

Article

Vibration Effects of Axial Preload in Ball Bearings of an Electric Drive Unit Using Multi-Body Dynamics Simulation

Seung Gyu Hwang , SangHan Lee , Donghee Kim, Seonhyeong Kim and Dong-Jun Kim *

Hyundai Mobis, 37, Cheoldobangmulgwan-ro, Uiwang-si 16082, Gyeonggi-do, Republic of Korea; hsg@mobis.com (S.G.H.); hurtooth@mobis.com (S.L.); seon0112@mobis.com (S.K.)

* Correspondence: dongjunkim@mobis.com

Abstract

The vibration characteristics of bearings in the Electric Drive Unit (EDU) of electric vehicles have a direct impact on the overall Noise, Vibration, and Harshness (NVH) performance. Due to the high speed of the drive motor, the importance of reducing shaft system vibrations has grown. Wave Washer (WW) is used to apply axial preload to bearings, thereby mitigating shaft system instability. In this study, the EDU was modeled using the Multi-Body Dynamics (MBD) simulation software ADAMS, and the effect of WW preload on bearing vibration was analyzed. The simulation results revealed that the dominant low-order vibration peak around 400 Hz was reduced by up to 30 dB. Notably, the vibration reduction effect was evident as the preload increased to level A, whereas only marginal improvements were observed beyond this level, indicating a saturation tendency. To provide a physical interpretation of this behavior, a simplified analytical stiffness formulation based on Hertzian contact theory was employed. The results suggest that the observed saturation tendency is associated with both the stabilization of rolling-element contact conditions and the nonlinear relationship between preload and bearing stiffness.

Keywords: electric drive unit (EDU); noise, vibration, and harshness (NVH); multi-body dynamics (MBD); wave washer (WW); preload

1. Introduction

The power transmission system of Electric Vehicles (EVs) is required to achieve low noise and high reliability, which elevates the importance of the performance of individual components within the system [1]. Unlike conventional internal combustion engine vehicles, EVs exhibit lower powertrain masking noise, making vibrations generated by drivetrain components more perceptible to vehicle occupants [1]. Bearings are among the key components of the drive system, supporting rotational motion while distributing loads [1]. Among the primary factors influencing bearing vibration, preload has a substantial impact on bearing stiffness, friction, durability, and dynamic vibration response; consequently, applying and controlling preload effectively has emerged as a central challenge for ensuring system reliability [2–4].

Wave Washer (WW) provides elastic force to apply appropriate axial preload to bearings, enabling absorption of assembly tolerances while accommodating dynamic loads. However, under operating conditions where multiple loads and motions are coupled, experimentally analyzing the influence of wave washer-based preload control on bearing vibration faces considerable limitations.



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Numerous studies have investigated the relationship between bearing preload and vibration characteristics using a variety of analytical, numerical, and experimental approaches. Under elastohydrodynamic lubrication (EHL) conditions, variations in preload for angular contact ball bearings were examined through combined experimental measurements and theoretical models to evaluate their effects on vibration characteristics and harmonic components [5]. In related work, the influence of preload conditions on the vibration response of ball bearings was analyzed, demonstrating that preload theoretically constitutes a key governing parameter in bearing dynamics [6]. Moreover, rotor–bearing–housing assemblies have been integrated using finite element models coupled with MSC ADAMS 2022.4-based Multi-Body Dynamics (MBD) simulations to elucidate the effects of structural flexibility on bearing dynamic behavior with high fidelity [7]. The applicability and validity of ADAMS-based MBD analysis have further been demonstrated by simulating misalignment conditions in rotating systems and evaluating the resulting acceleration responses through Fast Fourier Transform (FFT) analysis [8]. Additionally, both healthy and defective deep-groove ball bearings have been modeled using ADAMS, enabling quantitative fault diagnosis based on time-domain acceleration signals and FFT analysis. This approach is methodologically related to the time-domain colormap analysis employed in the present study [9].

From a reliability and design standpoint, guidelines for space-mechanism bearings emphasize that insufficient or excessive preload can elevate vibration levels and shorten service life, thereby underscoring the need to identify an optimal preload condition [10]. Research has examined the design and behavior of WW components: analytical and design models based on key geometric parameters—such as the number of waves, diameter, and thickness—have been formulated [11], and three-dimensional CAD together with finite element analyses have quantitatively characterized how load–displacement behavior and geometry govern preload characteristics [12]. From the EV NVH perspective, finite element-based dynamic NVH analyses of electric power steering systems have provided detailed insights into vibration originating from drivetrain components, reinforcing the necessity of simulation-based assessment—including finite element-based and MBD approaches—for EV drivetrain NVH evaluation [13].

In summary, although preload is widely recognized as a critical factor for controlling bearing vibration response, there remains a shortage of studies that systematically analyze time-domain acceleration responses through simulation and clarify nonlinear vibration behavior as a function of preload level. Recent studies have also investigated analytical and numerical approaches for predicting bearing stiffness, preload-dependent dynamic behavior, and rolling-bearing vibration characteristics, highlighting the importance of bearing stiffness in determining vibration responses under different operating conditions [6,14–17]. Furthermore, while previous studies have investigated the influence of preload on bearing vibration characteristics [2–6], limited attention has been paid to how preload affects vibration transmission behavior within the drivetrain system [7,13,18]. Considering that bearings act as major vibration transmission paths within the EDU [1,7,13], understanding how preload influences vibration propagation throughout the drivetrain system is essential for NVH-oriented design. In particular, studies that quantitatively investigate nonlinear vibration responses under varying preload levels and explain the associated vibration-reduction mechanisms in EV drivetrain applications are still limited [1,18].

In this study, an EDU system was modeled using an ADAMS-based MBD model to simulate bearing vibration responses in the time domain under varying preload conditions. Bearing acceleration data are visualized with colormap representations, and order-tracking analysis is used to intuitively identify vibration trends with changing preload. This approach clarifies the influence of WW on the vibration characteristics of the EDU and

provides foundational insights for determining optimal bearing preload in design applications. Unlike previous studies that have focused predominantly on frequency-domain analysis [1,5,19], the present study integrates time-domain colormap visualization with order-tracking analysis to quantitatively characterize nonlinear vibration responses due to preload variations under two representative loading conditions. In addition, the underlying mechanisms responsible for vibration reduction are discussed from the perspectives of vibration transmission characteristics and dynamic stabilization of the shaft-bearing system.

