



Article

The Influence of Misalignment on the Performance of the Herringbone Groove Journal Bearing-Rotor System

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Abstract

Misalignment of the herringbone groove radial bearing can lead to changes in performance and system abnormalities. To investigate the effects of different misalignment modes and magnitudes on the HGJB-rotor system, a coupled dynamic model was established. Based on this model, the influences of parallel misalignment and angular misalignment on bearing performance were analyzed, and the variation law of rotor vibration was revealed. The results indicate that the rotor motion trajectory and bearing dynamic coefficients (including critical journal mass and critical whirl frequency) exhibit time-varying characteristics. Specifically, compared with the aligned condition, a parallel misalignment of $\delta = 8.0 \times 10^{-6}$ m reduces the relative film thickness by 17.8% and increases the maximum film pressure by 1.85%. Meanwhile, an angular misalignment of $\theta_0 = 8.0 \times 10^{-4}$ rad results in a 45.9% reduction in relative film thickness and a 33.1% increase in maximum film pressure. Additionally, the increased misalignment magnitude enhances the rotor vibration amplitude significantly. For instance, the Y-direction displacement amplitude increases by 59.4% under the maximum parallel misalignment. Moreover, the misalignment also alters the axial trajectory of the rotor. Overall, different misalignment modes and magnitudes exert significant effects on the rotor vibration characteristics. The research findings provide theoretical support and technical references for the further development and engineering application of HGJBs.

Keywords: herringbone groove journal bearing; misalignment; dynamics; vibration characteristics



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1. Introduction

Herringbone groove journal bearings (HGJBs) generate a hydrodynamic oil film for load bearing by virtue of the pumping effect of herringbone grooves and the cyclic stair-step effect. Featuring a compact structural design, the bearings exhibit low vibration and noise emissions, along with minimal frictional power loss and wear [1,2]. Compared with conventional journal bearings, HGJBs offer superior operational stability and adaptability, which has led to their widespread application in high-speed rotating machinery, including turbocompressors and turbine generators [3,4]. Bearing misalignment is an inevitable issue in engineering practice, caused by improper installation, rotor deformation, or dynamic loads during operation. This issue has an adverse impact on both bearings and the entire mechanical system, leading to discernible performance degradation, such as reduced oil

film thickness, increased film pressure, and amplified rotor vibration. These effects directly undermine the operational reliability and service life of rotating machinery [5]. Consequently, the effects of misalignment on bearing-rotor systems have become a key research focus. However, the complexity of operating conditions and the strong coupling between bearing lubrication and rotor motion have impeded a comprehensive understanding of misalignment mechanisms. As a result, numerical simulation and experimental testing are indispensable for enhancing the predictive capability of system performance.

Existing studies on bearing misalignment have established important theoretical and experimental foundations. However, there are notable limitations in research methodology and application scenarios. The connections between research findings are fragmented, which hinders the further development of this field. Zhang et al. [6] conducted a comparative analysis of the effects of misalignment along two perpendicular directions on the stability of a rigid rotor–journal bearing system. It was found that the system was more sensitive to horizontal misalignment, providing insights into the directional dependence of misalignment effects. Nevertheless, the study was restricted to gas-lubricated aerostatic bearings and only concentrated on stability thresholds, neglecting the transient evolution of rotor vibration and oil film distribution. This leaves gaps in the understanding of dynamic coupling effects in hydrodynamic bearing systems. Sun et al. [7] conducted misalignment research by incorporating parameter uncertainty into the stability analysis of rotor-bearing systems. The results indicated that the misalignment reduces stability thresholds and increases the risk of subsynchronous vibration, highlighting the significance of misalignment in the robust design of such systems. However, the research employed a simplified model that treated bearings as linear stiffness damping elements, ignoring the nonlinear variations of oil film characteristics under misalignment conditions. Moreover, no distinction was made between parallel and angular misalignment, resulting in the failure to reveal their distinct impacts on system stability. Li et al. [8] proposed a novel theoretical model to investigate the effects of misalignment on the static and dynamic characteristics of micro-grooved journal bearings. They found that micro-grooves could alleviate pressure concentration caused by misalignment, offering guidance for the structural optimization of grooved bearings. Nevertheless, this research focused on static performance and steady-state dynamic coefficients, overlooking the transient interaction between rotor motion and oil film dynamics. Additionally, micro-grooves differed significantly from herringbone grooves in pumping effect and oil film distribution characteristics, making the research findings inapplicable to high-speed conditions.

Other relevant studies have further expanded the research scope, yet they still fail to address the core research requirements of the HGJB-rotor system. For example, Zhang [9] compared the steady-state characteristics of the bearings under pure static and hybrid operating conditions. It emphasized the sensitivity of bearing performance to horizontal deviations while neglecting the dynamic coupling between bearings and rotors. Li et al. [10] incorporated eccentricity into the stability analysis of bearing systems and verified the results experimentally. However, their research focused on flexible rotors with unbalanced disks, and the effects of misalignment were not considered. Han [11] adopted a similar research methodology for high-speed turbine generator rotors. Nevertheless, the study focused on axial clearance rather than misalignment. Huang [12] and Cao et al. [13] investigated the effects of temperature differences and bearing clearance on system dynamic behavior, respectively. However, their studies did not integrate the dynamic changes in the system induced by the misalignment. The dynamic response and stability of the rotor-bearing system are critical factors for ensuring the stable operation of high-speed rotating machinery [14–20]. Given the widespread application of HGJBs in rotating machinery, the misalignment effect has attracted increased research attention [21].

As discussed above, existing research on the misalignment effects of HGJBs has primarily focused on individual bearings and steady-state performance. It lacks systematic investigations into the quantifiable dynamic impacts of misalignment on the HGJB-rotor system. These impacts include the time-varying laws of rotor vibration amplitude, the quantitative sensitivity of bearing dynamic coefficients to different types of misalignments, and the nonlinear variations of key lubrication parameters with misalignment magnitude. Moreover, existing studies lack a coupled dynamic analysis that integrates the above quantitative indicators. This makes it difficult to provide accurate guidance for engineering applications that require vibration control and an extended bearing service life. To fill this research gap, a coupled dynamic model is established in this study. This model can characterize the transient dynamic process of the system, the temporal evolution of rotor whirl, and bearing dynamic performance. A comprehensive analysis is conducted on the effects of two types of encountered misalignments on the dynamic performance of the HGJB-rotor system.

2. Theory

As illustrated in Figure 1, a rigid rotor is supported by two HGJBs. The distance from point a on the left HGJB to the rotor's left end is defined as L_1 , and the distance from point a to the rotor's center of mass is denoted as L_a . For the right HGJB, the distance from point b to the rotor's right end is L_2 , while the distance from point b to the rotor's center of mass is L_b . Owing to the spatial distribution of the bearings and the inherent rotor imbalance, the rigid rotor is capable of translational and torsional motions. The rotational axis X , which is collinear with the axial coordinate, is the central axis about which the shaft rotates at an angular velocity of ω .

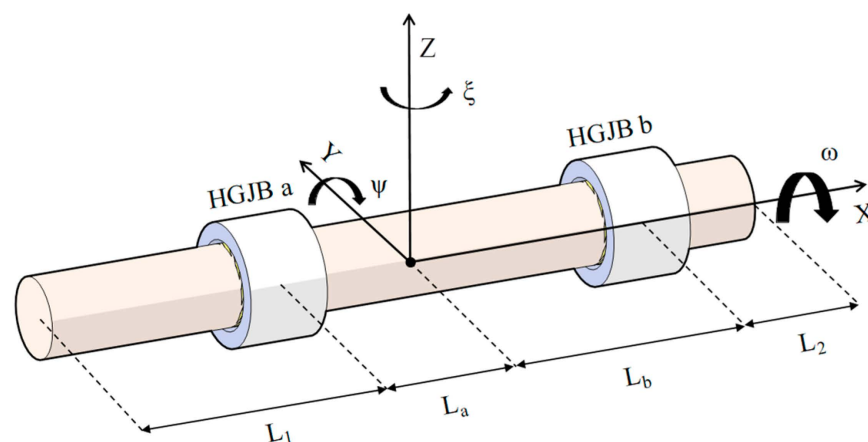


Figure 1. Coordinate system of the HGJB-rotor system.

Given the significant stiffness disparity between the bearings and the rotor, the rotor is modeled as a rigid body in this study. Based on the formulation proposed by Liu et al. [22], the motion equations of the rigid rotor are derived with targeted modifications implemented to incorporate the effects of two typical misalignment forms (parallel deviation and angular deviation). The specific modification approaches are elaborated. The core modification lies in the revision of bearing force terms. The original model in Ref. [22] is established under the assumption of perfect bearing–rotor alignment, where the hydrodynamic oil film distribution exhibits circumferential symmetry. To account for misalignment effects, the bearing force components (F_Z^a , F_Y^a , F_Z^b , F_Y^b) in the motion equations are re-derived by coupling the oil film pressure distribution characteristics under misaligned conditions. For parallel deviation (characterized by transverse displacement δ), the oil film thickness undergoes a uniform offset, and the bearing forces are adjusted. For angular

deviation (defined by tilt angle θ_0), the local oil film thickness varies linearly along the axial direction (with one side thinning and the other thickening), resulting in an asymmetric oil film pressure distribution. The asymmetric pressure distribution is integrated into the bearing force calculation to characterize the unbalanced lateral forces, which are induced by angular misalignment.

The pressure distribution characteristics under misalignment conditions are as follows: for parallelism deviation (transverse displacement δ), which adheres to the definition of parallelism tolerance in ISO 1101 [23], the oil film thickness experiences a uniform offset, and the bearing force is regulated by the change in the average film thickness. For angular deviation (tilt angle θ_0), corresponding to the angularity tolerance specified in ISO 1101 [23], the local oil film thickness varies linearly in the axial direction (one side is thinner and the other is thicker), leading to an asymmetric pressure distribution. This asymmetric distribution is incorporated into the bearing force calculation to characterize the induced unbalanced lateral forces. Reference [22] explores the moment balance caused by imbalance and differences in bearing forces. In the present study, the angular misalignment exerts a tilting moment on the rotor. The moment is incorporated into the moment balance equations to capture the coupling effect between the rotor's tilting motion and lateral displacement under the action of θ_0 .

