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Analysis of Grooves Used for Bearing Lubrication Efficiency Enhancement under Multiple Parameter Coupling

Chao Wang [1], Mingkai Wang [1] and Linbo Zhu [2,*]

[1] Key Laboratory of Education Ministry for Modern Design and Rotor-Bearing System, Xi'an Jiaotong University, Xi'an 710049, China; jannik@stu.xjtu.edu.cn (C.W.); mkwang@stu.xjtu.edu.cn (M.W.)
[2] School of Chemical Engineering and Technology, Xi'an Jiaotong University, Xi'an 710049, China
* Correspondence: linbozhu@mail.xjtu.edu.cn

Abstract: Adding axial groove structures to the surface of bearing inner ring is an effective way to enhance the bearing lubrication efficiency. In this paper, the angular contact ball bearing H7006C was taken as the research object, and through visual simulations and quantitative experiments, systematic analyses and discussions were carried out to find out the relationship between groove-enhancing performance and working conditions. Firstly, simulation models of standard bearing and groove-added bearing were established. By combining the Volume of Fluid (VOF) model, the enhancing mechanism of grooves was found. Secondly, the trend of groove-enhancing performance with the change of rotation speed was studied through simulations and quantitative experiments. On this basis, through multiple parameter coupling, the effects of oil supply amount and nozzle diameter on the groove performance were discussed. The results show that the bottom oil layer is the key for grooves to achieve the lubrication efficiency enhancement, and its distribution can reflect the groove-enhancing performance. The groove width that best adapts to the change of working conditions increases with the increase of oil supply amount and nozzle diameter. To maintain the stability of bottom oil layer, the nozzle diameter should be larger than the groove width. This research is of great significance to the application of grooves in the lubrication efficiency enhancement of high-speed bearing.

Keywords: bearing lubrication enhancement; groove structure; multiple parameter coupling

Citation: Wang, C.; Wang, M.; Zhu, L. Analysis of Grooves Used for Bearing Lubrication Efficiency Enhancement under Multiple Parameter Coupling. *Lubricants* **2022**, *10*, 39. https://doi.org/10.3390/lubricants10030039

Received: 19 February 2022
Accepted: 5 March 2022
Published: 7 March 2022

1. Introduction

The complex motions and collisions of bearing assemblies can seriously affect bearing performance [1]. Collisions can generate much heat, leading to a large temperature rise. Keeping bearing well-lubricated can reduce bearing friction loss, control the temperature rise, and improve bearing performance [2]. Among the common lubrication methods, jet lubrication is widely used in high-speed working conditions for its high lubrication efficiency [3]. In jet lubrication, oil or oil-air mixture is normally injected from the nozzle mounted on the side of the bearing into the bearing cavity, then adheres to the bearing assemblies or flows into the contact area. However, high-speed and high-pressure vortices form when the bearing rotates at high speed, preventing oil from flowing into the contact area, resulting in the decrease of bearing lubrication efficiency [4,5]. In order to improve the bearing lubrication, several studies from the perspective of the optimization of oil supply parameters have been carried out. Zhang [6] has studied the influence of different oil supply amount, number of nozzles, injection speed, and rotation speed on the two-phase flow field in the bearing cavity and compared the friction torque of the bearing under the nozzles of different diameters. Taking the bearing temperature as the index, Li [7] and Zheng [8] have studied the effects of the number of nozzles, rotational speed, and nozzle diameter on the bearing lubrication efficiency under jet lubrication through experiments.

