Вісник Тернопільського національного технічного університету
https://doi.org/10.33108/visnyk_tntu
Scientific Journal of the Ternopil National Technical University
2023, № 4 (112) https://doi.org/10.33108/visnyk_tntu2023.04
ISSN 2522-4433. Web: **visnyk.tntu.edu.ua**

UDC 539.3

DETERMINATION OF DYNAMIC CHARACTERISTICS OF THE CENTRIFUGE SHAFT

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Summary. *The paper presents the results of modeling the dynamic characteristics of the shaft of a laboratory centrifuge, which were compared with the results obtained analytically and experimentally. The obtained results showed the convergence of analytical and experimental data, in turn, the results obtained with the help of the KISSsoft software complex have overestimated values. The paper also provides determination of natural frequencies and forms of oscillations by the methods of the vibration theory.*
Key words: *centrifuge, natural frequencies, natural forms, Campbell diagram, KISSsoft.*

https://doi.org/10.33108/visnyk_tntu2023.04.032 *Received 29.09.2023*

Formulation of the problem. Centrifuges are used in various industries, in medical laboratories, agriculture, for separating mixtures into fractions consisting of substances having different densities.

Centrifuges are designed to provide a high degree of separation of the mixture. They have high-speed rotating elements in the form of rotors, which cause harmful vibrations. To ensure high quality separation, they must provide high rotational speeds and stability. these requirements can only be met if the dynamic characteristics are determined, the knowledge of which makes it possible to determine the critical frequencies, stable and unstable zones of the centrifuge movement.

Analysis of the available research results. The analysis of literature sources has shown that existing methods for calculating the dynamics of centrifuges are simplified [1–3] and are based on a model with a one-body rotation [4], which is the rotor, and are based on the kinetic momentum theorem, while the actual design of the centrifuge is a multi-mass system [5]. Based on a one-mass model, it is not possible to accurately determine the necessary information about the spectrum of the centrifuge's natural frequencies [6–7], which is important. The problem of determining the centrifuge motion leads to the need to consider the problem of motion of a system of bodies fixed on elastic supports [7–11]. Therefore, to solve this problem, a natural approach is to use the Lagrange equations of the second type [8]. The shaft is considered as a rotating elastic rod.

Objective of the research. To investigate the dynamics of shaft motion on the example of the PICO21 centrifuge by determining the natural frequencies and forms of oscillations, both by the methods of vibration theory and using the KiSSsoft software package, and to check the adequacy of the results obtained by comparing them with experimental data.

Statement of the problem. The RISO21 laboratory centrifuge (Fig. 1, Fig. 2) consists of such elements as a rotor 3 spinning around a vertical axis, which is also the shaft of an electric motor, whose rod 2 is located on the same axis. The stator of the motor 1 and the casing of the laboratory centrifuge are fixed on elastic supports-dampers. The supports 4 of the laboratory centrifuge casing are designed in such a way that the centrifuge can rotate relative to the fixed axes. The stiffness of the supports is the same when rotating about any horizontal axis [7, 8,

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11]. Compared to the masses of other bodies, the mass of shaft 5 is small, so the calculations did not take into account the weight of the shaft and considered it as a linear elastic beam.

In this research, the computational dynamic model of a centrifuge is considered as a multi-mass system with elastic elements, which takes into consideration the influence of gyroscopic effects [12]. The method under study reflects the determination of the spectrum of natural frequencies and natural oscillation forms of the centrifuge by taking into account the multimass and other design factors.