The modified motion equations are expressed as follows:

$$\begin{cases} m_r \ddot{z} = F_Z^a + F_Z^b + F_{Zi} + m_r g \\ m_r \ddot{y} = F_Y^a + F_Y^b + F_{Yi} \\ I_T \ddot{\xi} + I_P \dot{\psi} = F_Y^a l_a - F_Y^b l_b + M_{\xi i} \\ I_P \ddot{\psi} + I_T \dot{\xi} = -F_Z^a l_a + F_Z^b l_b + M_{\psi i} \end{cases} \quad (1)$$

In the given equation, the key parameters are defined as follows: m_r represents the rotor mass; I_T and I_P denote the lateral and polar moments of inertia of the rotor, respectively; z and y are the translational displacements along the Z and Y directions, respectively; ξ and ψ stand for the rotational angles, respectively; F_Z and F_Y are the bearing forces acting along the Z and Y directions, while F_{Zi} and F_{Yi} represent the unbalanced forces in the corresponding directions; l is the distance between the bearing midplane and the mass center of the rotor; $M_{\xi i}$ and $M_{\psi i}$ denote the imbalance-induced moments, respectively; and a and b are used to designate the two bearings individually.

To validate the rigid rotor assumption, the first lateral critical speed of the rotor is calculated by using the Rayleigh method, which is a well-recognized approach for estimating the critical speeds of rotor systems. The rotor is assumed to be made of structural steel with a density of $\rho = 7850 \text{ kg/m}^3$ and an elastic modulus of $E = 206 \text{ GPa}$. Based on the geometric parameters of the rotor ($D_r = 19.94 \text{ mm}$, $L_1 = 30 \text{ mm}$, $L_2 = 15 \text{ mm}$, $L_a = 70 \text{ mm}$, $L_b = 85 \text{ mm}$), the rotor is modeled as a stepped shaft, and its mass distribution and polar moment of inertia are derived accordingly. The cross-sectional area of the rotor is $A = \pi(D_r/2)^2 = 3.11 \times 10^{-4} \text{ m}^2$, and the total length of the rotor is $L_{total} = L_1 + L_a + L_b - (L_a + L_b - L_1 - L_2) = 130 \text{ mm}$. The total mass of the rotor is $m_r = \rho A L_{total} = 3.17 \text{ kg}$. The polar moment of inertia for the rotor is $I_P = \pi D_r^4 / 64 = 3.92 \times 10^{-11} \text{ m}^4$. The Rayleigh formula for the first lateral critical speed n_c is as follows:

$$n_c = \frac{30}{\pi} \sqrt{\frac{g \sum (m_i y_i^2)}{\sum (m_i y_i^2) + \frac{\pi^2}{4} \left(\frac{I_P l_i^3}{3 E I_{bi}} \right)}} \quad (2)$$

where $g = 9.81 \text{ m/s}^2$, m_i and y_i are the mass and static deflection of each rotor segment, I_{bi} is the bending moment of inertia of the bearing support (adopting the stable stiffness coefficient $k_{yy} = 1.85 \times 10^7 \text{ N/m}$ from Section 3.2 to equal the bearing support stiffness), and l_i is the length of each segment. Substituting the structural and material parameters into the formula, the first lateral critical speed of the rotor is calculated as $n_c = 35,200 \text{ rpm}$. The operating speed of the rotor in this study is $12,000 \text{ rpm}$, which is only 34.1% of the first critical speed ($12,000/35,200 \approx 0.34$).

For rotating machinery, when the operating speed is less than 40% of the first critical speed, the elastic deflection of the rotor is negligible (typically less than $1 \times 10^{-6} \text{ m}$), which is far smaller than the bearing clearance ($\approx 3 \times 10^{-5} \text{ m}$) and oil film thickness ($\approx 2 \times 10^{-5} \text{ m}$) in the system. This indicates that the rotor's elastic deformation has no significant impact on the oil film distribution, bearing force transmission, or rotor vibration characteristics. Thus, the rigid-rotor assumption is justified.

In practical operation, bearings may experience misalignment due to improper installation. When the axis of the bearing is parallel to the rotor shaft, but there is a lateral displacement δ in the transverse direction, it is called parallelism deviation, which can be seen in Figure 2a. When the bearing axis is not perfectly parallel to the rotor, but the bearing center remains in the direction of the rotor axis while being inclined at an angle θ_0 , this condition is known as angular deviation, as shown in Figure 2b.

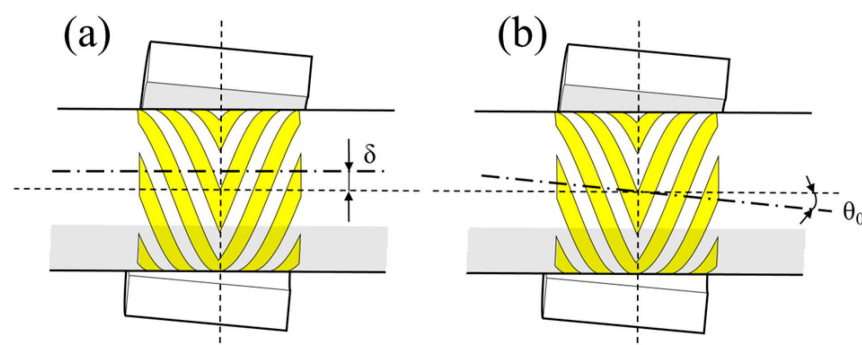


Figure 2. Two bias misalignment models. (a) Parallel misalignment; (b) angular misalignment.

3. Model Validation

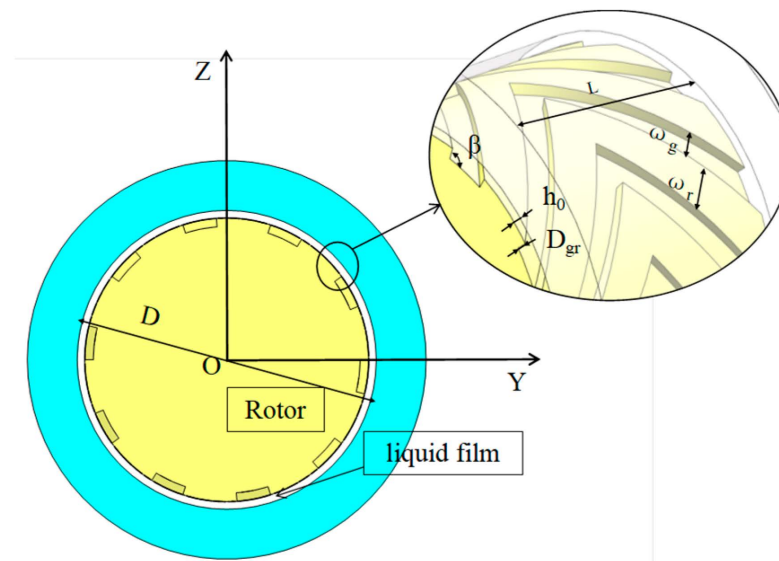
In this study, the herringbone journal bearing with the most widely used rectangular-groove symmetrically distributed configuration was investigated. Figure 3 illustrates the configuration of HGJB. Key structural parameters included aspect ratio L/D , groove width ratio $\alpha_b = \omega_g / (\omega_g + \omega_r)$, groove angle β , groove depth ratio $H_{gr} = 1 + (D_{gr}/h_0)$, ridge ratio γ , radial clearance C , and number of herringbone grooves N . The specific structural parameters can be found in Table 1. By employing a wrapping command, the axle journal was adorned with a herringbone groove pattern. The pertinent details regarding the rotor's structural characteristics can be found in Table 2.

Table 1. Geometric specifications of herringbone groove journal bearings (HGJB).

Parameter	Value	Parameter	Value
D (mm)	20	H_{gr} (-)	2
L/D (-)	1	D_{gr} (mm)	0.03
α_b (-)	0.3	h_0 (mm)	0.03
ω_g (mm)	1	C (mm)	0.03
B ($^\circ$)	158.7	N (-)	6

Table 2. Geometric specifications of the rotor.

Parameter	L_1 (mm)	L_2 (mm)	L_a (mm)	L_b (mm)	D_r (mm)
Value	30	15	70	85	19.94

**Figure 3.** The HGJB-rotor system structure.

3.1. Vortex Analysis

The transient simulations were carried out by COMSOL 6.3 Multiphysics. A transient implicit solver was adopted to ensure numerical stability. The oil film domain was discretized with a structured hexahedral mesh (minimum radial size: 5×10^{-6} m). A mesh independence study verified that the 2.5×10^5 element mesh struck a balance between accuracy and efficiency. Given the thin-film flow with a low Reynolds number ($Re < 1000$), the laminar flow model was employed. The convergence criteria were set as residual norms $< 1 \times 10^{-6}$ for flow/structural variables and a relative displacement change $< 5 \times 10^{-7}$, with a fixed time step of 1×10^{-5} s (1/50 of the rotor rotational period). Two-way fluid solid coupling (FSI) was implemented. The oil film pressure was transferred to the rotor as a surface force, while the rotor displacements updated the film thickness boundary condition in real time.

The primary objective of this study was to evaluate the accuracy of the proposed analysis model. By configuring the bearing surfaces and the rotor axis in a herringbone pattern, transient simulations were carried out to obtain a solid–liquid coupling field. The rotor rotated at a speed of 12,000 rpm. One of the bearings was misaligned with the rotor axis, which caused the rotor to undergo whirl due to mass imbalance. The centroid of the rotor was chosen as the reference point for conducting the transient analysis, and the trajectory in the YZ plane was obtained, as shown in Figure 4. The trajectory comprised a transient response and steady-state whirl. The equilibrium positions in the trajectory were determined by the gravity load, and the whirl near the equilibrium positions was the response to the mass imbalance.