Although changing the oil supply parameters is a feasible way to improve the bearing lubrication efficiency, the oil transfer into the contact area is still restricted by the strong air flow. Therefore, the way of the optimization of the oil supply structure to improve lubrication efficiency is gaining much attention. A novel guide-type nozzle designed by Guo [9,10] can improve the utilization rate of oil for using the capillary force to reduce the influence of the air core inside the nozzle, so oil at the outlets is of higher velocity and easy to cross the air curtain inside the bearing cavity. NSK Ltd. has manufactured Spinshot II nozzle, this type of nozzle injects oil to the surface of the inner ring, which reduces the influence of intense air flow on the oil transfer into the bearing contact area [11]. A similar type of nozzle has also been designed by NTN Corporation [12], with this nozzle oil injected into the counterbore of the inner ring and then transported into the bearing cavity by centrifugal force. Wang [13] compared the oil-air flow inside the bearing cavity between the traditional nozzle and Spinshot II nozzle, the results show that with Spinshot II more oil accumulates near both contact areas of the bearing. However, the inner-ring-oriented nozzles are still restricted, while oil has high flow resistance when flowing on the inner ring surface for its liquid-solid contact with the surface. Several scholars have proposed a method based on groove structure to further improve lubrication efficiency [14–18]. Yan [15] established a whole bearing simulation model with some axial groove structures added to the non-contact area of the bearing inner ring, then compared the lubrication state of the new bearing and the standard one. The results show that the one with grooves has more oil adhering to the ball surface. The flow process of oil in the axial groove has also been studied, it can be divided into three stages dominated by the injection velocity, capillary force, and viscous force, respectively. Through bearing lubrication performance experiments, the validity of using groove to improve the lubrication efficiency is also verified [16]. Based on this, studies on the influences of injection angle, injection speed, and groove width on the groove-enhancing capacity have been carried out [17,18].

However, previous studies are all single-factor analyses. In fact, the factors interact with each other and influence groove performance together. In this paper, through the oil flow simulations and the quantitative experiments, the basic relationship between rotational speed and groove enhancement performance is found, the influences of bearing oil supply amount and nozzle size on grooves are discussed, and the changing law of groove width is found.

2. Analysis of Lubrication Efficiency Enhancing Mechanism of Groove

2.1. Bearing Simulation Model

The H7006C angular contact ball bearing was treated as the research object in this paper. The axial grooves were evenly added to the non-contact area on the inner ring surface, as seen in Figure 1a. Due to its periodic geometry, only 1/17 of the model was taken to accelerate the calculation speed. Meanwhile, in order to monitor the flow of lubricating oil on the inner ring surface more intuitively and quantify the amount of oil entering the raceway, the bearing cage and ball were ignored. Through the extraction of fluidic domain, the simulation models of the H7006C bearing with groove and the standard H7006C bearing were obtained, as shown in Figure 1b.

The geometrical parameters and of the nozzle and groove and their positional relationship are shown in Figure 1c. The main parameters of the grooves include the groove width w and the groove depth h. In this paper, the research mainly focuses on the groove width w, the groove depth h is set as 0.2 mm. The nozzle whose diameter d is 0.5 mm is located on the side of the bearing, and the axial distance between nozzle and bearing side D is 0.5 mm and height H is 0.3 mm, and the injection angle is 30° to the horizontal. The simulation models were divided into unstructured cells by ICEM CFD, and the meshes near the non-contact area were refined to increase the calculation accuracy.

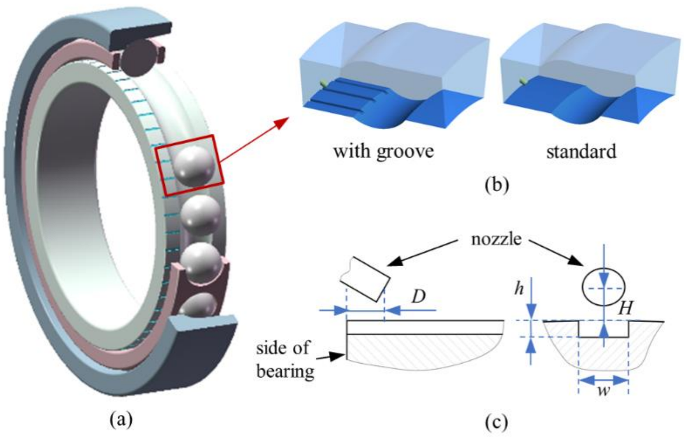

Figure 1. The bearing lubrication simulation model. (**a**) geometry model, (**b**) simulation model, (**c**) parameters and position relationship.

