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DEVELOPMENT OF PROJECT RECOMMENDATIONS TO PREVENT AUTOOSCILLATIONS AND VIBRATION OVERLOAD OF THE ROTARY MACHINE WORKING MEMBER

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Abstract: There were developed ways to minimize the rotor vibration activity in the plain bearing by readjusting design parameters, which eliminate the reasons of generating powerful low-frequency oscillations, imbalance position of rotor inertia axis, reduction of rotor oscillation amplitude, prevention of the loss of stability of the steady motion of rotors while maintaining bearing operating parameters. It has been proved that reduction of dynamic eccentricity is possible by increasing the oil angle, increasing the lifting force and reducing the rotor precession angle to 15-30 degrees. It has been defined that the most efficient way to prevent autooscillations is to increase the parameters of the curved oil wedge through selecting value ranges of relative clearance which is used while bearing designing or maintenance due to limitation of the range recommended by the stability nomogram.

KEYWORDS: rotary machine, theory of oscillation, rotor, rotor stability, dynamic balance stability loss

1 Introduction

Practical use of rotary machines (especially powerful ones) with hydrodynamic supports of the working member (plain bearings) developed on the contemporary experience and regulatory and procedural basis of design and development shows that rotor stability is provided by theoretically proved and tested geometrical relations of plain bearing and its operation conditions. These relations prove that in certain zones of the stability nomogram recommended to avoid spontaneous generation of autoocillations stability of the rotation axis is not achieved, as shown in Figure 1 [1]. This conditions the necessity to define them and apply in practice.

Choice and design of plain bearings ensure the stable operation of the rotary machine when its oscillation activity is minimal [2,3]. The rotor enters its equilibrium position area when it reaches operating modes, as shown in Figure 2 [1,4]. Figure 2 (b) shows the area of the rotor axis stable position.

When the rotor axis is at O_1, it means that the rotor axis is in its stable position. This happens when the rotary machine achieves its operating parameters.

Deviation of the rotor inertia axis (coordinates of O_1) can be determined by φ_a – the inclination angle of the load vector (lines of the centers, see Figure 2(c)). This parameter depends on the ratio of parameters (see Table 1) of the dynamic eccentricity – e (parameters of

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the hydrodynamic friction conditions) which causes occurrence of the centrifugal force $F=M\,e\,\omega^2$ and, as a result, excitation of mechanical oscillations with the frequency $f=\frac{n}{60}$.

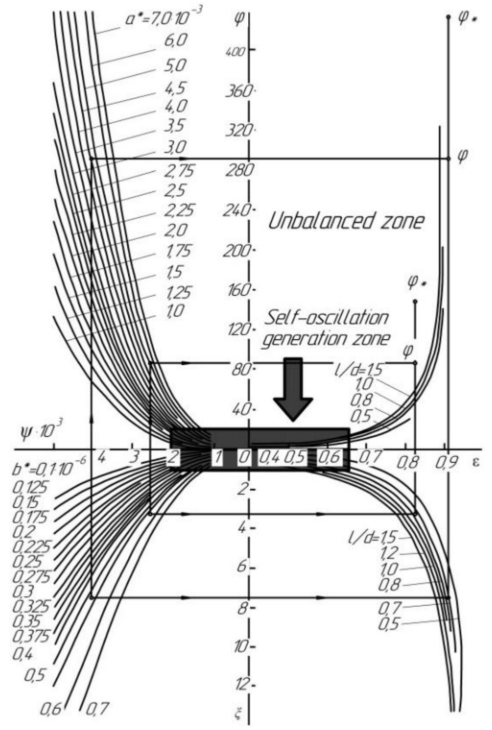
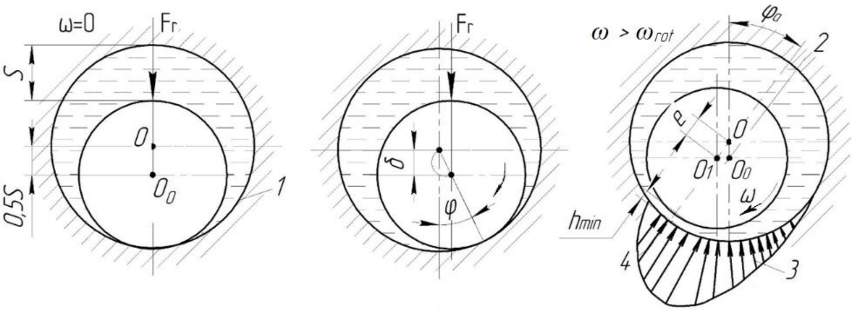


