Vibroprotective mechanism with the quasizero stiffness for seats of operators of construction and road machines

S V Tomleeva\(^1\), A G Saveliev\(^2\), G D Moiseev\(^2\) and P G Kolesnikov\(^3\)

\(^1\)Bryansk State Engineering-Technology University, 3, Dimitrova prospect, Bryansk, Russia
\(^2\)Moscow Automobile and Road Construction State Technical University (MADI), 64, Leningradsky prospect, Moscow, Russia
\(^3\)Siberian State Technological University, 82 Mira prospect, Krasnoyarsk, Russia

\(^*\)E-mail: tomkuzsv@mail.ru

Abstract. The paper considers vibration isolation device with a quasi-zero stiffness, which nonlinear element is made in the form of a closed elastic rod, an elastic line described by the circle. The authors list the features of stiffness and strength characteristics of the device, and select the parameters that ensure the full protection on a portion of the suspension.

1. Introduction

The effect of vibration on the human body has been studied to some extent [1], which resulted in requirements for the permissible level of vibration at the workplace, which should be taken into account when equipping the workplace of the operator of construction and road vehicles.

Approaches to solving this problem are following: passive vibration protection, where elements with unregulated (constant) and adjustable stiffness are used and shock absorber resistance coefficients are selected accordingly; active vibration protection based on the implementation of instantaneous counteraction to the input disturbance; pulsed vibration protection based on the use of systems with an abrupt change in the stiffness of the suspension elements. There is also a new developing direction of vibration protection based on the use of devices of forming a nonlinear stiffness characteristic having quasi-zero values in the position of static equilibrium of the protected object in elements of vibration protection. These devices function as passive, having the properties of active vibration protection devices. Elements that provide quasi-zero stiffness are hopping elements that have a different structural solution (Mises truss, popping membrane, and others).

2. Setting the problem

One of the main tasks in calculating a vibration isolator is to ensure a low frequency of its free vibrations, since the smaller it is, the wider the frequency range of the driving force, which ensures optimal vibration...
protection. Achieving a low frequency of free oscillations is possible only when using a "soft suspension" with elastic elements with a low stiffness coefficient. The presence of a segment with quasi-zero stiffness on the graph of the stiffness of the hopping elements gives reason to consider them as elastic elements of vibration protection systems.

The disadvantages of such systems are: a relatively small segment of the power characteristic, which provides optimal vibration protection; ensuring the position of the static equilibrium of the system when changing the weight of the protected object (weight of the driver). However, recently, these problems have been successfully solved [2]. When pneumatic springs and controlled dry friction are included in the vibration-protective system, it is possible to achieve an increase in the required length of the power characteristic up to 10 times and return the system to an unstable equilibrium position automatically.

In this paper, we consider the vibration-proof device of the driver's seat of a construction vehicle driver, which includes a non-linear suspension of quasi-zero stiffness.

3. Analytical Model
The scheme of the device is presented on the figure 1 [3].

![Figure 1. General view of the vibration protection device.](image)

The element forming the nonlinear stiffness function of the device is made in the form of a closed elastic rod. When mechanical vibrations are transmitted through the body 1 to the protected object 2 in a spring of constant stiffness 3, a dynamic elastic force arises due to displacement from the position of static equilibrium. In parallel with the spring, the hopping element 4, which is an elastic closed rod of rectangular cross-section, is also shifted in the vertical direction, slipping between two rollers pivotally attached to the protected object 2. The distance between the centers of the rollers is fixed and always less than the diameter of the elastic ring, therefore, when moving the rod between the rollers at the contact points, some forces arise in the vertical direction, which are opposite to the action of the spring elastic force 3. Thus, the external action forms two oppositely directed parallel forces that compensate each other friend and do not allow excitation to be transmitted to the protected object 2.

To analyze the vibration-protective qualities of the proposed device, it is necessary to obtain the power characteristic of the device, to make sure that it has segments characteristic of non-linear suspensions of quasi-zero stiffness.

