Research of the acceleration dynamics of the horizontal separation centrifuge for drilling mud cleaning

I A Ryndin¹, V N Brendakov²,³ and S N Kladiev¹
¹ National Research Tomsk Polytechnic University, Tomsk, Russia
² National Research Tomsk State University, Tomsk, Russia
³ STI National Research Nuclear University MEPhI, Russia
E-mail: ilyaryndin93@gmail.com, VNBrendakov@mephi.ru, Kladiev@tpu.ru

Abstract. This article presents a method for assessing the technical condition of a horizontal centrifugal centrifuge using vibration analysis methods. The study has been conducted on a centrifuge’s single-mass rotor model with a rigid and flexible shaft and flexible supports. Based on the results of the research recommendations for assessing the compliance of the bearings and deflection of the rotor shaft, the location of the natural frequency in the frequency spectrum of oscillations have been identified. An approach to damping vibrational oscillations of a rotor system using controlled acceleration of the rotor by means of a frequency-controlled electric drive is proposed.

1. Introduction

The drilling mud cleaning system based on a centrifuge is designed for deep cleaning of weighted and non-weighted solutions from drill cuttings when drilling oil, gas and other wells. It is used as part of the circulation systems of drilling rigs.

Effective drilling fluid cleaning systems, including modern vibrating strainers and centrifuges, by changing the operating modes, allow you to maintain a certain depth of mud cleaning and the total and active solids content in it, thereby regulating the technological properties of the drilling fluid and have a primary effect on speed drilling and economic well drilling performance [1].

When entering the drilling fluid in a centrifuge in the presence of centrifugal forces, it is separated into a condensed solid phase (sludge) and purified solution. The separation of the drilling fluid in the centrifuge occurs continuously, while the purified solution is returned to the circulating system.

An important task when working with a centrifuge is its racing. It should occur smoothly, without vibrating vibrations of the rotor system, which can lead to an imbalance of the system and the destruction of the bearing supports, and then the centrifuge itself.

2. Formulation of the problem

Oscillatory processes that occur during the operation of rotary machines, are created by the machine itself and are the inevitable factor affecting the operation of the system. As a rule, the presence of permissible vibration is laid at the design stage of the machine and does not pose a significant problem if the vibration is kept within acceptable limits. However, if the intensity of vibrations reaches a significant value, then there are threats of disruption to the normal operation of the machine. Harmful vibration contributes to premature wear of machine parts, because of vibration, fatigue failures of the supporting elements and bearing elements of the structure occur, the flow of technological processes and parameters of technological modes change, disturbances are introduced into the automatic control systems. To prevent the negative effects of vibration, it is necessary to diagnose the object vibration condition in a timely manner and, based on research, take measures to repair and vibro-protect the rotor machine [2].

3. Theory

During the industrial operation of the machine, the evaluation of their vibration characteristics is carried out in accordance with State Standard 10816 "Monitoring the condition of machines based on the results of vibration measurements on non-rotating parts." Measuring the characteristics of the
stator elements for most machines is quite sufficient for an adequate assessment of the state of the units. The control is carried out in the operating frequency range of the rotary machine. When conducting control by vibration equipment, the following parameters are measured: vibration displacement (mm); vibration velocity (mm/s) – derivative of vibration displacement over time; vibration acceleration (m/s²) – derivative of vibration velocity over time [3].

The control points are chosen based on the design features of the machine, as a rule, they are support bearing housings or other elements that most characterize the overall vibration state of the system. For the sake of completeness, measurements are carried out in three mutually perpendicular directions. The control method is quite simple: after measuring the vibration characteristics, their values are compared with the recommended levels of the four vibration state zones, which characterize the quality state of the machine and the possibility of its further operation. According to the aforementioned standard, the following zones are established: A – the vibrations are minor, characteristic of new machines or for those just commissioned; B - the machine is suitable for further operation; C – the machine is unsuitable for long-term operation, with a suitable opportunity for the machine to carry out repairs; D – vibration levels can cause machine breakdown, operation is prohibited [4], [5].

Consider a mathematical model of the rotor, the scheme of which is shown in Figure 1.

Figure 1. Scheme of single-mass rotor

The mass of the shaft, as compared with the mass of the rotor, will be neglected. We represent the rotor in the form of a circular disk whose center coincides with the point of attachment on the shaft and is denoted by \( O_2 \). The center of mass of the disk is denoted as \( O_1 \). In general, it may not coincide with the geometric center of the \( O_2 \) disk. The magnitude of the displacement of the center of mass of the rotor relative to the shaft axis is called eccentricity and is denoted by \( e \). We associate the beginning of the Cartesian coordinate system with the point of the disk center in the case of an undeformed shaft. The coordinate axes \( X \) – horizontal and \( Y \) – vertical lie in the central plane of the rotor disk. The \( Z \) axis coincides with the axis of the undeformed shaft and the axes of the two pivot bearings on which this shaft is attached.

