THE RESEARCH OF ROTATING SETS IN TURBOCHARGERS, THE INFLUENCE OF RELATIVE WIDTH OF SLIDING BEARINGS WITH A FLOATING RING ON STATIC AND DYNAMIC PROPERTIES

Mariusz Cygnar, Marek Aleksander

State Higher Professional School in Nowy Sącz
Staszica 1, 33-300 Nowy Sącz, Poland
e-mail: mcygnar@pwsz-ns.edu.pl

Aleksander Mazurkow, Grzegorz Budzik

Rzeszów Univeristy of Technology
Powstanców Warszawy Avenue 8, 35-959 Rzeszów, Poland
tel.: +48 17 8651640, fax: +48 17 8651150
e-mail: algarze@rz.onet.pl, budzik@prz.edu.pl

Abstract

This work is the continuation of the research concerning the dynamic model of a rotating set in turbocharger. In this model masses of rotors and a shaft have been modelled as concentrated masses. The rotating set has been propped on two supports forming lateral sliding bearings with a floating ring bearing. Each bearing is designed including floating ring bearing mass. The shaft of a rotating set spins at angular velocity $\omega_{1,}$ whereas a floating ring bearing spins at angular speed ω_2.

A mathematical model constitutes a system of differential equations, mutually coupled. The mathematical model has been solved by determining: acceleration, velocity and displacement in each knot. The paper shows the crucial influence of a bearing relative clearance on static and dynamic properties of a rotating set in a turbocharger. The authors suggest that the rotating set works stably only in these cases as for given parameters, we choose properly oil films' geometry. In the conclusion, the authors formulate proposal that might be useful for the designers who deal with these types of bearings.

Especially, the mathematical model of a turbocharger rotating set, work parameters of bearing knots, researches of static and dynamic properties are presented in the paper.

Key words: *turbocharger, rotating unit, sliding bearing with a floating ring bearing, rigidity and damping factor*

1. Introduction.

The author has paid attention to the model of the rotating set described in the Fig. 1. In this model a shaft is bearing mounted in two sliding bearings with a floating ring. Each bearing was modeled taking into account the mass of a floating ring. We have assumed that rotors in both a turbine and a compressor have different masses and different values of unbalance. Through rigidity and damping factors we measured the ability of both oil films to damp oscillations. The motion of the model is examined in two planes: OXZ and OYZ. Here presented rigidity factors (c_x, c_y) and damping factors (d_x, d_y) are studied as coupled values.

This work has described the influence of the relative width of bearing bushes ($B^*=2{\cdot}L/R_2$) on static properties of a rotating set.

2. Mathematical model of a turbocharger rotating set

The motion equations for a discrete model depicted in the Fig. 1 are written in the form of matrix in planes OXZ and OYZ:

$$
\begin{aligned}
&[M_x]\cdot[\ddot{x}]+[D_{xx}]\cdot[\dot{x}]+[D_{xy}]\cdot[\dot{x}]+[C_{xx}]\cdot[x]+[C_{xy}]\cdot[x]=[F_x(t)],\\
&[M_y]\cdot[\ddot{y}]+[D_{yy}]\cdot[\dot{y}]+[D_{yx}]\cdot[\dot{y}]+[C_{yy}]\cdot[y]+[C_{yx}]\cdot[y]=[F_y(t)],
\end{aligned}
\tag{1}
$$

where:
(M_x), (M_y) - matrixes of masses,
(D_{xy}), (D_{yx}), (D_{xx}), (D_{yy}) - matrixes of damping factors,
(C_{xy}), (C_{yx}), (C_{xx}), (C_{yy}) - matrixes of rigidity factors,
(F_x(t)), (F_y(t)) - matrixes of forces.

