

The Approach to Automating the Control of the Vibrational Field of a Technological Machine

Kruglov S.P.

Subdepartment of automation of production processes
Irkutsk State Transport University
Irkutsk, Russia
kruglov_s_p@mail.ru

Kovyrshin S.V.

Subdepartment of automation of production processes
Irkutsk State Transport University
Irkutsk, Russia
sergkpw@mail.ru

Bolshakov R.S.

Subdepartment of operational work management
Irkutsk State Transport University
Irkutsk, Russia
bolshakov_rs@mail.ru

Abstract – The article considers the task of controlling a technological vibration machine, designed for the vibration interaction of a granular working medium with details attached to the working body of a vibration machine. The computational scheme of this kind of machine is a solid body on two springs with adjustable stiffness, which can be attributed to symmetric systems that execute a flat oscillatory motion. The task of the automatic control system is to maintain the vibration field at the required level, in particular, the amplitude of oscillations under conditions of a priori uncertainty of the parameters of the vibration machine. A control system is proposed to be built on the basis of a stabilizing regulator with static adaptation, capable of performing a target function with the object nonlinearity, external perturbations and having a simple structure. The features and conditions of functioning of this regulator are considered. A demo example is given.

Keywords — *technological vibration machine, vibration field, nonlinear control object, adaptive control system*

I. INTRODUCTION

Vibrational technological processes in modern productions in recent years are becoming quite widespread in engineering, mining, construction industries, at the enterprises of metallurgical and chemical engineering [1–3]. An operational aspect of vibration technological machines is the presence of working bodies creating vibration fields that form certain conditions for the dynamic interactions of the working medium with the products being processed [4–6]. Technological processes using the effects of dynamic interactions have become widespread in the case of vibration transportation of materials, classification, orientation of parts, etc., which is reflected in the work of domestic and foreign experts [7–9].

The processes of dynamic interactions of elements of a vibrating medium and processing objects have a wide range of forms and arising dynamic effects, which requires searching and developing appropriate methods, ways and approaches to constructing mathematical models that are needed to determine the necessary conditions for the implementation of technological processes that ensure a high level of product

quality or the implementation of certain functions with working media [5, 10–12].

The construction of mathematical models of vibration technological machines is associated with the consideration of issues of the dynamics of solids with several degrees of freedom, forming spatial mechanical oscillatory systems containing mass-inertial elements in the form of solids interacting with supporting surfaces and between themselves through elastic and dissipative connections [1, 13, 14].

The characteristic aspects of the dynamic properties of technological vibration machines, displayed by the computational schemes in the form of mechanical oscillating systems, are the parameters of the vibrational field created by the movements of the machine's working body, which is often a solid body with a mass and corresponding inertia moment. In the physical sense, the vibrational field can be regarded as a definite spatial distribution of the amplitudes of oscillations of the points of the worktable. Rather simple forms of vibrational fields arise with movements of the working body with two, less often with three degrees of freedom [14, 15].

In this regard, much attention in the problems of the dynamics of vibration technological machines is given to symmetric systems, reducible, under certain conditions, to simplified models in the form of mechanical oscillatory systems with two degrees of freedom, executing a plane motion.

The use of simplified computational schemes of technological machines, which is reflected in the works [14, 16, 17], allows developing analytical approaches and methods for automatic adjustment and correction of dynamic states of vibration stands, based on a theoretical apparatus focused on creating automatic control systems for the monitoring, evaluation and control of vibration fields. These approaches found their way in [5, 6, 12].

The necessity to take into account the features of dynamic interactions of elements of technological machines when

implementing movements in automatic modes initiated the development of methods for structural mathematical modeling. Within the framework of this approach, a mechanical oscillatory system with several degrees of freedom, considered as a simplified linear model of a technological object, is interpreted by a structural diagram of a dynamically equivalent automatic control system.

In the proposed work, a methodological basis is developed in solving the problems of building automatic control systems for the dynamic states of the working bodies of technological vibration machines.

