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The analysis of the influence of structural parameters on static and dynamic properties of sliding bearings with a floating ring bearing

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Abstract

Turbochargers structure developments i.e. work of rotating sets in turbochargers with higher rotational speed imposes more on bearing knots. The recommended values of parameters in this paper were introduced to describe bearings with floating rings. The authors used computational examples to show the influence of presented ranges of a relative clearance of the inner bearing and quotients of radial clearances on work parameters of a sliding bearing with a floating ring such as: relative eccentricities, oil films maximum temperatures, oil films maximum pressures, oil films minimal heights, a journal velocity quotient and a floating ring as well as amplitudes of displacements of floating rings in bearing knots of a rotating set in a turbocharger.

Keywords: Turbocharger, rotating unit, oil film force, adiabatic model, eccentricity ratio.

Introduction

Turbochargers structure developments i.e. work of rotating sets in turbochargers with higher rotational speed imposes higher requirements on bearing knots (Fig.1). A bearing shaft described in Fig.1 is bearing mounted in lateral sliding bearings with a floating ring (Fig. 3). The right choice of structural characteristics of sliding bearings in an essential way influences the proper work of rotating sets. Therefore, we must use the appropriate design methods. Theoretical and experimental models constitute the base for these methods. These models should reflect very precisely the real conditions of work in a bearing.

Taking into account the results from the study of static and dynamic properties in the research Mazurkow (2009a) authors have recommended the following values of parameters which describe the geometry of sliding bearings with floating rings:

$$\psi_1 = \frac{C_{R1}}{R_1} = \frac{R_1 - R_{J1}}{R_1} \approx 4 - 5‰,$$ relative clearance of inner oil film,

$$C_R^* = \frac{C_{R2}}{C_{R1}} = \frac{R_2 - R_{J2}}{R_1 - R_{J1}} = 0{,}5 - 1{,}0,$$ quotient of radial clearance, (1)

$$B^* = \frac{B}{D_1} = \frac{B}{2 \cdot R_1} \le 0{,}5,$$ bearing bush relative width.

The aim of this research work is to use calculation examples and discuss the influence of presented ranges of a relative clearance of the inner bearing (ψ_1 and quotients of radial clearances (C^*_R) on work parameters of a sliding bearing with a floating ring such as: relative eccentricities (ε_1 & ε_2), maximum temperatures of oil films (T_{1max} & T_{2max}), maximum pressures of oil films (p_{1max}, p_{2max}), minimal heights of oil films (h_{1min} & h_{2min}), a velocity quotient of a journal and a floating ring ($\nu=\omega_1/\omega_2$) as well as amplitudes of displacements of floating rings in bearing knots of a rotating set in a turbocharger: (x_5, y_5, x_6 & y_6).

Dynamic model of a rotating set in a turbocharger

To deal with the subject the authors have developed the dynamic model of a rotating set in a turbocharger which is described in the Fig. 2. It has been accepted that rotors on either a turbine or a compressor side have different masses and different unbalance values. In this model each bearing is modeled including a floating ring bearing mass. Through rigidity and damping factors oil films capacity towards oscillations damping is also considered. The motion of the model is examined in two planes: 0XZ and 0YZ. Here, presented rigidity factors (c_x & c_y) and damping factors (d_x & d_y) are studied as coupled values.

A mathematical model of a presented physics model is constituted by motion equations which are written according to forces method for each mass of this model in planes OXZ and OYZ and can be expressed:

$$x_1 = -\alpha_{x11}\left[m_1\ddot{x}_1 - m_{n1}\delta_1\omega^2 \sin\omega t\right] - \alpha_{x12}\left[m_2\ddot{x}_2 + d_{x2}(\dot{x}_2 - \dot{x}_5)\right] - \alpha_{x13}\left[m_3\ddot{x}_3 + d_{x3}(\dot{x}_3 - \dot{x}_6)\right] - \\ - \alpha_{x14}\left[m_4\ddot{x}_4 - m_{n4}\delta_4\omega^2 \sin\omega t\right]$$
$$x_2 = -\alpha_{x21}\left[m_1\ddot{x}_1 - m_{n1}\delta_1\omega^2 \sin\omega t\right] - \alpha_{x22}\left[m_2\ddot{x}_2 + d_{x2}(\dot{x}_2 - \dot{x}_5)\right] - \alpha_{x23}\left[m_3\ddot{x}_3 + d_{x3}(\dot{x}_3 - \dot{x}_6)\right] - \\ - \alpha_{x24}\left[m_4\ddot{x}_4 - m_{n4}\delta_4\omega^2 \sin\omega t\right]$$
$$x_3 = -\alpha_{x31}\left[m_1\ddot{x}_1 - m_{n1}\delta_1\omega^2 \sin\omega t\right] - \alpha_{32}\left[m_2\ddot{x}_2 + d_{x2}(\dot{x}_2 - \dot{x}_5)\right] - \alpha_{x33}\left[m_3\ddot{x}_3 + d_{x3}(\dot{x}_3 - \dot{x}_6)\right] - \\ - \alpha_{x34}\left[m_4\ddot{x}_4 - m_{n4}\delta_4\omega^2 \sin\omega t\right]$$

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$$x_4 = -\alpha_{x41}\left[m_1\ddot{x}_1 - m_{n1}\delta_1\omega^2\sin\omega t\right] - \alpha_{x42}\left[m_2\ddot{x}_2 + d_{x2}(\dot{x}_2 - \dot{x}_5)\right] - \alpha_{x43}\left[m_3\ddot{x}_3 + d_{x3}(\dot{x}_3 - \dot{x}_6)\right] -$$
$$-\alpha_{x44}\left[m_4\ddot{x}_4 - m_{n4}\delta_4\omega^2\sin\omega t\right]$$
$$m_5\ddot{x}_5 + c_{x4}x_5 + d_{x4}\dot{x}_5 + c_{x2}(x_5 - x_2) + d_{x2}(\dot{x}_5 - \dot{x}_2) = 0$$
$$m_6\ddot{x}_6 + c_{x6}x_6 + d_{x6}\dot{x}_6 + c_{x3}(x_6 - x_3) + d_{x3}(\dot{x}_6 - \dot{x}_3) = 0$$
$$y_1 = -\alpha_{x11}\left[m_1\ddot{y}_1 - m_{n1}\delta_1\omega^2\sin\omega t + Q_1\right] - \alpha_{x12}\left[m_2\ddot{y}_2 + d_{x2}(\dot{y}_2 - \dot{y}_5) + Q_2\right] -$$
$$-\alpha_{x13}\left[m_3\ddot{y}_3 + d_{x3}(\ddot{y}_3 - \ddot{y}_6) + Q_3\right] - \alpha_{x14}\left[m_4\ddot{y}_4 - m_{n4}\delta_4\omega^2\sin\omega t + Q_4\right]$$
$$y_2 = -\alpha_{x21}\left[m_1\ddot{y}_1 - m_{n1}\delta_1\omega^2\sin\omega t + Q_1\right] - \alpha_{x22}\left[m_2\ddot{y}_2 + d_{x2}(\dot{y}_2 - \dot{y}_5) + Q_2\right] -$$
$$-\alpha_{x23}\left[m_3\ddot{y}_3 + d_{x3}(\ddot{y}_3 - \ddot{y}_6) + Q_3\right] - \alpha_{x24}\left[m_4\ddot{y}_4 - m_{n4}\delta_4\omega^2\sin\omega t + Q_4\right]$$
$$y_3 = -\alpha_{x31}\left[m_1\ddot{y}_1 - m_{n1}\delta_1\omega^2\sin\omega t + Q_1\right] - \alpha_{x32}\left[m_2\ddot{y}_2 + d_{x2}(\dot{y}_2 - \dot{y}_5) + Q_2\right] - \quad (2)$$
$$-\alpha_{x33}\left[m_3\ddot{y}_3 + d_{x3}(\ddot{y}_3 - \ddot{y}_6) + Q_3\right] - \alpha_{x34}\left[m_4\ddot{y}_4 - m_{n4}\delta\omega^2\sin\omega t + Q_4\right]$$
$$y_4 = -\alpha_{x41}\left[m_1\ddot{y}_1 - m_{n1}\delta_1\omega^2\sin\omega t + Q_1\right] - \alpha_{x42}\left[m_2\ddot{y}_2 + d_{x2}(\dot{y}_2 - \dot{y}_5) + Q_2\right] -$$
$$-\alpha_{x43}\left[m_3\ddot{y}_3 + d_{x3}(\ddot{y}_3 - \ddot{y}_6) + Q_3\right] - \alpha_{x44}\left[m_4\ddot{y}_4 - m_{n4}\delta_4\omega^2\sin\omega t + Q_4\right]$$
$$m_5\ddot{y}_5 + c_{y4}y_5 + d_{y4}\dot{y}_5 + c_{y2}(y_5 - y_2) + d_{y2}(\dot{y}_5 - \dot{y}_2) = Q_5$$
$$m_6\ddot{y}_6 + c_{y6}y_6 + d_{y6}\dot{y}_6 + c_{y3}(y_6 - y_3) + d_{y3}(\dot{y}_6 - \dot{y}_3) = Q_6$$