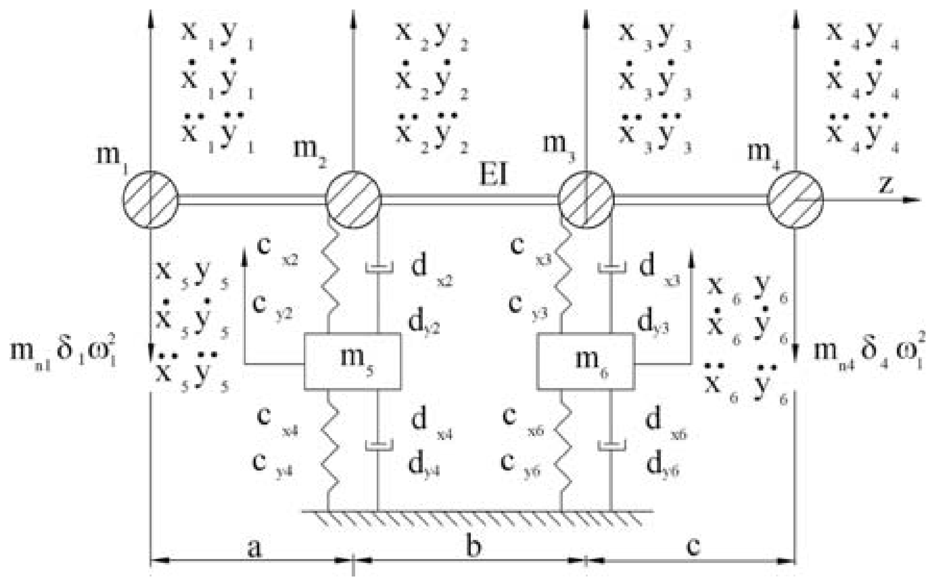

Fig.1 Model of a rotating set

The complete description of the dynamic model of a turbocharger is presented in the papers [2-3].

3. Work parameters of bearing knots

As it has been mentioned the shaft of the rotating set is proper on sliding bearings with a floating ring bearing. The constructional elements of a bearing are (Fig. 2): the fixed bearing bush (2), and the loosely fixed floating ring (1) separating the journal and the fixed bearing bush- known as the floating ring bearing. The oil is supplied into the outer and the inner bearings under the pressure through the holes (3) which are in the fixed bearing bush and the floating ring. The circumferential lubricant grooves (4) in the fixed bearing bush or the floating ring bearing provide regular oil feed to the lubricant gaps. The directions of oil flow in a bearing are shown in Fig. 2 (5).

Work parameters of a bearing are expressed by:

- Sommerfeld Number which for inner oil film equals:

$$S_{o1}=\frac{\eta(\omega_1+\omega_2)\cdot D_1B}{F}\left(\frac{R_1}{C_{R1}}\right)^2, \tag{2}$$

- Sommerfeld Number which for outer oil film equals:

$$S_{o2}=\frac{\eta\omega_2D_2B}{F}\left(\frac{R_3}{C_{R2}}\right)^2, \tag{3}$$

- relative eccentricities ε_1, ε_2,
- angular velocity of a floating ring ω_2 and the quotient $\nu = \frac{\omega_2}{\omega_1}$,
- displacement amplitudes in bearing supports: x_5, x_6, y_5, y_6.

