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# Rotor Rolling over a Water-Lubricated Bearing

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**Abstract**—The article presents the results of studying the effect of forces associated with secondary damping coefficients (gyroscopic forces) on the development of asynchronous rolling of the rotor over a water-lubricated bearing. The damping forces act against the background of other exciting forces in the rotor–supports system, in particular, the exciting forces of contact interaction between the rotor and bearing. The article considers a rotor resting on supports rubbing against the bearing and the occurrence of self-excited vibration in the form of asynchronous roll-over. The rotor supports are made in the form of plain-type water-lubricated bearings. The plain-type bearing's lubrication stiffness and damping forces are determined using the well-known algorithms taking into account the physical properties of water serving as lubrication of the bearing. The bearing sliding pair is composed of refractory materials. The lubrication layer in such bearings is thinner than that used in oil-lubricated bearings with white metal lining, and there is no white metal layer in water-lubricated bearings. In case of possible deviations from normal operation of the installation, the rotating rotor comes into direct contact with the liner's rigid body. Unsteady vibrations are modeled using a specially developed software package for calculating the vibration of rotors that rub against the turbine (pump) stator elements. The stiffness of the bearing liner with the stator support structure is specified by a dependence in the force–deformation coordinate axes. In modeling the effect of damping forces, the time moment corresponding to the onset of asynchronous rolling-over with growing vibration amplitudes is used as the assessment criterion. With a longer period of time taken for the rolling-over to develop, it becomes possible to take the necessary measures in response to actuation of the equipment set safety system, which require certain time for implementing them. It is shown that the gyroscopic damping components facilitate the developing rolling of the rotor over the bearing. If measures taken to decrease these components in the damping devices and bearings are met with success, the onset of asynchronous rolling-over with the growing amplitudes occurs after a longer period of time.

*Keywords:* rotor, stator, bearings, seals, asynchronous rolling-over, roll-over exciting forces, gyroscopic forces, rotor section center motion trajectory, rotor precession angular speed

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Forced outages of turbines for carrying out repair works and even catastrophic destructions of installations for different purposes often resulted from rubbing and subsequent rolling of the rotor over the stator. Such an emergency mode development scenario is recognized by many researchers [1–4]. Examples of the consequences from development of asynchronous rolling of a rotor over the stator are given in [5, 6–8].

One important issue relating to the problem of reducing dangerous consequences from rolling of a rotor over the stator is the period of time (so-called Abtime) taken for the turbine set protection system to come into action and the rolling-over development time to the point at which the vibration amplitudes reach dangerous levels for the strength of elements (studs connecting the turbine upper and lower casings, bearings, and rotor). These turbine components take the main forces that arise when the rotor rolls over the stator. The equipment set protection system takes a certain period of time to come into action. The results from modeling the process through which a rotor rolls over the stator shows that the time taken to reach the mode characterized by high forces with which the rotor acts on the stator elements as it rolls over the stator is noticeably shorter than the Abtime [7]. The possibility of an almost instantaneously developing mode in which a rotor rolls over the stator was for the first time pointed out by E.L. Poznyak in [3], who also introduced the notions of synchronous and asynchronous rolling-over, which are used in this study. The authors of [1, 2] presented the results from numerical simulation and experimental investigations aimed at studying the development of hazardous vibration arising during asynchronous rolling of the rotor over the stator for the model rotor detuned from the resonance by 7–8%.