2. Simulation Method

2.1. Multi-Body Dynamics Model of the EDU

Figure 1 illustrates the Multi-Body Dynamics (MBD) model of the power transmission system in the EDU, developed using MSC ADAMS 2022.4 software. The model consists of a rotor shaft, a pair of gears, a pair of splines, bearings, and a simplified housing that includes only mass, inertia, and modal analysis data. The rotor shaft is supported by three ball bearings (B1–B3), and a WW is applied to the left side of the B3 bearing. When the WW is applied, axial preload is introduced in the $-Z$ direction. The coordinate system is defined such that the shaft axis is the Z -axis, the radial directions are X and Y , gravity acts in the $+Y$ direction, and, in the vehicle installation configuration, the forward direction corresponds to $-X$.

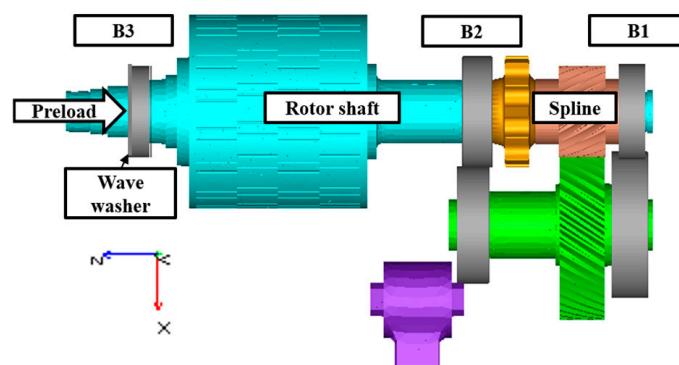


Figure 1. Multi-Body Dynamics (MBD) model of the EDU.

All shafts were modeled as flexible bodies generated through finite element analysis to account for bending and torsional deformation during operation. Structural characteristics derived from finite element analysis were incorporated into the bearing supports, spline connections, and gear-meshing regions to realistically represent the dynamic interactions among drivetrain components. The simplified housing was constrained using boundary conditions corresponding to actual mounting locations. The housing model was connected to the shaft-bearing system through the bearing supports, allowing the influence of boundary conditions and structural dynamics on vibration transmission behavior to be considered.

The principal parameters of the EDU MBD model used in this study—such as bearing properties, gear meshing stiffness, and the axial preload level—were configured based on actual EDU design data. Because these parameters are subject to industrial confidentiality, individual numerical values are not disclosed. Instead, the modeling methodology and parameter ranges were selected with reference to established practices reported in previous MBD simulation studies [7–9,18–23]. As noted in the literature, a simplified modeling strategy aimed at NVH characterization is suitable for assessing relative trends in EDU vibration behavior, and the present study adopts this approach to focus on the effect of axial preload variation on bearing-induced vibration characteristics while considering the structural flexibility and dynamic interactions of the drivetrain system.

The bearing stiffness characteristics used in the MBD model were obtained from detailed finite-element-based contact analyses. Contact interactions among the inner race, outer race, and rolling elements were modeled using MSC Nastran 2022.4, and the resulting stiffness characteristics were incorporated into the EDU MBD model. The bearing considered in this study is a deep-groove ball bearing. Therefore, the simplified analytical stiffness formulation introduced in Section 4 was not used to define the bearing characteristics in the simulation model, but rather to provide a physical interpretation of the observed preload-dependent vibration behavior.

Figure 2 presents the axial preload–deformation curve of the WW. When the WW is in its free-length state, no preload is generated; as the deformation grows, the preload increases accordingly. In this study, WW models with different stiffness values were applied so that the preload could be regulated within the deformation region. Through this setup, an axial preload—whose magnitude varies with the WW stiffness—was applied to the B3 bearing along the $-Z$ direction.

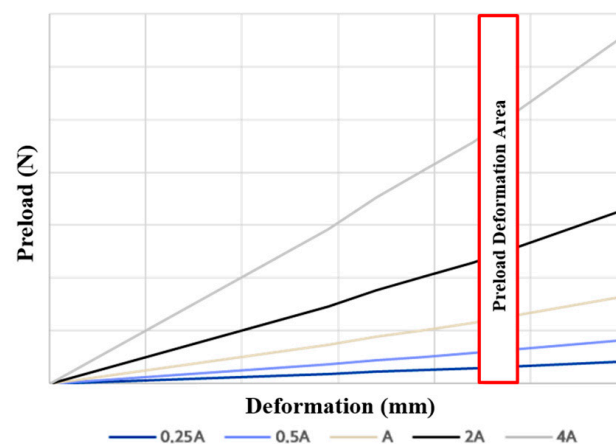


Figure 2. Relationship between Wave Washer deformation and preload.

2.2. Driving Scenarios and Simulation Conditions

The simulations were conducted for 12 different scenarios, as summarized in Table 1. To evaluate the influence of WW preload, the reference preload was defined as A , and six preload levels were compared: 0, $0.25A$, $0.5A$, A , $2A$, and $4A$. The actual preload value, denoted as A N, is represented by the symbol A throughout this paper for confidentiality reasons. Load conditions were categorized into two types based on load ratio: low load (approximately 20%) and high load (approximately 95%). For all scenarios, the rotational speed was ramped up from 0 to 14,000 rpm over 2 s and then maintained at 14,000 rpm for an additional 2 s. The results were examined to assess whether the model responses exhibited physically consistent trends with respect to variations in preload, load, and rotational speed.

Table 1. Simulation conditions.

Case	Load	Preload	Case	Load	Preload
1	20% (Low)	0	7	95% (High)	0
2		$0.25A$	8		$0.25A$
3		$0.5A$	9		$0.5A$
4		A	10		A
5		$2A$	11		$2A$
6		$4A$	12		$4A$

3. Vibration Analysis

In this study, acceleration data measured at the B3 bearing were analyzed as a function of the WW preload. The results corresponding to different load conditions were compared using colormap visualization and order-tracking analysis [20,24]. The colormap representation was employed to identify the overall response trends, after which dominant order components were further examined. The order components for detailed investigation were selected based on shaft rotational orders exhibiting noticeable changes in the colormap results.

