Tribological behavior of grooves textured thrust cylindrical roller bearings under dry wear

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Abstract
To prolong the serve life of roller element bearings (REB) and improve the reliability of mechanical system, the tribological behavior of the “washers-cage-rollers” system of grooves textured thrust cylindrical roller bearings (TCRB, 81107TN) under dry wear was researched. The pattern parameters include: width of grooves (WOG, 50, 100, and 150 μm), depth of grooves (DPOG, 7, 11, and 15 μm), as well as groove deflection angle (GDA, 0°, 45°, 90°, and 135°). The influence mechanism of grooves on the tribological properties of REBs is discussed. The results show that: As the GDA is 45°, the coefficient of friction (COF) and mass loss of bearing is the lowest among four angles. The average COFs of grooves textured bearings are much higher than that of smooth one, and their mass losses are all lower than that of smooth bearing. The influence of pattern parameters on the surface stresses of grooves textured bearings is weak. In this work, when the WOG is 50 μm and the DPOG is 7 μm, the wear loss of grooves textured bearing is the lowest, reduced by up to 75.6%. This work can provide a valuable reference for the raceway design and optimization of REBs.

Keywords
Rolling element bearing, friction and wear, surface texture, groove, dry wear

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Introduction
Rolling element bearings (REBs) are commonly used to support parts, bear large axial or radial loads, and limit the movements along the axial direction. REBs may be cheap, but its failure will be costly. As one of the main reasons of bearing failure, the friction and wear between moving parts (e.g. washers, rollers, and cage) of REBs, usually caused by contamination, poor lubrication, wrong fits, misalignments, etc., is undesirable and also inevitable, resulting in huge energy waste and material losses. To reduce the energy consumption of mechanical system and meet the increasingly stringent emission requirements, it is of great significance to reduce such waste and losses, postpone the transition
point between mild wear and severe wear of REBs, and prolong their mild wear periods.\(^1\)

The tribological performance of sliding/rolling tribopair depends on the surface roughness, surface hardness, contact type (e.g. point-to-surface, line-to-surface, or surface-to-surface), lubrication regime (e.g. dry wear, starved lubrication, boundary lubrication, mixed lubrication, and elasto-hydrodynamic lubrication), as well as the properties of materials.\(^2,3\) Among all above parameters, only surface characteristics (e.g. roughness, hardness, and topography) can be customized according to individual requirements or specific condition, partially or entirely, to provide us a feasible way to control the friction and wear behavior of contact surfaces.

Surface modification technologies are put forward in this case, including physical/chemical vapor deposition (P/CVD)\(^4,5\) laser remelting (LR)\(^6\) surface texture (ST)\(^7-11\) and so on. Thereinto, ST has been proved to be a cost-effective way to change the friction and wear performance of mechanical components through creating periodically distributed macro-/micro units, for example, pillar, pit, groove, and grid, in a geometry or pattern on the target surfaces.\(^12-18\) Several issues of the wear behavior of textured bearings have been investigated: lubrication, fatigue denting, abrasive wear, spalling, optimization, and so on. Rosenkranz et al.\(^19\) fabricated a dot-like pattern (periodicity of 6.5 \(\mu\)m and depth of 1 \(\mu\)m) on the washer surfaces of thrust cylindrical roller bearings (TCRBs, 81212, SAE 52100), and got an 83\% reduction in the wear loss of the laser-patterned bearing compared to that of bearings with a ground surface finish. Wang et al.\(^20\) set the maximization of load carrying capacity (LCC) of oil film as the optimization target, and developed a general parametric model of the groove bottom profile (including inner structure and depth profile) of thrust bearings to obtain the global-optimum profile of the groove texture bottom. Kumar et al.\(^21\) investigated the influence of different cross-sectional shapes of micro-dimples on the performance of thrust pad bearing, and optimized the dimple shapes from the viewpoint of LCC too.