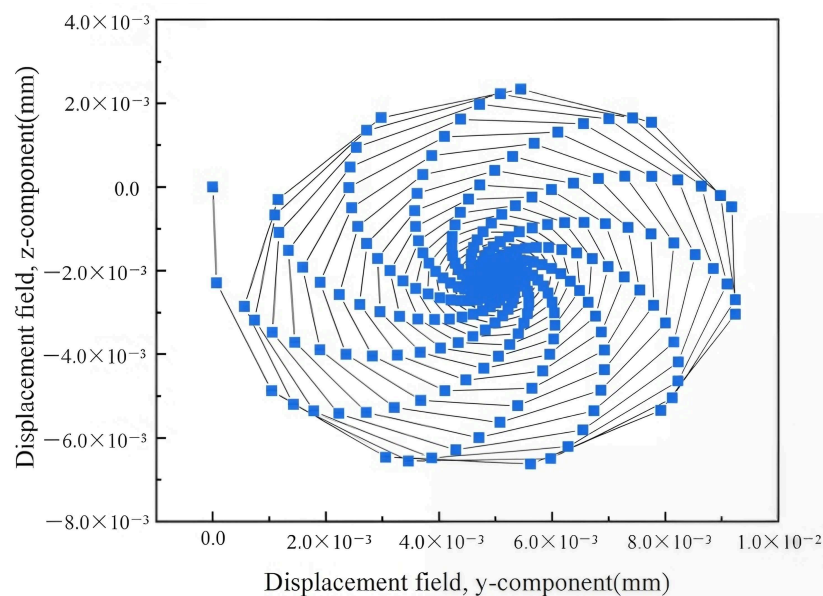


Figure 4. Rotor trajectory in the YZ plane of the displacement field.

In Figure 4, the shape of the rotor trajectory was quite regular. It can be observed that the rotor trajectory in the displacement field mainly demonstrated a pure cylindrical or conical motion. This motion phenomenon is consistent with the experimental results in Elia's study [24]. The experiment on spiral-grooved gas journal bearings indicated that the rotor under imbalance excitation exhibits half-frequency whirl with a dominant frequency of 0.45–0.47 times the rotational frequency. In this study, the rotor was rotated at 12,000 rpm ($f_0 = 200$ Hz), which is the same as the speed in the literature. As shown in Figure 5, the spectral analysis revealed that the dominant frequency of the rotor's acceleration was 92 Hz, which is 0.46 times f_0 . It fell within the range of Elia's experimental results (0.45–0.47 f_0), with a relative error of only 2.2%. This validated the accuracy of the proposed model.

The first lateral critical speed of the rotor (35,200 rpm) was significantly higher than the operating speed (12,000 rpm), confirming that the rotor operated within the sub-critical speed range. In this range, the rotor avoided resonance, and its dynamic response was dominated by mass imbalance and bearing force rather than elastic deformation. This further supported the validity of the rigid rotor assumption adopted in the model. The consistency between the simulated rotor trajectory (Figure 4) and experimental results [24] also indirectly verified that neglecting rotor elasticity does not compromise the accuracy of the model.

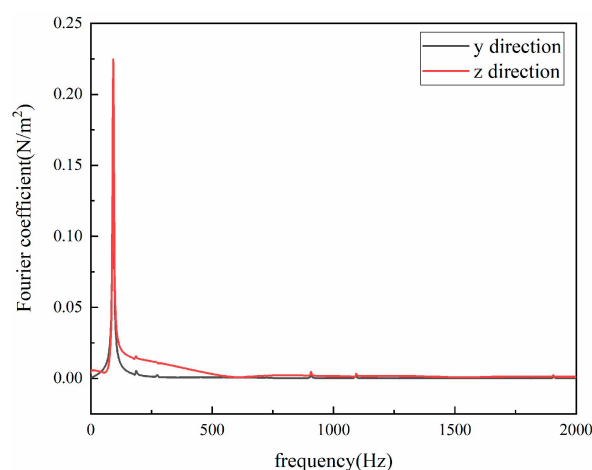


Figure 5. Acceleration spectrum analysis of bearings.

3.2. Dynamic Characteristic Analysis of Bearings

In this paper, the dynamic stiffness and damping coefficient of herringbone groove journal bearings were extracted by using the small disturbance method. The steady-state hydrodynamic pressure distribution in the oil film was solved by the Reynolds equation considering misalignment effects, as shown as follows:

$$\frac{\partial}{\partial x} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = 6\omega \frac{\partial h}{\partial x} + 12 \frac{\partial h}{\partial t} \quad (3)$$

where h denotes the oil film thickness, μ is the lubricant dynamic viscosity, p is the oil film pressure, ω is the rotor angular velocity, and t is time.

Small harmonic perturbations were imposed on the bearing's steady-state position and velocity to linearize the bearing force response. The perturbation amplitude was set to $\varepsilon_p = 1 \times 10^{-4}$ (much smaller than the steady-state eccentricity to satisfy the small perturbation assumption), and the perturbation frequency was set to $f_p < f_0 = 200$ Hz to avoid nonlinear distortions. The perturbed film thickness and pressure were decomposed into steady-state (h_0, p_0) and perturbation ($\Delta h, \Delta p$) components: $h = h_0 + \Delta h$, $p = p_0 + \Delta p$. Substituting the decomposed variables into Equation (3), the higher-order small quantities were neglected, and the linearized perturbation Reynolds equation was derived as follows:

$$\frac{\partial}{\partial x} \left(\frac{h_0^3}{\mu} \frac{\partial \Delta p}{\partial x} \right) + \frac{\partial}{\partial z} \left(\frac{h_0^3}{\mu} \frac{\partial \Delta p}{\partial z} \right) = 6\omega \frac{\partial \Delta h}{\partial x} + 12 \frac{\partial \Delta h}{\partial t} \quad (4)$$

Solving Equation (4) under the boundary conditions ($p(x) = p(x + 2\pi)$), the perturbed pressure Δp was integrated over the bearing surface to obtain linearized bearing forces, which are expressed as follows:

$$\begin{cases} \Delta F_y = k_{yy}\Delta y + k_{yz}\Delta z + c_{yy}\Delta \dot{y} + c_{yz}\Delta \dot{z} \\ \Delta F_z = k_{zy}\Delta y + k_{zz}\Delta z + c_{zy}\Delta \dot{y} + c_{zz}\Delta \dot{z} \end{cases} \quad (5)$$

where k_{ij} and c_{ij} are the dynamic coefficients.

By examining the fluctuations of these coefficients throughout the transient analysis period, it was possible to obtain stable bearing dynamic coefficients. Figure 6 illustrates the temporal evolution of the individual components of the dynamic stiffness coefficients specifically. The eccentricity ε and the tilt angle θ_0 were considered. The negative sign indicated direction only. It can be observed that the stiffness coefficients increased with time. The direct force stiffness coefficient and the cross-coupling force stiffness coefficient exhibited opposite directions but similar magnitudes of variation. After $t = 0.3$ s, the bearing stiffness reached a stable value: k_{yy} was at 1.85×10^7 N/m, and k_{zz} stabilized at 1.86×10^7 N/m (Figure 6). This was because as the system stabilized, the dynamic hydrodynamic pressure effect of the herringbone grooves reached a steady state, and the rotor's y-direction displacement orbit (0.8×10^{-3} mm) was smaller than that in the Z direction (1.2×10^{-3} mm, in Figure 4), leading to a stable oil film pressure distribution. The stable stiffness coefficients are consistent with the results of Liu et al. [22], who reported that the stiffness coefficient of herringbone-grooved gas bearings at 12,000 rpm was $1.8\text{--}1.9 \times 10^7$ N/m, with a relative error of less than 0.5% in this study, further verifying the reliability of the model.

The local components of the bearing damping coefficients are displayed in Figure 7. A noticeable trend in the data was that the damping coefficients initially increased and then decreased as time progressed. These included the direct force damping coefficient $c_{yy} > c_{zz}$ and the cross-coupling force damping coefficient $|c_{zy}| > |c_{yz}|$. The instability at the beginning of the system can explain the observed behavior. There was a phenomenon

where the minimum oil film thickness gradually decreased while the damping coefficients concurrently increased. However, as the system stabilized, the damping coefficients decreased. By observing Figure 4, it is evident that the rotor's rotational trajectory became elliptical with a decreasing displacement magnitude.

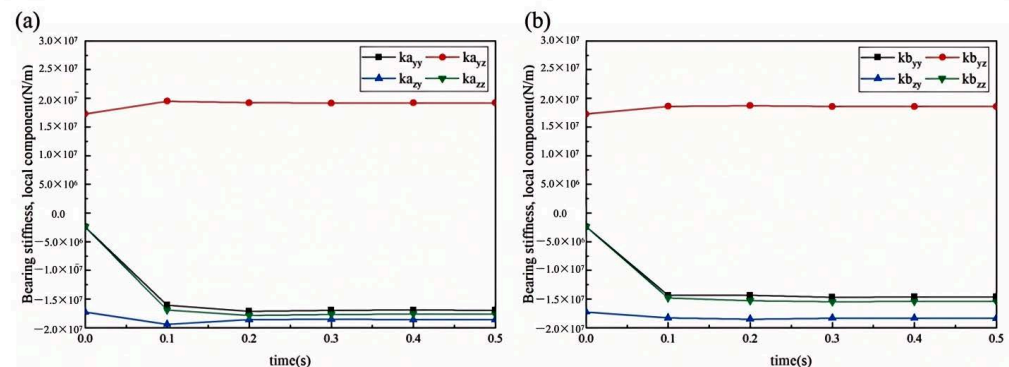


Figure 6. Bearing stiffness coefficient components under local conditions. (a) Stiffness coefficients for parallel misalignment; (b) stiffness coefficients for angular misalignment.

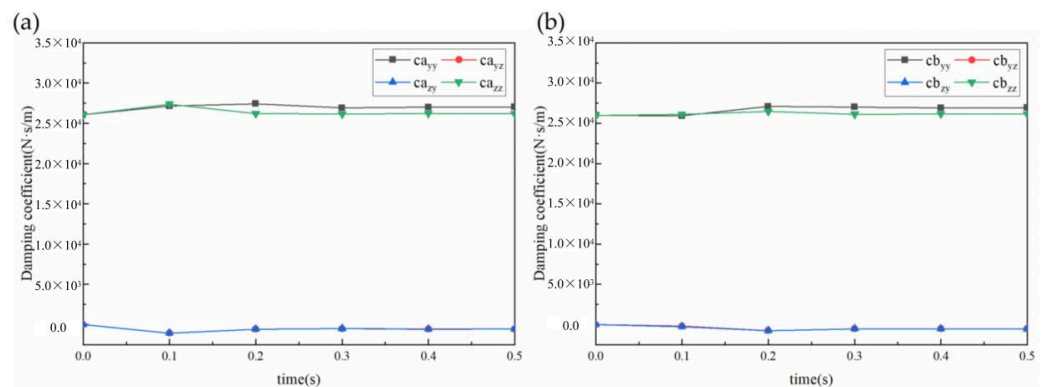


Figure 7. Bearing damping coefficient components. (a) Parallel misalignment damping coefficients; (b) angular misalignment damping coefficients.