3.1. Acceleration Data Analysis

Figure 3 shows the axial (Z-direction) acceleration response of the B3 bearing under the low-load condition as a function of preload. When a WW is applied, the acceleration increase region in the Z-direction shifts toward a higher rotational speed range. This behavior can be attributed to the axial preload applied by the WW, which increases the bearing stiffness and consequently shifts the vibration response toward higher speeds. For preload levels equal to or greater than A, little change in the acceleration response is observed. This indicates that once the WW stiffness exceeds a certain threshold, additional preload does not significantly affect the vibration characteristics. When sufficient stiffness is achieved, further increases in stiffness no longer yield additional reductions in axial vibration acceleration.

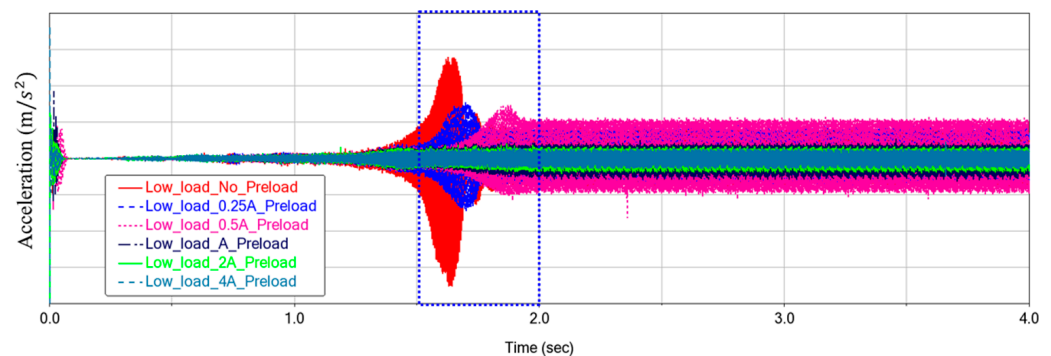


Figure 3. Z-direction acceleration responses of the B3 bearing under low-load conditions for all preload levels.

Figure 4 presents the radial (X, Y) acceleration responses of the B3 bearing under a low-load condition. Figure 4a,b show the acceleration responses in the X- and Y-directions, respectively. The red line represents the no-preload case, while the blue line corresponds to the case with an axial preload of 0.25A applied through the WW. In the absence of preload, insufficient axial force in the Z-direction leads to unstable vibration responses even in the radial directions (X, Y). In contrast, applying a small axial preload (0.25A) significantly reduces radial vibrations, indicating that axial bearing preload is effective in enhancing dynamic stability under low-load conditions.

3.2. Colormap Analysis

3.2.1. Colormap Analysis Under Low-Load Condition

Figure 5 illustrates the vibration colormap results of the B3 bearing under the low-load condition as a function of the applied WW preload. Figure 5a,b present the Y- and Z-direction vibration responses without preload, while Figure 5c,d correspond to a preload of 0.5A. Figure 5e,f show the Y- and Z-direction vibration responses when a preload level of A is applied.

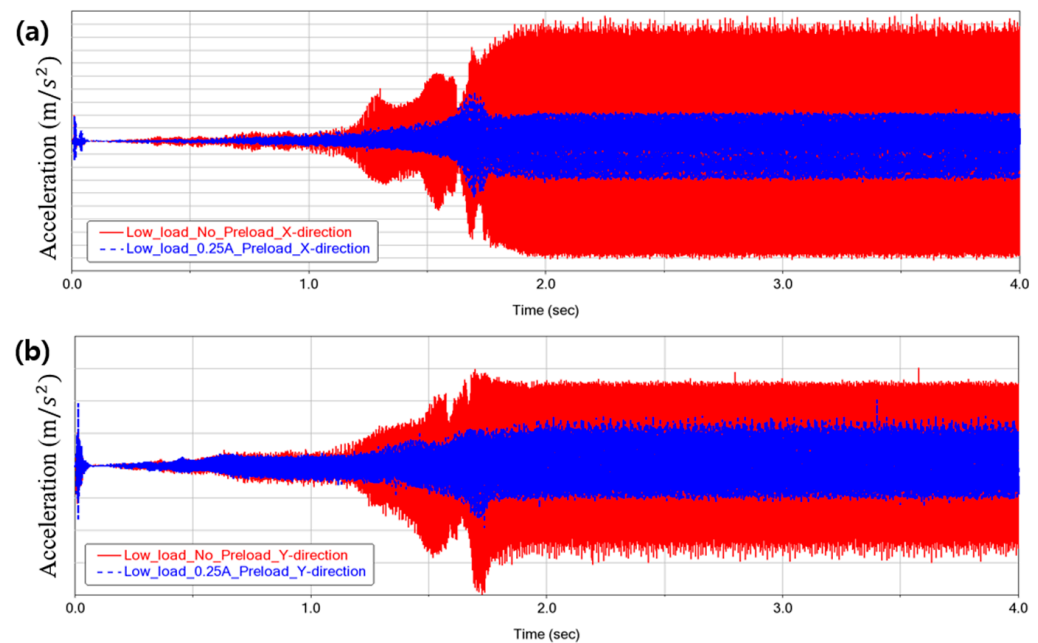


Figure 4. Radial (X, Y) acceleration responses of the B3 bearing under low-load conditions: (a) X-direction and (b) Y-direction. Red: no preload; blue: axial preload of 0.25A.

As shown in Figure 5a,b, when no preload is applied, a pronounced low-order vibration component appears around 400 Hz. In contrast, in Figure 5c–f, the application of WW preload up to level A effectively suppresses this low-order vibration component. This result indicates that the WW contributes to vibration reduction in both the radial (Y) and axial (Z) directions by applying axial preload to the shaft system. However, the colormap results obtained with higher preload levels of 2A and 4A do not exhibit a noticeable difference compared to the A preload condition. Therefore, to further investigate the preload-dependent vibration characteristics, a detailed comparison using order-tracking analysis was conducted.

