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"Research Note"

STUDY OF LUBRICANT COMPRESSIBILITY EFFECT ON HYDRODYNAMIC CHARACTERISTICS OF HEAVILY LOADED JOURNAL BEARINGS*

S. A. GANDJALIKHAN NASSAB, ** H. SOHI AND E. H. ZAIM

Dept. of Mechanical Engineering, School of Engineering, Shahid Bahonar University, Kerman, I. R. of Iran
Email: Ganj110@mail.uk.ac.ir

Abstract– By means of numerical techniques, the effect of lubricant compressibility on hydrodynamic behavior of heavily loaded journal bearings is studied in the present work. To reach this goal, the set of continuity and momentum equations for compressible two-dimensional lubricant flow in journal bearings are solved numerically by CFD method. The journal bearing under consideration has infinite length and the lubricant flow is assumed to be laminar and isoviscous. Although the lubricant is liquid, in the cases of high bearing loads, the variation of density with pressure may be important. Therefore, in the computations, the lubricant density is considered variable as a function of pressure. Considering the complex geometry in the physical domain, an attempt is made to transfer the set of governing equations into computational plane by means of conformal mapping. The transformed forms of equations are discretized by the control volume method and are solved using the SIMPLE Algorithm. Results show that the compressibility effect causes an increase in the generated hydrodynamic pressure, such that this effect is enhanced under the condition of high shaft rotational speed, small clearance and high eccentricity ratio.

Keywords– Compressibility effect, journal bearings, hydrodynamics, CFD

1. INTRODUCTION

Journal bearings designed for heavy-duty machinery usually work under sever operating conditions. In the cases of high bearing load, a great value of hydrodynamic pressure in the order of GPa may be produced in the bearing pressure zone. Under these conditions, the liquid oil experiences variable density, such that for higher values of hydrodynamic pressure, the lubricant compressibility may be important and must be considered in the computations. There are many researches in which the hydrodynamic characteristics of journal bearings have been analyzed by solving the Reynolds lubrication equation [1-2]. In those works, it is commonly assumed that the lubricant flow is incompressible. During the last two decades, there have been a few studies in which the hydrodynamic or THD characteristics of journal bearings were investigated by solving the Navier-Stockes and energy equation with CFD techniques. For the first time, Tucker and Keogh [3] obtained solutions to a set of exact governing equations for both cases of stationary and orbiting journal center. The governing equations including continuity, momentum and energy were written in a generalized cylindrical coordinate and discretized by the finite volume method. Difference equations were solved numerically using the SIMPLE Algorithm for pressure calculation. The results were presented in the form of a design chart for predicting the THD behavior of journal bearings. Besides, there are a few papers by Gandjalikhan Nassab [4-5] about hydrodynamic and THD analysis of journal bearings by CFD method. In those works, instead of using generalized cylindrical coordinate system [3] which cannot match the boundaries and apparently introduces an approximation in formulation, conformal

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**Corresponding author

mapping was used to generate an orthogonal grid. As a result, the governing equations were transformed in the computational domain. Transformed equations were discretized by the control volume method and solved by the SIMPLE Algorithm.

In all of the above works, the flow of liquid oil as the lubricant was considered incompressible. It is expected that in cases having a high bearing load in which a great value of lubricant hydrodynamic pressure is generated, the fluid compressibility may be important and should be considered for reliable performance prediction of journal bearings. To the best of the authors' knowledge, the compressibility effect of oil flow in journal bearings has not been studied by CFD method. Furthermore, all of the studies about compressibility effects are restricted to the gas lubricated bearings. Therefore, the present paper deals with the hydrodynamic analysis of journal bearings by solving the continuity and momentum equations using CFD method including lubricant compressibility. In this manner, the oil density is considered as a function of pressure based on an appropriate relation. Conformal mapping is used to generate an orthogonal grid and the governing equations are transformed for use in the computational domain. Discretized forms of the transformed equations were obtained by the control volume method and solved using the SIMPLE Algorithm. By this technique, the effect of lubricant compressibility on hydrodynamic behavior of journal bearings is fully explored.