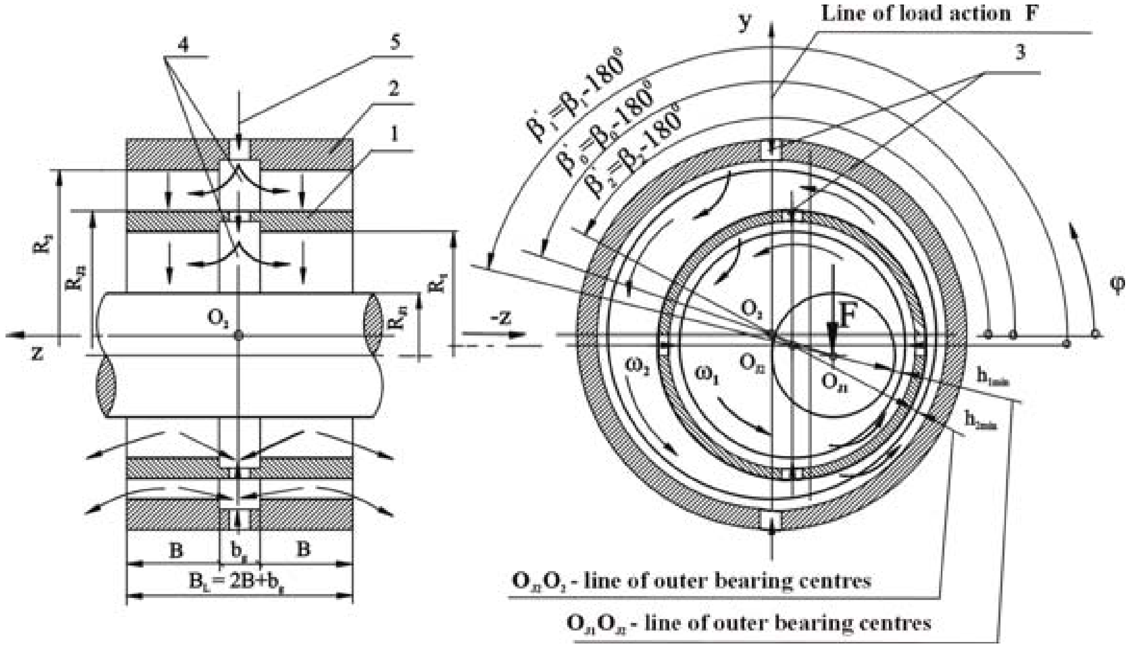

Fig.2. The sliding bearing with the floating ring: 1- the fixed bearing bush, 2- the floating ring, 3- the holes in the fixed bearing bush and the floating ring,4- circumferential lubricant grooves in the fixed bearing bush or the floating ring, 5- directions of oil flow in a bearing

4. Researches of static and dynamic properties

The analysis of static and dynamic properties of the rotating set in Fig. 1 has been carried out for given parameters described in Tab. 1.

Tab. 1. Given parameters from calculations of the rotating set

Concentrated masses in particular knots [$N\cdot s^2/m$]					
m_1	m_2	m_3	m_4	m_5	m_6
5.0	0.3	0.25	2.0	0.055	0.055
Bearing loads in knots: (F_{w2}, F_{w3}) [N]					
F_{w2}=700, F_{w3}=500					
Radial clearances in bearings, in knots: $(C_{R1}, C_{R2})_{w2}$, $(C_{R1}, C_{R2})_{w3}$ [m]					
$(C_{R1})_{w2,w3} = (C_{R2})_{w2,w3} = 0.05\cdot10^{-3}$,					
Journal rays for inner and outer bearings in knots: $(R_{J1}, R_{J2})_{w2}$, $(R_{J1}, R_{J2})_{w3}$ [m]					
$(R_{J1})_{w2,w3} = 15.82\cdot10^{-3}$			$(R_{J2})_{w2,w3} = 19.03\cdot10^{-3}$		
Bearing bushes relative widths in knots: $(B^*)_{w2}$, $(B)_{w3}$ [m]					
$(B^*=2\cdot B/R_2)_{w2,w3}$ = 0.25; 0.3; 0.4; 0.5; 0.6; 0.65; 0.7; 0.8; 0.9; 1.0					
Geometric parameters of rotating set					
a=0.055 [m]	b=0.075 [m]		c=0.045 [m]	$I_x=0.15\cdot10^{-6}$ [m^4]	
Material constants					
$E=1.915\cdot10^{11}$[N/m^2]			η=0.028 [Pa·s]		
Rotational speed of a shaft [rps]					
N_1=500					
Unbalance of rotating masses [$N\cdot s^2$]					
$N_{w1}=0.18\cdot10^{-4}$			$N_{w4}=0.5\cdot10^{-5}$		

For given parameters which are above described in the Tab. 1 we have determined: Sommerfeld Number (S_1, S_2), relative eccentricities (ε_1, ε_2), the quotient of angular velocities of a floating ring and a journal ($v = \omega_2/\omega_1$), we checked stability as well as calculated amplitudes of displacements in bearing knots for stable structural cases (x_5, x_6, y_5, y_6). The results of this study are given in Fig. 3-9.

