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Effects of surface texture on the lubrication performance of the floating ring bearing

Shiyuan Pei [a,*], Hua Xu [a,b], Meng Yun [c], Fanghui Shi [a,d], Jun Hong [a]

[a] Key Laboratory of Education Ministry for Modern Design and Rotor-Bearing System, Xi'an Jiaotong University, Xi'an 710049, China
[b] School of Mechanical Engineering, Xinjiang University, Urumqi 830046, China
[c] New Link College, Vancouver, Canada 1234940
[d] General Motors Company, Warren, MI 48090, USA

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ABSTRACT

To study the effects of the surface texture on the performance of floating ring bearings (FRB), we employ a deterministic lumped-parameter thermal model to estimate the FRB performance. A multiscale method, finite cell method, is used to improve the computational efficiency. The model is first validated through comparing the predicted results with measured data. Then, the effects of nine types of textures on the bearing behaviors are systematically studied. The results show that textures can considerately affect the FRB performance, such as significantly increasing the side leakage and reducing the temperature rise, which is desirable in many cases for the turbocharger development.

1. Introduction

To increase the power output and reduce the CO_2 emission of internal combustion engines, turbochargers have been extensively used in automobile industry recently. To support the rotor during the operation, the bearing system, including the radical and thrust bearings, is an essential element for the turbocharger. There are mainly three kinds of radical bearings used in the automotive turbocharger [1]: full floating ring bearing (FRB), semi-floating ring bearing (SFRB) and rolling element bearing. The last one can reduce much friction at a low rotor speed, but it does not show obvious advantage over the other fluid bearings at a high-speed range, because the temperature rise at the high speed leads to a significant reduction of the oil viscosity and thus the friction force for the film bearing. However, the lousy noise and the high cost as well as the limited lifetime prevent the rolling bearing from being used in many cases. On the other hand, The SFRB exhibits a higher temperature rise and frictional power loss [2] compared with the FRB. Hence, the FRB becomes a preferred choice due to its high reliability and low costs for many manufacturers in massive production.

Many investigators have studied the tribological performances of the FRB for many years, both theoretically and experimentally. Rohde and Ezzat [3] analyzed the behaviors of dynamically loaded floating bearings for automotive applications. They presented the power loss, minimum film thickness, and speed ratio in a parametric manner with parameters of inner and outer film clearances. Mokhtar [4] performed an optimization study of floating bearings and concluded that less power loss could be achieved by the adoption of floating bearings. Trippett and Li [5] found in the experiment that the ring speed ratio decreased with the increasing shaft speed, and they explained that the decrease in the speed ratio was caused by the viscosity decrease of the inner oil film due to its higher temperature than the outer oil film. San Andres et al. [6–8] proposed a lumped-parameter thermal energy balance model for the estimation of the FRB performance. The predicted results matched well with experimental data when the speed was moderate. Later on, San Andres et al. [9] developed a more complex thermo-hydrodynamic model that includes the Reynolds equation with variable oil viscosity and the thermal energy to evaluate the thermal energy transport in a SFRB. The model gives a good overview of the heat flow paths and points out the most relevant thermal effects on the SFRB.

Although extensive investigations of the FRB can be found from a collection of literature, there are still a couple of problems occasionally arising under certain circumstance. Recent study [10] reported that many instances have occurred where the outer surfaces of the FRB have exhibited damage from oil debris resulting in constant tone noise and subsequent warranty claims. The effect of oil debris was investigated on the sub-synchronous NVH, and the predicted frequency of these sub-synchronous forces matched well with measured data. However, resorts have to be found to resolve this problem. In addition, to pursue a higher

* Corresponding author.
E-mail address: peishiyuan@xjtu.edu.cn (S. Pei).

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compression ratio, the automotive turbochargers could operate at very high speeds from 100 k to 400 k rpm. Due to the viscous effects, the temperature rise of the film becomes very high, especially for the turbine side bearing. For instance, San Andres et al. [9] found that the film temperature in the inner film of a SFRB could be up to more than 200 °C under the 240 k rpm operation speed. When it exceeds the certain temperature, oil will be degraded or will even be burnt (about 210 °C for the SAE 5W30 oil [1]), which could cause the total failure of the turbocharger.

It is generally accepted that surface textures, which have attracted a lot of researcher's devotion [11–16] in the past two decades, can capture wear debris as micro-traps. Moreover, the textures can also serve as a micro hydrodynamic bearing to provide additional load capacities. Thus, the surface texture seems to be a good candidate to improve the performance of the FRB. However, to the best of the authors' knowledge, there is no literature investigating the tribological performances of the FRB affected by the surface texture.

In this paper, we will employ a deterministic lumped-parameter thermal model to estimate the lubrication performance of the FRB with nine kinds of textures. A multiscale method, finite cell method, will be used to improve the computational efficiency. The analysis program will be first validated through comprising the predicted results with measured data. Then, the effects of the textures on the bearing characteristics, which include the ring speed, journal and ring eccentricities, power loss, temperature rise and flow rate as a function of the rotor speed, will be systematically studied. Finally, several conclusions will be drawn according to the results.