A search of methods for increasing the time taken for the rotor to reach a mode in which it permanently

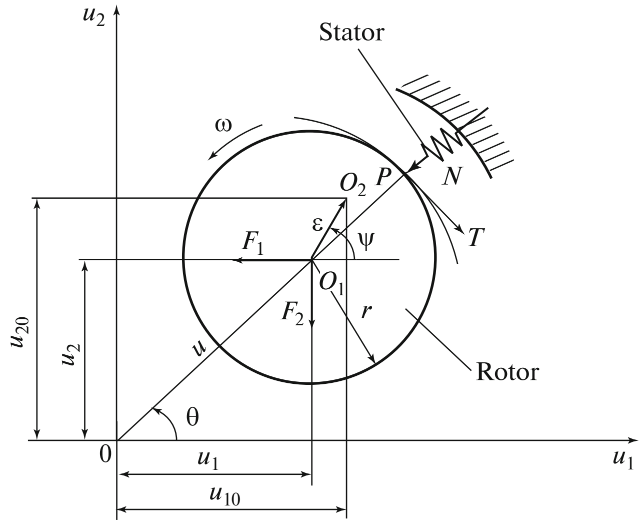


**Fig. 1.** Schematic presentation of the contact between the rotor and yielding support. $P$ is the point of contact between the rotor and support; $O$ is the bearing bore center; $O_1$ is the rotor section center; $O_2$ is the position of the rotor mass center; ω is the rotor angular rotation speed; $N$ is the support reaction force; $T$ is the sliding friction force; $R = M\omega^2\varepsilon$ is the force due to the rotor imbalance; ε is the abrupt loss of balancing; $M$ is the rotor mass; $r$ is the rotor radius; $u$ is the rotor section center displacement in the radial direction; ψ is the rotor turning angle at the moment in which the abrupt loss of balancing occurs; θ is the precession angle; $F_1$ and $F_2$ are the forces acting from the sides of rotor supports.

rolls over the stator after a few initial rubbings and, generally, methods for suppressing asynchronous rolling-over and forcing the asynchronous rolling-over to change to a less dangerous mode is considered to be of much importance, because the Abtime values differ very significantly for different protection systems.

Generally, damping forces play a positive role and facilitate suppression of a developing asynchronous rolling-over [5, 6]. However, as far as the damping matrix secondary coefficients are concerned, the forces determined by them (gyroscopic forces [9]) act in the rotor section center motion plane direction perpendicular to the speed components (for example, on the horizontal and vertical vibration directions). It should also be noted that the direction in which the damping force components act, as well as their values, change periodically depending on the rotor center position in its motion trajectory; i.e., these components act both tangentially with respect to the rotor motion trajectory (opposite to the speed direction) and in the direction perpendicular to the trajectory. The former ones retard the motion of the rotor and facilitate the development of its backward precession. Their action is similar to the action of friction forces arising when the rotor rubs against the stator. The effect of force components perpendicular to the rotor motion trajectory is considered in [7].

The damping forces appearing in the equations describing the vibration of a symmetrical rotor resting on two supports [6, 10] can be expressed by

$$-\bar{q} = B\dot{\bar{u}}, \tag{1}$$

where $B = \begin{bmatrix} b_{11} & b_{12} \\ b_{21} & b_{22} \end{bmatrix}$ is the matrix of dynamic damping coefficients (e.g., in the bearing lubrication layer of plain bearings); $\dot{\bar{u}} = \begin{bmatrix} \dot{u}_1 \\ \dot{u}_2 \end{bmatrix}$ is the rotor (rotor journal) speed vector), and 1 and 2 are the horizontal and vertical vibration directions.

It is of interest to separate the following forces from the damping components:

$$-q_{12} = b_{12}\dot{u}_2; \quad -q_{21} = b_{21}\dot{u}_1. \tag{2}$$

Forces (2) are called gyroscopic forces [9]. This name has been adopted in connection with deep elaboration of the theory of gyros, in which these forces were considered. The components $\begin{bmatrix} b_{11} & 0 \\ 0 & b_{22} \end{bmatrix}$ of the damping coefficient matrix are related to dissipative forces. Figure 1 represents the moment at which the rotor comes into contact with the bearing and shows the gyroscopic forces acting on it. The study objective is to determine the influence of damping force components (2) under the conditions of a developing rolling-over in which the rotor experiences the action of not only damping forces but also the forces of contact interaction between the rotor and bearing that excite the asynchronous rolling-over [5–7].

