COMPUTATIONAL MECHANICS 2021
36th conference with international participation
Srní
November 8 - 10, 2021

Calculation of stiffness and damping coefficients of hydrodynamic lemon-bore journal bearings

J. Pokorný[a], T. Návrat[a]

[a] *Institute of Solid Mechanics, Mechatronics and Biomechanics, Faculty of Mechanical Engineering, Brno University of Technology, Technická 2896, 616 69 Brno, Czech Republic*

1. Introduction

This paper deals with a calculation of stiffness and damping coefficients of the hydrodynamic lemon-bore journal bearings of the finite length. The hydrodynamic pressure distribution is described by the dynamic Reynolds equation solved using the finite volume method. The variable temperature in the circumferential and axial directions of the bearing is considered in the calculation resulting in changes of the lubricant viscosity and density. The proposed computational model involves the mixing of the lubricant, obtaining a steady-state solution of the pressure and temperature, and establishing the equilibrium position of the journal. Based on the small dynamic displacements of the journal from the equilibrium position, the stiffness and damping coefficients are determined. The results of these dynamic characteristics are presented in dependence on speed for the selected load of the particular lemon-bore bearing.

2. Theory

The position of the journal of the hydrodynamic bearing is given by the eccentricity e and the attitude angle ψ (Fig. 1). The journal is separated from the bearing bush by the lubricant film of the thickness h. There is the hydrodynamic pressure p in the lubricant film.

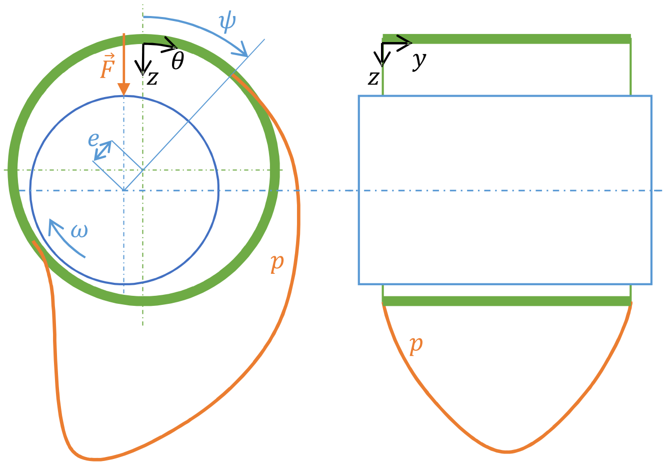

Fig. 1. Cylindrical hydrodynamic bearing in the static equilibrium position, the used coordinate system, [1]

The hydrodynamic pressure distribution is described by general Reynolds equation for dynamic lubrication

$$\frac{\partial}{R^2 \partial\theta}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial y}\right)=6\left(U\frac{\partial(\rho h)}{R\partial\theta}+2\frac{\partial(\rho h)}{\partial t}\right), \quad (1)$$

where R is the journal radius, θ and y are the circumferential and axial coordinates, ρ and η are the lubricant density and viscosity, U is the sliding velocity and t denotes time, [3].

The two-dimensional temperature distribution T in the lubricant film, assuming the constant temperature across the film thickness with adiabatic boundary conditions, follows the energy equation [3]:

$$q_\theta \frac{\partial T}{R\partial\theta}+q_y\frac{\partial T}{\partial y}=\frac{\eta U^2}{J\rho c_p h}+\frac{h^3}{12\eta J\rho c_p}\left[\left(\frac{\partial p}{R\partial\theta}\right)^2+\left(\frac{\partial p}{\partial y}\right)^2\right], \quad (2)$$

where $q_\theta = \frac{Uh}{2} - \frac{h^3}{12R\eta}\frac{\partial p}{\partial\theta}$ and $q_y = -\frac{h^3}{12\eta}\frac{\partial p}{\partial y}$ are the flow rates, J is the mechanical equivalent of heat, and c_p is the specific heat of the lubricant. The Reynolds equation (1) and the energy equation (2) are solved using the finite volume method.