2.2. Boundary Conditions

The calculations were carried out with Fluent. The inner ring of the bearing was set as a rotating wall, and the outer ring and nozzle wall were set as a stationary wall. The nozzle was set as mass-flow-inlet with oil amount m. The other boundaries were all set as pressure-outlet. The set-up of boundary conditions is shown in Figure 2.

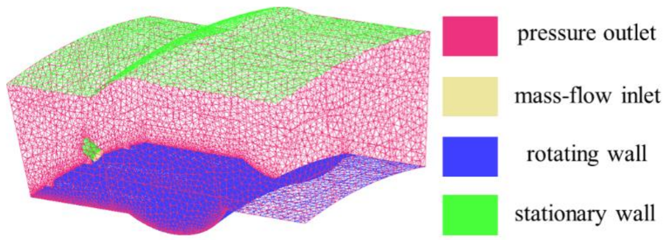

Figure 2. CFD mesh and boundary conditions.

The Volume of Fluid (VOF) model was used to track the oil-air interface, air was considered as the main phase, while the secondary phase is oil. The physical properties of oil are listed in Table 1. The standard k-ε turbulent model was used due to the intense air flow motion inside the bearing cavity.

Table 1. Physical properties of oil.

Physical Properties	Value	Physical Properties	Value
Density ρ (kg/m^3)	876	Surface tension γ (n/m)	0.04
Viscosity μ (kg/m·s)	0.058	Contact angle θ_{CA} (°)	52.8

2.3. Oil-Flow-Enhancing Mechanism of Grooves

Figure 3a shows the oil flow distribution on the inner ring surface of standard bearing and grooved bearing with m = 4 mL/s and bearing rotating speed n = 10,000 rpm. It can be found that more oil flows into the grooved bearing raceway, which shows that grooves can certainly improve the axially flowing capacity of oil. Figure 3b shows the oil distribution on the axial section of the groove of different bearings. Compared with the standard bearing, oil flows longer, and its velocity decreases more slowly. The oil distribution in the groove can be divided into two oil layers. The oil in the groove with a lower axial velocity is named as "bottom oil layer". The oil with a higher axial velocity, which flows on the top of "bottom oil layer", is named as "top oil layer". Although the oil in the bottom oil layer cannot flow into the bearing raceway, it changes the solid-liquid contact when oil flows on the standard bearing into fluid contact and decreases oil flowing resistance.

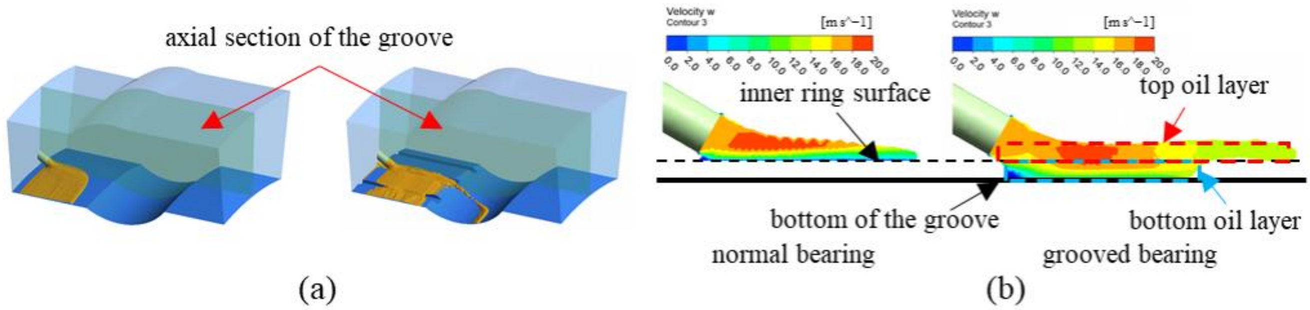

Figure 3. Oil distribution of bearing with m = 4 mL/s and n = 10,000 rpm. (**a**) Oil flow distribution on both bearings, (**b**) oil distribution on the groove axial section.

Since the bottom oil layer in the groove is the key to achieving the enhancement of bearing lubrication efficiency, its distribution area may affect its oil-flow-enhancing capacity. The oil distribution on the bottom wall of the groove was taken as the index D_{oil} to evaluate the enhancing capacity of the groove on bearing lubrication efficiency, which is defined as follows:

$$D_{oil} = \frac{A_{oil}}{A_{groove}} \tag{1}$$

where: A_{oil} is the area of the oil distribution on the bottom wall.