2. THEORETICAL ANALYSIS

The governing equations which are written for isoviscous two-dimensional, steady, compressible and laminar flow, consist of the conservations of mass and momentum. The non-dimensional forms of these equations in the Cartesian coordinate system, Fig. 1, can be written as

$$\frac{\partial \rho^* u^*}{\partial x^*} + \frac{\partial \rho^* v^*}{\partial y^*} = 0 \tag{1}$$

$$\frac{\partial}{\partial x^*}\left(u^{*2} - \frac{1}{\rho^* \mathrm{Re}}\frac{\partial u^*}{\partial x^*}\right) + \frac{\partial}{\partial y^*}\left(u^* v^* - \frac{1}{\rho^* \mathrm{Re}}\frac{\partial u^*}{\partial y^*}\right) = -\frac{\partial p^*}{\partial x^*} \tag{2}$$

$$\frac{\partial}{\partial x^*}\left(u^* v^* - \frac{1}{\rho^* \mathrm{Re}}\frac{\partial v^*}{\partial x^*}\right) + \frac{\partial}{\partial y^*}\left(v^{*2} - \frac{1}{\rho^* \mathrm{Re}}\frac{\partial v^*}{\partial y^*}\right) = -\frac{\partial p^*}{\partial y^*} \tag{3}$$

The dimensionless variables are defined as:

$$x^* = \frac{x}{c},\ y^* = \frac{y}{c},\ u^* = \frac{u}{\overline{V}},\ v^* = \frac{v}{\overline{V}},\ \rho^* = \frac{\rho}{\rho_0},\ p^* = \frac{p}{\rho_0 \overline{V}^2},\ \mathrm{Re} = \frac{\rho_0 \overline{V} c}{\mu}$$

In Eqs. (1) to (3) and also in Fig. 1, ρ, μ, p are the lubricant density, viscosity and pressure, respectively, $\bar{V}$ the shaft linear velocity, ω the shaft angular speed, e and c the bearing eccentricity and clearance, respectively, r_s the shaft radius, r_i the inner radius of the bush, (u,v) the lubricant velocity components, θ the angle in the direction of rotation, θ_l the half angle of groove span , and (x,y) and (ξ, η) are the coordinate systems in the physical and computational domains, respectively.

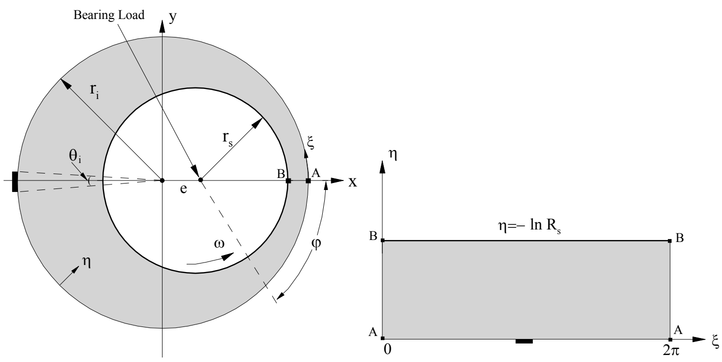
Because of the high bearing load and great value of lubricant hydrodynamic pressure, the oil density is considered as a variable property. The lubricant density as a function of pressure is obtained from the Dowson and Higginson [6] relation:

$$\rho = \rho_0 [1 + \frac{0.6p}{1 + 1.7p}] \tag{4}$$

In this expression, which is valid for mineral oils, ρ_0 is the lubricant density at atmospheric (zero gauge) pressure and p is the lubricant pressure in terms of GPa. It is clear that in the cases in which the lubricant flow is assumed incompressible, the value of ρ^* is equal to unity.

3. SOLUTION PROCEDURE

Due to the complexity of the geometry in the (x,y) plane, the physical domain is mapped into a rectangular (ξ,η) plane. An example of the physical and computational domains is shown in Fig. 1. The details of the relation between physical and computational coordinate systems along with the transformation functions are given in the previous works by the first author [4-5]. Finite difference forms of the partial differential equations were obtained by integrating over an elemental cell volume with staggered control volumes for the ξ- and η-velocity components. Other variables of interest were computed at the grid nodes. The discretized governing equations were numerically solved by the SIMPLE Algorithm of Patankar and Spalding [7]. Numerical solutions were obtained iteratively by the line-by-line method. The iterations were terminated when the sum of the absolute residuals was less than 10^{-4} for each equation. The details of the numerical approach are given in the previous paper by the first author [5].