The detailed discussion concerning the algorithm and the method of solving eqns. (2) is presented in works (Mazurkow, 2008; Mazurkow, 2009a).

Adiabatic model of a sliding bearing with a floating ring bearing

Oil to both bearings on a compressor and a turbine sides is supplied through a feeding duct which is in the middle frame. The structure of tubules which provide fresh oil causes that bearings are fed independently with a constant temperature and a constant feeding pressure (T_z & p_z). The detailed description of a structure and parameters which describe work of sliding bearings with floating rings (Fig. 3) is discussed in researches (Buluschek, 1980; Mazurkow, 1993; Mazurkow, 2008).

The adiabatic model of a short bearing has been accepted for calculations in this paper. This model is described in a dimensionless way by the following equations:

Equations of pressures distribution in lubricant gaps (i= 1, 2):

$$\frac{\partial^2 p_i^*}{\partial z^{*2}} = 6\cdot\frac{\eta^*}{h_i^{*3}}\cdot\frac{\partial h_i^*}{\partial\varphi_i}\cdot\left(\frac{B}{D_i}\right)^2, \qquad (3)$$

where: $p_i^* = p_i\cdot\dfrac{\psi_i^2}{\eta_0\cdot(\omega_{Ji}+\omega_i)}$,

Equations of lubricant gaps shapes (i= 1, 2):

$$h_i^* = \frac{h_i}{C_{Ri}} = 1 - \varepsilon_i\cdot\cos(\beta_i - \varphi_i), \qquad (4)$$

Equations of temperature distribution in lubricant gaps (i= 1, 2):

$$C_{zi}(\varphi_i, z^*; \eta^*)\cdot\frac{\partial T_i^*}{\partial z^*} = A_{zi}(\varphi_i, z^*; \eta^*), \qquad (5)$$

Where:

$$A_{z1} = -\left(\frac{\partial p_1^*}{\partial\varphi_1}\right)^2 - \frac{12\cdot\eta^{*2}}{h_1^{*4}}\cdot\frac{(\omega_1\cdot R_1 - \omega_2\cdot R_2)^2}{R_2^2(\omega_1+\omega_2)^2} - \left(\frac{2\cdot R_2}{B}\right)^2\cdot\left(\frac{\partial p_1^*}{\partial z^*}\right)^2$$

$$A_{z2} = -\left(\frac{\partial p_2^*}{\partial\varphi_2}\right)^2 - \frac{12\cdot\eta^{*2}}{h_2^{*4}} - \left(\frac{2\cdot R_4}{B}\right)^2\cdot\left(\frac{\partial p_2^*}{\partial z^*}\right)^2,$$

$$C_{zi} = \left(\frac{2\cdot R_i}{B}\right)^2\cdot\frac{\partial p_i^*}{\partial z^*}$$