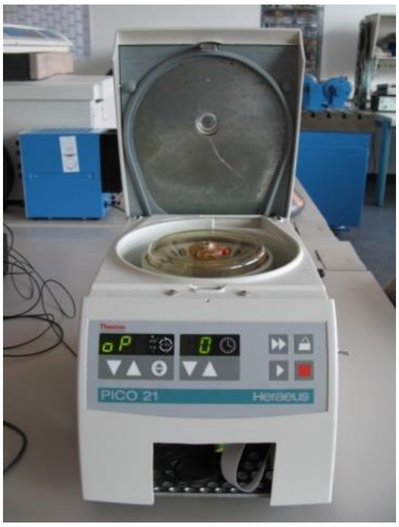

Figure 1. Laboratory centrifuge Pico 21 with a rotor

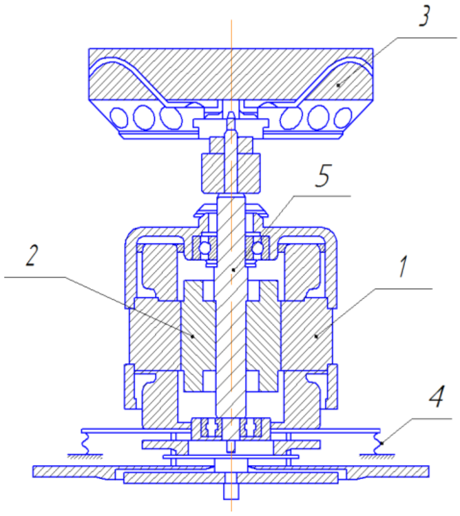

Figure 2. Sketch of the laboratory centrifuge Pico 21

As noted, the common approach to studying the dynamics of multimass systems is to use the Lagrange equation of the second kind [8].

$$\frac{d}{dt}\frac{\partial L}{\partial \dot{q}_i} - \frac{\partial L}{\partial q_i} = 0, \qquad (1)$$

where L is the Lagrange function, and q_i is the generalized coordinates. The Lagrange equations are differential equations that describe the motion of a system. The periodic motions of a centrifuge were considered. One of the variants of the Euler-Krylov angles was used to describe the motion of bodies [12].

The kinetic energy of the system is calculated as the sum of the kinetic energies of the bodies of which it consists, i.e. $T = T_1 + T_2 + T_3$, where T_1 – is the kinetic energy of the stator and centrifuge housing, T_2 – the kinetic energy of the motor rotor (anchor), and T_3 – the kinetic energy of the rotor. The potential energy is expressed through displacement as follows

$$\Pi = \frac{1}{2}\cdot(q_1, q_2, ..., q_i)\cdot\left\|C_{ij}\right\|\cdot(q_1, q_2, ..., q_i)^T, \qquad (2)$$

where c_{ij} are the components of the stiffness matrix, which is related to the yielding matrix $\left\|\delta_{ij}\right\|^{-1}$ by the relation $\left\|c_{ij}\right\| = \left\|\delta_{ij}\right\|^{-1}$, $i, j = \overline{1,12}$.

Using Lagrange's equations of the second kind, the differential equations of motion of the multimass system were obtained. Since the operating mode of the centrifuge is a trapezoidal cycle, i.e., start-operating speed-stop, we are interested in periodic motions, the desired functions were represented as harmonic functions that were substituted into the differential equations of motion (1). The result is a homogeneous system of linear algebraic equations. This system has non-trivial solutions provided that its determinant is zero. Since the components of the determinant depend on the rotational speed, the natural frequencies also depend on the shaft rotational speed [13–15]. The natural frequencies of the centrifuge shaft oscillations can either increase (direct precession) or decrease (inverse precession). As the shaft rotation speed increases, the frequency difference between the direct and inverse precession increases. The developed methodology was verified on test examples. The results of determining the natural frequencies are shown in Table 2.

To determine the natural frequencies and waveforms by the methods of vibration theory, the laboratory centrifuge shaft is considered as a system with two degrees of freedom [16] (Fig. 3).

The rotor mass and the total mass of the anchor and stator have the following values:

$$m_1 = m_r = 0,507kg,\ m_2 = m_a + m_s = 2,4 + 0,6 = 3kg .$$

To compare the determination of natural frequencies of oscillations, the force method, the Donckerley method, and the Rayleigh method are used.

