Ferrofluid lubrication for ball bearings to avoid starvation

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Abstract

Purpose – This paper aims to prevent oil starvation and improve the service life of the rolling bearings.

Design/methodology/approach – A thrust ball bearing with magnetic circuit structure is proposed for ferrofluid lubrication. With the aid of magnetic field, ferrofluid can be maintained in the contact area of rolling bodies to delay lubricant loss. Experiments are performed to ensure the validity of the designed bearing.

Findings – Compared with conventional lubricant, service life of the ferrofluid lubricated bearing can be prolonged under magnetic field. In addition, with a proper magnetic field distribution, lubricant starvation may be limited under the conditions of present experiments.

Originality/value – This work provides a method to control the starved lubrication of rolling bearings with restricted lubricant supply.

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Keywords Seals, Ferrofluids, Rolling bearings, Starved lubrication

Paper type Research paper

1. Introduction

Loss-of-lubricant from transmission system will lead to disastrous consequences for helicopters. Different works have been performed to allow the transmission system operating for more than 30 min after a failure of lubrication system (Mba *et al.*, 2012). However, such aim is not always achieved up to date.

Rolling bearing is one of the core components in transmission system and its reliability is highly dependent on lubrication. While in the event of a lubrication system failure, oil starvation is usually inevitable (Rashid *et al.*, 2015), which will induce metallic contacts, overheating and deformation or, in the worst case, bearing seizure. To prevent or delay the starvation, an intuitive approach is to confine the lubricant in the Hertz contact zone of the bearing. From the view point of surface design, surface patterns could be an effective way to guide the lubricant into the friction areas (Dai *et al.*, 2014; Grützmacher *et al.*, 2016; Dai *et al.*, 2018; Grützmacher *et al.*, 2019). From the view point of lubricant, developing functional fluid is also a valid path.

Ferrofluid (FF) is a suspension of single-domain ferromagnetic particles, with typical dimensions of about 10 nm, dispersed in appropriate carrier liquids (Holm and Weis, 2005). One of the interesting features of FF is that it exhibits normal fluid behavior coupled with magnetic property. In contrast with conventional oils, FF can be restricted at the desired location by an external magnetic field (Prajapati, 1995). From the view point of lubrication, such characteristic may bring a new idea to avoid oil starvation for bearings. The use of FF to control starvation in a ball–plate contact was confirmed by Andablo-Reyes *et al.* (2010). Ochoński (2007) presented some new structure designs of various sliding bearings lubricated and sealed with FF. Patel *et al.* (2017) experimentally studied the film pressure and temperature of a FF lubricated journal bearing. Recently, Xu *et al.* (2020) introduced a FF lubricated rolling bearing with improved operation behavior. However, the magnetic field distributed in the bearing race was generated by discrete columnar magnets, without forming magnetic circuit in the bearing. In addition, the starved lubrication and short service life still remained. Could the operation behavior be further increased when a magnetic circuit was introduced in bearing structure? How about the effect of the magnetic parameter on the bearing's operation behaviors? There is still little available knowledge about this.

This paper focused on a type of FF lubricated thrust ball bearing with a self-sealing structure. Tribological behaviors of the bearing with restricted lubricant supply were carried out to assess the impact of magnetic field on lubrication starvation. Besides, the static FF self-sealing capacity of the bearing was also explored. The application of using FF for bearing lubrication may provide an effective way to overcome and reduce the oil starvation.

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2. Structure of the bearing

Figure 1 shows the structure of the FF lubricated bearing. A standard thrust ball bearing (ID code: 51104, Material: GCr15) is chosen. The surface roughness of the race and the ball is about 0.3 μm. The top cover and gasket are made of low carbon steel with good magnetic conductivity. Non-magnetic aluminum alloy is used for the bottom cover. The magnetic field is generated by a ring NdFeB magnet (Operating temperature: <100°C, size: $\Phi35 \times \Phi29 \times 2$ mm) with an axial magnetization. A rubber ring is fixed between the magnet and lower race of the bearing. There is a certain clearance δ (0.5 mm) between the top cover and ring magnet to ensure proper operation.

