

Article

Effect of Groove Structures on Lubrication and Vibration Characteristics of Multi-Layer Composite Water-Lubricated Bearings

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Abstract

To meet the demand for water-lubricated bearings (WLBs) with low vibration, low noise and high load-carrying capacity in propulsion systems, this study designed and tested a three-layer composite WLB consisting of an inner phenolic working layer, a middle rubber damping layer and a glass-fiber-reinforced composite layer. The lubrication, vibration and wear behaviors of three bearings with different groove structures, namely a non-grooved bushing, a fully straight-grooved bushing and a fully spiral-grooved bushing, were comparatively investigated under combined variations in rotational speed (20–400 r/min), specific pressure (0.18–0.8 MPa) and water flow rate (5–20 L/min). The results demonstrate that both specific pressure and flow rate strongly govern the transition from mixed lubrication to hydrodynamic lubrication and the associated vibration response. As the specific pressure and water flow rate increase, the transition speed and coefficient of friction of grooved bearings, particularly straight-grooved bearings, increase markedly. Non-grooved bearings consistently maintain the lowest levels, while spiral-grooved bearings exhibit lubrication performance intermediate between the above two types. Under low-speed and heavy-load conditions, non-grooved bearings show the smallest increase in vibration amplitude. Grooves amplify high-frequency vibrations and inject medium- and high-frequency energy as rotational speed increases. Considering lubrication, vibration control, and wear resistance simultaneously, spiral-grooved bearings exhibit the most robust overall performance under realistic operating conditions. The results provide experimental evidence and practical design guidance for groove-structure selection in multi-layer composite WLBs operating under low-speed and heavy-load conditions.

Keywords: water-lubricated bearing; groove structures; lubrication characteristic; vibration characteristic



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1. Introduction

With the growing emphasis on green shipping and environmental awareness, water-lubricated bearings (WLBs) have found widespread application in mechanical equipment operating in aquatic environments such as ships [1,2]. As critical support components of propulsion systems, the performance and lubrication status of WLBs determine the operational efficiency and safety of ships [3]. During engine start-stop or operation under low-speed or heavy-load conditions, stern bearings are prone to bushing ablation and

spalling, generating severe friction vibrations and noise [4,5]. This significantly impacts the stealth capability, reliability, and crew comfort of underwater equipment.

Vibration and noise caused by friction have been a key research focus in recent years, and existing studies mainly investigate the lubrication and friction vibration behavior of WLBs from the perspectives of lining materials and groove structures. The commonly used materials for WLBs mainly include rubber (NBR), ultra-high molecular weight polyethylene (UHMWPE), polyamide (PA), phenolic resin-based composite materials, fiber-reinforced composite materials, etc. [6–11]. Kuang et al. [12] combined friction-vibration measurements on rubber stern bearings at high speed with a coupled model, revealing how rubber layer deformation and contact state affect friction-induced vibration. By incorporating different two-dimensional materials into cotton-reinforced phenolic resin laminates, Guo et al. reduced coefficient of friction (COF) fluctuations and mitigated stick-slip wear, thereby suppressing friction-induced vibration and noise [13]. Owing to the superior tribological performance under low-speed, heavy-load conditions compared with traditional rubber materials [14], phenolic resin-based composites were adopted as the inner lining material in this study.

As propulsion shaft systems evolve toward larger diameters, higher loads, and lower noise requirements, traditional single-layer bushing WLBs are increasingly unable to ensure adequate load-carrying, stability, and noise control. In response, multi-layer WLBs have been proposed, usually incorporating a rubber layer between the polymer bushing and the metal housing to enhance the lubrication performance of water-lubricated stern bearings [15]. Litwin et al. [16] experimentally investigated PTFE-NBR-bronze three-layer WLBs and showed that they operate in hydrodynamic lubrication under most working conditions, with lower start-up and running resistance than NBR bearings. Shen et al. [17] further developed fluid–structure interaction models for composite WLBs and revealed the effects of operating conditions and structural parameters on the lubrication performance of rubber-plastic bearings. These results highlight the advantages of multi-layer composite WLBs in demanding marine environments.

The arrangement, number and geometry of lubrication grooves strongly influence the hydrodynamic pressure distribution and thus the overall bearing performance [18]. Experimental studies by Litwin [19] showed that groove shape and distribution (e.g., square or circular grooves, uniform or local arrangements) significantly affect motion resistance and are decisive for the occurrence of hydrodynamic phenomena in three-layer bearings. Xie et al. [20] demonstrated that lubrication grooves can attenuate vibration and reduce wear by comparing the COF and vibration amplitudes of non-grooved and fully grooved WLBs. Wu et al. [21] further reported that groove type, width and depth significantly affect pressure drop and recovery in the groove region and thereby modulate elastohydrodynamic lubrication. In addition to square and circular grooves, spiral structures have recently been applied to WLBs because they can provide higher load capacity and more stable pressure distributions [22]. However, only a limited number of studies have examined the effects of spiral-grooved structures on water-lubricated radial bearings, with most research focusing on water-lubricated spiral-grooved thrust bearings [23]. Zhou et al. [24] reported that spiral-grooved WLBs exhibit better lubricating performance than straight-grooved WLBs in comparative experiments.

In addition to friction-induced vibration and noise, wear and the associated clearance evolution remain critical concerns for water-lubricated stern bearings operating under frequent start–stop, low-speed and heavy-load conditions [6,25]. Because water has a low viscosity and limited load-carrying capacity at low sliding speeds, mixed lubrication with intermittent asperity contact can persist locally, promoting adhesive wear, abrasive wear and third-body effects [26,27]. Such wear-induced local geometry changes may, in turn, alter

the film formation tendency and the stability of the vibration response. Groove structures can further influence wear by modifying leakage, through-flow, debris evacuation, and local contact intermittency [28,29].

Overall, multi-layer WLBs demonstrate significant potential in load-carrying capacity, lubrication stability and vibration damping owing to the hierarchical design of the materials. However, most existing studies either optimize lining materials or explore the grooving effect under normal working conditions. In contrast, systematic experimental evidence for multi-layer composite WLBs with different groove geometries under the heavy-load, low-speed and low-flow regimes typical of stern bearings remains scarce. In this context, this paper constructs a dedicated test bench and performs comparative tests on non-grooved, straight-grooved and spiral-grooved three-layer composite WLBs. By jointly analyzing the lubrication characteristics, frequency-resolved vibration responses and spatial wear distributions under heavy-load and low-flow conditions, the work aims to (i) reveal the mechanisms by which groove structures regulate hydrodynamic film formation and frictional excitation, and (ii) provide guidance for the structural design of high-performance WLBs in marine propulsion applications.

2. Test Bench and Test Procedure

2.1. Test Bench

The performance tests for the WLBs were carried out on the test bench schematically illustrated in Figure 1 and shown as a photograph in Figure 2. The bench primarily consisted of the drive module, transmission module, loading module, lubrication module and testing module. The test shaft was made of 45# steel, and the journal section in contact with the bushing was coated with tungsten carbide. The spindle was driven by a 30 kW servo motor with a maximum speed of 3000 r/min. Radial loads were applied to bearings 1# and 2# by a hydraulic loading system, and the load carried by the WLBs equaled the sum of the front and rear hydraulic forces. Two force sensors mounted above the loading device were used to monitor the applied loads. Cooling and lubricating water was supplied from a recirculating lubrication system. The bearing housing had an inlet at one end and an outlet at the other. Water from the supply tank was pressurized by a pump and injected into the test bearings through the inlet at the rear end, and then discharged from the outlet at the front end. Two triaxial accelerometers were mounted on the left and right ends of the bearing housing to measure vibration acceleration in the X (horizontal) and Z (vertical) directions. The detailed parameters of test sensors are listed in Table 1.

