

Article

Experimental Study on the Dynamic Characteristics of Needle Roller Bearings Under Periodic Impact Load

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Abstract

Needle roller bearings are characterized by rolling elements with relatively high length-to-diameter ratios and are widely used in mechanical systems with limited radial installation space. In gear transmission systems, periodic impact loads induced by gear meshing may be superimposed on steady radial loads, thereby altering the dynamic response of the bearing. However, the effects of the amplitude and frequency of periodic impact loading on the dynamic characteristics of needle roller bearings remain insufficiently understood. In this study, a needle roller bearing test rig capable of applying periodic impact loading was developed, and a multi-sensor measurement system was configured to measure outer-ring vibration, inner-ring motion, cage motion, and the friction torque of the bearing system. Dynamic tests were conducted under different amplitudes and frequencies of periodic impact loading. A reduction in bearing motion stability was observed under periodic impact loading, as evidenced by increased outer-ring vibration and enlarged cage motion in both the horizontal and vertical directions. As the loading amplitude increased, the RMS values of outer-ring acceleration, inner-ring displacement, and cage displacement increased, while an increase in the mean friction torque was also observed. The inner-ring trajectory expanded along the loading direction, and the whirling range of the cage trajectory increased. As the loading frequency increased, the RMS values of outer-ring acceleration, inner-ring displacement, and cage displacement increased, and the mean friction torque increased. In contrast, the whirling range of the cage trajectory decreased. These findings clarify the distinct effects of periodic impact-loading amplitude and frequency on bearing vibration, internal motion, and friction-torque characteristics and provide experimental support for the dynamic performance evaluation of needle roller bearings under impact conditions.

Keywords: needle roller bearing; periodic impact load; needle roller bearing test rig; dynamic characteristics



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1. Introduction

Needle roller bearings are characterized by large length-to-diameter ratios and are widely used in mechanical systems where radial installation space is limited and radial loads must be supported. In such systems, the dynamic performance of the bearing directly affects the operational behavior of the supported system and is strongly dependent on operating conditions. During operation, periodic impact excitations induced by gear meshing, rotor unbalance, or misalignment can be superimposed on the steady radial load.

These excitations can alter the bearing dynamic characteristics and may increase the risk of operational instability, jamming, and other failure modes [1].

Extensive studies have been conducted on bearing dynamics and the dynamic behavior of bearings under Impact Load. From the perspective of theoretical modeling, Ren [2] developed a dynamic model that incorporates the geometric constraints of cage pockets and multi-flexible-body coupling and introduced a half-sine impact function to obtain the variable-stiffness characteristics of rolling bearings under transient excitation. To further characterize cage dynamics under Impact Load, Guo [3], Huang [4], and Yao [5,6] employed nonlinear explicit dynamics methods (ANSYS/LS-DYNA) and the discrete element method (DEM) to simulate contact and collision between the cage and rolling elements, thereby clarifying the effect of impact excitation on the whirling trajectory of the cage centroid. Ma [7] further coupled variable-speed parametric equations and used numerical integration to capture frequency-bifurcation-induced energy concentration in the cage motion trajectory. Li [8] used an explicit finite element method (FEM) to analyze the evolution of local contact stress waves on the cage pocket wall during impact-load transmission. Wang [9] clarified the modulation effect of load fluctuation frequency on unstable cage whirling based on nonlinear stability analysis. Regarding the coupling between frictional energy consumption and lubrication interfaces, Han [10] and Tu [11] established roller-cage dynamic models based on mixed lubrication theory under time-varying loads and numerically analyzed friction power loss under different loading conditions. Lv [12], Zhang [13], and Miao [14] used the thermal-fluid–solid interaction (TFSI) method to reveal the effects of fluid inertia and turbulent stress on the evolution of lubrication film stiffness under impact disturbance.

In terms of experimental testing and validation, Shi [15] developed a needle roller bearing impact test rig and achieved in situ measurement of cage deformation under Impact Load. Savolainen [16] verified the feasibility of outer-ring fault identification based on the spectral characteristics of acceleration signals, providing an important reference for bearing condition monitoring.

Although substantial progress has been made in understanding the dynamic behavior of bearings under periodic impact loading, previous studies have focused primarily on theoretical modeling and numerical simulation, while experimental validation remains limited. In addition, most experimental investigations have relied mainly on outer-ring vibration measurements, whereas the dynamic responses of the inner ring and cage under strong periodic impact excitation have not been adequately characterized. Consequently, the effects of periodic impact loading and variations in its amplitude and frequency on the overall dynamic behavior of needle roller bearings remain insufficiently understood.

Accordingly, a test rig capable of applying periodic impact loading was developed, and a multi-sensor measurement system was configured to measure outer-ring vibration, inner-ring motion, cage motion, and bearing friction torque. The effects of periodic impact-loading amplitude and frequency on the dynamic characteristics of needle roller bearings were systematically investigated. The proposed testing method and the resulting findings provide a basis for understanding and evaluating the dynamic performance of needle roller bearings under periodic impact loading.

2. Needle Roller Bearing Test Rig for Periodic Impact Load and Dynamic Testing

2.1. Needle Roller Bearing Test Rig

As shown in Figure 1, the needle roller bearing test rig mainly consists of a motor, a coupling, a torque sensor, a support device, a loading bearing, a spindle, a loading device, and the bearing under test.

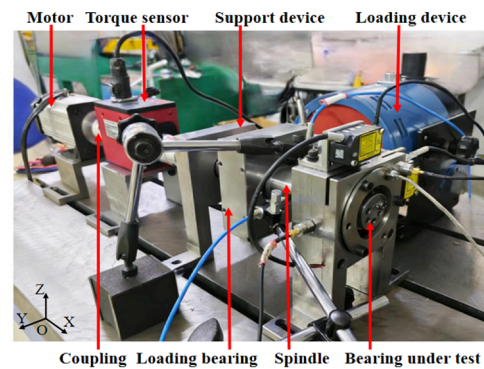


Figure 1. Bearing test rig.

The drive system of the test rig uses a brushless DC motor with stepless speed regulation as the power source. Power is transmitted to the spindle through a coupling, and a torque sensor is installed to acquire the fluctuations in the bearing-system friction torque in real time. The spindle system is rigidly supported by an angular-contact ball bearing mounted in a fixed seat (model: SKF 6005). This bearing arrangement carries the operating load and effectively constrains the axial displacement of the shaft system. The external radial load is applied through an independent loading seat, in which a cylindrical roller bearing (model: SKF N207) is installed to ensure stable loading. The test bearing was a needle roller bearing based on the nominal $K38 \times 46 \times 32$ roller assembly, but the standard inner and outer rings were replaced with custom-made rings. In particular, the actual inner-ring bore diameter was 30 mm, which was 8 mm smaller than the nominal 38 mm diameter of the standard assembly. Therefore, the ring dimensions and the associated contact geometry differed from those of the standard $K38 \times 46 \times 32$ configuration. The actual geometric parameters of the test bearing are listed in Table 1.

Table 1. Actual geometric parameters of the test bearing.

Parameter	Value	Unit
Number of rolling elements	20	
Outer raceway diameter	46	mm
Radial clearance	0.064	mm
Rolling element diameter	4	mm
Total rolling element length	25.5	mm
Inner raceway diameter	30	mm

2.2. Implementation of Periodic Impact Loading

Periodic impact load is usually generated by periodic excitations associated with rotor misalignment and unbalance in actual operating conditions. To apply periodic impact load, an independent impact exciter was arranged along the Y direction of the inertial coordinate system in the test rig. The loading schematic is shown in Figure 2a, where the output push rod of the exciter is connected to the loading bearing seat by bolts. The periodic impact load is accurately applied to the spindle system, thereby imposing periodic impact load on the bearing under test. The physical setup is shown in Figure 2b, and the system consists of an exciter, a force sensor, a support bearing, a loading bearing, and the bearing under test.

