Effect of the Sleeve Profile on the Stability of Rotor Operating in Multilobe Journal Bearings

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Abstract

Multilobe journal bearings with the sleeve of the 2-, 3 or 4- sliding lobes of cylindrical profile are applied in different types of rotating machinery. The design of such journal bearings, the number of lobes and oil grooves improves thermal state of bearing at higher speeds and the stability of operation.
The paper describes the results of the calculations of dynamic characteristics and determination of stability ranges of simple symmetric rotor operating in the offset types of multilobe journal bearings. The dynamic characteristics of supporting bearings are defined by four stiffness and damping coefficients of oil film. The iterative solution of Reynolds, energy and viscosity equations allows the obtaining of the load capacity of bearings and the required dynamic coefficients of oil film. Adiabatic, laminar oil film and the static equilibrium position of journal were assumed. The oil film pressure, temperature, viscosity fields and the oil film forces were the basis of the bearing dynamic characteristics and stability determination.

Keywords: multilobe, offset journal bearings, stability of rotor

1. Introduction

The stability of rotor operating in journal bearings can be determined on the basis of supporting bearings dynamic characteristics expressed by the stiffness and damping coefficients of oil film [1-6]. The multilobe bearings, mostly used in slightly loaded, high speed rotating machines are characterised by good damping of vibrations and good stability of operation [1-3]. Exemplary types of such bearings are the 2- (offset-halves) [6], 3-and 4-lobe offset bearings [7-11] that are applied in the turbine gearboxes [6].
The design of 2-, 3- or 4-lobe journal bearings, the number of lobes and oil grooves improves the thermal state of bearing and stability of operation [1-3]. These multilobe journal bearings can be manufactured as the bearings with cylindrical sliding surfaces [5], with pericycloid profile of bearing bore [7,8] or as the offset ones [6,9].
Typical multilobe (classic) journal bearing is composed of single circular sections whose centres of curvature are not in the geometric centre of the bearing. The geometric configuration of the bearing as a whole is discontinuous and not circular. The multilobe pericycloid journal bearings ("wave bearings" [7,8]) is characterised by continuous profile and multihydrodynamic oil films on the journal perimeter.
The characteristic feature of multilobe, offset journal bearings are that the circle inscribed in the bearing profile touches the end of the convex gap of the bearing [5,6] in the direction of journal rotation (Fig. 1).
The paper presents the effect of sleeve profile on the stability of simple symmetric, elastic rotor operating in 2-, 3- and 4-lobe offset journal bearings The oil film pressure,

temperature and viscosity fields that are required for the calculations of the bearing static and dynamic characteristics, have been obtained by iterative solution of the Reynolds', energy and viscosity equations. Adiabatic, laminar oil film and the static equilibrium position of journal were assumed. The dynamic characteristics of journal bearing are the basis of stability ranges determination [1-3]. All the stiffness and damping coefficients were calculated by means of perturbation method [1,6].

2. Stability of elastic rotor

Considered multilobe, offset type journal bearings are presented in Fig. 1. Their geometry can be found in [5,6].

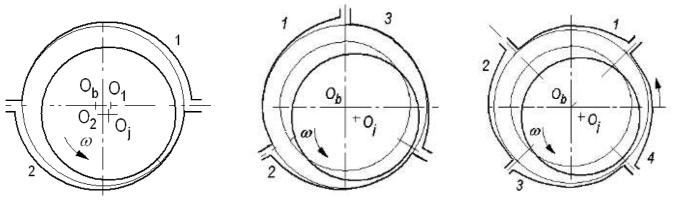

Figure 1. Lay-out of the 2-, 3-, and 4-lobe offset journal bearings; O_b, O_j, O_1, O_2 – centres of bearing, journal, upper and bottom lobe (2-lobe offset bearing), 1 ÷ 4 number of lobes

The journal bearing static and dynamic characteristics for adiabatic or diathermal model of oil film can be determined by the numerical solution of the oil film geometry, Reynolds, energy, viscosity [2,6] equations. The stiffness g_{ik} and damping b_{ik} coefficients allow the determination of stability ranges [1-8].

