Vibration protection system with nonlinear elastic and damping characteristics

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Abstract. One of the leading areas of improving the road construction machines is to increase the comfort and safety of the human operator by reducing vibration effects. Therefore, the most simple and available way to implement this direction is applying the passive vibration isolation of a human operator’s seat. Vibration protection mechanisms of such seats typically have one translational degree of freedom. Studying the single degree of freedom vibration protection systems under the various external influences, in particular, with kinematic excitation of base displacements is a relevant topic of investigations. The research objective is to develop a mathematical model of a vibration protection system with variable stiffness and damping coefficients of a vibration isolator, as well as to optimize the mathematical model parameters. The following problems were solved: the development of the calculation model, the compilation of the model differential equations system, the description of the dependences of the stiffness force and damping coefficient of the vibration protection system using two-point Hermite splines. A type of the kinematic excitation characterized by one parameter - a constant base velocity, the evaluation criterion of the system response to this kinematic excitation - the maximum object acceleration and the integral criterion for assessing the system response to a spectrum or excitation range at the different speeds were proposed. The parameters specifying the type of the dependences of the stiffness force and damping coefficient of the vibration isolator were optimized by the integral criterion. The results are represented in the form of the mathematical model formulas, displacements and accelerations graphs and optimized values of the system parameters.

Key-words: vibration, vibration protection, quasi-zero stiffness, mechanism, power characteristic, damping

1. Introduction
As the performance, power and speed of the road construction machinery are improved, the need to protect operators and machine components from the damage caused by vibration increases [1]. Increased force effects on the working attachments of machines from the processed media are almost always variable and cause vibrations of machine components and operators controlling these machines [2]. Significant vibrations reduce the service life of the transmission and other components, have a negative impact not only on the efficiency but also the reliability of machines. Vibration suppression systems play an important role in solving the issues of increasing the performance of road construction
vehicles, improving their systems for protecting operators from negative impacts during operating process [3]. The vibration isolation improvement of road construction vehicles operators is achieved mainly by optimizing the cab mounting system on the base chassis and the human operator's seat in the cab [4].

Vibration isolation systems providing a certain range of the constant force values in the general range of the vibration-isolated object displacements relative to the base link are commonly called the systems with a quasi-zero stiffness section [5, 6]. The theoretical foundations for studying such systems in Russia were laid by the scientific school of a professor P M Alabuzhev [7]. The use of these systems is considered to be a promising direction of the vibration protection [8], since in the quasi-zero stiffness section, the vibration acceleration of the protected object is minimal when the base link is moving [9]. This is a section of the graph “force-displacement”. If we consider the single degree of freedom mechanical systems, then such systems with a section of quasi-zero stiffness can be described as having some conditional or real physical spring (or a combination of springs). Moreover, this spring will generally have a variable stiffness and its stiffness is close to zero in the quasi-zero stiffness section [10].

Vibration protection systems with a section of quasi-zero stiffness are currently widely used for vibration isolation of various objects [11]. In the case of the above-mentioned combination of real physical springs, the individual elements of the system can form mechanisms with so-called negative stiffness [12].

Typically, the mechanical systems as a combination of springs and levers, as well as rotary and sliding joints are used to create a negative stiffness (when not restoring, but deviating force increases, providing a mechanism deviates from the equilibrium position) [13].

Mechanisms based on the pneumatic rubber-cord devices are also used for vibration protection [14]. A separate direction in the field of passive vibration isolation of objects is utilizing the vibration-isolating metamaterials [15].

Active vibration protection systems are supplied with the external energy, most often the electrical one [16]. Magnetorheological polymer materials possessing highly elastic properties are also used [17]. At the same time, the advantages of the passive vibration protection systems include greater reliability and durability, as well as their energy independence [18].

An example is one of the simplest entirely mechanical vibration protection systems with a single degree of freedom described in paper [19]. Its mechanism consists of several rotary and sliding joints, levers and two pre-tensioned coil cylindrical springs. Another comparatively simple mechanism with a section of quasi-zero stiffness is a scissor mechanism in combination with conventional cylindrical springs [20]. These systems provide quasi-zero stiffness in a relatively small range of displacements near the mid-position.

It is evident that any vibration protection systems must have a characteristic that provide the object vibration protection under the various external influences. If they involve the frequency range of periodic impacts, then protection should be provided in the low, medium and high frequency ranges [21], since vibrations can be caused by vibrations of various elements of construction and road vehicles with different weights and locations of working bodies, transmissions, engines, etc. [22].

