Systems of vibration isolators based on elements with linear characteristics

P D Balakin, O S Dyundik and I P Zgonnik
Omsk State Technical University, 11, Mira Ave., Omsk, 644050, Russia

Abstract. Actual problems are applied issues that are related to vibration isolation of machine assemblies and operator seats in machines. The publication presents some technical solutions of vibration isolator systems. These systems are simple in design and include elastic elements with a linear stiffness characteristic, but with a special arrangement to the vibration displacements of the protected object. In systems of vibration isolators "with leap" in a specific range the effect of quasi-zero stiffness of the support is realized with an ideal vibration isolation of the object. Technological and universal technical solution for the vibration-isolating seat of transport machine operator is described in the publication.

Key-words: vibration isolator, elastic member, natural oscillation frequency of system, quasi-zero stiffness, linear characteristic

1. Introduction
Increasing specific power of the transformed power flow in machine assemblies while reducing the materials consumption of machine parts is one of the most important areas of development of modern mechanical engineering. The transition to the use of composites and light alloys in the configuration causes increase the natural oscillation frequency of machine elements.

The abilities of structural materials with performance criteria such as strength and stiffness are used effectively in the configurations of modern machines, assemblies, components and machine elements. But traditional technical solutions of mechanical machine systems in the new conditions do not meet the criteria of vibration activity, which negatively affects the reliability and resource of machines, machine control systems and, most importantly, harms the health of service personnel and machine control operators. Consequently, for transport machines, the problem of vibration isolation of the operator from kinematic vibration excitation is actual.

2. Problem statement
The problems of finding a rational technical solution for the vibration-isolating seat of the transport machines operator involves evaluating a wide range of vibration excitation arising from the power plant, assemblies and components of the machine and from the roadway. The latter is low-frequency and particularly dangerous, its frequencies are close to the resonant frequencies of internal organs and individual parts of the operator's body, which is established experimentally [1].

When the frequency of vibration excitation approaches the resonant ones, the operator's functional state deteriorates: visual function decreases, movement coordination decreases, cardiac activity worsens, reaction time increases and so on.
The separation of the operator from the source of vibration excitation by a linear elastic element of low stiffness provides a technical solution for vibration isolation of this part of the seat of significant dimensions and large movements of the operator relative to the machine controls, which complicates the control of the machine.

The rationality of the technical solution of the transport machine operator's seat is estimated by the degree of complexity of the design and its technical feasibility. On the use of linear elastic elements in the support part [2], placed so that their combination together forms a "soft" support with a nonlinear characteristic, and, if possible, with areas where the effect of quasi-zero stiffness occurs, are pay attention [3, 4, 5].

3. Theory

On figure 1 vibration protection systems consist of three components: the source of vibration excitation (dithering) - S, the object of vibration protection (O) and the vibration isolators (VI). As a rule, the source and object are represented as solid bodies, and the dominant vibration displacement is uniaxial and occurs along some common "x" axis. Scheme (Fig. 1. b) according to which the carrier mass creates a kinematic excitation \( x = x(t) \), and the operator \( O \) support load and separated from the bearing member by a vibration isolator is used to simulate the operator's vibration protection.

Vibration isolator (VI) is designed to weaken the vibrational energy coming from the source (S), which eliminates its impact on the object. Usually, the vibration isolator is represented inertialess, and the intrinsic reactions \( R \) and \( R' \) must be minimal under the conditions of the problem. In fact, the VI is a buffer isolation of the S and O for the passage of vibrations, but retains the functional connection of the S and O.

For uniaxial vibration isolator reaction \( R \) and \( R' \) are equal in magnitude and opposite in sign \( \bar{R} = -R' \), response quantities of linear vibration isolator is proportional to the relative displacement \( \delta \) of the source and the object and a speed of deformation \( \delta' \) of the vibration isolator, i.e.

\[
R = s\delta + d\delta', \tag{1}
\]

when \( s \) and \( d \) are the stiffness of the elastic joints and the damping ratio of the vibration isolator. When \( d=0 \), the vibration isolator will only be elastic, structurally simple, so when creating a vibration-isolating seat, are adhere to the scheme with minimal damping.

According to (1) with limited deformation \( \delta \), the minimization of \( R \) is achieved due to the low stiffness "c" of the elastic elements. The stiffness must be sufficient to allow for the perception of a static load, namely the weight of the operator.

The effectiveness of the vibration isolator is implicitly estimated by the natural oscillation frequency of the system with the vibration isolator. The lower the natural frequency value, the system is potentially more effective for the purpose of vibration isolation [4].

