Roller vibroprotection mechanism with a quasi-zero stiffness section

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Abstract. A new numerical method for designing the profile of the support surface for moving the vibration protection mechanism pressure roller for the operator's seat in road and construction machines has been developed. The roller vibration protection mechanism of the type under consideration provides for the required force characteristic with a quasi-zero stiffness section of a given long length. This advantage of the roller vibration protection mechanism makes it possible to recommend it for application in vibroprotection systems for road and construction machinery operators. Difficulties in creating such mechanisms are caused by the nonlinear nature of the system of algebra-differential equations that describe the relationship between the static force characteristic and the geometry of the mechanism support profile. That is why it is difficult to obtain a solution in the analytical form of a support surface profile and requires a numerical solution. As an example of the efficiency of the proposed method, we obtained a support surface profile for a roller vibration protection mechanism with a given force characteristic in the form of a three-segment spline with 0.1 meters length quasi-zero stiffness section. The developed method lets us use the specified parameter values as the initial data. In the Simscape Multibody application of the MATLAB system, a simulation model of a dynamic vibration protection mechanism system was developed. It allows simulating movements and studying the modes of this mechanism at the design stage. We give an example of the time dependencies of the mechanism points coordinates obtained using a simulation dynamic model for damped free vibrations.

Key-words: vibration, vibration protection, quasi-zero stiffness, roller mechanism, constant force

1. Introduction

In the market of road and construction machinery there is an increase in its speed, efficiency and energy saving. The increased productivity often relates to the growing of force effects on the vehicles chassis elements and working attachments [1]. High force actions are usually changeable and result in vibration of a machine units and operators [2].

It is known that the level of vibrations in the workplace significantly affects the operator's performance capability [3]. High vibration level increases fatigue and the number of control errors. In turn, the number of control errors directly affects the performance of the machine and reduces it [4] which can finally lead to accidents.
Vibroprotection mechanisms with a quasi-zero stiffness section in the displacement working area are considered as promising vibration isolation systems [5]. Mechanical vibroprotection units with quasi-zero stiffness with one degree of freedom are characterized by a conditional spring with very low stiffness at a certain point in the force-displacement curve [6]. Their study is caused by existing various objects vibration isolation issues [7].

Vibration protection systems with quasi-zero stiffness are usually used as one of main elements in various structures with negative stiffness [8]. These can be not only mechanisms with rotational and translational hinges [9], but also metamaterials. An elementary cell of such metamaterial is constructed by parallel connection of elements with positive stiffness and elements with negative stiffness [10]. Vibration protection systems of the active type are developed on the basis of electromagnetic elements [11], including magnetorheological elastomers [12]. Besides, passive vibroprotection systems have some advantages, since they do not require any electrical or electromagnetic components [13].

Vibration protection systems of various machines must have good characteristics of vibration isolation in the low frequency range [14]. This is also very relevant for road and construction vehicles [15]. In addition, it is desirable that the vibration isolator of the human operator's seat or the cab of road construction vehicles provides the specified damping characteristic [16]. Promising are the vibration protection systems with adjustable configuration parameters [17]. The simplest general example of a mechanical system with a quasi-zero stiffness section is a mechanism consisting of two tension springs located perpendicular to each other [18]. The simplicity of this mechanism can be considered as its advantage. The disadvantage of such jumping mechanisms is a relatively small range of movements, where the stiffness really takes values close to constant. To reduce this disadvantage, mechanisms with an increased number of springs are used [19]. Mechanisms based on pneumatic and rubber-cord shells are also used [20]. However, all of them have a complicated structure and still have some disadvantages.

Vibroprotection mechanisms with roller elements for creating negative rigidity have huge potential [21]. Their use theoretically provides for creating rather large sections with constant stiffness. Let us consider the use of the simplest of these mechanisms with a single roller and a curved surface.

2. Problem statement

It is necessary to develop a dynamic model of the vibration protection system mechanism for the operator’s seat of road construction machines using a roller and a curved surface (figure 1).
Figure 1. Design scheme of a mechanism with a roller providing quasi-zero rigidity due to a curved support profile.

The coordinate system $OXY$ is rigidly bound to the machine's base chassis. In the design diagram, $OX$ axis is directed vertically and coincides with the direction of movement of the vibration-isolated object, i.e. a seat with an operator. $OY$ axis is perpendicular to $OX$ axis and forms a rectangular Cartesian coordinate system with it. Along $OX$ axis, the vertical force $F_x$ acts on the vibration-isolated object from the side of the vibration protection mechanism, while the horizontal compression force $F_y$ acts along $OY$ axis. The inclination of the curved profile at the point of contact with the roller is lettered $\alpha$. The tension spring is characterized by a coefficient of stiffness $c_y$ and an equilibrium length $y_e$. The local coordinate system $O_1X_1Y_1$ relates to the profile, the beginning of which coincides with the upper point of the profile.