Furthermore, when the motion and the texturing orientation are at different angles, there is also significant difference of tribological properties. Shi et al.\(^22\) proposed a new calculation method to estimate the relative fatigue life of high-speed and heavy-load ball bearing with surface texture, and found that the relative fatigue life increase with the increase of transverse texture, and decrease with the increasing longitudinal texture. Vladescu et al.\(^23\) assessed the ability of pockets of varying shapes and orientations to reduce the frictional losses of convergent-divergent bearing under a range of lubricant viscosities (from boundary lubrication, mixed lubrication to full film lubrication) and applied normal loads, and found that surface texturing is beneficial in the boundary and mixed regimes. Touche et al.\(^24\) researched the influence of the groove topography (top width and their orientation) on the friction in elasto-hydrodynamic lubrication (EHL) and mixed lubrication regimes, and found that the transverse grooves (perpendicular to the entrainment direction) enable the formation of local dimples in mixed regime, which locally increase the lubrication efficiency. Lu et al.\(^25\) evaluated the influence of anisotropically shaped textures on the behavior of sliding friction and sensitivity to sliding direction, and found that the real contact length variation rate is a major factor controlling the local friction response. Wang et al.\(^26\) investigated the influence of the grooved surface on the tribological behavior of railway brake systems, and found 45\(^\circ\) texturing surface shows a better ability in entrapping wear debris than that of 90\(^\circ\) texturing surface, indicating a greater potential in reducing the squeal instability of the friction system.

As a kind of REBs, TCRBs are commonly used in rotating instruments, for example, impact hammer, electric screwdriver, and torque wrench, to bear large axial impact load and limit the movements of parts along the axial direction. Due to their separability and convenience in texture preparation, TCRBs were also used to research the influence of surface texturing on the tribological performance of REBs.\(^19\) As shown in Figure 1, if the linear velocity of the center of one roller is set as VR, the linear velocity of its inner end, VR\(_a\), will be lower than VR, and the linear velocity of its outer end, VR\(_b\), will be higher than VR, although they have the same angle velocity, \(\omega\). This means that there are obvious sliding phenomena existed at both ends of rollers. Due to its complex rolling/sliding feature, the influence mechanism of grooves on the wear behavior of TCRBs is not clear yet.\(^27-29\)

When the fighter is doing tumbling and other violent movements, the aero-bearings (REBs) may be in the state of no lubrication or even dry friction. Therefore,
the wear resistance of aero-bearing, especially the friction and wear performance without lubrication, is the basis to ensure the stability and reliability of aero-engine. Dry wear, running with no lubrication oil, is the worst work condition for all bearings. The friction and wear properties of textured REBs under dry wear can best reflect the wear resistance and friction-reduction ability of the texture itself, which can also provide a reference for the research of tribological performance in other lubrication regimes. Therefore, based on previous works, a laser marking system (PL100-30W, China) was used to process grooves textured 81107TN bearings with different pattern parameters. A MMX-1A wear test rig was used to obtain their COFs under dry wear. An electronic analytical balance (EX225D, Ohaus) with a precision of 0.1 mg (0.01 mg readability) was used to measure their wear losses. A 3D surface profilometer (VK-1050, Keyence, Japan) was used to characterize their worn surfaces. The wear performance and the influence mechanism of grooves on the tribological properties of TCRBs were compared and discussed. This work would provide a valuable reference for the raceway design and optimization of REBs.

Materials and modeling

The sketch and dimensions of 81107TN bearing are shown in Figure 2. Among all parts in the figure, the shaft washer, housing washer, and rollers are fabricated by GCr15, and their harnesses are about HRC 60 ± 1. The cage is made of nylon, PA66. The basic performance parameters of 81107TN bearing are listed in Table 1.

The pattern parameters include: width of grooves (WOG, 50, 100, and 150 μm), depth of grooves (DPOG, 7, 11, and 15 μm), as well as groove deflection angle (GDA, 0°, 45°, 90°, and 135°, see Figure 3). The number of grooves is 64, that is, the angle between two adjacent grooves is 5.625°. Obviously, WOG and GDA are the controlling parameters to determine the final shape and area density of textured surface. The tribological performance of TCRBs with different GDAs (four groups, coded from T01 to T04, see Table 2) was compared first. Choosing the GDA with the best anti-wear resistance, the influence of different WOGs and DPOGs on the tribological properties of TCRBs (G05 and eight new groups, marked as G01-G04, G06-G09) was researched. A group of smooth TCRBs is also introduced as a reference, and coded as G10. To better compare and analyze the results, T02 and G05 are two different code numbers of the same group. Each group includes three bearings and was tested three times. Thus, the total number of consumed 81107TN bearings is $13 \times 3 = 39$.