To reveal the influence of magnetic field, four ring magnets with the same size but different surface magnetic field induction (labeled as B_1: 70 mT, B_2: 90 mT, B_3: 110 mT and B_4: 130 mT) were used in this work. Excessive strength of the magnetic field will affect the rolling behavior of the ball due to the strong interaction between the ball and race. Much lower magnetic strength cannot absorb the FF in the contact area effectively. The distribution of magnetic field in the bearing assembled with each magnet was analyzed by using an Ansys Maxwell 16.0 software. The magnetic and geometrical parameters input were in accordance with each bearing and the results were presented in Figure 2 (the area shown corresponding to the red rectangle in Figure 1). As can be seen, the magnetic circuit is formed in the designed bearing. More importantly, the contact area of the ball and the race exhibits the maximum magnetic flux density. Therefore, the FF will always be pulled into this area to hold a fluid film on the surface of rolling body. In addition, the higher surface field intensity of the magnet, the larger magnetic field distribution in contact area of the ball and race.

3. Experimental procedures

The tribological behaviors of the designed bearing under different lubricating conditions were conducted by using an MMW-1 tribometer. As shown in Figure 1, the bearing seat, bottom cover as well as the lower race remained stationary and the upper race connected with shaft was driven by a motor with an adjustable rotate speed. Before tests, each part of the bearing was cleaned ultrasonically in acetone and dried at the temperature of 40°C in oven. Two types of condition (axial load/speed: 500 N/1200 rpm and 800 N/2000 rpm) were applied, corresponding to the contact pressures of 2.86 and 3.32 GPa. The volume of the lubricant dropped to the race each time was strictly controlled at 0.02 ml via a micro syringe. Variations of friction torque and temperature were measured by a torque sensor and a thermocouple, respectively. Before an abrupt change of the friction torque, the stable operation time was defined as the service life the bearing. Each group of tests was repeated for three times. After that, worn surface of the bearing race was observed by an optical microscope (Keyence Corp., Japan) and 3 D profile (Bruker Corp., USA).

Self-sealing capacity is another important feature of the designed bearing. As shown in Figure 1, when FF is injected into the clearance δ, a liquid O-ring will be formed. The static sealing capacity of the FF in the clearance was explored by using a pressure testing platform. As can be seen in Figure 3, a piston cylinder was connected with the bearing and the piston driven by a lead-screw moved down at the speed of 0.5 mm/s. Due to the FF sealing in the clearance, air pressure in the bearing cavity increased gradually, and the pressure value was measured by an air pressure sensor.

In the experiments, commercial FF consisting of Fe_3O_4 nanoparticles with the average size of 15 nm was used. The saturation magnetization was 15.9 kA/m, corresponding to the particle volume fraction of about 4.9%. These magnetic particles covered with a surfactant of oleic acid were dispersed in diester as carrier liquid. The fluid is stable and no solid-liquid separation appears in the used sample. The FF presents the viscosity of about 79 mPa•s, a bit higher than its carrier liquid (diester, 50 mPa•s). For comparison, tribological performance of the bearing lubricated with pure diester (conventional lubricant) was also tested.