Due to the smallness of the values of $\alpha$, the following assumptions were made to simplify the solution of the problem: 1) the point of contact with the roller is located on a horizontal line connecting the rotation axis of the roller and the center of the operator's seat sliding joint; 2) the trajectory of the point on the rotation axis of the roller when it moves along the profile surface $y=f(x)$ is equidistant to this profile.

This mechanism should provide the specified force characteristic $F_x=f(x)$ in the displacement working area. Given that vertical displacement of the operator’s seat of road construction machines in motion can reach relatively high values, exceeding the range of displacement of their vibration isolation mechanisms, the force characteristic of the mechanism should include sections of smooth stroke limitation in addition to the quasi-zero stiffness section.

The characteristic can be set as either analytical expressions, or a vector of discrete numeric values (sequences of numbers). In both cases, it can have a form of a graph (figure 2).

![Figure 2](image) Smooth force characteristic of a vibration protection mechanism with a quasi-zero stiffness section in the range of 0.1...0.2 m given in the form of a three-segment spline (example)

To solve the problem, we need to find the profile $y=f(x)$ of the curved surface on which the roller moves, which corresponds to the specified force characteristic $F_x=f(x)$.

3. Theory

Let us denote: $\dot{y} = \frac{dy}{dx}$.

For the scheme shown in figure 1, the following algebra-differential relations are valid:
Relations (1) can be written as a system of two nonlinear algebra-differential equations:

\[
\begin{align*}
\tan(\alpha) &= \dot{y}; \\
F_y &= (y - y_x) \cdot c_y; \\
F_x &= F_y \cdot \tan(\alpha)
\end{align*}
\] (2)

In turn, system (2) can be reduced to a single first-order differential equation having the following form:

\[
\dot{y} = \frac{F_x}{y \cdot c_y - y_x \cdot c_y}
\] (3)

Differential equation (3) can be solved by applying numerical methods. For certain forms of the force characteristic \( F_x = f(x) \), the corresponding profile \( y = f(x) \) that meets the conditions of the problem may not exist. The result of numerical solution of the differential equation (3) is the profile \( y = f(x) \) of the vibration protection mechanism, along which the roller moves.

To study the dynamic properties of the described vibration protection mechanism, its simulation mathematical model was developed in the Simscape Multibody application of the MATLAB system (figure 3).

**Figure 3.** Simulation mathematical model of a roller vibroprotection mechanism with a section of quasi-zero stiffness

This model includes prismatic sliding joints blocks ( ) connected in a chain. The first two blocks of the type described (located on the left in figure 3) specify, respectively: vertical movements of the base chassis of a road or construction vehicle and vertical movements of the sliding joint 1 of the design scheme of the vibration protection mechanism (figure 1).

The third Prismatic Joint block, located to the right in figure 3, sets the horizontal movement of the roller axis relative to the vibration protection mechanism (sliding joint 2 of the design scheme).

The key feature of the developed simulation model, allowing to set the force characteristic of the vibration protection mechanism with a section of quasi-zero stiffness, is the consistent use of the Spline ( ) and Point on Curve Constraint ( ) blocks.

The former of these blocks, Spline, defines a three-dimensional curve (spline) that passes through the specified points. The coordinates of curve points are set in the block settings window. The curve can be open or closed. The intermediate values between neighboring specified points are calculated using cubic interpolation. In this example, a closed three-dimensional spline was used based on the obtained profile of the vibration protection mechanism \( y = f(x) \). The third coordinate \( z \) was set equal to zero for all points in the profile.

The Point on Curve Constraint block creates a kinematic constraint in the mechanical model between the point on the rotation axis of the roller and the profile curve of the vibration protection mechanism.
$y=f(x)$. This constraint allows the roller axis to move only along the curve of the profile that is rigidly connected to the base chassis of the machine. Serial connection of the two mentioned blocks as shown in figure 3 creates a kinematic constraint corresponding to the design scheme (figure 1). In the settings of the Prismatic_Joint3 block (sliding joint 2 of the design scheme), the spring stiffness coefficient $c_y$ and its equilibrium length $y_e$ are set. In addition, in the settings of the Prismatic_Joint2 block (sliding joint 1 of the design scheme), the coefficient of damping of vertical oscillations of the seat is set. The stiffness coefficient of the Prismatic_Joint2 block is assumed to be zero.