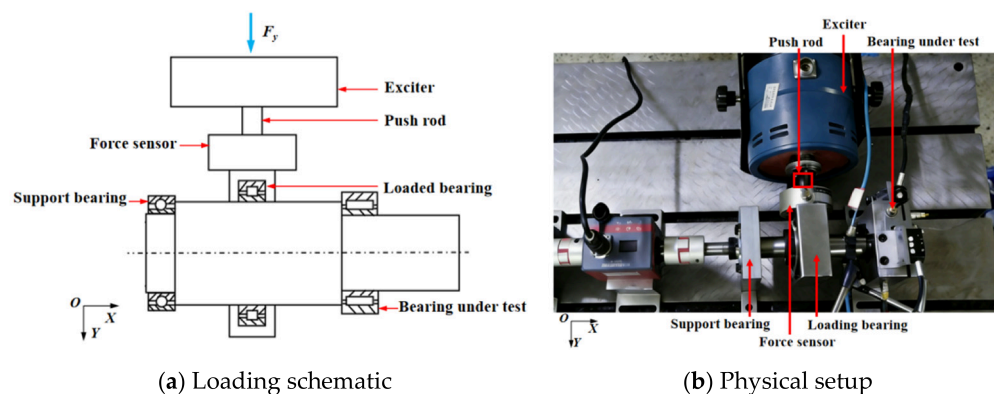


Figure 2. Periodic impact-loading system.

2.3. Multi-Physical-Quantity Testing of Bearing Dynamics

(1) Testing method

Multiple sensors were arranged to measure the dynamic characteristics of the bearing, including the cage displacement, inner-ring displacement, bearing vibration, and bearing-system friction torque. The sensor arrangement is shown in Figure 3, and the sensor composition and specifications are listed in Table 2.

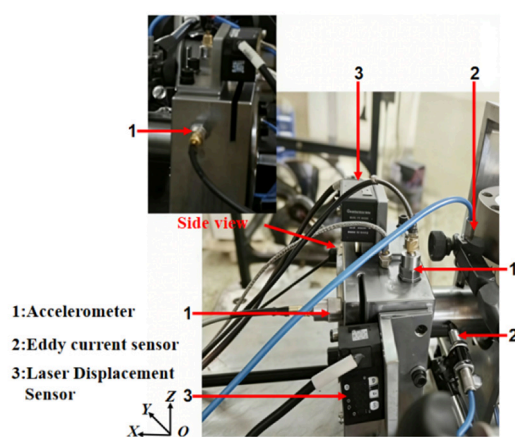


Figure 3. Sensor arrangement.

An integrated optical sensing scheme was designed to measure cage displacement because the measurement of cage motion in needle roller bearings is constrained by the outer ring. Four symmetric observation windows were precision-machined on the outer ring at equal angular intervals of 90° , with each window having a width of 16 mm and a depth of 7.5 mm. The modified outer ring is shown in Figure 4a and provides the necessary measurement channels for cage displacement measurement. As shown in the schematic in Figure 4b, two laser displacement sensors were arranged orthogonally to enable laser incidence on and reflection from the cage surface, thereby allowing the cage displacement to be measured. The detailed sensor arrangement is shown in Figure 4c.

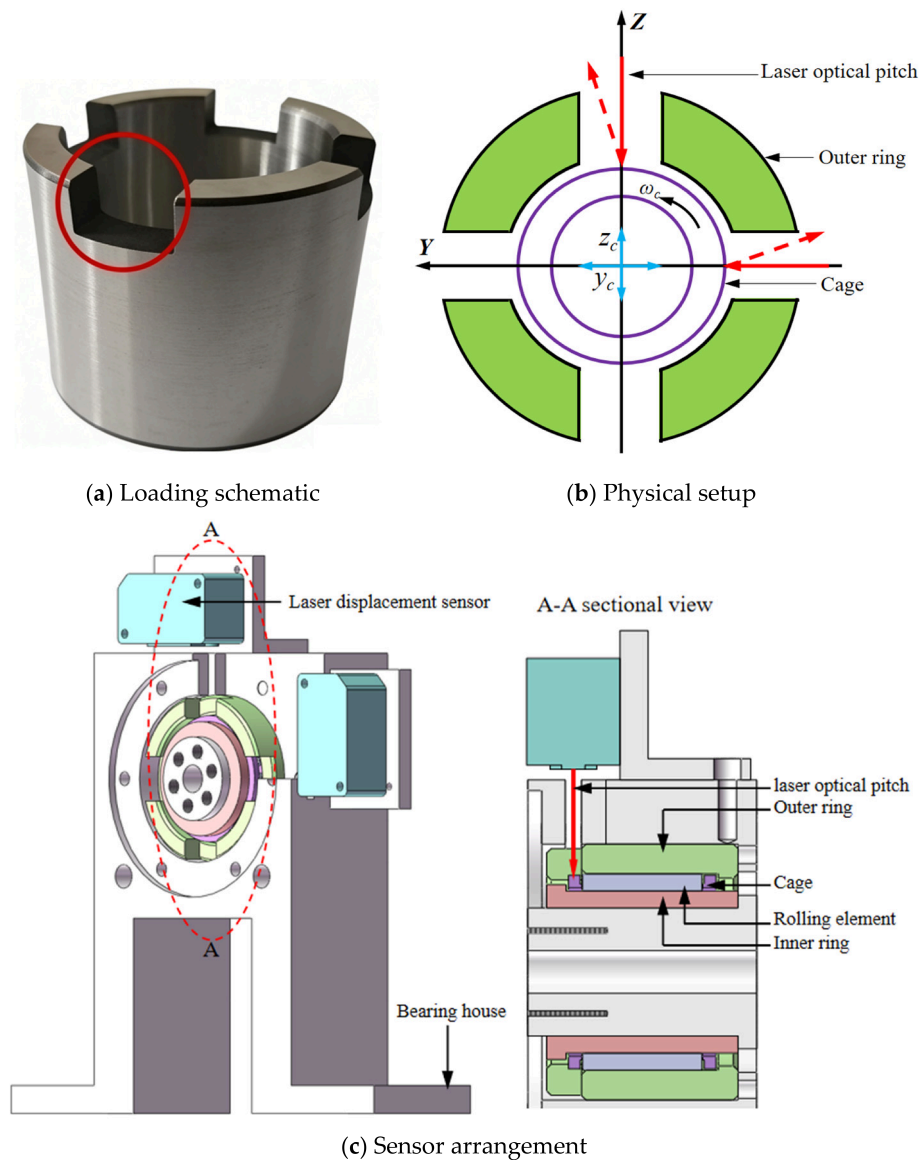


Figure 4. Cage condition testing.

Two eddy-current sensors were arranged outside the spindle and near the root of the inner ring (No. 2 in Figure 3) to measure the vertical and horizontal displacements of the inner ring.

Acceleration sensors were arranged on the top and side of the bearing seat (No. 1 in Figure 3) to measure the vertical, horizontal radial, and axial vibration acceleration signals.

The torque sensor shown in Figure 1 was installed to measure the bearing-system friction torque during operation.

The CA-YD-136 acceleration sensor, 6600XL eddy-current sensors, HG-C1030 laser displacement sensors, DYN-200 torque sensor, and DYLF-102 force sensor were used according to the factory calibration specifications. No additional in-house calibration, reference-input calibration, or curve fitting was performed before the experiments.

Table 2. Sensor composition and specifications.