Prior to wear tests, all bearings were pre-treated as the following steps: First, remove the antirust grease of bearings, wash and dry them; Second, process different groove patterns on the raceways of shaft washers (T01, T03-T04, G01-G09) only; Third, polish the textured shaft washers with SiC papers (from #400 to #2000 grades).
textured bearings is 1.1–1.3 μm, and that of smooth group is 0.693 μm; Finally, clean the polished washers in a ultrasonic cleaner (VGT-1620QTD, China) for 15 min, and dry them by a hot-air blower. The parameters of laser marking system are listed as follows: laser wave length of 1064 μm, laser power of 30 W, scanning speed of 100 mm/s, and frequency of 72 kHz.

The tribological properties of grooves textured TCRBs were tested using a MMW-1A wear test rig (Huaxing, Jinan, China) with a customized 81107 tribo-pair. As shown in Figure 4, the tribo-pair consists of: rotating header, upper chucking, oil baffler, 81107 shaft washer, 81107 cage/rollers, 81107 housing washer, lower chucking, pressure cushion block, dynamometer disk, 51108 thrust bearing, bumper rod, and wire fixing bolts for friction force measurement. Through repeated trials, the parameters of the rig were set as the following: vertical loading force, 2950 ± 100 N, about 30% of the ultimate fatigue load of 81107TN bearing (see Table 1); rotating speed, 250 rpm; test duration, 18,000 s. All tests were carried out at room temperature (20°C) and 30% humidity.

**Experimental and simulation results**

Note that: The COF curve of each group in this section is the mean value curve of three tests. The mass loss of each group is the difference between averages of nine measurements of three bearings before and after tests.

**Tribological behavior of grooves textured bearings with different GDAs**

Figure 5 shows the COF curves of grooves textured bearings with different GDAs (0°, 45°, 90°, and 135°) under dry wear, when the WOG is 100 μm and the DPOG is 11 μm. The COF curve of smooth bearing and its average coefficient of friction (ACOF) line are also shown in the figure as references. The label “T01: 100-11-0” means “Group code: WOG-DPOG-GDA.” All ACOFs and mass losses of bearings (T01-T04 and G10) under dry wear are listed in Table 3. Obviously, the ACOFs of all textured bearings are higher than that of smooth one. While their wear losses are all lower than that of smooth bearing. As shown in Table 3, with the increase of the GDA, the ACOFs and wear losses of grooves textured bearings decrease first and then get higher, that is, there is an apparent inflection point of the friction-reducing effect of the GDA, that is, 45°. Among four groups, the ACOF and mass loss of T02 is the lowest. Its mean-variance of wear loss is also the minimum. This is the reason of the GDA selection of G01–G09.

Figure 6 compares the worn surfaces of the shaft washers of bearings with different GDAs (T01–T04, G10) under dry wear. As shown in the figure, there are serious wear marks and large high temperature region on the worn surface of smooth bearing (G10). The wears of T01–T03 are apparently slighter than those of T04 and G10, and there is no serious furrow on their raceways. The wear of T04 is more serious than those of T01–G03, but slighter than that of G10. This is consistent with the results in Lu et al.25.

| Sample no. | GDA (°) | WOG (μm) | DPOG (μm) | Area Density (%) |
|------------|---------|----------|-----------|-----------------|
| T01        | 0       | 100      | 11        | 4.69            |
| T02/G05    | 45      | 100      | 11        | 6.15            |
| T03        | 90      | 100      | 11        | 7.06            |
| T04        | 135     | 100      | 11        | 6.15            |
| G01        | 45      | 50       | 7         | 2.34            |
| G02        | 45      | 50       | 11        | 2.34            |
| G03        | 45      | 50       | 15        | 2.34            |
| G04        | 45      | 100      | 7         | 6.15            |
| G06        | 45      | 100      | 15        | 6.15            |
| G07        | 45      | 150      | 7         | 7.03            |
| G08        | 45      | 150      | 11        | 7.03            |
| G09        | 45      | 150      | 15        | 7.03            |
| G10        | —       | —        | —         | —               |

