supercritical carbon dioxide turbochargers. Lu et al. [19-21] performed several experimental studies on the vibration control of gear shafting using ISFD. The experimental results proved that ISFDs can effectively suppress the impact vibration of gear meshing and can efficiently control the vibration in a wider frequency band.

In this paper, a single-disk rotor test rig with base excitation is developed to study the response of the rotor system under various frequencies and excitation forces, as well as to explore the effect of the ISFDs on the dynamic characteristics of the rotor. The vibration suppression of a rotor system by ISFDs under the action of mass imbalance and base excitation is experimentally investigated, and the lateral vibration law of the shaft relative to the stator is studied.

1 Integral squeeze film damper

Squeeze film dampers (SFDs) are commonly used as vibration absorbers in aero-engines, since they provide good vibration reduction performance, and have been widely used in other rotating machinery. Traditional SFDs usually use a centering elastic support, which produces viscous damping by squeezing the oil film vibration modulation in an annular gap. This support provides excellent vibration reduction effect, reduces the rotor bending, reduces the dynamic load exerted on the bearing when the rotor passes through the critical speed, and improves the rotor system stability[22]. However, under severe conditions, such as heavy load and sudden unbalance, where the design scope of work is exceeded, traditional SFDs will exhibit a high degree of nonlinear characteristics, which will lead to nonlinear vibration phenomena, such as lock up, bistable response, non-synchronous precession, and even chaotic motion[23].

Compared to traditional SFDs, the design method and performance of ISFDs are significantly different. ISFDs are fabricated using wire cut electrical discharge machining (WEDM) technology. The piecewise design of the squeeze film region prevents the circumferential flow of the squeeze oil film and solves the highly nonlinear problem of traditional SFDs.

A schematic diagram of the ISFD structure is shown in Fig. 1. The structure consists of 2 parts: the outer ring and the inner ring. The outer ring and the bearing seat are secured by a transition fit, while the inner ring and the outer ring of the rolling bearing are interference-fitted. An S-type elastomer connects the outer ISFD ring and the inner ISFD ring, and the gap between the outer ring and the inner ring forms the squeeze film region. In case of failure, where the rotor system vibrates, the vibration will be transmitted from the rolling bearing to the ISFD inner ring, and the squeezing action will generate a squeezing effect, such that the vibration of the rotor will be effectively controlled and the stability of the rotor system will be improved. The basic parameters of the ISFD structure are listed in Table 1.

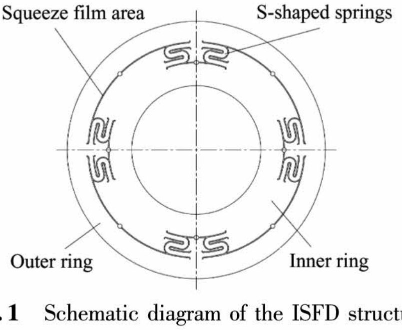


Fig. 1 Schematic diagram of the ISFD structure

Table 1 ISFD structure parameters

Parameter	Value
Axial length (mm)	10
Inner diameter (mm)	30
Outer diameter (mm)	60
Radial height (mm)	20.3
Radial thickness (mm)	4.9
Oil film clearance (mm)	0.2
Distribution angle of the S-shaped spring (°)	52

Studies have shown that the damping force generated by the extruded oil film in an ISFD can be linearized, and the equivalent physical model can be observed in Fig. 2[17]. K_{ij} and C_{ij} is the stiffness and damping coefficient of the equivalent physical model, where i and j represent x and y directions. Different stiffness and damping magnitudes can be obtained by altering the ISFD axial length, radial height, radial thickness, oil film clearance, S-type elastomer distribution angle, and other structural parameters.

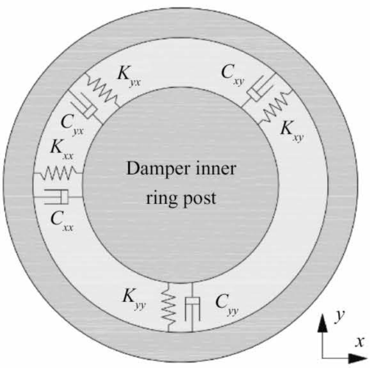


Fig. 2 ISFD equivalent physical model

2 Dynamic model

Firstly, the Jeffcott rotor system is taken as the research object, and rotor dynamic models of rigid and ISFD support under base excitation are established, providing a theoretical basis for experimental research.

2.1 Dynamic model of rigid support rotor

A schematic diagram of the Jeffcott rotor system model of the rigid support structure under base excitation is shown in Fig. 3. The two ends of the rotor adopt a rigid support structure, which is composed of a bearing seat, a rigid sleeve, and a deep-groove ball bearing. The excitation signal is output by the exciter, providing base excitation for the rotor system. In Fig. 3(b), f_B represents the nonlinear supporting force of the deep-groove ball bearings.

