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Effect of compound dimple on tribological performances of journal bearing

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ABSTRACT

The effect of the compound dimple on the tribological performances of a journal bearing is studied. In doing so, the tribological performances for the bearing with the compound dimple and simple dimple are studied using a fluid structure interaction (FSI) method and compared further. Numerical results show that the compound dimple can supply the larger load-carrying capacity and lower friction coefficient due to its twice hydrodynamic action in comparison with the simple dimple. Moreover, the above improvement depends on the geometry sizes of the compound dimple, the dimple interval, and working parameters of the bearing.

1. Introduction

With increasing demand for reducing energy consumption, the surface texturing is attracting intensive attentions of tribological researchers due to its role in reducing friction. The associated efforts are accelerated due to the work by Etsion et al. in which surface texturing was applied successfully to mechanical seals [1]. The simple research works conducted experimentally or numerically in the past year are involved in effects of the geometry size, distribution and shape of dimple on tribological performances for the tribological pairs. Among the dimple shapes used, spherical and rectangular dimples are often employed due to their good behaviors in improving tribological performances of the tribological pairs. Typically, Galda et al. and Zum Gahr et al. found that a spherical dimple with an appropriate size and distribution shows a better lubrication behavior [2,3]. The CFD result by Han et al. showed that the load-carrying capacity of lubricant increases but its friction coefficient decreases for the parallel sliding surface with the spherical dimples [4]. Owing to the wide application of a journal or thrust bearing at various rotary machineries, the surface texturing is also used for improving tribological performances for this kind of bearings. Etsion et al. pointed out that the friction coefficient of the parallel-thrust bearing partially with spherical dimples is reduced by more than 50% in comparison with that of the untextured bearing [5]. Through a series of tests, Lu et al. found an apparent reduction in the friction of a journal bearing due to the coupling effect of the spherical and cylindrical dimples located on the bearing inner surface [6]. The first author of the present work found that the cavitation area of a journal bearing with spherical dimples decreases in comparison with that of the untextured bearing [7]. The numerical result from Fillon et al. showed that the maximum film pressure of the textured journal bearing can be improved through spherical dimples with an appropriate geometry size and distribution [8]. Galda et al. found that the wear amount of a steel ring fabricated with spherical dimples is smaller than that of one having dimples of long drop shape [9].

Beside the spherical dimple research works, effects of a rectangular dimple on performances for the tribological pairs were also studied in the past years. Gherca et al. found that among the rectangular, triangular and parabolic dimples, the rectangular one has a high efficiency in providing the load-carrying capacity of the lubricant between parallel sliding surfaces [10]. Experimental results from Meng et al. indicated that the friction between parallel sliding surfaces with rectangular dimples decreases due to the dimple effect [11]. Ling et al. made an adhesion experiment of a drill with rectangular dimples [12]. Their result showed that the adhesion decreases because of the dimple effect and the drill's lifetime is lengthened, when these dimples were distributed in an overlapped triangular way. Wang et al. obtained the optimal ratio of a rectangular dimple's depth to the clearance between parallel textured surfaces, with which the best lubrication performance for tribological pairs can be obtained [13]. Kango et al. found that the friction coefficient of a journal bearing with the appropriately

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distributed rectangular dimples can be reduced compared with that of the untextured bearing [14]. Additionally, Shen et al. pointed out that a cylindrical dimple with a rectangular cross section can result in a large load-carrying capacity of the lubricant for a thrust bearing [15].

So far, the dimple shapes employed in the research works above are basically simple, mainly either spherical, rectangular, or cylindrical etc. However, the improvement in the tribological performances by the simple dimples is usually limited.

As an extended exploration of the simple dimple used in the past research works, the compound structure characterized by two-layer pores is used for textured surfaces, which is termed as the compound dimple in the present study. For the said compound dimple, the pore shape in the first and second layers is rectangular and spherical, respectively. This shape layout is due to the consideration of the combining merits of the above two kinds of dimples, as stated previously. The compound dimple can be manufactured through the laser texturing based on the simple dimple obtained with the same method. Then, take a water-lubricated journal bearing for example, the compound dimple's influences on its tribological performances are then investigated with a fluid–solid interaction (FSI) method. Choosing this kind of bearings is due to the consideration of its limited carrying-capacity because of the low water viscosity, although it is characterized by efficiency and energy conservation and pollution-free. Through the FSI method, the deformation of the bearing inner surface can be incorporated into the film thickness in solving the film pressure, and the obtained film pressure is in turn considered in solving the bearing deformation, which is a two-way coupling process. In the investigation, the bearing models with the compound and simple dimple shapes (i.e., rectangular dimple) are established. Based on these models, the hydrodynamic lubrication actions of the compound and simple dimples are solved with the Navier–Strokes equation since it can overcome limitations of the Reynolds equation in predicting lubrication performances for tribological pairs at conditions with obvious lubrication film inertia forces or small film thickness ratios [16,17]. Meanwhile, the bearing's mechanical performances such as the deformation and stress are solved with constitutive equations of solid. Next, the tribological performances for the journal bearing with the compound and simple dimples are compared. Finally, the associated conclusions are drawn.