**Figure 4.** Sketch of the customized 81107 bearing tribo-pair.
1. Rotating header; 2. Upper chucking; 3. Oil baffler; 4. 81107 shaft washer; 5. 81107 cage/rollers; 6. 81107 housing washer; 7. Lower chucking; 8. Pressure cushion block; 9. Dynamometer disk; 10. 51108 thrust bearing; 11. Bumper rod; 12. Wire fixing bolts for friction force measurement.
Tribological behavior of grooves textured bearings

Coefficients of friction. To better compare the difference of COF curves, the ACOF line and COF curve of smooth bearing (G10) are both added as references in subsequent figures too. Figure 7 shows the COF curves of bearings. Thereinto, Figure 7(a), (c) and (e) show the COF variations of bearings with fixed WOG. Figure 7(b), (d) and (f) show the COF variations of bearings with fixed DPOG.

As shown in the figure, the COF curves of bearings under dry wear can be divided into two stages: running-in stage and fluctuating-running stage, whether the textured or smooth. Apparently, the COFs of all grooves textured bearings are higher than that of smooth group in this condition.

Wear losses and worn surfaces. Figure 8 compares the mass losses of the shaft washers and the ACOFs of bearings (G01–G10) under dry wear. As shown in the figure, the mass losses of grooves textured bearings are all lower than that of smooth one (5.75 mg). Thereinto, the loss of G01 is the lowest among all bearings and only 1.40 mg. The loss of G03 is also quite lower than those of other textured bearings, only 2.32 mg. The losses of G02, G05, G07, and G09 are close, and about two-thirds of that of smooth group. The losses of G04, G06, and G08 are quite high and just slightly lower than that of G10. Figure 9 shows the worn surfaces of shaft washers of G01–G10 under dry wear. Evidently, there are apparent high-temperature marks on the raceways of all bearings. Based on their worn surfaces, the wear pattern of bearings under dry wear is mainly abrasive wear combined with fatigue pitting. Compared with other bearings, the fatigue pitting of G02, G04, G06, G08, and G10 is much more serious.

Simulations of the grooves textured “washers-cage-rollers” system

A 3D model of 81107TN bearing was established (see Figure 10) and static simulations were carried out through ANSYS Workbench. The applied vertical load was the same as that in wear tests, that is, 2950 N, and the lower surface of the housing washer was set as a fixed support. To better compare and discuss the difference of equivalent stresses between grooves textured and smooth bearings, a path, located 15 μm below the raceway of the shaft washer, is introduced. Its direction is shown in Figure 10(b), from the inside to the outside. The node number of grooves textured bearings is about
480,000–6,200,000 and the number of element is about 270,000–310,000. They are quite higher than those of smooth bearing, whose node number is about 75,000 and its element number is only about 21,000.

Figure 11 shows the equivalent stress contours around the paths of bearings (T01–T04) with different GDAs, when the WOG is 100 μm and the DPOG is 11 μm. Figure 12 shows their equivalent stresses along the paths. The stress of smooth bearing is also added as a reference. As shown in the figure, the surface stresses of all grooves textured bearings are higher than that of

Figure 7. COF curves of bearings under dry wear: (a) WOG = 50 μm, (b) DPOG = 7 μm, (c) WOG = 100 μm, (d) DPOG = 11 μm, (e) WOG = 150 μm, and (f) DPOG = 15 μm.

Figure 8. Wear losses of shaft washers and ACOFs of bearing (G01–G10) under dry wear.
smooth one. Among four groups, the average stress of T03 (90°/C176) is the highest and about 7.65 MPa. This is because its raceway is divided into separate rings by closed grooves. As a result, its effective contact area (ECA) is significantly reduced compared with other bearings, and there are obvious stress peaks between two adjacent rings (see Figure 11(c)). The average stress of T01 (0°) is the lowest and equals to 5.67 MPa, which is only slightly higher than that of smooth bearing (4.89 MPa). This is because the length of groove units is the shortest in this case, and its ECA is the biggest among four groups. The average stresses of T02 and T04 are almost the same, about 6.68 MPa.