II. STATEMENT OF THE RESEARCH TASK

A technological object (vibration table) is considered; its computational scheme in linear simplification is a system that performs flat oscillations in coordinates y_1, y_2 and y_0, φ , consisting of an extended solid body (M, J), as shown in Fig.1.

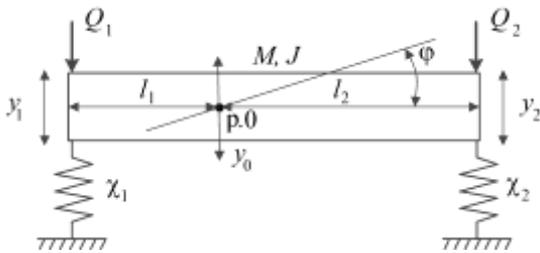


Fig. 1. The computational scheme of the vibration table

A solid body (table) is a working body, whose vibrations provide, under certain conditions, the implementation of the technological process created by the vibration interactions of a granular working medium with parts attached to the working body. In turn, the working body is brought into working condition by vibration exciters installed on it.

There are the following designations in the figure: M, J are the mass and central moment of the table inertia; $p.0$ is the center of gravity of the table, remote from the left and right edges of the table by the arms l_1, l_2 , respectively; y_0, y_1, y_2 are coordinates of the linear motion of the table in the center of masses, on the left and right edge, respectively; φ is the coordinate of the angular motion of the center of masses; χ_1, χ_2 is the adjustable stiffness of the springs (air bellows with adjustable pressure can act as such springs); Q_1, Q_2 are forces of interaction of the presented mechanical system with the external medium, we assume that $Q_i = Q - \mu_i \dot{y}_i + \xi_i$, $i = \{1, 2\}$, where Q is the exciting force, which is a harmonic signal with constant amplitude A_{exc} and frequency ω_{exc} , μ_i are coefficients of viscous friction applied to the left and right edge of the table respectively, ξ_i is the centered noise that simulates a force effect on the processed part and other external perturbations. The vibration table parameters $M, J, l_1, l_2, \mu_1, \mu_2$ are considered unknown and may drift over time.

Let us take the following assumptions:

- the table is an absolutely solid body;
- table oscillations are carried out with a small amplitude near the equilibrium state.

The task is to build an automatic control system that provides the values of the amplitudes of oscillations y_1, y_2 (we denote them as A_{y_1}, A_{y_2} , respectively) at a specified constant level with a wide range of values, regardless of the parameters $M, J, l_1, l_2, \mu_1, \mu_2$. We note that the equality of the oscillation amplitudes also corresponds to the problem of minimizing angular oscillations φ .

It is known [8] that this kind of a mechanical system is described by the following system of equations:

$$\begin{aligned} \ddot{y}_1 (Ma^2 + Jc^2) + \ddot{y}_2 (Mab - Jc^2) + \mu_1 \dot{y}_1 + \chi_1 y_1 &= Q - \xi_1; \\ \ddot{y}_1 (Mab - Jc^2) + \ddot{y}_2 (Mb^2 + Jc^2) + \mu_2 \dot{y}_2 + \chi_2 y_2 &= Q - \xi_2; \\ \varphi &= c(y_2 - y_1), \end{aligned} \quad (1)$$

where $a \triangleq l_2 / (l_1 + l_2)$; $b \triangleq l_1 / (l_1 + l_2)$; $c \triangleq 1 / (l_1 + l_2)$;

This control object with input signals in the form of stiffness of springs (χ_1, χ_2), and with output signals in the form of vibration amplitudes (A_{y_1}, A_{y_2}) is strongly nonlinear [18]. When solving problems similar to the one under consideration, systems of automatic stabilization of the amplitude of oscillations in the form of tracking systems using P- and PI-controllers are widely known [19–21]. However, these systems are effective only for linear control objects and those close to them. Under conditions of strong nonlinearity of the control object, the regulator must additionally implement dynamic compensation based on the known nonlinearity of the control object [22] or to have an algorithm for readjusting its parameters according to operation modes. All this requires a priori information about the parameters of the control object, which is not available in the task assigned. Another way implemented in practice is to experimentally adjust the parameters of the regulator according to the modes. As a rule, regulators of this type are used only in a small area of adjustable amplitudes and variations of the other parameters of the vibration table.