3.2.2. Colormap Analysis Under High-Load Conditions

Figure 6 presents the Y-direction (radial) vibration colormap results of the B3 bearing under the high-load condition with increasing WW preload. Figure 6a shows the Y-direction vibration response without preload, while Figure 6b,c correspond to preload levels of 0.5A and A, respectively.

When no preload is applied, a significant vibration component is observed around 400 Hz. However, as the WW preload is increased up to level A, this vibration component is effectively reduced. This result suggests that, under high-load conditions, the axial preload applied by the WW increases the bearing stiffness, thereby reducing radial vibration.

On the other hand, when higher preload levels of 2A and 4A are applied, no noticeable additional vibration reduction is observed in the colormap results; therefore, these cases are omitted for clarity. Furthermore, unlike the low-load condition shown in Figure 5, the vibration variations in other directions are relatively small under the high-load condition. This behavior can be attributed to the high torque applied under the high-load condition, which is presumed to induce an effect similar to that of axial preload. Consequently, to more precisely compare the vibration characteristics according to the preload level, order-tracking analysis was conducted.

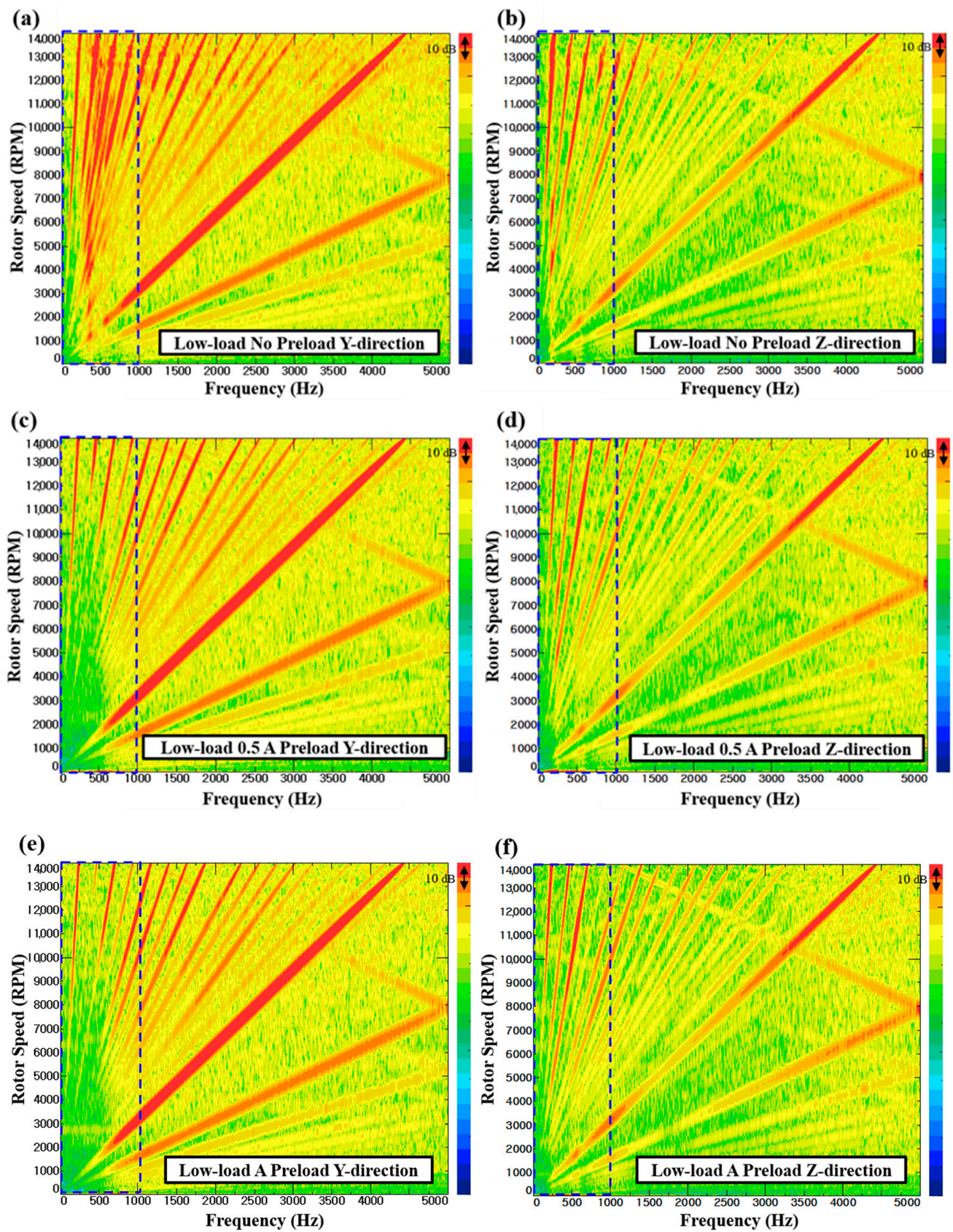


Figure 5. Vibration colormaps of the B3 bearing under low-load conditions: (a) No preload, Y-direction, (b) No preload, Z-direction, (c) 0.5A preload, Y-direction, (d) 0.5A preload, Z-direction, (e) A preload, Y-direction, and (f) A preload, Z-direction.