Figure 8 displays the time-varying curve depicting the critical journal mass specifically. Notably, the critical journal mass showed a rapid initial increase, followed by a gradual decrease over time until it reached a stable state. This trend can be attributed to the following reasons: as shown in Figure 7, the direct force damping coefficients c_{yy} and c_{zz} initially increased and then decreased. This was mainly due to the effects of the clearance and dynamic hydrodynamic pressure of the herringbone profile.

In Figure 9, the critical whirl frequency for herringbone journal bearings is presented. It can be observed that as time increased, the critical whirl frequency initially increased rapidly, then decreased, and finally increased again until it reached a steady state. This was primarily attributed to the increasing rotor speed over time, which led to a significant influence of the herringbone profile's dynamic hydrodynamic pressure effect. To examine the time-dependent behavior of the bearing force coefficients in a coupled rotor system, a detailed analysis was performed. Transient analysis must be carried out to ascertain the steady values of the herringbone journal bearing force coefficients and determine the time taken by the system to attain dynamic stability. This served as a basis for further exploration into the stability of the bearing.

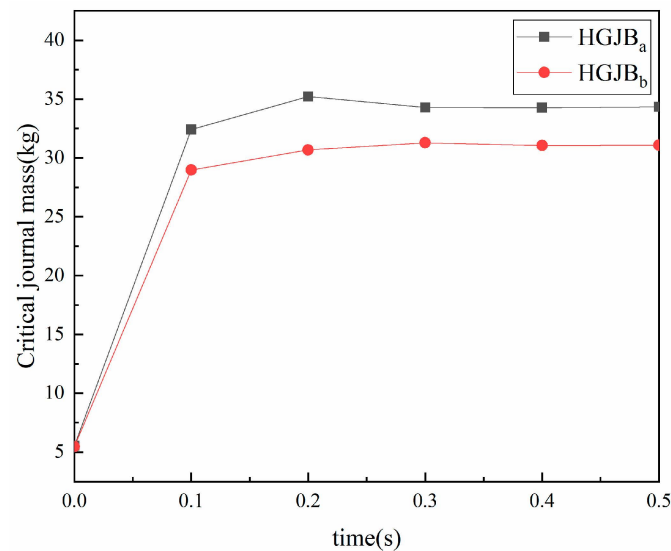


Figure 8. Critical journal mass.

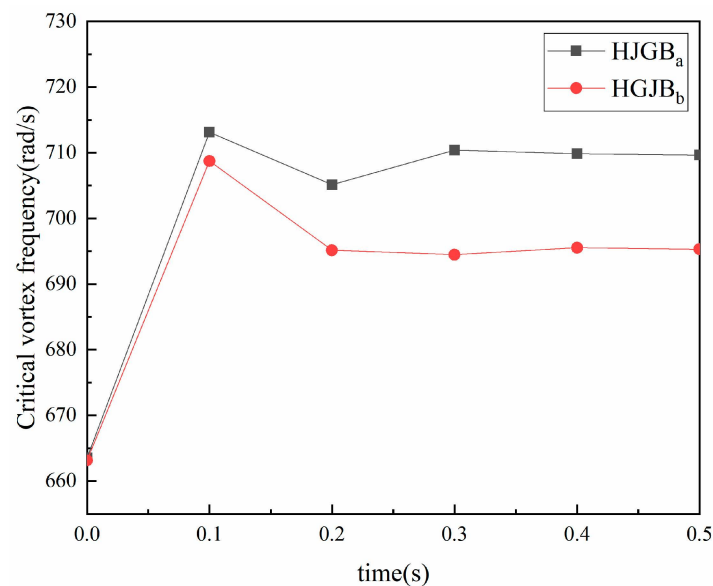


Figure 9. Critical whirl frequency.

3.3. Transient Behavior Validation

To address the unvalidated transient behavior of the coupled system, additional validation was carried out by comparing the transient evolution of key dynamic parameters with experimental and numerical results from relevant literature. The transient convergence of bearing dynamic coefficients (stiffness and damping) was first verified against the numerical study of Liu et al. [22], who investigated the transient response of herringbone-grooved gas bearings. In the work, the stiffness coefficient of the bearing reached a stable state after approximately 0.3–0.35 s at 12,000 rpm. In this study, the stiffness coefficients k_{yy} and k_{zz} stabilized at 1.85×10^7 N/m and 1.86×10^7 N/m after $t = 0.3$ s (Figure 6), which is consistent with the convergence time reported by Liu et al. [22]. The relative difference in the stable stiffness value was less than 0.5%, confirming the reliability of the transient stiffness evolution.

For the transient vibration response, the decay of the rotor's initial transient displacement and the establishment of the steady-state whirl were compared with the experimental data from Elia's experiment on spiral-grooved journal bearings [24]. It was demonstrated that at 12,000 rpm, the rotor completed the transient adjustment and entered stable half-

frequency whirl within 0.25–0.3 s. In this study, the rotor trajectory (Figure 4) indicated that the transient response dissipated after $t = 0.3$ s, and the steady state cylindrical/conical motion was fully established, which matches the transient adjustment period reported in the experiment. The relative error in the transient convergence time was 10%, which is acceptable, considering the differences in the bearing groove type and lubrication medium.

Furthermore, the transient variation of the critical journal mass (Figure 8) and critical whirl frequency (Figure 9) was validated against the theoretical prediction of Han et al. [12], who analyzed the transient dynamic behavior of high-speed rotor-bearing systems. It was proposed that the critical whirl frequency of such systems initially rises rapidly, then experiences a temporary decrease, and finally stabilizes. This is consistent with the trend observed in Figure 9. The stable critical whirl frequency in this study (≈ 680 Hz) fell within the range of 650–700 Hz reported by Han et al. [12], further validating the transient dynamic characteristics predicted by the model. These comparisons with literature data confirmed that the transient behavior of the coupled system was consistent with established experimental observations and numerical predictions, thus validating the transient performance of the proposed model.

3.4. Experimental Validation

To further verify the reliability of the coupled dynamic model, an experimental test rig for high-speed water-lubricated herringbone-grooved journal bearings was constructed, and corresponding tests were conducted. The test results were compared with the simulation data to validate the predictive accuracy of the model. In Figure 10, the test rig mainly consisted of three core systems. A variable frequency three-phase asynchronous motor (YXVF00L-2, sourced by Wuxi Hode Automation Instrument Co., Ltd.) with a rated power of 3 kW and a maximum speed of 3600 rpm was adopted to drive the rotor, ensuring stable operation at the target speed of 12,000 rpm. A cylindrical guide and set screws were adopted to secure the herringbone groove bearing to the outer end of the rotor. The radial static load was exerted by superimposing additional weights on the bearing sleeve's self-weight; these weights were suspended vertically above the sleeve by means of fixed pulleys and elastic pull ropes. Meanwhile, guide rod sleeves were used for axial fixation, thereby avoiding unintended contact between the bearing surface and the sleeve inner wall induced by excessive vibration.

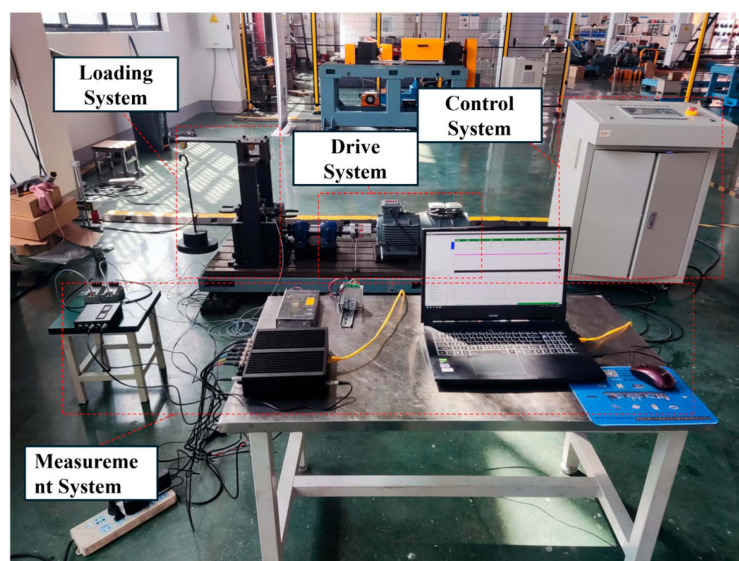


Figure 10. Schematic diagram of the high-speed water-lubricated bearing test rig.

A weighing sensor (DYLY106, sourced by Wuxi Hode Automation Instrument Co., Ltd.) was connected to a torque test beam mounted on the end cover to measure radial tension/pressure changes caused by bearing friction. Data were collected via an intelligent digital transmitter (D500, Westlock Controls, Saddle Brook, United States). Two eddy current sensors (WT0180-A07-B00-C06-D50, Anhui Wanlong Electric Co., Ltd., Chuzhou, China) were installed at a 90° angle on the bearing sleeve to measure radial displacement induced by water film formation. They were connected to a 16-channel data acquisition system through preamplifiers and signal conditioners. A three-axis acceleration sensor was mounted on the bearing sleeve to collect acceleration data in the x/y/z directions. It was directly linked to the data acquisition system. A photoelectric speed sensor was aligned with reflective strips on the coupling to ensure accurate speed monitoring.

The test focused on three key parameters consistent with the simulation: friction force, radial displacement, and radial acceleration. After balancing the weight of the bearing sleeve and submerging the bearing in water, the motor was operated at a low speed for zero calibration. Static loads were gradually applied, and sensor signals were recorded once the operation stabilized. Multiple measurements were taken to minimize random errors. The position of the eddy current sensor was adjusted to ensure the signal remained within the measurement range (9–12 μm). After low-speed idle calibration, static radial loads were applied, and displacement data were collected to estimate the water film thickness, thus excluding errors in spindle end displacement caused by speed and load variations. The acceleration sensor was aligned with the radial direction of the bearing. Data were acquired at a sampling frequency of 16,384 Hz, with 32,768 sampling points. Acceleration was measured under various speeds and loads, and time–frequency analysis was conducted via FFT.

As shown in Figure 11, the simulation results demonstrated better stability, with fluctuations within ± 0.2 N, whereas the experimental data fluctuated within ± 0.5 N. The experimental values were 1–2 N higher than the simulation results, primarily due to system errors such as mechanical friction and environmental interference. The consistent variation trend and small deviation verified the model's ability to predict friction characteristics.