A_{groove} is the area of the bottom wall.

Meanwhile, the amount of oil that flows into the bearing raceway was taken as the other index M_{oil}. Both indexes were acquired through CFD-post, and the methods are shown in Figure 4.

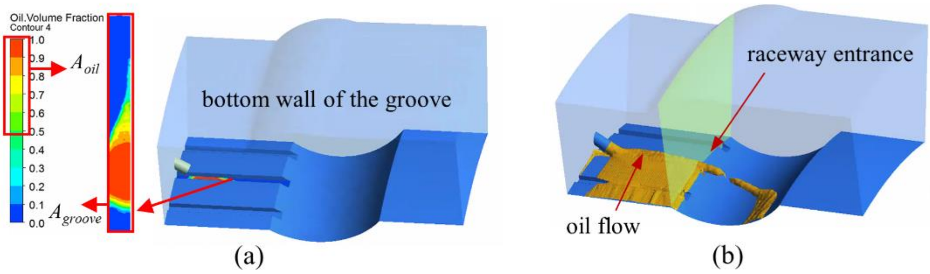

Figure 4. Index acquisition methods. (**a**) D_{oil}, (**b**) M_{oil}.

A_{oil} is the sum of the areas where the oil volume fraction is no less than 0.5. It should be stressed that if no bottom oil layer exists in the groove, the index D_{oil} was taken as 0. For instance, as shown in Figure 5, the oil distribution on the axial section of the groove is similar to that of the standard bearing, which indicates that only a low amount of oil can be enhanced by the groove. In this case, oil is more likely to spread and deflect circumferentially, which is hard for the bottom oil layer to form.

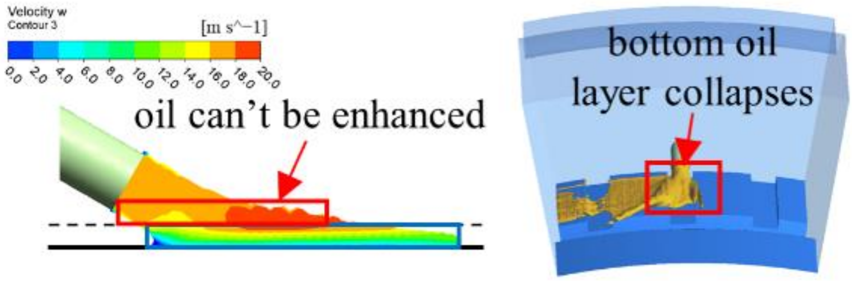

Figure 5. The oil flow collapses inside the groove.

3. Effect of Rotation Speed on the Enhancing Capacity of Grooves

3.1. Simulation Results and Discussion

Figure 6 shows both indexes of different groove widths with the change of bearing rotation speed, where the bars stand for index M_{oil} and curves stand for D_{oil}. When the bearing rotation speed is 1000 rpm, the wider the groove is, the larger M_{oil} is. This is because more oil can be filled in the groove with the increase of the groove width, thereby improving the efficiency on the oil flow. However, M_{oil} decreases with the increase of the rotation speed due to the increased centrifugal force and the deflection effect. Besides, the wider grooves show a more obvious downward trend. The reason is that as the groove width increases, the restriction of the groove side walls on the oil flow is reduced, oil is more susceptible to the deflection of the bearing, resulting in the decrease of the stability of the bottom oil layer, and the drag reduction effect on the top oil layer is greatly reduced, which can be seen in Figure 4.