Figure 13 shows the equivalent stresses along the paths of bearings (G01–G09). Thereinto, Figure 13(a), (c) and (e) show the equivalent stresses along the paths of bearings with fixed WOG. Figure 13(b), (d) and (f) show the equivalent stresses along the paths of bearings with fixed DPOG. As shown in the figure, the equivalent stresses of the points in the contact areas between rollers and raceways are significantly higher than those of points in the non-contact regions. The stresses at the
outer sides of raceways are also a little higher than those at the inner sides, whether the textured or smooth. Compared with the stress of smooth group, the peak-stresses of the two sides of raceways of all grooves textured bearings are almost the same and even get smaller. The maximum stresses of G01–G09 in the contact area between rollers and raceway are all lower than 15 MPa, only about 1.5–2 times higher than that of smooth bearing, with two other stress peaks at the positions where grooves passed through.

By comparing the curves in Figure 13(a), (c) and (e), the following results can be obtained: when the WOG is fixed, the surface stress becomes higher with the
increase of DPOG. But as the DPOG is too big, the stress may decrease instead, like G09. Specifically, when the WOG is 50 and 100 μm, the stress curves mixes together and are hard to distinguish, indicating the effect of the DPOG on the stresses of grooves textured surfaces is weak and un-apparent. As the WOG is 150 μm, the condition is better and three curves can be distinguished. Likewise, when the DPOG is fixed, 7, 11, or 15 μm, as the WOG is 100 μm, the stresses are relatively higher than the values while the WOG is 50 and 150 μm (see Figure 13(b), (d) and (f)). The stress curves of textured bearings all mix together and are very difficult to distinguish, indicating the weak influence of the WOG on the surface stresses in this case.

This is because the raceways of textured bearings are divided into different regions by groove units, which act as the shallow U-shaped thin-wall embedded reinforcements (see Figure 14), due to the “local quenching effect” during laser marking process.\(^5,33\) When the WOG is fixed and small, the integrity of raceways is gradually weakened with the increase of DPOG, but the strengthening effect of U-shaped thin-wall embedment is gradually enhanced according to classical theoretical mechanics. Correspondingly, there are obvious and serious surface stress concentrations along the edges of grooves under the rolling of rollers in this condition. However, as the WOG and DPOG are too big, like G09, the integrity of raceways is further weakened, and the strengthening effect of U-shaped thin-wall embedment is also reduced for its lower depth-width ratio. As a result, the stresses along the edges of grooves are reduced and the average surface stress gets smaller.

Taken together, the influence of pattern parameters on the surface stresses of grooves textured bearings is weak, whether the WOG or DPOG. That is, the groove units cannot significantly increase the contact stresses of textured bearings. As the WOG is 100 μm, or the DPOG is 11 μm, the surface stress of bearing is relatively higher than other groups.

**Discussion**

The tribological data of bearings (G01–G10) under dry wear are listed in Table 4. Generally, as shown in the table, the ACOFs of grooves textured bearings are much higher than that of smooth one, and their mass losses are all lower than that of G10. The loss of G01 is 1.40 mg, the lowest among all bearings in this work.

**Effect of pattern parameters of the grooves on the tribological behavior**

**Groove deflection angle.** To better analyze the influence of different GDAs on the tribological behavior of 81107TN bearings, a maximum volume coefficient of groove units (MVCG), $\Phi$, is introduced. Its physical meaning is the total volume of all groove units of textured shaft washer and defined as follows (when the GDA $\neq 90^\circ$):

$$\Phi = 64 \times \frac{(52/2 - 35/2) \cdot \text{WOG} \cdot \text{DPOG}}{\cos(\text{GDA})} \times 10^{-5} (1)$$

Based on the equation (1), the MVCG of T01–T02, T04 is 5.984, 8.464, $-8.464$, respectively. The MVCGs of G01–G09 are listed in Table 4. The MVCG of T03 can be calculated as:

$$\Phi = 2\pi \cdot \text{WOR} \cdot \text{WOG} \cdot \text{DPOG} \cdot \text{NG} \times 10^{-5} (2)$$

Wherein, WOR is the rotating radius of rollers, mm; and NG is the number of grooves of textured shaft washer. In this work, as shown in Figure 3, the WOR is 22.25 mm and the NG is 6, so the MVCG of T03 is 9.222.