The equations of motion for the journal and the centre of elastic shaft are given in matrix form by Eqn. (1) [4].

$$M \cdot \ddot{x} + B \cdot \dot{x} + C \cdot x = \hat{a}\cos \omega t + \hat{b}\sin\omega t \tag{1}$$

where: *M*, *B*, *C* –matrices of mass, damping and stiffness, $\hat{a}, \hat{b}$ - coefficients of dynamic constraints, ω- angular velocity, (s^{-1}).

After transformations of Eqn. (1) the real and imaginary part was obtained [1,4]. The stability of elastic rotor-bearing system is analysed based on the following characteristic frequency equation of 6-th order with regard to (λ/ω) [1-6].

$$c_6\,\lambda^6 + c_5\,\lambda^5 + c_4\,\lambda^4\; c_3\,\lambda^3 + c_2\,\lambda^2 + c_1\,\lambda + c_0 = 0 \tag{2}$$

1.

2. Solution assumption for Eqn. (2) is $\lambda_j = - u_j + iv_j$ ($1 \le j \le 6$), with u as damping and v representing the self-vibrations. Stability of the linear vibrations of system occurs only when all real parts of eigenvalues λ_j are negative. The coefficients c_0 through c_6 in Eqn. (2) are the functions of a_0 , b_0 , g_{ik}, b_{ik}.

$$c_0 \div c_6 = f(a_0, b_0, g_{ik}, b_{ik}) \quad (3)$$

where: a_0 - ratio of angular velocity ω to the angular self-frequency of stiff shaft, $a_0 = (\omega / \omega_c)^2$, ω_c - angular self frequency of stiff rotor, $\omega_c = \sqrt{c^* / m}$, b_0 - ratio of Sommerfeld number to the relative elasticity of shaft, $b_0 = So/c_s$, c* - shaft stiffness, (N m^{-1}), c_s - relative elasticity of shaft, $c_s = f/\Delta R = g/(\omega_c^2 \cdot \Delta R)$, f- static deflection of shaft, (m), F.- resultant force of oil film (N), F_{stat} - static load of bearing, (N), g - acceleration of gravity, (m s^{-2}), g_{ik} - dimensionless stiffness coefficients, $g_{ik} = So(\Delta R/F_{stat})$, g'_{ik}, - stiffness coefficients, (N/m), b_{ik}- dimensionless damping coefficients, $b_{ik} = So(\Delta R/F_{stat})\,\omega \cdot b'_{ik}$, b'_{ik} - damping coefficients, (N s/ m), *m* - mass of the rotor, (kg), *So* - Sommerfeld number, $So = F \cdot \psi^2 / (L \cdot D \cdot \eta \cdot \omega)$, S_{ok} -critical Sommerfeld number, $S_{ok} = So\ \omega/\omega_c$

Coefficients of characteristic Eqn. (2) depend on the stiffness and damping coefficients, Sommerfeld number So, relative elasticity of shaft c_s and the ratio of angular velocity to the critical angular velocity of stiff rotor. The expression determining the ratio of boundary angular speed Ω_b to the critical ω_c one, and the stability of rotor [1-4], is:

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

where: A_0, A_1, A_2, A_3, A_4, consist the stiffness g_{ik} (i=1,2 and k=1,2) and damping b_{ik} (i=1,2 and k=1,2) coefficients [1-6], Ω_b - boundary angular velocity (s^{-1})

3. Results of calculations

The stability of simple elastic, symmetric rotor (Jeffcott rotor) was determined with the use of the calculated dynamic characteristics The calculations included the non-dimensional load capacity S_0 and journal displacement ε as well as the static equilibrium position angles α_{eq}, too. The offset journal bearings under consideration have the length to diameter ratio L/D=0.6, L/D=0.75 and L/D=0.8. Different lobe relative clearances ψ_s (for the 3-lobe pericycloid bearing its relative eccentricity was λ* = 0.25 [6,7]) and rotational speeds were assumed. The bearing relative clearances were ψ = 0,9‰, ψ = 1.5‰ and ψ = 2.7‰. The feeding oil temperature was T_0=40^{0}C and the corresponding thermal coefficients K_T [2,8] were 0.139, 0.215 and 0.315; $K_T = \omega \cdot \eta_0 / (c_t \cdot \rho \cdot g \cdot T_0 \cdot \psi^2)$ where: c_t - specific heat of oil, (J/kgK), g - acceleration of gravity (m/s^2), η_0 - dynamic viscosity of supplied oil (Ns/m^2), ρ - oil density, (kg/m^3), ψ - bearing relative clearance, (‰).