Figure 1 Structure diagram of the FF-lubricated rolling bearings

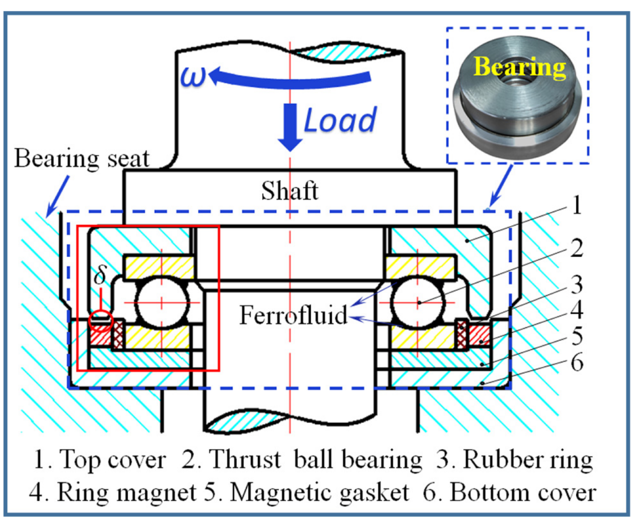
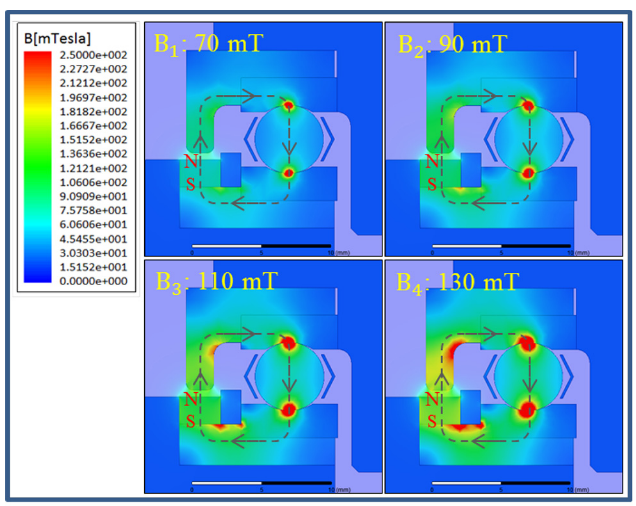
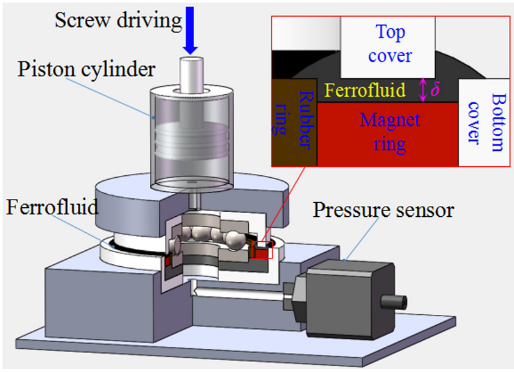


Figure 2 Magnetic field distribution in the bearing



Note: The area shown corresponding to the red rectangle in Figure 1

Figure 3 Sketch map of the self-sealing test platform



4. Results and discussion

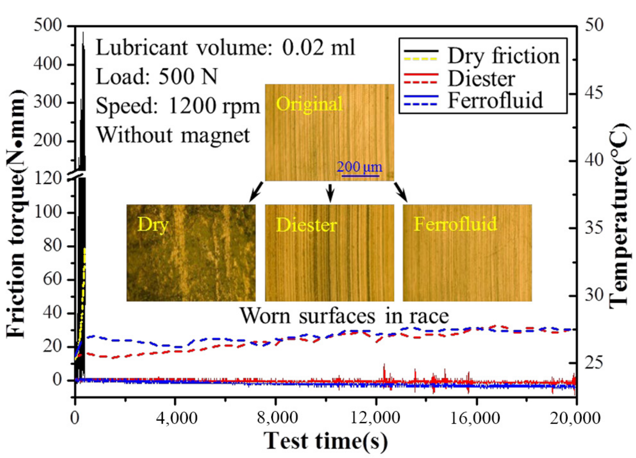
Figure 4 presents the preliminary experimental results of the bearing (without magnet) lubricated with diester and FF. And the result under dry friction condition is also given for comparison. As can be seen, the friction torque and temperature under dry friction condition increased rapidly at the beginning of the test, accompanying with loud noise and obvious vibration. The result indicates that the bearing cannot work without lubricant. Significant changes appeared when 0.02 ml lubricants were added to the bearing race. It can be seen that the value of the friction torque reached about 0, and it remained stable during the test time of 2×10^4 s. Besides, the temperature of the bearing rose smoothly. After the experiment, the worn surfaces of the bearing raceway were observed, as shown in Figure 4 inset. Extensive and severe wear associated with scuffing can be found in the bearing race after dry friction. However, the worn surface only presents deep and narrow grooves when lubricated with diester. Compared with diester lubrication, a narrow superiority of FF manifests itself in anti-wear property, as the variations of friction torque and temperature are almost the same. As can be seen, there is an invisible difference between the original and worn surface lubricated with FF. As pointed in Trivedi *et al.* (2017), the nanoparticles in FF may enter the interface of the ball and race, playing as filler to partially reduce the contact. In addition, these particles show the ability of repairing worn surface to improve the wear resistance (Wang *et al.*, 2008).

As shown in Figure 4, whether lubricated with diester or FF, the friction torque and temperature both remained low and stable. Such result indicates that effective lubrication film has been formed in the contact zone throughout the test time. To further figure out the different operation behaviors, especially the influence of magnetic field, severe experimental condition of the higher load and speed was used in the following tests.