To stabilize the amplitude of oscillations, a sufficiently effective method of extremal control [23, 24] is also known, which provides control adjustment in the direction of the anti-gradient of the objective function. However, this approach is justified only if there is one global extremum of the objective function. In the problem of stabilizing the amplitude of a body having more than one elastic element, there can be many extremes. This is well known from the analysis of the amplitude-frequency characteristics of similar objects having many natural frequencies. Extremal control is possible here only in the region with one extremum, which needs to be somehow determined, and this region moves and transforms when adjusting the spring stiffness.

One of the possible solutions of the problem assigned is to build an adaptive control system. Work [25] would be an example. In this work it is proposed to implement a “combined robust-periodic control loop synthesized using the hyper-stability criterion” to ensure a specified amplitude of oscillations. However, this solution assumes complete information about the installation parameters, and the adaptation is carried out only to the payload mass placed on the vibration unit.

Other approaches are known to be used for building adaptive control systems. For example, with direct and indirect (identification) adaptive control [26]. However, they require the current estimation of the parameters of either the control object or the regulator parameters, which complicates the control system. Since the quality of transients in the problem under consideration is not critical, the most appropriate solution here is to use a simpler way of implementing adaptive control — in the form of a stabilizing regulator with static adaptation [27], which will be further considered.

III. THE STATIC ADAPTATION METHOD

Consider a stable (mandatory condition), non-linear and non-stationary scalar control object. Let its dynamics be described by the following differential equation (without taking into account the initial input and output conditions):

$$x(t) = \frac{b_m p^m + b_{m-1} p^{m-1} + \dots + b_0}{a_n p^n + a_{n-1} p^{n-1} + \dots + a_0} u(t) + \Delta x = \tilde{x}(t) + \Delta x(t), \quad (2)$$

where $u(t), x(t)$ is the input and output of the object, respectively; $\Delta x(t)$ is the external perturbation on the object, reduced to its output; $\tilde{x}(t)$ is the first term in the right part of (2), i.e. the output of the object without external influence; t is current time; unknown parameters of the object, which are generally described by dependencies: $b_i \triangleq b_i(x, t)$, $a_j \triangleq a_j(x, t)$, $i = \overline{1, m}$, $j = \overline{1, n}$, $i \leq j$; external perturbation $\Delta x(t)$ is also considered unknown; $p \triangleq d/dt$ is a differentiation operator; we consider that signals u and x are directly measured.

Another mandatory requirement for the source control object is to accept the condition:

$$k(x, t) \triangleq \tilde{x}_{std} / u_{const}; \quad \text{sign}[k(x, t)] = \text{const}, \quad (3)$$

where $k(x, t)$ is a nonzero gain factor of an object whose sign is known, in the problem in question it is strictly positive (hereinafter we will denote it as k); u_{const} is any permanent control; \tilde{x}_{std} is the value of the output of an object that is stable in condition after the end of the transient, corresponding to u_{const} ; the transient time is approximately known.

From the first equality (3) it follows that the coefficient of the control object can be determined on the steady-state process in the form of its evaluation (at first we consider that there is no external perturbation, $\Delta x \equiv 0$):

$$\hat{k}_{std} = x_{std} / u_{const}, \quad (4)$$

where \hat{k}_{std} is the gain ratio estimate for the steady state process; x_{std} is the steady-state value of the output of the control system corresponding to the constant control u_{const} .

Hence, the required control (u_{req}) that provides the achievement of the control goal without taking into account the dynamics of the transition process is:

$$u_{req} = x_{spec} / \hat{k}_{std}, \quad (5)$$

where x_{spec} is the specified output value (in the problem considered $x_{spec} > 0$).