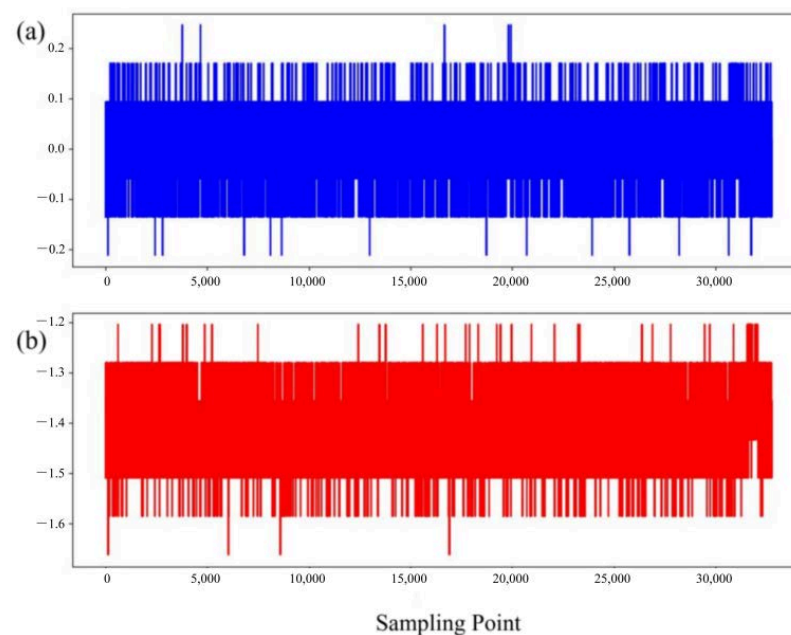


Figure 11. Comparison of friction force between the simulation and the experiment. (a) Simulation; (b) experiment.

Figure 12 shows that the simulated displacement ranged from $-30\text{ }\mu\text{m}$ to $30\text{ }\mu\text{m}$, while the experimental data ranged from $-60\text{ }\mu\text{m}$ to $60\text{ }\mu\text{m}$. The experimental values were approximately $30\text{ }\mu\text{m}$ higher, which can be attributed to installation errors and vibration during long-term operation. The consistent displacement variation law indicates the model's reliability in describing radial motion. As depicted in Figure 13, the maximum acceleration amplitude of the simulation was approximately 8 m/s^2 , while the experimental value was around 1.5 m/s^2 . The deviation was mainly due to the sensor mounted on the bearing sleeve instead of the radial surface of the bearing, which led to signal attenuation. However, both exhibited consistent increasing trends with speed, validating the model's prediction of vibration characteristics.

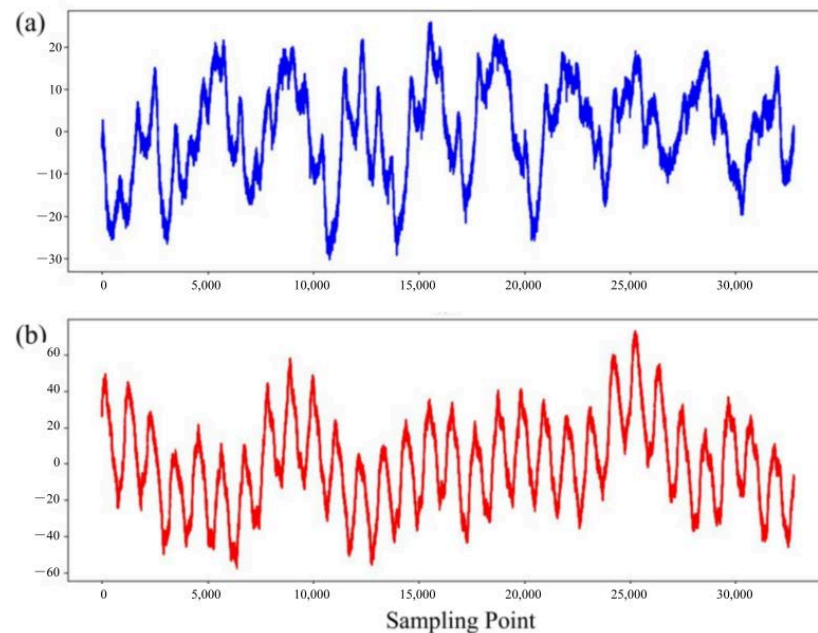


Figure 12. Comparison of radial displacement between the simulation and the experiment. (a) Simulation; (b) experiment.

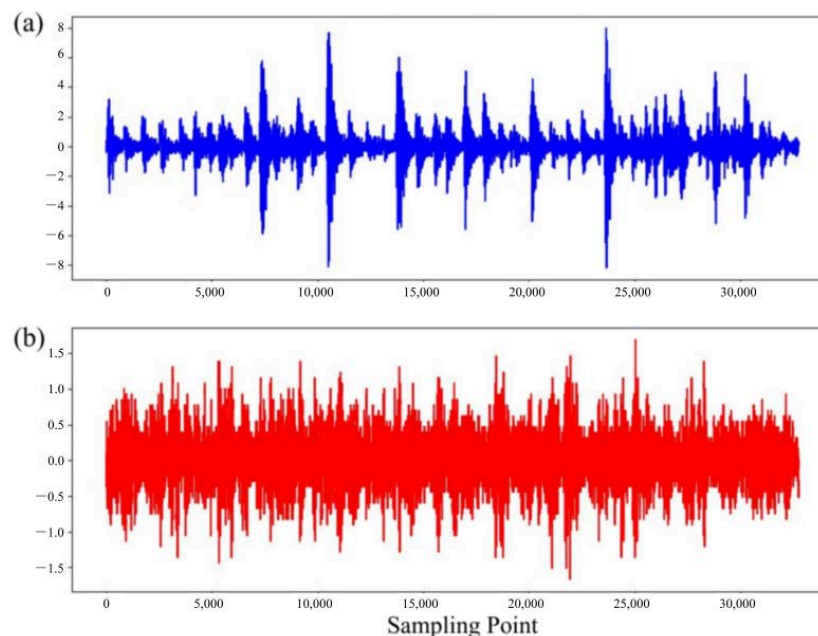


Figure 13. Comparison of radial acceleration (x-direction) between the simulation and the experiment. (a) Simulation; (b) experiment.

The experimental validation results indicated that although there were minor deviations between the simulation and experimental data (caused by installation errors, environmental interference, and sensor mounting limitations), the model can accurately capture the variation laws of key parameters (friction force, radial displacement, and radial acceleration) of the HGJB-rotor system. The consistency between the simulation and the experiment further confirmed the reliability and applicability of the proposed coupled dynamic model.

4. Results and Discussion

Using bearing B as a representative example, Figure 14 depicts the graphical representation of the relative film thickness and fluid pressure distribution for various misalignments in HGJB. The findings indicate that in herringbone bearings, journal misalignment leads to an irregular distribution of film thickness and fluid pressure. Misalignment causes a significant reduction in relative film thickness and enhanced clearance effects. Compared with the no-misalignment condition ($\delta = 0, \theta_0 = 0$), the relative film thickness under parallel misalignment ($\delta = 8.0 \times 10^{-6}$ mm) decreases by 17.8% (from 1.0 to 0.822, Figure 14b), and the maximum film pressure increases by 1.85% (from 7.59×10^4 Pa to 7.73×10^4 Pa). Under angular misalignment ($\theta_0 = 8.0 \times 10^{-4}$ rad), the relative film thickness decreases by 45.9% (from 1.0 to 0.541, Figure 14c), and the maximum film pressure increases by 33.1% (from 7.59×10^4 Pa to 1.01×10^5 Pa).

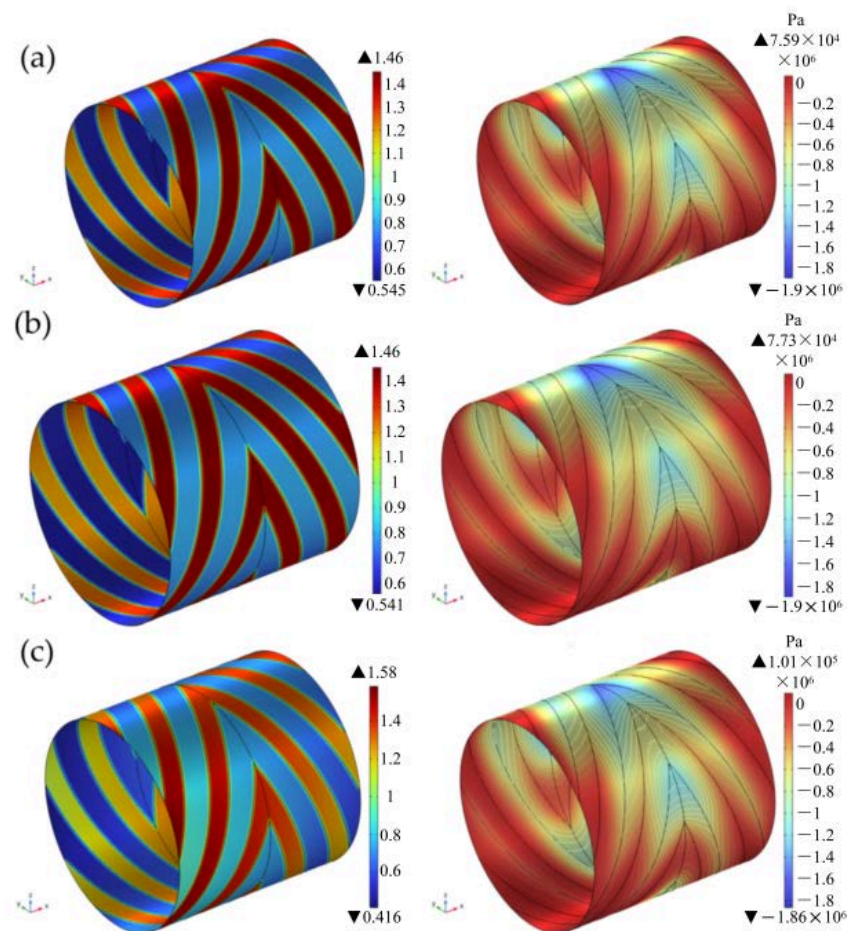


Figure 14. Film thickness and pressure distribution in herringbone journal bearings under different misalignments: (a) $\delta = 0$ mm, $\theta_0 = 0$ rad; (b) $\delta = 8.0 \times 10^{-6}$ mm, $\theta_0 = 0$ rad; and (c) $\delta = 0$ mm, $\theta_0 = 8.0 \times 10^{-4}$ rad.