The cold lubricant is supplied from the reservoir as the bearing operates. Therefore, the mixing of the lubricant must be considered to determine the temperature change in the lubricant film [2]:

$$T_{1,n} = T_0 + c_{mix}\frac{q_{2,n-1}}{q_{s,n}+q_{2,n-1}}(T_{2,n-1}-T_0), \quad (3)$$

where $T_{1,n}$ is the temperature at the leading edge of the nth lobe, T_0 is the lubricant supply temperature, c_{mix} is the mixing factor, $q_{2,n-1}$ is the exit flow at trailing edge of the upstream lobe and $T_{2,n-1}$ is its temperature, and $q_{s,n}$ is the side leakage of the lobe.

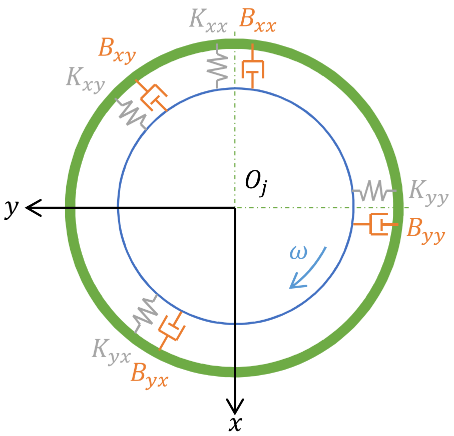

Fig. 2. Scheme of the hydrodynamic bearing stiffness and damping coefficients, [5]

Due to the temperature dependence of the viscosity and density of the lubricant, equation (1) is solved simultaneously with (2) and (3) until a steady-state solution is found. At the same time, the static equilibrium position of the journal is sought, resulting in the balance between the external load and the hydrodynamic pressure [4]. The stiffness and damping coefficients are determined using small dynamic displacements about this equilibrium position. The reaction forces of the lubricant film are functions of this small displacements:

$$\boldsymbol{f} = \boldsymbol{K}\,\boldsymbol{U} + \boldsymbol{B}\dot{\boldsymbol{U}}, \quad (4)$$

where $\boldsymbol{f}$ is the force response vector, $\boldsymbol{U}$ is the displacement excitation vector, $\dot{\boldsymbol{U}}$ is the velocity excitation vector, $\boldsymbol{K}$ and $\boldsymbol{B}$ are the stiffness and damping matrices both containing four coefficients (Fig. 2).

The central finite differences are used to establish these dynamic coefficients. The excitation of the journal is performed separately in two directions, with one direction half of the coefficients is obtained, with the other direction the rest [6].

3. Results

Results are presented for the lemon-bore bearing with parameters given in Table 1.

Table 1. Hydrodynamic lemon-bore bearing parameters

Journal radius	$R = 45mm$
Bearing length	$l = 63mm$
Radial clearance	$c = 0.075mm$
Preload	$\delta = 0.7$
External load	$F = 5\,670N$ (average pressure $p_m = 1MPa$)
Rotational speed	$n = 2\,000 \div 30\,000rpm$
Lubricant type	oil TB 46
Lubricant supply temperature	45°C

Fig. 3 shows the static equilibrium position of the bearing for different rotational speeds. The direction of rotation is considered clockwise. The lubricant film thickness is not in scale; it was 50 times magnified. The maximum values of the pressure at $2000rpm$ and $30000rpm$ were $4.3MPa$ and $7.6MPa$, respectively. The minimum lubricant film thicknesses were $28\mu m$ and $69\mu m$ for these rotational speeds.