(a) Physical plane (b) Computational plane

Fig. 1. Schematic of the physical and computational domains

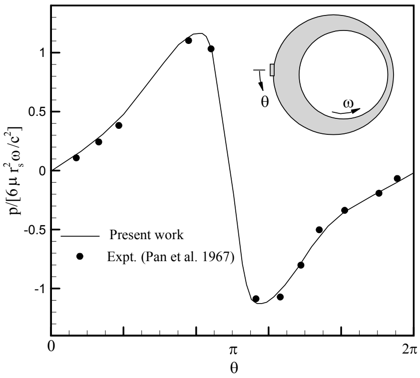
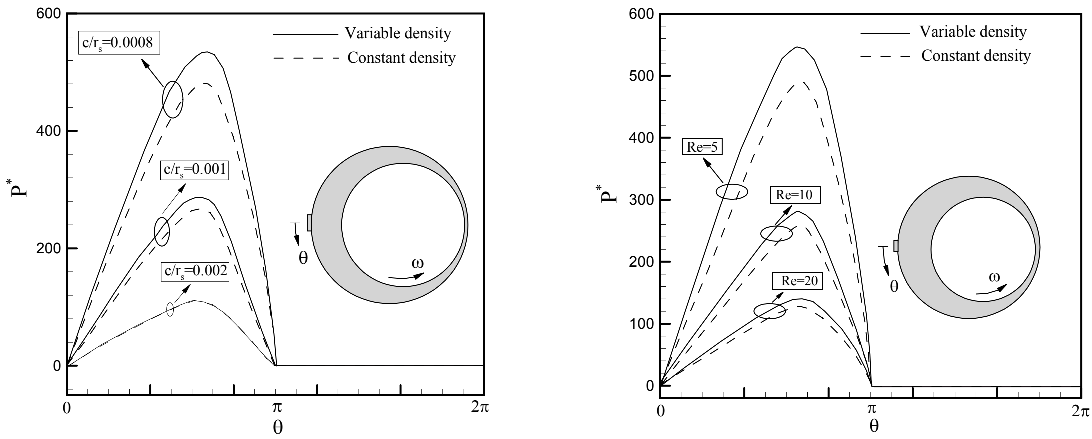


Fig. 2. Circumferential pressure distribution on the shaft surface, $Re=134, c/r_s=0.01, \varepsilon=0.68$

During iterative solution, whenever the pressure falls below the value of cavitation pressure at a grid point (ξ_i, η_j), the entire section, $(\xi = \xi_i)$, passing through that point is considered to be located in the cavitated region. In this way, the approximate locations of boundaries of the cavitated region at each iteration can be determined. Based on the experimental observations by Heshmat [8], the lubricant pressure in the cavitated zone remains constant equal to the oil vapor pressure. It should be mentioned that at normal temperatures, the lubricant vapor pressure is close to absolute zero pressure and therefore the same value is used in the present computations. Numerical calculations were performed by writing a computer program in FORTRAN. As for the grid test to obtain grid-independent solutions, an optimum grid of 51×21, with clustering near the journal and bush surfaces, was used for the flow field calculations.

4. NUMERICAL RESULTS

First, to confirm the numerical findings, the test bearing of Ref. [1] was simulated and the results were compared with experimental data in Fig. 2. It should be noted that this test case with a very large clearance ratio ($c/r_s = 0.01$), which is not a typical operating condition, is selected because under this condition, the viscous dissipation can be neglected, such that the isoviscous assumption for lubricant flow employed in the present computations becomes more accurate. As seen in Fig. 2, the general agreement of the present results with the experimental data is quite good and the values of minimum and maximum pressures and their predicted locations are reasonably close to those of the measurements. As mentioned before, the main goal of the present study is to investigate the effect of lubricant compressibility on the hydrodynamic behavior of a journal bearing. To address this goal, several test cases are analyzed, such that the bearings which are under the study have an axial groove located on the line passing through the centers at the maximum radial gap. The pressure distributions on the shaft surface with and without considering the compressibility effect are shown in Fig. 3a at three different values of the clearance ratio. It is seen that the compressibility effect causes an increase in the generated hydrodynamic pressure, especially at small clearance ratio. Such that for c/r_s=0.0008, there is about a 10% increase in the value of maximum pressure because of the flow compressibility. It should be noted that, based on the density-pressure relation (Eq. (4)), density variation becomes considerable under high values of the lubricant pressure in the order of GPa. Therefore, for heavily loaded journal bearings with high values of hydrodynamic pressure, the compressibility effect may be important. From Fig. 3a, for c/r_s=0.0008, one can compute the value of maximum pressure $p_{\max} = 520 \times \rho \overline{V}^2 \approx 0.3\,GPa$, causing lubricant compressibility to become important. Also, it can be found from Fig. 3a that the compressibility effect decreases by increasing in clearance ratio, such that for c/r_s=0.002, the lubricant pressure distribution is not affected by lubricant compressibility.



(a) $\mathrm{Re}=10,\ e/c=0.4$ (b) $c/r_s=0.001,\ e/c=0.4$

Fig. 3. Effect of lubricant compressibility on hydrodynamic pressure

In Fig. 3b, the pressure distributions on the shaft surface are plotted for three different values of the Reynolds number. This figure shows that by increasing the shaft rotational speed that leads to high values of Re, the compressibility effect decreases. But one should recall that the lubricant pressure is nondimensionalized by the dynamic pressure $\rho \overline{V}^2$, which differs materially from one curve to the next. For example, the value of dynamic pressure at Re=20 is four times greater than that of Re=10. By considering this fact, it can be concluded that the lubricant compressibility has more influence on generated hydrodynamic pressure for the bearings with high rotational speed.

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