Figure 5 shows the typical variations of friction torque and temperature of bearings with ring magnet. As can be seen, the bearing lubricated with diester operated properly for the initial 5.3×10^3 s, and then the friction torque revealed a sudden increment from almost 0 to 400 N·mm. In addition, the corresponding temperature kept growing from the beginning of the test and also presented a sharp explosion. With the increase of the normal load, Hertzian contact stress enhances, drawing a higher demand in lubrication. Worse, under the growth of the speed, centrifugation may play a dominate effect on oil loss. Figure 5(a) inset shows the trace amounts of diester left in the bearing race after 5 min test. For visual observation, red stain was added into the diester in advance. It indicates that most of the diester was splashed out the race and the bearing temperature increased quickly, which led to a further decline in the viscosity. Thus, the lubrication condition got worse gradually, starved lubrication occurred and failure first appeared for the bearing lubricated with diester.

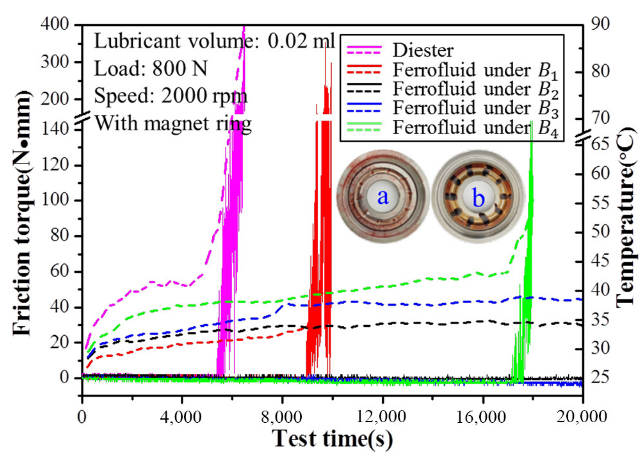
When lubricated with FF, different operation performances appeared due to the effect of magnetic field. In comparison to the diester, the service life increased to 8.9×10^3 s as the bearing equipped with ring magnet B_1. According to the results of magnetic field analysis (see in Figure 2), the contact zone of the ball and race shows the highest magnetic flux density. So long as the FF is dropped on the bearing race, it will be

Figure 4 Variations of friction torque and temperature of bearing without magnet



Notes: The solid refers to friction torque and the dashed refers to temperature

Figure 5 Variations of friction torque and temperature of bearing with magnet



Notes: The solid refers to friction torque and the dashed refers to temperaturea spontaneously absorbed and restricted in the contact interface. Figure 5(b) inset gives the appearance of residual FF in the upper race after 5 min test as well. As observed, in spite of a certain spread, most of the FF was concentrated in the contact position, which indirectly verifies the validity of the magnetic structure designed in the bearing. Due to the lower magnetic field, FF escapes from the race bit by bit and oil starvation is unavoidable.

With the increasing of the magnetic field, the service life continued to extend and the bearings fixed with magnet B_2 and B_3 presented low friction torque within the 2×10^4 s test time. Compared with the outcome reported in Xu *et al.* (2020), the lubrication life increased obviously even at a higher speed condition. Besides, the temperature growth was also mild. It can be deduced that the lubrication mechanisms have been transferred from starvation to fully flooded condition. On the one hand, the enhanced magnetic field in the contact area may exclude the centrifugal effect to some extent and prevent lubricant loss (Xu *et al.*, 2020). As mentioned in Grützmacher *et al.* (2017), an earlier lubricant film breakdown may be induced by higher centrifugal forces. On the other hand, with the increasing of the applied magnetic field, the load carrying capacity of the squeeze film increases, which will improve the lubricity (Shukla and Kumar, 1987; Patel and Deheri, 2007). Therefore, the FF lubricated bearing equipped with magnets B_2 and B_3 exhibits better operation behavior.

Unexpectedly, lubrication failure appeared once again despite applying the magnet B_4 with the highest surface magnetic field. It is known that magnetic particles in the carrier liquid randomly orient in the absence of a magnetic field. When a magnetic field is brought closer to FF, these particles will orient themselves toward the direction of external field. The orientation of the particles causes additional resistance to the flow (Rosensweig, 1985). As shown in Figure 2, the bearing fixed with magnet B_4 possesses the strongest magnetic field in the contact area of the ball and the race. Obviously, excessive magnetic field will lead to a higher flow resistance. As a result, the internal shear force increases and the directly manifestation is the quickest temperature rise among the four FF lubricated bearings. Meanwhile, particles also prefer to agglomerate in the contact area due to the high magnetic field, which may also cause to the increase of the friction.