The differential equations of motion of the system are

$$\begin{cases} m_D\ddot{x}_D + c_D\dot{x}_D + k_S(x_D - x_B) = m_D e_\mu \omega^2 \cos\omega t \\ m_D\ddot{y}_D + c_D\dot{y}_D + k_S(y_D - y_B) = m_D e_\mu \omega^2 \sin\omega t - m_D g \\ m_B\ddot{x}_B = f_{Bx} + \dfrac{k_S}{2}(x_D - x_B) \\ m_B\ddot{y}_B = f_{By} + \dfrac{k_S}{2}(y_D - y_B) - m_B g \\ m_O\ddot{x}_O + k_O x_O = F_x - f_{Bx} \\ m_O\ddot{y}_O + k_O y_O = F_y - m_O g - f_{By} \end{cases} \tag{1}$$

where m_D is the concentrated mass on the disk; c_D is the viscous damping of the disk caused by the aerodynamic effect; k_S is the stiffness of the shaft, (x_D, y_D), (x_B, y_B), and (x_O, y_O) are the displacements of the

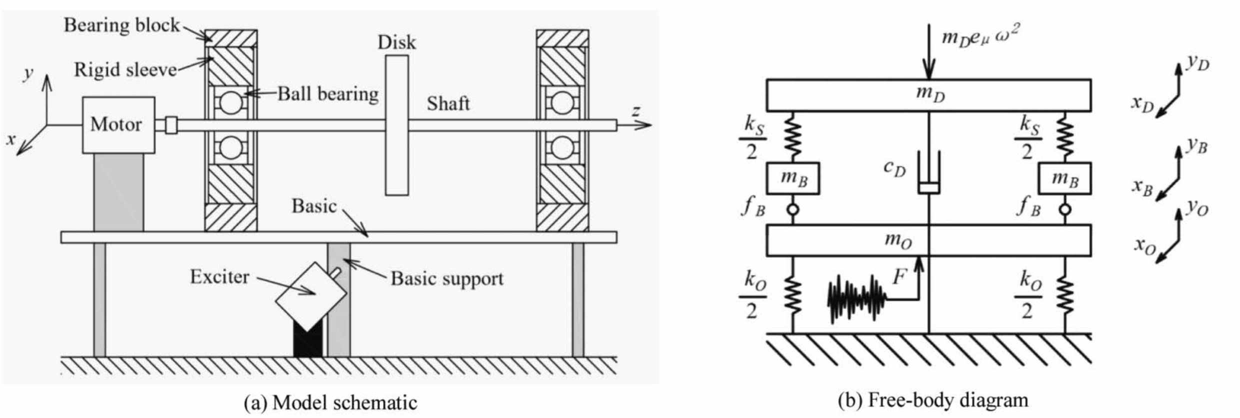


Fig. 3 Dynamic model of a rotor supported on ball bearings with the rigid sleeve subjected to base excitation

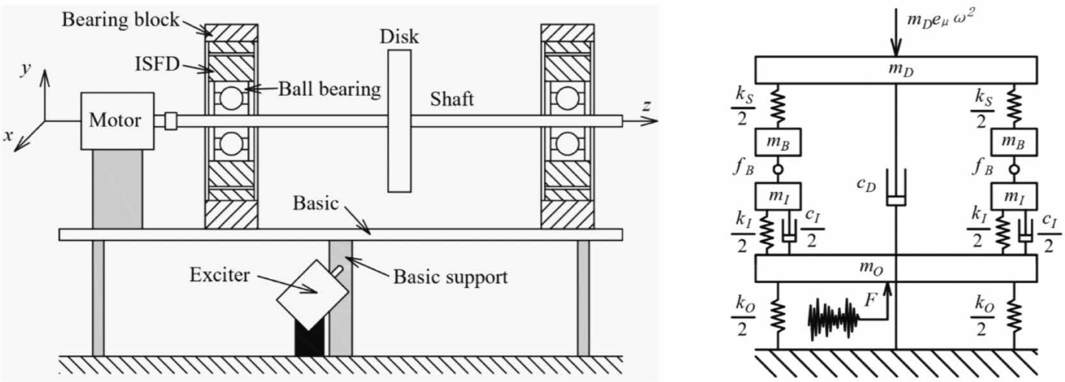
disk, the journal of the deep-groove ball bearing, and the base in a fixed coordinate system, respectively; e_μ is the unbalanced mass eccentricity of the disk; $(\cdot)$ denotes d/dt; ω is the angular velocity of the rotor; m_B is the centralized mass of the journal position and the inner ring position of the deep-groove ball bearings; k_O

is the support stiffness of the base; m_O is the centralized mass of the base and the outer ring position of the deep-groove ball bearings; f_{Bx} and f_{By} are the supporting forces of the deep-groove ball bearings in the x and y directions, respectively; F_x and F_y are the exciting forces of the exciter on the base in the x and y directions, respectively.

2.2 Dynamic model of ISFD supported rotor

A Jeffcott rotor system model of the ISFD supported structure under base excitation is established, and its schematic diagram is shown in Fig. 4. Both ends of the rotor adopt an ISFD support structure, which consists of a bearing seat, the ISFD, and deep-groove ball bearings. The excitation force is output by the exciter, providing base excitation for the rotor system. In Fig. 4(b), f_B represents the nonlinear supporting force of the deep-groove ball bearings.