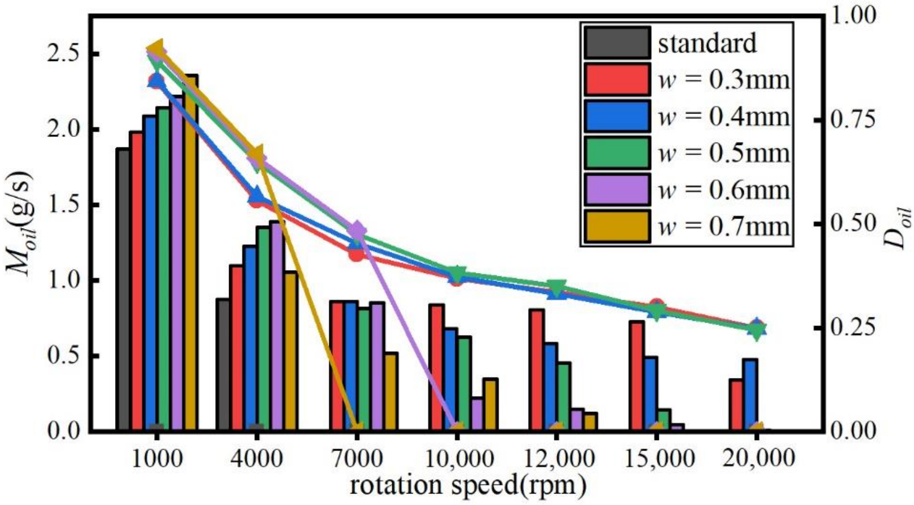

Figure 6. Change in the enhancing capacity of different grooves.

From the curves that show the change of D_{oil} with rotation speed, it can be seen that the at lower rotation speed, D_{oil} is larger with wider grooves, but when rotation speed increases to 10,000 rpm, no more bottom oil layer exists in the grooves with width w 0.6 or 0.7 mm. The downward trend of D_{oil} is gentler when the width w is 0.3 or 0.4 mm. It can be seen that the downward trend of M_{oil} is similar to that of D_{oil}, which indicates that the distribution of the bottom oil layer can reflect the enhancing capacity of the grooves. In the working conditions described above, the groove with width w = 0.3 mm behaves well and can adapt to the changes of rotation speed best. This kind of width is called "the most adaptive width" in the following part of the article. Besides, for each certain rotation speed, there is an optimal width that has the best enhancing capacity, this is called "the optimal width" in the following part of the article.

3.2. Experimental Verification

3.2.1. Experimental System

The experimental system is shown in Figure 7a. It was made up of a motorized spindle, mechanical spindle, electronic balance, oil supply structure, and a simplified bearing inner ring. The simplified bearing inner ring, called "simplified ring" in the following part of the article, is divided into the upstair area and the downstair area, as shown in Figure 7b. The upstair area seen as the non-contact area has grooves and the downstair area seen as the bearing raceway can collect oil by using oil-absorbing tissue. The ring was installed on the end of the mechanical spindle. The nozzle of oil supply structure was set at the top of the ring. The inclination angle between the nozzle and the horizontal plane was 30°. During the experiments, first, the spindle speed was set to the experimental speed and then the oil supply structure was turned on. After 30 min, the increased weights of tissues were treated as the results, which reflects the flowing capacity of oil. To improve the accuracy of the results, seven experiments were carried out for each rotation speed of each ring, five from which with the smallest variance were selected, and the average value of them was taken as the final result.

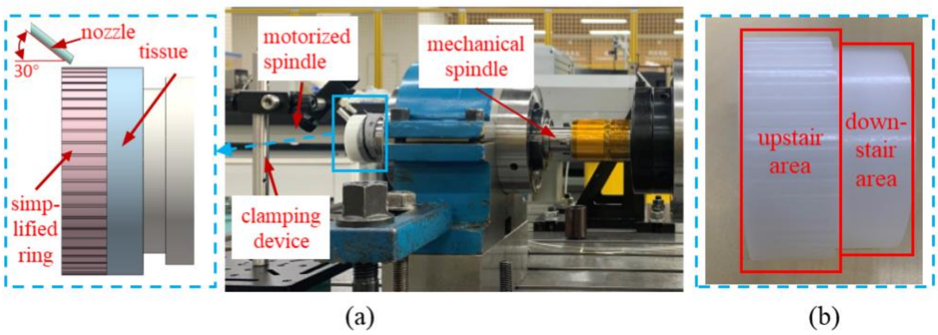

Figure 7. Experiment system and the simplified ring. (**a**) experiment system (**b**) simplified ring.

