Effects of surface texturing on the tribological performance of the rolling bearing system

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Abstract: Surface texturing is one important means to improve the tribological performance of friction pairs. Meanwhile, the friction of the rolling bearing is related to the sliding behavior of the contact surfaces. At present, there are few research work on the textured rolling bearings. Therefore, it would be meaningful to study the potential of surface texturing in improving the performance of rolling bearing system. This paper developed one EHL (elasto-hydrodynamic lubrication) model for the rolling bearing system. In particular, the roller/outer ring conjunction was used as the research object. Effects of surface texturing on the tribological performance of the rolling bearing systems were studied under different bearing speeds. As a main conclusion, the local oil film thickness of the rolling bearing system can be increased after surface texturing. It seems that the textured surface can obtain a special benefit on the tribological properties of rolling bearing system.

1. Introduction

Surface texturing is one important means to improve the tribological performance of friction pairs [1]. When the friction pairs are under the fully flooded lubrication, the textures can be used as the small hydrodynamic bearings, resulting in additional hydrodynamic support. In this way, the direct contact of the friction pair surfaces can be effectively avoided, thus friction can be reduced [2]. When the friction pairs are under the starved lubrication, surface textures may act as micro-reservoirs to reserve and release oil to the interfaces of contacting parts [3]. In addition, wear debris may be trapped within surface textures to reduce wear and, subsequently, to prolong the life of mechanical components[4]. Surface texturing has gained wide attention worldwide [3, 5, 6].

The rolling bearings are sometimes called the antifriction bearings, because that they have the smaller friction power than the sliding bearings in the work. In fact, the friction of the rolling bearing is related to the sliding behavior of the contact surfaces. Rolling bearings are usually operated under the boundary/mixed lubrication. Surface texturing is one important method to reduce friction, which has various effects in different lubrication conditions. As early as 1992, Akamatsu et al. found that the use of the randomly distributed textures can improve the life of the rolling bearings under the mixed/boundary lubrication [7]. After that, based on the input data from Akamatsu et al. [7], Zhai et al. analyzed the effects of surface texturing on the fatigue life of rolling bearing [8]. At present, there are few research work on the textured rolling bearings. Therefore, it is necessary to study the potential of surface texturing in improving the performance of the rolling bearing system. In this paper, an EHL model would be developed for the rolling bearing system. The effects of surface texture in rolling bearing system will be studied by numerical simulations.
2. Model description

2.1. Governing equations

In this paper, the needle bearing was studied as the research object. Figure 1 shows the overview of the roller/outer ring conjunction. Due to the fact that the length of roller is long enough, the lubrication problem of the roller/outer ring conjunction can be regarded as an EHL problem. The clearance between the roller and the outer ring of the bearing can be expressed as:

\[ h(x) = h_0 + \frac{x^2}{2R_{or}} + v(x) \]  

where \( h_0 \) denotes the minimum oil film thickness, \( v(x) \) represents the elastic deformation amount. \( R_{or} \) is the equivalent curvature radius, whose expression is given as follows:

\[ R_{or} = \frac{R_o R_r}{R_o + R_r} \]  

where \( R_o \) is the curvature radius of the outer ring, \( R_r \) is the curvature radius of the roller.

When the outer ring of bearing is textured, the clearance between the rolling body and the textured outer ring of bearing can be given as:

\[ h(x) = h_0 + \frac{x^2}{2R_{or}} + 0.5h_g \left( \cos \left( x' \pi / r_g \right) + 1 \right) + v(x), \quad x' \in \Omega \]  

where \( h_g \) is the depth of texture, \( 2r_g \) represents the width of texture features. \( x' \) is located in the local coordinate axis. The local coordinate is located in the center of texturing features [1]. \( \Omega \) is the area occupied by one texture. The textures are well-distributed on the surface of outer ring of bearing.

It should be noticed that, according to the structure of the rolling bearing, when the roller is being in pure rolling contact, the rotation speed of roller can be calculated as [9]:

\[ n_2 = \frac{1 + 2s_g}{2s_g(1 + s_g)} n \]  

Meanwhile, the revolution speed of roller is given as [9]:

\[ n_g = \frac{1}{2(1 + s_g)} n \]  

where \( s_g = \frac{R_r}{R_i} \), which denotes the ratio of the radius of roller and the radius of inner ring of bearing.