However, the use of relations (4), (5) is impractical for a nonlinear non-stationary object, since requires a multiple system exit to the steady-state process, obtaining the estimate (4) on it and the corresponding recalculation u_{req} to gradually approach the specified steady-state process on the stabilization problem.

To eliminate this problem, we require that estimate (4) be carried out continuously, with the control u being close to u_{const} . For the considered control object, this means that the control speed must be much lower than the time of the object transient. Hence, instead of (4) and (5), we take the relations:

$$\hat{k} = x/u, \quad u = f_{lpf}(x_{spec} / \hat{k}), \quad (6)$$

where \hat{k} is the estimate of the gain ratio of the control object on the current control; f_{lpf} is the function of low-pass filtering with a cut-off frequency is much less than the reciprocal of the transient time of the object under consideration.

In accordance with the obtained solution (6), we will build the control system presented in Fig.2.

Figure 2 shows the following notation: \tilde{x}_0 is the initial value \tilde{x} (hereinafter the subscript "0" means the initial moment of time); the low frequency filter is represented as a selectable dynamic link with the aperiodic dynamics with the transition period is much more than the transient of the control object, the filter forms the control of the object (u); $\bar{x} = x_{spec} / x_{lim}$; there is a signal at the filter input ($\bar{x}u$); the limiter is used to eliminate random negative signal values x : $x_{lim} \geq 0$.

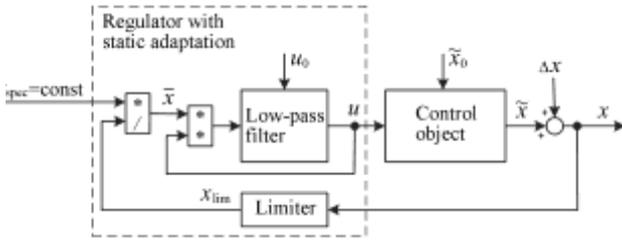


Fig. 2. The structure of the proposed stabilizing control system

A characteristic difference of the scheme in Fig. 2 from the classical tracking system with a PID controller is the use of the “multiplication-division” block at the input of the closed control loop instead of the “addition-subtraction”. In this regard, the control system according to Figure 2 is non-linear, and therefore the analytical analysis of its functioning is significantly complicated. Next, we consider the properties of this control system only at a qualitatively-logical level.

We consider the individual particular properties of this system with different initial values \tilde{x}_0 and u_0 , provided that there is no restriction in the system ($x_{lim} \equiv x$) and there is no external perturbation ($\Delta x \equiv 0$):

- if $u_0 = 0$, then from Fig. 2 and the properties of the object it follows that the further behavior of the system corresponds to $u \equiv u_0 = 0$ and $\tilde{x} \equiv x \rightarrow 0$, this state of the system will be called the “dead point”. One should not bring the system into this state;
- if $\tilde{x}_0 = \tilde{x}_{std0} = x_{spec}$, where \tilde{x}_{std0} corresponds to $u_0 \neq 0$, then $\bar{x}_0 = 1$ and the further behavior of the system will be: $\tilde{x} \equiv x \equiv \tilde{x}_{std0} \equiv x_{spec}$, $\bar{x}_0 \equiv 1$, $u \equiv u_0 \neq 0$, which corresponds to the steady state of the control system with the control goal achieved;
- if $\tilde{x}_0 > x_{spec}$, then $\bar{x}_0 < 1$ and this causes a decrease in the signal u , i.e. $\dot{u} < 0$;
- if $\tilde{x}_0 < x_{spec}$, then $\bar{x}_0 > 1$ and this causes an increase in u , i.e. $\dot{u} > 0$.

It is not difficult to determine that the rate of change of signal u in the last two cases is proportional to the magnitude of signal \bar{x} . If the change in signal u results in a stable target state of the control system, then this control limit value will be denoted as u_{std} .

Based on the obtained properties, we consider the behavior of the control system for different ratios $\tilde{x}_0 = x_0$ and u_0 (recall that $k > 0$). We also consider that there is no external perturbation and limitation x .