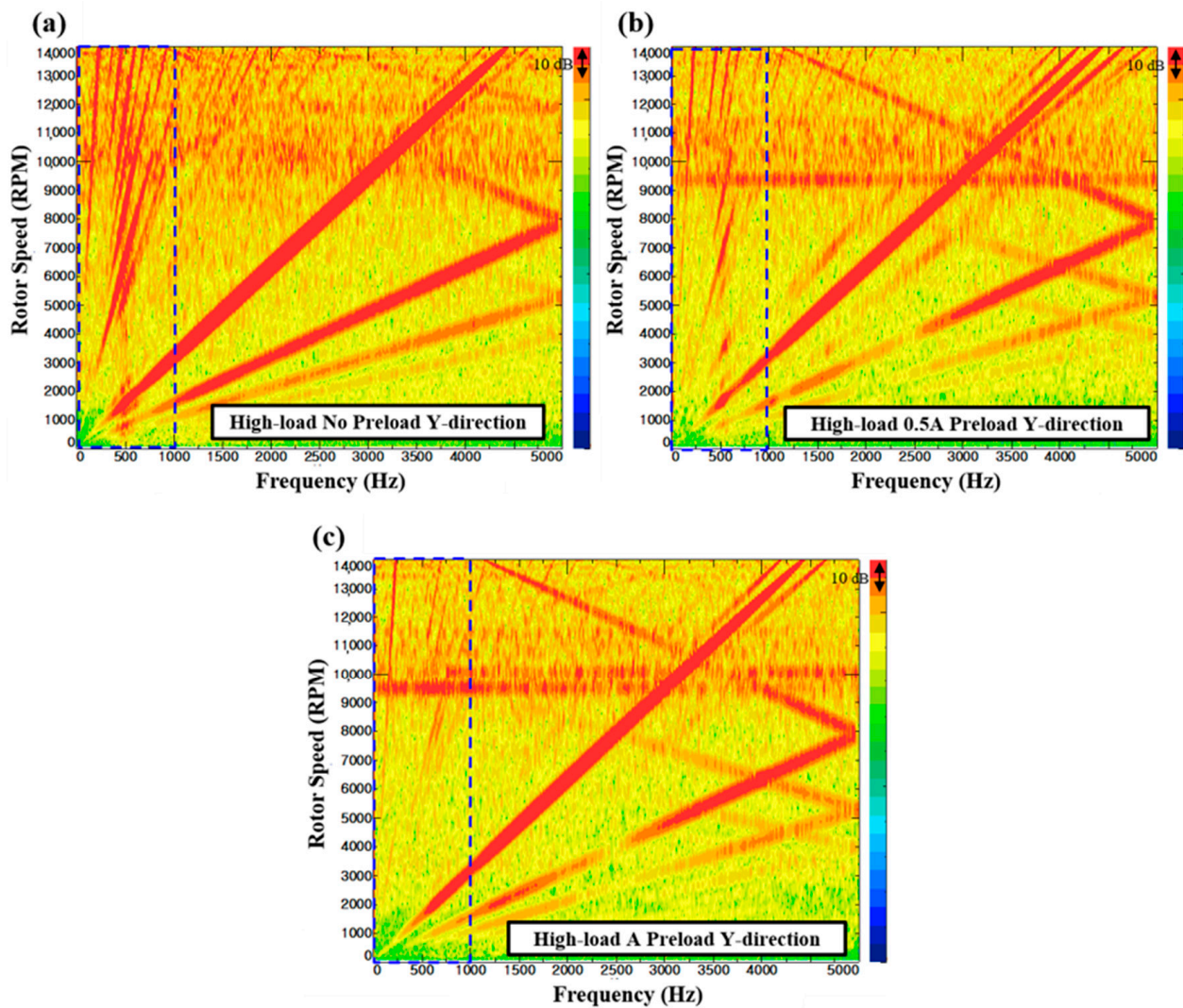


Figure 6. Vibration colormaps of the B3 bearing under high-load conditions: (a) No preload, Y-direction, (b) 0.5A preload, Y-direction, and (c) A preload, Y-direction.

3.3. Order-Tracking Analysis

Order-tracking analysis was employed to separate and visualize vibration data according to rotational orders as a function of shaft speed. Based on the colormap analysis results, additional comparative analyses were conducted for the dominant shaft rotational orders (1st, 2nd, 3rd, and 5th) that exhibited noticeable sensitivity to the WW preload. Through this approach, the trends in specific order vibration responses according to preload variation were more clearly identified. The no-preload condition was used as the baseline reference for all vibration comparisons. The reported vibration reductions correspond to the maximum reduction in order-tracked vibration amplitude relative to the no-preload condition.

3.3.1. Order-Tracking Analysis Under Low-Load Conditions

Figure 7 presents the order-tracking analysis results of the B3 bearing under the low-load condition with varying WW preload. Figure 7a, 7b, 7c, and 7d correspond to the 1st-, 2nd-, 3rd-, and 5th-order components, respectively. Each component represents the Nth-order vibration associated with shaft rotation, and clear differences in vibration levels are observed within the frequency band around 400 Hz depending on the applied WW preload.