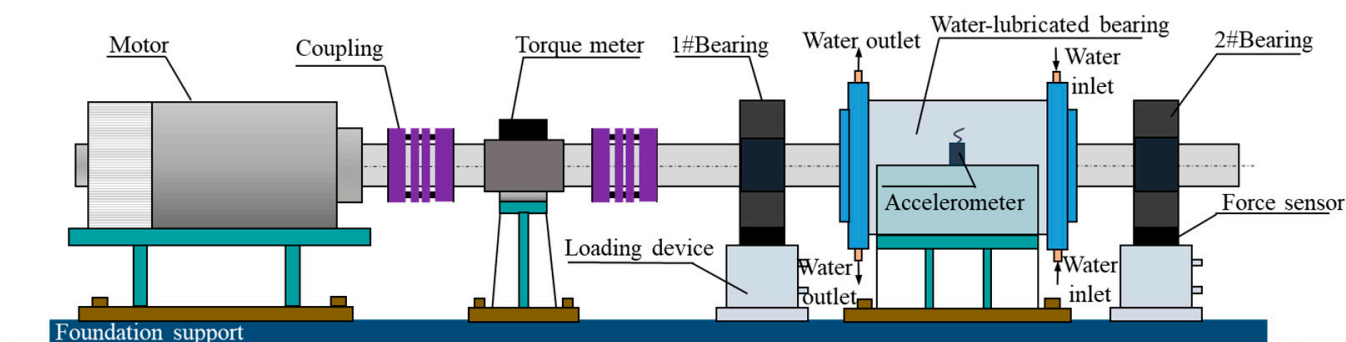
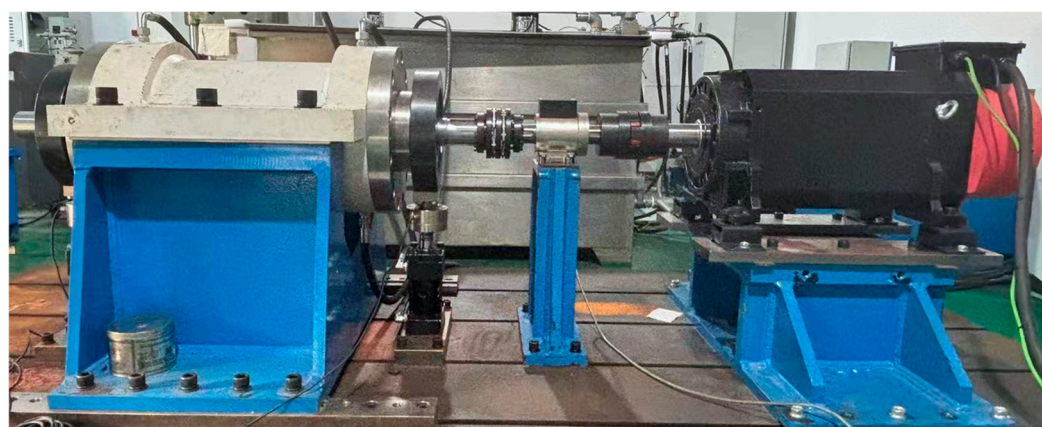


Figure 1. The structure of water-lubricated bearing test bench.

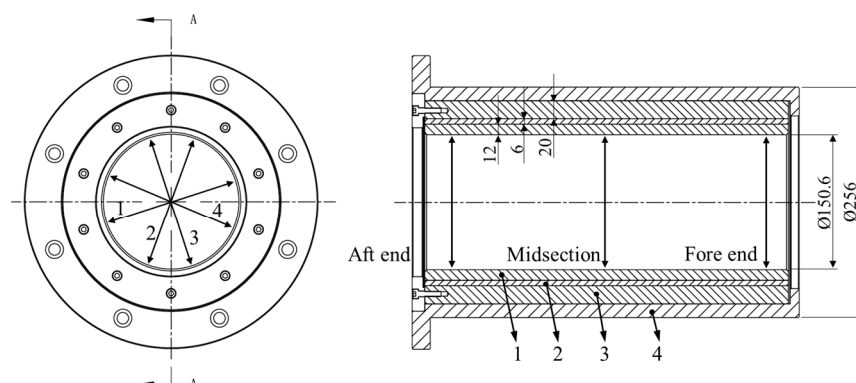
Table 1. Parameters of test sensors.

Part	Parameters	Value
Torque meter	Brand	Force China
	Measure range	$\pm 1000 \text{ N}\cdot\text{m}$
	Accuracy class	0.5
Load sensor	Brand	SMOWO
	Measure range	0–50 kN
	Sensitivity	$2.0 \pm 0.5\% \text{ mV/V}$
Flow meter	Brand	XMSJ
	Measure range	4–50 L/min
	Sensitivity	$\pm 1\%$
Accelerometer	Brand	DHDAS
	Sensitivity	$9.89 \text{ mV}/(\text{m/s}^2)$
	Frequency range	10–10,000 Hz

**Figure 2.** Physical diagram of water-lubricated bearing test bench.

2.2. Test Bearing

The test bearing employed a three-layer composite WLB. As shown in Figure 3, the composite bushing was press-fitted into the steel bearing shell, and the three-layer bearing liner consisted of an inner phenolic working layer, a middle rubber damping layer, and a glass-fiber reinforced composite layer. The basic properties of the three-layer bearing material are listed in Table 2.



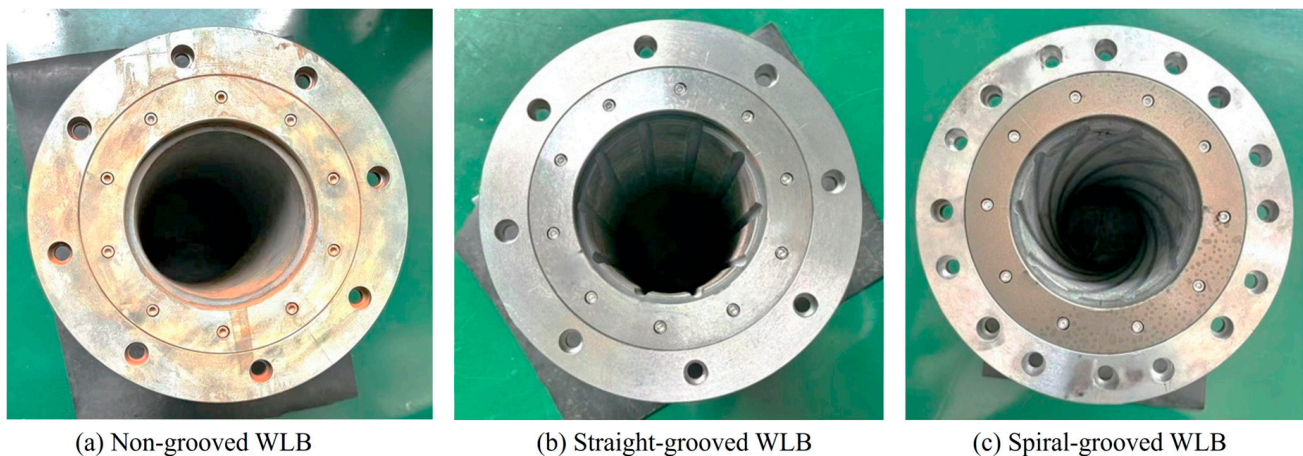
1-Inner phenolic working layer; 2-Middle rubber damping layer; 3-Glass-fiber reinforced composite layer; 4-Bearing shell

Figure 3. Schematic diagram of three-layer composite WLB structure and wear measurement locations. (Unit: mm).

Table 2. The material properties of the three-layer bearing bushes.

Material	Density (g/cm ³)	Elastic Modulus (MPa)	Poisson Ratio	Hardness
Phenolic resin-based composites	1.3	8500	0.3	85 (shore D)
Rubber	1.15	20	0.47	75 (shore A)
Glass-fiber reinforced composite	1.8	10,000	0.29	88 (shore D)

Each type of WLB tested had a distinctive structural design, as illustrated in Figure 4. Bearing I featured a non-grooved bushing. Grooved bearings incorporated lubrication grooves along the entire circumference of the bushing. Bearing II adopted a circular straight-grooved design, while Bearing III employed a spiral-grooved structure with a spiral angle of 22°. The detailed structural parameters of the three bearings are summarized in Table 3.

**Figure 4.** Physical diagram of water-lubricated bearings with different groove structures.**Table 3.** Structural parameters of water-lubricated bearings.