1). If $0 < u_0 \leq u_{std}$, and $\tilde{x}_0 \leq x_{spec}$ ($\bar{x}_0 \geq 1$), then this results in a smooth (due to the operation of the low-pass filter)

increase u until $u = u_{std}$ occurs, i.e. there is desired sustainable movement of the control system.

2). If $u_0 > u_{std}$, and $\tilde{x}_0 > x_{spec}$ ($\bar{x}_0 < 1$), then it gives a smooth decrease u before, as in the previous case, the desired target sustainable movement of the control system.

3). If $0 < u_0 \leq u_{std}$, and $\tilde{x}_0 > x_{spec}$ ($\bar{x}_0 < 1$), then this will lead to the fact that in the first stage u will slowly decrease, but due to the faster movement of the output of the control object, the signal \tilde{x} will quickly decrease until the condition $\tilde{x} = x \leq x_{spec}$ ($\bar{x} \geq 1$) occurs, i.e. the first of the cases under consideration, if this moment is regarded to be initial for it, and, consequently, this will result in a stable desired state of the control system.

4). If $u_0 > u_{std}$, and $\tilde{x}_0 \leq x_{spec}$ ($\bar{x}_0 \geq 1$), then the movement of the system, as in the previous case, is divided into two stages: the first is a slow increase of u and a rapid increase of \tilde{x} before the condition $\tilde{x} = x > x_{spec}$ ($\bar{x} < 1$) occurs, in the second stage the system moves according to the second case out of the considered ones.

Note that the logic of the functioning of the control system in the four cases considered is valid if the system does not get into the “dead point”. For this, as follows from the logic of reasoning with $\text{sign}(x_{spec}) = \text{sign}(\tilde{x}) = \text{sign}(x) = \text{sign}(k) > 0$, it suffices to assign:

$$u_0 > 0. \quad (7)$$

Now consider the effect of a non-zero external perturbation on the operation of the control system. Although its action shifts the gain ratio estimate by (6), it does not change the logic of the system, if the possibility of the system entering the “dead point” is eliminated. And this requires the fulfillment of the condition (in the problem in question $x_{spec} > 0$):

$$\Delta x < x_{spec}. \quad (8)$$

Indeed, if conditions (8) are not fulfilled, the signals u and u_{std} will have different signs, and while moving $u \rightarrow u_{std}$ the signal u should cross the zero point, which is the “dead point”, that is, the signal u will “stick” at this zero point. In this case, $\tilde{x} \rightarrow 0$, and $x \rightarrow \Delta x$. Although the state of the system will converge to a steady motion, but the target setting for stabilizing the output of the control system at x_{spec} will be performed with an error whose modulus is $|\Delta x - x_{spec}|$.

Let us estimate the effect on the adjustment of the nonlinearity of the control object, and with local extremes. Nonlinearity will be considered as a function $x_{std} = k(x_{std})u_{std}$. It can be argued that if the function

$k(x_{std})$, with the constancy of its sign (taken by condition), is unambiguous and changes quite smoothly – not faster than the control changes – then the control system will reach the target state. To prove this thesis, fig. 3 shows the process of adjusting the control system with an arbitrary nonlinearity $k(x_{std})$ — the curve shown in the figure.

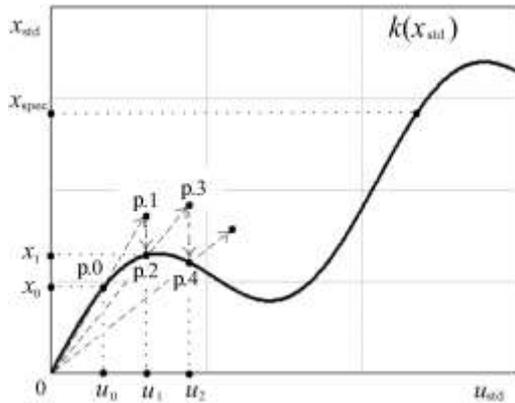


Fig. 3. The adjustment of the control system with non-linear object

We suppose that there is no external perturbation, the input and output variables are measured exactly, the system functions at discrete points in time (these assumptions do not change the essence of further reasoning).