When the GDA is $0^\circ$ and $90^\circ$, as shown in Table 3, the ACOF of T01 (0.05514) is slightly lower than that of T03 (0.05534). This is because when the GDA is $0^\circ$, its MVCG is only 5.984, which means that its capacity to collect and hold wear debris is limited. A large amount of nylon powder will remain on the raceway when the grooves are filled, although the debris can be thrown out partially under the action of centrifugal force. In this case, the thickness of nylon film is thick and uneven. As the GDA is $90^\circ$, six grooves are all closed. Its MVCG is 9.222, a relatively high capacity to collect debris, and the amount of nylon powder left on the raceway is fewer than that of T01 in its initial period, but increase very soon, for the nylon debris in the system cannot be thrown out effectively in this condition.

![Cross section of groove unit of G09 when wear test is finished under starved lubrication.](image14.png)

Figure 14.
condition. As a result, compared with that of T01 (see Figure 15), the nylon film of T03 is much thicker and more uneven. Due to the high COF between nylon and steel as well as the well mechanical properties of nylon, the coefficient of T01 is slightly lower than that of T03, while its wear loss (4.52 mg) is a little higher than the loss of T03 (4.17 mg). In addition, there are obvious “arch-shaped extrusion marks” on the raceways of T01 and T03, which also should be attributed to the large amount of nylon powder left in the system. Based on their worn surfaces, the wear pattern of T01 is abrasive wear and fatigue pitting, and that of T03 is serious abrasive wear and fatigue pitting.

When the GDA is 135°, the COF and mass loss of T02 is the lowest, whose wear pattern is slight abrasive wear and slight fatigue pitting. This is because its MVCG is 8.464, the powder left on the raceway is much fewer than other GDAs for the nylon powder can be thrown out effectively. As a result, the nylon film is the thinnest and the most even among four groups, which is the reason for the lowest COF and wear loss of T02.

**Table 4. Tribological data of bearings (G01–G10) under dry wear and their MVCGs.**

| Group No. | WOG (μm) | DPOG (μm) | ACOF  | Wear loss (mg) | MVCG  |
|-----------|-----------|-----------|-------|----------------|-------|
| G01       | 50        | 7         | 0.06493 | 1.40           | 2.69307 |
| G02       | 50        | 11        | 0.05799 | 3.73           | 4.23197 |
| G03       | 50        | 15        | 0.05381 | 2.32           | 5.77086 |
| G04       | 100       | 7         | 0.04979 | 5.07           | 5.38614 |
| G05       | 100       | 11        | 0.0502  | 3.67           | 8.46393 |
| G06       | 100       | 15        | 0.05442 | 5.11           | 11.54173 |
| G07       | 150       | 7         | 0.06816 | 3.31           | 8.07921 |
| G08       | 150       | 11        | 0.05635 | 4.80           | 12.6959 |
| G09       | 150       | 15        | 0.05871 | 3.68           | 17.31259 |
| G10       | —         | —         | 0.04733 | 5.75           | 2.69307 |

**Figure 15. Nylon film left on the raceway of T01 when wear test is finished under dry wear.**

The GDA is 45°, the COF and mass loss of T02 is the lowest, whose wear pattern is slight abrasive wear and slight fatigue pitting. This is because its MVCG is 8.464, the powder left on the raceway is much fewer than other GDAs for the nylon powder can be thrown out effectively. As a result, the nylon film is the thinnest and the most even among four groups, which is the reason for the lowest COF and wear loss of T02.

**Width of grooves and depth of grooves.** When bearings run under dry wear, as the WOG increases from 50 to 100 μm, the ACOFs of G01–G06 decrease first and then get higher (see Figure 8). Among six groups, the ACOFs of G04 and G05 are relatively lower than those of other groups (G01–G03, G06). Specifically, when the WOG is 50 μm, as shown in Figure 7(a), three curves are relatively high and easy to be distinguished, indicating the influence of DPOG on the COF variation is significant. As the WOG is 100 μm, the curves in Figure 7(c) mix together and cannot be distinguished, which means the influence of the DPOG is weak. While the WOG is 150 μm, three curves can be distinguished again (see Figure 7(e)). The ACOFs of bearings (G07, G08, and G09) decrease first and then get higher with the increase of DPOG, that is, there is an inflection point of the influence of DPOG on the COF. Equally, when the DPOG is 7 μm, three curves can be distinguished easily (see Figure 7(b)). As the DPOG is 11 and 15 μm, the curves mix together and are hard to distinguish (see Figure 7(d) and (f)). That is, there is a stable point of the influence of WOG on the COF variation.