Exemplary results of the calculations of dynamic characteristics and stability are shown in Fig. 2 through Fig. 15. The stiffness g_{ik} and damping b_{ik} coefficients are given in Fig.2 through Fig.7. The stability ranges can be observed in Fig. 8 through Fig. 15.

Fig. 2 and Fig. 3 show the stiffness and damping coefficients of the offset-halves bearing with the lobe relative clearance ψ_s =2. The largest values have the coefficient g_{22} for the Sommerfeld numbers over 0.44. The values of the coupled damping coefficients b_{12} and b_{21} are very close (e.g. Fig. 3). The damping coefficient b_{11} is larger than the ones b_{12} and b_{21} in the range of Sommerfeld numbers S_0 from 0 to about 0,6. At increasing values of Sommerfeld numbers, the coefficient b_{11} is smaller than the coupled terms b_{12} and b_{21} (Fig. 3).

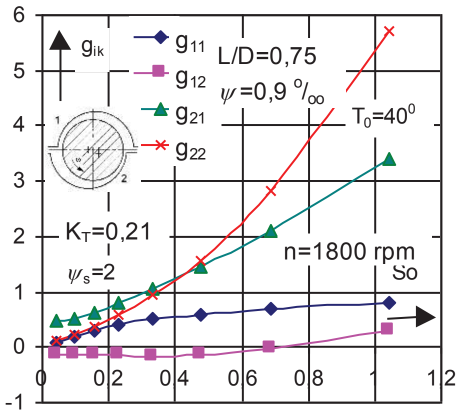

Figure 2. Stiffness coefficients of offset-halves offset- journal bearing

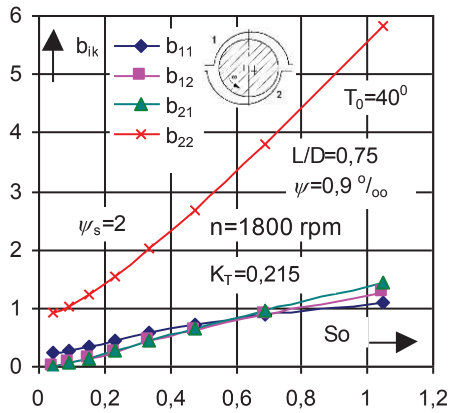

Figure 3. Damping coefficients of halves journal bearing

The stiffness and damping coefficients of the 3-lobe offset journal bearing are presented in Fig. 4 and Fig. 5. The bearing length to diameter ratio L/D=0.8, clearance ratio ψ = 1.5 ‰ and lobe relative clearance ψ_s=2 were assumed. Heat number K_T was K_T=0.315.
The largest values have the coefficients g_{22} and b_{22} but the smallest g_{12} and b_{21} (Fig. 4 and Fig.5). The values of the coupled damping coefficients b_{12} and b_{21} are very close and b_{11} is larger than the coupled coefficients (Fig. 5).
The stiffness and damping coefficients of 4-lobe offset bearing are given in Fig. 6 and Fig. 7. The largest values has the stiffness coefficient g_{21} and the smallest value the coefficient g_{11} (Fig. 6); In the range of Sommerfeld number from nil to about 0.12 the coefficient g_{22} is larger than g_{12} but at higher Sommerfeld numbers there is reverse dependence. The damping coefficient b_{22} is the largest and the coupled coefficients have the smallest, equal values, i.e. $b_{12}=b_{21}$ (Fig. 7).
The stability ranges of symmetric rotor operating in 2-lobe offset bearings with the lobe relative clearance ψ_s =1 (Fig. 8 - cylindrical sliding surfaces) and ψ_s =2 (Fig. 9 - lemon shaped sliding surfaces) show the difference. The rotor running in the bearings characterized by the value of lobe relative clearance ψ_s =2 shows larger range of stability (e.g. Fig. 8 and Fig. 9).
Exemplary results of stability ranges that were obtained for 3-lobe offset journal bearing at different values of lobe relative clearance ψ_s and at different relative elasticity of shaft