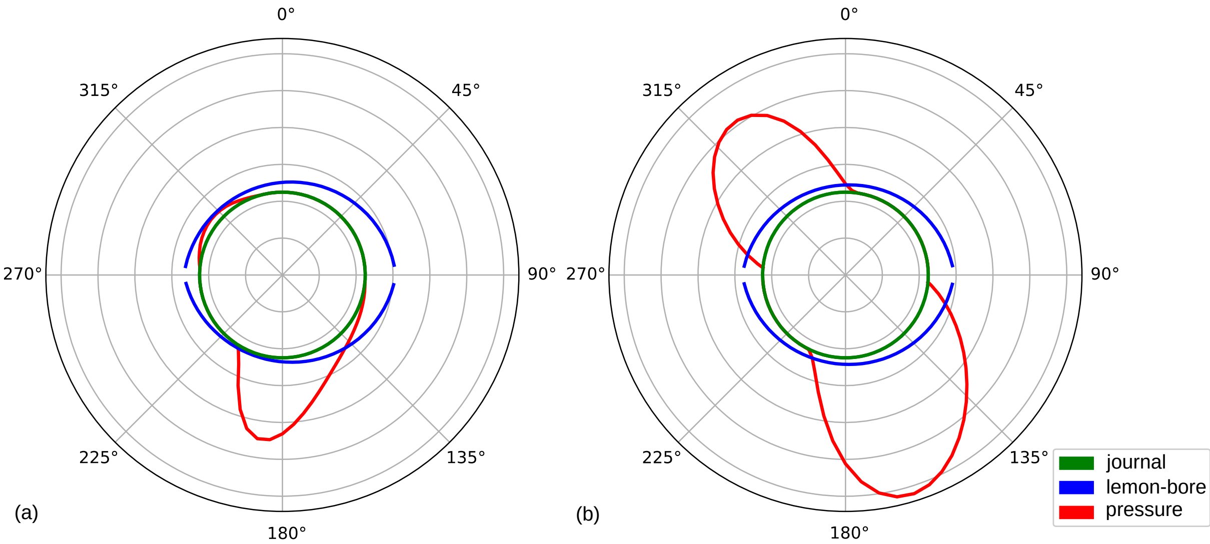

Fig. 3. Hydrodynamic pressure at the static equilibrium position: a comparison, $n = 2000rpm\ (a)$, $n = 30000rpm\ (b)$

Fig. 4 (a) shows the hydrodynamic pressure distribution at $30000rpm$. The pressure is shown not only at the bearing mid-plane (Fig. 3 (b)) but across the entire length of the bearing. Fig. 4 (b) shows the comparison of the temperature change in the circumferential direction for different rotational speeds. Due to mixing, the temperature change at the leading edge of the lobes is not zero.

Fig. 5 shows the dependence of the stiffness and damping coefficients on the rotational speed. The coordinate system is indicated in Fig. 2. The direct stiffness coefficient in the load direction increases with speed while all the damping coefficients decreases. The cross-coupled damping terms have the same value. Although the cross-coupled stiffness coefficients have a destabilizing effect, it is much smaller than in the case of a plain cylindrical bearing. Therefore, lemon-bore journal bearings are used more than cylindrical.

