# Dynamic Characteristics and Stability of High Speed Cylindrical Journal Bearing

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**Abstract**

The operation of plain circular journal bearing at high speed is restricted by the excessive temperature that is generated in the oil film and by the loss of stability. Low costs of machining, high-transmitted loads are the advantages of these types of bearings. Better knowledge of bearing operation and its stability from the point of hydrodynamic theory of lubrication and theory of vibration can be affected by modifications of bearing design. These modifications should result in the higher speeds of operation, decrease in power losses and better bearing stability. The basis of stability determination is the dynamic characteristics of bearing. For the plain circular journal bearing the dynamic characteristics has been calculated. Stability of high speed, symmetric rotor was determined, too. Different values of bearing length to diameter ratio, and relative clearance were assumed. Reynolds', energy and viscosity equations were solved by means of iterative procedure. Adiabatic oil film, laminar flow in the bearing gap as well as aligned orientation of journal in the bearing was considered.

***Keywords:*** cylindrical journal bearing, dynamic characteristics, stability

## 1. Introduction

Cylindrical journal bearings are the most popular, simple and inexpensive bearing type that is applied in the different bearings systems [1-6]. They can carry large load compare to the multilobe journal bearings but there is the restraint on high-speed operation. This restraint results from high temperature that is generated in the oil film as well from stability problems. Industrial applications of cylindrical journal bearings can be found in accelerating gear trains, planetary gears, and turbine gearboxes. In many applications the cylindrical journal bearings support rotors that operate at the speeds up to e.g. 30000 rpm [4]. The proper choice of bearing parameters, e.g. bearing clearance, oil supplied temperature and pressure [4-6] permits operation of the bearing at high speeds.

The improvement of turbomachinery efficiency and the reduction of costs yield to higher rotor weights and to enlarged specific bearing loads. Thus, the bearings are the limiting factor for the turbomachinery design and its efficiency [5]. Considering the costs of bearing design, the decrease in power losses and oil supply, it is possible to achieve these tasks in cylindrical journal bearings by introducing the design changes in bearings. These changes can include the inlet of oil to the sliding surfaces and type of supply e.g. side oil supply [6].

The loss of motion stability and the generation of self-vibrations are determined by the system parameters such as the unit load of bearing, bearing clearance and the bending stiffness of the shaft. It was stated in [4] that an effect of bearing clearance on the stability threshold is not explicit and identical for different unit loads of bearing. In the case of small loads, an increase in the bearing clearance can cause both the decrease and the increase in the stability threshold that depending on the clearance [4].

The stability of journal bearings includes the problems of hydrodynamic theory of lubrication and theory of vibrations. The bases for the investigation of bearing stability are the dynamic characteristics of bearing that are determined by four stiffness and four damping coefficients [1-3].

This paper presents the results of calculations of dynamic characteristics of the plain circular journal bearings operating at high speeds and different bearing clearance. The dynamic characteristics were applied for the determination of simple symmetric rotor stability. The applied, author's code of calculations and results allow for cylindrical bearing design changes towards application at higher speeds of operation.

## 2. Dynamic characteristics and bearing stability

The geometry of the oil film gap of cylindrical journal bearing (Fig. 1) can be described on the assumption of the parallel axis of journal and bearing bush, by Eqn. (1) [1, 3].

$$\overline{H} = 1 - \varepsilon \cdot \cos(\varphi - \alpha) \qquad (1)$$

where: $\overline{H} = h/(R - r)$ – dimensionless oil film thickness, $h$ – oil film thickness (m), $R$ – sleeve radius (m), $r$ – journal radius, $\alpha$ – attitude angle, $\varphi$ – peripheral co-ordinate.

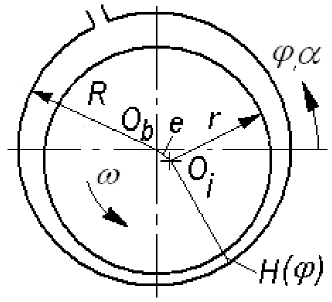


Figure 1. Geometry of cylindrical journal bearing; $e$ – eccentricity, $O_b$, $O_j$ – bearing and journal radius, $\omega$ - angular velocity

The journal bearing performances for adiabatic model of oil film can be determined by the numerical solution of the oil film geometry, Reynolds, energy and viscosity equations on the assumption of static equilibrium position of the journal [6, 7]. The oil film pressure distribution was defined from equation (2).