Vibration protection systems can have not only fixed but also adjustable parameters [23], besides nonlinear changes are possible not only in the stiffness, but also in the damping properties [24].

When developing new and improving existing vibration protection systems, the key issue is evaluating the systems behaviour under the external influences. For this purpose, studies are conducted using the developed mathematical models of vibration protection systems and the system response to the external influences is evaluated. Various approaches and parameters for evaluating the systems can be applied [25]. For example, in the process of the kinematic harmonic excitation, the vibration isolation coefficient, which is the ratio of the object acceleration to the base acceleration, is widely used [26]. In this case, the harmonic excitation is characterized by at least two independent parameters: the amplitude and frequency. It is appropriate to use such an asymmetric excitation of the system, which will be characterized by a single parameter and to investigate the vibration protection system behaviour in a certain range of this impact.
2. Problem statement
A dynamic model of the vibration protection system mechanism of the operator's seat of the road construction vehicles with adjustable parameters of stiffness and damping is considered (figure 1).

Figure 1. The calculation scheme of a single degree of freedom vibration protection system with variable stiffness and damping coefficients of the vibration isolator.

The model includes a concentrated mass of the vibration-isolated object, a conditional spring and a conditional damper. The following symbols were accepted: $m$, $z$ are the mass and vertical coordinate of the vibration-isolated object (human operator’s seat) in a fixed coordinate system; $c$ is the variable stiffness coefficient; $b$ is the variable damping coefficient; $z_{op}$ is the base coordinate in a fixed coordinate system; $z_{1}$ is the object coordinate relative to the base; $F$ is the stiffness force acting on the object’s body from the conditional spring.

In contrast to the known models with constant coefficients, the stiffness and damping coefficients of the considered model of the vibration isolator are variable functions of the relative coordinate $z_{1}$.

For the purpose of simplifying, it is appropriate to set the kinematic asymmetric excitation of the system, which will be characterized by a single parameter. Besides, the system response to this excitation in a wide range of this parameter variation from zero to the preset maximum value should be examined. Therefore, it is necessary to propose a criterion for evaluating the system response to a certain excitation.

Moreover, the criterion for evaluating a higher-level vibration protection system, which characterizes the system responses to excitations over the entire specified range of changing the parameter characterizing this excitation is required to be introduced. Finally, the parameters of the vibration protection system should be optimized by a higher-level criterion.

3. Theory
The dynamics equation for the mass vertical displacement of the vibration-isolated object of the seat, based on the Lagrange-d'Alembert principle can be represented as follows [27]:
\[ m \cdot \ddot{z} + b \cdot \dot{z} + F = 0 \]  
(1)

where:

\[ F = c \cdot z_1; \quad z_1 = z - z_{op} \]  
(2)

Hereinafter the points indicate the first time derivatives parameters. The stiffness force \( F \) like the stiffness coefficient \( c \) is a generally nonlinear coordinate function \( z_1 \) \((F=f(z_1), \quad c=f(z_1))\). It is thus advisable to proceed from setting the stiffness coefficient function as the initial data directly to setting the stiffness force function \( F=f(z_1) \).

To describe the dependencies of the stiffness force \( F=f(z_1) \) and damping coefficient \( b=f(z_1) \), two-point Hermite splines with the highest second-order derivatives were used. Analytical expressions of Hermite splines have the following form \([28]\):

\[ F = \text{sgn}(z_1) \left( s_{1F} \cdot |z_1|^3 + s_{2F} \cdot |z_1|^2 + s_{3F} \cdot |z_1| \right); \quad b = b_b + s_{1b} \cdot |z_1|^3 + s_{2b} \cdot |z_1|^2 + s_{3b} \cdot |z_1| \]  
(3)

where \( s_{1F}, s_{2F}, s_{3F}, s_{1b}, s_{2b}, s_{3b} \) are the constant coefficients depending on the coordinates and the first derivative of the Hermite spline endpoint:

\[ s_{1F} = \frac{3 \cdot \dot{F}_{\text{max}} z_{1kon}^2 - 6 \cdot \dot{F}_{\text{max}} z_{1kon}^3}{z_{1kon}^2 3}; \quad s_{2F} = \frac{7 \cdot \dot{F}_{\text{max}} z_{1kon}^3 - 15 \cdot \dot{F}_{\text{max}} z_{1kon}^4}{z_{1kon}^2 3}; \quad s_{3F} = \frac{10 \cdot \dot{F}_{\text{max}} - 4 \cdot \dot{F}_{\text{max}} z_{1kon}^2 \cdot \dot{F}_{\text{max}}}{z_{1kon}^2 3}; \]  
(4)