Equations of balance of powers and frictional moments:

$$F_{L1y}^* = F_1^*, F_{L2y}^* = F_2^*, F_{L1x}^* = 0, F_{L2x}^* = 0 \qquad (6)$$

Where:

$$F_1^* = \frac{2\cdot\psi_1^2}{\mathrm{B}\cdot\mathrm{R}_1\cdot\eta_0\cdot(\omega_1+\omega_2)}\cdot\mathrm{F}, \qquad F_2^* = \frac{2\cdot\psi_2^2}{\mathrm{B}\cdot\mathrm{R}_3\cdot\eta_0\cdot\omega_2}\cdot\mathrm{F}$$

Equation of frictional moments balance in a bearing,

$$M_2^* = M_3^*,$$

$$\alpha_2\int_{-1}^{1}\int_{0}^{2\pi}\left(\frac{1}{2}\cdot h_2^*\cdot\frac{\partial p_2^*}{\partial\varphi_2} + \frac{\eta^*}{h_2^*}\right)d\varphi_2 dz^* =$$
$$= \alpha_1\cdot\frac{\mathrm{R}_1}{\mathrm{R}_3}\int_{-1}^{1}\int_{0}^{2\pi}\left(\frac{1}{2}\cdot h_1^*\cdot\frac{\partial p_1^*}{\partial\varphi_1} + \frac{\eta^*}{h_1^*}\cdot\alpha_3\right)d\varphi_1 dz^* + \psi_1\cdot\varepsilon_1\cdot\sin\beta_1 \qquad (7)$$

Fig.1. A turbocharger with rotors: the centrifugal rotor of a compressor & the centripetal rotor of a turbine