A model of the rotor–supports system is presented in [5, 10]. The criterion used to estimate the effect the damping force secondary components $b_{ij}$ ($i \neq j$) have on the development of asynchronous rolling of the rotor over the bearing is based on the time for which asynchronous rolling-over with the growing vibration amplitudes develops. Measures taken to increase the time for which asynchronous rolling-over develops (to slow it down) are a positive factor from the viewpoint of limiting the consequences resulting from rolling the rotor over the stator (which are sometimes catastrophic in nature) [5, 6, 10, 11] because it opens up the possibility for the equipment set safety system to perform operations on closing the stop valves and disconnecting the generator from the network under the conditions of a developing emergency situation. This also facilitates deceleration of the rotor as it comes into contact and its escape from the resonance zone.

To model the effect of damping force components on the development of rolling-over, a symmetrical rotor with water-lubricated support plain bearings was selected. The bearings' sliding pair is composed of refractory metals. The lubrication layer in such bearings is thinner than that in white metal-lined oil-lubricated bearings, in particular, due to the difference between the dynamic viscosity coefficients of oil and water (which differ from each by 30–40 times). In view of this circumstance, the risk of rubbing in water-lubricated bearings is considerably higher due to a smaller lubrication film thickness and due to the rotor coming in direct contact with the liner body when rubbing occurs.

The white metal layer used in oil-lubricated plain bearings creates insignificant resistance to rubbing within the limits of its thickness, and the hazard of rubbing increases only when the rotor comes into contact with the liner body. Therefore, the gap in bearings with a white metal lining is commensurable with the gap between the rotor and stator in the turbine flow path as far as the hazard of rolling-over to occur is concerned. A small gap between the bearing's rigid body and journal and the increased sliding friction coefficient as compared with that in bearings with a white metal lining are factors that facilitate the occurrence of rotor rolling over the stator. For example, for plain bearings with an oil lubrication system and a white metal lining, the sliding friction coefficient for rubbing $\chi = 0.06$–$0.07$, whereas the value of $\chi$ for water-lubricated bearings may be higher by a factor of two or more. According to the data of [12], under the conditions of liquid lubrication, $\chi = 0.001$–$0.005$; in case of small gaps and high-viscous lubrication, $\chi = 0.01$–$0.03$; under the conditions of boundary lubrication, $\chi = 0.1$–$0.2$. A water film decreases friction only slightly, and the sliding friction coefficient for rubbing depends completely on how accurately the bearing's friction pair surfaces have been processed. The condition of contact surface after a few rubbings can only change for the worse; i.e., the coefficient $\chi$ can increase taking into account the "rubbing history."

The evolvement of the rolling-over mode was modeled using the developed software package aimed at calculating the vibration of rotors involving their rubbing against the turbine (pump) stator elements. The algorithm for numerically simulating the development of the rotor's rolling over the bearing is given in [5, 6, 10].

The initial equations describing the rotor vibration taking into account gyroscopic forces are given by

$$\begin{aligned} m\ddot{u}_{10} + b_{11}\dot{u}_1 + b_{12}\dot{u}_2 + k_{11}u_1 + k_{12}u_2 &= 0; \\ m\ddot{u}_{20} + b_{21}\dot{u}_1 + b_{22}\dot{u}_2 + k_{21}u_1 + k_{22}u_2 &= 0; \end{aligned} \tag{3}$$

$$\begin{aligned} & m\ddot{u}_{10} + b_{11}\dot{u}_1 + b_{12}\dot{u}_2 + k_{11}u_1 + k_{12}u_2 \\ & \quad + N\cos\theta - T\sin\theta = 0; \\ & m\ddot{u}_{20} + b_{21}\dot{u}_1 + b_{22}\dot{u}_2 + k_{21}u_1 + k_{22}u_2 \\ & \quad + N\sin\theta + T\cos\theta = 0. \end{aligned} \tag{4}$$

Equations (3) describe the motion of a rotor in the gap (without coming into contact with the bearing), and equations (4) describe the same but with taking into account the rotor's coming in contact with the bearing. By alternately solving equations (3) and (4) with checking the gap in the bearing, it becomes possible to model the vibration of the rotor, including its rubbing against the bearing. Equations (3) and (4), and the quantities appearing in them are described in more detail in [5, 6, 10].