The hydrodynamic pressure would be generated in the convergent-divergent gap. For the roller/outer ring conjunction, the hydrodynamic pressure can be predicted by the following equations:

\[ \frac{\partial}{\partial x} \left( \rho \frac{\partial p}{\partial x} \right) = 12U_s \frac{\partial (1 - \theta) \rho h}{\partial x} \]  

\[ p + \theta - \sqrt{p^2 + \theta^2} = 0 \]

where \( h \) is the oil film thickness, \( p \) is hydrodynamic pressure, \( \theta \) is the cavity fraction. It is
noteworthy that the cavity fraction $\theta$ is related to the cavitation. The term $\theta$ is influenced by the pressure $p$. As shown in Eq. (7), the term $\theta$ is set to 0, when the hydrodynamic pressure $p$ is greater than the cavitation pressure ($A$ value of 0 is used as the cavitation pressure in this study). When the cavitation takes place, the hydrodynamic pressure $p$ is equal to the cavitation pressure, the term $\theta$ is greater than zero and meets the condition $\theta > 0$.

In addition, $U_s$ is the entrainment speed, which can be obtained by $U_s = (u_1 + u_2)/2$. $u_1$ is the speed of the upper surface, $u_2$ is the speed of the lower surface. For the roller/outer ring system, $u_1 = \frac{2\pi R_1}{60n_2}$ while, $u_2 = 0$.

### 2.2. Rheological relationship

In this work, the relationship between the lubricant density and pressure would be predicted by the model of Dowson and Higginson [10]. The corresponding expression is given as follows:

$$\rho = \frac{C_1+C_2p}{C_1+p} \rho_0$$

where $\rho_0$ is the lubricant density. $C_1$ and $C_2$ are constant coefficients. They depend on the lubricant under consideration. In this work, $C_1 = 0.59 \times 10^9$ and $C_2 = 1.34$ are adopted in the simulation [10].

In addition, it is widely accepted that the lubricant viscosity is influenced by the thermoviscous effect, the piezoviscous effect and the shear thinning behavior. The piezoviscous effect can be considered by the Roelands equation [11], while, the behavior of the shear thinning fluid can be characterized by the Cross equation [12]. Therefore, by combining the Roelands equation with the Cross equation, the viscosity expression can be given as:

$$\mu = \mu_0 e^{-[(\mu_0+9.67)/(1+5.1\times10^{-9}p)^{\alpha}]-1} \left[ n_3 + \frac{1-n_3}{1+n_1n_3} \right]$$

where $\mu_0$ is the viscosity at atmospheric pressure. $\alpha$ is the lubricant parameter. It depends on the lubricant under consideration. $n_1$ and $n_2$ are parameters that depend on the lubricant under consideration. $n_3$ is the ratio of the high shear ratio viscosity to the low shear rate viscosity. This parameter is related to the lubricant under consideration. Meanwhile, the shear rate $\dot{\gamma}$ can be obtained by the following equation:

$$\dot{\gamma} = \frac{|v|}{h}$$

In addition, the corresponding data of oil 5W30 are listed as follows: the lubricant viscosity ($\mu_0$) is 0.04867 Pa.s, the lubricant temperature is 313.15 K, the lubricant density ($\rho_0$) is 850 Kg m$^{-3}$, the ratio of the high shear ratio viscosity to the low shear rate viscosity ($n_3$) is 0.71.

### 2.3 Elastic Deformation

In this paper, the elastic deformation of surface is considered. For the one-dimensional linear contact problem, the elastic deformation amount can be obtained by [13]:

$$\nu(x) = -\frac{2}{\pi E} \int_{s_1}^{s_2} p(s) \ln(s-x)^2 \, ds + c$$

where $s$ is an additional coordinate on the $x$ axis. It represents the distance between the arbitrary line load $p(s)$ and the origin of the coordinates. $p(s)$ is the load distribution function. For the EHL problem, $p(s)$ represents the hydrodynamic pressure distribution; $s_1$ and $s_2$ are the starting and ending coordinates of the load; $c$ is the undetermined constant, which can be incorporated into $h_0$. $E$ is the equivalent modulus of elasticity, and its formula is given as follows:

$$\frac{1}{E} = \frac{1}{2} \left( \frac{1-v_o^2}{E_o} + \frac{1-v_r^2}{E_r} \right)$$

where $E_o$ is the elastic modulus of the outer raceway, and $E_r$ is the elastic modulus of the roller; $v_o$ is the Poisson ratio of the outer raceway, and $v_r$ is the Poisson ratio of the roller.