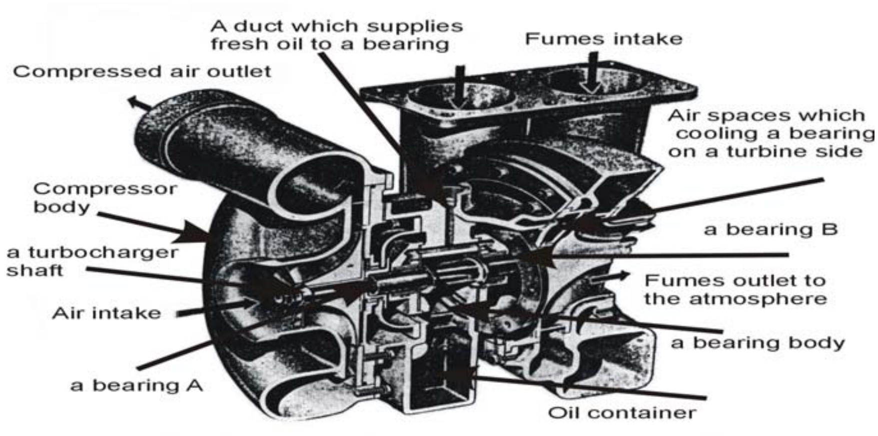

Fig. 2. Model of a rotating set

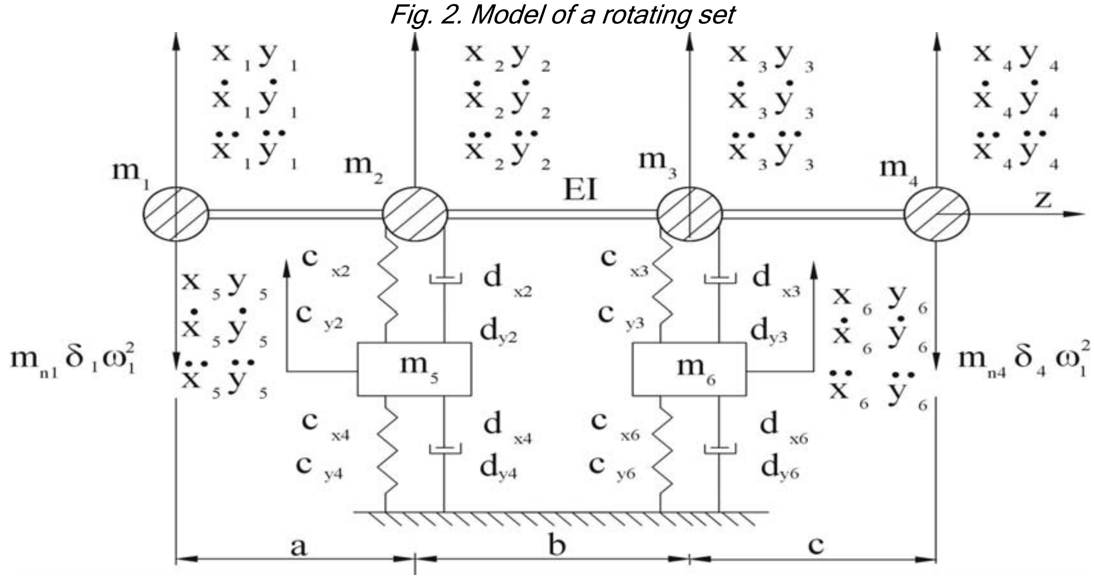

Fig. 3. A sliding bearing with a floating ring: 1-a floating ring bearing, 2-a fixed bearing bush, 3-holes through which oil is supplied to a bearing, 4-a circumferential groove, 5-directions of oil flow in a bearing

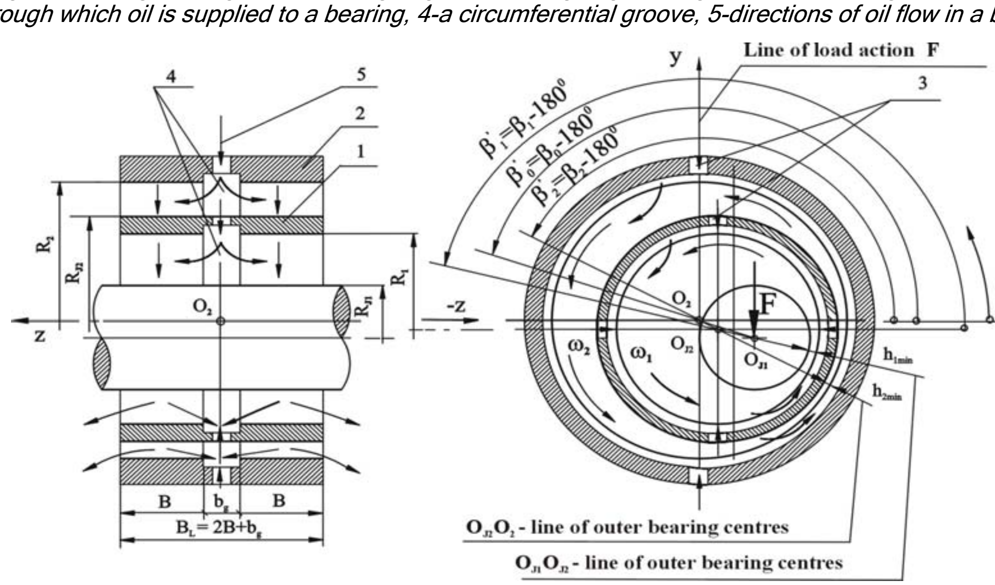

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Table 1. Given parameters in a short bearing model

Geometric parameters	Parameters concerning bearing work
R_{Ji} - radius of bearing journals,	F- bearing load,
R_i - inner radius of bearing bushes	ω_1- angular speed of bearing journal
	T_0- reference temperature
	T_z- feed oil temperature
	$\eta=\eta(T)$- oil dynamic viscosity
	c_p, ρ, λ- specific heat, density, oil conductivity.