$$\frac{\partial}{\partial \varphi}\left(\frac{\overline{H}^3}{\eta}\frac{\partial \overline{p}}{\partial \varphi}\right) + \left(\frac{D}{L}\right)^2 \frac{\partial}{\partial \overline{z}}\left(\frac{\overline{H}^3}{\overline{\eta}}\frac{\partial \overline{p}}{\partial \overline{z}}\right) = 6\frac{\partial \overline{H}}{\partial \varphi} + 12\frac{\partial \overline{H}}{\partial \phi} \qquad (2)$$

where: $\overline{p}$ – dimensionless oil film pressure, $\overline{p} = p\psi^2/(\eta\,\omega)$, $p$ – oil film pressure (MPa), $D$, $L$ – bearing diameter and length (m), $t$ – time (sec), $\overline{z}$ – axial co-ordinates, $\phi = \omega t$ – dimensionless time, $\overline{\eta}$ – dimensionless viscosity, $\psi$ – bearing relative clearance, $\psi = \Delta R/R$ (‰), $\Delta R$ – bearing clearance, $\Delta R = R - r$ (m).

It has been assumed for the pressure region that on the bearing edges the oil film pressure $p(\varphi, z) \geq 0$, ($p(\varphi, z) > 0$ – side oil supply [6]) and in the regions of negative pressure, $p(\varphi, z) = 0$. The boundary conditions for the oil film pressure and temperature take into account the inlet pressure and temperature. The oil film pressure distribution computed from Eqn. (2) has been introduced in the transformed energy equation [6, 7]. Temperature values $T(\varphi, z)$ on the boundaries ($z = \pm L/2$) were determined by mean of the parabolic approximation [6]. Temperature and viscosity distribution were found by the iterative solution of equations (1), (3) and energy one [7].
The equations of motion for the journal and the centre of elastic shaft are given in matrix form by Eqn. (3). All the stiffness and damping coefficients were calculated by means of perturbation method [1-3].
The motion of simple symmetric rotor can be described by the following equation [3]:

$$M \cdot \ddot{x} + B \cdot \dot{x} + C \cdot x = \hat{a}\cos\omega t + \hat{b}\sin\omega t \tag{3}$$

where: $M$, $B$, $C$ –matrices of mass, damping and stiffness, $\hat{a}, \hat{b}$ coefficients of dynamic constraints.

The coefficients of the characteristic frequency equation of 6-th order [3] depend on the stiffness $g_{ik}$ and damping $b_{ik}$ coefficients, Sommerfeld number $S_0$, relative elasticity of shaft $c_s$ and the ratio of angular velocity to the critical angular velocity of stiff rotor. As the result of transformations, the expression that determines the ratio of boundary angular speed $\omega_b$ to the critical $\omega_c$ one, and determines the stability of rotor, has the form [3]:

$$\left(\frac{\omega_b}{\omega_c}\right) = \frac{1}{1 + b_0 \cdot \dfrac{A_3}{A_1}} \frac{A_2 \cdot A_3^2}{A_1^2 + A_1 \cdot A_3 \cdot A_4 + A_0 \cdot A_3^2} \tag{4}$$

where: $b_0$ – ratio $S_0/c_s$, $c_s$ – relative elasticity of shaft, $c_s = f/\Delta R = g/(\omega_c^2 \cdot \Delta R)$, $f$ – static deflection of shaft, (m), $A_0 \div A_4$ are the combination of eight coefficients (four stiffness $g_{ik}$ and four damping $b_{ik}$), $S_0$ - Sommerfeld number, $S_0 = F \cdot \psi^2 /(L \cdot D \cdot \eta \cdot \omega)$, $S_{ok}$ – critical Sommerfeld number, $S_{ok} = S_0 \cdot \omega / \omega_c$.

## 3. Results of calculations

The results of theoretical calculations provide the bearing static and dynamic characteristics as well as the stability of simple, symmetric one mass rotor. The first one includes the Sommerfeld number $S_0$ and the static equilibrium position angles $\alpha_{eq}$. The data applied for the calculations are given in Table 1; the heat number $K_T$ depends on the applied speed, bearing clearance and supplied oil temperature ($K_T = \omega \cdot \eta_0 /(c_t \cdot \rho \cdot g \cdot T_0 \cdot \psi^2)$, where: $c_t$ – specific heat of oil, (J/kgK), $g$ – acceleration of gravity (m/s$^2$), $T_0$ – temperature of supplied oil, (°C), $\eta_0$ – dynamic viscosity of supplied oil, Ns/m$^2$, $\rho$ – oil density, (kg/m$^3$).

Comparison of the values of journal displacements ε and static equilibrium position angles $\alpha_{eq}$ that were obtained by author's calculation and experimental investigation of Glienicke [2] is shown in Fig. 2 and Fig. 3; calculated values are close to the values of Glienicke.