where \( z_{1kon} \) is the value of the coordinate \( z_1 \) at the right point of the two-point spline \( F=f(z_1) \); \( \dot{F}_{\text{max}} \) is the stiffness force value \( F \) at the right point of the two-point spline \( F=f(z_1) \); \( z_{1kon} \) is the value of the coordinate \( z_1 \) at the right point of the two-point spline \( b=f(z_1) \); \( b_0 \) is the damping coefficient value at the left point of the two-point spline \( z_1=0 \); \( \dot{b}_{\text{max}} = (b_{\text{plus}} + b_0) \) is the value of the damping coefficient \( b \) at the right point of the two-point spline \( b=f(z_1) \); \( b_{\text{plus}} \) is the additive component of the damping coefficient function at the right point.

The left point in both splines used has a zero abscissa value \( z_1=0 \), while the right point has a positive abscissa value. Equations (3) of Hermite splines include additions (using the absolute values of the coordinate \( z_1 \) and its sign function) necessary for the correct calculation of the functions values for the negative values of the abscissa \( z_1 \).

Gravity was not taken into account in the differential equation (1). The value of the stiffness force \( F \) at the left point of the two-point spline \( F=f(z_1) \) was assumed to be zero (the section of the so-called quasi-zero stiffness). The model under consideration is simpler, but simultaneously it is completely identical in its behavior to a more complex model taking into account the object's gravity. In the latter model, it is necessary to add a constant term compensating the gravity in statics to formula \( F \).

The first derivatives of the functions \( F \) and \( b \) at the left points of the two-point splines accepted zero values.

Kinematic asymmetric excitation of the vibration protection system was set as a uniform increment of the base coordinate:

\[ \dot{z}_{\text{op}} = v_{\text{const}}; \quad z_{\text{op}} = v_{\text{const}} \cdot t \]  
(5)

which starts at the zero point of the transition time \((t=0)\) and continues infinitely. Thus, the only parameter of the base displacement constant velocity \( v_{\text{const}} \) characterizes the kinematic asymmetric excitation.

The variables values of the displacement \( z \) and speed \( \dot{z} \) of the object's absolute coordinate in a fixed coordinate system at the initial time were assumed to be zero. The described effect on the system can be characterized as a stepped variation of the base velocity at the time zero.
The differential equation (1) solved relative to the higher derivative has the following form:

\[ \dddot{z} = -\frac{F + b \cdot \dot{z}}{m} \]  

(6)

Given that \( \dot{z} = v_{\text{const}} - \dot{z} \) and by replacing the variable \( \dot{z} = u \), the differential equation (6) can be represented in the Cauchy form, i.e. as a system of two first-order differential equations:

\[
\begin{align*}
\dot{u} &= -\frac{F + b \cdot (v_{\text{const}} - u)}{m}, \\
\ddot{z} &= u.
\end{align*}
\]

(7)

The expanded form is:

\[
\begin{align*}
\dot{u} &= -\left( s_1 \cdot \dot{z}^4 + s_2 \cdot \dot{z}^4 + s_3 \cdot \dot{z}^4 \right) + \left( b_0 + b_1 \cdot \dot{z}^4 + b_2 \cdot \dot{z}^4 \right) \cdot \left( v_{\text{const}} - u \right) \\
\ddot{z} &= u.
\end{align*}
\]

(8)

The differential equations system (8) can be solved by the known numerical methods [4, 7].

The maximum acceleration of the load in a fixed coordinate system \( \dddot{z}_{\text{max}} \) achieved during the transition process was accepted as a criterion for evaluating the vibration protection system behaviour of the lower hierarchical level and is calculated by the equation:

\[ \dddot{z}_{\text{max}} = \max \left\{ \dddot{z} \left( t \in [0, T] \right) \right\} \]  

(9)

where \( T \) is the final simulation time.

For a certain range of changes of the base displacement constant velocity value \( v_{\text{const}} \), the criterion for evaluating the higher level vibration protection system is the integral of the function \( \dddot{z}_{\text{max}} = f \left( v_{\text{const}} \right) \):

\[ q = \int_0^{v_{\text{max}}} \dddot{z}_{\text{max}} \, dv_{\text{const}} \]  

(10)

where \( v_{\text{max}} \) is the maximum value of the base displacement velocity \( v_{\text{const}} \) in the range under consideration \( v_{\text{const}} \in [0, v_{\text{max}}] \).