The differential equations of motion of the system are as follows:



(a) Model schematic (b) Free-body diagram

Fig. 4 Dynamic model of a rotor supported on ball bearings with the ISFD subjected to base excitation

$$\begin{cases} m_D\ddot{x}_D + c_D\dot{x}_D + k_S(x_D - x_B) = m_D e_\mu \omega^2 \cos\omega t \\ m_D\ddot{y}_D + c_D\dot{y}_D + k_S(y_D - y_B) = m_D e_\mu \omega^2 \sin\omega t - m_D g \\ m_B\ddot{x}_B = f_{Bx} + \dfrac{k_S}{2}(x_D - x_B) \\ m_B\ddot{y}_B = f_{By} + \dfrac{k_S}{2}(y_D - y_B) - m_B g \\ m_I\ddot{x}_I + \dfrac{c_I}{2}(\dot{x}_I - \dot{x}_O) + \dfrac{k_I}{2}(x_I - x_O) = -f_{Bx} \\ m_I\ddot{y}_I + \dfrac{c_I}{2}(\dot{y}_I - \dot{y}_O) + \dfrac{k_I}{2}(y_I - y_O) = -f_{By} - m_I g \\ m_O\ddot{x}_O + k_O x_O = F_x + c_I(\dot{x}_I - \dot{x}_O) + k_I(x_I - x_O) \\ m_O\ddot{y}_O + k_O y_O = F_y + c_I(\dot{y}_I - \dot{y}_O) + k_I(y_I - y_O) \\ \qquad - m_O g \end{cases} \tag{2}$$

where m_I is the centralized mass of the outer ring of the deep-groove ball bearing and the ISFD inner ring, c_I is the damping coefficient of the ISFD, k_I is the stiffness of the ISFD, and (x_I, y_I) is the displacement of the ISFD inner ring and the deep-groove ball bearing outer ring in a fixed coordinate system.

3 Introduction of experimental devices

3.1 Support structure

As mentioned above, two different supporting structures are used in the experiments: the rigid support and the ISFD support. Images of the two support types are shown in Fig. 5. Fig. 5(a) shows the rigid support, which is the traditional support configuration, consisting of a rigid sleeve, a deep-groove ball bearing, a bearing seat, an end cover, and a rubber O-ring. Fig. 5(b) shows the ISFD support consisting of an integral squeeze film damper, a deep-groove ball bearing, a bearing seat, an end cover, and a rubber O-ring. In the ISFD support structure, a closed oil chamber is formed by the rubber O-ring, the end cover, and the bearing seat, which provides the ISFD with enough damping fluid to form a squeeze film.

3.2 Test rig set-up

Fig. 6 shows the experimental platform with the single disk rotor system under base excitation, which includes a motor, a rotor, a disk, the ISFD support, the base, a base fixing bracket, an exciter, and other main components.

A Jeffcott rotor is used in the test bench. The span of the shaft is 420 mm, the diameter of the shaft is 10 mm, the thickness of the disk is 15 mm, and the diameter of the disk is 78 mm. The two ends of the rotor are supported either by the rigid or the ISFD support structure, which consists of the bearing base, a rigid sleeve, the ISFD, and a deep-groove ball bearing.

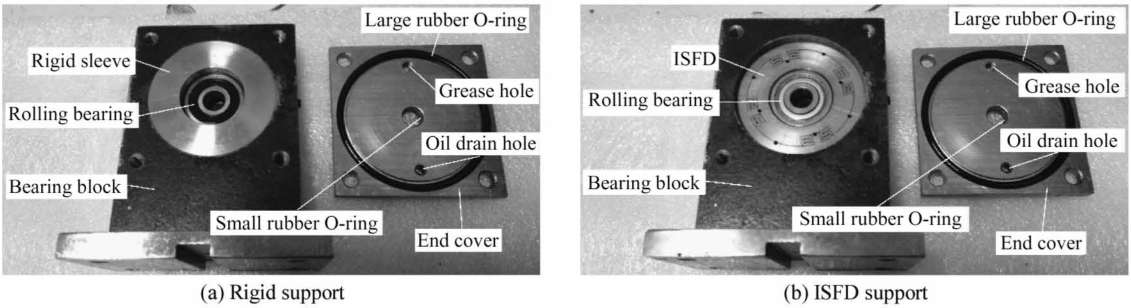
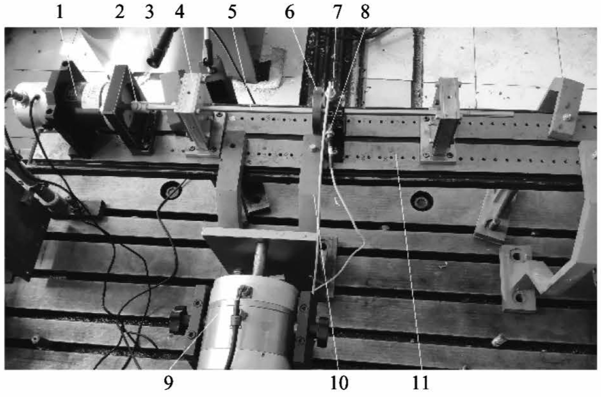


Fig. 5 Images of the rigid and ISFD support structures

The exciter, which could exert a 0 – 100 N, 0 – 100 Hz sinusoidal base excitation force on the rotor system, is attached to the base fixing bracket through bolts. During the experiments, the LC8008 multi-channel fault diagnosis system for mechanical equipment is used to collect vibration displacement signals. The relative displacement of the rotating shaft near the disk and the base is measured by an eddy current sensor. Eddy current sensor 1 captures the relative displacement in the vertical direction (Y-direction) and eddy current sensor 2 captures the relative displacement in the horizontal direction (X-direction). The layout of the experimental set-up can be observed in Fig. 6.