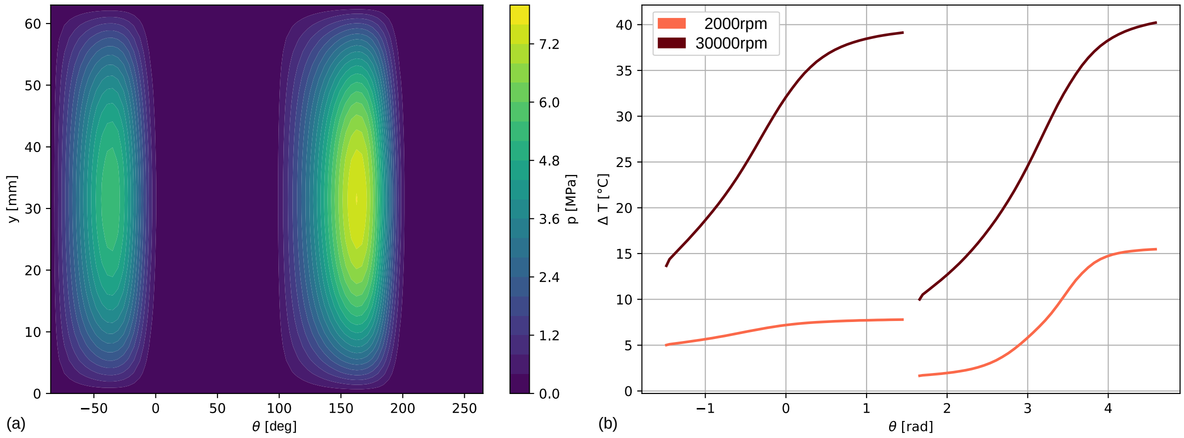

Fig. 4. Hydrodynamic pressure distribution at $30000rpm$ (a), Temperature change at the bearing mid-plane (b)

4. Conclusions

In the presented paper, the dynamic solution of the Reynolds equation was used to determine the stiffness and damping coefficients of the lemon-bore hydrodynamic journal bearing. The energy equation with lubricant mixing was coupled with the Reynolds equation to incorporate the temperature changes in the lubricant film. Thus, the variable viscosity and density of the lubricant were considered in the circumferential and axial direction of the bearing. Therefore, the steady-state solution of the static equilibrium position of the journal was obtained. Based on this position, the dynamic characteristics of the bearing were established. The speed dependence of these characteristics was presented for particular lemon-bore bearing.

The proposed approach can be recommended for determining the dynamic characteristics not only of lemon-bore bearings but also of plain journal bearings and other types of bearings with fixed geometry. It offers reasonable results within a short time. Further research might be focused on fixed-pad bearings or tilting-pad journal bearings.

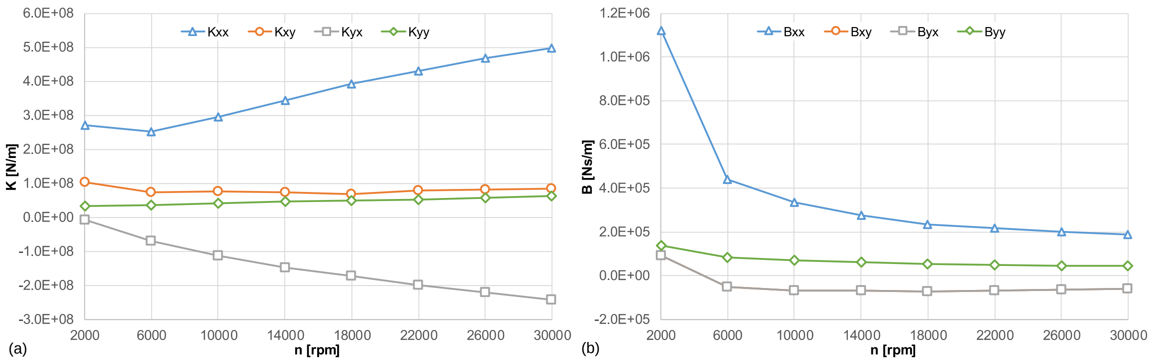

Fig. 5. Dynamic characteristics: a speed dependence, stiffness coefficients (a), damping coefficients (b)

References

[1] Harnoy, A., Bearing design in machinery: engineering tribology and lubrication, Marcel Dekker, New York, 2003.

[2] Heshmat, H., Pinkus, O., Mixing inlet temperatures in hydrodynamic bearings, Journal of Tribology 108 (1986) 231–244.

[3] Huang, P., Numerical calculation of lubrication: Methods and programs, John Wiley & Sons, Singapore, 2013.

[4] Pokorný, J., Calculation of static equilibrium position of hydrodynamic journal bearings, in: Zolotarev, I., Radolf, V. (Eds.), Engineering Mechanics 2019, Institute of Thermomechanics of the Czech Academy of Sciences, Svratka, 2019, pp. 291–294.

[5] San Andrés, L., Modern lubrication theory: Gas film lubrication, Notes 15, Texas, 2010.

[6] Tiwari, R., Rotor systems: Analysis and identification, 1st ed., CRC Press, Boca Raton, 2017.