1. Determination of the first natural frequency by the method of forces

The equations of the force method are as follows:

$$\begin{cases} w_1 = -\delta_{11} m_1 \ddot{w}_1 - \delta_{12} m_2 \ddot{w}_2 \\ w_2 = -\delta_{21} m_1 \ddot{w}_1 - \delta_{22} m_2 \ddot{w}_2 \end{cases} \quad (3)$$

For calculation, we take the displacement as $w_1 = W_1 \cos pt;\ w_2 = W_2 \cos pt$ and substitute it into the system of equations (3).

To obtain a nontrivial solution to the system of equations, we equate the determinant to zero:

$$\det \begin{vmatrix} \omega^2 \delta_{11} m_1 - 1 & \omega^2 \delta_{12} m_2 \\ \omega^2 \delta_{21} m_1 & \omega^2 \delta_{22} m_2 - 1 \end{vmatrix} = 0 \quad (4)$$

Expanding the determinant, we obtain the characteristic equation with respect to ω^2, from which we determine the squares of the natural frequencies ω_1^2, ω_2^2:

$$\omega^4 \left(\delta_{11}\delta_{22} - \delta_{12}\delta_{21} \right) m_1 m_2 - \omega^2 \left(m_2 \delta_{22} + m_1 \delta_{11} \right) + 1 = 0 \quad (5)$$

The displacements $\delta_{11}, \delta_{22}, \delta_{12}, \delta_{21}$ were calculated using the Vereshchagin method. To determine the corresponding displacements, bending moment diagrams were constructed from the action of unit forces $\overline{X_1} = 1, \overline{X_2} = 1$ applied at the places of concentrated masses $m_1,\ m_2$.

Substituting the obtained displacement values into the characteristic equation, we obtain the value of the first natural frequency $\omega_1 = 144,44\sqrt{EI}$.

Next, determine the first natural frequency of the waveform.

$$\begin{vmatrix} \omega_i^2 \delta_{11} m_1 - 1 & \omega_i^2 \delta_{12} m_2 \\ \omega_i^2 \delta_{21} m_1 & \omega_i^2 \delta_{22} m_2 - 1 \end{vmatrix} \begin{Bmatrix} W_{11} \\ W_{22} \end{Bmatrix} = 0 \quad (6)$$

Taking into account the first natural frequency $\omega_1 = 144,44\sqrt{EI}$ and normalized $W_{11} = 1$, we obtain the first natural waveform from the solution of equation (6):

$$\begin{vmatrix} \omega_i^2\delta_{11}m_1 - 1 & \omega_i^2\delta_{12}m_2 \\ \omega_i^2\delta_{21}m_1 & \omega_i^2\delta_{22}m_2 - 1 \end{vmatrix} \begin{Bmatrix} 1 \\ W_{12} \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \rightarrow \{W\}_1 = \begin{Bmatrix} 1 \\ 0,544 \end{Bmatrix} \quad (7)$$

The intrinsic waveform is shown graphically in Fig. 6.

2. Determination of the first natural frequency by the Donckerley method

The Donckerley method is a modal analysis method used to analyze dynamic systems and oscillations by determining the modal parameters of the system, such as natural frequencies. This method involves that the system oscillates with a fairly stable period and has no significant changes in its dynamics during the observation interval

$$\frac{1}{{\omega_1}^2} = \frac{1}{\left(\omega_1'\right)^2} + \frac{1}{\left(\omega_2'\right)^2}, \quad (8)$$

where $\left(\omega_1'\right)^2 = \frac{1}{\delta_{11}m_1}$, $\left(\omega_2'\right)^2 = \frac{1}{\delta_{22}m_2}$.

After the calculations, taking into account the obtained displacements, the corresponding masses, and substituting them into Equation (8), the value of the first natural frequency is obtained. The results of the calculations are shown below.