3.2.2. Results and Discussion

Several simplified rings were manufactured, including one ring without groove as the standard bearing, and three rings with different grooves widths, respectively 0.3, 0.5, and 0.7 mm. The increased weights of oil on these four rings were achieved when the spindles rotated at the speed of 1000, 3000, and 5000 rpm., as shown in Figure 8. The results show that at each rotation speed, the weight increase of rings with groove widths of 0.3 and 0.5 mm is larger than the weight increase of the ring without groove, which shows that proper grooves can enhance the oil flow capacity and let more oil flow into bearing raceway. Secondly, at the same speed, as the groove width increases, the weight increase shows a trend of first increasing and then decreasing, indicating for each rotation speed, there is an optimal width for the grooves. The decreasing trend of weight increase of the ring with a groove width of 0.5 mm is sharper than that of the ring with a groove width of 0.3 mm with the change of speed. This result shows a similar phenomenon to the simulation one that the decline of M_{oil} of wider grooves is larger than that of narrower grooves.

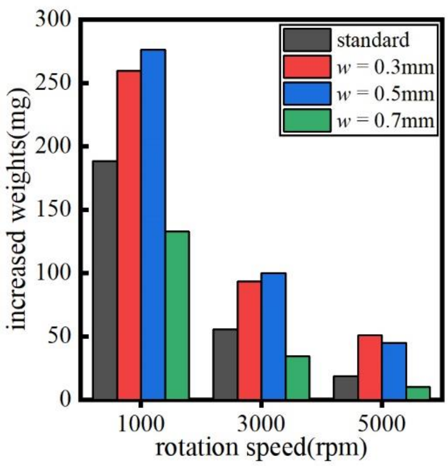

Figure 8. The increased weights of tissue on different rings.

4. Influence of Oil Supply Situation on the Enhancing Capacity of the Grooves

4.1. Influence of Oil Supply Amount on the Groove Width under Different Rotation Speed

Since oil supply is related to the bearing lubrication performance, it may affect the enhancing capacity of the grooves. The bars and the curves in Figure 9 represent the indexes M_{oil} and D_{oil} at oil supply amounts of 4 mL/s and 8 mL/s, respectively. As can be seen, at each rotation speed, the amount of oil flowing into the bearing raceway increases with the increase of oil supply amount. The reason is when the oil supply amount increases, the oil sprayed from the nozzle has higher kinetic energy, which can help resist friction loss, so that the oil flowing on the non-contact area surface may have a larger axially flowing speed, thus improving its axial flowing capacity. Besides, more oil flowing into the groove is beneficial to the form of the bottom oil layer and provides a better drag reduction effect for the top oil layer. In addition, it can be seen from the downward trend that the most suitable groove width value increases with the increase of the oil supply amount, from 0.3 mm at 4 mL/s to 0.5 mm at 8 mL/s.