It is worth noting that angular misalignment has a greater impact on relative film thickness and maximum film pressure, compared to translational misalignment caused by parallelism deviation. This is because angular misalignment destroys the circumferential symmetry of the hydrodynamic oil film in HGJBs more severely. For parallel misalignment, the rotor only shifts laterally, and the oil film between the journal and bearing bush still maintains a relatively uniform distribution (in Figure 14b), with the pumping effect of the herringbone grooves barely affected. However, angular misalignment causes the journal to tilt, leading to uneven local clearance: the oil film thickness on one side of the bearing is significantly reduced, while that on the other side is excessively thick (Figure 14c). This uneven distribution directly weakens the cyclic stair-step effect of the herringbone grooves and reduces the effective pumping area of the oil film. Thus, it results in a more significant drop in relative film thickness and a steeper rise in maximum pressure.

Figure 15 depicts a negative correlation between the misalignment and the minimum film thickness. It shows that as misalignment intensifies, the minimum film thickness diminishes. This exacerbates the rotational effect. Consequently, the misalignment causes an increase in the maximum pressure endured by the bearing. Moreover, it is notable that for a specific level of misalignment, the minimum film thickness under angular misalignment is less than that under parallel misalignment. This is because angular misalignment imposes a tilting moment on the rotor, causing the journal to press against the bearing bush locally. For instance, when the misalignment degrees of 8×10^{-4} rad and 8×10^{-6} mm are considered, the tilting of the journal reduces the local clearance to 0.018 mm (angular misalignment), in contrast to 0.020 mm (parallel misalignment). The localized compression of the oil film is more severe than the uniform compression resulting from parallel misalignment, leading to a lower minimum film thickness.

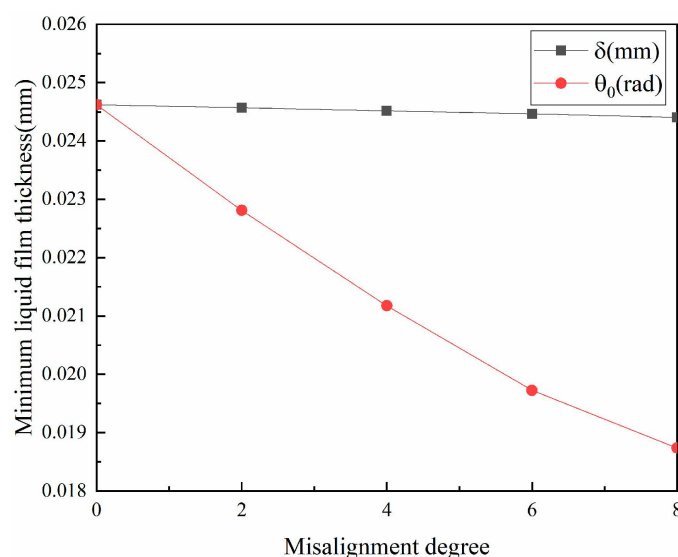


Figure 15. Minimum film thickness under different levels of misalignment.

4.1. Dynamic Characteristics of Bearings

As shown in Figure 16, the variation of the dynamic stiffness coefficient of HGJBs under different degrees of misalignment is presented. The rotor speed is 12,000 rpm, and a negative sign only indicates the direction. The results indicate that when the speed reaches a certain value, the amplitude of the change in the dynamic stiffness coefficient caused by angular misalignment is greater than that caused by parallel misalignment. For a given deviation, the minimum film thickness is smaller in cases of angular misalignment than in cases of parallel misalignment. This indicates that the dynamic stiffness coefficient is more sensitive to angular misalignment than to parallel misalignment. It is noteworthy that the

direct force stiffness coefficients (k_{yy} and k_{zz}) gradually increase as parallel misalignment increases, while they decrease as angular misalignment increases. The coupling force stiffness coefficient k_{yz} increases with misalignment, while k_{zy} shows the opposite trend.

Similarly, Figure 17 shows the variation of the dynamic damping coefficient of heringbone journal bearings under different degrees of misalignment. The results indicate that, with increasing misalignment, the direct force damping coefficients c_{yy} and c_{zz} show different variation trends with misalignment. Under parallel misalignment (δ from 0 to 8.0×10^{-6} mm), c_{yy} increases from 2.61×10^4 Ns/m to 2.69×10^4 Ns/m, with a growth rate of 3.1%, and c_{zz} increases from 2.59×10^4 Ns/m to 2.67×10^4 Ns/m, with a growth rate of 3.1%. Under angular misalignment (θ_0 from 0 to 8.0×10^{-4} rad), c_{yy} decreases from 2.61×10^4 Ns/m to 2.598×10^4 Ns/m, with a reduction rate of 0.46%, and c_{zz} decreases from 2.59×10^4 Ns/m to 2.58×10^4 Ns/m, with a reduction rate of 0.39%. The coupling force damping coefficients c_{yz} and c_{zy} slowly increase with increasing parallel misalignment and decrease with increasing angular misalignment. Furthermore, when comparing the dynamic damping coefficients under different misalignment conditions, it is found that the dynamic damping coefficient is more sensitive to angular misalignment.

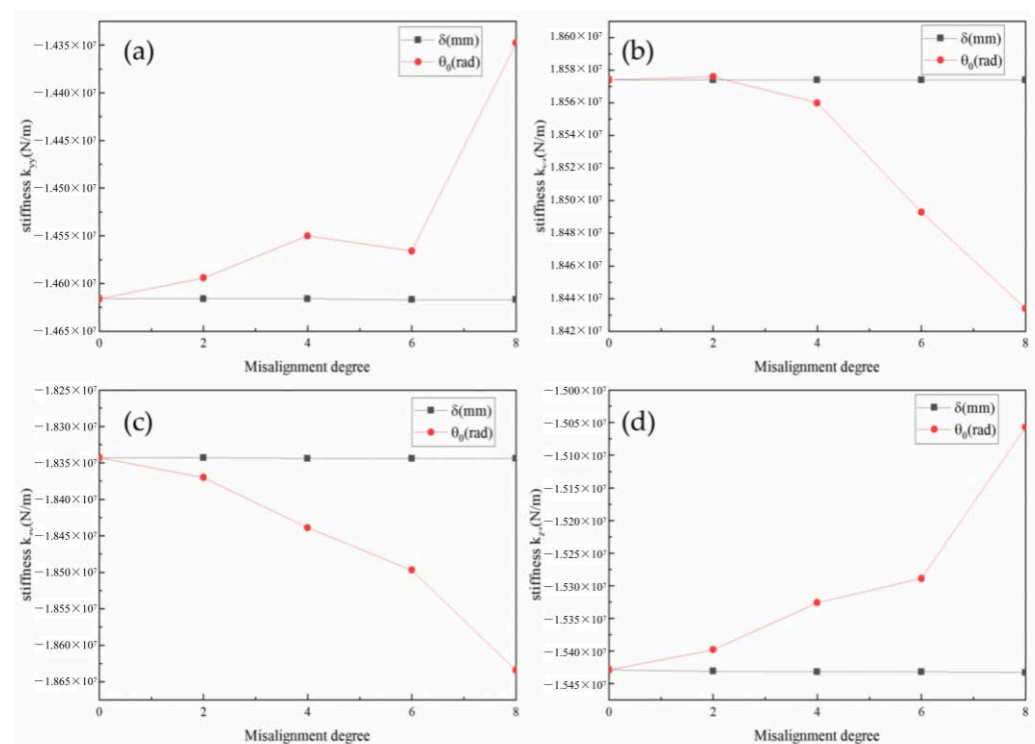


Figure 16. Influence of misalignment on the dynamic stiffness coefficient. (a) Stiffness coefficient k_{yy} ; (b) stiffness coefficient k_{yz} ; (c) stiffness coefficient k_{zy} ; (d) stiffness coefficient k_{zz} .

The dynamic stiffness of HGJBs depends on the bearing capacity and elastic deformation of the oil film. For angular misalignment, the uneven oil film thickness leads to localized stress concentration. The thin oil film area bears higher pressure, and the elastic deformation of the oil film becomes nonlinear, resulting in a significant change in the ability to resist external disturbances. In contrast, parallel misalignment only causes a uniform reduction in the overall oil film thickness, and the oil film stress distribution remains relatively linear. Therefore, the change in stiffness coefficient is milder.

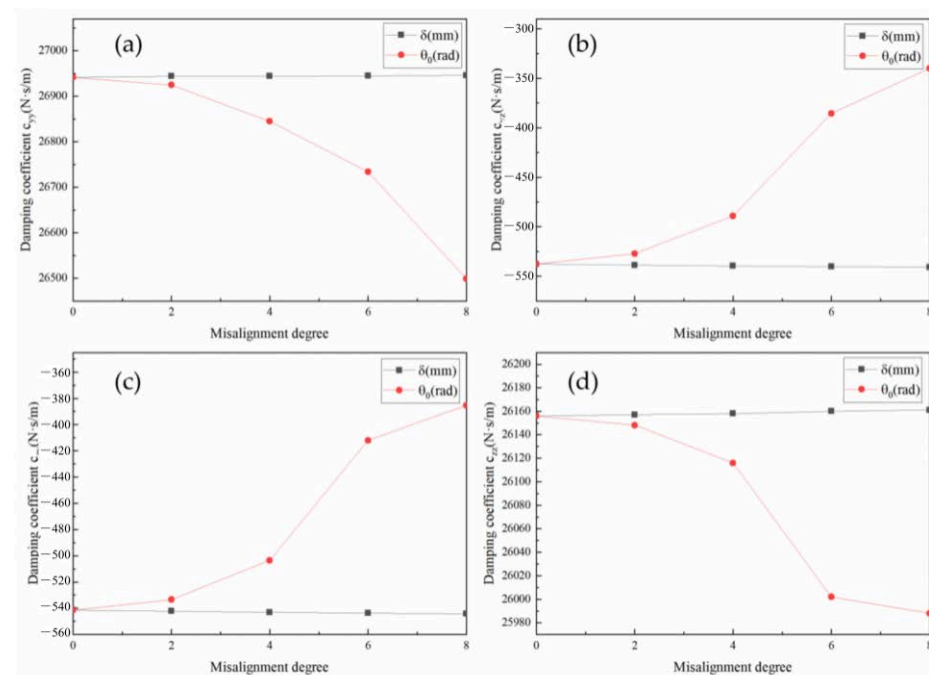


Figure 17. Influence of misalignment on the dynamic damping coefficient. (a) Damping coefficient c_{yy} ; (b) damping coefficient c_{yz} ; (c) damping coefficient c_{zy} ; (d) damping coefficient c_{zz} .