The detuning degree of the rotor–support system from the resonance is 5%; the rotor journal radius is 0.106 m; the sliding friction coefficient (with water used as lubricant and with the duly processed contacting surfaces) is equal to 0.1, and the relative gap in the bearing with the journal concentrically arranged in the bearing is equal to 0.0028. The nonlinear stiffness characteristic of the support together with the bearing liner is represented by the deformation versus force dependence taking into account the gap in the bearing. The stiffness of the support together with the liner (the damping in the support is not taken into account) is equal to $0.362 \times 10^8$ kN/m. An abrupt loss of rotor balancing (separation of a mass equal to 0.025% of the rotor mass at a radius of 1 m at the time moment $\tau = 0$) is considered as a factor exciting rotor vibration involving its rubbing. Transient vibration triggered by the abrupt loss of balancing starts from the equilibrium position of the rotor journal in the bearing bore. It is important to note that, as was shown in [5, 6, 11], the further development of the asynchronous rolling-over mode is governed by the dynamic properties of the rotor–supports system and is little dependent on the initial cause that led to the occurrence of vibration with rubbing. Of the totality of forces acting on the rotor, the model takes into account the forces due to imbalance, the forces caused by contact interaction when the rotor rubs against the stator, and damping force components (1).

Figure 2 shows the results obtained from numerically simulating the development of rolling-over involving the rubbing of the rotor journal against the bearing liner for $b_{12} = b_{21} = 0$. With $b_{12} = b_{21} \neq 0$, the general development pattern of the rolling-over mode changes only slightly. An unsteady pattern of rubbings is observed, which is due to some scatter of collision parameters from revolution to revolution. The rotor motion is accompanied by its collisions and rebounds with transition to asynchronous rolling of the rotor over the stator (see Fig. 2f). The dashed curve shows the circle representing the gap in the bearing within which the rotor journal moves without rubbing. An increase of the secondary components $b_{ij}$ $(i \neq j)$ entails reduction of the time at which the rolling-over mode with the growing vibration amplitudes starts to develop (Fig. 3).

The dependence shown in Fig. 3 is nonlinear in nature, with the general trend toward decreasing the selected parameter (the time to the moment at which the rolling-over starts to develop) with increasing the coefficients $b_{ij}$ $(i \neq j)$. The time taken for the rolling-over to develop is by 12–15% longer if there are no forces determined by the secondary coefficients in the