Table 1. Data applied for the calculations

| No. | $\psi$ [‰] | $n$ [rpm] | $v$ [m/s] | $\omega$ [$s^{-1}$] | $T_0$ [°C] | $\eta$ [$Ns/m^2$] | $K_T$ [-] |
|---|---|---|---|---|---|---|---|
| 1. | 2,7 | | | | 23 | 0.022 | 0.39902 |
| 2. | 10 | 30000 | 62.83 | 3141.53 | 23 | 0.022 | 0.02782 |
| 3. | 2,7 | | | | 40 | 0.014 | 0.13966 |
| 4. | 10 | | | | 40 | 0.014 | 0.01018 |

Exemplary stiffness and damping coefficients as well as the stability ranges are given in Fig. 4 through Fig. 9 and Fig. 10, Fig. 1, respectively.
The values of the stiffness and damping coefficients determined at different oil supply temperatures and for the bearing with the length to diameter ratio L/D=1.0 and the clearance $\psi$ = 2.7 ‰, can be observed in Fig. 4 through Fig. 7. The values of coefficients are slightly larger at higher temperatures (e.g. Fig.4 and Fig. 6 or Fig. 5 and Fig. 7).
An effect of bearing clearance on the dynamic characteristics was considered for the bearing relative clearance $\psi$ = 2,7 ‰ and $\psi$ = 10,0 ‰. An increase in the bearing relative clearance $\psi$ at assumed oil supply temperature does not show the significant changes in calculated stiffness and damping coefficients in the range of Sommerfeld numbers from nil up to $S_0$=1.5 (Fig. 6 through Fig. 9). At larger bearing clearance there is a greater range of Sommerfeld numbers (e.g. Fig. 8 and Fig. 9).

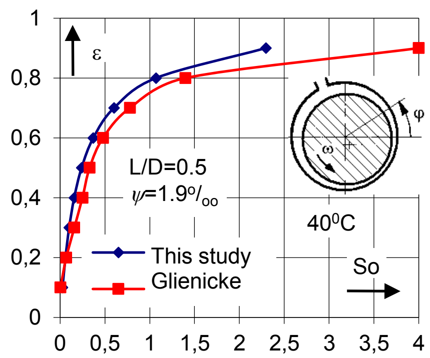


Figure 2. Sommerfeld number of cylindrical journal bearing

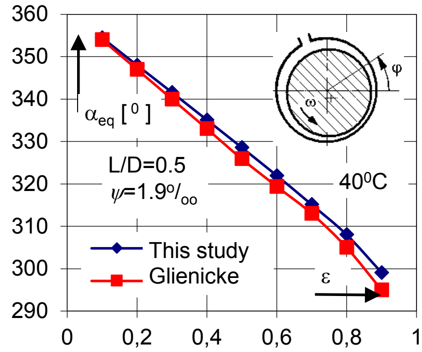


Figure 3. Static equilibrium position angle of cylindrical journal bearing

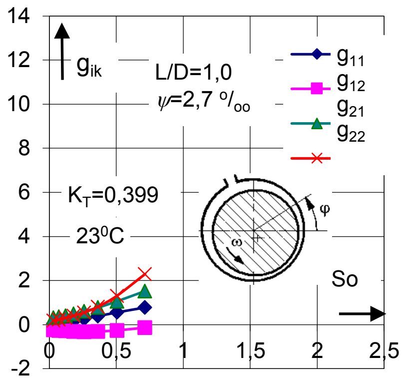


Figure 4. Stiffness coefficients of cylindrical journal bearing ($T_0 = 23°C$)

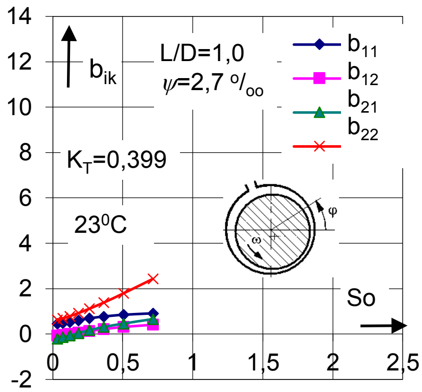


Figure 5. Damping coefficients of cylindrical journal bearing ($T_0 = 23°C$)

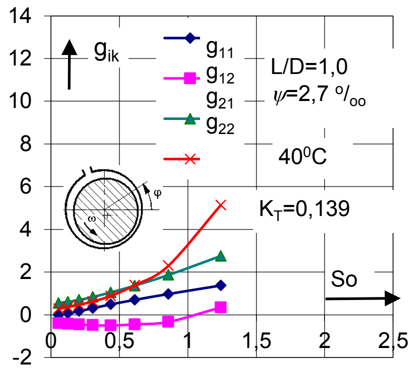


Figure 6. Stiffness coefficients of cylindrical journal bearing ($T_0 = 40°C$)