1-Motor; 2-Coupling; 3-Photoelectric sensor; 4-ISFD support; 5-Shaft; 6-Disk; 7-Eddy current sensor 1; 8-Eddy current sensor 2; 9-Exciter; 10-Base fixing bracket; 11-Base

Fig. 6 Test rig of the single disk rotor system under base excitation

4 Results and discussion

4.1 Effect of different support structures on rotor system response under sinusoidal base excitation

In order to investigate the effect of the different supports on the dynamic characteristics of the rotor, the rotor speed is set to 3 300 rpm, the excitation force applied by the exciter to the rotor system is 75 N, and the frequency is 65 Hz. In order to accurately capture the response characteristics, the displacement of the rotor relative to the base with the rigid or the ISFD support is recorded. Amplitude-time and amplitude-frequency comparison diagrams of the rotor vibration responses in the X and Y directions at the measuring points under different support structures are obtained and can be seen in Fig. 7 – Fig. 10.

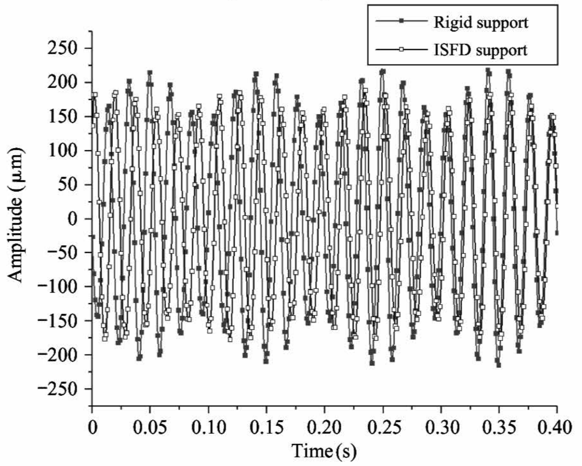


Fig. 7 Amplitude-time comparison diagram of the rotor in the X-direction under different supporting structures

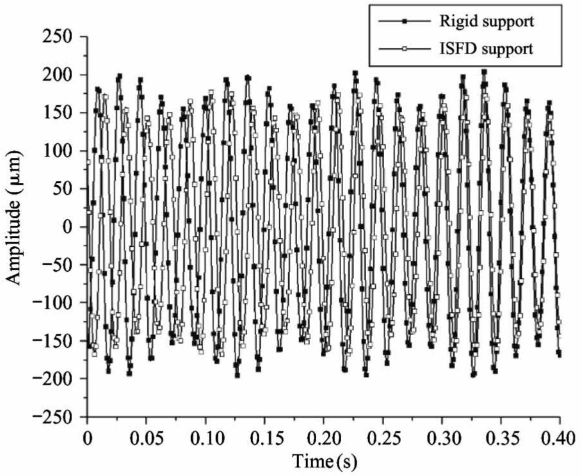


Fig. 8 Amplitude-time comparison diagram of the rotor in the Y-direction under different supporting structures

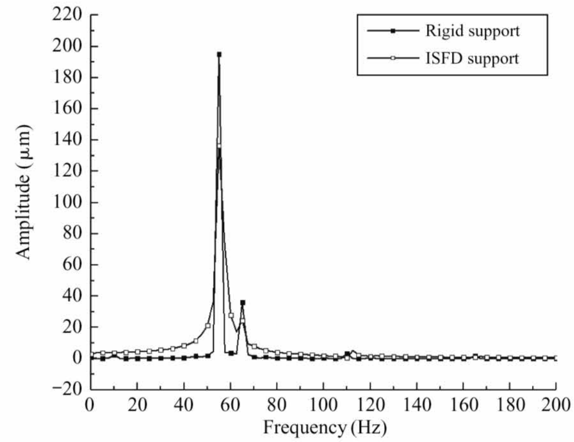


Fig. 9 Amplitude-frequency comparison diagram of the rotor in the X-direction under different supporting structures

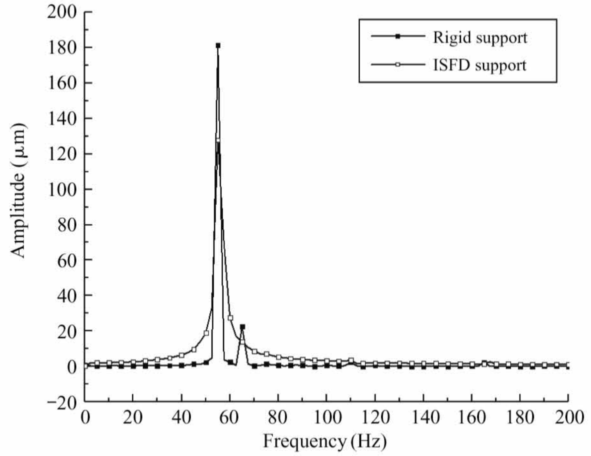


Fig. 10 Amplitude-frequency comparison diagram of the rotor in the Y-direction under different supporting structures