Sensor	Model	Quantity	Specifications
Acceleration sensor	CA-YD-136	3	Sensitivity: 3 pC/g Measurement range: 1000 g Frequency response: 35,000 Hz Frequency-response error: 5% Maximum transverse sensitivity: $\leq 5\%$ (Sinocera Piezotronics, Yangzhou, China)
Eddy-current sensor	6600XL	2	Sensitivity: 2 mV/ μm ; Frequency response: 0–10 kHz Additional sensitivity error: $\pm 20\%$ (Kaihang Intelligent Equipment, Huanggang, China)
Laser displacement sensor	HG-C1030	2	Sampling rate: 666 Hz Measurement range: 30 ± 5 mm Linearity error: $\pm 0.1\%$ FS Repeatability: 10 μm (Panasonic, Osaka, Japan)
Torque sensor	DYN-200	1	Measurement range: ± 20 N·m Sampling frequency: 800 Hz Non-linearity: $\leq 0.1\%$ FS (Daysensor, Bengbu, China)
Force sensor	DYLF-102	1	Measurement range: 800 kg Sampling frequency: 888 Hz Combined accuracy: $\geq 0.05\%$ FS (Daysensor, Bengbu, China)

(2) Dynamic model of the bearing

A simplified dynamic model of the bearing was established for the interpretation of the experimental phenomena observed under periodic impact loading. The model was not intended for quantitative prediction. The governing equations describe the loading state of the bearing under periodic impact loading and provide a theoretical framework for interpreting the response characteristics observed in the experiments. The model formulation is presented as follows:

The force analysis of the needle roller bearing is shown in Figure 5. Q_{ij} and Q_{oj} denote the contact forces between the rolling element and the inner and outer raceways, respectively; $F_{\mu ij}$ and $F_{\mu oj}$ denote the drag forces exerted by the inner and outer raceways on the rolling element, respectively; F_{cj} denotes the centrifugal force acting on the rolling element; F_{crj} denotes the normal force exerted by the cage on the rolling element; $F_{\mu crj}$ denotes the tangential friction force exerted by the cage on the rolling element; F_{Dj} denotes the flow resistance exerted by the lubricant on the revolution of the rolling element; ω_i denotes the rotational speed of the inner ring; ω_c denotes the rotational speed of the cage; ω_r denotes the rotational speed of the rolling element; and φ_j denotes the angular position of the j -th rolling element.

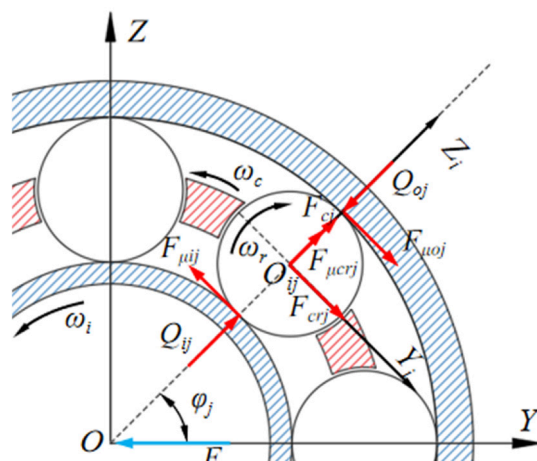


Figure 5. Inner ring and cage coordinate system.

Based on the force analysis shown in Figure 5, the differential equations of motion for the outer ring, inner ring, and cage were established to support the subsequent interpretation of the experimental observations.

The differential equation of motion for the outer ring is given as follows:

$$\begin{aligned} m_o \ddot{y}_o &= \sum_{j=1}^N Q_{oj} \cos \varphi_j - \sum_{j=1}^N F_{\mu oj} \sin \varphi_j \\ m_o \ddot{z}_o &= \sum_{j=1}^N Q_{oj} \sin \varphi_j + \sum_{j=1}^N F_{\mu oj} \cos \varphi_j \end{aligned} \quad (1)$$

Here, m_o is the mass of the outer ring.

The differential equation of motion for the inner ring is given as follows:

$$\begin{aligned} m_i \ddot{y}_i &= -F_y - \sum_{j=1}^N Q_{ij} \cos \varphi_j + \sum_{j=1}^N F_{\mu ij} \sin \varphi_j \\ m_i \ddot{z}_i &= -\sum_{j=1}^N Q_{ij} \sin \varphi_j - \sum_{j=1}^N F_{\mu ij} \cos \varphi_j \end{aligned} \quad (2)$$

Here, m_i is the mass of the inner ring.

The differential equation of motion for the cage is given as follows:

$$\begin{aligned} m_c \ddot{y}_c &= \sum_{j=1}^N (-F_{crj} \cos \varphi_j - F_{\mu crj} \sin \varphi_j) \\ m_c \ddot{z}_c &= \sum_{j=1}^N (-F_{crj} \sin \varphi_j + F_{\mu crj} \cos \varphi_j) \end{aligned} \quad (3)$$

Here, m_c is the mass of the cage.

3. Dynamic Characteristics of Needle Roller Bearings Under Periodic Impact Loading

A periodic impact load was applied as the external excitation, and the corresponding impact load function is expressed as follows:

$$F_y = F_{re} \sin(2\pi f_j t) \quad (4)$$

Here, F_{re} is the impact load amplitude and f_j is the impact load frequency.

The test bearing was operated under intermittent drop-feed lubrication using No. 46 lubricating oil. The ambient temperature during the experiments was approximately

15–30 °C. All experimental trends reported in this study were obtained through three independent repeated tests conducted under each corresponding test condition, with standard deviations representing inter-test variations falling within 0.07.

The test conditions listed in Table 3 to investigate the dynamic characteristics of needle roller bearings under periodic impact load.

Table 3. Test conditions.

Rotational Speed (r/min)	Frequency (Hz)	Amplitude (N)
1500	0	0
1500	10	20

Figure 6 shows the time-domain waveform and frequency spectrum characteristics of the outer-ring vibration acceleration in the vertical direction. The results show that, under periodic impact load, new frequency components f_j and harmonics nf_j appear in the frequency spectrum. In addition, the amplitude of the outer-ring vibration acceleration increases after the periodic impact load is applied.

As the periodic impact load is transmitted from the inner ring to the outer ring through the rolling elements, the contact between the rolling elements and the outer ring is intensified. The force Q_{oj} exerted by the rolling elements on the outer ring is therefore increased, resulting in an increase in the amplitude of the vertical acceleration $\ddot{z}_o$.

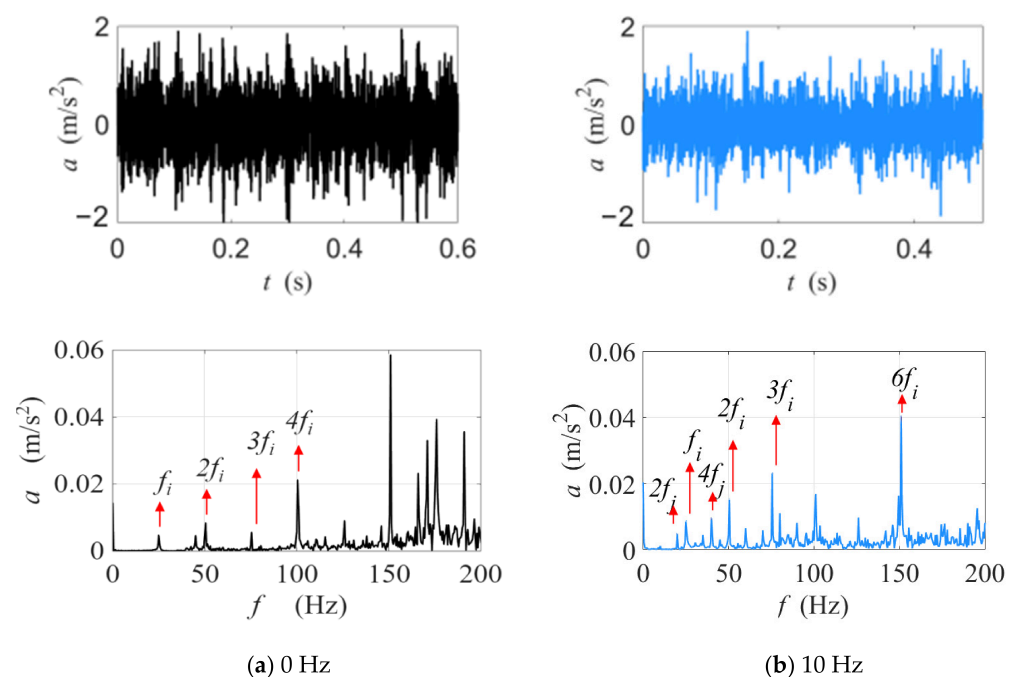


Figure 6. Vertical acceleration of the outer ring.