2. Method

2.1. Governing equation

A thermo-hydrodynamic bearing model based on the procedure described in the past literature [7] is employed. It includes the Reynolds equation for each film and a lumped-parameter thermal energy balance model. Thermal expansion of the journal, the ring and the bearing are also considered. Fig. 1 shows the schematic view of a floating ring. The hydrodynamic pressures, P_i and P_o, generated within the inner and outer thin film are described by the following Reynolds equation:

$$\frac{\partial}{\partial x_i}\left(\frac{h_i^3}{12\mu_i}\frac{\partial p_i}{\partial x_i}\right)+\frac{\partial}{\partial y_i}\left(\frac{h_i^3}{12\mu_i}\frac{\partial p_i}{\partial y_i}\right)=\frac{R_J\omega_J+R_1\omega_R}{2}\frac{\partial h_i}{\partial x_i} \tag{1a}$$

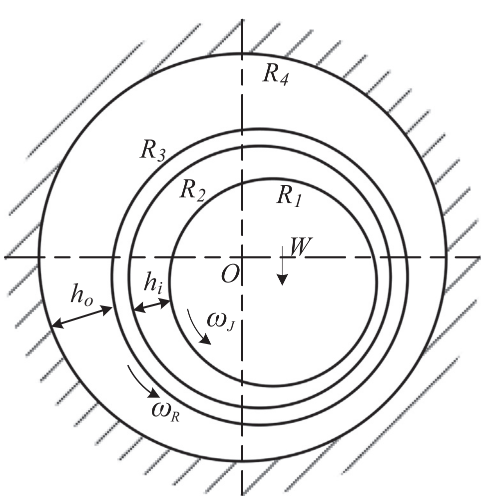

Fig. 1. Schematic view of a floating ring.

$$\frac{\partial}{\partial x_o}\left(\frac{h_o^3}{12\mu_o}\frac{\partial p_o}{\partial x_o}\right)+\frac{\partial}{\partial y_o}\left(\frac{h_o^3}{12\mu_o}\frac{\partial p_o}{\partial y_o}\right)=\frac{R_2\omega_R}{2}\frac{\partial h_o}{\partial x_o} \tag{1b}$$

where $0\le x\le 2\pi R_{i,o}$, $0\le z\le B_{i,o}$. Detailed model formulation please refers to the past literature [7] and their treatments are strictly applied in present study.

2.2. Surface topology

To account for the geometrical dimensions of the textures, the film thickness $h_{i,o}$ in Eq. (1) and (2) at a specific point within the domain consists of two parts: one function describes the macro geometry of the bearing h_i and the other function describes micro geometry of the surface texture h_2. Mathematically, the following representation of the two parts (h_i and h_2 of the film thickness for inner and outer film are used:

$$h_{i1}=C_i(1-e_{xJ}\cos\theta-e_{yJ}\sin\theta) \tag{2a}$$

$$h_{o1}=C_o(1-e_{xJ}\cos\theta-e_{yJ}\sin\theta) \tag{2b}$$

where

$$h_{i2}=\frac{C_i}{a}\cos(N_x\theta)\cos\left(2\pi N_y\frac{z}{B_i}\right) \tag{3a}$$

$$h_{o2}=\frac{C_o}{a}\cos(N_x\theta)\cos\left(2\pi N_y\frac{z}{B_o}\right) \tag{3b}$$

$$C_i=C_{i0}-\delta_j+\delta_{ri} \tag{4a}$$

$$C_o=C_{o0}-\delta_b+\delta_{ro} \tag{4b}$$

where

$$C_i=C_{i0}-\delta_j+\delta_{ri} \tag{5a}$$

$$C_o=C_{o0}-\delta_b+\delta_{ro} \tag{5b}$$

where subscript *i,o* denote for inner and outer film; θ is the circumferential coordinate; *z* is the axial coordinate; *B* is the width of the film; e_j and e_R are the journal and ring center vector displacements; *C* is the operating radical clearance, which is equal to the nominal bearing clearance plus the change in the clearance due to thermal expansion; δ is the radial thermal expansion. Parameter *a* denotes for the texture amplitude parameter and is kept constant for all cases as 2, which implies that the amplitude of textures is equal to half of the corresponding inner and outer nominal clearance at the room temperature in all cases. Nine kinds of textures (transverse, isotropic and longitude pattern with corresponding concave, concave-convex and convex profile) are numerically generated using the parameters listed in Table 1 and are visualized in Fig. 2.

Table 1
Texture generation parameters.

Type	Texture number		Texture thickness modification
	Circumferential N_x	Axial N_y	
Smooth	0	0	–
A	32	0	When $h_2>0$, $h_2=0$
B	32	0	–
C	32	0	When $h_2<0$, $h_2=0$
D	32	8	When $h_2>0$, $h_2=0$
E	32	8	–
F	32	8	When $h_2<0$, $h_2=0$
G	0	8	When $h_2>0$, $h_2=0$
H	0	8	–
I	0	8	When $h_2<0$, $h_2=0$