According to the amplitude-time comparison diagrams of Fig. 7 and Fig. 8, the vibration in the rotor system is effectively reduced when the ISFD support is used. In Fig. 9 and Fig. 10, it can be seen that the frequency spectrum mainly includes the rotational frequency of the rotor and the base excitation frequency, in which the amplitude corresponding to the rotational frequency is generated by the unbalanced vibration caused by the uneven mass distribution of the rotating disk. In addition, the amplitude corresponding to the base excitation frequency is generated by the excitation force produced by the exciter and transmitted to the rotor through the bearing seat. Through comparative analysis, it can be deduced that the ISFD support demonstrated a good vibration reduction effect on the base excitation frequency and the rotational frequency of the rotor with the reduction of vibration being more than 30%. More specifically, the amplitude of the rotor in the X-direction at a rotational frequency (55 Hz) decreased from 195.1 μm to 136.5 μm, and at an excitation frequency of 65 Hz, it decreased from 36.0 μm to 24.5 μm. The amplitude of the rotor in the Y-direction at a rotational frequency of 55 Hz decreased from 181.6 μm to 127.9 μm, and at an excitation frequency of 65 Hz, it decreased from 22.6 μm to 13.9 μm.

4.2 Effect of ISFD on rotor system response under sinusoidal excitation conditions with different frequency

In order to investigate the effect of ISFD on rotor system response under sinusoidal excitation with different frequency, the speed of the rotor is set to 3 300 rpm and the exciter applies sinusoidal excitation of different frequency to the rotor system. The magnitude of the excitation force is 75 N and the frequencies are 0 Hz, 5 Hz, 10 Hz, ⋯, 100 Hz. The response characteristics of the rotor system using the two different support structures is investigated. The vibration responses of the rotor in the X and Y directions under base excitations with different frequency are measured. In order to facilitate the observation of the vibration response trend, after the measured data are processed, spectrum waterfall diagrams are plotted (Fig. 11 – Fig. 14).

As it can be seen in Fig. 11 and Fig. 12, in the rotor system with rigid support, when the excitation frequency is close to 65 Hz or 90 Hz, there are 2 amplitude peaks in the spectrogram: the one corresponding to the rotor rotational frequency (55 Hz) and the other is the corresponding amplitude of the base excitation frequency. When the excitation frequency is not proximal to 65 Hz or 90 Hz, only the amplitude corresponding

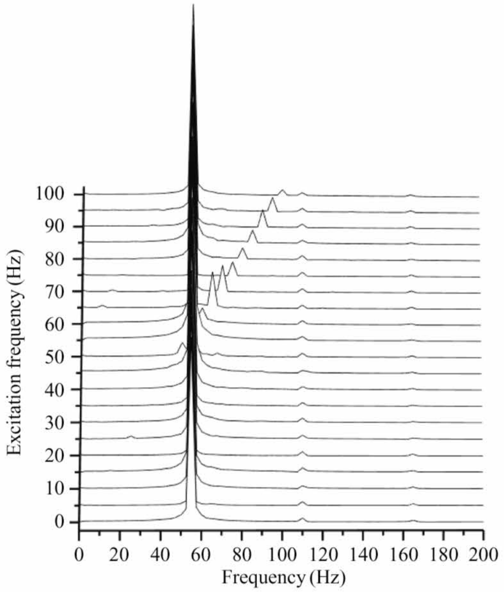


Fig. 11 Spectral waterfall diagram of the rigid support rotor response in the X-direction under base excitations with different frequencies

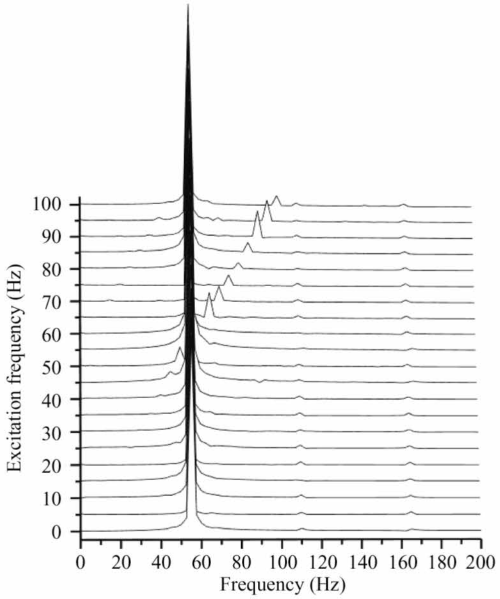


Fig. 12 Spectral waterfall diagram of the rigid support rotor response in the Y-direction under base excitations with different frequencies

to the rotor rotational frequency (55 Hz) could be observed in the spectrogram, while the amplitude corresponding to the base excitation frequency is not present.