For a uniaxial linear system, the natural oscillation frequency \( k_0 \) is determined by a known formula
\[ k_0 = \sqrt{\frac{c}{m}}, \quad (2) \]

when \( m \) is mass of the support load object. \( m = \frac{G}{g} \), when \( G \) is weight of the object, \( g \) is acceleration of gravity.

\[ c = \frac{G}{\delta_{st}}, \quad \text{where} \quad \delta_{st} \] is the static deformation of the vibration isolator, then

\[ k_0 = \sqrt{\frac{g}{\delta_{st}}}, \quad (3) \]

The damping coefficient of the system is determined by a known formula

\[ n = \frac{v}{2m}. \quad (4) \]

In case of force excitation, the amplitude of the reaction force \( R_0 \) transmitted to the object:

\[ R_0 = \frac{H}{\sqrt{k_0^2 + 4n^2p^2}}, \quad (5) \]

when \( H \) and \( p \) are amplitude and frequency of the exciting force. Effectiveness of the vibration isolator is determined by the ratio \( k_R = \frac{R_0}{H} \) or \( k_R = \frac{cx_0}{H} \), when \( x_0 \) is maximum relative displacement support load of the body (object).

For kinematic excitation, the external force excitation \( F(t) = 0 \). Induced vibrations of the base (Fig. 1b) is taken as a harmonic \( x = x_0\sin pt \). Then the amplitude \( A \) of the object's acceleration is determined:

\[ A = \frac{x_0p^2}{\sqrt{(k_0^2 - p^2)^2 + 4n^2p^2}}, \quad (6) \]

For a absolute elastic vibration isolator is \( v = 0 \), therefore is \( n = 0 \), then

\[ A = \frac{x_0p^2k^2}{k_0^2 - p^2}, \]

and the effectiveness of the vibration isolator is additionally evaluated by the vibration isolation coefficient \( k \)

\[ k = \frac{A}{x_0p^2}, \quad (7) \]

which depends on the frequency \( p \) of vibration excitation and it is desirable for vibration isolation that the coefficient has a value \( k \leq 1.0 \).

4. Results discussion

An engineering interest is a scheme with a load-bearing elastic element with a linear stiffness characteristic. The carrier element 1 in Fig. 2 in the initial state, it takes the weight load from the force of the weight of the protected object. In the case of static deformation of the load-bearing element 1, the corrective elastic elements 2, having their initial pressure deformation, are located horizontally and perpendicular to the vectors of the weight force and external force excitation. This scheme refers to schemes «with leap», the force effect of which is that in the starting state, the corrective elastic elements are mutually balanced and do not have a projection on the vertical axis of the OY. The pressure force of the correction elements forms a projection along the OY axis when an external force \( P \) is applied along this axis in any direction and an object's displacement \( \delta_y \). The projection of the
pressure force actively affects the system of forces that create movement for the vibration protection object.

The engineering calculation of parameters and the principle of operation of the technological vibration isolator will be given. Calculate the force interactions in the support «with leap» (Fig. 2) under the following initial conditions $a = 1,0 \text{n.u.}$; mass of the protected object is 70 kilo, its weight is $G = 700 \text{ N}$; the initial deformation of the load-bearing element 1 under the force of the object's weight is $\Delta_1 = 0,5 \text{n.u.}$, the initial deformation of each of the corrective elements 2 in the start position is $(y=0) \Delta_2 = 0,5 \text{n.u.}$, then the stiffness of the load-bearing elastic element is $C_1 = \frac{700N}{0.5\text{n.u.}} = 1400 N/\text{m.u.}$; the stiffness of a single corrective elastic element is $C_2 = 500 N/\text{n.u.}$, when $\text{n.u.}$ – nominal unit for excluding the influence of the scale factor. Equation of force statics in the starting position:

$$C_1\Delta_1 – mg = 0.$$  

(8)

Consider the separate action of forces on a mobile platform, setting its vertical displacement $\delta_y$ at intervals 0,1 n.u. The load-bearing elastic element will create a force $F_1 = C(\Delta_1 + \delta_y)$, when $\delta_y = 0$, $F_1 = 700 \text{ N}$. The graph of the function $F_1 = F_1(\delta_y)$ is represented by the upper part in Fig. 3. Weight of the object and the force from the corrective elements create a force $P = G + 4C_2(\Delta_2 - \Delta l_2) \sin \alpha$,

$$P = G + 4C_2(\Delta_2 - \Delta l_2) \sin \alpha,$$

(9)

when $\alpha = \arctg \frac{\delta_y}{\alpha/2}$, the elongation of the correcting element $\Delta l_2 = \sqrt{a^2 + \delta_y^2} - \frac{a}{2}$. The graph of the change in the force $P = P(\delta_y)$ is represented by a solid line in Fig. 3 at the bottom. The position of variables, but multidirectional forces $F_1 = F_1(\delta_y)$ and $P = P(\delta_y)$, acting along the axis of OY, if are mirror the graph $F_1 = F_1(\delta_y)$ relative to the axis of $\delta_y$. From the graphs (Fig. 3) it is clear that the values of these forces are close in modulus in a wide range of object movement from $\delta_y = 0$ to $\delta_y = 0,4 \text{n.u.}$ When $\delta_y = 0$ and $\delta_y = 0,3$, he modules of these forces are completely matched, which indicates that the stiffness of the support under consideration is equal to zero and, consequently, the natural oscillation frequency of the system with the object is equal to zero. Thus, a technical solution with a