Fig. 1 Parameters of the bearing design and operating mode to determine the rational area of rotor position stability

The oscillation amplitude depends significantly on the quality of the working member balancing adjustment.



(a) (b) (c)

Fig. 2 Rotor positions in the plain bearing in different phases of the operating mode: (a) position when the rotor stops moving; (b) position when the rotor begins moving; (c) the stable position at operating mode

During the operation of rotary machines (especially powerful ones), there occur vibrations caused by the loss of stability of the steady motion of rotors in hydrodynamic plain bearings [2,5,16]. At the same time, parameters and operating modes of the bearing conform to the current methods of plain bearing design.

The phenomenon of instability of the shaft and bearing center line position, instability of angles β, dynamic eccentricity *e* (see Table 1) in the working support with the increased intensity of rotor oscillations is practically understudied. The intensity of rotor self-oscillations can reach such great values that their amplitudes will exceed resonance amplitudes of highly imbalanced rotors [3,6,10,11,13].

The article aims to the theoretical explanation of conditions of the rotor dynamic balance stability loss, critical deviation of the rotor inertia axis (eccentricity) from bearing geometrical axis, increased vibration activity, petterns of shaft autooscillation generation in the bearing within the designed zones that correspond to the parameters of balanced area of operation, temperature, load, rotor rotation velocity, increased uncontrolled imbalances [12,14,15,17-19].

These patterns and factors should be sorted out in the bearing design methodology, namely choice of bearing design parameters that consider conditions of rotor axis (inertia axis) stability maintenance within the working loads.

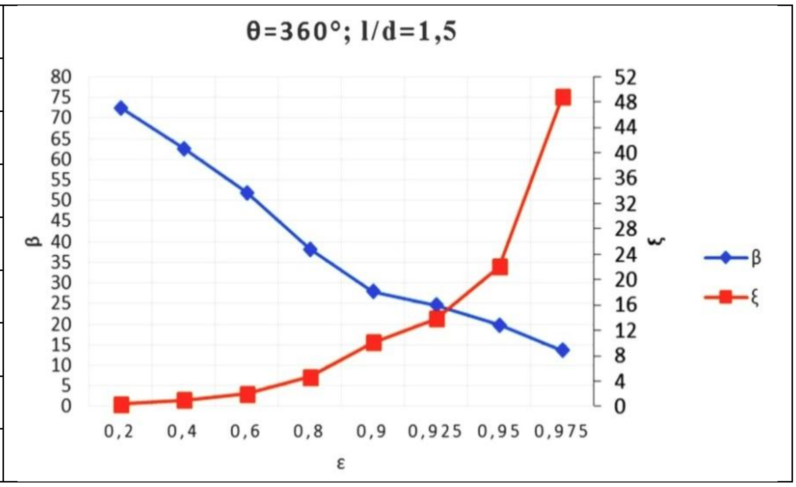
2 Methods and material

In modern rotary machines (especially powerful ones) with hydrodynamic supports of the working member (plain bearings) the designed position of the rotor axis in the bearing is determined by the set of design parameters which ensure the balanced rotation of the rotor [2,5,12,13].

Patterns of the rotor position are determined by the parameters of Table 1 that demonstrates the influence of the combination of design and mode factors on the balanced position of the rotor, where e is relative eccentricity of the rotor axis; β is the angle of balanced position of load vector F and the center line OO_1; ζ ψ is bearing capacity ($\zeta = \frac{F\psi_{ef}^2}{dl\mu_{ef}\omega}$).