From above-mentioned, under a constant excitation force condition, when the excitation frequency approached the frequency of the first critical speed (3 900 rpm) of the rotor system (65 Hz), the corresponding amplitude of the base excitation frequency in the spectrum diagram increased significantly, indicating that the base excitation can excite the natural frequency of the rotor system, which is very dangerous for the rotor system. When the excitation frequency approached 90 Hz, the amplitude corresponding to the base excitation frequency in the spectrum increased again, possibly due to that the frequency of the base excitation is close to the natural frequency of the rotor system base, which resulted in the resonance phenomenon. Due to that the stiffness of the base bracket is large, when the excitation frequency is far from 65 Hz or 90 Hz, the effect of the excitation force on the response of the rotor system is weak, since the amplitude corresponding to the base excitation frequency is not noticeable in the spectrum. As it can be seen in Fig. 13 and Fig. 14, similar to the rigid support rotor system, when the base excitation frequency of the ISFD-supported rotor system approached 65 Hz or 90 Hz, the corresponding amplitude of the base excitation frequency increased significantly, indicating that the base excitation of these 2 frequencies has the greatest impact on rotor system response.

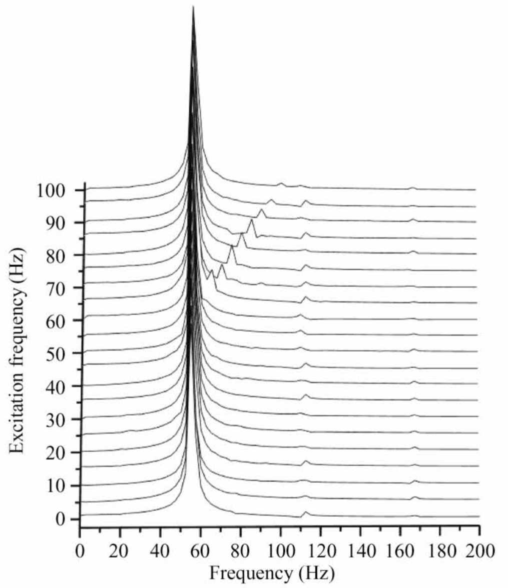


Fig. 13 Spectral waterfall diagram of the ISFD support rotor response in the X-direction under base excitations with different frequencies

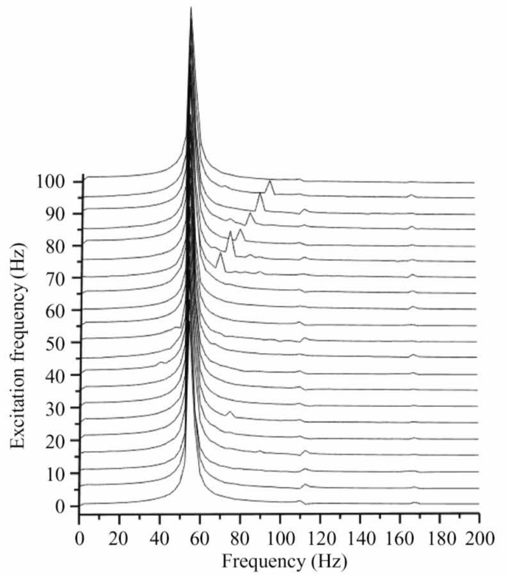


Fig. 14 Spectral waterfall diagram of the ISFD support rotor response in the Y-direction under base excitations with different frequencies

By comparing the responses of the rotor system, it is found that the vibration of the rotor system is significantly decreased after the ISFD support structure is adopted, and the corresponding amplitudes of the rotational and base excitation frequencies are reduced. In order to compare the response of the rotor system under the two different supporting structures, the amplitudes corresponding to the base excitation frequency and the rotational frequency with partial excitation frequencies in the spectrum waterfall diagram are identified and the results are presented in Table 2.

Table 2 Amplitude comparison between the response of the rigid and the ISFD support rotor system under base excitation conditions with different frequencies

Base excitation frequency (Hz)	Measurement direction	Corresponding amplitudes of the rotational frequency (μm)		Vibration reduction (%)	Corresponding amplitudes of the base excitation frequency (μm)		Vibration reduction (%)
		Rigid support	ISFD support		Rigid support	ISFD support	
50	X direction	193.9	143.4	26.0	14.2	13.6	4.2
	Y direction	183.7	137.7	25.0	17.4	10.3	40.8
60	X direction	186.1	146.3	21.4	16.3	15.6	4.3
	Y direction	173.1	139.3	19.5	7.5	19.8	-164.0
65	X direction	195.1	136.5	30.0	36.0	24.5	31.9
	Y direction	181.6	127.9	29.6	22.6	13.9	38.5
70	X direction	188.4	146.2	22.4	26.5	16.9	36.2
	Y direction	177.8	138.0	22.4	13.9	18.3	-31.7
80	X direction	193.2	134.9	30.2	11.9	16.4	-37.8
	Y direction	178.3	130.0	27.1	6.3	12.3	-95.2
90	X direction	191.4	141.9	25.9	18.2	10.0	45.1
	Y direction	181.7	134.1	26.2	24.1	15.8	34.4
100	X direction	191.2	132.7	30.6	6.2	5.7	8.1
	Y direction	179.6	126.1	29.8	9.2	2.2	76.1