From the viewpoint of mass losses, when the WOG is 50 μm, the losses of G01–G03 are quite lower than other bearings, especially for G01, which is only 1.4 mg. As the WOG is 100 and 150 μm, the mass losses are relatively high. There is an inflection point of the influence of DPOG on the wear losses too, especially when the WOG is 50 or 150 μm. Likewise, as shown in Figure 8, there are inflection points of the influence of WOG on the wear losses, whether the DPOG is 7, 11, or 15 μm.
According to their MVCGs, the wear behavior of grooves textured bearings under dry wear can be classified into the following three cases: when the MVCG is small (2.69–5.77), that is, G01–G03, with the increase of the WOG or DPOG, and the nylon powder remained on the raceway is reduced. However, due to the limited capacity of grooves to collect and hold debris, the nylon film is thick but getting thinner. As a result, the COFs of bearings become smaller although still relatively high. Owing to the well mechanical properties of nylon, the thick nylon films can effectively protect the textured raceways, which is the reason for their low wear losses in this case. The obvious COF fluctuations of G01–G03 should also be attributed to the more nylon powder left on their raceways (see Figure 7(a)). As the MVCG is moderate (5.39–11.54), that is, G04–G06, the capacity of grooves is enhanced. The nylon powder left on the raceway is reduced. As a result, the COF fluctuations of G04–G06 are quite smaller than those of G01–G03 (see Figure 7(c)). The nylon film forms slowly and their thicknesses are thinner than those of T01–T03, resulting in the relatively low COFs and high mass losses of them. Compared with the curve of smooth bearing, the reason for the evident peak-time delays of COF curves of G04 and G05 should also be attributed to the reduced nylon powder remained on the textured raceways and the slow formation of nylon films. As for G06, its peak-time is a little earlier than that of smooth bearing. This is because the MVCG reaches the transient point of the system. From this point on, the friction behavior of the bearing is very close to that of smooth bearing, especially the peak-time. Furthermore, when the WOG is 100 μm, their high surface stresses may cause the serious fatigue pitting of material, which is another reason for the high losses of them. While the MVCG is big (8.08–17.31), that is, G07–G09, the capacity of grooves to collect and hold wear debris is much higher than the amount of nylon powder generated in the system, and the metal debris and powder are more likely to be thrown out by the centrifugal force, which further induce the fast rising of system temperature. Owing to the well mechanical properties of nylon, the nylon powder melts quickly and a nylon film forms. This is the first influence mechanism of grooves on the tribological behavior of textured bearings under dry wear, whether the textured or smooth. As the WOG is 50 μm and the DPOG is 7 μm, that is, G01, its mass loss is the lowest. Compared with the smooth bearing, the loss of G01 can be reduced by about 75.6%, and the variance of three repeated tests is almost zero, indicating the friction-reducing effect of G01 is repeatable and credible.

**Influence mechanisms of grooves on the tribological properties of bearings**

Besides the influence of groove units on the friction and wear properties of rolling bearings listed in Long et al., when TCRBs are tested under dry wear, due to the direct contact between rollers and pockets of nylon cage, there is a large amount of nylon powder generated in the “washers-cage-rollers” system during wear test. Nylon powder left on the raceway increases the rolling resistance of the system and even causes the jam of rollers, which further induce the fast rising of system temperature. With the rapid increase of system temperature, the nylon powder melts quickly and a nylon film forms. This is the first influence mechanism of grooves on the tribological behavior of textured 81107TN bearings (see Figure 16). However, owing to the obvious circumferential linear velocity difference of two ends of rollers (see Figure 1), the thickness of nylon films along the radius direction is not uniform (see Figure 15). The films on the both sides of raceways are almost completely worn out, while the films on the middle of raceways are basically intact. Finally, the frictional-induced heat, nylon powder, as well as the system temperature gets to a dynamic balance. The COF of bearing begins to drop and gets stable gradually, indicating the end of its running-in stage. For the COF of nylon-steel tribo-pair is much higher than that between steel and steel, the formation of uneven nylon film is the main reason for the high COF peaks of all bearings under dry wear in their running-in stages.