Figures 7 and 8 show the inner-ring motion trajectory and the time-domain waveform and frequency spectrum characteristics of the horizontal displacement, respectively. The results show that, under periodic impact load, combined frequency components $f_i + f_j$ appear in the frequency spectrum. The inner-ring motion trajectory changes from fluctuations within ± 0.05 mm in both the z_i and y_i directions to fluctuations within ± 0.05 mm in the z_i direction and within ± 0.5 mm in the y_i direction, namely the impact direction.

As the impact load amplitude acting on the inner ring is increased, the displacement of the inner ring in the loading direction, namely the Y direction, is increased, whereas no obvious change is observed in the orthogonal direction, namely the Z direction. In addition,

the combined frequency component $f_i + f_j$ appears in the frequency spectrum. This phenomenon can be attributed to the nonlinear coupling between the vibration components f_i and f_j , which is caused by the bearing clearance, nonlinear contact stiffness, and the time-varying contact state under periodic impact excitation.

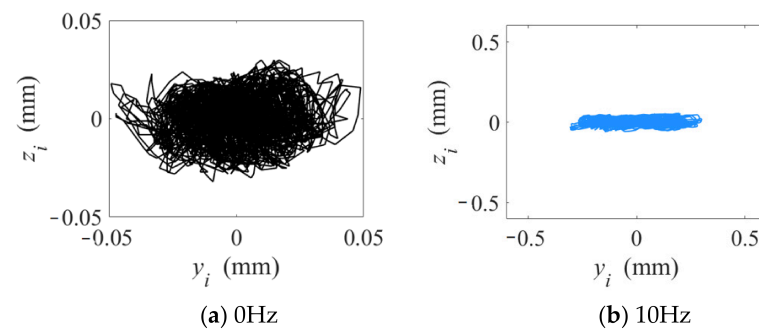


Figure 7. Motion trajectory of the inner ring.

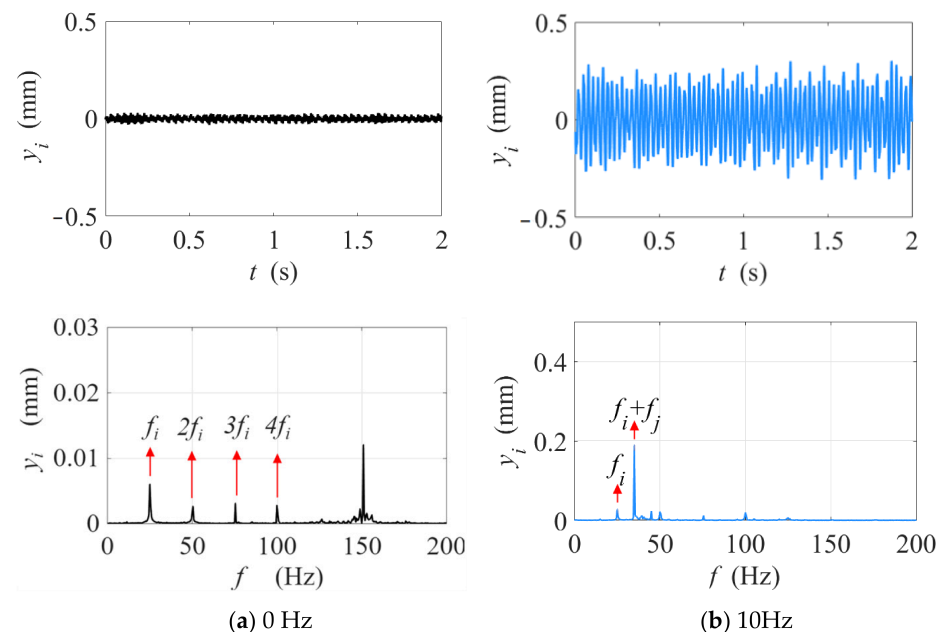


Figure 8. Horizontal displacement of inner ring.

Figures 9–11 show the cage trajectory and the time-domain waveforms and frequency spectrum characteristics of the horizontal and vertical displacements, respectively. The results show that, under periodic impact load, the cage trajectory changes from fluctuations within ± 0.1 mm in both the z_i and y_i directions to whirling over a range greater than ± 0.1 mm in both directions. In addition, the amplitudes of the cage characteristic frequency increase in both the horizontal and vertical directions.

Under the periodic impact load, the cage whirl is mainly induced by the periodic redistribution of the contact forces inside the bearing. The applied load can be regarded as a time-varying excitation, which periodically changes the normal contact force between the rolling elements and raceways. As the rolling elements pass through the loaded zone, the circumferential distribution of contact force, tangential traction, and rolling element-pocket interaction force becomes non-uniform. Consequently, the resultant force acting on the cage does not always pass through its geometric center, causing the cage center to deviate from the bearing axis and to perform an orbital motion. This orbital motion is observed as cage whirl.

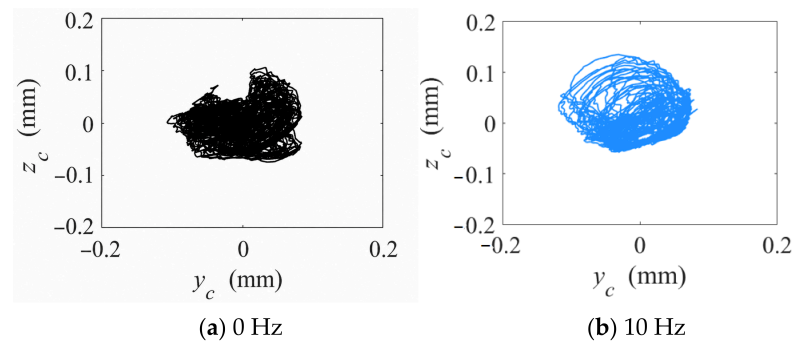


Figure 9. Motion trajectory of the cage centroid.

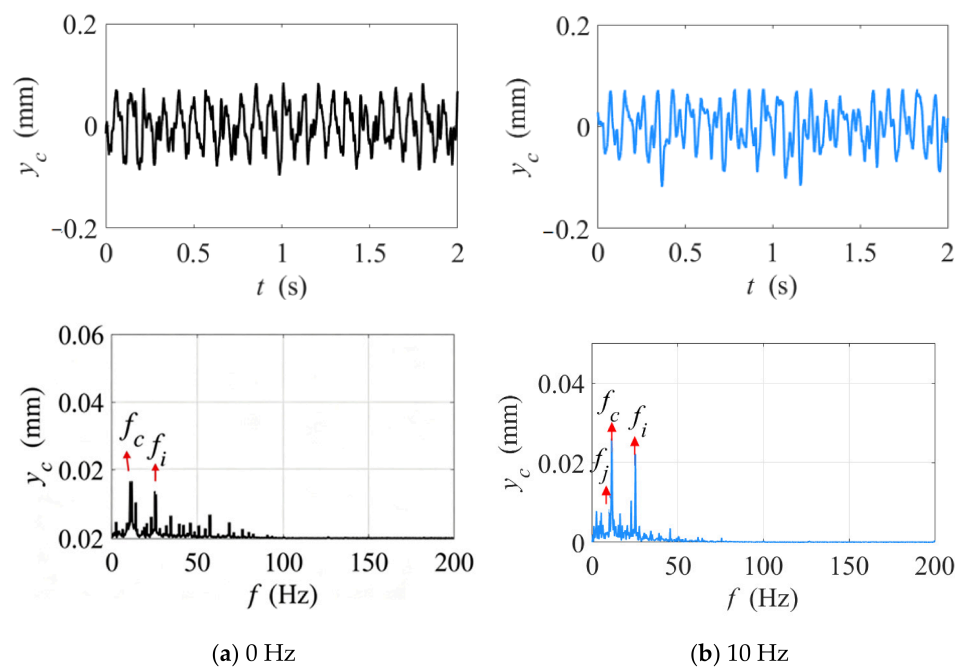


Figure 10. Horizontal displacement of cage.