Figure 6 presents the morphologies of the worn surface in bearing race after tests. As can be seen, bearing lubricated with diester shows severe adhesive wear with material peeling inside the wear track. Besides, the pits can also be found according to the 3-D image. The width of black wear scar in race decreases obviously when lubricated with FF under B_1 and the adhesive zone reduces a lot. Similar features are also found for the bearing equipped with magnet B_4. In comparison to the original one, no significant changes appear as the bearing fixed with magnets B_2 and B_3.

Figure 6 Morphology of the worn surface in the race

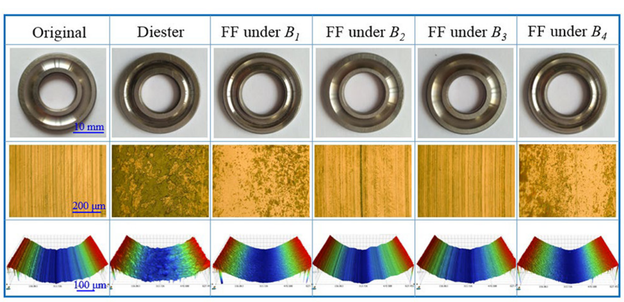


Figure 7 shows the results of the static self-sealing capacity of the designed bearing. In the absence of the magnetic field, the sealing capacity of the FF in the gap was almost negligible. As the bearing equipped with magnet, the air pressure in the bearing cavity grew linearly due to the uniformly moving down of the piston. And a sudden drop in pressure means the FF sealing film broken or seal failure. Then, a new balance is established between the gas compression and leakage. The peak value is defined as the critical sealing capacity of the bearing (ΔP). It shows that the critical sealing capacity increases with the increment of the surface magnetic field strength of the magnet. The maximum ΔP of the bearings was up to 8 KPa when applied with magnet B_4.

As mentioned in Berkovsky *et al.* (1993), the pressure difference sustained by a FF drop can be simply written as:

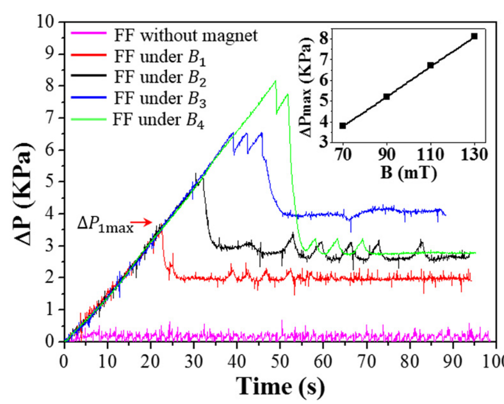
$$\Delta P = M_s B_{max} \tag{1}$$

where M_s is the saturation magnetization of the FF and B_{max} is the maximum value of magnetic field induction in the working gap. Figure 7 inset presents the relationship between ΔP and B_{max}, assuming surface magnetic field induction of each magnet as the maximum in the clearance (see in Figure 2 and 3). It seems that ΔP is proportional to ΔB, which is consistent with equation (1). Compared with traditional labyrinth and lip seals, the bearing with FF seal has some advantages, such as simplicity in structure, easy maintenance and high reliability.

5. Conclusions

Starved lubrication often occurs in rolling bearings under the harsh conditions. In this paper, an FF-lubricated rolling bearing is designed with the magnetic field concentrated at the contact of ball and race. Thus, FF can be confined on the surface of rolling bodies to promote lubricant supply. Meanwhile, an unintentional FF self-sealing structure is also formed between the bearing's top

Figure 7 Self-sealing capacity of the designed bearing with different magnets



cover and the magnet. Compared with conventional lubricant, service life of the FF lubricated bearing can be prolonged obviously with the addition of magnetic field. Moreover, with the proper magnetic field distribution, lubricant starvation may be limited under the conditions of this study. The pressure tests indicate that FF in the clearance between magnet and top cover has a self-sealing property and the pressure-resistant capacity of the fluid film is proportional to the maximum magnetic induction in the clearance.

This work presents a preliminary result on bearing starvation controlling by FF lubrication. From the view of industrial applications, further work, such as magnetic field optimization, FF parameters, synergy of the field and FF as well should be carried out.

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