### 3. Solution procedure

As shown in Figure 2, the involved flow chart of numerical scheme is given in detail. The sequence of
computation involves with:

(I) Some parameters should be assumed. These parameters include the pressure distribution $p$, the lubricant density $\rho$, the lubricant viscosity $\mu$ and the minimum oil film thickness $h_0$.

(II) Based on Eqs. (6)-(10), the hydrodynamic pressure $p$ can be obtained.

(III) According to Eqs. (11)-(12), the elastic deformation of surface can be predicted.

(IV) The oil film thickness is updated based on Eqs. (1)-(3).

(V) Steps (I)-(IV) should be repeated until the error between the updated values of pressure (which is obtained at the current iteration) and the previous values of pressure (which is obtained at the previous iteration) is below or within an allowable precision ($10^{-6}$).

(VI) In the calculation, the sum of the hydrodynamic support is assumed to equilibrate the applied load. If the convergence criterion of load is satisfied, the calculation is completed, otherwise, the minimum oil film thickness ($h_0$) should be adjusted. In this way, the calculations of both the hydrodynamic pressure and the elastic deformation need to be repeated.

![Image of the simulation flow chart](image)

Figure 2 The simulation flow chart

4. Results and discussion

The simulation conditions used in this section are given as follows: The lubrication temperature is 353.15 K; The lubrication viscosity is 0.01247 Pa.s; The effective length of roller is 18 mm, and its diameter is 4.5 mm; The diameter of outer ring is 56 mm; The bearing speed is 2000 RPM. It means that the speed of the roller/outer ring model is 1.338 m/s. The applied load of the roller/outer ring conjunction is 200 N. Under the applied load, the half width of Hertz contact ($b_{HZ}$) is 17.64 $\mu$m. The calculation length along the $x$ direction in the simulation is $6b_{HZ}$. $X$ is the dimensionless term of $x$, which is obtained by $X = x/b_{HZ}$. The starting point of the dimensionless $X$ is -4. The ending point of the dimensionless $X$ is 2. When the surface of outer ring is textured, the texture width is 1.7639 $\mu$m, the texture depth is 0.1764 $\mu$m, and the texture space is 8.8195 $\mu$m. The detailed parameters of the corresponding rolling bearing are shown in table 1.
### Table 1 Rolling bearing parameters

| Parameters                        | Values  |
|----------------------------------|---------|
| Diameter at inner raceway        | 47 mm   |
| Diameter at outer raceway        | 56 mm   |
| Elastic modulus at inner raceway | 200 GPa |
| Poisson's ratio at inner race way| 0.3     |
| Elastic modulus at outer raceway | 200 GPa |
| Poisson's ratio at outer race way| 0.3     |
| Diameter of rollers              | 4.5 mm  |
| Number of rollers                | 24      |
| Elastic modulus of rollers       | 205 GPa |
| Poisson's ratio of rollers       | 0.3     |
| Effective length of rollers      | 18 mm   |

#### 4.1. Validation Case

In order to verify the calculation code of this model, the results of the current model would be compared with the results of Huang et al. [13]. It is noteworthy that the simulation conditions are consistent with the literature. The comparison results of numerical simulation are given in Figure 3. According to the results, it is found that only a difference of 0.5% in peak pressure is detected. It appears that the results achieved from the present model and the reference from Huang et al. match considerably well.