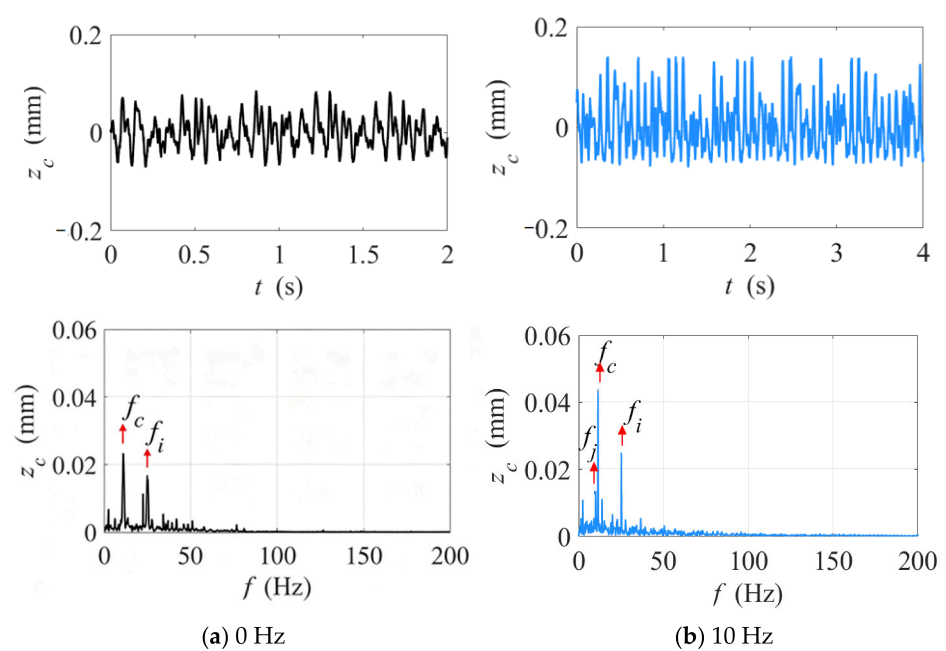


Figure 11. Vertical displacement of cage.

As the periodic impact load F_y in the Y direction is transmitted to the cage through the rolling elements, the normal force F_{crj} exerted by the rolling elements on the cage is increased instantaneously. According to Equation (3), m_c remains unchanged, whereas F_{crj} increases, thereby increasing the time-domain amplitude.

The test results show that the periodic impact load leads to a slight increase in the amplitude of the outer-ring vibration acceleration, an enlarged fluctuation range of the inner-ring motion trajectory along the Y direction, and increased amplitudes of the cage characteristic frequency in both the horizontal and vertical directions.

Therefore, a comparative study was conducted under different impact load amplitudes and frequencies, and the influence of impact load parameters on the dynamic characteristics of needle roller bearings was further investigated.

4. Effects of Periodic Impact Load Parameters on the Dynamic Characteristics of Needle Roller Bearings

4.1. Effect of Periodic Impact Load Amplitude on the Dynamic Performance of Needle Roller Bearings

The effect of impact load amplitude was investigated by keeping the impact load frequency and rotational speed constant while varying the impact load amplitude. The variations in bearing vibration acceleration, inner-ring displacement, cage displacement, and bearing-system friction torque were then analyzed. The test conditions are listed in Table 4.

The impact load amplitudes of 20, 180, and 300 N were selected to cover low, intermediate, and high levels within the stable operating range of the test rig. These discrete levels were used to examine the sensitivity of the bearing dynamic response to periodic impact load. Therefore, the results should be interpreted as a controlled parametric investigation rather than a complete representation of all practical operating conditions.

Table 4. Test conditions.

Rotational Speed (r/min)	Frequency (Hz)	Amplitude (N)
1500	10	20
1500	10	180
1500	10	300

Figure 12 shows the time-domain waveform and frequency-spectrum characteristics of the outer-ring vibration acceleration in the vertical direction. The results show that, as the impact load amplitude increases, the amplitude of the outer-ring vibration acceleration exhibits a slight increasing trend. The RMS value of acceleration increased from 0.60 m/s^2 to 0.63 m/s^2 and 0.67 m/s^2 as the excitation amplitude increased, corresponding to increases of 5.0% and 11.7%, respectively, compared with the initial condition. This indicates that a higher periodic impact load amplitude enhanced the vibration intensity of the bearing system.

As the impact load amplitude acting on the inner ring is increased, larger periodic impact load is transmitted from the inner ring to the outer ring through the rolling elements. Consequently, the contact between the rolling elements and the outer ring is intensified, and the force Q_{oj} exerted by the rolling elements on the outer ring is increased, thereby increasing the amplitude of the vertical acceleration $\ddot{z}_o$.

Figures 13 and 14 show the inner-ring motion trajectory and the time-domain waveform and frequency spectrum characteristics of the horizontal displacement, respectively. The results show that, as the impact load amplitude increases, the fluctuation range of the inner-ring motion trajectory increases along the y_i direction, namely the impact di-

rection. The vibration displacement amplitudes of the inner ring and $f_i + f_j$ also show increasing trends. The RMS value of inner-ring displacement increased from 0.129 mm to 0.208 mm and 0.260 mm as the periodic impact load amplitude increased, corresponding to increases of 61.2% and 101.6%, respectively. This indicates that a higher periodic impact load amplitude significantly enhanced the effective displacement response of the inner ring.

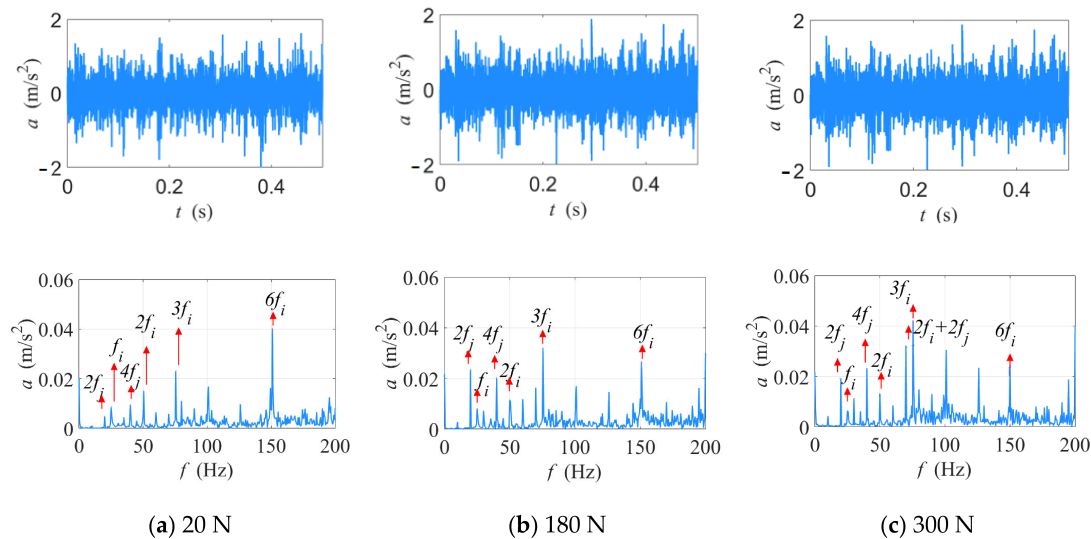


Figure 12. Vertical acceleration of outer ring.

