Evaluation of effect of oil film of rotor bearing

L B Alekseeva, V V Maksarov
Saint-Petersburg Mining University, 2, 21st Line V.O., 199106, St. Petersburg, Russia
E-mail: Lbalek@rambler.ru

Abstract. The high-rpm rotors were subjected to the dynamic analysis. Oscillations of a rotor spinning in gapped bearings were considered. It was stated that the rotor necks motion pattern depends on a lot of factors: a ratio of static and dynamic loads on the bearing, radial clearance size, presence of oil film between a neck and a bearing, elastic and inertial properties of a mounting group. The most unfavourable mode where static and dynamic loads are equal was detected without taking into account the oil film impact. The impact of oil film on the bearing assembly dynamics is significant in high-rpm rotors. The presence of oil film can possibly cause rotor buckling failure and self-starting. Rotor motion stability in small was studied. Herewith, various schemes were considered. Expressions, determining the stability zones of a rigid rotor on the fixed support and the supports with elastic and inertial elements, were given.

1. Introduction
Rotors apply various elastic elements built in between a rotor and a machine casing. They allow separating a rotor-casing system from resonant modes, thus, reducing the exciting force imposed on the casing and, eventually, on the base. However, it is not always possible to utilize elastic elements with the required compliance due to the design constraints applied to the peak rotor displacement and restrictions of large static stresses inside them [1, 2].

Vibrations in low-frequency range are normally characterized by harmonicity while the major disturbance source is rotating elements imbalance. Other vibratory sources such as the misalignment of separate mechanisms connected by joints, violation of the geometric pattern of flexible joints (e.g. rotor neck ovality), and elements accelerated motion can be distinguished within this range.

2. Reasons for rotors vibration emerging and methods to reduce them. The sources of vibration adherent to rotors can be divided into three types: mechanical sources; sources emerging due to machine operation; aerodynamic ones. Mechanical sources of vibration are conditioned by constructive and technological peculiarities of manufacturing different types of rotors. The most wide-spread sources of vibration in all types of rotors are rotating elements imbalance; accessories drives misalignment; violation of the geometric pattern of flexible joints.

Rotor imbalance occurs due to the misalignment of its principal central axis of inertia with the axis of rotation. While spinning, an unbalanced rotor is subjected to dynamic forces, whose components being held in some plane (which intersects the axis of rotation) vary depending on rotor spin rate $\omega$. The degree of imbalance is counted as value $mr$, where $m$ is some imbalance mass set at distance $r$ from the axis of rotation. Rotor imbalance can be caused by various factors [3, 4].
There can be the following constructive reasons: violation of preloads along the counterparts mounting surface; violation of previously gained balance due to frequent rotor reassembling; wrong choice of working tolerance causing radial motion variation.

The possible technological reasons are as follows: defects of mechanical operation (violation of rotor circular symmetry, bending deflections caused by permanent deformation); assembly defects (elements bends caused by uneven heating and cooling under the conditions of shrink fit, misalignment, etc.); different conditions of running on testing grounds and with full-scale testing.

Misalignment can occur in air-cooled and water-cooled turbine generators and large direct-current machines due to rotors thermal unbalance. Uneven heating or cooling of a rotor generates asymmetric flows and even rotor bending. The strength generated by heat imbalance can reach the value from 0.4 to 0.6 of the rotary mass in the turbine-driven generator.

A gap in the conjugation pivot-bearing is the element influencing the dynamics of the whole system. When determining the degree of imbalance and the gap size, it is possible to apply the kind of working mode involving the neck impacting the bearing. The evoked exciting forces have a wide frequency spectrum. The critical speeds spectrum of the rotor having non-linear boundary data conditioned by the gaps in bearings is constant. Rotor natural oscillation frequencies are the functions of the oscillatory amplitudes of rotor ends in bearings gaps. Only a gap with no regard to conjugation pivot-bearing geometry deviation leads to the situation when an unbalanced force gives a casing not a purely harmonic disturbance but a polyharmonic force with the frequencies multiple of rotor spin rate even under the conditions of the neck small-amplitude oscillation. Moreover, the oil layer generated in bearings when spinning the rotor can bring self-oscillation [5].

Therefore, special devices such as vibration isolators, which dilute the mechanical influence on the object of protection against vibration, are adopted. One of the methods to suppress oscillation is to apply a dynamic absorber, which is adjusted to an object and brings additional dynamic force diluting the influence on the object of protection against vibration from the source end.

3. **Model of rotor oscillation in bearings with gap.** A rotor neck motion pattern depends on the correlation of dynamic load posed by an unbalanced centrifugal force of rotor inertia and static load of the bearing. If dynamic load is smaller than the static one, the center of the rotor neck will execute oscillatory motion through a circumference arc. The most unfavourable mode is when static and dynamic loads are equal. The neck pulls away from the bearing. The oscillatory amplitude can reach the point of 90°. This phase triggering starts depending on not only the values of the mentioned forces but also the size of the gap. Eventually, it leads to the bearing increased vibration and its early mechanical failure.