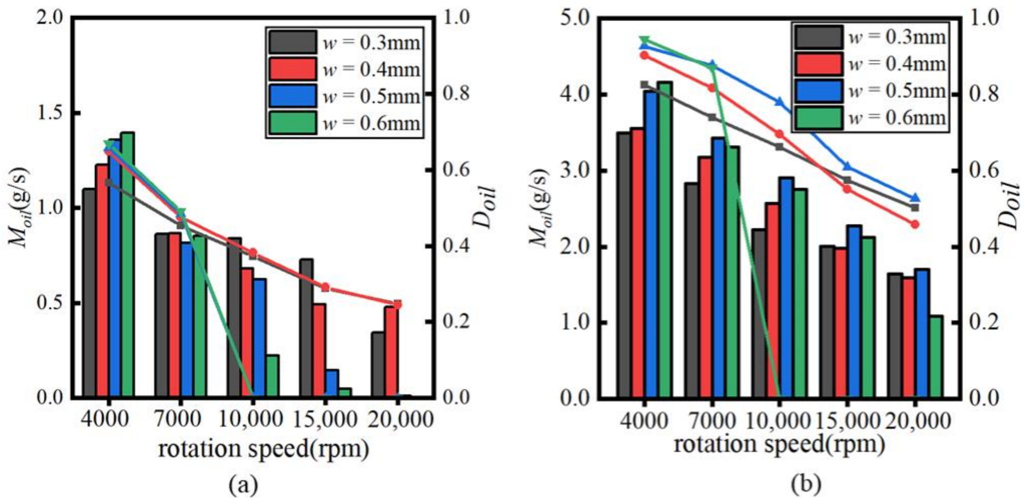

Figure 9. M_{oil} and D_{oil} under different oil supply amounts. (**a**) Oil supply amount m = 4 mL/s, (**b**) Oil supply amount m = 8 mL/s.

For D_{oil}, the results show that as the oil supply amount increases, D_{oil} at each groove width increases. This is because the increase in the oil supply amount can increase the oil amount of the bottom oil layer in the groove and can supplement the oil that leaves the groove structure due to the deflection of the bearing, which effectively improves the dynamic stability of the bottom oil layer. Besides, the bottom oil layer can form in the groove with w = 0.6 mm at high rotation speed with oil supply amount increasing and its upward trend is more obvious than that of groove with w = 0.3 mm, which is owing to the larger capacity of wider groove for oil to feed in. Therefore, the improvement of oil-flow-enhancing capacity of wider grooves seems more obvious than the narrower ones.

4.2. Influence of Nozzle Diameter d on the Groove Width under Different Rotation Speed

The change of the nozzle diameter d will affect the oil flow area, so the enhancing capacity of the groove may also change. The indexes M_{oil} and D_{oil} at nozzle diameter of 0.4 and 0.6 mm are shown by bars and curves of Figure 10, respectively. The wide groove is better at low speed, but the downward trend is greater. Besides, M_{oil} of each groove width at each rotation speed decreases with the increase of nozzle diameter. The reason is that there is a negative relationship between the nozzle diameter and the sprayed velocity when the oil supply amount is a fixed value, oil has lower kinetic energy if it is sprayed from larger nozzles, after it enters the grooves, it will get easier to flow out of the grooves due to the deflection caused by the bearing rotation, resulting in a shorter bottom oil layer and a weaker drag reduction efficiency.

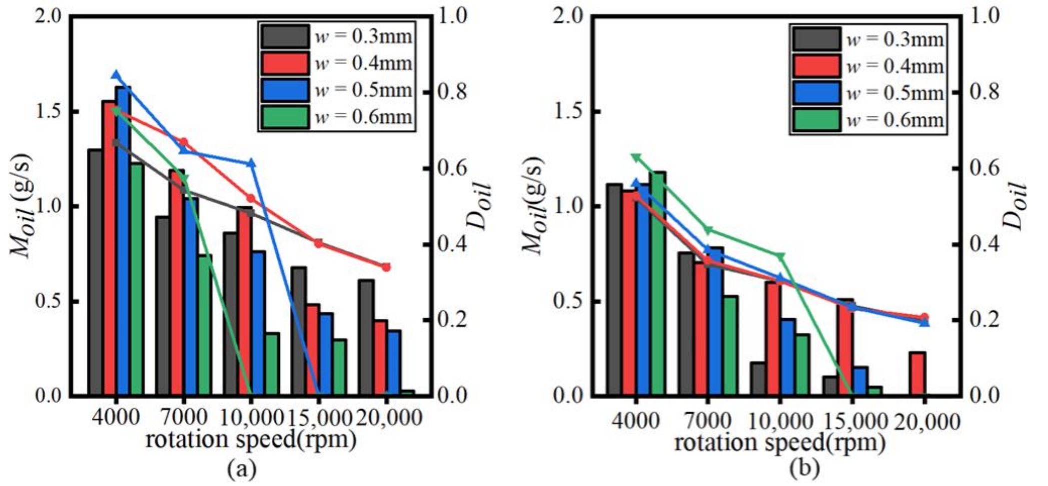

Figure 10. M_{oil} and D_{oil} under different oil supply amount. (**a**) Nozzle diameter d = 0.4 mm, (**b**) nozzle diameter d = 0.6 mm.