Table 1 Rotor axis position parameters

№	e	β	ζ
1	0.2	72.5236	0.4273
2	0.4	62.6588	1.0005
3	0.6	51.9692	1.9724
4	0.8	38.1601	4.6824
5	0.9	27.9610	10.1382
6	0.925	24.6184	13.8256
7	0.95	19.8007	22.0440
8	0.975	13.5971	48.8429



Results of the experimental study of vibration characteristics (see Table 2) [7] carried out on classical balanced rotary machines (centrifugal compressor K-1500, see Figure 3) show that inertia axis position is not constant (see Figure 4(a)). Horizontal vibrations are predominant in rotors. This used to be explained by the fact that horizontal stiffness of the machine is less than vertical one, but the excitation oscillating force is horizontal although such a mode is acceptable with allowable actual levels of oscillation activities (see Table 2).

Under steady load modes, the rotor axis is not in the stable position and has complex motion trajectories relative to the dynamic balance point. This results in significant vibration loads which differ from the design ones as evidenced by hodographs of rotor oscillations in the plane of the supports (see Figure 4(a) and (b)).

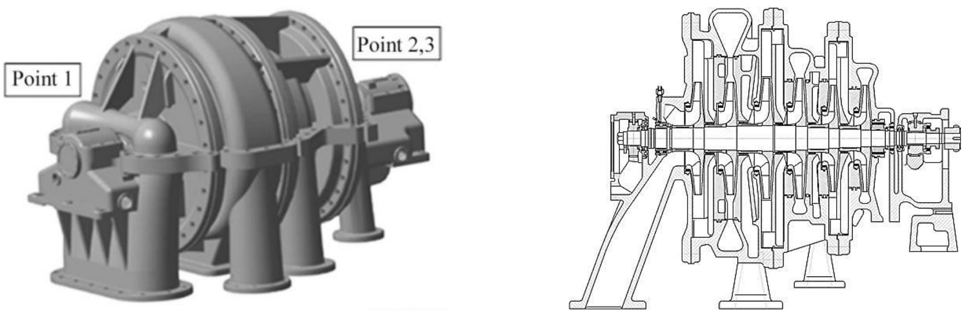


Fig. 3 Centrifugal compressor K-1500

Table 2 Results of measuring shaft vibration velocity and displacement

№ supports	V, mm/sec rms value						A, mkm (peak to peak)					
	vertical	horizontal right	horizontal left	axial	support left	support right	vertical	horizontal right	horizontal left	axial	support left	support right
Point 1	0.630	1.031	1.096	0.843	0.721	0.803	8.99	11.014	9.31	12.48	6.57	7.652
Point 2, 3	0.49	1.553	1.263	0.98	0.836	0.797	5.596	13.329	12.41	14.47	12.964	8.251