The impact of oil film on the bearing assembly dynamics is rather sufficient in high-rpm rotors. In this case there is a possibility of buckling failure and self-starting. Due to the impact of oil film, the actual critical speed can be 30-50% lower than the design speed. This possibility is stipulated by the fact that the oscillation strength can be high. It can be conditioned by the situation when the oscillatory amplitudes, caused by oil film, frequently exceed the amplitudes of resonance oscillations of the poorly balanced rotors and there is cyclic stress in the axis. In addition, there can be troubles related to the possibility of loosening the forced fit of the mating parts, bushes impairment and jamming of a pivot in a bearing.

The fundamental obstacles occurring in the process of computation of the stability of the rotor motion along the oil film are linked with determining the bonds rigidity and damping [6].

When assessing the stability of neck motion along the oil film, the following aspects will be taken into account: firstly, both disturbed and undisturbed motions occur under the conditions of the same forces (sources of energy); secondly, stability will be considered at the infinitely great timespan.

In general, hydrodynamic forces represent non-linear functions of the corresponding coordinates, speeds, determining the position of a neck in a bearing. It is possible to linearize the expression for these forces in case of slight displacements. The presence of such forces predetermines the possibility of buckling failure of the neck on oil film.
If the speed and load are constant, the neck will occupy a definite position on a so-called curve of mobile equilibrium, characterized by relevant eccentricity $\chi = \varepsilon / \Delta$, where $\varepsilon$ is eccentricity; $\Delta$ is a radial clearance.

Herewith, external forces and constraining force of the oil layer are balanced. If a neck is influenced by small disturbances, the neck center will deflect from the curve of mobile equilibrium and will execute vibrations with the growing amplitude. Let us denote such start position as an unstable one. It can occur in a situation when $\chi < 0.7$. The impact of loads and compliances of support can significantly influence stability zones altering.

4. **Stability of rigid rotor on rigid supports.** Let us introduce the fixed system of reference $xyz$. It ought to be noted that the system with an oil film is not conservative. The oscillations in the $x$- and $y$-directions in the situation under consideration are connected and the differential equations of the rotor progressive motion will be as follows [7]:

$$\begin{cases}
m_1 \ddot{x} + 2 F_x = 0; \\
m_1 \ddot{y} + 2 F_y = 0,
\end{cases}$$

(1)

where $m_1$ is a rotor mass; $F_x$, $F_y$ are linearized values of hydrodynamic forces.

Let us define the system solution (1) in

$$x = \bar{x} e^{\omega t}; \quad y = \bar{y} e^{\omega t}.$$ 

(2)

To create the stability zones, the characteristic equation whose degree is equal to 4 was derived. It is rather favorable to apply Hurwitz criteria. Figure 1 demonstrates the boundaries of rotor stability in the plane of parameters $\varphi$, $\beta$

$$\varphi = \frac{2 \mu l}{m_1 \psi \omega_p}; \quad \beta = \omega / \omega_p,$$

where $\mu$ is oil dynamic viscosity; $l$ is the bearing length; $\psi = \Delta / r$; $r$ is neck radius; $\omega$ is angular velocity; $\omega_p$ is rotor speed of operation.

![Figure 1. Rotor stability zones on rigid supports](image)

5. **One-dimensional model of rotor oscillation on rigid supports.** Let us consider a one-dimensional model of rotor oscillation on antivibration supports containing the first absorption stage with the rigidity $C_1$, the second absorption stage with the rigidity $C_2$, the linear damper (antivibrator) with the rigidity $C_3$ [7, 8].
Figure 2. One-dimensional model of rotor oscillation on rigid supports

Figure 2 shows: $m_1, m_2, m_3$, which are the values of mass of a corresponding rotor, spacer, damper; $C_1, C_2, C_3$ are the values of rigidity of the corresponding elastic linkages.

An oil film reveals its elastic properties in high-rpm rotors and these properties influence the machine dynamics. An oil film is suggested to be applied as the first absorption stage. It enables to simplify the construction of vibroisolating assembly.

The impact on antivibration supports, which contain masses connected both in series and in parallel and forming two-stage shock absorption with a dynamic absorber, was studied.