Structural Parameters	Bearing I	Bearing II	Bearing III
Lining radius r /mm	75	75	75
Bearing length L /mm	403	403	403
Radius clearance c /mm	0.3	0.3	0.3
Groove count	0	10	10
Groove depth h /mm	0	6	6

2.3. Test Procedure

The overall test procedure for the WLBs is illustrated in Figure 5. During the preparation stage, water absorption tests were first performed on the WLBs for 48 h, and the inner diameter of each WLB was measured every 24 h at room temperature. Subsequently, thermal expansion tests were carried out across a temperature range from -5°C to 60°C . To ensure a stable frictional contact state between the shaft and bearing, a 24 h continuous running-in test was then carried out, consisting of three cycles of 8 h each. In order to evaluate the lubrication performance of WLBs with different groove structures, two types of tests were conducted: load characteristics and water flow rate characteristics. The load characteristics tests were performed at specific pressures of 0.18, 0.28, 0.5 and 0.8 MPa,

while the flow rate characteristics were assessed at water flow rates of 5, 10 and 20 L/min. The specific pressure was calculated as follows:

$$P = \frac{W}{DL} \quad (1)$$

where P is the specific pressure, MPa; W is the external load borne by the bearing, N; D is the diameter of the test bearing, mm; L is the length of the test bearing, mm.

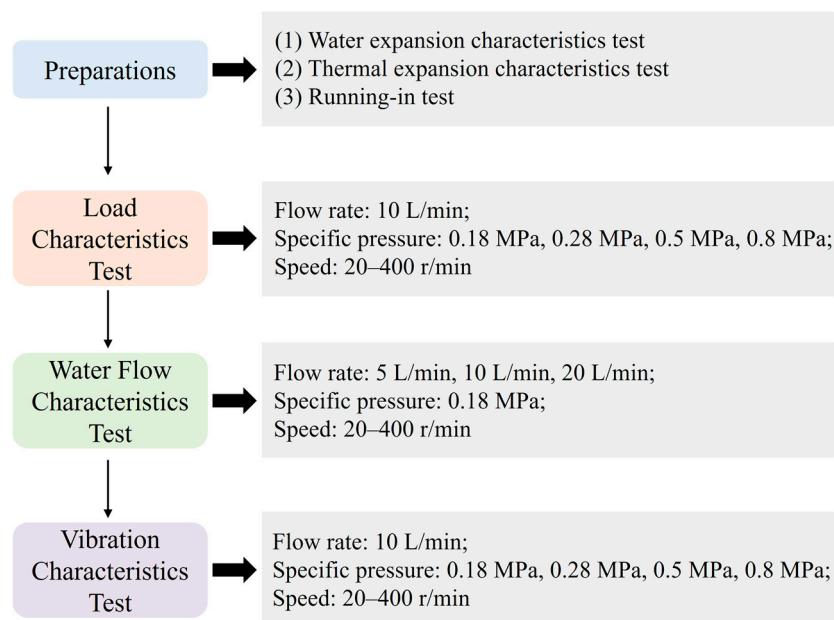


Figure 5. Flowchart of test procedures.

To minimize eccentric loading and misalignment under the double-ended hydraulic loading scheme, the two force sensors were calibrated prior to testing, with the force difference controlled within ± 50 N at each target load. Shaft–housing alignment was verified using a dial indicator before each test, and comparable vibration levels measured at the left and right ends confirmed the absence of abnormal misalignment during operation. The water supply was provided by a recirculating loop. The flow rate was adjusted via a bypass line while the supply pressure was maintained constant using a pressure regulator, and the target flow rate was verified by a flow meter before data acquisition. For each operating condition, after the target rotational speed, specific pressure, and water flow rate were reached and the signals became stable, friction torque and vibration acceleration were recorded for subsequent analysis. The COF was calculated from the measured torque according to Equation (2):

$$f = \frac{M}{RW} \quad (2)$$

where f is the COF; M is the friction torque measured during the experiment, N·m; R is the radius, m; W is the external load borne by the bearing, N.

During the load characteristic tests, vibration signal measurements were conducted simultaneously. To reduce the influence of background vibration, a rubber damping pad was installed under the motor and torque meter. The sampling frequency was 25.6 kHz, the analyzed frequency range was 10–10,000 Hz, and the sampling time for each condition was 30 s. The vibration signals were analyzed both in different frequency bands and in terms of the total vibration level (TVL). In this study, the TVL was expressed in “dB” and defined as the energetic sum of the band-limited vibration levels. The measured spectrum was divided into low-, mid- and high-frequency bands (10–300 Hz, 300–1200 Hz and 1200–10,000 Hz,

respectively), which roughly correspond to the overall rotor–bearing dynamic response, fluid-induced excitation, and friction-related vibration, respectively. Let a_L , a_M and a_H denote the root mean square (RMS) acceleration in these three bands, the corresponding band levels L_L , L_M , L_H (dB) were given by

$$L_i = 20 \log_{10} \left(\frac{a_i}{a_{ref}} \right) \quad (i = L, M, H) \quad (3)$$

where a_{ref} is the reference acceleration, typically set to 10^{-6} m/s^2 . The TVL over 10–10,000 Hz was then computed as

$$L_{TVL} = 10 \log_{10} (10^{L_L/10} + 10^{L_M/10} + 10^{L_H/10}) \quad (4)$$

For vibration analysis, each 30 s record was divided into five consecutive 6 s sub-windows. TVL (and the band-limited vibration levels, if applicable) was computed for each sub-window, and the mean value across the five sub-windows was reported as the representative vibration level for that condition. The corresponding standard deviation across the five sub-windows was reported as an estimate of within-record variability (i.e., time fluctuation within a single record rather than repeat tests).

Prior to power spectral density (PSD) analysis, each vibration record was detrended (mean removed) and high-pass filtered (4th-order IIR; half-power frequency of 0.1 Hz) to suppress low-frequency drift. The PSD was estimated using Welch’s method: the signal was divided into overlapping segments, windowed with a Hann window, and averaged across segments. For the full-band frequency–speed map, a 1 s window with 50% overlap and an N_{FFT} of 32,768 was used. For low-frequency inspection, a 5 s window with 50% overlap was employed, together with a larger window to improve frequency resolution, and the PSD was plotted over 0–50 Hz.

After each test, the bench was shut down, the test bearing was removed, and its wear was measured using an inside micrometer with a resolution of 0.001 mm. The measurement schematic is shown in Figure 3. Taking the installation position of the bearing on the test bench as a reference, measurements were made at three axial sections along the bearing: the fore end (motor side), the midsection, and the aft end (away from the motor). The longitudinal section of the bearing served as the axial reference, and wear measurements were conducted in a cross-shaped pattern. Wear amount was defined as the difference between the initial and post-test inner diameter at each measurement position.

3. Results and Discussion

3.1. Load Characteristics Test

Figure 6 presents the variation in the COF with rotational speed for WLBs under specific pressures ranging from 0.18 to 0.8 MPa at a constant water flow rate of 10 L/min. Overall, the COF exhibits a pronounced downward trend with increasing rotational speed, remaining stable in the high-speed range. This behavior indicates that, as the speed increases, the lubrication state gradually changes from boundary or mixed lubrication to hydrodynamic lubrication. Here, the rotational speed corresponding to the onset of the COF plateau (i.e., the rapid-decrease region transitioning to gradual change) is referred to as the transition speed. As shown in Figure 6a,b, at 0.18 MPa and 0.28 MPa, the transition speed of the non-grooved bearing is approximately 80 r/min, whereas that of the grooved bearings reaches about 120 r/min. As shown in Figure 6d, at 0.8 MPa, the transition speed of the non-grooved bearing shifts to around 120 r/min, while the straight-grooved bearing further increases to about 280 r/min and the spiral-grooved bearing to roughly 160 r/min. It can be observed that increasing specific pressure shifts the transition speeds of all three

bearing configurations towards higher values, with the straight-grooved bearing exhibiting the largest shift and the non-grooved bearing the smallest. This shift suggests a delayed transition toward a more hydrodynamic-dominated regime as the load increases.