First, consider the option when the supports are absolutely rigid. In this case, the engine is located in the middle of the shaft, i.e. \( l_1 = l_2 \). When the rotor rotates at an angular velocity, centrifugal force arises. In this case, the \( O_2 \) point will shift in the \( X – Y \) plane only as a result of the shaft deflection. This offset will be denoted as \( \xi \), taking into account the projections on the coordinate axes, we have \( \xi^2 = x_e^2 + y_e^2 \). The center of mass of \( O_1 \) will also have an offset for which, taking into account the projections on the axes, can be written \( R^2 = x_m^2 + y_m^2 \). From geometric considerations, the following dependencies can be obtained: \( x_m = x_e + e \cdot \cos(\omega t) \), \( y_m = y_e + e \cdot \sin(\omega t) \). When the shaft rotates, two forces act on it, the force of elasticity associated with the deformation of the shaft and the force of inertia associated with the movement of the center of mass of the system. The equilibrium condition of the rotor is defined as the equality to zero of the resultant of all forces. Thus, it is possible to obtain an equation describing the movement of the rotor.
\[ F_c = -c \dddot{x}; \]
\[ F_m = -ma = -mR''; \]
\[ F_d + F_m = 0 \]

where \( c \) is the rigidity coefficient of the shaft, \( m \) is the mass of the rotor, \( R'' \) is the second time derivative of the displacement of the center of mass.

We introduce the concept of the natural frequency of the shaft oscillation \( \Omega = (c/m)^{1/2} \), and now in the projections on the coordinate axes we can write:

\[
\frac{d^2 x_b}{dt^2} + \Omega^2 \cdot x_b = e \cdot \omega^2; \\
\frac{d^2 y_b}{dt^2} + \Omega^2 \cdot y_b = e \cdot \omega^2 \cdot \sin(\omega t).
\]

The solution of such an inhomogeneous system of differential equations will be sought as the sum of the general solutions of the homogeneous system and the particular solution of the inhomogeneous system. In this case, you can write:

\[
\begin{align*}
  x_b &= C_1 \cdot \cos(\Omega t) + C_3 \cdot \sin(\Omega t) + C_5 \cdot \cos(\omega t) \\
  y_b &= C_4 \cdot \cos(\Omega t) + C_6 \cdot \sin(\Omega t) + C_8 \cdot \sin(\omega t).
\end{align*}
\]

The first two terms in each equation describe shaft oscillations with frequency \( \Omega \), i.e. free vibrations with the natural frequency of the shaft. The third term in each equation describes the forced oscillations of the shaft with a given frequency \( \omega \). The presence of friction forces leads to the rapid damping of free vibrations, therefore, taking into account the duration of the separation process, we will consider only forced vibrations.

In this case, the substitution of the solution \( x_b \) and \( y_b \) in the original system of differential equations allows to obtain the value of the constants:

\[
C_8 = C_6 = \frac{e \cdot \omega^2}{\Omega^2 - \omega^2}.
\]

4. Experimental results

For a flexible shaft with an eccentricity \( e \), in which absolutely rigid supports are used, there are natural frequency \( \Omega \) and rotate with angular velocity \( \omega \), it is possible to estimate the shaft deflection by the form:

\[
\begin{align*}
  x_b &= \frac{e \cdot \omega^2}{\Omega^2 - \omega^2} \cdot \cos(\omega \cdot t) \\
  y_b &= \frac{e \cdot \omega^2}{\Omega^2 - \omega^2} \cdot \sin(\omega \cdot t) \\
  \xi &= \left| \frac{e \cdot \omega^2}{\Omega^2 - \omega^2} \right|
\end{align*}
\]

Figure 2 shows an example of the calculation of the shaft deflection on rigid supports. The calculation is made for the relative values of deflection and rotational speed. The resonant increase in the amplitude of oscillations of the shaft occurs when \( \omega \to \Omega \), i.e. we have \( \xi \to \infty \), which can cause the destruction of the shaft.
If the supports have a finite rigidity value, this has a significant effect on the magnitude of the natural frequencies of the rotor oscillations. In this case, the total displacement of a point associated with the center of the rotor will have a value \( \xi = \xi + \xi_0 \), where \( \xi_0 \) is the displacement caused by the deformation of the supports.