Let the system be in the steady state (x_0, u_0) at the initial moment of time, i.e. at point 0 according to the figure. At this moment, the gain ratio of the object is estimated, whose value corresponds to the line connecting the origin with point 0. The control (denoted as u_1), corresponding to point 1, is calculated according to dependence (6). According to this control, due to the stability of the object, the system will be at point 2, corresponding to the steady state (x_1, u_1) . Next, a new estimate of the gain ratio and similar sequential movement of the system to the points 3 and 4. This movement occurs until the controlled variable x reaches the desired steady value x_{spec} . When this is achieved, the system will be in a steady state, as indicated by previous arguments.

Similar reasons can be produced for the nonstationarity of the object.

IV. FEATURES OF NUMERICAL SIMULATION

Consider the vibration table according to fig. 1 and relations of (1) with parameters: $m = 50\text{kg}$; $J = 7.2\text{kg m}^2$; $l_1 = 0.4\text{m}$; $l_2 = 0.6\text{m}$; $\mu_1 = \mu_2 = 500\text{Nm/s}$; external force Q has the parameters: $A_{exc} = 300\text{N}$, $\omega_{exc} = 9\text{Hz}$; spring stiffness coefficients (χ_1, χ_2) have a range of changes $5 \cdot 10^4 \div 2 \cdot 10^6 \text{ N/m}$.

To track the specified values of oscillation amplitudes y_1 and y_2 , two control systems were used according to Fig. 2 (to control two springs), where the amplitudes of these oscillations were used as a signal x (we assume that they are directly measured), and the inverse values of stiffness of adjustable springs were used as a signal u . The signal Δx corresponds to an external perturbation and is a centered random signal with a uniform distribution density with a root-mean-square deviation, generating an additional 10% deviation of the signals y_1 and y_2 , as well as the perturbation generated by a stepwise change in the amplitude of the signal Q by 100N at time 10s and step decreasing the frequency of the disturbing signal at 3Hz at time 15s.

By preliminary testing it was established that the vibration table transient time is about 0.8s. Therefore, an aperiodic link with a unitary gain ratio and a time constant of 1.5s was used as a low-pass filter (its transition process was about 4.5s [22]). For both control systems it was selected that signal $u_0 = 0.5 \cdot 10^{-6} \text{ m/N}$ - corresponds to the maximum stiffness of the springs, $x_0 = 0$. Note that the operating conditions of the control system (7) and (8) are satisfied.

Fig. 4 presents the results of the study of the functioning of control systems for the proposed method. The simulation was carried out in the Matlab environment.

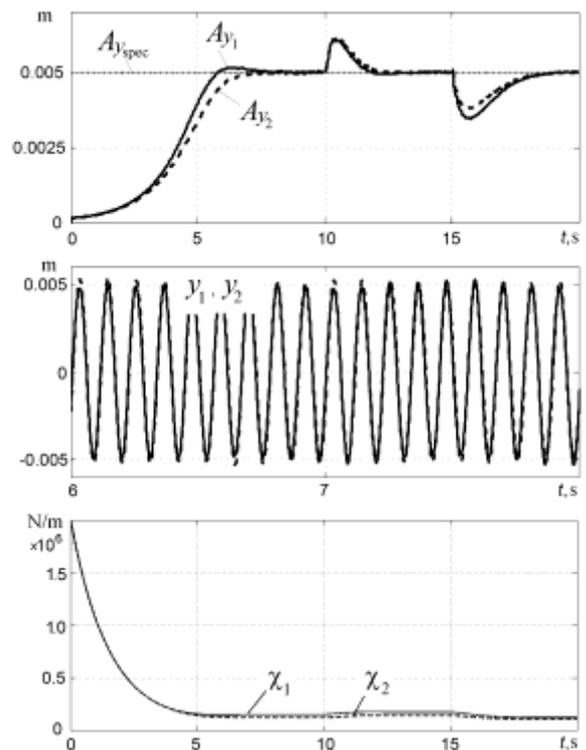


Fig. 4. Research of the proposed control system

From the figure we can observe that the specified value of the amplitudes of oscillations ($A_{y_{spec}} = 5mm$) is reached, and external perturbations are also countered. Angle oscillations (φ) are no more than 0.1 degrees. The curves y_1 and y_2 shown in the figure on the 6-8s time interval are the same in other parts of the stable amplitude of oscillations.