3. Determination of the first natural frequency by the Rayleigh method

The Rayleigh method is a superposition method used to analyze dynamic systems and oscillations by dividing the system into its constituent elements, determining their natural frequencies, and superposing these oscillations to obtain a general characteristic of the system. This method allows to determine the fundamental natural frequency of the system by analyzing the energy products of harmonic components

$${\omega_1}^2 = \frac{\sum_{i=1}^{2} Q_i W_i}{\sum_{i=1}^{2} m_i {W_i}^2}, \quad (9)$$

where $Q_i = m_i g$, i.e. $Q_1 = 0,507 \cdot 10 = 5,07N$, $Q_2 = 3 \cdot 10 = 30N$.

The values W_i are determined using the Vereshchagin method.

Analysis of numerical results. The results of determining the natural frequencies of oscillations analytically based on the Lagrange equation of the second kind are shown in the graphical form of the dependence of the natural frequency on the rotational speed, taking into account gyroscopic effects (Campbell diagram) in Fig. 4.

To verify the methodology for determining the natural frequencies, experimental studies of the centrifuge were carried out using laboratory equipment (Fig. 5) from the Institute of Mechanics of the Otto von Guericke University in Magdeburg (Germany).

To determine the critical frequencies, the centrifuge was accelerated in the range from 0 to 12000 rpm.

For the measurement of displacements, two triangulation displacement sensors Opto NCDT 2220 (micro-epsilon) ILD 2220-100 lasers were used, the beams of which are directed at an angle of 90° to the side surface of the rotating rotor of the laboratory centrifuge, on which a mirror tape was fixed. An amplifier of the NP-3414 type built into the laser was used to measure the signals (Fig. 5).

The signal was received using the DS-2000, which amplifies and extends the signal to the vibration sensor. To analyze the signal, a DS-0227 multi-channel station was used to process the data. The results of data processing and the trajectory of the rotating body were displayed on a PC screen.

In order to study the vibrations of the centrifuge shaft, vibration patterns were recorded at different operating speeds, which were used for further analysis. Spectral analysis of the vibrations made it possible to determine the natural frequencies of the laboratory centrifuge.

Based on the results obtained, a Campbell diagram was experimentally constructed (Fig. 4), which shows the dependence of natural frequencies on the rotational speed and demonstrates the influence of gyroscopic effects on natural frequencies and enables determining the operating modes in which resonant oscillations are possible.

To verify the reliability of the results obtained, the natural frequencies of oscillations of the centrifuge shaft were also calculated using the KISSsoft software package [17]. The data given in Table 1 were taken for modeling the shaft.

Table 1

Shaft dimensions

Diameter of shaft section	Length of shaft section
$d_1 = 3mm$	$L = 5mm$
$d_2 = 4,5mm$	$L = 12mm$
$d_3 = 4mm$	$L = 1mm$
$d_4 = 4,5mm$	$L = 10mm$
$d_5 = 15mm$	$L = 36mm$
$d_6 = 18mm$	$L = 14,2mm$
$d_7 = 16mm$	$L = 56,8mm$
$d_8 = 8mm$	$L = 8mm$
$d_9 = 4mm$	$L = 4mm$

A graphical representation of the shaft design in the KISSsoft software, according to the above dimensions, is shown in Figure 3.

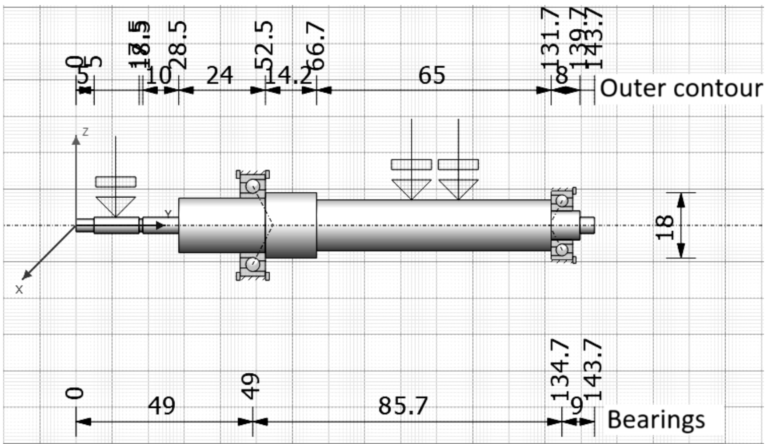

Figure 3. Schematic representation of the rod system

The results of simulation using the KISSsoft program are shown in Table 2 and Fig. 7.