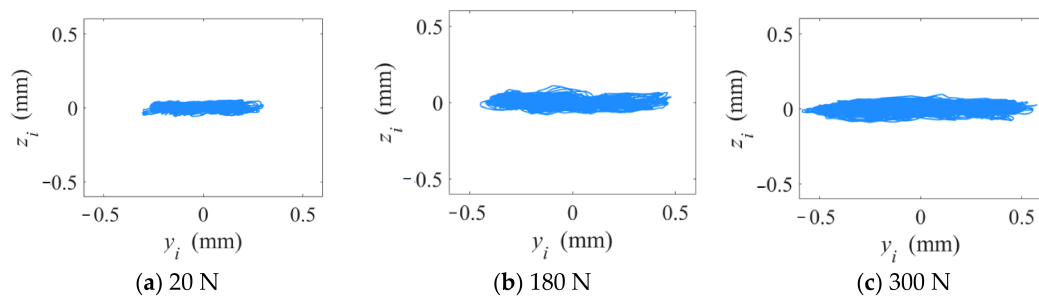


Figure 13. Motion trajectory of the inner ring.

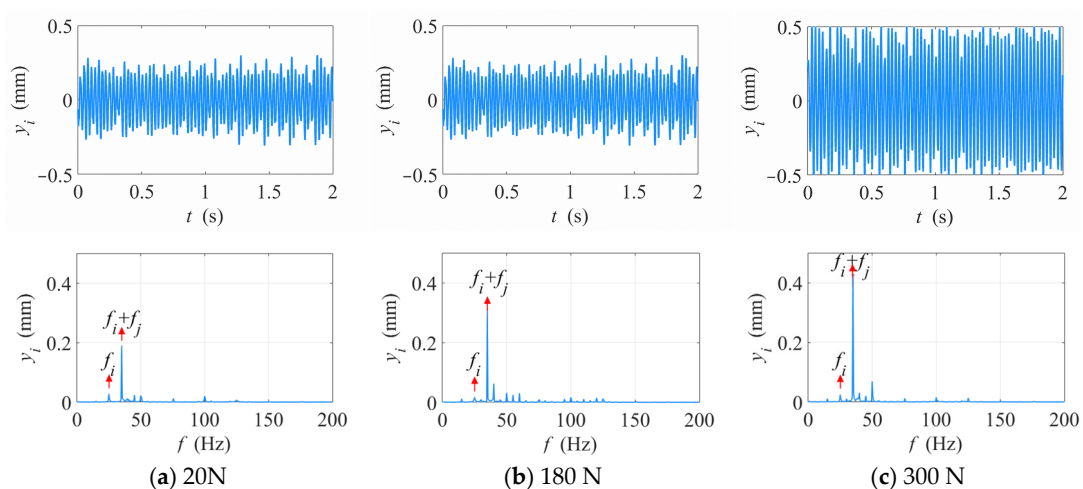


Figure 14. Horizontal displacement of inner ring.

As the impact load amplitude acting on the inner ring is increased, the instantaneous displacement of the inner ring is mainly increased in the loading direction, namely the Y

direction, whereas only a slight change is observed in the orthogonal direction, namely the Z direction.

Figures 15 and 16 show the cage trajectory and the time-domain waveform and frequency spectrum characteristics of the horizontal displacement, respectively. The results show that, as the impact load amplitude increases, both the whirling range of the cage trajectory and the amplitude of the cage characteristic frequency increase. The RMS value of cage displacement increased from 0.038 mm to 0.045 mm and 0.052 mm as the periodic impact load amplitude increased. The spectral amplitude at the cage characteristic frequency increased from 0.026 mm to 0.030 mm and 0.037 mm, corresponding to increases of 14.3% and 40.9%, respectively.

As the impact load amplitude acting on the inner ring is increased, the load interaction between the cage and the rolling elements is enhanced, and the contact between the rolling elements and the cage is intensified. Consequently, the normal force F_{rcj} exerted by the rolling elements on the cage is increased. According to Equation (3), the cage accelerations z_c and y_c increase, leading to the expansion of the whirling trajectory. With increasing impact load amplitude, the periodic pulse amplitude of F_{rcj} is marked.

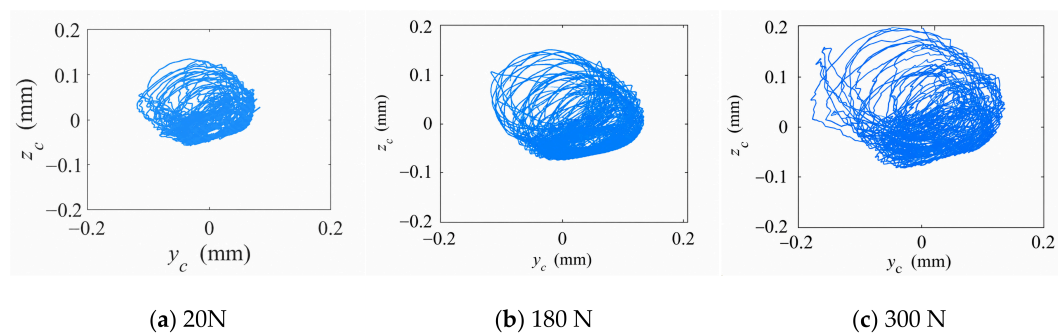


Figure 15. Motion trajectory of the cage.

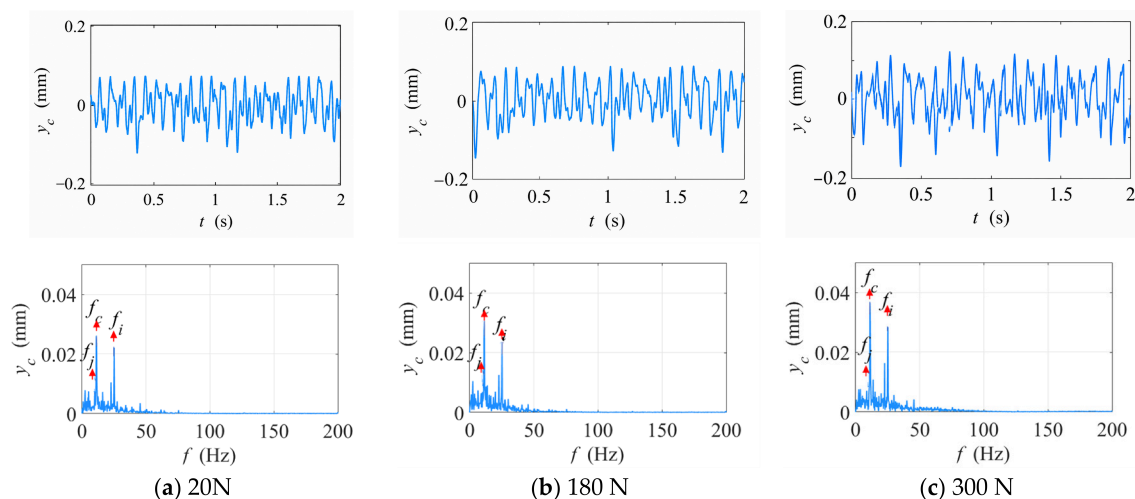


Figure 16. Horizontal displacement of cage.

Figure 17 shows the amplitude curve of the bearing system friction torque. The results show that the amplitude of the bearing system friction torque increases with increasing impact load amplitude. The mean friction torque increased from 0.397 N·m to 0.408 N·m and 0.440 N·m corresponding to increases of 2.9% and 11.0%.

The increase in friction torque should not be attributed solely to intensified contact. Larger transient variations in the contact load and relative motion among the rolling elements, raceways, and cage are induced by a higher periodic impact load amplitude.

Consequently, rolling–sliding motion, lubricant film shear, and contact-related energy dissipation may be increased, thereby contributing to the measured increase in friction torque. The presence of stronger dynamic interactions within the bearing system is supported by the concurrent increases in vibration and displacement responses. However, because drop-feed lubrication was adopted, the lubrication state could not be monitored directly during the tests. In addition, the local contact temperature, contact stiffness, and slip ratio were not measured. Therefore, the individual contribution of each mechanism cannot be quantified in the present study.