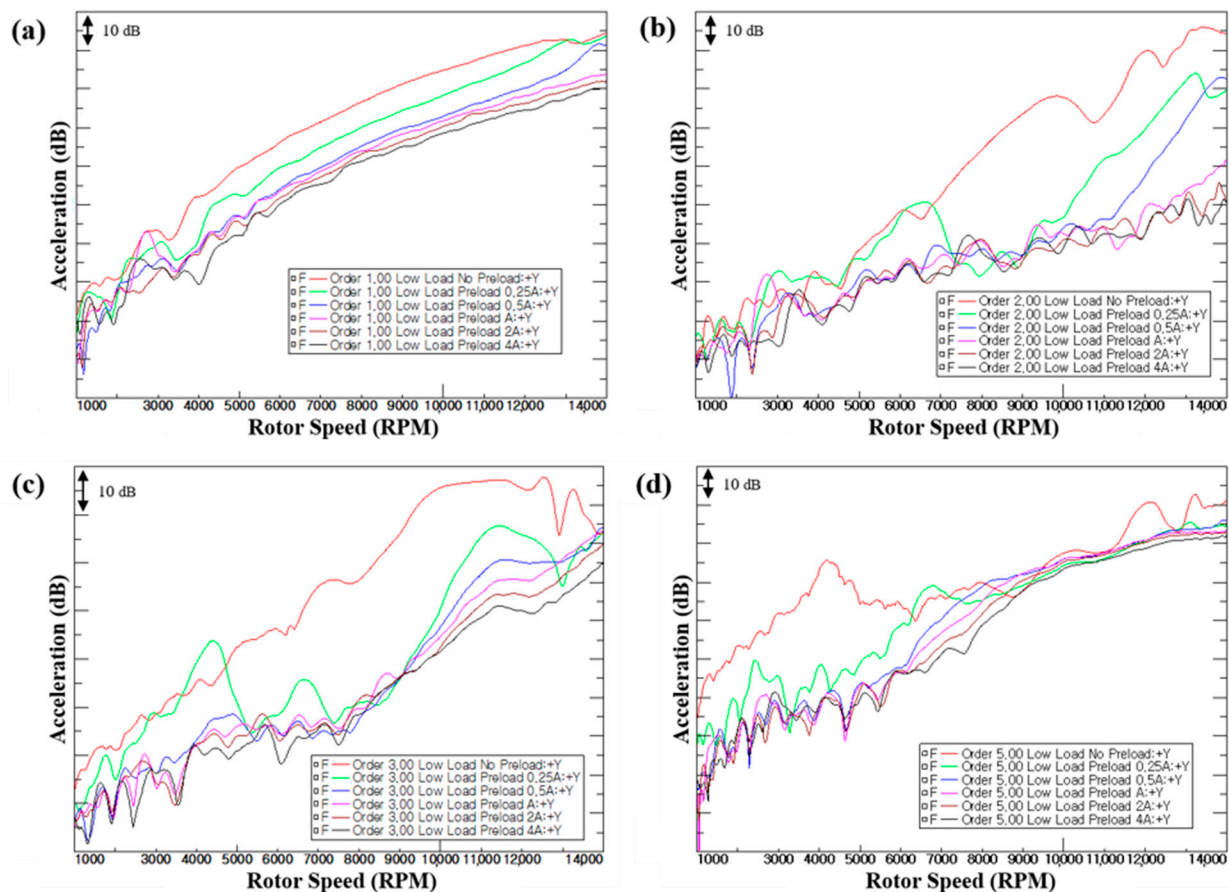


Figure 7. Order-tracking vibration curves of the B3 bearing under low-load conditions for different preload levels: (a) 1st, (b) 2nd, (c) 3rd, and (d) 5th orders.

For the 1st-order component shown in Figure 7a, vibration reduction is evident over the entire RPM range when the WW is applied. Relative to the no-preload condition, the peak order-tracked vibration amplitude is reduced by more than approximately 15 dB above 4000 RPM. For the 2nd-order component shown in Figure 7b, the vibration amplitude tends to increase with RPM when the preload is insufficient. Relative to the no-preload condition, the peak order-tracked vibration amplitude in the 8000–11,000 RPM range is reduced by approximately 30 dB when the preload increases to level A; however, further improvements beyond this level are limited.

The 3rd-order component shown in Figure 7c exhibits a trend similar to that of the 2nd-order component, where insufficient preload leads to increased vibration at higher RPMs. Relative to the no-preload condition, the peak order-tracked vibration amplitude in the 8000–12,000 RPM range is reduced by approximately 20 dB when the preload increases to level A.

For the 5th-order component shown in Figure 7d, relative to the no-preload condition, the peak order-tracked vibration amplitude is reduced by up to approximately 20 dB in the 4000–6000 RPM range at preload levels of A and above. In contrast, the vibration reduction achieved at the preload level of 0.25A is approximately 10 dB lower than that achieved under preload levels of A and above, indicating a noticeable preload-dependent difference in vibration suppression performance in the corresponding RPM range. Although vibration reduction is observed up to the preload level A, no significant change is noted beyond this level.

Overall, as shown in Figure 7, increasing the WW preload up to level A results in vibration reduction across all examined orders (1st, 2nd, 3rd, and 5th). However, the

additional influence of WW preload becomes marginal at levels above A. These results indicate that reinforcing bearing stiffness through an appropriately preloaded WW is necessary within specific RPM ranges and is effective for vibration reduction under low-load operating conditions.

3.3.2. Order-Tracking Analysis Under High-Load Conditions

Figure 8 presents the order-tracking analysis results under the high-load condition with varying WW preload. Figure 8a, 8b, and 8c show the Y-direction (radial) 1st-, 2nd-, and 5th-order components, respectively. In addition, Figure 8d,e present the Z-direction (axial) 2nd- and 3rd-order components. Each order corresponds to the Nth-order component of shaft rotation, and distinct differences are observed in the vibration response within the frequency band around 400 Hz depending on the WW preload.

For the 1st-order component shown in Figure 8a, relative to the no-preload condition, the peak order-tracked vibration amplitude is reduced by up to approximately 15 dB above 12,000 RPM at the preload level of 4A. In contrast, the vibration reduction achieved at the preload level of 0.25A is approximately 5 dB lower than that obtained at preload levels of 2A and above, suggesting that insufficient preload leads to increased peak order-tracked vibration amplitude in the high-RPM region.

For the 2nd-order component shown in Figure 8b, relative to the no-preload condition, the peak order-tracked vibration amplitude is reduced by up to approximately 15 dB above 10,000 RPM at preload levels of A and above. In contrast, the vibration reduction achieved at the preload level of 0.25A is approximately 5 dB lower than that achieved under preload levels of A and above, indicating a noticeable preload-dependent difference in vibration suppression performance in the high-RPM region.

For the 5th-order component shown in Figure 8c, relative to the no-preload condition, the peak order-tracked vibration amplitude is reduced by up to approximately 20 dB in the 4000–8000 RPM range at preload levels of A and above. In contrast, the vibration reduction achieved at the preload level of 0.25A is approximately 10 dB lower than that achieved under preload levels of A and above, indicating a noticeable preload-dependent difference in vibration suppression performance in the mid-RPM region.

In the Z-direction (axial), for the 2nd-order component shown in Figure 8d, relative to the no-preload condition, the peak order-tracked vibration amplitude is reduced by up to approximately 10 dB in the 8000–11,000 RPM range at preload levels of 2A and above. In contrast, the vibration reduction achieved at the preload level of 0.25A is approximately 5 dB lower than that achieved under preload levels of 2A and above, indicating a noticeable preload-dependent difference in vibration suppression performance in the corresponding RPM range.

Likewise, for the 3rd-order Z-direction (axial) component shown in Figure 8e, relative to the no-preload condition, the peak order-tracked vibration amplitude is reduced by up to approximately 15 dB in the 6000–8000 RPM range at preload levels of 2A and above. In contrast, the vibration reduction achieved at the preload level of 0.25A is approximately 10 dB lower than that achieved under preload levels of 2A and above, indicating a noticeable preload-dependent difference in vibration suppression performance in the corresponding RPM range.