Results of the research. Since the natural frequencies in dynamic systems depend on the gyroscopic effects arising from rotation, the Campbell diagram (Fig. 4) contains branches of natural values that are formed by splitting the gyroscopic forces. Forward and backward precession curves can appear if the movement is in the direction of rotation. Forward precession curves usually occur due to unbalanced rotation, while backward precession curves can occur due to other periodic forces.

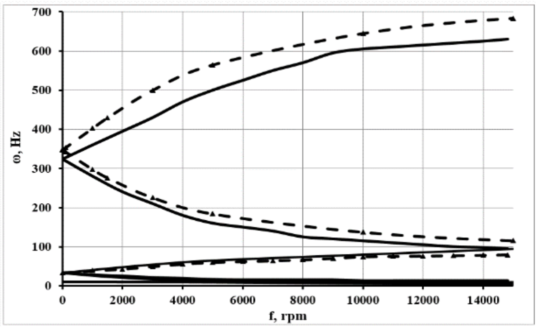

Figure 4. Campbell diagram, plotted experimentally and analytically

Figure 5. Experimental bench with equipment

Fig. 4 shows the dependence of natural frequencies on rotational speed, taking into account gyroscopic effects. The dotted line shows the values obtained analytically, and the solid line shows the data obtained experimentally. The results show that at low frequencies, the measurement accuracy is quite high and is within 5%. As the rotational speed increases, the accuracy begins to decrease, and depending on the rotational speed, it varies up to 20%.

Table 2 shows a comparative analysis of the first three natural frequencies without taking into account gyroscopic effects, obtained by three methods: experimentally, analytically, and as a result of simulation in the KiSSsoft software package.

Table 2

Comparative values of the first three natural frequencies

p_i	Experimental value, Hz	Calculation value, Hz	KISSsoft value, Hz
p_1	10	9.8	25.12
p_2	34.375	33.774	92.39
p_3	323.75	327.71	310.71

As a comparison, the values of the first natural frequency of bending vibrations of the shaft obtained by the methods of vibration theory are given below:

- by force method: $\omega_1^2 = 20862,9EI\ c^{-2} \to \omega_1 = 144,44\sqrt{EI}\ c^{-1}$
- by Donckerley method: $\omega_1^2 = 28384,9EI\ c^{-2} \to \omega_1 = 168,47\sqrt{EI}\ c^{-1}$
- be Raileigh method: $\omega_1^2 = 29180,7EI\ c^{-2} \to \omega_1 = 170,81\sqrt{EI}\ c^{-1}$

Discrepancy in the results obtained:

$$\delta_{\text{Donckerley}} = \left|\frac{170{,}81 - 168{,}47}{170{,}81}\right| \cdot 100\% = 1{,}37\%$$

$$\delta_{\text{force method}} = \left|\frac{168{,}47 - 144{,}44}{168{,}47}\right| \cdot 100\% = 14{,}26\%$$

From the values obtained, it can be concluded that the Rayleigh method gives overestimated values of the natural frequency compared to other methods, since the force method gives more accurate results.