The characteristic motion equation of 12-power was obtained on the basis of the motion equations of the system under research. D-decomposition method is applied to create stability zones. The required transformations resulted in the derivation of the system of equations defining the rotor stability boundary.

\[
\begin{aligned}
\lambda^* = \lambda^* (\theta f A - q f B) + \varphi f B = 0; \\
\lambda (\lambda^* - \psi s A + \varphi f B b B) = 0.
\end{aligned}
\]

where $\lambda$ is the value of the imaginary components of a characteristic equation; $b, t, s, q, f$ are certain parameters which depend on the position of neck in a bearing; $A,B,C$ are the polynomials of 8-power $\lambda$, whose coefficients depend on the characteristics of rotor and antivibration supports.

Let us consider a more complicated scheme including a two-stage vibration isolation of rotors with a dynamic damper (antivibrator) (Figure 3). The motion equations of such system are as follows:

\[
\begin{aligned}
m_1 \ddot{x}_1 + 2F_1 = 0; \\
m_1 \ddot{y}_1 + 2F_2 = 0; \\
m_2 \ddot{x}_2 + K_s (x_2 - x_1) + K_s x_2 - 2F_3 = 0; \\
m_2 \ddot{y}_2 + K_s (y_2 - y_1) + K_s y_2 - 2F_4 = 0; \\
m_3 \ddot{x}_3 + K_s (x_3 - x_1) = 0; \\
m_3 \ddot{y}_3 + K_s (y_3 - y_1) = 0.
\end{aligned}
\]
where \( m_2, m_3 \) are the masses of spacers, dampers; \( K_1, K_2 \) are the factor of rigidity of the second absorption stage and elastic linkages “antivibrator - spacer”; \( x, x_1, x_2, y, y_1 \) are the rotor and spacers masses central displacement.

![Computational scheme](image)

**Figure 3.** Computational scheme

An oil wedge occurring as a result of neck movement inside a bearing can be used as the first absorption stage. An oil wedge factor of rigidity is \( K_1 \).

Let us obtain a characteristic equation with power 12 as a result of transformations. Under such circumstances it is rather unfavourable to apply Hurwitz criterion to handle the tasks in general terms. The present study uses D-decompositions method, which consists in depicting the imaginary axis of roots on the plane of the test parameters influencing the system stability.

The analysis of the impact of antivibration supports characteristics on rotor stability zones altering.

It was shown that the zone of rotor unstable motion expands as the rigidity of the second absorption stage reduces. This widening is especially significant for the values:

\[
\gamma^2 = \frac{C_2}{m \omega_s} < 1 ,
\]

where \( \omega_s \) is antivibrator adjustment frequency.

For values \( \gamma^2 > 1 \), rotor stability boundaries on the supports with two-stage absorption virtually coincide with the rotor stability boundaries on the elastic supports with the values of rigidity of the second absorption stages.

For \( \gamma^2 < 1 \) the increase of the damper mass in comparison to the spacer mass decreases the rotor marginal stability. However, this decrease can be neglected with regards to the actual values of antivibration support masses (10-15% of the rotor mass) and to scope \( \beta < 1.1 \) of the rotor speed.

Under the conditions of equal free-running frequency and damper adjustment frequency, a rigid rotor can lose stability only if the rotating velocity is twice larger than the damper adjustment frequency.

### 6. Conclusion

1. The studies presented in the paper are a part of a general algorithm of the gear under design. They will enable one to obtain source data for providing approximate calculation of the gear components stress and stiffness.
2. The disturbance sources in low-frequency, mid-frequency and high-frequency ranges have been defined. The major disturbance sources in each range have been singled out. General disturbance ranges and general methods of diminishing disturbance in the source have been presented.

3. The scheme of inner elastic-inertial protection against vibration, consisting of a rotor, casing, spacers and antivibrators has been introduced.

4. Vibroisolation stability boundaries and the role of oil film in the system rotor-casing have been specified.

References

[1] Grigoriev N V 1974 Vibration energy of machinery. (Mechanical engineering)
[2] Maksarov V V, Olt J, Krasnyy V A 2016 Improving fretting resistance of heavily loaded friction machine parts using a modified polymer composition Agronomy Research 14 (1) 1023 - 1033
[3] Tondl A 1971 Dynamics of rotors of turbogenerators (Leningrad Energy)
[4] Merkin D R 1971 Introduction to the theory of stability of motion (Moscow Science)
[5] Maksarov V, Madissoo M, Olt J 2013 Increasing the Effectiveness of the Cutting Process in the Course of Milling Journal of Mechanics & Industry Research 1 (4) 75 - 81
[6] Pozniak E L 1980 Vibrations of rotors. Vibration engineering. Oscillations of machines, constructions and their elements 3 130 – 189
[7] Olt J, Tärgla T, Liyvapuu A 2016 Mathematical Modelling of Cutting Process System Engineering Mathematics I. Proceedings in Mathematics & Statistics, 178 173-186
[8] Gupta A and, Sharma A 2015 Piezoelectric Energy Harvesting via Shoe Sole International Journal of New Technology and Research 1(6) 10-13