Let's consider one of the parts symmetrically to the vertical (Figure 2). The action of the rollers on the rod is replaced by the action of a concentrated force P applied at the point of contact of the roller with the rod. To determine the stiffness function of this element, we will use the technique described in
[4], according to which it is necessary to find the vertical displacement of the point of application of force \( P \) and divide the force into this displacement:

\[
C = \frac{P}{f}.
\]  

\( \text{Figure 2. Calculation scheme of the core element.} \)

We restrict ourselves to considering a rod of small curvature, since in our case the condition \( h/R \ll 10 \) is fulfilled, where \( R \) is the radius of curvature of the elastic line of the rod, \( h \) is the height of the section of the rod.

To calculate the displacement of the rod, we use the Mohr method. For rods of small curvature, the displacement of the point of application of force is determined by the formula:

\[
f = \int_0^1 \frac{Mds \partial M}{EJ \partial P},
\]

Where \( E \) – elastic modulus of the rod material, \( J \) – the moment of inertia of the cross section of the rod, \( M \) – the bending moment acting in the section of the rod, \( P \) - the acting force. The expression of bending moment includes \( Y_B, M_B \). To determine them, we use the condition for fixing the end section B of the rod:

- the vertical movement of point B is zero \( (f_B = 0) \);
- the angle of rotation of the cross section of the rod at point B is zero \( (\theta_B = 0) \).

These conditions are expressed by the formulas:

\[
\begin{align*}
  f_B &= \frac{1}{EJ} \int_s M \frac{\partial M}{\partial Y_B} \, ds = 0, \\
  \theta_B &= \frac{1}{EJ} \int_s M \frac{\partial M}{\partial M_B} \, ds = 0
\end{align*}
\]  

(3)
Having solved the resulting system of equations, we find \( Y_B, M_B \). Substituting them into the expression of the bending moment \( M \), we find \( f \) by the formula (2):

\[
f = \frac{R^3}{El} \left[ P \left( \frac{\pi}{2} - \frac{\alpha}{2} + \frac{3 \sin 2\alpha}{4} + (\pi - \alpha)(\cos \alpha)^2 \right) + Y_B \left( -\sin \alpha - \frac{\pi}{2} + \frac{\alpha}{2} - \frac{\sin 2\alpha}{4} - \cos \alpha(\pi - \alpha) \right) + M_B \left( \sin \alpha + \cos \alpha(\pi - \alpha) / R \right) \right]
\]

where \( Y_B = \frac{P}{\pi} (\pi - \alpha + 0.5 \sin 2\alpha) \);

\( M_B = \frac{PR}{\pi} (\pi - \alpha + 0.5 \sin 2\alpha - \sin \alpha - \cos \alpha (\pi - \alpha)) \).

Knowing the stiffness function of the elastic rod defined by formula (1), we obtain the force characteristic \( F \) of the elastic element in the \( x \)-axis direction:

\[
F = P \cos \gamma \cos \alpha,
\]

where \( P = C \cdot v \);

\( v = \Delta - R(1 - \sin \alpha) \);

\( \Delta \) – specified preload of the center point at which \( x = 0, \alpha = \frac{\pi}{2} \).

Since \( \gamma = 90^\circ - \alpha \), when eventually

\[
F = \frac{1}{2} P \sin 2\alpha
\]

The graph of the power characteristic \( F(x) \) is shown in Figure 3 by a dashed line.

### 4. Results

The calculations were carried out with the following numerical values: \( h = 0.002m \) – section height of a flexible rod; \( b = 0.012 m \) – section width; \( R = 0.20m \) – radius of curvature of the elastic line of the rod; \( E = 2.1 \times 10^5 MPa \).
Figure 4. Graph of the power characteristics of the vibration protection device.

As can be seen, in the vicinity of zero, the stiffness of the device is close to zero. Thus, this vibration protection device refers to the type of devices with quasi-zero stiffness and has all their properties.

5. Conclusion
As a result of the calculations, the power and stiffness characteristics of the vibration protection device with the hopping element in the form of a closed rod are obtained. The nature of the graphs shows that the calculated device refers to the type of devices with quasi-zero stiffness and, with a certain selection of parameters, can provide optimal vibration protection over a wide range of input disturbance frequencies.

References
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