On the whole, the existence of nylon film can significantly reduce the influence of pattern parameters (WOG and DPOG) on the wear properties of bearings under dry wear, whether the textured or smooth.
Conclusions could be drawn: experimental and characterization data, the following dry wear. Through the comprehensive analysis of the properties of grooves textured 81107TN bearings under a laser marking system and a MMX-1A wear test rig were used to reveal the friction-reducing and anti-wear behavior of the “washers-cage-rollers” system of REBs, To research the influence of groove units on the wear properties of grooves textured 81107TN bearings.

Finally, the groove units can significantly reduce the ECA between rollers and raceway, and further increase the contact stress of textured surface (see Figure 13). According to the classical coulomb friction formula, the friction will increase as the external load is fixed. Consequently, the COF is also bound to increase, which is another reason for the high ACOFs of grooves textured bearings under dry wear and also the third influence mechanism of grooves on the tribological behavior of textured 81107TN bearings.

Conclusions
To research the influence of groove units on the wear behavior of the “washers-cage-rollers” system of REBs, a laser marking system and a MMX-1A wear test rig were used to reveal the friction-reducing and anti-wear properties of grooves textured 81107TN bearings under dry wear. Through the comprehensive analysis of the experimental and characterization data, the following conclusions could be drawn:

(1) The average surface stresses of T01–T04 are quite small and all lower than 8 MPa, whether the GDA is 0°, 45°, 90°, or 135°. The ACOFs of grooves textured TCRBs with different GDAs are all higher than that of smooth group (0.04733), while their mass losses are all lower than that of smooth bearings (5.75 mg). Among four groups, the COF and mass loss of T02 are the lowest and equal to 0.0502 and 3.67 mg, respectively. Its wear pattern is slight abrasive wear and slight fatigue pitting. The COF and wear loss of T04 are the highest and equal to 0.05845 and 4.55 mg, respectively. Its wear pattern is serious fatigue pitting and abrasive wear. The wear pattern of T01 and T03 is abrasive wear combined with slight fatigue pitting, and there are obvious “arch-shaped extrusion marks” on their raceways.

(2) When TCRBs are tested under dry wear, their ACOFs are much higher than that of smooth one, and their mass losses are all lower than that of smooth bearing. There is an inflection point of the influence of DPOG and also a stable point of the influence of WOG on the COFs. The existence of un-even nylon film reduces the influence of pattern parameters on the wear properties, and directly affects the final tribological behavior of bearings, whether the textured or smooth. In this work, when the WOG is 50 μm and the DPOG is 7 μm, that is, G01, its wear amount is the lowest, only 1.40 mg, reduced by about 75.6% compared with the smooth bearing.

(3) The influence of pattern parameters on the surface stresses of grooves textured bearings is weak and unapparent, whether the WOG or DPOG. That is, the groove units cannot significantly increase the contact stresses of textured bearings. The stresses in the middle of textured raceways are only slightly higher than that of smooth one, with almost the same peak-stresses at the two sides of raceways. When the WOG is 100 μm, or the DPOG is 11 μm, the surface stresses of bearings are relatively higher than other groups.

(4) The MVCG is a key parameter to reflect the friction and wear performance of grooves textured bearings. The final tribological properties of grooves textured bearings under dry wear are determined by the following factors: the MVCGs of bearings, surface contact stress, nylon film characteristics, the U-shaped thin-wall strengthening of grooves, fatigue peeled of material along edges of grooves, loads, etc. According to the obtained tribological data, to obtain better wear resistance of rolling bearing under dry friction, the pattern parameters of grooves textured TCRBs are recommended as follows: the WOG, ≤ 50 μm; the DPOG, ≤ 7 μm.

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Data Availability
The original codes and data used to support the findings of this study are available from the corresponding author upon request.

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