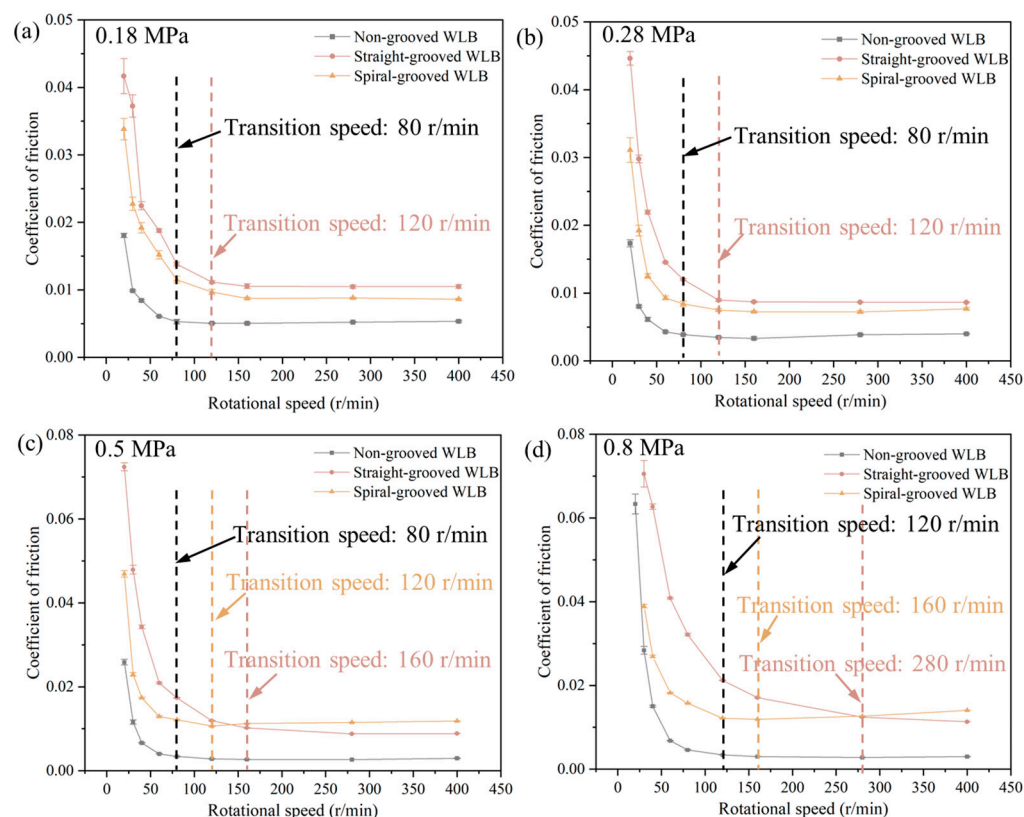


Figure 6. Variation of the COF of WLBs with rotational speed at a water flow rate of 10 L/min under different specific pressures: (a) 0.18 MPa; (b) 0.28 MPa; (c) 0.5 MPa; (d) 0.8 MPa.

Under the same specific pressure and speed, the non-grooved bearing shows the lowest COF. This outcome is consistent with reduced leakage and the strongest wedge effect, although leakage and film pressure were not directly measured. For bearings with a global groove structure, the grooves divide the continuous hydrodynamic wedge into multiple segments. The water film thickness increases within the grooves, which may reduce the local hydrodynamic pressure and, consequently, the effective load-carrying capacity [29]. As a result, the COF of grooved bearings is higher than that of the non-grooved bearing. The spiral-grooved bearing exhibits intermediate COF, which is consistent with a pumping-assisted replenishment mechanism reported in prior comparative studies [24]. It is worth noting that under low-speed and heavy-load conditions, the friction torque of both straight-grooved and spiral-grooved bearings increases dramatically within a short period, leading to lock-up. Consequently, the COF data for grooved bearings at 20 r/min are absent in Figure 6d.

3.2. Water Flow Characteristics Test

Figure 7 shows the relationship between the COF and rotational speed for WLBs under a specific pressure of 0.18 MPa and water flow rates ranging from 5 to 20 L/min. As the rotational speed increases, the COF follows a typical Stribeck-type trend: it first decreases rapidly with speed and then enters a plateau with only gradual changes. Under the present boundary conditions, increasing the supply flow rate may enhance axial through-flow, which weakens the pressure peak of the hydrodynamic wedge and reduces the effective load-carrying capacity. Consequently, a higher linear speed is required to establish a

fully load-carrying water film under otherwise comparable operating conditions. When water flow increases from 5 L/min in Figure 7a to 10 L/min in Figure 6a and 20 L/min in Figure 7b, the transition speed of the non-grooved bearing rises from 60 r/min to 80 r/min. For grooved bearings, the inherent leakage through the grooves is already significant, so the influence of external flow variations on the transition speed is relatively limited and the transition speed remains almost unchanged within the investigated range. Additionally, the additional shear between the shaft and bearing bushings in grooved bearings and pumping work increases viscous dissipation, slightly raising the COF [30]. Simultaneously, the grooves may disrupt the hydrodynamic wedge and guide leakage flow along the axial direction, which could contribute to the higher COF of grooved bearings under identical conditions.

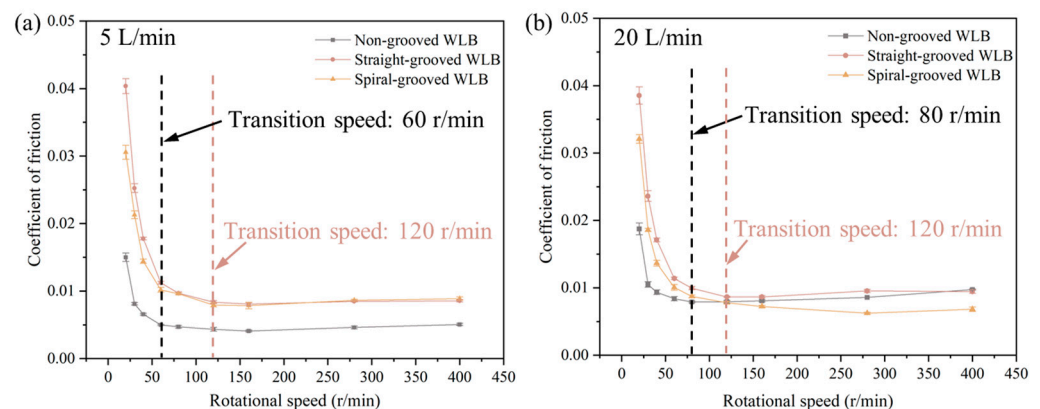


Figure 7. Variation of the COF of WLBs with rotational speed under different water flow rates: (a) 5 L/min; (b) 20 L/min.

3.3. Vibration Characteristics Test

Vibration acceleration signals in the X and Z axes were measured using accelerometers mounted on both sides of the bearing housing plate. The measured TVLs from the two sides were comparable, and no abnormal features were observed. Therefore, for brevity, the following analysis focuses on the X- and Z-direction vibration measured on the left side. To analyze the effect of specific pressure on vibration characteristics, 30 r/min was selected as the reference rotational speed for comparison. This speed represents a low-speed operating regime in the present test campaign. Consequently, at 30 r/min the hydrodynamic contribution is limited and the vibration response is more sensitive to variations in pressure, which facilitates comparing the pressure-dependent trends among different groove configurations.

As shown in Figure 8, the TVLs over the entire frequency range increase monotonically with specific pressure for all three groove configurations. The TVL in the X-axis direction for the non-grooved bearing increases by 5.3%, while that in the Z-axis direction increases by 8.3%. The straight-grooved bearing is more sensitive to specific pressure, with the TVL increasing by 9.1% in the X-axis and 10.3% in the Z-axis. For the spiral-grooved bearing, the TVL changes only slightly in the range of 0.18–0.5 MPa. When the specific pressure increases to 0.8 MPa, the TVL in the X-axis increases by 6.3%, and that in the Z-axis increases by 7.2%. In all cases, the variation in vibration amplitude in the Z-axis direction consistently exceeded that in the X-axis. This suggests that monitoring vibration along the loading direction is more sensitive to load-induced lubrication deterioration and may be advantageous for condition monitoring under the present setup.