Table 2. Influence of structural parameters on static & dynamic characteristics

Characteristics			
Static			Dynamic
$h_{1,2,min}\downarrow$	$T_{1,2\,max}\downarrow$	$p_{1,2max}\downarrow$	$A_p(x_5, x_6, y_5, y_6)\downarrow$
$\psi_i\downarrow$	$\psi_i\downarrow$	$\psi_i\uparrow$	$\psi_i\downarrow$
$C^*_R\downarrow$	$C^*_R\uparrow$	$C^*_R\uparrow$	$C^*_R\downarrow$
			$m_p=m_5=m_6\uparrow$
$B^*\downarrow$	$B^*\uparrow$	$B^*\uparrow$	$B^*\uparrow$
			$N_{w1}\downarrow$, $N_{w4}\downarrow$
$T_z\uparrow$	$T_z\downarrow$	$T_z\downarrow$	

Table 3. Given parameters for calculations of sliding bearings with floating rings

Given parameters	Structural tasks			
Journal rotational speed n_1 [rps]	300			
Journal radius R_{J1} [m]	$15{,}82\cdot10^{-3}$			
Relative width B^*	0,5			
Relative clearance of inner oil film ψ_1	4‰	4‰	5‰	5‰
Relative clearance of inner oil film ψ_2	1,7‰	3,4‰	2,1‰	4,2‰
Quotient of radial clearances C^*_R	0,5	1,0	0,5	1,0
Bearing load [N]	$F_{tu}=700$, $F_{sp}=600$			
Reference temperature T_0 [^{0}C]	20^0			
Feed oil temperature T_z [^{0}C]	30^0			
Feed oil pressure p_z [MPa]	0,1			
Floating ring bearing thickness [m]	$2{,}97\cdot10^{-3}$			
Oil type: $\eta(T)$ [Pa·s], $\rho(T)$ [kg/m^3], $c_p(T)$ [J/kg·^{0}C], λ [W/m·^{0}C],	$\eta(T)=\eta_o\cdot e^{a_\eta\cdot(T-T_0)+b_\eta\cdot(T-T_0)^2}$, $\rho(T)=a_\rho+b_\rho\cdot T+d_\rho\cdot T^2$, $c_p(T)=a_c+b_c\cdot T+d_c\cdot T^2$, $\lambda=0{,}145$, gdzie: $\eta_o=0{,}1084$, $a_\eta=-0{,}55291\cdot10^{-1}$, $b_\eta=-0{,}239\cdot10^{-3}$, $a_\rho=896{,}25$, · $b_\rho=-1{,}437$, $d_\rho=0{,}625\cdot10^{-2}$, $a_c=1802{,}1$, $b_c=2{,}878$, $d_c=0{,}87\cdot10^{-2}$			

Table 4. Given parameters for calculations of the rotating set

Concentrated masses in particular knots [kg]					
m_1	m_2	m_3	m_4	m_5	m_6
5,0	0,3	0,25	2,0	0,055	0,054
Geometric parameters of a rotating set					
a=0,055 [m]	b=0,075 [m]	c=0,045 [m]	$I_x=0{,}15\cdot10^{-6}$ [m^4]		
Young Modulus					
$E=1{,}915\cdot10^{11}$[N/m^2]					
Unbalance of rotating masses [N·s^2]					
$N_{w1}=0{,}18\cdot10^{-4}$	$N_{w4}=0{,}5\cdot10^{-5}$				

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Where calculations respectively amount to:

$$\alpha_1 = \frac{R_1 \cdot \eta_0 \cdot (\omega_1 + \omega_2)}{\psi_1} \cdot \frac{B}{F},\ \alpha_2 = \frac{R_3 \cdot \eta_0 \cdot \omega_2}{\psi_2} \cdot \frac{B}{F}, \alpha_3 = \frac{R_1 \cdot \omega_1 - R_2 \cdot \omega_2}{R_1 \cdot (\omega_1 + \omega_2)}$$