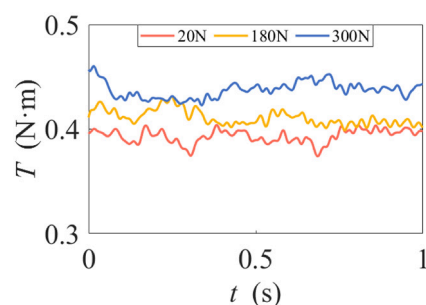


Figure 17. Bearing system friction torque.

4.2. Effect of Periodic Impact Load Frequency on the Dynamic Performance of Needle Roller Bearings

The effect of impact load frequency on the dynamic performance of needle roller bearings was investigated by keeping the impact load amplitude and rotational speed constant while varying the impact load frequency. The variations in bearing vibration acceleration, inner-ring displacement, cage displacement, and friction torque were then analyzed. The test conditions are listed in Table 5.

The frequencies of 5, 25, and 50 Hz were selected to cover low, intermediate, and high levels within the stable operating range of the test rig. These discrete levels were used to examine the sensitivity of the bearing dynamic response to periodic impact load. Therefore, the results should be interpreted as a controlled parametric investigation rather than a complete representation of all practical operating conditions.

Table 5. Test conditions.

Rotational Speed (r/min)	Frequency (Hz)	Amplitude (N)
2000	5	100
2000	25	100
2000	50	100

Figure 18 shows the time–domain waveform and frequency spectrum characteristics of the outer-ring vibration acceleration in the vertical direction. The results indicate that the amplitude of the outer-ring vibration acceleration in the vertical direction increases with increasing impact load frequency. The RMS value of acceleration increased from 0.76 m/s^2 to 0.91 m/s^2 and 1.05 m/s^2 as the excitation frequency increased, corresponding to increases of 19.7% and 38.2%, indicating a higher vibration intensity of the bearing system.

As the impact load frequency acting on the inner ring is increased, the number of contacts per unit time between the outer ring and the rolling elements increases. Consequently, the outer ring is subjected to the subsequent impact before equilibrium is fully restored. In the time domain, this process induces a superposition effect in the vibration waveform, resulting in an increase in the acceleration amplitude.

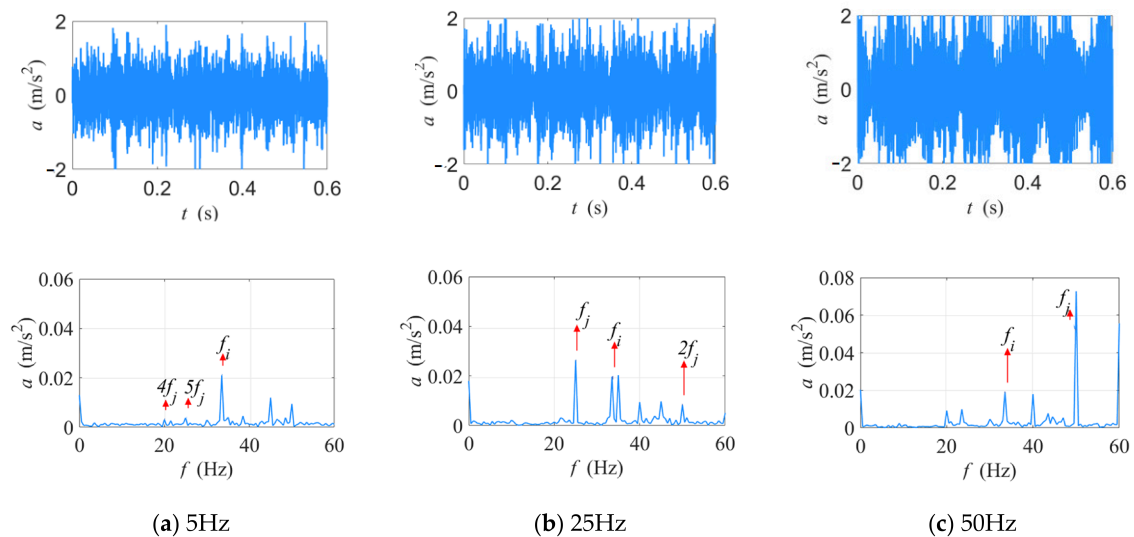


Figure 18. Vertical acceleration of outer ring.

Figures 19 and 20 show the inner-ring motion trajectory and the time-domain waveform and frequency spectrum characteristics of the horizontal displacement, respectively. The results indicate that, as the impact load frequency increases, the inner-ring motion trajectory expands along the y_i direction, namely the impact direction. The RMS value of inner-ring displacement increased from 0.193 mm to 0.197 mm and 0.207 mm as the impact frequency increased, corresponding to increases of 2.1% and 7.3%, respectively, indicating a slight enhancement in the effective displacement response.

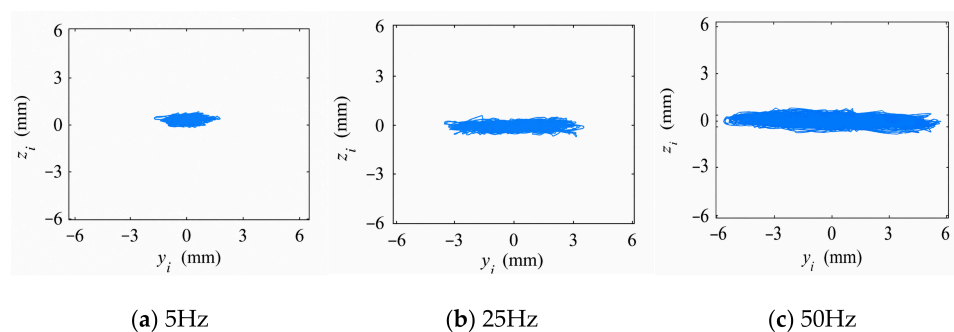


Figure 19. Motion trajectory of the inner ring.

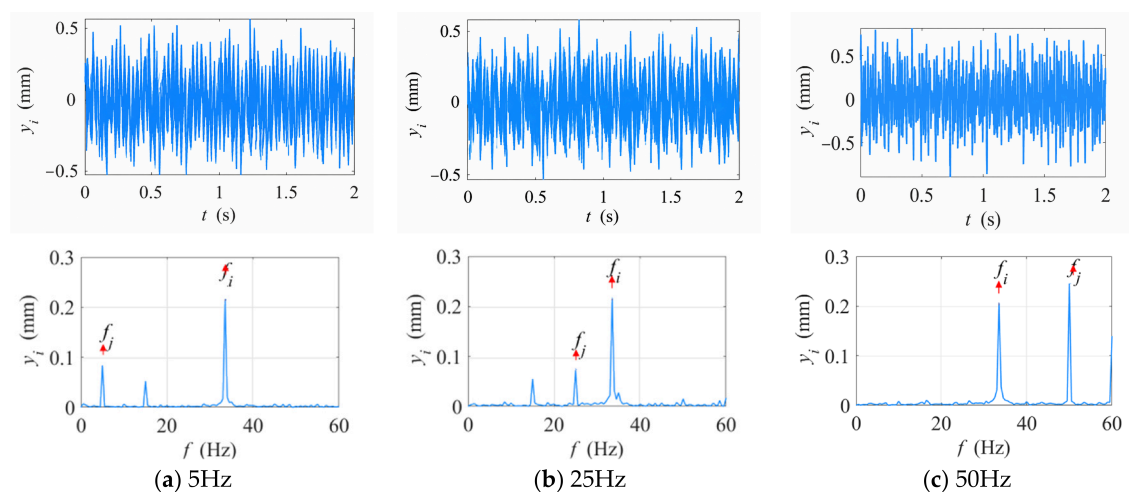


Figure 20. Horizontal displacement of inner ring.