It is necessary to minimize the criterion (10) by the design parameters values of the vibration protection system. These are the parameters characterizing the curves size and shape of the system stiffness and damping features: \( b_0, z_{\text{konF}}, F_{\text{max}}, z_{\text{konb}}, b_{\text{plus}}, F_{\text{max}}, b_{\text{plus}} \).

The dimensionless coefficients \( k_F \) and \( k_b \), which characterize the shape of the curves \( F=f(z_1) \) and \( b=f(z_1) \) in the normalized relative coordinates should be considered instead of the values of the derivatives values \( F_{\text{max}} \) and \( b_{\text{plus}} \). The listed parameters have the following dependencies:

\[ F_{\text{max}} = k_F \cdot \frac{F_{\text{max}}}{z_{\text{konF}}}; \quad b_{\text{plus}} = k_b \cdot \frac{b_{\text{plus}}}{z_{\text{konb}}} \]  

(11)

Resulting from the substitutions of two optimized variables, the list of the optimized parameters of the system took the following form: \( b_0, z_{\text{konF}}, F_{\text{max}}, z_{\text{konb}}, b_{\text{plus}}, k_F, k_b \).

4. Experimental results
At a certain value of \( v_{\text{const}} \), the transition process can be simulated independently, i.e. a system of differential equations (8) can be solved at the constant values of the parameters \( b_0, z_{1\text{konF}}, F_{\text{max}}, z_{1\text{konb}}, b_{\text{plus}}, k_F, k_b \), as well as at the constant value of the vibration-isolated object mass \( m=200 \text{ kg} \).

A set of values of the above parameters defines the curves shape of the system stiffness and damping characteristics. As an example, figure 2a shows the curves of the stiffness and damping characteristics for the random values \( b_0=20 \text{ N/(m/s)}, z_{1\text{konF}}=0.1 \text{ m}, F_{\text{max}}=1000 \text{ N}, z_{1\text{konb}}=0.1 \text{ m}, b_{\text{plus}}=2000 \text{ N/(m/s)}, k_F=1.2, k_b=2.5 \).

The time dependences of the absolute acceleration, absolute displacement and relative displacement of the vibration-isolated object with the characteristics shown in figure 2a, at the base displacement speed \( v_{\text{const}}=0.1 \text{ m/s} \) and mass of \( m=200 \text{ kg} \) are given. In other words, the results of the transition process single simulation for the values of \( b_0, z_{1\text{konF}}, F_{\text{max}}, z_{1\text{konb}}, b_{\text{plus}}, k_F, k_b \) are given. This combination of the variable parameters values is not optimal by the integral criterion \( q \), i.e. by the sum of the maximum accelerations for all values of the base velocities. To minimize the value of the criterion \( q \), the values of the variable parameters were optimized.

During the computational experiment, the variation range of the base displacement velocity \( v_{\text{const}} \) was 
\[
0 \leq v_{\text{const}} \leq 1 \text{ m/s}.
\]

When varying the velocity \( v_{\text{const}} \), the sampling interval was 0.02 m/s.

At each value of \( v_{\text{const}} \) from the assessed range, the transition process was simulated independently, i.e. a system of differential equations (8) was solved at the constant values of the optimized parameters \( b_0, z_{1\text{konF}}, F_{\text{max}}, z_{1\text{konb}}, b_{\text{plus}}, k_F, k_b \), as well as at the mass constant value of the vibration-isolated object \( m=200 \text{ kg} \).

After completing the cycle of changing the values of \( v_{\text{const}} \) in the range of \( v_{\text{const}} \in [0;1] \text{ m/s} \), the integral criterion \( q \) value was calculated by formula (10). Moreover, according to the optimization algorithm, the values of some optimized parameters \( b_0, z_{1\text{konF}}, F_{\text{max}}, z_{1\text{konb}}, b_{\text{plus}}, k_F, k_b \) were changed and the cycle of varying the values \( v_{\text{const}} \) including calculating \( q \) was repeated.