![Figure 3](image)

**Figure 3** The oil film thickness and pressure distribution results for the EHL problem

#### 4.2. Textured system results

Figure 4 (a) shows the results about the distributions of oil film thickness when the outer ring is textured or not. In order to reveal the influence of surface texturing on the oil film thickness distribution, the enlarged image of oil film thickness distribution is presented in Figure 4 (b). As shown in Figure 4 (b), in the middle of the lubrication region, the oil film is approximately uniform. In the outlet of the lubrication region, the oil film begins to fall, forming the necking phenomenon. The necking phenomenon is one typical characteristic of the steady lubrication of the line contact. Moreover, it is observed that the presence of surface texturing can affect the neck position slightly, affecting the minimum oil film thickness. In the current case, the minimum oil film thickness is increased by 2.219%. In addition, Figure 4 (c) shows the results about the hydrodynamic pressure when the outer ring is textured or not. In addition, Figure 4 (c) also gives the distribution of Hertz contact pressure. As shown in Figure 4 (c), it can be found that the pressure distribution is similar to that of Hertz in the middle of the lubrication region. At the necking position, the oil film pressure has a significant secondary pressure peak. Surface texturing would influence the pressure distribution, which significantly affects the...
secondary pressure peak.

Figure 4 The comparison results for the roller/outer ring conjunctions with or without textures on the surface of outer ring (the bearing speed is 2000 rpm): (a) the oil film thickness distribution, (b) the enlarge drawing of the oil thickness distribution, (c) the hydrodynamic pressure distribution

The bearing speed can affect the entrainment speed of roller/outer ring conjunction. Different entrainment speeds will result in different distributions of oil film thickness and hydrodynamic pressure. Figure 5 shows the distributions of oil film thickness as well as the hydrodynamic pressure distribution when the bearing speed is 1000 RPM. It should be pointed out that when the bearing speed is 1000 RPM, the entrainment speed is 0.669 m/s. According to the results, in the current case, the minimum oil film thickness is increased by 1.338 %.

Figure 5 The comparison results for the roller/outer ring conjunctions with or without textures on the surface of outer ring (the bearing speed is 1000 rpm): (a) the oil film thickness distribution, (b) the hydrodynamic pressure distribution

Figure 6 shows the distributions of oil film thickness and hydrodynamic pressure when the bearing speed is 6000 RPM. It is noteworthy that when the bearing speed is 6000 RPM, the corresponding
The entrainment speed is 4.014 m/s. The results show that the minimum oil film thickness is increased by 1.612% in the current case. The increase of the minimum oil film thickness can effectively avoid the direct contact of the friction pairs. In this way, the wear of the friction pairs can be reduced, which can result in the improvement of the tribological properties of the friction pairs.

![Figure 6](image)

**Figure 6.** The comparison results for the roller/outer ring conjunctions with or without textures on the surface of outer ring (the bearing speed is 6000 rpm): (a) the oil film thickness distribution, (b) the hydrodynamic pressure distribution

In fact, the tribological problem of the rolling bearing system belongs to the non-conformal contact problem. With respect to the conformal contact, applying the textures on the non-conformal contact surfaces is controversial [14]. Previous experiments and numerical simulations show that a beneficial effect on the performance under the elasto-hydrodynamic lubrication can be obtained by the application of the proper surface textures. The local/average oil film thickness of the friction pairs under rolling/sliding condition can be increased [3]. At the same time, a large number of elasto-hydrodynamic lubrication studies show that: surface texturing would provide a positive effect when the size of the surface textures is less than the dimension of Hertz contact [15-17]. In contrast, when the size of surface textures is greater than the dimension of Hertz contact, the textured surface would have a negative effect [18-20]. The textures of different parameters would lead to the various tribological properties of friction pairs (including positive effects or negative effects). However, the optimization of surface texturing is beyond the scope of this paper. The purpose of this paper is to prove that surface texturing is also a potential way to improve the performance of rolling bearings.

5. Conclusion

In order to study the potential of surface texturing in improving the performance of the rolling bearing system, this paper develops one elasto-hydrodynamic lubrication model for the rolling bearing system. The roller/outer ring conjunction was taken as the research object. The effects of surface texturing were studied under different bearing speeds.

The results show that the suitable surface textures can effectively increase the minimum oil film thickness of the friction pairs. The increase of the minimum oil film thickness can effectively avoid the direct contact of the friction pairs. In this way, the wear of the friction pairs can be reduced and the involved tribological properties can be improved. At the same time, it is found that the effects of surface texturing are influenced by the bearing speeds. Different bearing speeds will result in various tribological performances.