As it can be seen in Table 2, the maximum amplitude reduction in the rotor rotational frequency, after the ISFD support structure is installed, is 30.6%, indicating that the ISFD can effectively reduce the vibration of a rotor system caused by unbalanced forces. The maximum amplitude reduction in the base excitation frequency with partial excitation frequency, after the ISFD support structure is installed, is about 40%, suggesting that the ISFD support structure can effectively suppress the vibration of the rotor system caused by base excitation. At an individual excitation frequency (such as 80 Hz), the vibration of the rotor is increased, while the vibration amplitude is maintained at a smaller value. To sum up, it can be seen that under base excitations of different frequencies, when the rigid support is replaced by the ISFD support, the vibration of the rotor system is significantly improved and the corresponding amplitudes of the rotational and base excitation frequencies are reduced. This indicates that the ISFD can effectively improve the vibration of rotor systems caused by unbalanced forces and different base excitation frequencies to a certain extent.

4.3 Effect of ISFD on rotor system response under sinusoidal base excitation conditions with different excitation forces

According to the spectrum waterfall diagrams in Fig. 11 and Fig. 12, when the excitation frequency is 65 Hz and 90 Hz, the base excitation has a great effect on the rotor system. In the following experiment, a sinusoidal base excitation of 65 Hz is applied to the rotor system in order to investigate the effect of ISFD on the response of the rotor system under sinusoidal base excitation conditions with different excitation forces. The rotor speed is again set to 3 300 rpm, while the exciter applies sinusoidal excitation with different excitation forces to the rotor system. The magnitude of the excitation frequency is 65 Hz and the applied forces are 0 N, 15 N, 30 N, ⋯, 90 N. In order to observe the variation trend of the vibration response, after the measured data are processed, spectrum waterfall diagrams are plotted (Fig. 15 – Fig. 18).

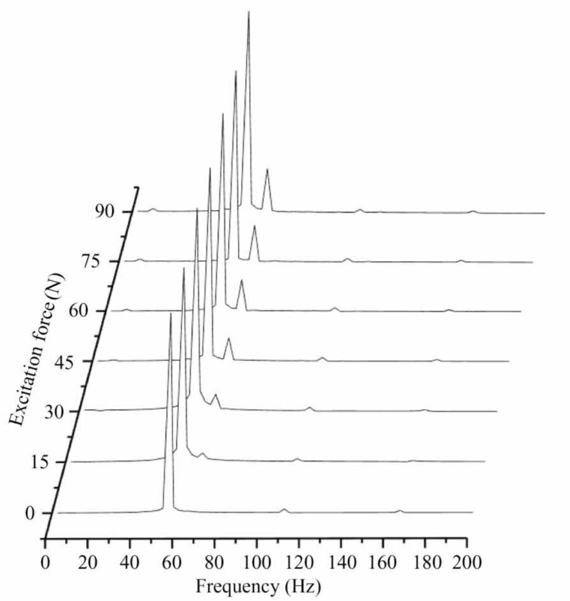


Fig. 15 Spectral waterfall diagram of the rigid support rotor response in the X-direction under base excitations with different forces

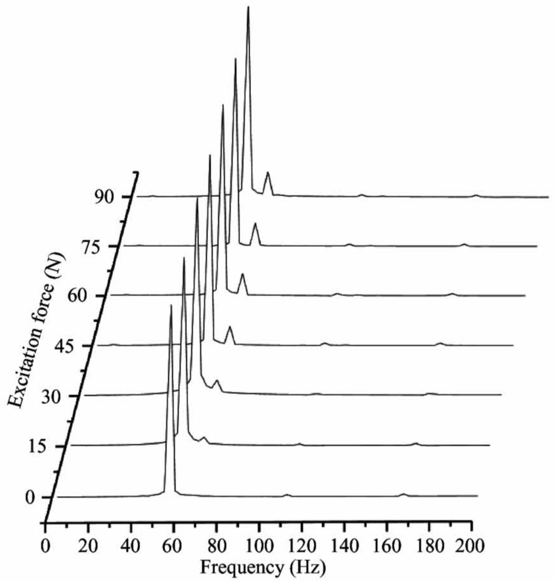


Fig. 16 Spectral waterfall diagram of the rigid support rotor response in the Y-direction under base excitations with different forces

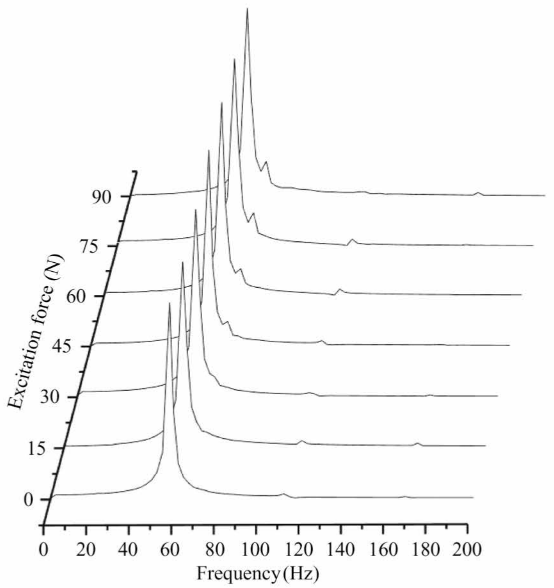


Fig. 17 Spectral waterfall diagram of the ISFD support rotor response in the X-direction under base excitations with different forces

As it can be seen in Fig. 15 – Fig. 18, in the rigid support rotor system, as the excitation force increased, the amplitude corresponding to the base excitation frequency kept increasing. However, in the ISFD support rotor system, this trend is not observed due to the damping effect of the squeeze oil film.