The equilibrium condition of the rotor remains in force, i.e. \( F_{el} + F_{in} = 0 \). However, it should be borne in mind that the force of elasticity in the case of compliant supports has the form \( F_{el} = -c_r \cdot \xi_r \), where \( c_r \) is the rigidity of the rotor taking into account the flexibility of the supports. Since there are two supports, we can write \( F_{el} = -2c_0\xi_0 \), where \( c_0 \) is the rigidity of each support. For a complete displacement of the point of the geometric center of the disk, you can write:

\[
\xi_r = \xi + \xi_0 = -\frac{F_{el}}{c} - \frac{F_{el}}{2 \cdot c_0} = -\left( 1 + \frac{1}{2 \cdot c_0} \right) \cdot F_{el}.
\]

\[
c_r = \frac{c}{1 + \frac{c}{2 \cdot c_0}}.
\]

Solving the system of differential equations of the second order, describing the equilibrium condition of the rotor, for the natural frequency of oscillation of the rotor, taking into account the compliance of its supports \( (\Omega_r) \), we can get:

\[
\Omega_r = \Omega \cdot \sqrt{\frac{1}{1 + \frac{c}{2 \cdot c_0}}}
\]

From this expression, it is clear that the natural frequency of oscillation of the rotor significantly depends on the rigidity of the supports, and varies in the range from \( \Omega_r = 0 \), with \( c_0 = 0 \), to \( \Omega_r = \Omega \), with \( c_0 \to \infty \).

Figure 3 shows how the frequency response of a flexible single-mass rotor on elastic supports varies depending on the rigidity of elastic supports.
Figure 3. Frequency response of a flexible single-mass rotor on elastic supports

\[1 - c/c_0 = 0.1; \ 2 - c/c_0 = 0.5; \ 3 - c/c_0 = 1; \ 4 - c/c_0 = 5; \ 5 - c/c_0 = 10\]

If, at the transition through resonance, we have a system acceleration with an angular acceleration of \(2\lambda\), then we write for the rotation frequency \(\omega = \omega_0 + 2\lambda t\).

We present our own system frequency as

\[\Omega = \sqrt{\frac{c}{\mu}}; \ \mu = \frac{m}{1 + \frac{c}{2\cdot c_0}}\]

the solution can be obtained in the form:

\[x_b = e \cdot \Omega \cdot \int_0^t \exp \left[ -\frac{\mu}{m} \cdot (t - \tau) \right] \cdot \sin \left[ \Omega \cdot (t - \tau) \right] \cdot \cos(\omega_0 \tau + \lambda \tau^2) d\tau + \frac{e}{1 - \left( \frac{\omega_0}{\Omega} \right)^2} \cdot \cos(\Omega t);\]

\[y_b = e \cdot \Omega \cdot \int_0^t \exp \left[ -\frac{\mu}{m} \cdot (t - \tau) \right] \cdot \sin \left[ \Omega (t - \tau) \right] \cdot \sin(\omega_0 \tau + \lambda \tau^2) d\tau + \frac{e}{1 - \left( \frac{\omega_0}{\Omega} \right)^2} \cdot \sin(\Omega t).\]

Figure 4 shows the result of the calculation of the frequency response of a flexible single-mass rotor mounted on absolutely rigid supports for various values of the rotational acceleration of the system.
Figure 4. The frequency response of a flexible single-mass rotor when moving through critical speed with different rotational accelerations

5. Discussion of the results

As can be seen from this figure, the dynamic transition through the critical frequency allows not only to reduce the maximum amplitude value of the vibratory displacement, but also to shift this maximum to the region of large frequency values. Thus, choosing the supports rigidity and acceleration correctly, you can optimize the operation of a single-mass rotor on elastic supports, minimizing the destructive vibrational forces at the resonant frequency.

As an example of a single mass rotor in the first approximation, we can consider a precipitation centrifuge of the brand OGSh-353K-09 with auger discharge of sediment. Specifications are shown in Table 1.

| Table 1. Technical parameters of OGSh-353K-09 |
|-----------------------------------------------|
| Internal rotor diameter, (mm)                  | 350          |
| Maximum rotor speed; (rev/min)                | 3000         |
| Relative auger rotation speed; (rev/min)      | 30           |
| Weight of centrifuges without drive, vibration isolation, cooling systems for seals, turbo drives; (kg) | 1520         |
| Mass of centrifuge complete with drive, anti-vibration device, automation kit; (kg) | 2870         |
| Rotor length; (m)                             | 2,8          |

