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Mixed-lubrication analysis of marine stern tube bearing considering bending deformation of stern shaft and cavitation

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ABSTRACT

A mixed-lubrication model has been developed to study tribological characteristics of a marine stern tube bearing. The finite difference method is employed to obtain numerical solutions of the Reynolds equation using combined Newton–Raphson and iterative relaxation methods. The differential equation of deflection curve of the propeller shaft is used to describe the bending deformation of propeller shaft. The displacement superposition method is used to solve the differential equation of deflection. The influences of cavitation and different boundary conditions, the Reynolds boundary condition and Jacobsson–Floberg and Olsson (JFO) boundary condition, as well as the bending deformation of shaft, on the Stribeck Curves are discussed in this paper.

1. Introduction

Costs associated with stern tube bearing problems are high because it forces a ship out of service. The risk of uneven contact and seizure of stern tube bearing increases if misalignment of a propeller shaft exists. Stern tube bearings are exposed to different operating conditions in terms of load, speed and temperature. The conditions range from pure hydrodynamic when the lubricating film is sufficiently thick to severe asperity contact when the lubricating film is too thin between the lubricated surfaces. Oh and Goenka [1–3] developed a method to solve for point contact problem for dynamically loaded rigid journal bearings to a dynamically loaded flexible connecting rod bearing. Vincent [4] proposed a numerical procedure, which incorporated an algorithm of cavitation based on the work of Elrod [5], to analyze the cavitation in dynamically loaded journal bearings using mobility method. The mobility method developed by Booker [6–7] is widely used for analyzing rigid journal bearings due to its computational efficiency when compared to others approaches, such as the multi-grid techniques based on the Elrod algorithm and the finite element methods of analysis. O'Donoghue [8] performed an experimental and theoretical study of a finite journal bearing, and in their work, the finite difference method (FDM) was employed to solve the two-dimensional Reynolds equation, and the Reynolds boundary condition was used to deal with cavition. All of the above studies were based on the assumption of the aligned journal. Sun and Gui indicated that the shaft became deformed when acted by a force, which resulted in the misalignment of the journal in bearing hole, thus the lubrication state of journal bearing may be influenced. And in their study, in order to describe the status of misalignment, the film thickness of a misaligned bearing was given by misalignment parameters (β and γ). Vijayaraghavan and Keith [10] analyzed a finite grooved misaligned journal bearing considering cavitation and starvation effects. Bouyer and Fillon [11] conducted an experimental study of misaligned plain journal bearing, which demonstrated that misalignment has significant role when the rotational speed or load is low. Wang [12] presents a mixed-EHL analysis of effects of elastic deformation, as well as axial and twisting misalignments on the performance of a coupled journal thrust bearing system. However, all above studies gave no attention to the interaction between the lubrication properties and deformation of journal. Under operating conditions, the load acted on journal causes the deformation of journal, and this deformation will affect the lubrication pressure distribution and asperity contact forces in mixed lubrication analysis; meanwhile, the lubrication pressure and asperity contact also act on the journal to influence the deformation of journal. Moreover, when the width of bearing is large, such as the marine stern tube bearing, the deformation of shaft in the bearing housing cannot be described by only two angles (β and γ). The shaft in the bearing housing is under the concentrated load of propeller screw in addition to the lubrication pressure and asperity forces, and the effect of bending deformation of propeller shaft.

To characterize the range of different operating conditions, the Stribeck curve has been used to identify the critical journal velocity at which the transition from hydrodynamic lubrication to mixed lubrication takes place or at which a certain acceptable coefficient of friction is exceeded [13]. de Kraker [14] calculated

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the Stribeck curves for the water lubricated journal bearings to investigate the effect of elastic behavior of the bearing without considering the influence of boundary and deformation of propeller shaft. In this paper, a computational method for solving the mixed-lubrication problem with JFO's mass conserving cavitation conditions [15,16] is presented. The influences of different boundary conditions (the Reynolds boundary condition and JFO boundary condition), as well as the deformation of propeller shaft, on the Stribeck curves are also discussed.

2. Model

The mixed lubrication model of marine stern tube bearing covers all regimes of lubrication from hydrodynamic to boundary lubrications, which includes following submodels:

(a) Mixed lubrication model. In many instances of hydrodynamic lubrication (HL), direct contact between the asperities will still occur in spite of the presence of HL film. It is suggested that hydrodynamic theories used to describe the lubrication of bearings should include surface roughness effects.
(b) Asperity contact model. If the lubricating film separating the surfaces is such that it allows some contact between the asperities. In the contact areas between bearing and shaft, the asperity contact pressure carries part of the load on the bearing, and the hydrodynamic lubrication pressure carries the remainder of the load.
(c) Bending deformation model of shaft. The forces (including the external force, the film pressure and asperity contact forces) acting on the propeller shaft causes the deformation of the shaft; meanwhile, the deformation of shaft influences the distribution of film pressure and contact forces of the hydrodynamic lubrication.