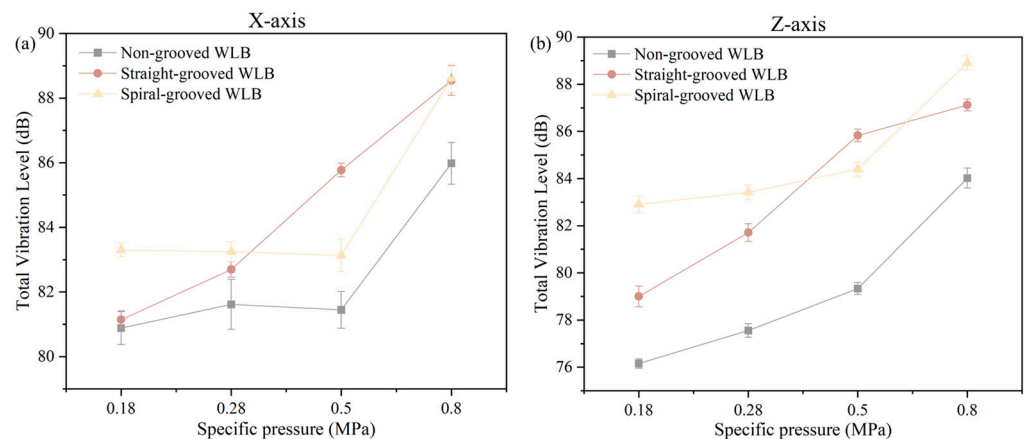


Figure 8. Comparison of total vibration levels of WLBs under different specific pressures at 30 r/min: (a) vibration levels in the X-axis; (b) vibration levels in the Z-axis.

Figures 9–11 show how the band-limited vibration levels in the low-, mid- and high-frequency ranges, as well as the TVL over 10–10,000 Hz, vary with specific pressure for the three WLBs. The band-limited results in Figure 9 indicate that, for the non-grooved bearing, the increase in TVL is mainly concentrated in the low-frequency band, while the mid- and high-frequency components change only slightly. As shown in Figure 10, the band-limited results for straight-grooved bearing exhibit that in addition to the increase in low-frequency levels, the high-frequency band also shows a significant rise, leading to an overall increase in vibration levels. As depicted in Figure 11, the increase in TVL for the spiral-grooved bearing is primarily driven by low-frequency components, accompanied by a moderate rise in the high-frequency band, whereas the mid-frequency range remains almost unchanged. Research by Ouyang et al. [31] revealed that during the mixed lubrication stage, the effective stiffness and damping provided by the water film decrease, leading to amplification of low-order responses. This phenomenon becomes more pronounced as the load increases. For grooved bearings, the abrupt geometric change at the groove edges may induce frictional fluctuations, causing frequency components to shift toward higher frequencies [32]. Furthermore, previous studies indicate that both the shear layer forming along the groove edges and the periodically detached vortices near the groove bottom represent small-scale, rapid processes that similarly transfer energy from the low-frequency band to the mid-to-high-frequency band [30,33,34]. As the specific pressure increases, the thickness of the water film decreases and the velocity gradient increases, which further injects energy into higher frequencies. Accordingly, in grooved bearings the increase in specific pressure leads not only to an enhancement in low-frequency vibration but also to a more pronounced growth in the high-frequency range.

The variation in the band-limited vibration levels with rotational speed at 0.8 MPa is shown in Figure 12. In the low-speed range below 120 r/min, according to Figure 6d, the bearings operate in a mixed lubrication regime. Accordingly, as shown in Figure 12a,b, the vibration amplitudes in the X and Z axes for the non-grooved bearing exhibit a trough between 60 r/min and 80 r/min, followed by a low-frequency rebound near 120 r/min. As illustrated in Figure 12c–f, the straight-grooved bearing exhibits a distinct low-frequency peak between 30 r/min and 120 r/min, while the spiral-grooved bearing shows periodic low-frequency fluctuations. In this regime, the TVL variation is mainly governed by the low-frequency band. As rotational speed continues to increase, the high-frequency vibration amplitudes of the grooved bearings rise significantly, causing the TVL to increase rapidly between 280 r/min and 400 r/min. Similar to the effect of specific pressure, the influence of rotational speed on the Z-axis vibration is more pronounced than on the X-axis.

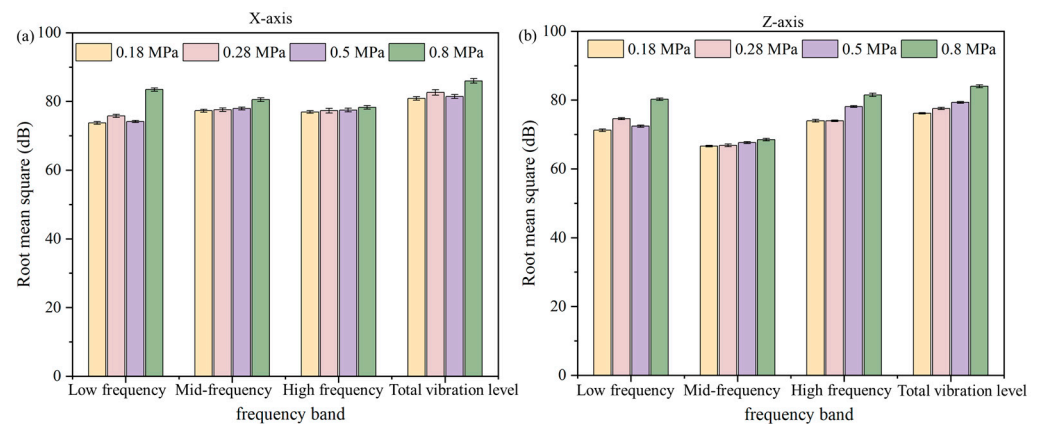


Figure 9. Vibration characteristics of non-grooved WLBs at 30 r/min under different specific pressures: (a) RMS in the X-axis; (b) RMS in the Z-axis.

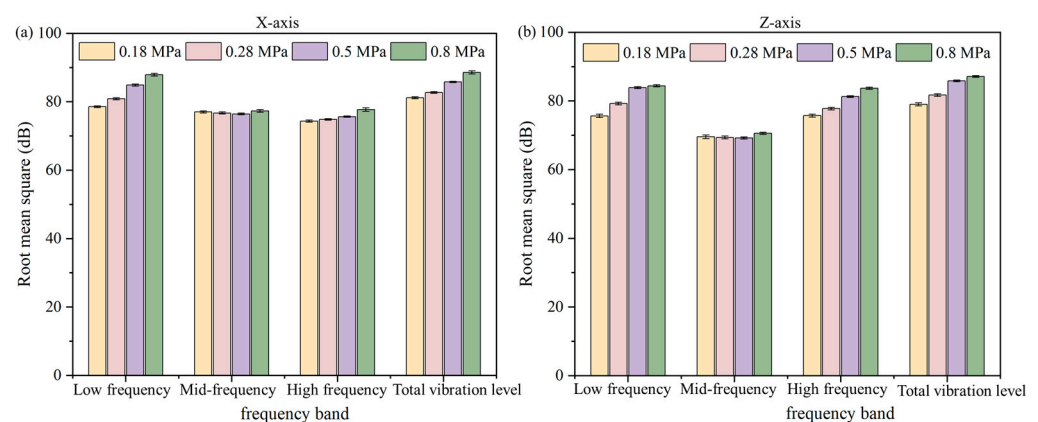


Figure 10. Vibration characteristics of straight-grooved WLBs at 30 r/min under different specific pressures: (a) RMS in the X-axis; (b) RMS in the Z-axis.

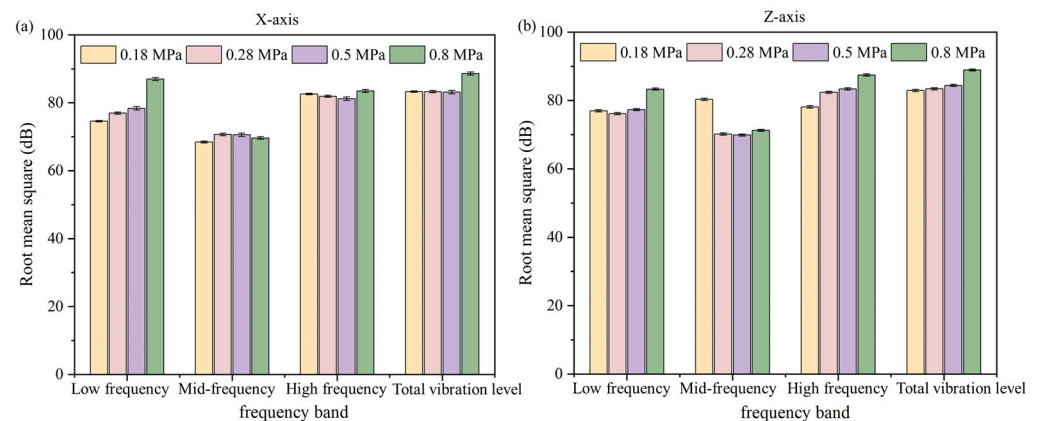


Figure 11. Vibration characteristics of spiral-grooved WLBs at 30 r/min under different specific pressures: (a) RMS in the X-axis; (b) RMS in the Z-axis.