For non- working zone: $\varphi_i^p < \varphi_i < \varphi_i^k, \frac{\partial p_i^*}{\partial \varphi_i} = 0$

For working zone:

$$\varphi_i^k \le \varphi_i \le \varphi_i^p, \frac{\partial p_i^*}{\partial \varphi_i} = A_i \cdot \left(z^{*2} - 1\right) \cdot \left[\frac{\cos\varphi_i \cdot (1 - \varepsilon_i \cdot \cos\varphi_i)^3 - 3(1 - \varepsilon_i \cdot \cos\varphi_i)^2 \cdot \varepsilon_i \cdot \sin\varphi_i}{(1 - \varepsilon_i \cdot \cos\varphi_i)^6}\right]$$

The described above mathematical model of a bearing constitutes the coupled system of equations. This system of equations has been solved in a numerical way, the algorithm and the method of solving equations has been discussed in work (Mazurkow, 2009a).

Table 1 depicts accepted parameters.

The following resultant parameters are searched:

Distributions of pressure and temperature:

$$p_i^* = p_i^*(\varphi_i, z^*),\ T_i^* = T_i^*(\varphi_i, z^*)$$

- Structural characteristics such as:
- Angular speed of a floating ring (ω_2).
- Motion resistances in a bearing (μ_i).
- Oil consumption which flows through a bearing (Q_{boi}).
- Positions of centres' lines: a journal, a floating ring bearing, a fixed bearing bush ($\varepsilon_1, \varepsilon_2, \varepsilon_0, \beta_1, \beta_2, \beta_0$).
- Speed quotient of a bearing journal and a floating ring $\nu=\omega_1/\omega_2$.

Other models of a sliding bearing with a floating ring namely: the adiabatic model of a finite-length bearing, the isothermal model of a finite-length bearing and the isothermal one of a short bearing are described in these works (Buluschek, 1980; Domes, 1980; Krause, 1987; Mazurkow, 1993; Mazurkow, 2009a).

Table 5. Resultant parameters

Resultant parameters, turbine side F_{tu}=700 [N]				
Structural tasks	1: ψ_1= 4‰, C^*_R=0,5	2: ψ_1= 4‰, C^*_R=1,0	3: ψ_1= 5‰, C^*_R=0,5	4: ψ_1= 5‰, C^*_R=1,0
Amplitudes of displacements of a floating ring 10^{-6} [m]	x_5= 0,140, y_5= 0,250,	x_5= 0,380, y_5= 1,210,	x_5= 0,161, y_5= 0,409,	x_5= 0,350, y_5= 1,510,
Relative eccentricities	ε_1=0,29 ε_2=0,25	ε_1=0,25 ε_2=0,48	ε_1=0,42 ε_2=0,416	ε_1=0,41 ε_2=0,54
Oil films maximum temperatures [^{0}C]	T_{1max}=58 T_{2max}=64	T_{1max}=55 T_{2max}=55	T_{1max}=53 T_{2max}=59	T_{1max}=48 T_{2max}=55
Oil films maximum pressures [MPa]	p_{1max}=4,0 p_{2max}=2,5	p_{1max}=3,2 p_{2max}=2,6	p_{1max}=4,5 p_{2max}=3,4	p_{1max}=5,4 p_{2max}=3,6
Oil films minimal heights [mm]	h_{1min}=0,045 h_{2min}=0,024	h_{1min}=0,048 h_{2min}=0,034	h_{1min}=0,046 h_{2min}=0,023	h_{1min}=0,047 h_{2min}=0,036
Velocity quotient of a bearing journal & a floating ring	ν=5,151	ν=3,241	ν=5,541	ν=2,311
Resultant parameters, compressor side F_{sp}=600 [N]				
Structural tasks	1: ψ_1= 4‰, C^*_R=0,5	2: ψ_1= 4‰, C^*_R=1,0	3: ψ_1= 5‰, C^*_R=0,5	4: ψ_1= 5‰, C^*_R=1,0
Amplitudes of displacements of a floating ring bearing 10^{-6} [m]	x_6= 0,055, y_6= 0,190,	x_6= 0,230, y_6= 0,900,	x_6= 0,140, y_6= 0,353,	x_6= 0,360, y_6= 1,160,
Relative eccentricities	ε_1=0,25 ε_2=0,19	ε_1=0,22 ε_2=0,34	ε_1=0,36 ε_2=0,36	ε_1=0,31 ε_2=0,41
Oil films maximum temperatures [^{0}C]	T_{1max}=56 T_{2max}=68	T_{1max}=52 T_{2max}=58	T_{1max}=51 T_{2max}=63	T_{1max}=49 T_{2max}=55
Oil films maximum pressures [MPa]	p_{1max}=3,2 p_{2max}=2,2	p_{1max}=2,8 p_{2max}=1,8	p_{1max}=3,4 p_{2max}=3,6	p_{1max}=3,1 p_{2max}=1,6
Oil films minimal heights [mm]	h_{1min}=0,048 h_{2min}=0,026	h_{1min}=0,052 h_{2min}=0,042	h_{1min}=0,051 h_{2min}=0,025	h_{1min}=0,054 h_{2min}=0,046
Velocity quotient of a bearing journal & a floating ring bearing	ν=4,085	ν=2,211	ν=3,673	ν=2,171