As the impact load frequency acting on the inner ring is increased, the subsequent impact is applied before the instantaneous displacement induced by the preceding impact has fully decayed. Under this continuous excitation, the transient displacement of the inner ring in the y_i direction accumulates, resulting in a marked expansion of the motion trajectory along this direction.

Figures 21 and 22 show the cage trajectory and the time-domain waveform and frequency spectrum characteristics of the horizontal displacement, respectively. The results show that the whirling range of the cage trajectory decreases with increasing impact load frequency. The RMS value of cage displacement increased from 0.056 mm to 0.057 mm and 0.061 mm as the impact frequency increased, corresponding to increases of 1.8% and 8.9%.

As the periodic impact load frequency acting on the inner ring is increased, the number of contacts per unit time between the cage and the rolling elements increases. Under repeated excitation, the instantaneous displacement of the cage induced by the periodic impact load is constrained, resulting in a reduced whirling range of the cage trajectory.

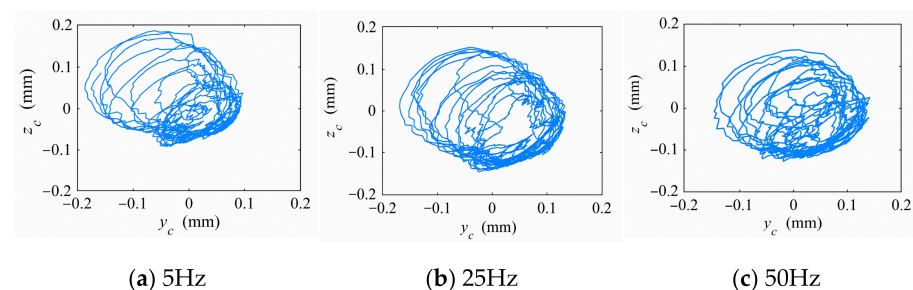


Figure 21. Motion trajectory of the cage.

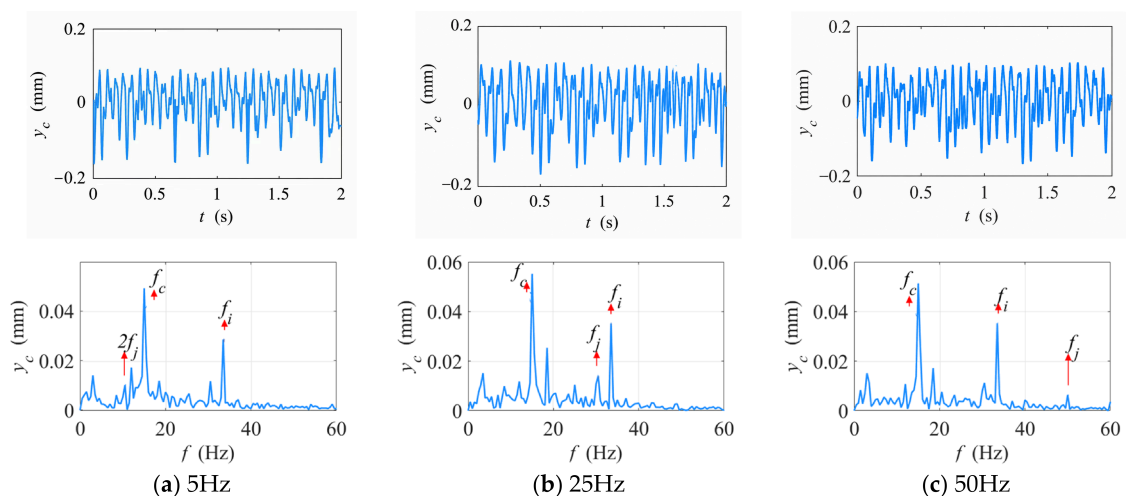


Figure 22. Horizontal displacement of cage.

Figure 23 shows the amplitude curve of the bearing system friction torque. The results show that the amplitude of the bearing system friction torque increases with increasing impact load frequency. The mean friction torque increased from 0.148 N·m to 0.160 N·m and 0.181 N·m, corresponding to increases of 8.3% and 22.3%.

An increase in the mean friction torque of the bearing system was observed as the periodic impact load frequency was increased. The interval between successive loading cycles was shortened at a higher frequency, and the transient dynamic response may therefore not have been fully attenuated. More frequent fluctuations in the contact load and relative motion among the rolling elements, raceways, and cage may consequently have been induced. Rolling–sliding motion, lubricant film shear, and frictional energy dissipation may thus have been increased. The increase in friction torque was accompanied by

enhanced vibration and displacement responses. However, because drop-feed lubrication was adopted, the lubrication state could not be monitored directly during the tests. In addition, the local contact temperature, contact stiffness, and slip ratio were not measured. Therefore, the individual contributions of these mechanisms cannot be quantified in the present study.

As for physical explanation, the increase in friction torque with increasing excitation frequency can be physically explained by the enhanced rate of dynamic energy dissipation. For a periodic impact load with a given load amplitude, increasing the excitation frequency does not necessarily increase the peak load, but it increases the loading and unloading rate. Therefore, the contact state between the rolling elements and raceways changes more rapidly, which promotes microslip, contact hysteresis, and transient traction losses. Meanwhile, the rolling elements interact with the cage pockets more frequently, leading to stronger sliding and impact-related energy dissipation. The lubricant film is also subjected to more rapid squeeze and shear deformation, which increases viscous dissipation. This explains why the friction torque increases as the frequency of the periodic impact load increases.

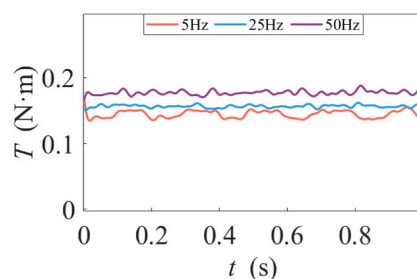


Figure 23. Bearing system friction torque.

The present results qualitatively agree with previous studies showing that impact loading and variable operating conditions can alter the contact state and dynamic stability of bearing cages [3,4,10,12]. In particular, the increase in vibration and internal displacement with impact amplitude indicates an intensified transient interaction within the roller-cage-raceway system. This trend is consistent with the reported sensitivity of cage dynamics to impact excitation and time-varying loads [3,4,10]. In addition, the increase in friction torque with impact frequency may be related to more frequent transient contact-force fluctuations, local slip, and changes in the lubrication state, as suggested by transient lubrication and time-varying-load studies [10,12].

5. Conclusions

The dynamic characteristics of needle roller bearings under periodic impact loading were experimentally investigated by varying the loading amplitude and frequency. The conclusions of this study are limited to the tested needle roller bearing, test rig, shaft speed, lubrication condition, and the investigated ranges of periodic impact loading amplitude and frequency. The conclusions are as follows:

Periodic impact loads disrupt the motion stability of the bearing. This phenomenon manifests as increased vibration-acceleration amplitude of the outer ring and aggravated cage motion in both the horizontal and vertical directions. These results reveal that periodic impact loads amplify internal motion fluctuations of the roller-cage-raceway system, which further degrades the operational stability of the bearing.

The dynamic response of the bearing was intensified with increasing periodic impact load amplitude. The RMS values of acceleration, inner-ring displacement, and cage displacement increased, and the mean friction torque also increased. In particular, the inner-

ring motion trajectory expanded along the impact load direction, while the whirling range of the cage trajectory increased. The bearing vibration, internal motion, and friction-torque were affected by periodic impact load amplitude.

Different response trends were observed with increasing periodic impact load frequency. The RMS values of acceleration, inner-ring displacement, and cage displacement generally increased, and the mean friction torque also increased. However, the whirling range of the cage trajectory decreased, indicating that the effects of periodic impact load frequency on inner-ring motion and cage motion were not fully consistent. Therefore, the influence of impact-load frequency should be evaluated according to the specific motion feature considered.

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