By comparing the response between the rigid support and the ISFD support rotor system, it can be seen that the vibration of the rotor system is effectively reduced after the ISFD support structure is adopted,

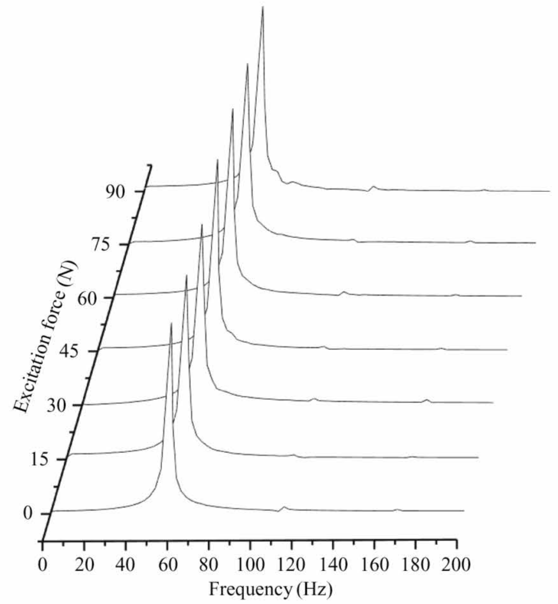


Fig. 18 Spectral waterfall diagram of the ISFD support rotor response in the Y-direction under base excitations with different forces

and the corresponding rotation frequency and base excitation frequency amplitudes are reduced. In order to compare the changes in rotor system response under the 2 configurations, the amplitudes corresponding to base excitation and rotational frequencies under different base excitation forces are identified and the results are listed in Table 3.

As it can be seen in Table 3, compared to the rotor system with the rigid support structure, the ISFD support structure could effectively suppress the rotor vibration caused by unbalanced forces, and the reduction in vibration amplitude reaches 32%. At the same time, the corresponding amplitude of the rotor excitation frequency is reduced, and the reduction reaches more than 40%. Based on the above analysis, under the action of different base excitation forces, when the rigid support is replaced by the ISFD support, the vibration of the rotor system is significantly improved, and the corresponding amplitudes of the rotational and base excitation frequencies are reduced. This indicates that the ISFD can effectively improve the vibration of rotor systems caused by unbalanced forces and base excitation of different excitation force magnitude to a certain extent.

5 Conclusions

The dynamic characteristics of a single disk rotor-rolling bearing-ISFD system under base excitation are investigated. The response of the rotor system under sinusoidal base excitation with different frequencies and

Table 3 Amplitude comparison between the response of the rigid and the ISFD support rotor system under base excitation conditions with different forces

Excitation force (N)	Measurement direction	Corresponding amplitudes of the rotational frequency (μm)		Vibration reduction (%)	Corresponding amplitudes of the base excitation frequency (μm)		Vibration reduction (%)
		Rigid support	ISFD support		Rigid support	ISFD support	
30	*X* direction	193.5	135.5	30.0	16.5	14.5	12.1
	Y direction	186.3	126.7	32.0	14.9	11.5	22.8
45	*X* direction	184.5	142.1	23.0	22.4	18.2	18.8
	Y direction	180.4	134.8	25.3	18.3	12.5	31.7
60	*X* direction	189.5	140.7	25.8	30.2	20.0	33.8
	Y direction	181.3	133.2	26.5	21.0	12.6	40.0
75	*X* direction	183.3	136.5	25.5	35.0	24.5	30.0
	Y direction	178.8	127.9	28.5	22.2	13.9	37.4
90	*X* direction	193.8	137.5	29.1	42.1	25.6	39.2
	Y direction	181.7	131.3	27.7	24.3	14.4	40.7

excitation forces is experimentally studied. The effect of the ISFD on the dynamic characteristics of the rotor is investigated. Useful conclusions are drawn, which are as follows:

(1) Comparative experiments demonstrate that when a certain sinusoidal base excitation is applied to a rotor system, the ISFD can significantly inhibit the vibration related to unbalanced forces and sinusoidal base excitation.

(2) In the case where sinusoidal base excitation with different frequencies is applied to the rotor system, when the base excitation frequency approached the first critical speed of the rotor system (65 Hz) or the natural frequency of the rotor system (90 Hz), strong vibration in the rotor system appeares, indicating that the base excitation of these 2 frequencies has a greater effect on rotor system response. It indicates that the ISFD can effectively improve the vibration of rotor systems caused by unbalanced forces and different base excitation frequencies to a certain extent.

(3) In the case where sinusoidal base excitation with different exciting forces is applied to the rotor system, when the base excitation force is increased, the vibration response of the rotor system becomes larger and larger. In addition, it is found that the ISFD has a good vibration reduction effect on sinusoidal base excitation with different exciting forces.

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