c_s are shown in Fig. 10 and Fig. 11. At the values of the relative elasticity of shaft under consideration there is an increase in the stability at the increase in the lobe relative clearance ψ_s (e.g. the curves for c_s=0.1 in Fig. 10 and Fig. 11). The coefficient tg τ in the range of larger critical Sommerfeld numbers S_{ok} is the measure of stability properties of bearing [2,3]. Larger values of angle τ mean the larger range of stability, i.e. at assumed load of bearing there is higher boundary of stability Ω_b/ω_c [2,3].

For the comparison task, the stability ranges of rotor operating in 3-lobe pericycloid journal bearing and in classic 3-lobe bearing are shown in Fig.12 and Fig. 13. It results from the comparison that the 3-lobe offset bearing has better stability properties than the 3-lobe pericycloid and 3-lobe classic bearings.

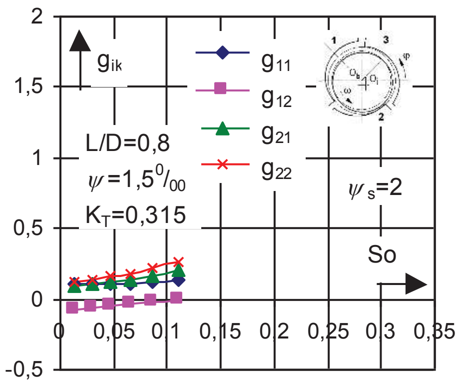