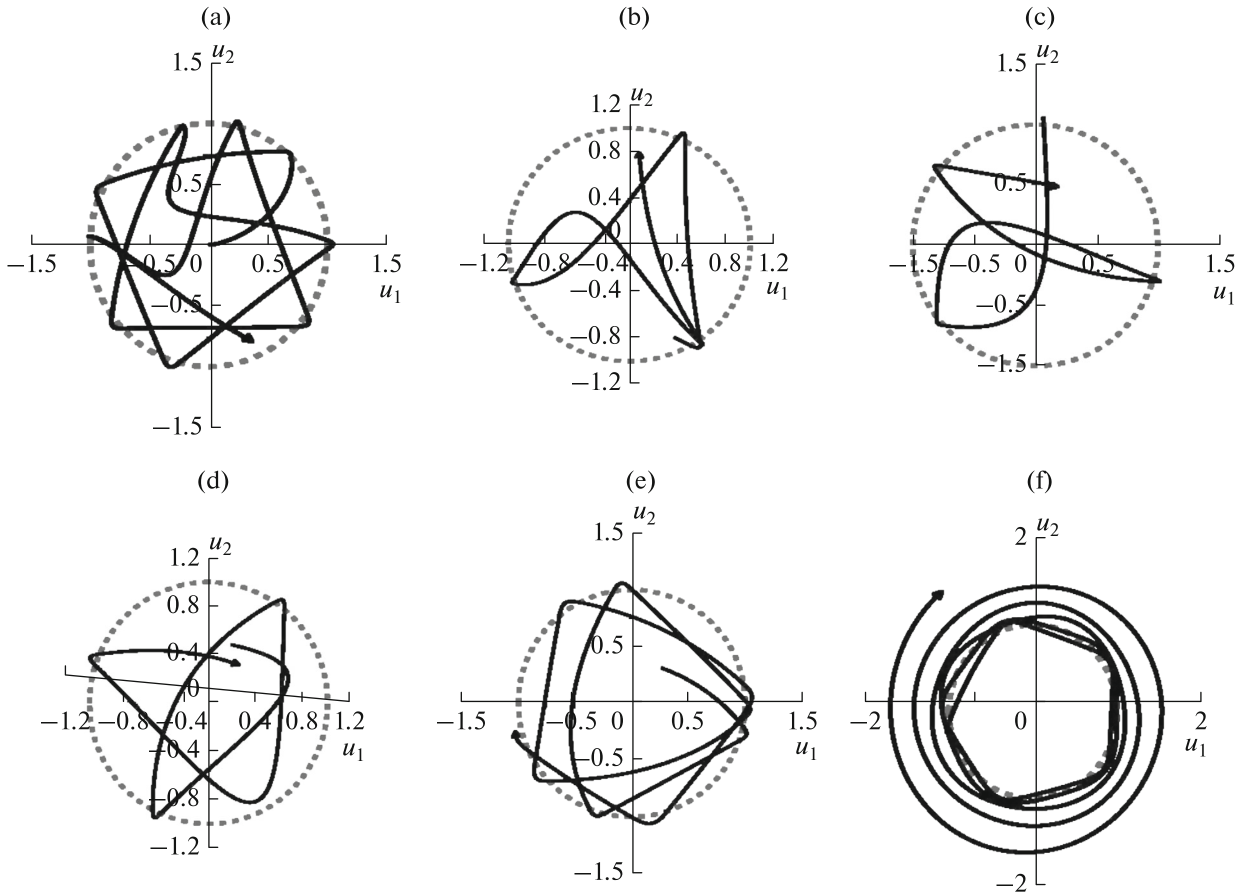


**Fig. 2.** Rotor journal center motion trajectory when rubbing occurs after the abrupt loss of balancing at $b_{12} = b_{21} = 0$. $\tau$, s: (a) 0–0.07, (b) 0.07–0.1, (c) 0.1–0.13, (d) 0.13–0.16, (e) 0.16–0.19, and (f) 0.19–0.21.

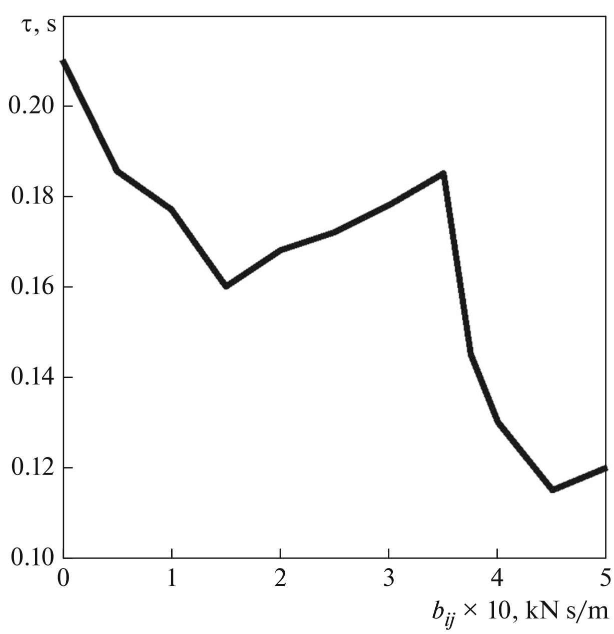


**Fig. 3.** Time to the moment at which asynchronous rolling of the rotor over the bearing starts as a function of the gyroscopic forces governed by the damping matrix secondary coefficients $b_{ij}$ ($i \neq j$).