**Figure 2.** Examples of the curves of the system stiffness and damping characteristics (a) and the single simulation results of the system transition process at \( v_{\text{const}}=0.1 \text{ m/s} \): the object accelerations in a fixed coordinate system (b); the object and base coordinates in a fixed coordinate system (c); the object coordinates relative to the base (d).
Simplex optimization from the starting point with the values of the optimized parameters \(b_0\), \(z_{1\text{kon}\, F}\), \(F_{\text{max}}\), \(z_{1\text{konb}}\), \(b_{\text{plus}}\), \(k_F\), \(k_b\) given above resulted in obtaining the optimal combination of these parameters: \(b_0 = 854.73\, \text{N/(m/s)}\), \(z_{1\text{kon}\, F} = 0.2\, \text{m}\), \(F_{\text{max}} = 2592.6\, \text{N}\), \(z_{1\text{konb}} = 0.05357\, \text{m}\), \(b_{\text{plus}} = 474.48\, \text{N/(m/s)}\), \(k_F = 2.045\), \(k_b = 1.296\).

5. Results discussion
During optimization, the maximum deviation value of the coordinate \(z_{1\text{max}} = \max (z_1)\) was limited by the penalty function method: \(z_{1\text{max}} \leq 0.1\, \text{m}\). The maximum deviation of the coordinate \(z_1\) is reached at the maximum value of the base displacement velocity of \(v_{\text{const}} = 1\, \text{m/s}\).

Figure 3a shows two curves of the functional dependencies \(z_{1\text{max}} = f(v_{\text{const}})\) used for calculating the value of the vibration protection system evaluation criterion \(q\) for two combinations of the optimized parameters values: nonoptimal and optimal ones.

![Figure 3. Curves for calculating the value of the vibration protection system evaluation criterion \(q\) for two combinations of the optimized parameters (a) 1 is nonoptimal, 2 is optimal; the curves of the system stiffness and damping characteristics for the optimized parameters optimal combination (b).](image)

A significant decrease in the value of the integral criterion for evaluating the vibration protection system \(q\) during optimization is observed. The curves of the stiffness and damping characteristics of the vibration protection system for the variable parameters values optimal combination are presented in figure 3b. Besides, the maximum deviation values of the coordinate \(z_1\) for the initial and optimal combination of the variable parameters values assumed close values of 0.113 and 0.1\, \text{m} correspondingly.

6. Conclusions
1. The calculation scheme of a translational single degree of freedom vibration protection system with variable stiffness and damping coefficients of the vibration isolator was developed. The values of the vibration isolator stiffness and damping coefficients are the functions of the vibration-isolated object coordinate relative to the base of the vibration isolator.
2. A dynamic model of a single degree of freedom vibration protection system mechanism of the road construction machines operator’s seat was developed. The model is presented as a system of two first-order differential equations and can be solved by the known numerical methods.
3. The functional dependences of the stiffness force and damping coefficient of the vibration protection system are proposed to be described by using the analytical expressions of two-point Hermite splines with the highest second-order derivatives. They make it possible to describe the quasi-zero stiffness range of the vibration isolation system for the small deviations and...
simultaneously to take into account the stiffness increase at the large deviations of the vibration-isolated object’s coordinate relative to the base of the vibration isolator.

4. It was suggested to examine the response of the developed vibration protection system to the asymmetric kinematic excitation in the form of a uniform increment of the base coordinate at the constant velocity, which starts at the time zero. Such excitation is characterized by the only parameter: the constant velocity of the base displacement $v_{\text{const}}$. The criterion for evaluating the vibration protection system response to the asymmetric kinematic excitation was the maximum acceleration of the vibration-isolated object in a fixed coordinate system.

5. Since the maximum acceleration of the vibration-isolated object depends on the value of the base displacement velocity $v_{\text{const}}$, which can change values in the operational conditions, the integral of the maximum acceleration function of the base displacement velocity is proposed to be used as a complex criterion evaluating the higher hierarchical level vibration protection system. The limits of changing the base displacement velocity from zero to 1 m/s at the interval of 0.02 m/s were considered.

6. Using the developed dynamic model, the simplex method was used to optimize seven parameters characterizing the curves shape of the stiffness force functions and damping coefficient of the vibration protection system. It was established that the value of the complex integral criterion evaluating the vibration protection system can be reduced during optimization in comparison with the random initial values of seven optimized parameters. Furthermore, the maximum deviation value of the vibration-isolated object’s coordinate can be simultaneously reduced relative to the base of the vibration isolator.

7. The developed mathematical model of the dynamic vibration isolation system with variable stiffness force and damping coefficient can be applied to describe the dynamics of the road construction machines operator’s seat, as well as makes it possible to simulate displacements and study the vibration protection system modes at the design stage.

7. References

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