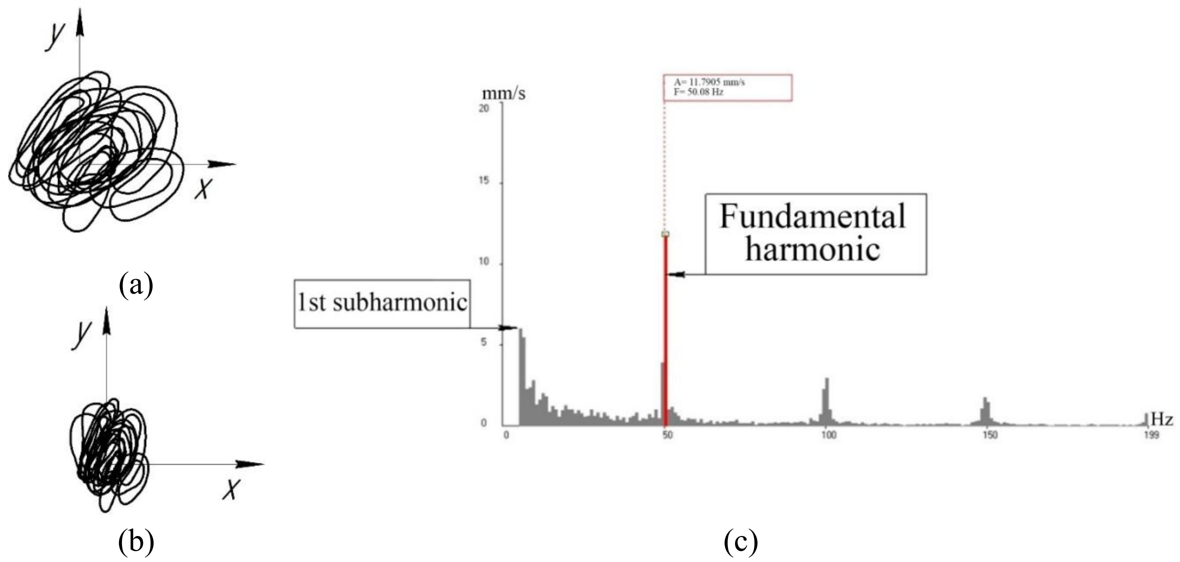
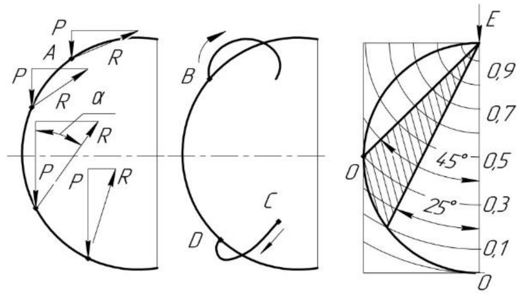


Fig. 4 Experimental determination of hodographs and oscillation frequency content: (a) front support oscillation hodograph; (b) rear support oscillation hodograph; (c) oscillation frequency content

Hodographs of rotor axis oscillations (see Figure 4(a) and (b)) have an extremely complex trajectory and suggest some oscillation activity even during the operation within the guaranteed balanced area (see Figure 1) with the possibility to switch over to cylindrical half-speed vortex [8, 9].

To explain the established mode of shaft motion it is necessary to re-evaluate the impact of factors of conditions and stability limit of the shaft balanced state.

Unit rotors rested on plain bearings with axis at O_1 perform the oscillation motion (see Table 2) [7] with respect to the balance point O_1, as shown in Figure 2 (c). This is due to the fact that the oscillation motion amplitude is affected by the value of dynamic eccentricity formed when the rotor switches from the point O_0 to the point O_1 and driven by a certain set of design and mode factors (see Table 1) even in case of no residual unbalance of the rotor. Here, the diagnostic feature of the oscillation motion of the kind (see Figure 4(c)) [7] is predominance of amplitudes of the fundamental harmonic in the oscillation spectrum. Such rotor behavior is conditioned by the cycling of position parameters of the load vector (β. F_H. F_V). Actual position of the point O_1 is conditioned by the shaft axis travel to the point of the balanced state along the trajectory determined by Gumbel semicircle (see Figure 5) [8] where the angle β is regulated by laws of the mode and design factors. It influences the rotor behavior and specifies the laws of its precession, as shown in Table 1 [8].



(a) (b) (c)

Fig. 5 Trajectories of the shaft axis travel when the rotor reaches the operating mode (Gumbel semicircle): (a) dependence of the oil layer stiffness and the shaft position in the bearing; (b) cyclic vortex movements of the shaft; (c) return of the shaft to equilibrium state

These oscillations (see Table 2) are due to oscillations of the load vector angle β (the angle between the balanced position of the load vector F and the line of the centers OO_0), as shown in Figure 6 [2].