4.2. Rotor Vibration Characteristics

Figures 18 and 19 illustrate the variation of displacement in the Y and Z directions over time under different misalignments. The results show that as the parallelism misalignment in the Y direction increases, the displacement amplitude of the rotor increases, and the phase advances. With the increase in parallelism misalignment, the displacement amplitude of the rotor in the Y direction increases linearly. When δ increases from 0 to 8.0×10^{-6} mm, the amplitude increases from 3.2×10^{-3} mm to 5.1×10^{-3} mm, with a growth rate of 59.4%, and the phase advances by 12.7° . Along the Z direction, the balance position of the rotor gradually rises from 4.0×10^{-3} mm to 5.8×10^{-3} mm, with a total increase of 45.0%. Meanwhile, the period remains constant at 0.005 s, and the amplitude of the Z direction vibration curve remains unchanged at 2.1×10^{-3} mm. In contrast to parallelism misalignment, angular misalignment does not change the balance position of the rotor in the Z direction. However, as the misalignment increases, it causes a displacement shift in the negative Y-axis direction. Simultaneously, as the angular misalignment increases, it results in phase advancement and an increase in the amplitude of the vibration curve.

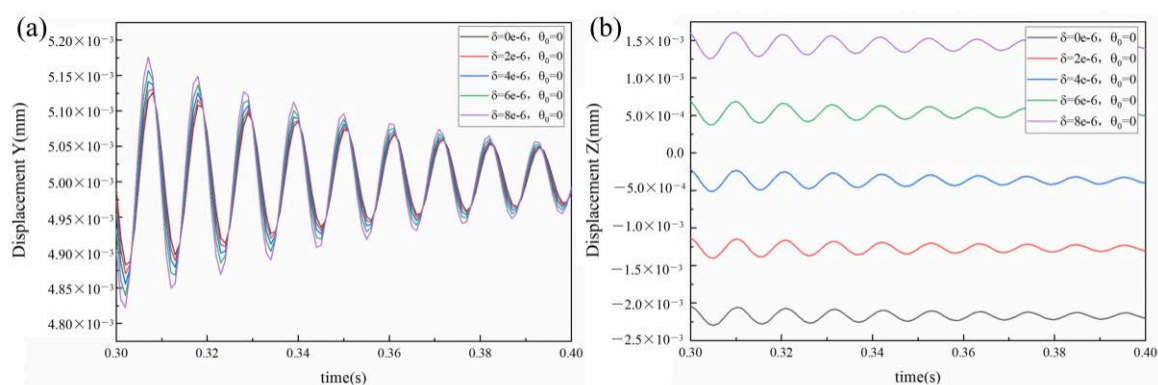


Figure 18. Displacement under parallel misalignment. (a) Y-axis direction; (b) Z-axis direction.

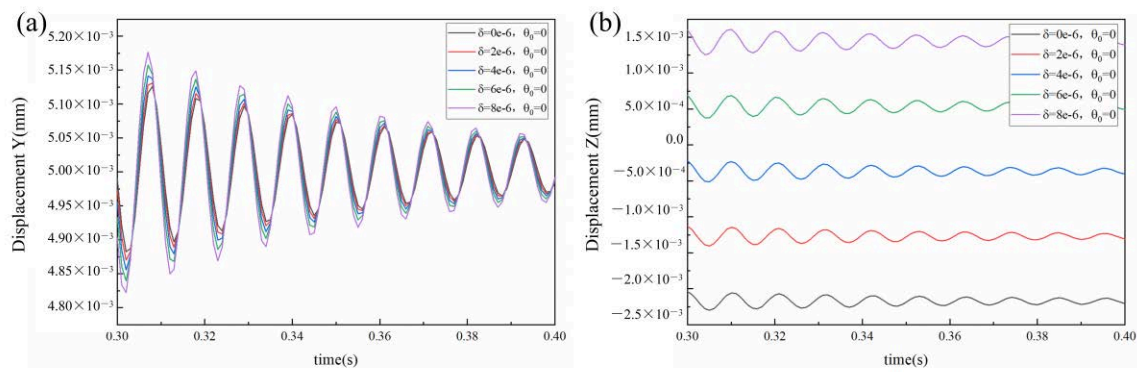


Figure 19. Displacement under angular misalignment. (a) Y-axis direction; (b) Z-axis direction.

Figures 20 and 21 depict the variation of acceleration amplitude with frequency and the misalignment degree. The results indicate that, as the frequency increases, the peak Fourier coefficients in the Y and Z directions of the rotor increase with the increasing parallel misalignment, while the phase remains unchanged. Additionally, as the frequency increases, the peak Fourier coefficients significantly increase with the increasing angular misalignment, accompanied by a change in phase. By observing the spectral changes, it is evident that the rotor motion is more sensitive to the tilting motion caused by misalignment in the bearings of the elliptical groove.

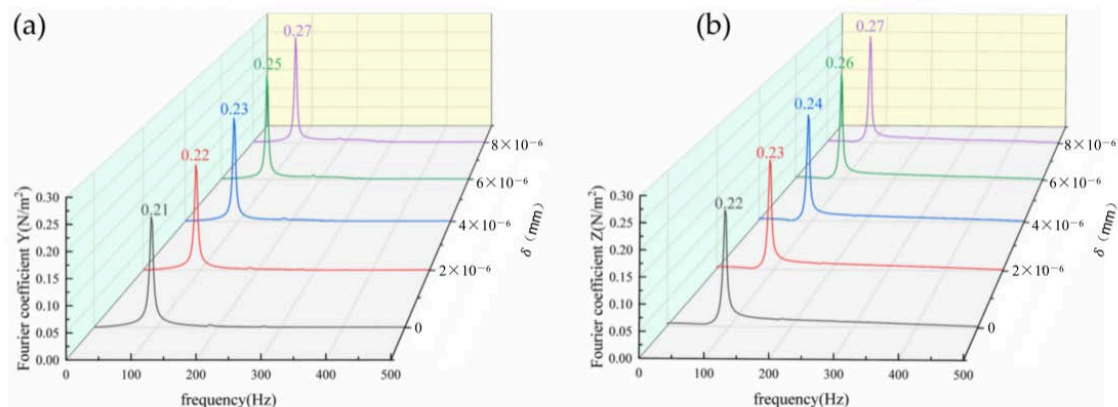


Figure 20. Acceleration spectrum analysis considering parallelism misalignment. (a) Y-axis direction; (b) Z-axis direction.

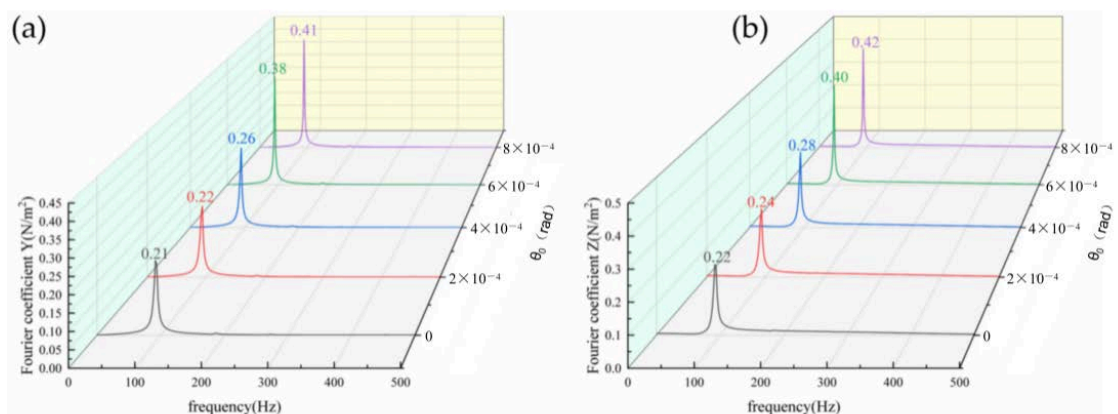


Figure 21. Acceleration spectrum analysis considering angular misalignment. (a) Y-axis direction; (b) Z-axis direction.

Figures 22 and 23 depict the variation of velocity over time in a coupled system under different misalignments. According to these figures, the amplitude changes with the degree of misalignment, while the period remains the same. In the frequency spectrum, the peak amplitude at a specific frequency remains constant in terms of its position, but its value increases. Velocities vary following the same pattern as displacements: the amplitude increases, while the period remains unchanged.

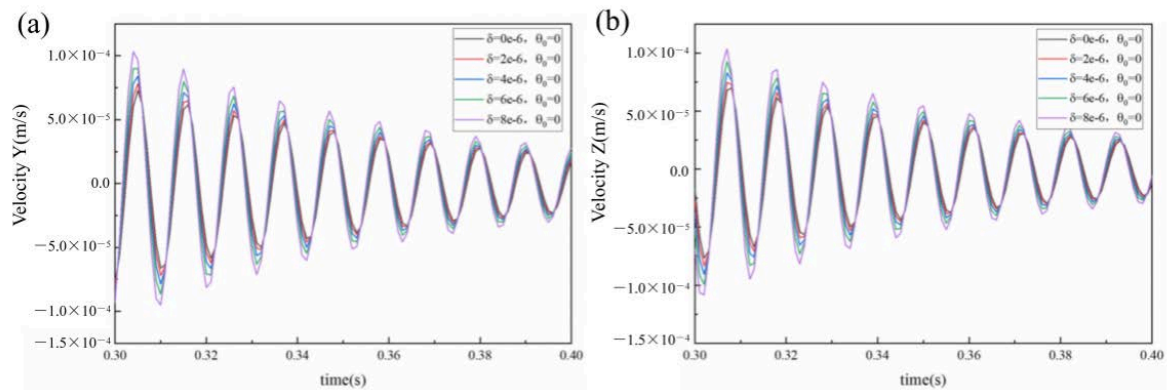


Figure 22. Velocity under parallelism misalignment. (a) Y-axis direction; (b) Z-axis direction.