Across all examined order components, increasing the WW preload up to 2A results in an overall reduction in the order-tracked vibration levels; however, additional effects beyond the 2A preload level are limited. These results indicate that, even under high-load operating conditions, reinforcing bearing stiffness through the application of a WW with an appropriate preload is required within specific RPM ranges and is effective in reducing vibration levels.

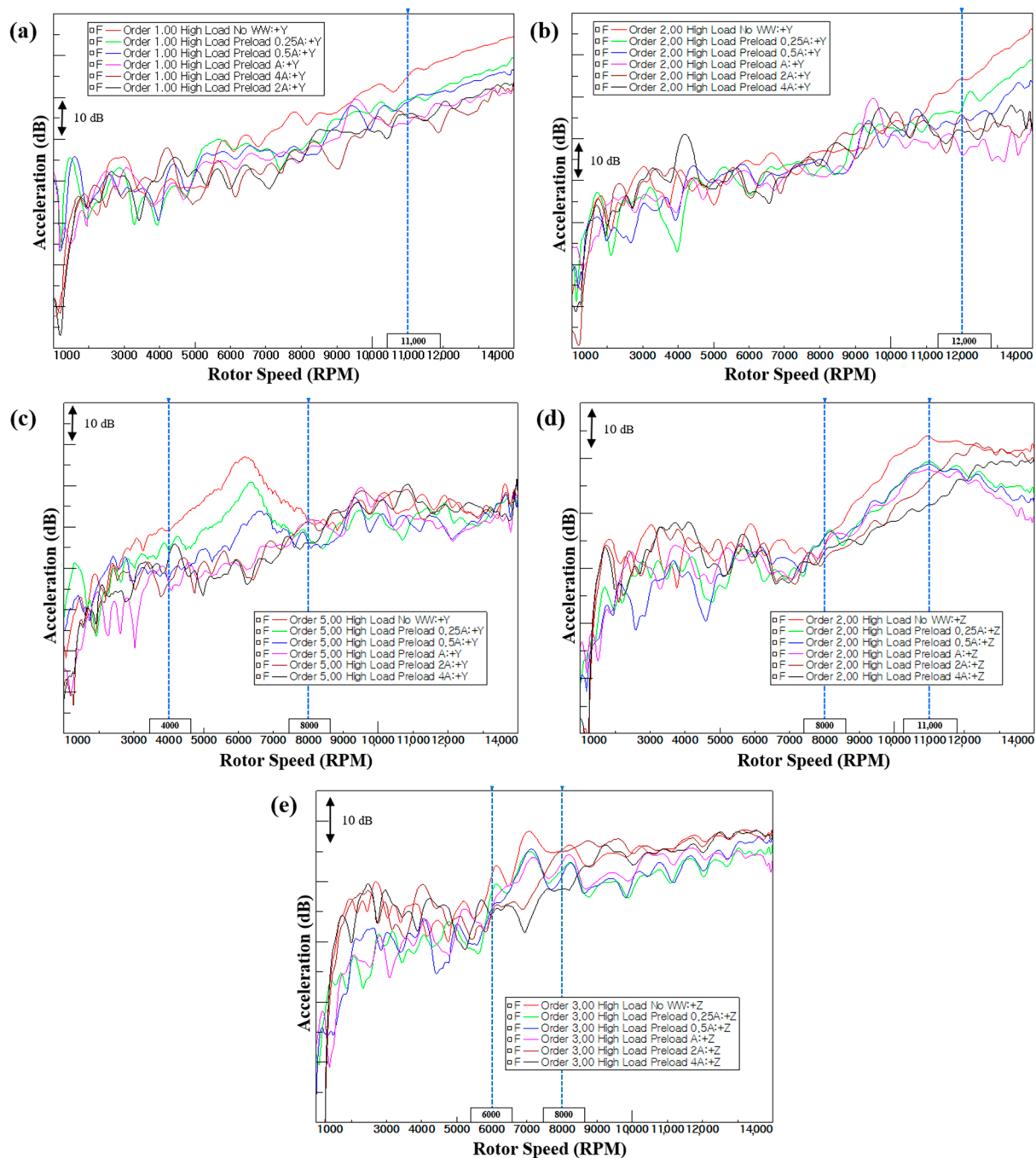


Figure 8. Order-tracking vibration curves of the B3 bearing under high-load conditions for different preload levels: (a) 1st, (b) 2nd, and (c) 5th orders in the Y-direction; (d) 2nd and (e) 3rd orders in the Z-direction.

Furthermore, by comparing the results shown in Figures 7 and 8, it is observed that the application of a WW under low-load conditions leads to vibration reduction over a wider RPM range and is relatively more effective than under high-load conditions. The preload required to achieve vibration stabilization under the high-load condition is 2A, which is higher than the preload level A that was sufficient under the low-load condition. This suggests that, as the applied load increases, the system is subjected to larger excitation forces, thereby requiring a higher preload to achieve an equivalent stabilizing effect.

4. Discussion

In this study, changes in the vibration response of an EDU system according to bearing preload variation were analyzed, and the influence of preload on vibration characteristics was investigated from a physical perspective. The discussion focuses on the mechanisms responsible for vibration reduction with increasing preload and the tendency for additional improvements to diminish beyond a certain preload level. Based on Hertzian contact theory and the analytical bearing formulation proposed by Hernot et al. [14], together with the assumptions adopted in the present study, a simplified analytical preload–stiffness relationship was introduced to provide a physical interpretation of the observed preload-dependent vibration behavior. It should be noted that the analytical formulation was not used to define the bearing stiffness characteristics in the MBD model, but rather to interpret the preload-dependent vibration behavior observed in the simulations.