Figure 4. Stiffness coefficients of 3-lobe offset journal bearing

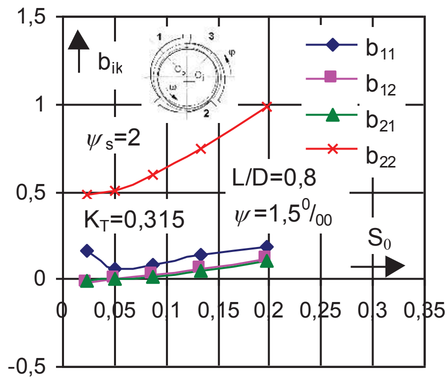

Figure 5. Damping coefficients of 3-lobe offset journal bearing

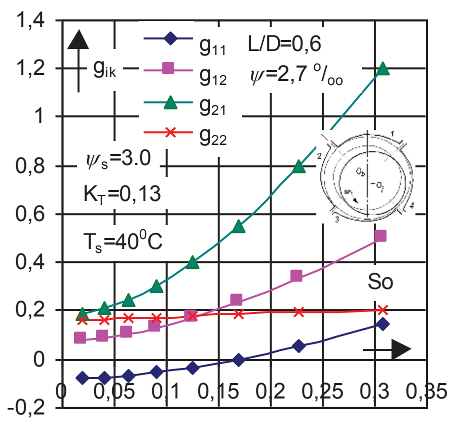

Figure 6. Stiffness coefficients of 4-lobe offset journal bearing

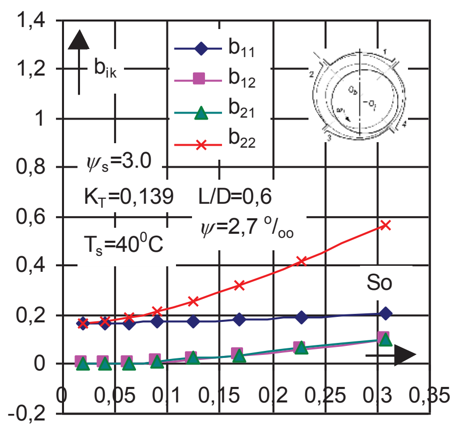

Figure 7. Damping coefficients of 4-lobe offset journal bearing

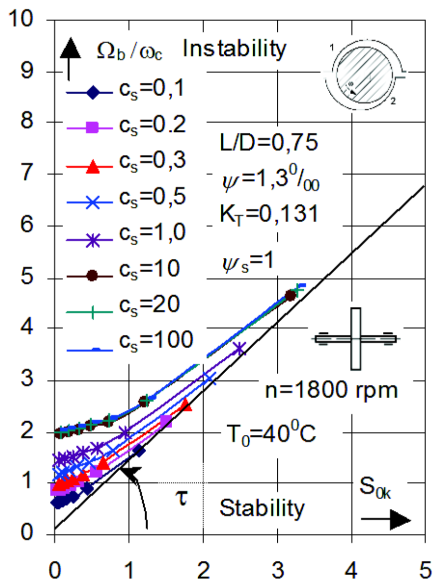

Figure 8. Stability chart of rotor operating offset-halves journal bearings (ψ_s =1)

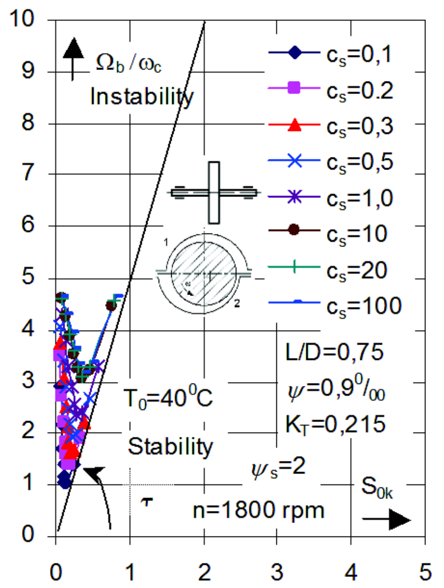

Figure 9. Stability chart of rotor operating in offset-halves journal bearings (ψ_s =2)

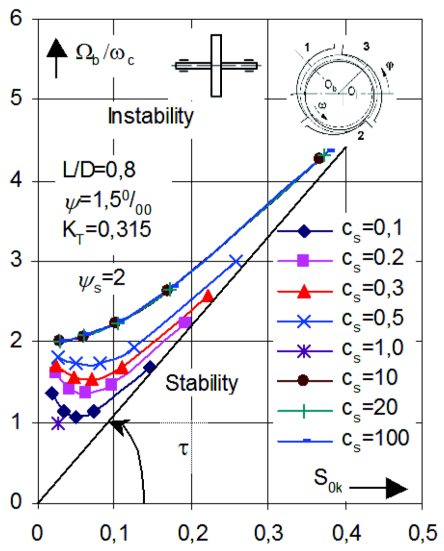

Figure 10. Stability chart of rotor operating 3-lobe offset journal bearing

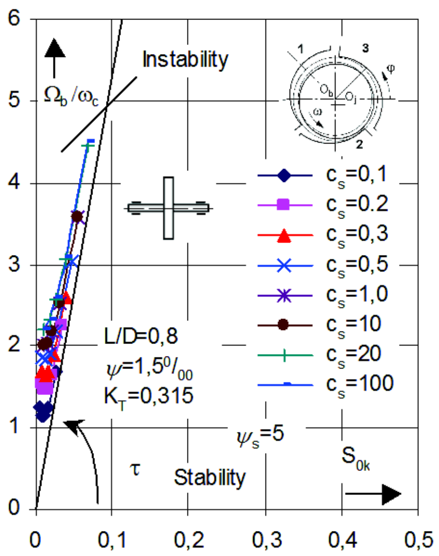

Figure 11. Stability chart of rotor operating in in 3-lobe offset journal bearing