For D_{oil}, at a low rotation speed, the smaller nozzle has a larger D_{oil} because it sprays faster oil, which helps the expansion of the bottom oil layer. However, with the increase of rotation speed, the bottom oil layer of the smaller nozzle is less stable. When the nozzle diameter is larger than 0.5 mm, the bottom oil layer can form and remain stable in the groove of width 0.5 mm, but it becomes unstable when the nozzle diameter is 0.4 mm. Besides, the increase of nozzle diameter results in the increase of stability of the bottom oil layer in wider grooves, the oil layer under the 0.6 mm nozzle does not collapse until 15,000 rpm, while under other diameters, it collapses at 10,000 rpm. The downward trend of D_{oil} at each nozzle diameter is also similar to that of M_{oil}. When nozzle diameter d is 0.4 and 0.5 mm, the most adaptive width of groove is 0.3 mm, but when nozzle diameter d is 0.6 mm, the most adaptive width of groove increases to 0.4 mm. From the curves, can we learn that at high rotation speed D_{oil} of groove of width 0.3 mm almost equals that of groove of width 0.4 mm, ideally the oil which can be enhanced can be seen in Figure 11, however, for a fixed oil supply amount, the amount of enhanced oil decreases with the increase of nozzle diameter, this amount becomes the lowest when groove width is 0.3 mm, resulting in a significant shrink of M_{oil} at high speed. To maintain the stability of the bottom oil layer, it is suggested that the groove width should be no more than the nozzle diameter.

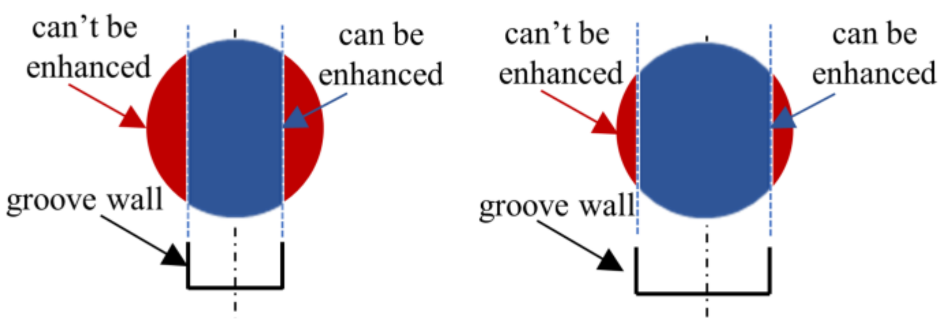

Figure 11. Simplified schematic diagram of enhanced oil.

5. Conclusions

By analyzing the flow characteristics of oil on the surface of different bearings, the enhancing mechanism of groove was analyzed. The enhancing capacity of different grooves was compared through simulations and experiments and the coupling effects of rotation speed, oil supply amount, nozzle diameter on the enhancing capacity were discussed. The conclusions are as follows:

1. The oil distribution in the groove can be divided into the top oil layer and the bottom oil layer. The bottom oil layer is the key for groove to achieve oil flow enhancement.
2. The trend of the experimental results is in good agreement with the trend of the simulation results. Compared with that of the groove of width 0.5 mm, the enhancing capacity of groove of width 0.3 mm seems more adaptive to the increase of rotation speed from 1000 to 5000 rpm.
3. The most adaptive width increases with the increase of oil supply amount and nozzle diameter. To achieve better enhancing capacity, the groove width should be no bigger than the nozzle diameter.

Author Contributions: Conceptualization, C.W., M.W. and L.Z.; Methodology, C.W.; Software, L.Z.; Validation, C.W. and L.Z.; Formal analysis, C.W. and M.W.; Investigation, C.W. and M.W.; Resources, L.Z.; Data curation, C.W.; Writing—original draft preparation, C.W.; Writing—review and editing, L.Z.; Visualization, C.W. and M.W.; Supervision, L.Z.; Project administration, L.Z.; Funding acquisition, L.Z. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded the National Key R&D Program of China (No. 2020YFB2007900).

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Conflicts of Interest: The authors declare no conflict of interest.

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