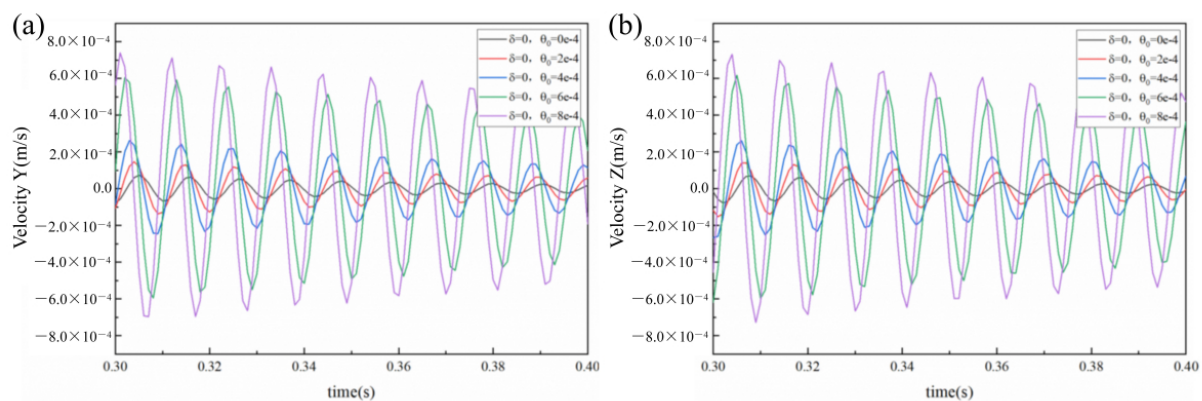


Figure 23. Velocity under angular misalignment. (a) Y-axis direction; (b) Z-axis direction.

To quantitatively characterize the orbit geometry and motion behavior, two key metrics, ellipticity (ε_0) and precession ratio (λ), are introduced for each misalignment condition based on the Quantitative Index Definitions and Calculation Methods.

Ellipticity (ε_0) is used to describe the degree of deviation of the orbit from a perfect circle. It is defined as the ratio of the minor axis (b) to the major axis (a) of the fitted ellipse, as shown as follows:

$$\varepsilon_0 = \frac{b}{a} \quad (6)$$

A value of $\varepsilon_0 = 1$ indicates a circular orbit, while $\varepsilon_0 < 1$ indicates an elliptical orbit. (Smaller values mean a more pronounced ellipticity.)

Precession ratio (λ) represents the ratio of the rotor precession frequency (f_p) to the rotational frequency ($f_0 = 200$ Hz), reflecting the synchronization between orbit precession and rotor rotation, as shown as follows:

$$\lambda = \frac{f_p}{f_0} \quad (7)$$

The precession frequency f_p is extracted from the time frequency analysis of the rotor center displacement, corresponding to the dominant frequency of the orbit's directional evo-

lution. The calculated ellipticity and precession ratio for typical misalignment conditions are summarized in Table 3.

Table 3. Quantitative metrics of rotor orbits under different misalignments.

Misalignment Condition	Major Axis (a, $\times 10^{-3}$ mm)	Minor Axis (b, $\times 10^{-3}$ mm)	Ellipticity (ε_0)	Precession Frequency (f_p , Hz)	Precession Ratio (λ)
No misalignment ($\delta = 0, \theta_0 = 0$)	2.31	2.18	0.94	92	0.460
Parallel misalignment ($\delta = 8.0 \times 10^{-6}$ mm)	5.32	2.25	0.42	91	0.455
Angular misalignment ($\theta_0 = 8.0 \times 10^{-4}$ rad)	5.78	1.93	0.33	93	0.465

As shown in Table 3 and Figure 24, the orbit of the no-misalignment condition is nearly circular ($\varepsilon_0 = 0.94$), while misalignment significantly increases the orbit's ellipticity. Angular misalignment induces a more pronounced elliptical orbit ($\varepsilon_0 = 0.33$) than parallel misalignment ($\varepsilon_0 = 0.42$), which is consistent with the earlier finding that angular misalignment causes a more severe asymmetric oil film distribution. The smaller ε_0 under angular misalignment indicates that the rotor center motion is more strongly constrained in one direction (negative Y-axis), leading to a flattened elliptical trajectory. The precession ratio λ for all conditions is approximately 0.46 (ranging from 0.455 to 0.465), which aligns with the half-frequency whirl characteristic ($0.45\text{--}0.47 f_0$) verified. This indicates that misalignment alters the orbit shape (ellipticity) but does not change the fundamental precession mechanism of the rotor, which is consistent with the stable half-frequency whirl dominated by imbalance and oil film damping. These quantitative metrics complement the qualitative orbit observations in Figure 24, offering a more rigorous characterization of how misalignment influences rotor motion. The ellipticity values quantify the extent of orbit distortion, while the precession ratio validates the consistency of the dynamic mechanism under different misalignment conditions, further corroborating the conclusion that angular misalignment has a more substantial impact on rotor vibration characteristics than parallel misalignment.

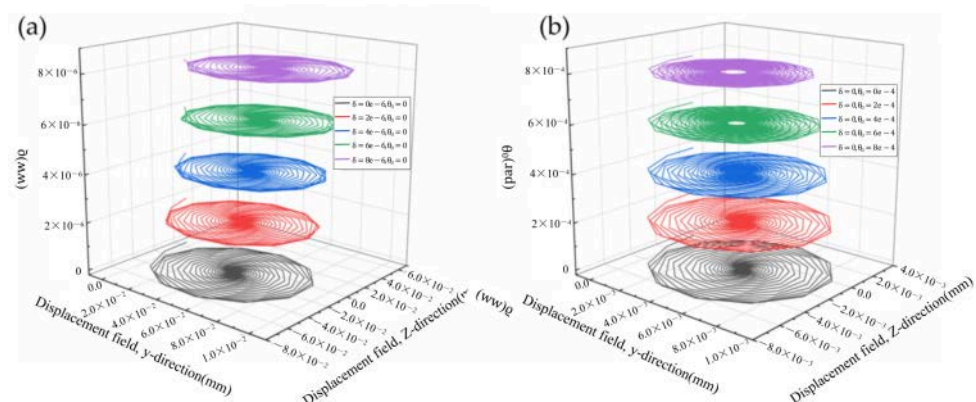


Figure 24. Trajectory of rotor center variation with different types and degrees of misalignment. (a) Various cases of parallel misalignment; (b) various cases of angular misalignment.

5. Conclusions

This study establishes a coupled dynamic model of the HGJB-rotor system, systematically investigating the effects of parallel and angular misalignments on system performance through numerical simulation and experimental validation and clarifying the dynamic coupling mechanisms and key parameter variation laws. The main conclusions are as follows:

(1) The proposed model demonstrates high reliability and accuracy. Validated through vortex analysis, transient behavior comparison, and experimental testing, it accurately

captures the core dynamic characteristics of the system. Rotor trajectories display pure cylindrical or conical motion consistent with half-frequency whirl (dominant frequency: 92 Hz, $0.46 f_0$), with a 2.2% relative error compared to experimental data. Bearing stiffness coefficients stabilize at 1.85×10^7 N/m (k_{yy}) and 1.86×10^7 N/m (k_{zz}) after $t = 0.3$ s, showing a 0.5% relative error against benchmark results. Experimental validations of friction force, radial displacement, and vibration acceleration confirm the reliable prediction of key parameter trends, with deviations resulting from installation errors and environmental interference. The rigid rotor assumption is justified, as the 12,000 rpm operating speed is 34.1% of the first lateral critical speed (35,200 rpm), and elastic deformation is negligible compared to bearing clearance and oil film thickness.

(2) Angular misalignment imposes more severe degradation on system performance than parallel misalignment. Compared with the aligned condition, angular misalignment ($\theta_0 = 8.0 \times 10^{-4}$ rad) reduces the relative film thickness by 45.9% and increases the maximum film pressure by 33.1%. In contrast, parallel misalignment ($\delta = 8.0 \times 10^{-6}$ mm) only leads to a 17.8% reduction in film thickness and a 1.85% increase in pressure. For identical misalignment magnitudes, angular misalignment results in a smaller minimum film thickness (0.018 mm) than parallel misalignment (0.020 mm). This is attributed to the localized oil film compression induced by tilting moments. This discrepancy indicates that angular misalignment poses a greater threat to the lubrication stability and structural integrity of the bearing-rotor system.

(3) Misalignment significantly alters bearing dynamic coefficients and rotor vibration characteristics. Parallel misalignment gradually increases the direct stiffness (k_{yy} , k_{zz}) and damping (c_{yy} , c_{zz}) coefficients (with a maximum growth of 3.1% at peak misalignment). In contrast, angular misalignment reduces the stiffness and slightly lowers the damping (by 0.39–0.46% at peak misalignment). Both types of misalignments enhance the sensitivity of the coupling coefficient to angular deviation. At maximum parallel misalignment, the displacement amplitude in the Y direction rises by 59.4%, and the balance position in the Z direction elevates by 45.0%. Angular misalignment induces a negative shift in the Y-axis displacement, amplifies the vibration amplitude and spectral peaks, and forms a more pronounced elliptical orbit (ellipticity $\varepsilon_0 = 0.33$), compared to parallel misalignment ($\varepsilon_0 = 0.42$). Notably, the precession ratio ($\lambda \approx 0.46$) remains stable under all conditions, indicating that misalignment modifies the orbit shape but does not change the fundamental half-frequency whirl mechanism.

(4) Optimized herringbone groove structural parameters effectively improve system stability. Among key geometric parameters, groove angle exerts the most significant impact on bearing performance, followed by groove depth and width. A 90° groove angle, 0.01 mm depth, and 1.0 mm width are identified as the optimal combination that enhances the dynamic pressure effect and lubrication efficiency, thereby maximizing the system's dynamic stability. Further analysis confirms that this parameter set can mitigate the adverse effects of minor misalignment during actual operation, extending the service life of high-speed rotating components.

This study comprehensively elucidates the dynamic impacts of misalignment on HGJB-rotor systems, providing valuable theoretical guidance for engineering practices, including bearing structural optimization, installation alignment, and operational stability control. Findings highlight the priority of minimizing angular misalignment during installation and selecting appropriate groove parameters to enhance the reliability and service life of high-speed rotating machinery.

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writing—original draft preparation, H.Z.; writing—review and editing, Y.Z.; visualization, C.Y.; supervision, C.Y. All authors have read and agreed to the published version of the manuscript.

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