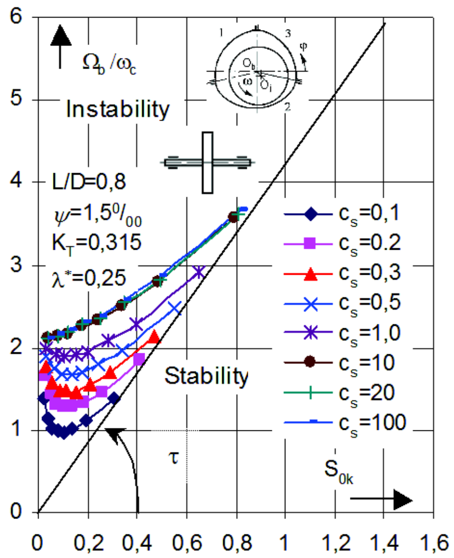

Figure 12. Stability chart of rotor operating in 3-lobe pericycloid journal bearing

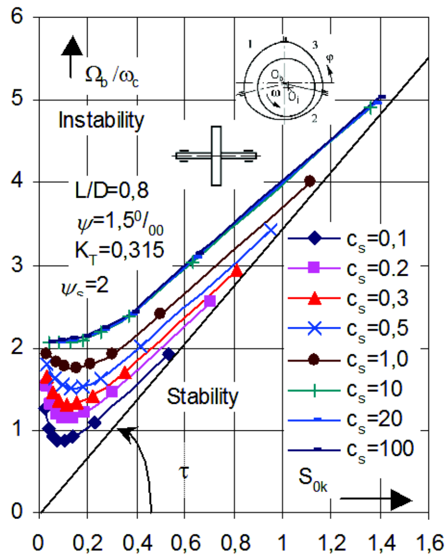

Figure 13. Stability chart of rotor operating in 3-lobe journal bearing

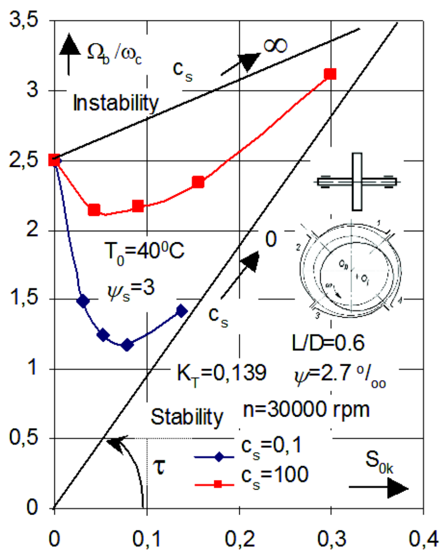

Figure 14. Stability of elastic rotor operating in 4- lobe offset journal bearing at the lobe relative clearance $\psi_s = 3$

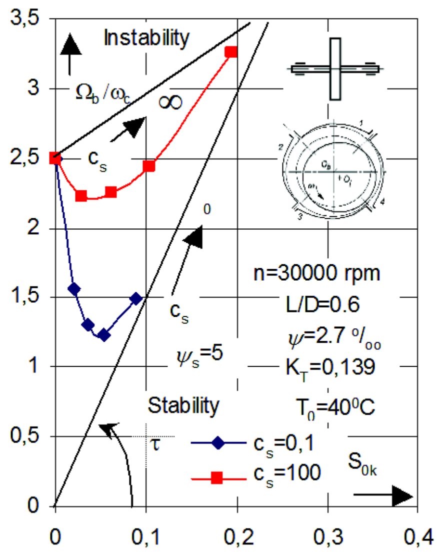

Figure 15. Stability of elastic rotor operating in 4-lobe offset journal bearing at the lobe relative clearance $\psi_s = 5$

4. Conclusions

Dynamic characteristics including the stability ranges of the chosen types of offset bearing with different sleeve profiles and on the assumption of adiabatic model of oil film, were obtained by means of perturbation method. Investigation that were carried out at assumed geometric and operating parameters, various relative shaft elasticity values, allows to draw the following conclusions:

1. Static and dynamic characteristics of the considered journal bearings of different sleeves profiles can be obtained from developed program of numerical calculations.
2. The offset type multilobe journal bearings show better stability than the classic multilobe bearings.
3. An increase in the relative elasticity of shaft increases the range of rotor stability.
4. The results of developed program form the input data for the investigation and analysis into the stability of different types of multilobe journal bearings.

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