4. Experimental results
For a given smooth force characteristic of a vibration protection mechanism with a quasi-zero stiffness section, shown in figure 2, using the above method, the profile of the support surface on which the roller moves was obtained as an example (figure 4). The profile is shown in the local coordinate system $O_1X_1Y_1$, the beginning of which coincides with the upper point of the profile, the axes are aligned with the corresponding axes of $OXY$ coordinate system (figure 1).

![Figure 4](image)

**Figure 4.** The profile of the support surface on which the roller moves, in the local coordinate system $O_1X_1Y_1$ (example)

For the same surface, we obtained the functional dependence of the profile angle $\alpha$ on the vertical coordinate $x_1$ (figure 5).
Figure 5. Functional dependence of the profile angle $\alpha$ on the vertical coordinate $x_1$ (example).

The increment step of the vertical coordinate of the profile $x_1$ took the value $dx_1=0.0001$ m. Spring stiffness coefficient $c_y=60000$ N/m, its equilibrium length $y_e=0.2$ m. The specified force characteristic (figure 2) corresponded to the mass of the payload (a vibration-isolated object, i.e. a seat with a human operator) $m=203.943$ kg.

After obtaining the profile of the support surface on which the roller moves, based on the developed dynamic model of the vibration protection mechanism (figure 3), a computational experiment was conducted to study the attenuation of free oscillations of the vibration-isolated object. The damping coefficients of the mechanism along the vertical and horizontal axes took the values $b_x=100$ N/(m / s); $b_y=100$ N/(m/s). The initial value of the vertical coordinate took the zero value $x=0$ m.

The simulation results of the dynamic fluctuations are shown in figure 6 in the form of time dependencies: vertical coordinate of the movable part of the vibroprotection mechanism $x$ (figure 6a), the horizontal coordinates of the movable part of the vibroprotection mechanism $y$ (figure 6b) and the render window of points movements in a vibration isolation mechanism (figure 6c). The top left point coincides with the origin of $OXY$ coordinate system, the bottom left point – with the movable part of the sliding joint 1. On the right there is the support profile and the mobile point of is contact with the roller.
5. Discussion

The functional dependence of the profile angle $\alpha$ on the vertical coordinate $x_1$ (figure 5) shows that the profile angle decreases slightly at the quasi-zero stiffness section. The value of the angle in this section is about $9 \ldots 10^\circ$. The range of changes in the values of the horizontal coordinate of the profile is about 0.055 m while the vertical profile length is 0.3 m.

The analysis of the time dependencies of $x$ and $y$ coordinates of the roller in the $OXY$ coordinate system for free damping oscillations of the vibroprotection mechanism (figure 6) shows that the oscillations are damped with a gradual decrease in frequency, as the degree of entering the reflectors area outside the quasi-zero stiffness section decreases.

6. Conclusion

Summing up, the following conclusion can be made:

1. We have developed the design scheme of the mechanism with a roller and a curved surface. This mechanism can be used as a vibration protection system for the operator's seat in road construction vehicles. The mechanism comprises two sliding joints directed perpendicular to each other, a rotational joint (roller axis), a roller and a support curved surface on which this roller moves.

Figure 6. The results of simulation of vibroprotection mechanism oscillations (example): (a) - time dependence of the $x$ coordinate; (b) - time dependence of the $y$ coordinate; (c) is the window for visualization of vibroprotection mechanism point movements in MATLAB (on the left - points of $OXY$ coordinate system, on the right - support profile and its contact point with the roller)
2. For this mechanism, we have developed a method for generating the profile of a curved surface along which the roller of the vibration protection mechanism moves. Based on the described method we can create a profile, the movement of the roller along which provides a given force characteristic of the vibration protection mechanism, provided that the latter is continuous and smooth.

3. By using the developed technique for generating the profile of a curved surface, as an example, we obtained a profile for the force characteristic, represented as a three-segment spline with a section of quasi-zero stiffness in the range of displacements of 0.1...0.2 m. The total range of vertical movement of the mechanism, set by the force characteristic, was 0.3 m. At the same time, the range of changes in the values of the horizontal coordinate of the profile was about 0.055 m. The range of changes in the profile angle was about 50 degrees. In the quasi-zero stiffness section, the angle of inclination was about 10 degrees with a spring stiffness coefficient of 60000 N/m and an equilibrium spring length of 0.2 m. The mass of the vibration-protected object in this case was 203.943 kg, which approximately corresponds to the mass of the seat and an operator of a real machine and the value of the force on the quasi-zero section equal to 2000 N.

4. The developed method lets us use the specified parameter values as the initial ones. Additional research has shown that for a certain combination of values of the initial parameters, the problem does not have a solution, i.e. the desired profile cannot be obtained.

5. The simulation model of a dynamic system vibration isolation mechanism developed in the application Simscape Multibody of the MATLAB system let us simulate movements and study the modes of the mechanism at the design stage.

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