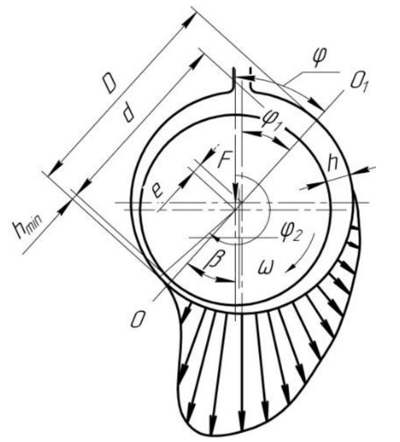


Fig. 6 Deviation of the rotor inertia axis from the support geometrical axis due to peculiarities of the hydrodynamic mode of rotor rotation

$$\beta = arctg \frac{\int_{\varphi_1}^{\varphi_2} \int_0^l p \cdot sin\varphi \cdot d\varphi \cdot dz}{\int_{\varphi_1}^{\varphi_2} \int_0^l p \cdot cos\varphi \cdot d\varphi \cdot dz} \tag{1}$$

where p is the value of specific pressure (a variable parameter on the trajectory of the rotor axis travel from the point O_0 to the point O_1) on the bearing surface from acting loads (change of the specific pressure on the bearing is possible at changes of the rotary machine operating loads as well); φ is the angle determining the geometrical parameters of the rotor stability that depends on design and mode parameters:

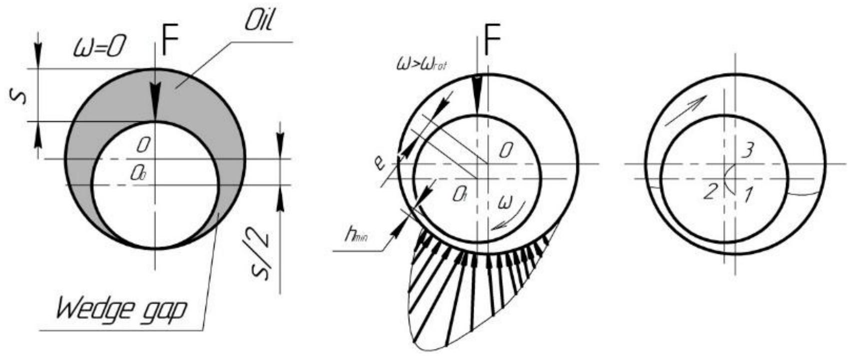
$$\varphi = \frac{2M \cdot \omega \cdot \Psi^3}{l \cdot \mu} \tag{2}$$

where M is the mass, kg; μ is dynamic viscosity of the lubricant, Pa·s; ω is angular velocity; ψ is effective (estimated eccentricity) relative bearing clearance (ψ = Δ/d), %; l is the length of the bearing along the generatrix, mm.

The rotor axis travel from the point O_0 to the point O_1 leads to reduction of vertical and raise of the horizontal component F that explains growth of horizontal oscillations of the rotor axis, as shown in Table 2.

3 Results and discussion

The level of rotor oscillations is proportional to the angle β which in turn depends on p, ω, ψ. When the value ψ is minimal, the oil wedge and its lifting capacity are minimal as well (see Figure 7(a)). When ω increases, the bearing capacity of the oil layer increases and reaches the required level when the shaft is displaced in the direction of the rotation. The angle of the oil wedge is increased automatically and the corresponding lifting force is provided, as shown in Figure 7(b). At the same time, the machine vibration increases due to the dynamic eccentricity growth.



(a) (b) (c)

Fig. 7 Oscillation displacement of the shaft centre position relative to the plain bearing centre: (a) boundary lubrication mode; (b) mixed lubrication mode; (c) trajectory of the shaft center displacement

When the rotor reaches the operating mode, the velocity changes and the load on the bearing of the shaft centre position will displace towards the bearing centre. The bearing running clearance will change from 0 to h_{min}, as shown in Figure 7(b). Then if the bearing load and velocity are constant, the centre of the rotor traces out a trajectory of its displacement in the bearing in point 2, as shown in Figure 7(c). Numbers 1, 2, 3 in the figure show the characteristic points of the trajectory of movement of the rotor rotation centre [6] at different design parameters of the bearing.

The position of point 1 corresponds to the maximum vertical loading (see Figure 2(a)) when the bearing capacity reserves are not realized and the rotor may start touching the insert while rotating.