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The analysis of the influence of structural parameters on static and dynamic properties of sliding bearings with floating rings bearing

The influence of structural parameters on static and dynamic characteristics is described in the Table 2. This kind of a description was for the first time presented for a sliding bearing with a cylindrical bush in a research work (Swiderski, 1995). The influence character of particular parameters on static and dynamic properties in slide bearings with a floating ring bearing was presented in research works (Mazurkow, 1993; Mazurkow, 2009a; Mazurkow, 2009b; Mazurkow, 2009c).

Table 2 shows that the influence of structural parameters on static and dynamic characteristics can be on one hand, contradictory e.g. the feed oil temperature and the minimal height of oil film or on the other hand, compatible e.g. the feed oil temperature and the oil film maximum temperature.

Calculation examples

To describe the influence of values of recommended parameters on work parameters of a bearing and a rotating set in a turbocharger we have accepted:

- Relative clearance of the inner oil film, ψ_1=4‰ , ψ_1=5‰,
- Quotient of radial clearances, C^*_R= 0, 5, C^*_R= 1, 0.

Other given quantities are presented in Tables 3 & 4. The calculations results are described in Table 5.

Conclusions

Taking into account the results from the study we can see the essential influence of geometric parameters of sliding bearings on work parameters of a rotating set in a turbocharger. Having examined the results it can be noticed:

- Lateral sliding bearings with a floating ring will work with the following values of a relative eccentricity : ε_1= <0,22; 0,42>, ε_2= <0,19; 054>.
- In all examined tasks maximum temperatures and maximum pressures will have values below acceptable values. For alloys e.g. on a Pb-base the range of their usage according to DIN 31652 amount from p_{lim}=16 MPa (T_{lim}= 100^{0}C) to 25 MPa (T_{lim}= 50^{0}C).
- Minimal height of lubricant films will assure a fluid friction, $h_{1,2min}$=<0,023; 0,054> mm.

  The smallest values of displacements amplitudes of floating rings bearing appeared in the task No.1, where a relative clearance of the inner bearing and the quotient of radial clearances respectively came to: ψ_1=4‰ and C^*_R=0,5. However, the biggest amplitudes of displacements were in the task No.4, for them given above values came to: ψ_1=5‰ and C^*_R=1, 0. The maximum, the six fold increase of amplitude of oscillations displacements appeared for a coordinate y_6. In the conclusion, the authors state that geometries of sliding bearings with floating rings have a significant influence on dynamic properties of rotating sets in turbochargers therefore; these geometries should be accepted for design calculations.

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