To further illustrate the evolution of frequency components with rotational speed, Figure 13 presents the time-domain waveforms and power spectral density (PSD) distributions across 0–10 kHz for bearings of different structures operating at 0.8 MPa under varying rotational speeds. In the time domain, the signals exhibit a mixed feature of broadband random vibration superimposed with periodic components. As the rotational speed increases, the overall fluctuation level tends to rise, indicating a progressive enhancement of excitation intensity with speed. The PSD map provides a clearer view of how the frequency contents evolve with speed. In the low-frequency range, narrowband components

correlated with speed can be identified, which are attributable to speed-synchronous components and their harmonics. Meanwhile, in the mid-to-high frequency range, a broadband elevation of spectral energy is observed as speed increases, characterized by an overall lift of PSD across a wide band rather than the amplification of a few discrete peaks at fixed frequencies. Such a broadband uplift is often reported in friction-induced vibration and noise scenarios where intermittent asperity contacts and impacts introduce impulsive and stochastic excitations [35]. Accordingly, the observed signature is consistent with an increased contribution of asperity-related friction excitation, although fluid-induced excitation cannot be fully ruled out.

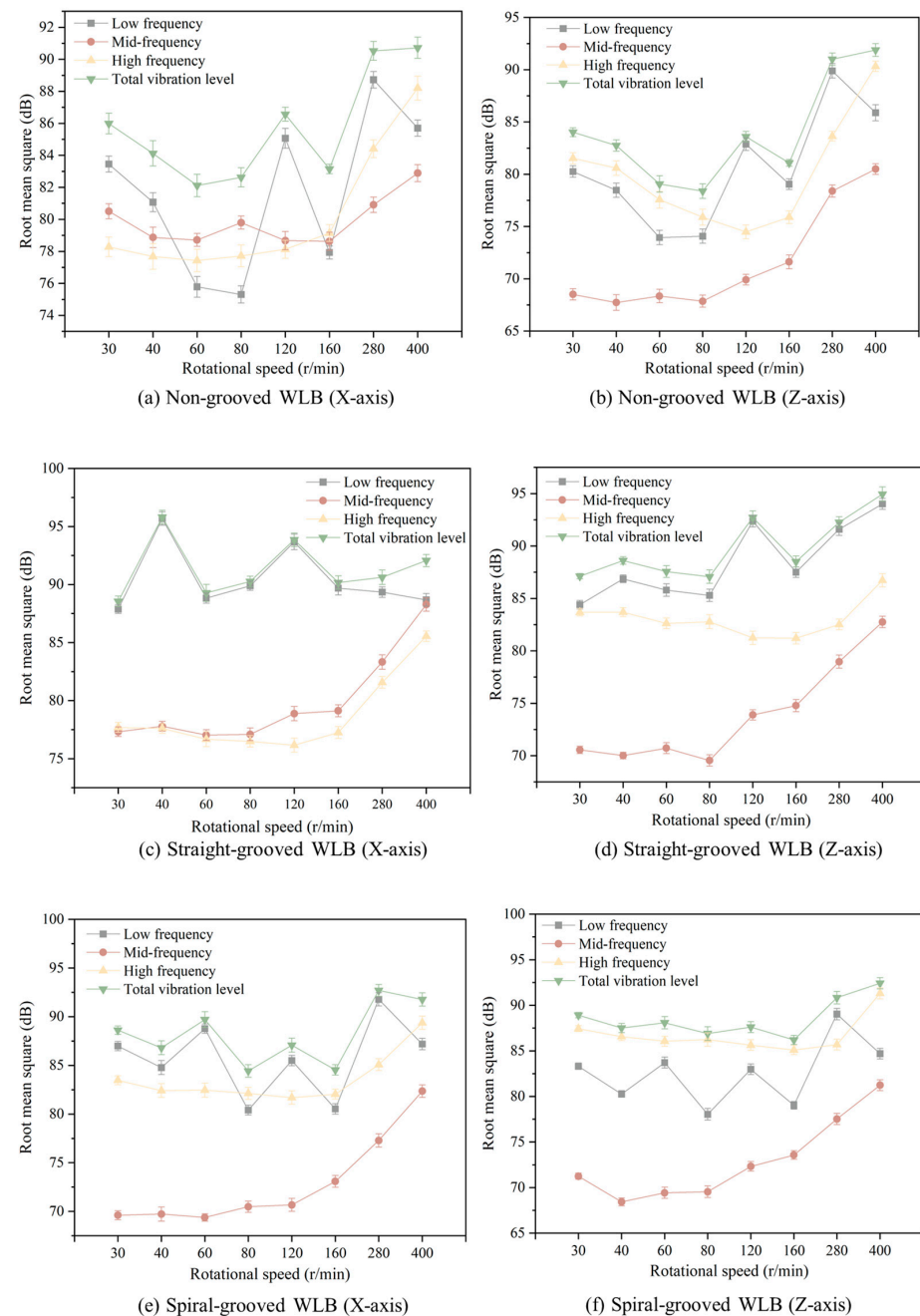


Figure 12. Vibration characteristics of water-lubricated bearings at 0.8 MPa as a function of rotational speed: (a) non-grooved WLB in the X-axis; (b) non-grooved WLB in the Z-axis; (c) straight-grooved WLB in the X-axis; (d) straight-grooved WLB in the Z-axis; (e) spiral-grooved WLB in the X-axis; (f) spiral-grooved WLB in the Z-axis.

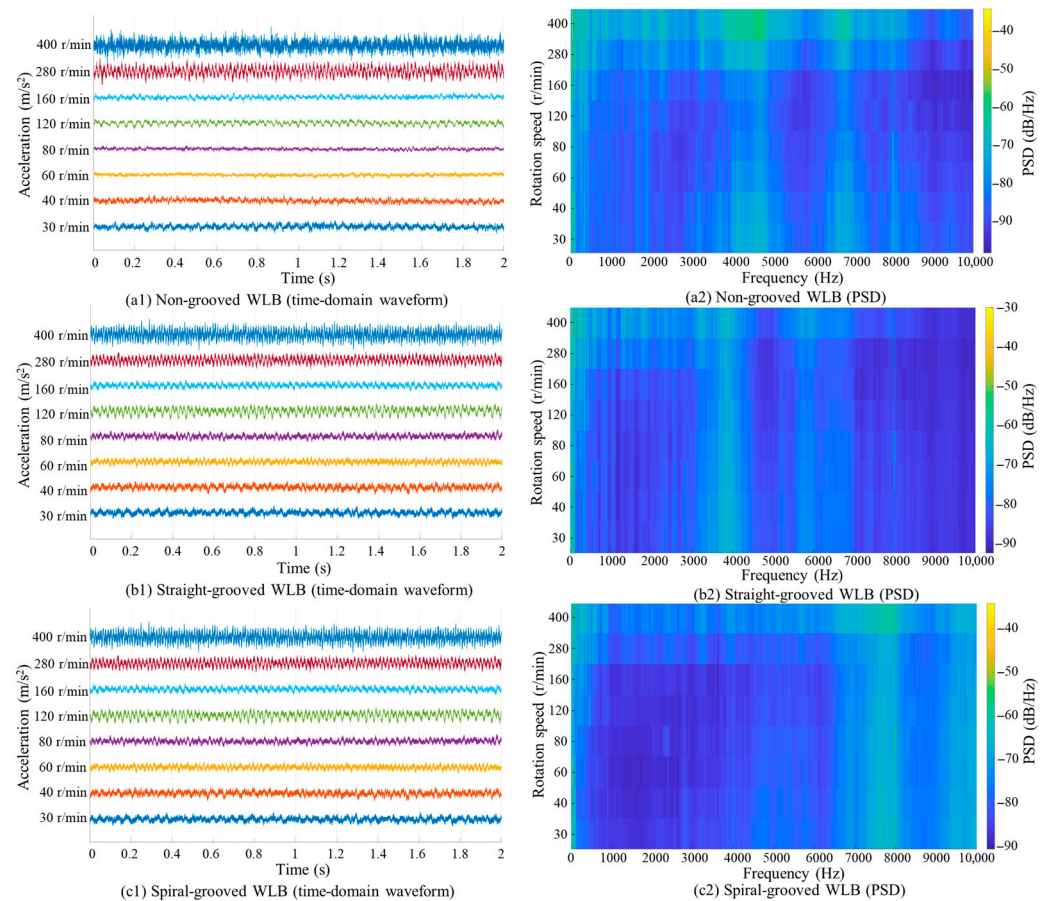


Figure 13. Time-domain waveforms and PSD of WLB signals at different rotational speeds for various groove configurations: (a1,a2) non-grooved WLB; (b1,b2) straight-grooved WLB; (c1,c2) spiral-grooved WLB.