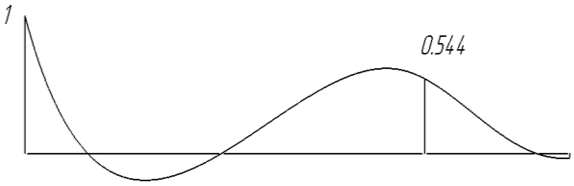

Figure 6. The first form of oscillations, obtained by force method

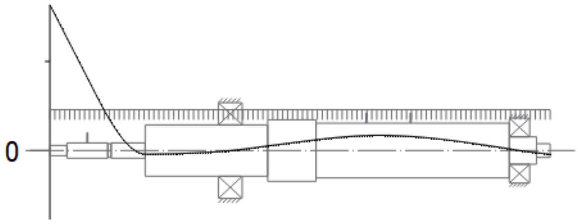

Figure 7. Shaft displacement, obtained by the KISSsoft

Figures 6 and 7 show that the waveforms obtained with KISSsoft are similar to those obtained with the force method.

Conclusions. The Campbell diagram showing the dependence of natural frequencies on the rotational speed with consideration of gyroscopic effects is constructed.

As a result of simulation using the KISSsoft software, a set of natural frequencies was obtained, which can be used to track the discrepancy between the obtained frequencies and the values obtained by experimental and analytical methods. This error is due to inaccuracies in the calculations and simulations in the KISSsoft software.

The results obtained make it possible to carry out the further necessary correction in order to improve the characteristics of the structure to increase both the reliability and the life of the system.

The graphs of natural waveforms reflecting the relative amplitude of movement of a structural element are constructed. The graph obtained using the methods of vibration theory is similar to the graph obtained using the KISSsoft software.

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УДК 539.3

ВИЗНАЧЕННЯ ДИНАМІЧНИХ ХАРАКТЕРИСТИК ВАЛУ ЦЕНТРИФУГИ

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Резюме. *Наведено результати моделювання динамічних характеристик валу на прикладі реальної лабораторної центрифуги PICO21 із застосуванням програмного комплексу KISSsoft як багатомасової системи. Представлена розрахункова модель враховує вал лабораторної центрифуги, ротор у якому розміщуються мензурки з речовинами різної фракції, анкера та статора. Отримані значення порівнювали з результатами, отриманими аналітичним та експериментальним шляхом. Аналітично власні частоти коливань валу центрифуги розраховували з застосуванням рівняння Лагранжа другого роду для динамічної моделі як багатомасової системи, оскільки існуючі методи визначення динамічних характеристик спрощені, розрахунки проводили як для одномасових систем та засновані на теоремі про кінетичний момент, що, у свою чергу, не описує реальну модель центрифуги, оскільки не враховує вплив усіх тіл, які створюють вібрації в конструкції. Представлена розрахункова модель враховує також вплив гіроскопічних ефектів, які виникають у результаті роботи лабораторної центрифуги. В результаті розрахунків побудовано діаграму Кемпбелла, яка відображає залежність власних частот коливань від швидкості обертання. Також, у свою чергу, використовуючи побудовану діаграму, можна визначити резонансні частоти, що надає можливість встановити до- та післярезонансні зони стійкої роботи центрифуги. Результати показали збіжність отриманих аналітичних та експериментальних даних, у свою чергу, результати, отримані за допомогою програмного комплексу KISSsoft мають завищені значення. Для перевірки достовірності отриманих значень виконано визначення власних частот коливань методами теорії коливань шляхом застосування методу сил, методу Донкерлі та методу Релея. Проведено порівняльний аналіз отриманих результатів. У результаті аналізу встановлено, що метод Релея дає завищені значення власних частот коливань у порівнянні з іншими методами. Власні форми коливань валу лабораторної центрифуги розраховували за допомогою представлених у роботі методів теорії коливань та програмного комплексу KISSsoft. Результати розрахунків показали схожу залежність, що підтверджує адекватність результатів моделювання валу.*

Ключові слова: *центрифуга, власні частоти, власні форми, діаграма Кемпбела, KISSsoft.*

https://doi.org/10.33108/visnyk_tntu2023.04.032 *Отримано 29.09.2023*