Study on optimization of the circumferential and axial waviness geometrical configuration of hydrodynamic journal bearing

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Abstract
This paper is focused on using GA genetic algorithm to find the optimal performance with respect to shape optimization in three dimensions for the hydrodynamic journal bearing. The mathematical model for film thickness was drawn using Fourier series function and axial waviness value ($\Delta$) to represent the journal bearing in circumferential and axial direction, respectively. The objective was then to determine the Fourier coefficients and axial waviness value ($\Delta$) that maximized the load capacity subjected to a given set of constraint. Optimized results show that the presence of cos wave in axial direction, with a positive dimensionless amplitude (+A) and waviness number $m = 0.633$, improves the load capacity by (8-10) % over the cylindrical plain bearing with the same arbitrary shape and size; in general, the increasing order of Fourier series ($n$), an axial dimensionless amplitude and $L/D$ ratio cause the change in load capacity to become more evident.

Keywords: Axial geometrical configurations; Axial waviness number; Shape optimization; General film thickness; Genetic algorithm

1. Introduction
During the last 30 years various theoretical and experimental investigations have been conducted regarding unconventional journal bearing configuration behavior in the case of hydrodynamic lubrication. The bearing performance is influenced by a number of parameters, such as the bearing geometry, length to diameter ratio, the viscosity of the working fluid, and the surface velocities. The profile of the bearing surfaces plays an important role in the performance since this determines the gap geometry and hence the pressure generated. A circular shape is the simplest profile for hydrodynamic journal bearings and is the most widely used design. However, with improved understanding of hydrodynamic theory several new bearing shapes have been proposed including elliptical bearings, dislocated cylindrical/dislocated elliptical bearings, and multiple-lobe. For example, Leung et al. [1] first examined a bearing having a spherical shape, but their treatment showed that its behavior was similar to that of a comparable cylindrical bearing. On the other hand, the results obtained by El-Gamal [2] in his analysis for a symmetrical wedge shaped bearing, showed that for small wedge angle the bearing was found superior to the plain cylindrical one. El-Gamal [3] extended the analysis to include bearings of different axial curvilinear shapes; convex, concave and wavy geometrical configurations were examined, and improved load capacity was found for the shapes considered over the plain cylindrical bearing. His results also showed that the concave geometry is superior to the others for long bearings, while the wavy shape is the best for short bearings. Lin and Hamrock [4] used a non-Newtonian fluid to study the effect of surface irregularity on lubrication in line contacts. Lin [5] also studied the hydrodynamic lubrication of journal bearings with three-dimensional asperities machined in the liners. He found that when the asperty height is increasing the load capacity increases, the side leakage flow increases and friction variable decreases.

The starting point for most bearing shape development is still limited to classical geometrical bearing shapes. In this work a method that starts to find the best wavy geometrical configuration in axial direction in case the circumferential shape is arbitrary. Shape optimization avoids the necessity of a trial and error procedure for selecting bearing profiles. Improved designs are found through an optimization process using GA subject to a given set of constraints. GA has recently become a popular tool to solve such problems as they are not driven by gradient search processes. Some researchers used shape optimization of hydrodynamic journal bearings as the study point; for example, Song, et al. [6, 7], studied optimal designs of short cylindrical journal bearings by enhancing an artificial life optimization algorithm taking the radial clearance,
the length to diameter ratio, and the average viscosity as design variables. The concept of using a Fourier series to model lubricant film shape is not new. Lebeck [8] and Matsuda, et al. [9, 10] used Fourier series to describe the film thickness around the circumference of a mechanical face seal. In Xiaoping Pang, et al. [11], a Fourier series function was used to represent the general film geometry of the bearing in circumferential direction; their results showed that the load capacity increases by increasing the order of Fourier series. The maximal load capacity of the experimental bearings of the order 2 and 3 and the optimized result is almost fully achieved at a profile of order \( n = 2 \). Hashimoto and Matsumoto [12] optimized an elliptical journal bearing using a hybrid optimization technique combined with a direct search method and sequential quadratic programming. The design variables were the vertical and horizontal clearance, the slenderness ratio, and the orientation angle.

A more thorough approach starts with a generalized bearing shape. Clearly, the shape produced must be capable of being manufactured. Fortunately, the manufacturing capability for arbitrary geometries has been advanced substantially by Swanson [13].

In 2003, Boedo and Eshkabilov [14] studied arbitrary bearing shapes and presented results for shape optimization to achieve an optimum profile that maximizes load capacity. They used discrete points on the bearing bush profile as design variables and a genetic search algorithm. However, their method requires some method to fit a smooth curve to these design points, and indeed some non-smooth points were observed in the optimized curve. Asimow [15] applied the Newton-Raphson method to determine the optimal bearing length and diameter that minimized an objective function based on friction loss and shaft twist. Dowson [16] and EL-Sherbiny et al. [17] carried out an optimal computerized bearing design by taking the radial clearance, the slenderness ratio, and the eccentricity as design variables, using minimum power loss as the design objective. Both Asimow [15] and Dowson [16] used a cylindrical journal bearing as the design basis.

All the prior works are based on optimizing the circumferential or axial bearing shape. Xiaoping Pang [11] and Boedo and Eshkabilov [14] optimized the bearing shape in circumferential direction only, while Hassain El Gamal [18] optimized the model for axial configuration only. In this article, an attempt to find the optimum shape in both directions, a new dimensional generalized film thickness profile is proposed and used as the basis for shape optimization in both directions. The maximum load capacity is taken as the design objective function, Fourier series coefficients and axial waviness value of the general film thickness profile are taken as design variables. An efficient shape optimization model is then formulated and an optimized result is obtained using the PDE and GA toolboxes in MATLAB. The optimized results are compared with those in previous published work.

Fig. 1. Journal bearing geometrical configuration and the coordinate system.

2. Three dimensional journal bearing mathematical model

The journal bearing geometry configuration and the coordinate system are given in Fig. 1. The first step is to build the thin film flow governing equation using the Reynolds equation. The bearing is assumed to be steadily loaded, and the flow is Newtonian, laminar, incompressible, and transient effects (i.e., squeeze-film action) are neglected. Under these conditions, a general form of the Reynolds equation reduces to the following form:

\[
\frac{\partial}{\partial x} \left( \frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( \frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = 6U \frac{\partial h}{\partial x}
\]  

(1)

where, \( p \) is the local film pressure, \( \mu \) is the lubricant viscosity and \( U \) denotes the fluid velocity at the journal surface. By using,

\[
\bar{p} = \frac{p}{\mu \omega r / h_{\text{min}}} \quad \lambda = \frac{x}{L/2} \quad x = r \theta \quad \bar{h} = h / h_{\text{min}}. 
\]  

(2)

The dimensionless Reynolds equation can be derived as:

\[
\frac{\partial}{\partial \theta} \left( \bar{R}_{\lambda} \frac{\partial \bar{p}}{\partial \theta} \right) + \frac{1}{(L/D) \lambda} \frac{\partial}{\partial \lambda} \left( \bar{R}_{\lambda} \frac{\partial \bar{p}}{\partial \lambda} \right) = 0 \frac{\partial \bar{h}}{\partial \theta}
\]  

(3)

where, \( L \) and \( D \) are the length and diameter of the bearing, respectively.

Subject to the boundary conditions: