Floating ring journal bearings of multilobe design in turbochargers

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Abstract. The floating ring bearings system of turbocharger operates at high rotational speeds and high temperatures on the inlet of exhaust gases. Usually, such system applies the cylindrical profile of floating ring and stationary sleeve. The changes of the geometry of inner sliding surface of stationary sleeve as well as external or inner sliding surface of floating ring into multilobe surfaces allow for obtaining additional operation properties. It can be expected the changes in the static and dynamic characteristics of bearing what results in higher speeds of operation and better stability. The paper considers application of multilobe sliding surfaces in the design of turbocharger as well as the computation possibilities of such design bearings. Exemplary results of the computation of static and dynamic characteristics for the chosen types of multilobe bearings support their application in high speed rotating machinery including the floating ring bearings of turbochargers. Applied code and the results of this work allow calculating the stability of rotors operating in the bearings with the turbulent model of oil film and particularly the high speed floating ring bearings.

1. Introduction
Cylindrical journal bearings are the most popular, simple and inexpensive bearing type that is applied in the different bearings systems including the turbochargers. 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. The journal bearings are usually the limiting factor for the turbomachinery design and for its efficiency; this factor is important in case of turbochargers bearings system, too. The turbocharger shaft and turbine wheel assembly rotates at speeds up to 300,000 rpm. Turbocharger life should correspond to that of the engine, which could be 1,000,000 km for a commercial vehicle. Only semi floating or floating ring journal bearings specially designed for turbochargers can meet these high requirements at a reasonable cost. Typical turbocharger bearings apply cylindrical profile of sleeve and floating ring what assures small power loss, good damping of vibration and very good stability of operation [1-9]. The rotor and bearings of turbocharger can be considered as tribosystem operating at high speeds and high thermal loads [2,7]. Within the structure of this tribosystem the distinction can generally be drawn between of three parts, i.e. journal, sleeve and cylindrical sleeve placed between of these parts. The rotating parts of turbocharger are separated by oil films allowing the operation at very high rotational speeds free of vibrations and without the loss of motion stability [3,10]. Reliable design of turbocharger is assured by the calculation, analysis and correct choice of the static and dynamic characteristics of bearings [2,9] for the laminar or turbulent oil film [11]. The knowledge of these characteristics allows the thermal analysis of bearing, its deformation and dynamic analysis including, e.g. the calculation of the critical speeds of system rotor/bearing. The changes of the geometry of inner sliding surface of stationary sleeve as well as external or inner sliding surface of floating ring into multilobe surfaces allow for obtaining additional operation properties [12]. It can be expected the changes in the static and dynamic characteristics of bearing [13,14] what results in higher speeds of operation and better stability. The design of multilobe journal
bearings, the number of lobes and oil grooves improves the thermal state of bearing and stability of operation [1-2]. These multilobe journal bearings can be manufactured as the bearings with cylindrical operating surface [13], with pericycloid profile of bearing bore [15] or as the offset ones [14]. 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” [15] is characterised by continuous profile and multihydrodynamic oil films.

Combination of cylindrical and multilobe profiles in the design of floating ring bearings can give the reliable and durable design allowing on the increase in the rotational speeds of rotor [10,12]. The newly-developed bearings should meet the rotor dynamics criteria, be interchangeable with existing bearings and the power losses as well the oil supply requirements should be maintained.

The manufacturers of turbochargers do not give any specific information on the types of applied bearings or on the calculation codes. The method of solution and code of floating ring-bearing calculation should rely on a load and moment balance on the inner and outer surfaces of the ring. It requires the numerical solution of the Reynolds, energy and viscosity equations for both films. Recirculation of the oil in the bearing system, heat and mass transfer between the inner and outer films, must be included.

The program developed by the author allows obtaining the static and dynamic characteristics of cylindrical and multilobe journal bearings including oil film pressure, temperature, viscosity distributions, static equilibrium position angles, load capacity, minimum oil film thickness, friction loss, oil flow. Laminar and turbulent adiabatic oil film, static equilibrium position of journal and parallel axis of journal and bearing can be assumed. Transfer of the oil film pressure and temperature distributions into the commercial programs as e.g. ANSYS allows obtaining the deformation of bearing system elements [16].

The paper presents multilobe design solutions of the floating ring bearings that can be applied in the bearings systems of turbochargers. It points on the possibility of application of author’s code of numerical calculation in the design process of turbochargers journal bearings.

2. Design of the bearings of turbochargers
The general view of turbochargers used in small and large engines with the location of one piece semi-floating journal bearing is shown in ‘figure 1’; both rotor disks are overhang and the bearing is kept from rotating by pinning the flange on its end, by means of notch for anti-rotation pin. The rotor of turbochargers with semi-floating ring and the floating rings are presented in ‘figure 1’.

Figure 1. Cut-away models of turbochargers; a) with semi-floating ring bearing, 1 – outlet of exhaust gases, 2 - turbine, 3 – one piece semi-floating journal bearing, 4- compressor, 5- inlet of compressed air; b) with floating ring bearings; 6- floating rings
Floating ring bearings of different design are presented in ‘figure 2’; mostly applied is the bearing showed in ‘figure 2a’. Multilobe solutions show ‘figure 3b’ and ‘figure 2c’ [12]. The variants with the floating ring of modified internal or external surfaces can be characterized as follows:

1) at cylindrical external surface of floating ring, the inner surface of floating ring (bore of sleeve 2) has the respective number of segments of the radii larger than the radius of internal profile (‘figure 2b’),
2) at cylindrical internal surface of floating ring, the external surface of floating ring (bore of sleeve 2) has the respective number of segments of the radii smaller than the radius of its external profile (‘figure 2b’),
3) in case of cylindrical internal surface of stationary sleeve, the external and internal surfaces of floating ring (sleeve 2) have the respective number of segments of radii smaller than the radius of external radius of external surface of floating ring as well as larger than the radius of internal profile for the internal surface of floating ring (‘figure 2c’),
4) at multilobe bore of stationary sleeve, the external surface of floating ring (sleeve 2) has the cylindrical profile but the internal surface (bore) of floating ring is multilobe with the radii larger than the radius of circle inscribed in the internal, multilobe profile.

![Figure 2](image)

**Figure 2.** Floating ring bearings; a) – cylindrical ring and sleeve, b) three-lobe inner and external profile; inner segments of radii larger than the radius of circle inscribed into the inner profile and the external segments with the radii smaller than the radius of circle described on the external profile, c) three-lobe profile in the bore of floating sleeve and in the bore of sleeve; 1 - stationary sleeve, 2- floating ring, 3-journal

General view of floating ring bearings manufactured by author are showed in ‘figure 3’. Depending on the requirements they can obtain the profiles: classic multilobe, pericycloid or offset.

![Figure 3](image)

**Figure 3.** General view of floating ring bearings; a- cylindrical floating ring and cylindrical sleeve, b - cylindrical floating ring and multilobe (three-lobe) sleeve; 1- the bearing casing made as the cylindrical (a) and three-lobe sleeve (b), 2 – floating ring, 3 – oil supply
3. Geometry of bearing

Application of multilobe journal bearings is strictly connected to the geometry of lubricating gap. The knowledge of the geometry relations among the different types of these bearings is of great deal in finding the best solution for the required application.

The geometry of the oil film gap of multilobe or pericycloid journal bearings (‘figure 4’) generally describes equation (1) [14,15].

\[ H_i(\phi) = H_{i,c} + H_{i,L,P}(\phi) \]  

where: \( H \) - dimensionless oil film thickness, \( H = h/(R-r) \), \( h \) - oil film thickness, (m), \( \phi \) - peripheral co-ordinate, 1- inner oil film, 2- outer oil film.

‘Figure 5b’ shows the dependence between the pericycloid relative eccentricity \( \lambda^* \) and the lobe relative clearance \( \psi_s \).

![Figure 4](image_url)

**Figure 4.** Geometry of 2-lobe journal bearings; a) geometry of the oil film of two-lobe pericycloid and two-lobe classic journal bearings \( H_{max} \), \( H_{min} \) - maximum and minimum oil film thickness at concentric orientation of journal and bush axis, \( p_s \) - oil supply pressure, \( \gamma \) - angular co-ordinate of the lobe centre, \( R_L, R_p \) - radius of bearing lobe and pericycloid, \( O_b, O_j, O_l \) - centre of: bearing, journal, bottom and upper bearing lobe, 1, 2, 3 - lobe numbers, b) the pericycloid eccentricity \( \lambda^* \) versus lobe relative clearance \( \psi_s \).

The first term of the right side of equation (1), which gives the oil gap thickness for the eccentric orientation of journal in the bearing bush has the following form:

\[ H_{i,c} = 1 - \varepsilon_i \cdot \cos(\phi_i - \alpha) \]  

where: \( \varepsilon \) - relative eccentricity, \( \alpha \) - attitude angle of centres line \( i \) - refers to the oil films

The oil film thickness \( H_{i,L}(\phi) \) of multilobe profile of bearing for concentric orientation of journal and sleeve axis describes equation (3).

\[ H_{i,L}(\phi) = \psi_{i,s} + (\psi_{i,s} - 1) \cdot \cos(\phi_i - \gamma_i) \]  

where: \( \psi_{i,s} \) - lobe relative clearance \( \psi_{i,s} = (R_L-r)/\Delta R \) (\( \psi_{i,s} = 1 \) for cylindrical bearing), \( \Delta R \) - bearing radial clearance \( \Delta R = R-r \) (m), \( r, R \) - journal and sleeve radius (m),
The oil film thickness of pericycloid bearing in concentric position of journal and sleeve can be expressed in non-dimensional terms by equation (4) (‘figure 4a’):

\[ H_{ip} = \lambda^* (1 + \cos n \varphi_i) \]  

where: \( \lambda^* \) - relative eccentricity of pericycloid \( \lambda^* = m/\Delta R \), \( m \) – eccentricity of pericycloid (m) \( m = R_p - r \), \( n \) – number of lobes (e.g. \( n = 3 \) for three-lobe bearing).

The comparison of classic three-lobe and three-lobe pericycloid journal bearings requires, that the geometry of bearings lobes have to be chosen as close as possible in the characteristic points of both profiles, i.e. at the angles \( \varphi = 30^\circ, 90^\circ, 150^\circ, 210^\circ, 270^\circ \) and \( 330^\circ \). In concentric position of journal and bearing and in mutual points of both bearing profiles the values of the oil film geometry are equal (‘figure 5’).

![Figure 5. Gap geometry of three-lobe pericycloid and three-lobe classic journal bearings (the relation between pericycloid eccentricity \( \lambda^* = 1.0 \) and the lobe relative clearance \( \psi_s = 5.0 \) determined at the journal relative eccentricity equal nil, \( \varepsilon = 0 \) and at the lobe angle 120°).](image)

**4. Static and dynamic characteristics of bearing**

The procedure of the determining of oil film pressure, temperature and viscosity distributions consists in the solution of Reynolds, energy and viscosity equations [10-15]. These equations were derived on the assumption that in the bearing gap exists the adiabatic laminar or turbulent flow of non-compressible Newtonian fluid. The oil film pressure distribution was obtained from the Reynolds equation:

\[ \frac{\partial}{\partial \phi} \left( \frac{H_i^3}{\eta K_z} \frac{\partial \tilde{p}}{\partial \phi} \right) + \frac{\partial}{\partial z} \left( \frac{H_i^3}{\eta K_z} \frac{\partial \tilde{p}}{\partial z} \right) = 6 \frac{\partial \tilde{H}_i}{\partial \phi} + 12 \frac{\partial \tilde{H}_i}{\partial \phi} \]  

where: \( D, L \) - bearing diameter and length (m), \( \tilde{p} \) - dimensionless oil film pressure, \( \tilde{p} = p \varphi^2 / (\eta \omega) \), \( p \) - oil film pressure, (MPa), \( K_z \) - turbulence correlation coefficients, \( z \) - axial co-ordinate, (m), \( \tilde{z} \) - dimensionless axial co-ordinate \( \tilde{z} = 2z / L \), \( \tilde{H} \) -dimensionless oil viscosity, \( \tilde{H} = \nu/\eta_0 \), \( \eta \) - dynamic viscosity, (Ns/m²), \( \eta_0 \) - viscosity of supplied oil, (Ns/m²), \( \psi \) - relative clearance, (‰), \( \psi = \Delta R/r \), \( \phi = \omega t \) – dimensionless time, \( \omega \) - angular velocity, (s⁻¹), \( t \) – time, (s).
In case of turbulent oil film [11] the dynamic viscosity should be considered as the effective one, i.e. 
\[ \eta(1+\nu/v) \text{ (} \nu \text{ -kinematical viscosity, } v \text{ - whirl viscosity of turbulent oil film)} \text{ and the energy equation has the form expressed by dimensionless equation (6).} \]

\[ \bar{u} \frac{\partial \bar{T}}{\partial \bar{\phi}} = K_T \cdot \bar{\eta}(1 + \frac{v}{\nu}) \cdot [(\frac{\partial \bar{u}}{\partial \bar{y}})^2 + (\frac{\partial \bar{w}}{\partial \bar{y}})^2] \]  

(6)

where: \( \bar{u}, \bar{w} \) - dimensionless average velocities in peripheral and axial direction.

Introducing the correlation factor \( \bar{r}_c \) [11] that is defined by the ratio of tangential stress \( \bar{(\tau^*)_{Coullet-turbulent}} \) (for turbulent Couette flow) to the shearing stress \( \bar{(\tau^*)_{Coullet-laminar}} \) in laminar Couette flow averages on the height of the lubricating gap the tangential stress of the turbulent flow:

\[ \bar{r}_c = \frac{\bar{(\tau^*)_{Coullet-turbulent}}}{\bar{(\tau^*)_{Coullet-laminar}}} = \frac{\bar{\tau}_{xyw} + \frac{H}{2} \frac{\partial \bar{p}}{\partial \bar{\phi}}}{\bar{\eta} / \bar{H}} \]  

(7)

where: \( \bar{\tau}_{xyw} \) - dimensionless wall shear stress.

The applied procedure of the calculation of bearing static characteristics consists in the solution of oil film geometry, Reynolds’s, energy, viscosity and heat transfer equations [11, 13]. The oil film thickness, pressure, and temperature obtained from this solution allow using these values in determining the load capacity, static equilibrium position angles, minimum oil film thickness, maximum oil film temperature, oil flow and friction loss [10, 15].

It has been assumed for the pressure region that the oil is supplied under pressure into the grooves but on the bearing edges \( (z = \pm L/2) \) as well as in the regions of negative pressures the pressure values are nil, i.e., \( \bar{p}(\phi, z) = 0 \) [14].

The oil film temperature distribution equation obtains the following form

\[ \frac{1}{\bar{\eta}} \frac{\partial \bar{T}}{\partial \bar{\phi}} = K_T \int_{-L/D}^{L/D} \left[ \frac{\bar{H}_1^3}{H} \left( \frac{1}{K_x} \left( \frac{\partial \bar{p}}{\partial \bar{\phi}} \right)^2 + \frac{1}{K_z} \left( \frac{\partial \bar{p}}{\partial \bar{z}} \right)^2 \right) \right] d\bar{z} \]  

(8)

where: \( \bar{T} \) - dimensionless oil film temperature, \( K_T \) - thermal coefficient.

The viscosity \( \bar{\eta} \) that has to be determined and the pressure gradients \( \frac{\partial \bar{p}}{\partial \bar{\phi}} \) or \( \frac{\partial \bar{p}}{\partial \bar{z}} \) are in the right side of equation (6) then both the temperature distribution, viscosity and pressure \( \bar{p}(\phi, z) \) as well the correlation coefficients \( K_x, K_z, \bar{r}_c \) occurring in the case of turbulent flow has to be determined by means of iteration [11].

Coefficient \( K_T \) results from transformation of energy equation [15]; thermal coefficient \( K_T = \omega \cdot \eta_0/(c_t \cdot \rho \cdot g \cdot T_0 \cdot \nu^2) \) where: \( c_t \) - specific heat of oil, (J/kgK), \( g \) - acceleration of gravity (m/s²), \( T_0 \) - temperature of supplied oil, (°C), \( \rho \) - oil density, (kg/m³). The developed code of computation [14, 15] allows obtaining the results for different geometrical and exploitation parameters of bearings. Integration of oil film pressure and temperature distributions gives the components of oil film force, which are the functions of co-ordinates \( (\epsilon, \alpha) \) and velocities \( (\dot{\epsilon}, \dot{\alpha}) \) [10]. Increments of the resultant force, after expansion in the Taylor series of the first order, are defined as the stiffness and damping coefficients [10, 14]. All the stiffness and damping coefficients were calculated by means of
perturbation method [14]. The stability of rotor-bearing system is analysed based on the characteristic frequency equation of 6-th order; this equation depend on the stiffness and damping coefficients, Sommerfeld number So, relative elasticity of shaft cs and the ratio of angular velocity to the critical angular velocity of stiff rotor. As result of transformations it was obtained the expression that determines the ratio of boundary angular speed \( \omega_b \) to the critical \( \omega_c \) one, and determines the stability of rotor [14]:

5. Results of calculations

The program developed by author allows obtaining the static and dynamic characteristics of wide family of journal bearings including the tilting-pad bearings. However, it would be necessary to modify this program for the floating ring journal bearings of multilobe profiles.

Exemplary results that can be obtained by application of the code of numerical computation are presented in ‘figure 6’ through ‘figure 15’; computed parameters are presented versus Sommerfeld number \( (S_o = F \cdot \psi^2 / (L \cdot D \cdot \eta \cdot \omega) \) with \( F \) - resultant oil film force (N). In all these results different bearings profiles as well as geometric (e.g. \( L/D \) – relative length of bearing, bearing relative clearance, lobe relative clearance \( \psi_s \), pericycloid relative eccentricity \( \lambda^* \)) and operational (rotational speed, supplied oil temperature \( T_0 \), angular length of groove \( \varphi_g \), oil viscosity) were used.

‘Figure 6’ shows the oil film pressure and temperature distributions of cylindrical (‘figure 6a’) and three-lobe (‘figure 6b’) journal bearing. Maximum value of temperature corresponds the peripheral coordinate of the end of oil film pressure distribution – this is characteristic for the diathermal oil film and it was verified in experimental investigation [13,14].

![Figure 6](image-url)

**Figure 6.** Exemplary oil film pressure and temperature distribution in cylindrical (a) and three-lobe (b) journal bearings for different relative eccentricities of journal \( \varepsilon \) and diathermal model of oil film

One of the most important parameters of bearings is the maximum oil film temperature \( T_{max} \) – it is presented in ‘figure 7’ and ‘figure 8’; there is a difference in these temperatures for cylindrical and three-lobe journal bearings (‘figure 8’).

Friction losses \( F_f \) of two types of bearings for turbulent model of oil film are given in ‘figure 9’. The friction losses show an increase at the increase in turbulence as results of decrease in lubricant viscosity. It causes the increase in the Reynolds number, turbulence intensity and such situation generates the increase in the friction losses above the laminar oil film.

Figure 10’ presents the friction losses of three type of multilobe journal bearings; among these bearings the larges friction losses show four-lobe journal bearing and the smallest for the pericycloid profile of sleeve.

In ‘figure 11’ there are the friction losses of three-lobe classic and pericycloid bearings at different extent of oil grooves. The values of friction losses \( F_f \) decrease at the increase in the oil groove extent \( \phi_g \) (e.g. ‘figure 11a’ or ‘figure 11b’ - \( F_f \) at \( \phi_g =0^\circ \) and \( F_f \) at \( \phi_g =40^\circ \)).
Figure 7. Maximum oil film temperature of adiabatic and diathermal model of oil film

Figure 8. Maximum oil film temperature of cylindrical and three lobe journal bearing

Figure 9. Friction losses of three-lobe pericycloid pericycloid (3LP) and three-lobe (3L) journal bearings journal turbulent model of oil film

Figure 10. Friction losses of three-lobe (3LP), three- and four-lobe (3L, for 4L) bearings

Figure 11. Effect of oil groove on the friction losses of three-lobe and three-lobe pericycloid journal bearings
The stability ranges that were determined for the cylindrical journal bearing with the length to diameter ratio $L/D=0.5$ and two different values of bearing relative clearance are showed in ‘figure 12’ and ‘figure 13’. $(S_{ok} - \text{critical Sommerfeld number}, \omega_b/\omega_c - \text{relative elasticity of shaft, } c_s = f/\Delta R, f - \text{static deflection of shaft})$ the ranges of stability and angle $\tau$ determines the stability are larger at smaller values of bearing clearance. ‘Figure 14’ and ‘figure 15’ present the stability ranges of three-lobe pericycloid and three-lobe classic bearings, respectively.

\[ \omega_b/\omega_c \]

**Figure 12.** Stability chart for the cylindrical journal bearing ($T_0=23^\circ C$)

**Figure 13.** Stability chart for the cylindrical journal bearing ($T_0=23^\circ C$)

\[ \frac{K_T}{T_0} = 0.0278 \]

\[ L/D=0.5 \]

\[ v = 2.7 \% \]

\[ \tau = \frac{c_s}{0.1} \]

\[ c_s = \frac{0.2}{0.3} \]

\[ c_s = \frac{0.5}{1.0} \]

\[ c_s = \frac{20}{100} \]

**Figure 14.** Stability chart for the pericycloid three-lobe journal bearing

**Figure 15.** Stability chart for the multilobe journal bearing
6. Final remarks
The multilobe journal bearings with discontinuous and continuous (pericycloid) profile of bearing bore can be applied in the semi floating and floating ring bearings systems of turbochargers. Exemplary results for the chosen multilobe bearings support their application in high speed rotating machinery including the turbochargers. Lower temperatures of operation, lower maximum oil film temperatures are very convenient from the point of reliable and durable bearings systems. The friction losses of multilobe bearings as compare to the cylindrical bearings can cause some disadvantages in application of these bearings. However, the higher rotational speeds and better stability can compensate the friction losses. The design changes in the oil supply such as, e.g. the length and angular width of oil grooves can affect the friction losses towards their decrease. Code of numerical calculations that was developed by author allows the obtaining the static and dynamic characteristics of bearings that can be applied in bearing systems of turbochargers both small and large power engines. Layout of applied code and the results of this work allow calculating the bearing characteristics and stability of rotors operating in the bearings with the turbulent oil film and particularly the floating ring bearings.

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