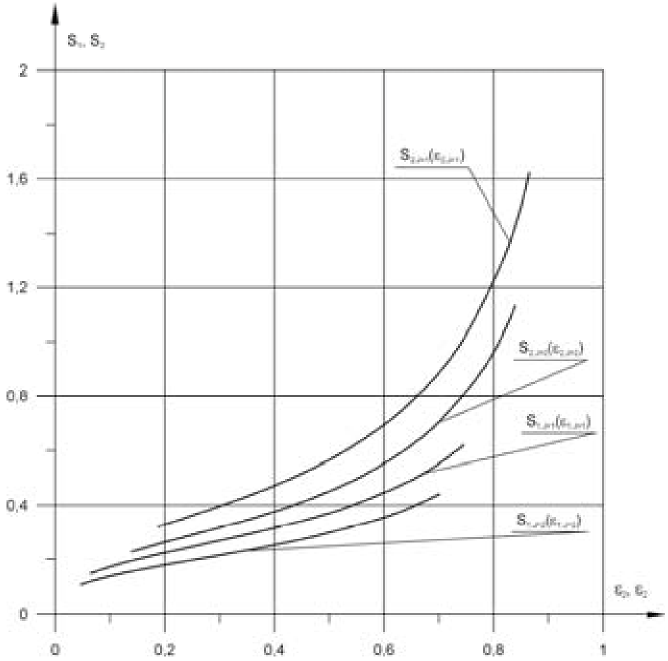

Fig. 3. The influence of relative eccentricities on Sommerfeld Numbers

a) b)

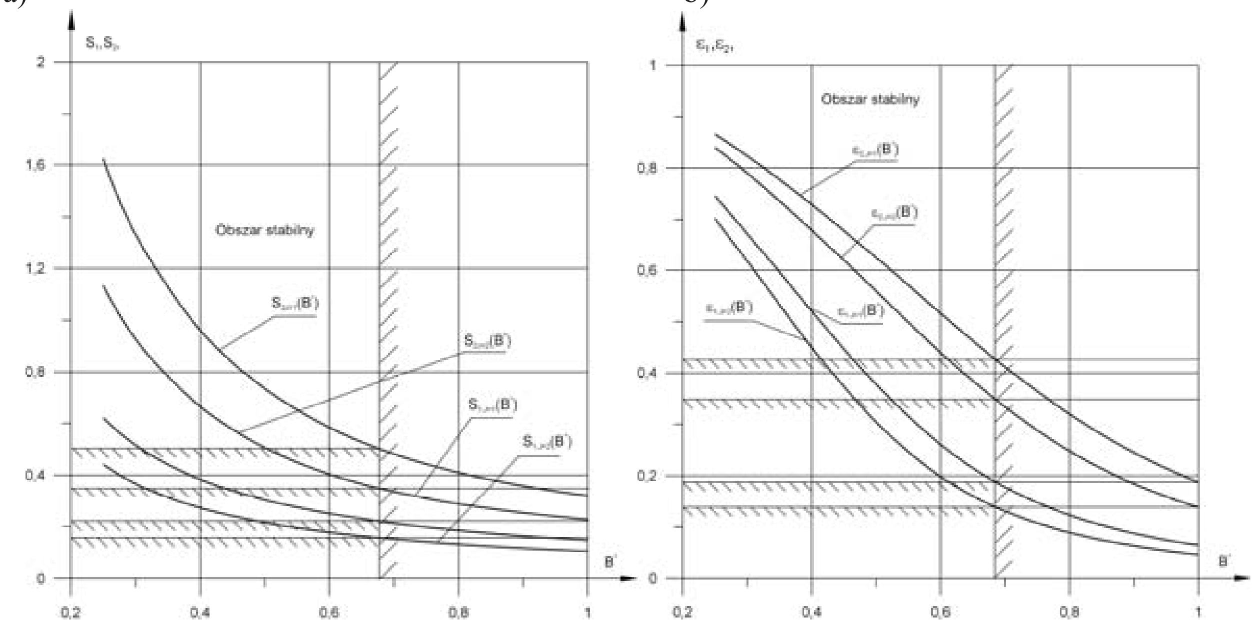

Fig.4. Stable work ranges, the influence of relative width of bearing bushes on: a) Sommerfeld Numbers, b) relative eccentricities

As we can see in the Fig. 3, when relative eccentricities increase the values of Sommerfeld Numbers also go up. Sommerfeld Numbers for both oil films are bigger in a turbine for which we set greater loads of a compressor.