Similar results with the indicated perturbations were obtained for other specified amplitudes: $A_{y_{spec}} = 0.2 \div 17mm$, which corresponds to the implementation of the full range of amplitudes of oscillations in the subresonant zone of the vibration table. The same results were obtained for different specified values of amplitudes for y_1 and y_2 from the indicated range, as well as other perturbations.

V. CONCLUSION

The presented theoretical substantiations of a stabilizing regulator with static adaptation show the possibility of effective control of the vibrofield of the technological object under consideration under conditions of a priori uncertainty. The synthesized control system is easy to build and counters moderate perturbations.

References

- [1] R.F. Ganiev, V.O. Kononenko, *Oscill of solids*. Moscow: Nauka Publ., Fizmatgiz, 1976, 432 p.
- [2] I.F. Goncharevich, K.V. Frolov, *Theory of Vibration Engineering and Technology*. Moscow: Nauka Publ., 1981, 320 p.
- [3] P. Nessler, *Modeling and control in mechanical systems*. Uppsala Univ. Sweden. Retrieved from: <http://www.it.uu.se> 2005.
- [4] V.K. Astashev, V.I. Babitsky, I.I. Wolfson. et al., *Dynamics of machines and machine control*. Moscow: Mashinostroenie Publ., 1988, 240 p.
- [5] A.P. Babichev, I.A. Babichev, *The Basics of Vibration Technology*. Rostov-on-Don: The DGTU Center Publ., 2008, 693 p. Retrieved from: <https://elibrary.ru/item.asp?id=19569061>
- [6] Yu.R. Kopylov, *Dynamics of vibro-impact hardening processes: a monograph*. Voronezh: Voronezh: Nauchnaya Kniga Publ., 2011, 568 p.
- [7] P.A. Lontsikh, S.V. Eliseev, *Dynamic quality of machines and equipment as a tool to ensure production reliability and process competitiveness*. Irkutsk: IrGTU Publ., 2014, 322 p.
- [8] A.V. Eliseev, V.V. Sel'vinsky, S.V. Eliseev, *Dynamics of vibrational interactions of elements of technological systems taking into consideration unilateral constraints*. Novosibirsk: Nauka Publ., 2015, 332 p.
- [9] A.V. Eliseev, A.I. Artyunin, D.H. Nguyen, K.Ch. Vyong, "Vibration field control of a technological machine based on changes in reduced inertial properties. Vibration technologies, mechatronics and controlled machines", pp. 68–77 [Coll. of sci. articles on mat. of XII int. sci. and techn. Conf., in 2 parts. Kursk, 18–20 May 2016].
- [10] L.Ya. Banakh, M.L. Kempler, *Vibrations of mechanical systems with regular structure*. Springer series: Foundation of Engineering Mechanics, 2011, 350 p.
- [11] C.W. de Silva, *Vibration. Fundamentals and Practice*. Boca Raton, London, New York, Washington, D.C.: CRC Press, 2000, 957 p.
- [12] A.V. Eliseev, V.V. Sel'vinsky, S.V. Eliseev, *Dynamics of vibrational interactions of elements of technological systems taking into consideration unilateral constraints*. Novosibirsk: Nauka Publ., 2015, 332 p.
- [13] S.V. Eliseev, S.V. Kovyrrshin, R.S. Bolshakov, "Specific aspects of the construction of compact elastic elements in mechanical oscillatory systems. Interactions with the elements of the system and the form of the connections", *Modern technologies. System Analysis. Modeling*, iss. 4(36), pp. 61–70, 2012.