It is necessary to point out that there are some limitations in this paper. The model of constant load and constant speed used in the simulation is a relatively simple theoretical model. The study of the effects of surface texturing on the performance of the rolling bearings should take into account the actual working conditions of the rolling bearings. One more advanced theoretical model should be established.

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References
[1] Gu C., Meng X., Xie Y., Yang Y. Effects of surface texturing on ring/liner friction under starved lubrication [J]. Tribology international, 2016, 94: 591-605.
[2] Gu C., Meng X., Xie Y., Zhang D. The influence of surface texturing on the transition of the lubrication regimes between a piston ring and a cylinder liner [J]. International Journal of Engine Research, 2016;
[3] Sudeep U., Tandon N., Pandey R. K. Performance of Lubricated Rolling/Sliding Concentrated Contacts With Surface Textures: A Review [J]. Journal Of Tribology-Transactions Of the Asme, 2015, 137(3): 031501-031501-031511.
[4] Gu C., Meng X., Xie Y., Li P. A study on the tribological behavior of surface texturing on the nonflat piston ring under mixed lubrication [J]. Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, 2016, 230(4): 452-471.
[5] Gropper D., Wang L., Harvey T. J. Hydrodynamic lubrication of textured surfaces: A review of modeling techniques and key findings [J]. Tribology international, 2016, 94: 509-529.
[6] Gu C., Meng X., Zhang D., Xie Y. A transient analysis of the textured journal bearing considering micro and macro cavitation during an engine cycle [J]. Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, 2017, 231(10): 1289-1306.
[7] Akamatsu Y., Tsushima N., Goto T., Hibi K. Influence of surface roughness skewness on rolling contact fatigue life [J]. Tribology Transactions, 1992, 35(4): 745-750.
[8] Zhai X., Chang L., Hoeprich M., Nixon H. On mechanisms of fatigue life enhancement by surface dents in heavily loaded rolling line contacts [J]. Tribology Transactions, 1997, 40(4): 708-714.
[9] Wen S. Z., Huang P. Tribological principle [M]. Tsinghua university press, 2012.
[10] Dowson D., Higginson G. R. Elasto-hydrodynamic lubrication: the fundamentals of roller and gear lubrication [M]. Pergamon Press, 1966.
[11] Roelands C. J. A. Correlational aspects of the viscosity-temperature-pressure relationship of lubricating oils [D]; TU Delft, Delft University of Technology, 1966.
[12] Frølund K., Schramm J., Tian T., Wong V., Hochgreb S. Analysis of the piston ring/liner oil film development during warm-up for an SI-engine [J]. Journal of Engineering for Gas Turbines and Power, 2001, 123(1): 109-116.
[13] Huang P. Numerical calculation methods of elastohydrodynamic lubrication [M]. Tsinghua university press, 2013.
[14] Gachot C., Rosenkranz A., Hsu S. M., Costa H. L. A critical assessment of surface texturing for friction and wear improvement [J]. Wear, 2017: 21–41.
[15] Andersson P., Koskinen J., Varjus S., Gerbig Y., Haefke H., Georgiou S., Zhmud B., Buss W. Microlubrication effect by laser-textured steel surfaces [J]. Wear, 2007, 262(3-4): 369-379.
[16] Wang X. L., Liu W., Zhou F., Zhu D. Preliminary investigation of the effect of dimple size on friction in line contacts [J]. Tribology international, 2009, 42(7): 1118-1123.
[17] Sudeep U., Pandey R. K., Tandon N. Effects of surface texturing on friction and vibration behaviors of sliding lubricated concentrated point contacts under linear reciprocating motion [J]. Tribology international, 2013, 62: 198-207.
[18] Wang X. L., Kato K. Improving the anti-seizure ability of SiC seal in water with RIE texturing [J]. Tribology Letters, 2003, 14(4): 275-280.
[19] Kovalchenko A., Ajayi O., Erdemir A., Fenske G., Etsion I. The effect of laser texturing of steel surfaces and speed-load parameters on the transition of lubrication regime from boundary to hydrodynamic [J]. Tribology Transactions, 2004, 47(2): 299-307.
[20] Podgornik B., Vilhena L. M., Sedlacek M., Rek Z., Zun J. Effectiveness and design of surface texturing for different lubrication regimes [J]. Meccanica, 2012, 47(7): 1613-1622.