As Sommerfeld Numbers (Fig. 4) grow up, the quotient (v) decreases. The growth of relative width (B) causes the rise of the quotient (v) due to stronger resistances in a bearing.

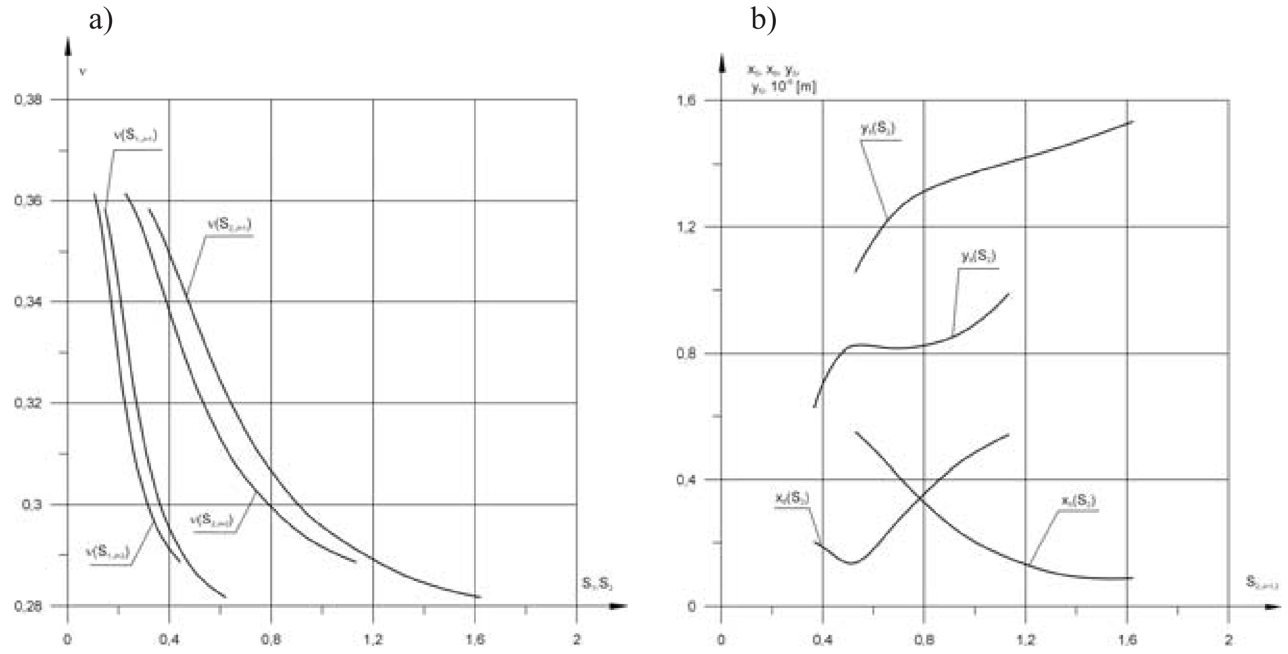

Fig. 5. a) The velocity quotient of the bearing journal and the floating ring in Sommerfeld Number function, b) The velocity quotient of the bearing journal and the floating ring in relative width of bearing bushes function