For this centrifuge is used electric drive for rotor rotation. Type AIMM180M2 with the parameters specified in table 2.

| Table 2. Parameters AIMM180M2 |
|------------------------------|
| Power; (kW)                  | 30             |
| Rotation frequency; (rev/min)| 3000           |

Auxiliary drive motor centrifuges type AIMM132MU2,5 with the parameters listed in table 3.

| Table 3. Parameters AIMM132MU2,5 |
An external view of the centrifuge OGSh 353K-09 is presented in Figure 5.

| Power; (kW) | 11 |
|-------------|----|
| Rotation frequency; (rev/min) | 150 |

**Figure 5. Centrifuge OGSh 353K-09**

For most technological processes, the rotor of the centrifugal centrifuge will smoothly accelerate from 30 to 300 using a variable frequency induction motor. This is due to the fact that there is a limitation on the magnitude of the angular acceleration when transmitting torque from the induction motor to the centrifuge’s rotor through the V-belt transmission, therefore it is necessary to take into account the effect of imbalance on the vibrating state of the rotor, the location of the natural frequency in the subcritical frequency spectrum of mechanical vibrations. As a consequence, approaches to damping vibration vibrations of the rotor system are defined [6], [7], [8].

**6. Conclusion**

As a result of the study of the mechanical system of a single-mass rotor, it was established that:

1. The model of a single-mass rotor with intermediate supports in the first approximation corresponds to the rotor of the OGSh-353K-09 centrifuge in terms of geometrical dimensions and drive power.

2. The design parameters of the centrifuge rotor affect the number of natural frequencies of the mechanical system (in particular, for single-mass systems with isotropic supports there is one resonance frequency per \( \Omega = 75 \text{ s}^{-1} \))

3. The value of the rigidity of the shaft and the supports affect the location of the resonance frequency on the frequency spectrum (with an increase in the total rigidity, the resonance frequency shifts to a zone of higher frequencies).

4. The limiting value of the rotational acceleration with a uniform acceleration of a centrifuge with a frequency-controlled induction motor should not exceed 15 s\(^2\), which corresponds to the time of the centrifuge start during 6.47 seconds. With an acceleration of 5 s\(^2\), the minimum start time is 11.2 s.

5. With a smoother acceleration of the unbalanced rotor with the achievement of the working rotation frequency which is outside the resonance zone that takes place in practice, it is necessary to apply special measures to overcome the natural frequencies of the system using special control laws. In this case, measures should be taken for faster passage of the resonant
frequencies during acceleration and excluding the accidental stopping of the acceleration process to any of the natural frequencies of the system.

6. The electric drive of the screw accelerates synchronously with the electric drive of the centrifuge rotor and after the end of the acceleration, the screw rotation frequency is set with a lag of 0.3...1%. The value of the lag of the screw speed behind the rotor speed depends on the characteristics of the divided suspension and the degree of draining of the sediment.

7. The obtained data allow us to develop simulation models of a mechanical system with an unbalanced rotor under Simulink MatLab to study the torque control system, when an unbalanced rotor passes through resonance.

7. References
[1] Svarovsky L 2001 Solid-Liquid Separation Butterworth-Heinemann – p 554
[2] Records A Sutherland K 2001 Decaunter Centrifuge Handbook (Elsevier Science) p 440
[3] Krishnan R Electric 2001 Motor Drivers Modeling, Analysis and Control (Prentice Hall) p 626
[4] Merkl R and Steiger W 2012 Properties of decanter centrifuges in the mining industry Minerals and Metallurgical Processing 29 (1) pp 6-12
[5] Shelton J, Smith J R and Gupta A 2011 Experimental evaluation of separation methods for a riser dilution approach to dual density drilling Journal of Energy Resources Technology Transactions of the ASME 133 (3), pp 1-5
[6] Kladiev S N, Slobodyan S M and Pischulin V P 2014 Automation of Preparation of Uranium Solutions Tsvetnye Metally pp 77-82.
[7] Vajda I, Dementyev Y N, Negodin K N, Kojain N V, Udut L S and Chesnokova I A 2017 Limiting Static and Dynamic Characteristics of an Induction Motor Under Frequency Vector Control Acta Polytechnica Hungarica pp 7 – 27
[8] Kladiev S N, Kolodnikov I A, Maksimov I A and Pishchulin V P 2012 Improving the Preparation of Solutions pre-extraction with Providing Automated Start of Centrifuges in Apparatus for Reprocessing of Spent Nuclear Fuel Tsvetnye Metally pp 90-95