The section from point 1 to point 2 is the transition to the area of dynamic balance and steady operation of the bearing when the force response of the oil layer to excitation loading impulse is proportional to the value of this impulse. Such mode of the rotary machine operation is possible for bearings with design parameters in the area of approximation to ψ_{max} within the stable balanced operation (see Figure 1). Oscillation activity depends on the value of dynamic eccentricity and the working member mass considering the residual imbalance value and has mechanical oscillations frequency equal to $f=\frac{n}{60}$.

For bearings with design parameters at the area ψ_{min} (see Figure 1), rotors can lose stability and enter the autooscillation mode (see Table 2). This is most common when the value β is the highest, due to various reasons including low ψ values, the insufficient lifting force of the oil wedge (small oil angles).

For these bearings at poit 2 (see Figure 7(c)) vertical load on the plain bearing is reduced (see Figure 5) and the axis starts travelling to point 3 due to oil layer wedging increase when the angle β grows. In this case the lifting force of oil layer exceeds the required one and so called "readjustment" [6] occures in the bearing response to external excitation. Thus, readjustment determines negative curvature of a rotor center trajectory. For instance, a unit force impulse causes the bearing response within two revolutions explaining occurrence of the first oscillation subharmonic, as shown in Figure 4(c).

The rotor moves back but travels along the trajectory farther than necessary (see Figure 4(a)). Further on, the rotor which has travelled beyond the balance point is exposed again to the excess impulse from the oil layer of the bearing acting in the direction of the set mode point. As a result, the rotor travels beyond the balance point to the initial point or even farther. This phenomenon leads to endless aoutuoscillations of the rotor which rests upon the oil wedge with "oil waves" travelling form input to output. Such types of oscillations cause low frequency vertical and radial vibrations (see Figure 4(b)).

The dynamic responce of the oil layer to load fluctuation that determines the trajectory inflection at point 2 is of great importance for understanding the processes inside the bearing. The curve of the shaft axis travel to and beyond inflection point 2 has different (positive and negative) values of load acceleration.

Significant increase of the first subharmonic means that the shaft enters the cascade mode of motion in the bearing at frequency equaling the first oscillation subhurmonic. This phenomenon is still understudied. Increase of loads on supports and rotor parts and other consequences requires further studying as well, since the power cycle lasts two shaft rotations, as shown in Figure 7(c).

Unit rotors with the increased level of the first subhurmonic oscillaions lose stability and enter the mode of radial autooscillations due to oscillations of rotor shaft load vectors (within two periods of rotation) that can result in different vibration loads.

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CONCLUSION

Stability nomogram application (see Figure 1) where rotor stability is provided by the theoretically proved and tested geometrical relation of plain bearing and its operating conditions has some risks of autooscillations and vibration overloads in the pre-resonant zone.

When designing plain bearings using stability nomograms (see Figure 1), reduction of the dynamic eccentricity e is the only way to support the rotor dynamic balance stability by minimization of deviation of the rotor inertia axis (eccentricity minimization) from the bearing geometrical axis within the ranges of rotor rotation load and velocity.

The easiest way to reduce dynamic eccentricity e is to increase the oil angle, its lifting force, reduce β to 15-30° due to limitation of the range $\psi_{min} < \psi < \psi_{extr}$ till $e_{min} = F(\Delta\ \psi)$ recommended by the stability nomogram (within the area of plain bearing balanced operation).

The most efficient way to prevent autooscillations is to increase the parameters of the curved oil wedge through selecting value ranges of relative clearance ψ ($4.6 \cdot 10^{-5}\sqrt{\lambda}\ \sqrt{\frac{l}{d}}$ ÷ $9.2 \cdot 10^{-5}\sqrt{\lambda}\ \sqrt[3]{\frac{l}{d}}$; where λ is the mode characteristcs equal to $\mu\frac{n}{k}$; μ is oil viscosity; n is rotation frequency; k is specific pressure on the bearing surface), which is used while bearing designing or maintenance.

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