damping matrix. It should be noted that the values of damping matrix secondary coefficients $b_{ij}$ ($i \neq j$) are of the same order as those of the main coefficients $b_{ii}$ ($i = 1, 2$). Such ratio between the main and secondary damping coefficients corresponds, e.g., to the damping values in the lubrication film of plain cylindrical bearings at the working rotor rotation frequency. In segmental bearings, their design features are such that the values of both secondary damping and secondary stiffness coefficients are brought to almost zero.

## CONCLUSIONS

(1) The damping forces governed by the gyroscopic damping components facilitate the developing rolling of the rotor over the bearing. Measures taken to decrease their values in damping devices and bearings result in a longer period of time to the moment at which asynchronous rolling-over with growing amplitudes starts to develop.

(2) Those who design bearings and damping devices should try to decrease the values of damping matrix secondary coefficients $b_{ij}$ ($i \neq j$). Segmental plain bearings with close-to-zero values of secondary

damping and stiffness coefficients can serve as examples of such design solutions.

(3) When the secondary damping coefficients in the considered rotor–bearings system are excluded, the period of time to the moment at which asynchronous rolling-over starts to develop increases by 12–15%. The longer the period of time to the moment at which the rolling-over starts to develop, the longer the time margin available for the equipment set safety system components to come in action.

## REFERENCES


1. R. Gasch, "Vibration of large turbo-rotors in fluid-film bearings on an elastic foundation," J. Sound Vib. **47**, 53–73 (1976).
2. R. J. Williams, "Parametric study of reverse whirl instability using an analytikal equilibrium model," BRITE_EURAM Project ROSTADYN Technical Report No. 4.17 (1996).
3. E. L. Poznyak, "Torsional impact in the shaft line at sudden and strong imbalance," Mashinovedenie, No. 5, 66–74 (1987).
4. L. Ya. Banakh, "Contact problems in rotor systems," in *Proc. 22nd Int. Conf. on Vibroengineering, Moscow, Oct. 4–7, 2016*; Vibroeng. Proc. **8**, 90–96 (2016).
5. V. F. Shatokhin, Doctoral Dissertation in Engineering (All-Russia Thermal Engineering Inst., Moscow, 2014).
6. V. F. Shatokhin, *Oscillations of Turbounit Rotors with Generation Roll of the Rotor over the Stator (Methods of Modeling and Software Tools)* (Lambert Acad., Dusseldorf, 2016).
7. V. F. Shatokhin, "Influence of the damping components in supports on the shape of the manifestation of the generation roll of the rotor over the stator," in *Proc. 9th Int. Sci.-Tech. Conf. on Problems of Vibrations, Vibration Adjustment, Vibration Monitoring and Diagnostics of the Equipment of Power Stations, Moscow, Nov. 21–24, 2017* (Vseross. Teplotekh. Inst., Moscow, 2017).
8. V. F. Shatokhin, "Forces exciting generation roll at rotor vibrations when rotor-to-stator rubbing," Therm. Eng. **64**, 480–489 (2017). doi 10.1134/S0040601517070072
9. O. G. Zav'yalov, Doctoral Dissertation in Engineering (St. Petersburg State Univ., St. Petersburg, 2009).
10. A. G. Kostyuk, V. F. Shatokhin, and O. A. Volokhovskaya, "Motion of an imbalanced rotor when it rubs against the stator," Therm. Eng. **59**, 87–95 (2012).
11. V. F. Shatokhin, "Development of the generation roll of the imbalanced rotor over the stator," Elektr. Stn., No. 11, 17–28 (2013).
12. P. I. Orlov, *Fundamentals of Machine Design: Handbook*, 3rd ed. (Mashinostroenie, Moscow, 1988; Mir, Moscow, 1976).


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