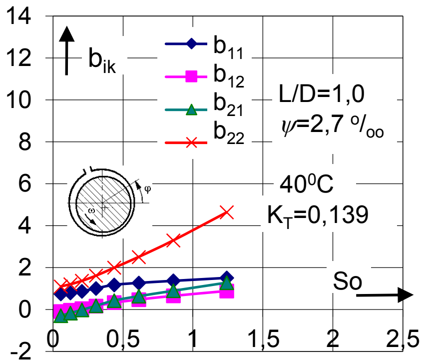


Figure 7. Damping coefficients of cylindrical journal bearing ($T_0 = 40°C$)

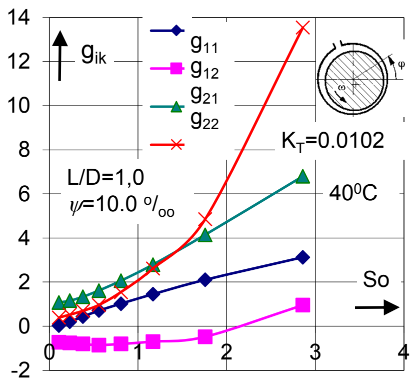


Figure 8. Stiffness coefficients of cylindrical journal bearing ($T_0 = 40°C$)

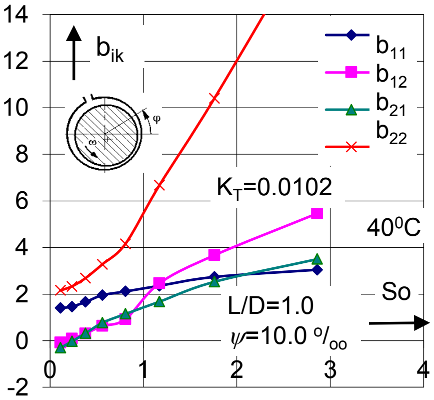


Figure 9. Damping coefficients of cylindrical journal bearing ($T_0 = 40°C$)

The stability ranges were determined for the bearing with the length to diameter ratio $L/D$ = 0.5 and two different values of bearing relative clearance. It results from Fig. 10 and Fig. 11 that the ranges of stability are larger at smaller values of bearing clearance; angle $\tau$ that determines the stability range is larger at the smaller bearing relative clearance.

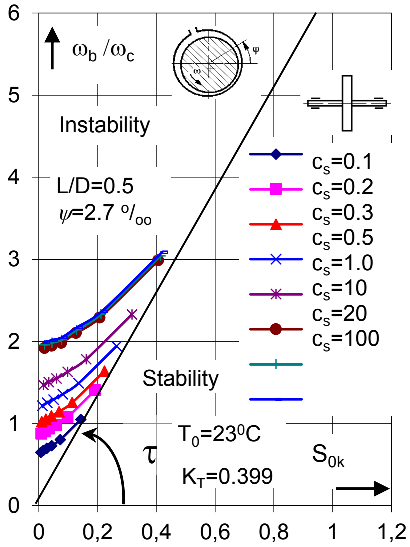


Figure 10. Stability chart for the cylindrical journal bearing ($T_0 = 23°C$)

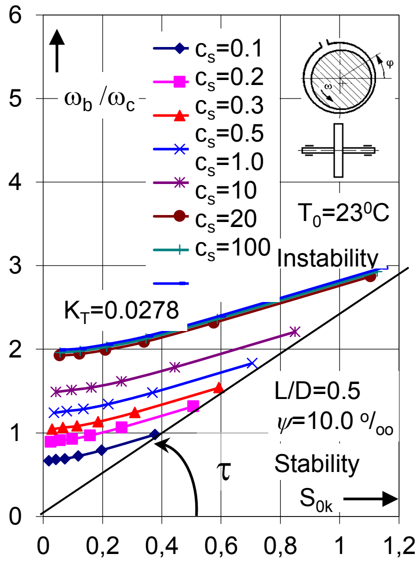


Figure 11. Stability chart for the cylindrical journal bearing ($T_0 = 23°C$)

## 4. Final remarks

The values of Sommerfeld number and static equilibrium position angles of own calculation are close to the values of Glienicke experimental investigation what certifies the correctness of developed code of calculation.

The results of the calculation of dynamic characteristics of cylindrical journal bearing show an effect of bearing relative clearance and supplied oil temperature on the stiffness and damping coefficients of oil film as well as on the stability of cylindrical bearing. It can be concluded that the variation of above-mentioned parameters can be applied for the design changes of considered bearing allowing at the same time for the operation at higher speeds.

Additional effects on the increase in the operational speeds can be achieved by means of the modification of inlet of supplied oil to the sliding surface of bearing as well as by pressurized side oil supply [6]; the developed code of calculations allows for theoretical investigation into these effects and for the obtaining the static and dynamic characteristics comprising the basic parameters of bearing such as load capacity, static equilibrium position angles, power losses, stiffness and damping coefficients and the stability ranges.

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