2. Governing equations

When the bearing operates, the cavitation of lubricant often occurs. In the process of the cavitation, the cavitation is often accompanied by the growth of gas bubbles. The variation in the bubble radius can be approximately described with the Rayleigh–Plesset equation, that is,

$$\rho_l\left[\frac{3}{2}\dot{R}_v^2+R_v\ddot{R}_v\right]=p_v-p-\frac{2\sigma}{R_v}-\frac{4\mu_l}{R_v}\dot{R}_v \tag{1}$$

where ρ_l and μ_l are the density and viscosity of the liquid lubricant, and σ is the surface tension of the lubricant. The symbols p and p_v are the ambient pressure of the bubble and the pressure of the vapor in the lubricant. For the bubble with the radius R_v, whose radius variation per unit time is denoted by $\dot{R}_v$ and whose acceleration is denoted by $\ddot{R}_v$.

When the lubricant is partly vapored, the relative amount of the volume fraction of vapored and liquid lubricant, denoted by α_v and α_l, meets the following relation:

$$\alpha_l+\alpha_v=1 \tag{2}$$

where α_v is obtained with the following expression:

$$\alpha_v=\frac{\alpha_a}{\alpha_a+(1-\alpha_a)R_a^3/R_v^3} \tag{3}$$

Here, the gas nuclear radius R_a is taken to be 1 μm and the volume fraction of nucleation sites α_a is set as 5.0×10^{-4}.

In the simulation, the liquid and vapored lubricants are assumed to have common velocity and temperature. As a usual dealing, the liquid lubricant is looked as incompressible Newtonian fluid without body force and its flow is laminar and steady-state. Thus, the ambient pressure of the gas bubble in Eq. (1) can be determined by solving the following continuity equation of the lubricant and Navier–strokes equation simultaneously, whose expressions are, respectively, given as follows:

$$\frac{\partial}{\partial x_i}(\rho u_i)=0 \tag{4}$$

$$\frac{\partial}{\partial x_i}(\rho_l u_i u_j)=-\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu_l\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\right]+\frac{\partial}{\partial x_j}(-\rho\overline{u_i'u_j'}) \tag{5}$$

where u_i ($i=1, 2, 3$) represents the velocity of the liquid lubricant in the coordinate directions x_j($j=1, 2, 3$), that is, x-, y- and z-directions shown in Fig. 1(a). The Reynolds stress $-\rho\overline{u_i'u_j'}$ is solved by the standard $k-\varepsilon$ model [18].

Once the film pressure is obtained through Eq. (5), the composite load-capacity of lubrication film on the bearing surface A can be computed as:

$$W_f=\iint_A prd\varphi dz \tag{6}$$

where φ is the circumferential angle from the maximum film thicknesses.

The frictional force acting on the shaft due to the viscosity shear force τ of lubricant can be expressed as:

$$F_f=-\iint_A \tau rd\varphi dz \tag{7}$$

Once the load-carrying capacity and frictional force are obtained, the friction coefficient can be computed according to the following relation:

$$f=\frac{F_f}{W_f} \tag{8}$$

In the solution process, the finite volume method is employed to solve the lubrication performances for the lubricant through the CFX module of the software ANSYS 14.5 version, while the bearing's mechanical performances are solved with the finite element method employed through the static structure module of this version.

In addition, the exchange of data with the said two dealings is executed in corresponding iteration stagger of this software.

3. Simulation model

A three-dimensional bearing model with ten compound dimples is modeled, whose three dimensional schematic and geometry sizes are shown in Fig. 1. In Fig. 1(b), the cylindrical shaft with radius r rotates at a rotational speed n against the bearing bushing with inner radius R_1. The elastic modulus and Poisson ratio for the bearing with density of 7850 kg/m^3 are 200 GPa and 0.3, respectively. When the load is applied vertically downward at the geometry center of the shaft O_2, its eccentricity occurs. The eccentricity distance between the point O_2 and geometry center for the bearing O_1 is denoted by e. The attitude angle of the shaft is denoted by α, which starts from the positive y-direction (i.e., load direction). The clockwise circumferential angle φ showing the change in the film thickness starts from the maximum film thickness h_{max}. In the present study, the compound dimple shown in Fig. 1(c) is introduced, whose first and second layers