Time-domain waveforms and PSD maps indicate that all three bearing types exhibit a trend of broadband energy elevation and waveform thickening as rotational speed increases. However, the groove structure significantly alters the intensity and distribution of high-frequency random components. Among them, the spiral-grooved WLB demonstrates a stronger broadband background and more pronounced high-frequency energy enhancement zones at most speeds. Zoomed low-frequency PSDs (0–50 Hz) at representative speeds (160 and 280 r/min) are further presented to examine the speed-related components in Figure 14a–c. The rotational frequency and its harmonics define the characteristic frequencies of the rotating system. However, the peaks around 1X and 2X are not consistently dominant, and the spectra are characterized by a pronounced broadband background with multiple local maxima. As shown in Figure 14d, to quantify the contribution of broadband high-frequency contents, the high-frequency band energy ratio E_{HF} by E_{0-50} is defined as

$$E_{HF} \text{ by } E_{0-50} = \frac{E_{HF}}{E_{0-50}} \quad (5)$$

where E_{HF} represents the integrated power in the 1200–10,000 Hz band, E_{0-50} denotes the integrated power in the 0–50 Hz band.

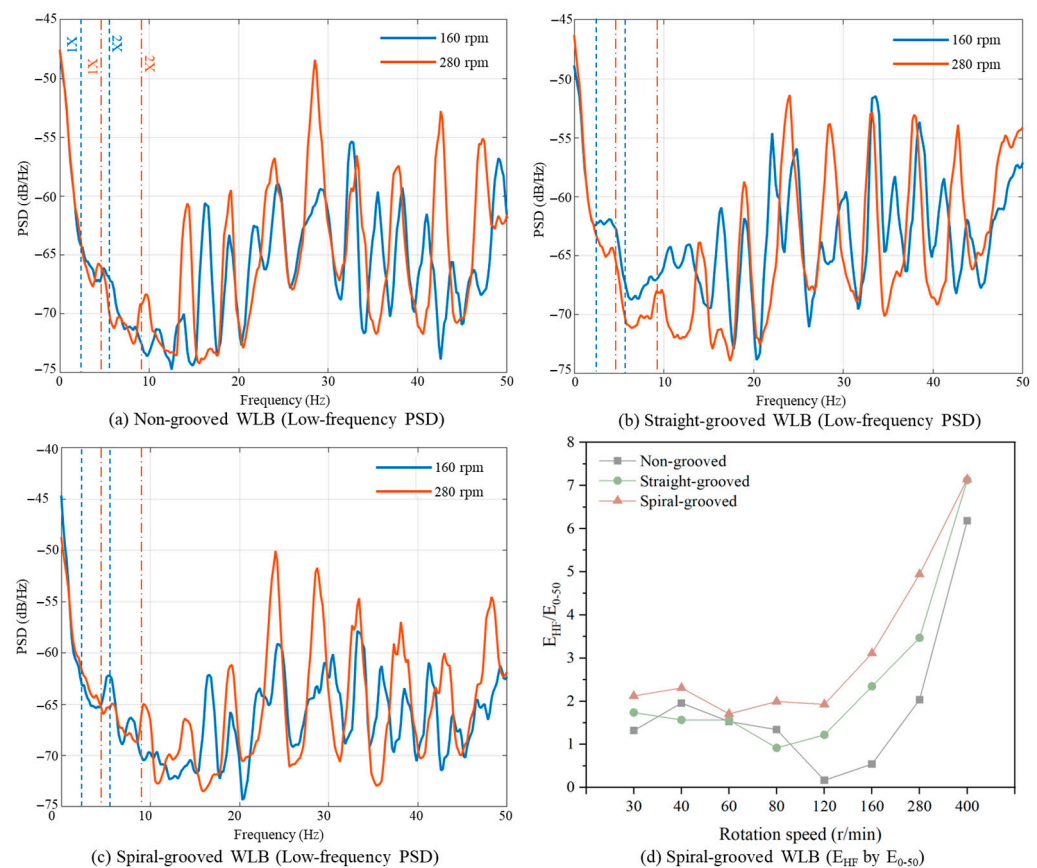


Figure 14. Low-frequency spectral characteristics of WLBs with different groove configurations at rotational speeds of 160 r/min and 280 r/min: (a) non-grooved WLB; (b) straight-grooved WLB; (c) spiral-grooved WLB; (d) evolution of the indicator E_{HF}/E_{0-50} with rotational speed for WLBs. Dashed lines mark the synchronous components (e.g., $1\times$ and its harmonics) in the low-frequency band.

This ratio characterizes the relative strength of high-frequency broadband components compared to low-frequency synchronous components. The ratio increases with speed, indicating an enhanced relative contribution of high-frequency broadband components at higher speeds. It also differs across bearing groove types, indicating that groove structures alter the proportion of high-frequency vibration components within overall vibration.

3.4. Wear Analysis

After the tests, the bearings were disassembled for morphological observation, and the wear measurements were conducted using the method described in Section 2.3. The wear properties of the bearings were evaluated by measuring the wear at several axial locations of the bearing. The wear amounts at different positions are plotted in Figure 15, and the corresponding post-test surface morphologies are shown in Figure 16. From the morphologies, the load-carrying region is located in the lower half of the bushing, where the wear is also most pronounced. As shown in Figure 16a, the non-grooved bearing exhibits distinct circumferential wear marks and stepped edge wear at the aft end, indicating that this end likely experienced mixed-lubrication conditions for a prolonged period under the tested operating history. As shown in Figure 16b, the wear on the straight-grooved bearing is primarily concentrated on the lower three ribs, predominantly distributed axially. As shown in Figure 16c, the spiral-grooved bearing displays a more uniform overall wear pattern, with only slight wear visible along the spiral direction. Compared with the other two bearings, the spiral-grooved bearing has a markedly smaller overall worn area. The

wear amount results are consistent with the observed morphologies. Among the three designs, the non-grooved bearing experiences the most severe wear, particularly at the aft end, where the maximum wear depth reaches 0.092 mm.

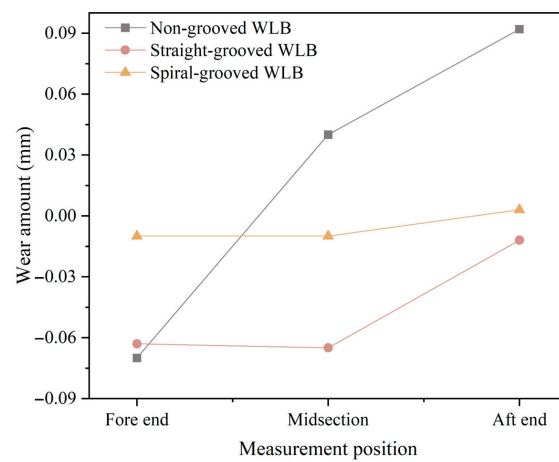


Figure 15. Wear amount at different positions of water-lubricated bearings with different groove structures.