- [14] S.V. Eliseev, *Applied system analysis and structural mathematical modeling (dynamics of transport and technological machines: connectivity of motions, vibrational interactions, lever linkages)*, Irkutsk: IrGUPS Publ., 2018, 692 p.
- [15] S.V. Eliseev, P.A. Lontsikh, V.B. Kashuba, E.A. Parshuta, E.V. Kaimov, D.H. Nguyen, "New approaches in the dynamics of mechanical elastic-inertia systems with lever linkages", *Science and education in the life of modern society*, pp. 75–85, 2015 [A collection of scientific papers on the materials of the Int. Sci. and Pract. Conf. in 12 parts]. Tambov.
- [16] S.V. Eliseev, N.K. Kuznetsov, R.S. Bolshakov, D.H. Nguyen, "About the possibilities of using additional inertial constraints in the problems of dynamics of technical systems", *Bulletin of Irkutsk State Technical University*, no. 5(112), pp. 19–36, 2016. Irkutsk.
- [17] S.V. Eliseev, A.I. Artyunin, *Applied theory of oscillations in problems of the dynamics of linear mechanical systems*. Novosibirsk: Nauka Publ., 2016, 459 p.
- [18] *Vibrations in technology: A reference manual*. In 6 vols. Ed. Board: V.N. Chelomey (chair), vol. 4. Moscow: Mashinostroenie Publ., 1981., *Vibration processes and machines*. In Lavendel E.E. (ed.), 1981, 509 p.
- [19] Yu.A. Buryan, P.D. Balakin, V.N. Sorokin, "On the issue of stabilization of the oscillation amplitude of a mechanical system", *Omsk Scientific Herald*, no. 2(130), pp. 38–44, 2014.
- [20] E.A. Zagrivny, V.V. Ivanik, "Stabilization of the oscillation amplitude of an autoresonance asynchronous electric drive of a reciprocating rotational motion of a dynamically balanced drill string on a carrying cable", *J. of Mining Institute*, vol. 189, pp. 91–94, 2011. St. Petersburg.
- [21] V.M. Shestakov, E.V. Belokuzov, A.E. Epishkin, "Synthesis of laws governing the operation modes of automated vibration systems", *Elektrichstvo Publ.*, no. 11, pp. 31–36, 2013.
- [22] *Methods of classical and modern theory of automatic control*. In 5 vols. vol. 3: *Synthesis of regulators of automatic control systems*. In K.A. Pupkov, N.D. Egupov (eds.). Moscow: The Publ. of Bauman MGTU, 2004, 616 p.
- [23] B.R. Andrievsky, I.I. Blekhman, Yu.A. Bortsov, S.V. Gavrilov, V.A. Konoplev, B.P. Lavrov, N.D. Polyakhov, O.P. Tomchin, A.L. Fradkov, V.M. Shestakov, *Control of mechatronic vibration units*. In Blekhman I.I., Fradkov A.L. (eds.). St. Petersburg: Nauka Publ., 2001, 278 p.
- [24] Yu.A. Buryan, V.N. Sorokin, A.A. Kapelukhovskiy, "The control system of the radiation intensity of elastic waves downhole generator", *Omsk Scientific Herald*, no. 1(97), pp. 75–79, 2011.
- [25] E.L. Eremin, D.A. Telichenko, N.P. Semichevskaya, L.V. Chepak, E.A. Shelonok, *Control of technical systems in conditions of uncertainty*. Blagoveshchensk. AmSU Publ., 2014, 211 p.
- [26] *Methods of classical and modern theory of automatic control*. In 5 vols. vol. 5: *Methods of the modern theory of automatic control*. In K.A. Pupkov, N.D. Egupov (eds.). Moscow: The Publ. of Bauman MGTU, 2004, 784 p.
- [27] S.P. Kruglov, "The stabilizing controller with static adaptation", *Information technology and mathematical modeling in controlling complex systems: an electron. Sci. J.*, no. 2, pp. 1–11, 2019. Retrieved from: <http://ismm-irgups.ru/toma/23-2019>, free. Title from the screen. Rus., Engl. (reference data: 06/05/2019).