Figure 2. Loading scheme of support «with leap»

Figure 3. Power characteristics of support
similar arrangement of linear elastic elements makes it possible to create effective vibration-isolating supports, including in the structures operator’s seats of transport machines.

Figure 4. Operator's seat mechanism

Also, consider to the scheme of vibration-isolating support in Fig. 4. The scheme uses an elastic element with a linear stiffness characteristic. In the starting position, the elastic element is embedded as a square diagonal, in the working configuration of the OABC lever scheme with a single kinematic size of the links as a rhombus diagonal. The diagonal of the rhombus is equal to a, i.e. OA = AB = BC = OS = a. This technical solution of the support is structurally simple and promising as a operator’s seat of transport machine.

The problem of determining the natural oscillation frequency of the support with the operator is solved. The weight Q is also assumed to be Q = 700 N, the stiffness of the elastic element must provide a given vertical displacement of the sitting h of the seat, which will then cause deformation $\Delta l$ of the elastic element. The coupling between h and $\Delta l$ is obtained from geometric relations

$$
\Delta l = \frac{h}{\cos\left(\frac{90 - \alpha}{2}\right)}.
$$

(10)

Weight Q of the operator is compensated by the force N of the elastic element, from the equilibrium condition there is:

$$
N = \frac{Q}{\cos\left(\frac{90 - \alpha}{2}\right)},
$$

(11)

and the calculated stiffness of the elastic element C is follows:

$$
C = \frac{N}{\Delta l} = \frac{Q}{h}.
$$

(12)

Since the masses of the mechanical system of the seat without an operator are insignificant, they are not used in the calculation of the oscillations frequency $k_0$, then taking into account $m=Q/g$, where g is acceleration of gravity, are get

$$
k_0 = \sqrt{\frac{C}{m}} = \sqrt{\frac{Qg}{hQ}} = \sqrt{\frac{g}{h}}.
$$

(12)

For example, let's take the vertical displacement of the sitting seat h = 0.2 m, then
\[ k_0 = \sqrt{\frac{9.8m/s^2}{0.2m}} = \sqrt{\frac{49}{s^2}} = \frac{7}{s} \] or \[ f = \frac{k_0}{2\pi} = \frac{7}{6.28} \approx 1Hz, \]

this indicates the effectiveness of the vibration isolation of the proposed structurally simple and easily implemented technical solution for the vibration isolation seat of the transport machine operator.

5. Conclusion
1. It is presented that a vibration protection system with minimal damping is perspective for vibration isolation of operators and machine assembly.
2. It is proved that an effective vibration isolation system is created by using a set of elastic elements with a linear stiffness characteristic. These elements are specially oriented and especially located in relation to the direction of force or kinematic excitation to endow the vibration isolator with nonlinear properties that are realized in a specific range of machine performance characteristics.
3. The natural oscillation frequency is the main criteria for evaluating the effectiveness of the vibration isolation system. In systems «with leap» in the direction of elastic forces, it is possible to create conditions of quasi-zero stiffness in a specific range of characteristics, which is ideal and necessary for vibration protection of transport vehicle operators, in particular.
4. On the bases of using conversion of motion and elastic elements with linear characteristics of rigidity, it is possible to synthesize a structurally simple and technological vibration isolation systems for personnel in a wide range of options of vibration effect.

6. References
[1] Alabuzhev P M Gritchin A A and Kim L I 1986 Vibration protection systems with quasi-zero stiffness (Leningrad: Mashinostroenie)
[2] Buryan Yu A and Silkov M V 2019 Vibration isolation support with the effect of quasi-zero stiffness (Omsk: OmGTU)
[3] Buryan Yu A and Silkov M V 2017 Design and evaluation of vibration isolation of the support for process equipment using quasi-zero stiffness (Omsk: OmGTU)
[4] Zotov A N and Hisamov 2018 Vibration isolator with a given power characteristic Alley of Science 6 438–6
[5] Zhou J Wang X and Mei V Characteristic analysis of a quasi-zero-stiffness vibration isolator 2018 Journal of Physics: Conference Series 8 397–4