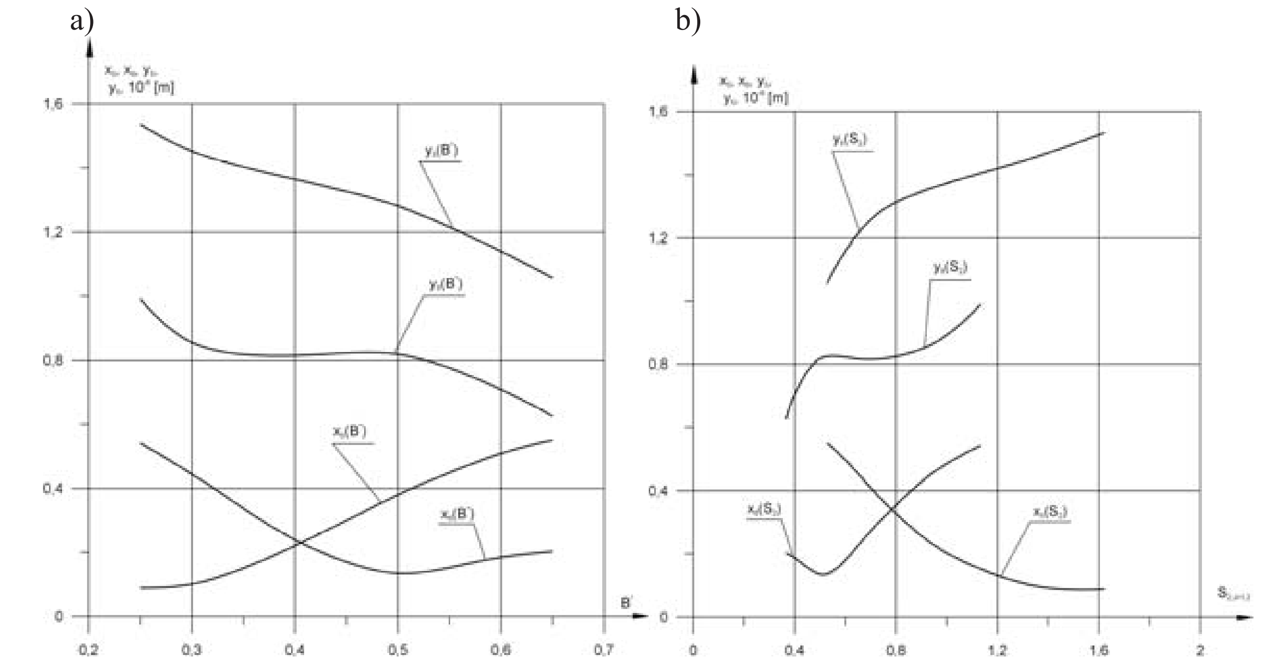

Fig. 6. a) Amplitudes of displacements in bearing knots in the function of relative width of bearing bushes, b) Amplitudes of displacements in bearing knots in Sommerfeld Numbers function

The rotating set (Fig. 5) stops working steady for the relative width value $B^{*} \geq 0.68$. For such given parameters the rotating set will work stably for the following Sommerfeld Numbers: $S_{2,J=2} \geq 0.37$, $S_{2,J=1} \geq 0.53$, $S_{1,J=2} \geq 0.163$, $S_{1,J=1} \geq 0.229$ and relative eccentricities $\varepsilon_{2,J=2} \geq 0.44$, $\varepsilon_{2,J=1} \geq 0.37$, $\varepsilon_{1,J=1} \geq 0{,}2$, $\varepsilon_{1,J=2} \geq 0{,}15$.

When Sommerfeld Numbers increase (Fig. 6) the amplitudes of displacements go up x_6, y_5, y_6 while the amplitude x_5 goes down. We should notice that displacements towards the coordinate y are bigger than displacements towards the coordinate x. When the relative width grows the amplitudes of displacements decrease x_5, y_5, y_6 however, the amplitude x_5 increases.

5. Conclusions

Taking into account the results of this study, we can see the significant influence of Sommerfeld Numbers and relative eccentricities on work stability of the rotating set. When Sommerfeld Numbers and relative eccentricities increase but precisely when they exceed minimal boundary values, the rotating set starts working steady.

Also, relative width of bearing bushes has the crucial influence on the oscillation level. The rotating set stops working stably for bigger values (B^*).

To obtain the lowest oscillation level of the rotating set we should make the compromise between bearings load and the relative width. Having analyzed the results of this work, we suggest that the relative width should equal $B^* \approx 0.5$.

6. References

[1] Kaniewski, W., *Warunki diatermicznego filmu smarnego*. Zeszyty naukowe Politechniki Łódzkiej. Zeszyt specjalny, Z. 14, 1977.

[2] Mazurkow, A., *Niepublikowane materiały Katedry Konstrukcji Maszyn Politechniki Rzeszowskiej*, Rzeszów 2007.

[3] Mazurkow, A., *The research of dynamic properties of rotating units in turbochargers.* Scientific Problems of Machines Operation and Maintenance, Vol .43, 2(154), Radom 2008.