2.1. Journal bearing geometry

When the shaft and bearing have relative rotational velocities with respect to each other, the amount of eccentricity adjusts itself until the film pressure and the asperity contact pressure is balanced the external loads. Fig. 1 depicts the basic geometric of a journal bearing. The film pressure and the contact pressure depend on the journal eccentricity, bending deformation of the journal, the relative angular velocity, the viscosity of the fluid lubricant, and the journal bearing geometry and clearance. Usually, the effects of hydrodynamic pressures are looked as contribution of two different actions: wedge and squeezing. The squeeze action relates the radial journal motion with the generation of load carrying capacity in the lubricant film, whilst the wedge action deals with the relationship of the relative rotational velocity of the journal and bearing ability to produce such pressure.

The film thickness of a misaligned bearing was given by misalignment parameters (β and γ) [9] as

$$h_\theta = c + e_0 \cos(\theta - \varphi) + z \tan\gamma \cos(\theta - \beta - \varphi) \tag{1}$$

where c is radial clearance of bearing; z is bearing width direction; θ is the circumferential direction of bearing; φ is the attitude angle, i.e., an angle between the applied load direction and the center line of the journal and the bearing.

The deflect angle in each section is different for bearings with high value of width, and the film thickness of the journal bearing considering the deformation of shaft can be expressed as

$$h_{i\theta} = c + e_i \cos(\theta - \phi) \tag{2}$$

where, $e_i = \sqrt{(\upsilon_{ix} + e_{ix})^2 + (\upsilon_{iy} + e_{iy})^2}$, e_i is the eccentricity of the ith section with the bending deformation of shaft, e_{ix} and e_{iy} are the components of vector eccentricity without bending in x and y directions (calculated by the displacement superposition method, described in Section 2.5).

For a given external loading condition, the static equilibrium position of the journal center can be found by equating the hydrodynamic forces and asperities contact forces with the externally applied load. A two-dimensional Newton–Raphson search technique is applied for calculation of the equilibrium position of the shaft center. The equilibrium equation for the hydrodynamic force, contact force and external load $\vec{P}$ is shown as

$$\vec{P} + \vec{F}_{oil} + \vec{W}_{asp} = 0 \tag{3}$$

where the hydrodynamic force $\vec{F}_{oil} = \int_0^L \int_0^{2\pi} p d\theta dL$ (film pressure p is determined by mixed lubrication model, described in Section 2.2)

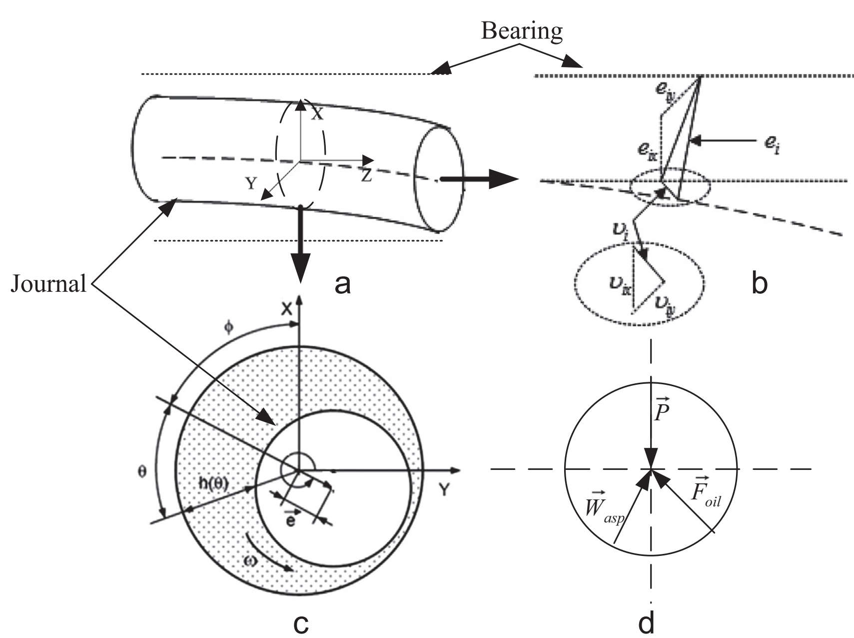

Fig. 1. Basic geometry of journal bearing.