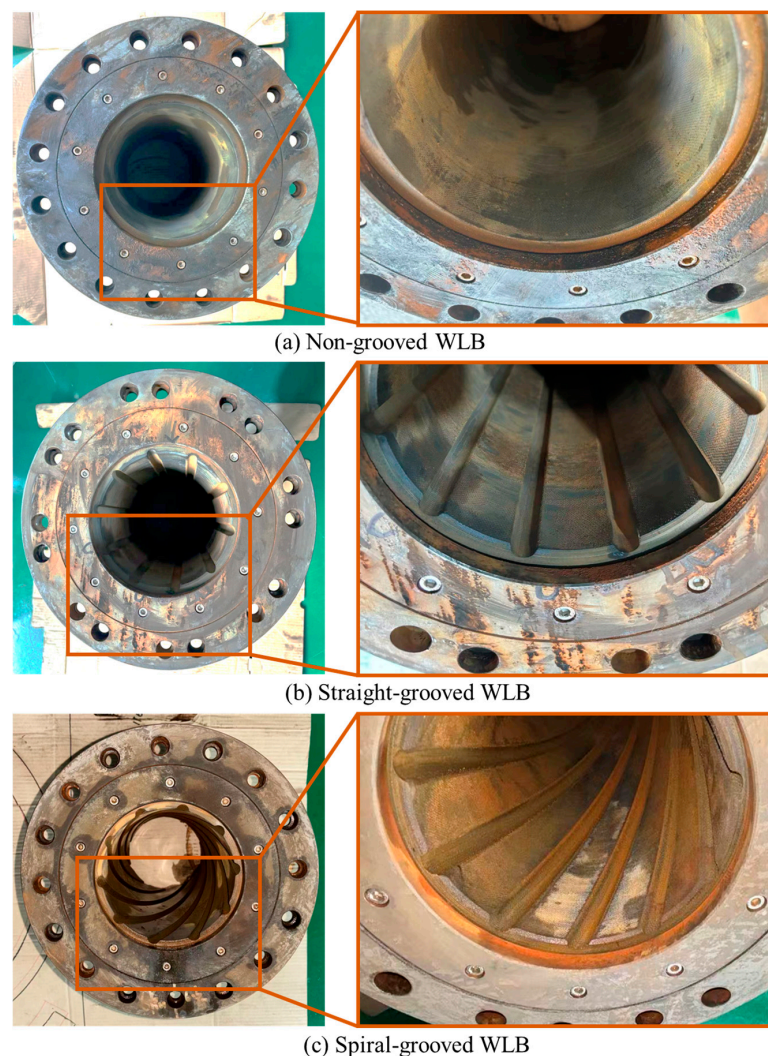


Figure 16. Wear morphologies of WLBs with different groove structures after testing: (a) non-grooved WLB; (b) straight-grooved WLB; (c) spiral-grooved WLB.

The above wear observations, together with the frequency-resolved vibration results, support the lubrication-vibration coupling mechanism discussed in Section 3.3. As schematically illustrated in Figure 17a, when the water film is not fully stabilized (mixed-lubrication state), groove-edge effects can promote the redistribution of vibration energy toward mid- and high-frequency bands, as summarized by the qualitative band contributions in Figure 17b for the grooved designs. As the water film becomes more continuous and stable, non-grooved bearings tend to exhibit weaker high-frequency contributions. However, the absence of dedicated flow channels (Figure 17c, non-grooved WLB) may increase the risk of lubrication deficiency and potentially constrain coolant renewal and convective heat dissipation, consistent with the observed non-uniform wear distribution. Straight-grooved bearings facilitate water supply and debris evacuation via axial through-flow and enhanced exchange at groove openings (Figure 17c, straight-grooved WLB), yet the segmented wedge and intensified leakage/through-flow may enhance groove-shoulder shear and flow instability, contributing to additional mid- to high-frequency components and viscous dissipation under sustained operation. By contrast, under heavy loads and low speeds, spiral grooves may provide moderate pumping and lateral debris guidance (Figure 17c, spiral-grooved WLB), helping sustain lubrication and coolant exchange without excessive pressure relief. Overall, within the investigated operating envelope and boundary conditions, the spiral-grooved bearing shows a more balanced frequency-band contribution and a more uniform wear pattern, suggesting a practical trade-off among lubrication behavior, vibration response, and wear resistance.

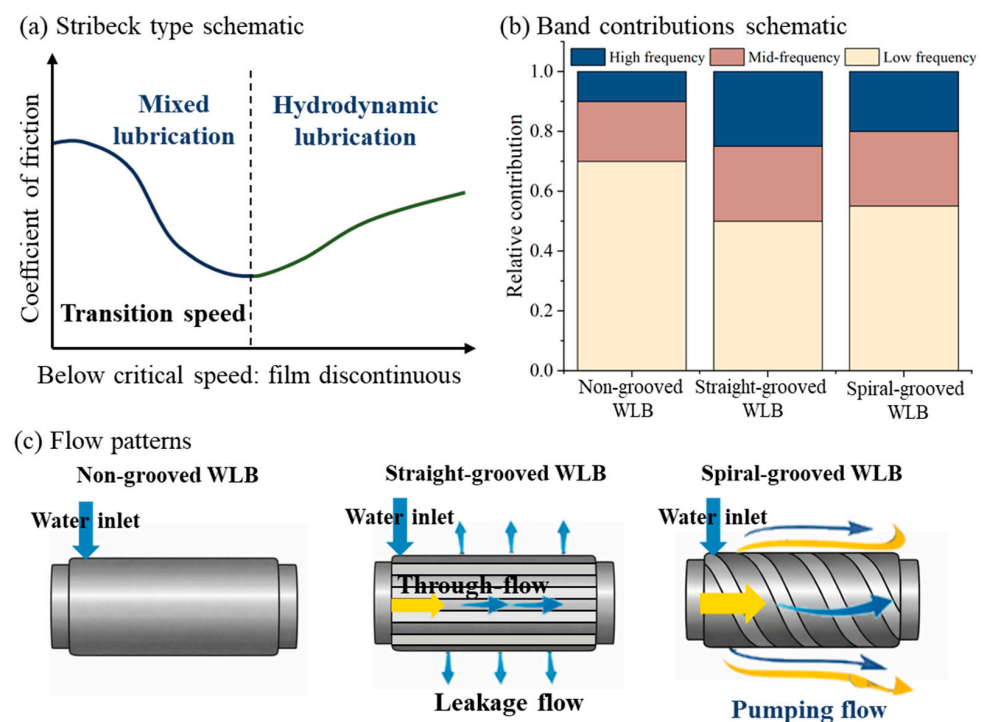


Figure 17. Schematic (qualitative) summary of lubrication–vibration coupling mechanisms for three groove configurations. (a) Stribeck-type transition and transition speed for water-film stabilization; (b) Qualitative summary of the relative contributions of low-/mid-/high-frequency bands inferred from the measured TVLs under the investigated operating conditions (particularly low-speed, heavy-load conditions); (c) Characteristic flow supply/evacuation paths for non-grooved, straight-grooved, and spiral-grooved designs (schematic). Panels (b,c) are qualitative schematics intended to summarize the observed trends rather than quantitative results.

4. Conclusions

To satisfy the stringent operational requirements of WLBs in underwater equipment, this study experimentally investigated the lubrication, vibration and wear characteristics of three-layer composite WLBs with three different groove configurations: a non-grooved bushing, a fully straight-grooved bushing and a fully spiral-grooved bushing. The main conclusions are as follows:

- (1) Specific pressure and water flow rate jointly govern the transition from boundary/mixed lubrication to hydrodynamic lubrication. Higher specific pressure and larger flow rates both shift the transition speed markedly towards higher values. Grooved bearings, especially the straight-grooved design, exhibit the largest increase in transition speed and higher COF in the hydrodynamic regime due to enhanced leakage and segmentation of the hydrodynamic wedge, whereas the non-grooved bearing maintains the lowest COF once a continuous water film is established.
- (2) Under low-speed and heavy-load conditions, the TVL of bearings increases with rising specific pressure, and the vibration component measured along the loading direction exhibits pronounced sensitivity to load-induced variations under the present test configuration. This suggests that vibration monitoring aligned with the dominant loading direction can be a practical strategy for condition monitoring in WLBs.
- (3) Groove structures can amplify vibration not only in the low-frequency band under mixed lubrication but also in the high-frequency band as operating speed increases. The straight-grooved design shows the most pronounced high-frequency growth, whereas the spiral-grooved design exhibits a more moderate increase across frequency bands.
- (4) Among the tested designs, the spiral-grooved bearing exhibits a more uniform wear pattern and a smaller worn area, indicating a more robust trade-off among lubrication stability, vibration control, and wear resistance under the investigated conditions.

Future work will build upon this foundation by integrating frequency-resolved vibration characteristics with wear indicators to develop a digital twin and condition monitoring framework for WLBs, enabling real-time health assessment and life prediction.

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