For simplicity, the present derivation assumes that:

1. the preload is applied as a pure axial load;
2. all rolling elements remain in continuous contact with the raceways under preload;
3. the load is uniformly distributed among all rolling elements;
4. the contact angle is assumed to remain constant throughout the derivation.

$$F_a = C\delta_a^{3/2} \quad (1)$$

where F_a is the applied axial preload, δ_a is the axial deformation induced by preload, and C is a constant determined by the bearing geometry and contact conditions. Since the axial stiffness K_a is defined as the slope of the load–displacement relationship,

$$K_a = \frac{dF_a}{d\delta_a} \quad (2)$$

By differentiating the above equation and eliminating the deformation term δ_a , the following relationship can be obtained:

$$K_a \propto F_a^{1/3} \quad (3)$$

This relationship indicates that the axial stiffness of the bearing increases in proportion to the cube root of the applied preload. Consequently, stiffness increases continuously with preload, whereas the rate of stiffness increase gradually decreases, resulting in a nonlinear preload–stiffness relationship. In addition, preload reduces the internal clearance of the bearing and stabilizes the contact conditions between the rolling elements and raceways, producing a relatively large vibration reduction effect at low preload levels. However, once sufficient preload is achieved, the additional stiffness gain becomes relatively limited, leading to a gradual saturation of the vibration reduction effect.

The simulation results presented in Section 3 indicate that the preload level required for vibration stabilization varies with operating load conditions. Under low-load conditions, vibration reduction tended to saturate at approximately preload level A, whereas under high-load conditions, stabilization was observed only at approximately 2A. This tendency suggests that preload effectiveness is governed not only by preload-induced stiffness enhancement but also by the magnitude of the excitation forces acting on the bearing–shaft system. Under higher loading conditions, larger excitation forces may be present, requiring higher preload levels to achieve sufficient contact stabilization and vibration suppression. Therefore, the observed load-dependent differences in vibration stabilization cannot be explained solely by the simplified preload–stiffness relationship but should be

interpreted as the combined effect of preload-induced stiffness enhancement and operating-load-dependent excitation conditions.

Furthermore, preload influences not only vibration amplitude but also the dynamic characteristics of the system. The simulation results shown in Figure 3 indicate that the dominant vibration response shifted toward higher rotational-speed regions as preload increased. This tendency can be interpreted as a consequence of the increase in the natural frequency of the shaft-bearing system caused by the preload-induced increase in bearing stiffness. Therefore, preload should be regarded not only as a means of vibration reduction but also as a design parameter capable of altering the resonance characteristics and vibration-sensitive operating regions of the system.

Overall, the vibration reduction effect observed with increasing preload, as well as the diminishing return of this effect at higher preload levels, can be explained by the combined influences of contact-condition stabilization and the nonlinear preload-stiffness relationship. In addition, the associated changes in system dynamic characteristics appear to play an important role in the observed vibration behavior.

5. Conclusions

In this study, the bearing vibration characteristics of an Electric Drive Unit (EDU) were investigated according to the axial preload level of a Wave Washer (WW) using Multi-Body Dynamics (MBD) analysis. In addition, a Hertzian-contact-based analytical stiffness formulation was derived and employed to interpret the preload-dependent vibration behavior observed in the simulations. The main conclusions are summarized as follows.

- (1) Under low-load conditions, the axial (Z) direction acceleration response exhibited a tendency for the vibration peak region to shift toward higher RPM ranges as the WW preload increased up to a certain level. This behavior suggests that the application of axial preload by the WW increases the effective bearing stiffness.
- (2) Under low-load conditions, vibration reduction was observed in the radial (Y) direction acceleration at high RPMs even with relatively low preload levels. This result demonstrates that axial preload not only influences axial vibration characteristics but also contributes to the reduction in radial vibration.
- (3) When the WW was applied, low-order vibration components around 400 Hz were significantly reduced. This effect is considered to be associated with the reduction in bearing internal clearance and the stabilization of contact conditions resulting from the applied axial preload.
- (4) Based on Hertzian contact theory and the simplified analytical formulation adopted in this study, the preload-dependent vibration behavior can be interpreted using the relationship ($K_a \propto F_a^{1/3}$), and the vibration reduction effect was found to exhibit a saturation tendency beyond a certain preload level.
- (5) The vibration reduction effect of the WW was more pronounced under low-load conditions than under high-load conditions. Under high-load operation, the influence of preload becomes relatively smaller compared to the system load, thereby requiring a higher preload level to achieve a comparable vibration reduction effect.
- (6) The vibration reduction effect of the WW was most prominent in low-order components (1st, 2nd, 3rd, and 5th orders), with each order exhibiting different effective RPM ranges. Therefore, order-based frequency-response analysis using order-tracking techniques is essential for predicting and designing optimal WW application conditions for vibration reduction.

Through this study, cases in which vibration levels were reduced by more than 30 dB around 400 Hz were confirmed when the WW was applied, demonstrating a clear improvement in NVH performance. In addition, a recurring tendency of significant vibra-

tion increase under insufficient preload conditions was observed across multiple orders. When vibration-sensitive orders overlap with critical NVH regions in actual vehicle operation, the application of a WW may provide valuable baseline design guidance for EDU system development.

However, the present study was conducted based on MBD simulations and does not fully account for all variables encountered in real driving environments. In particular, various load fluctuations and operating conditions that may occur in vehicle operation were not explicitly considered. Therefore, the results presented herein are limited to trend analysis based on simulation, and quantitative validation under real vehicle conditions remains necessary. In addition, temperature- and friction-related effects were not explicitly considered in the present model and may influence the actual vibration response of the bearing system through changes in contact conditions and effective bearing stiffness. Furthermore, the present study focused specifically on the influence of axial preload variation introduced by the WW while maintaining the remaining drivetrain design parameters unchanged. Future studies should consider experimental verification under various load and environmental conditions to improve the accuracy of the analysis and establish practical design criteria applicable to real EDU systems.

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Nomenclature

Roman Symbols

C	Constant determined by bearing geometry and contact conditions
F_a	Applied axial preload (N)
K_a	Axial stiffness of the ball bearing (N/mm)

Greek Symbols

δ_a	Axial deformation induced by preload (mm)
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Abbreviations

EDU	Electric Drive Unit
MBD	Multi-Body Dynamics
NVH	Noise, Vibration, and Harshness
WW	Wave Washer

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