Some issues of improving the accuracy of the system configuration of the vibration with inertial converter

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Abstract. In this work considered issues of elastic and dissipative characteristics for vibration protection systems with inertial converter. Represented the results of numerical simulation with taking account of the proposed. The condition for setting the frequency offset is determined and ways to improve the effectiveness of vibration protection are proposed.

1. Introduction

The ideology of increasing the durability of the equipment is to increase the endurance limit of the material of the workpiece. To do this we need to reduce the voltage concentration. Reducing the voltage concentration is achieved by providing effective reduction of friction and alternating voltage, by reducing alternating loads. In this way, the solution of the problem of increasing the efficiency of vibroprotection systems, along with the protection of the human operator, allows to solve the problem of increasing the durability of the equipment.[1-8].

Usually, the equipment operates in a specific frequency range, where it is affected by oscillations in a limited frequency range. Reducing vibrations in this range can improve equipment durability. The main problem is to provide the low frequency range and the required energy intensity. Also at startup transient processes take place shock processes, which can cause self-oscillation processes. All this leads to the fact that the real durability of the workpiece is much less than the durability of the material from which it is made, as the local voltage concentration for workpieces of complex construction increases. In this way, in some cases, it's require combined protection from vibrations and shocks. Studies of oscillations that occur during shock are carried out taking into account the fact that the shock effect is described by a half-sine, a square wave, a full sine wave, or is modeled by a delta function. Considering that the effectiveness of vibro protection during shock effects depends on the interval of static loads, it is necessary to reduce it. A significant factor in the selection of structural elements is the energy intensity of vibration-proofing devices. Pneumatic vibration protective systems have the best energy, but it is very inertial. Piezoactive systems, on the contrary, have high speed, but are prone to rapid destruction at high vibration amplitudes. To reduce the impact and resonance oscillations, a significant increase in the rigidity of the elastic characteristic is necessary, and the damping in the zone of static equilibrium should be minimal. Intermediate solution are hydraulic vibration protection systems. To solve problems of both shock and vibration damping, it is extremely important to solve the problem of determining
effective damping. Consider the question of choosing damping for a vibration protection system containing an element with increased inertia.

2. Math Modeling
Consider the processes occurring in the system of vibration protection using the example of a hydraulic support with a built-in inertial converter of the hydraulic type. Let the object of vibroprotection be modeled with a mass $m_1$ fixed to the base by means of an elastic element with rigidity $C_1$ with a motion converter embedded in it containing an elastic element with rigidity $C_{it}$, inertiality on relative motion $m_{it}$, with total damping $b_{it}$.

![Figure 1. Model of the vibration protection system](image)

We guess that the damping in the elastic element and the additional active regulator varies linearly and can be represented simply as $b_1$. The scheme of the described technical solution is presented in Figure 1. Let the object be influenced by the harmonic effect $F_0$. The response to this effect will be the reaction of the base $N$. These reactions can be described by equations 1 and 2.

\[
F_0 = (m_1 + m_{it})x + (b_1 + b_{it})x + (c_1 + c_{it})x
\]

\[
N = (m_{it})x + (b_1 + b_{it})x + (c_1 + c_{it})x
\]

The model of a dynamic system can be described by the system of equations (3) or (4)

\[
\begin{align*}
\dot{m}_1 x &= F_0 - c_1 x - b_1 x - p_1 A \\
\dot{p}_1 &= p_2 \\
\dot{m}_{it} x &= p_2 A - c_{it} x - b_{it} x \\
\dot{m}_{it} x &= -N - c_1 x - b_1 x - c_{it} x - b_{it} x \\
R(x) &= -N(x)
\end{align*}
\]

The condition for selecting spring rigidity.

The introduction of additional rigidity is closely related to the need to take into account the frequency settings of the system and the additional circuit. A situation is possible in which it will be necessary to abandon the use of a spring perceiving static rigidity or creating a system with negative static rigidity. Similar problems in mechanical systems did not exist since the mass of the object was always greater than the mass of the quencher. In systems with inertial motion converters, it is theoretically possible to create inertial effects capable of exceeding the inertia of the object itself.

The choice of rigidity is a solution to the system of equations...
The condition for the equivalence of damping under test will be the equality of damping in the passive vibration isolator and systems with inertial motion converters. This requires compliance with the conditions

\[
10 = \frac{1}{2 \cdot \pi} \cdot \frac{C_1 + C_n}{m_1 + m_{sp}}
\]

\[
C_n = 6.28^2 \cdot f_w \cdot m_p
\]

\[
C_b > C_1 > 10000
\]

\[
m_1 > 4 \cdot 10^3
\]

(5)

Structural damping in metal springs is determined similarly to the damping of a metal vibration isolator.

\[
b_1 \cdot j \cdot \omega = \left( b_2^m + b_2^s + b_2^l \right) j \cdot \omega
\]

(6)

Channel damping

\[
b_i^l \cdot j \cdot \omega = \frac{D^l \cdot l \cdot \rho \cdot \omega}{K_i}
\]

Ki - coefficient taking into account losses in the inertial channel in the process of energy conversion.

The condition for optimal damping of the vibration protection system is a solution to the dependency system

\[
\omega_p = \sqrt{\frac{C_1 + C_n}{m_1 + m_{sp}}}
\]

\[
\omega_p = \sqrt{\frac{2}{1 + \frac{C_1 + C_n}{\infty}}}
\]

\[
n = \sqrt{\frac{\left(1 + \frac{C_1 + C_n}{\infty}\right) \left(1 + \frac{2 \cdot (C_1 + C_n)}{\infty}\right)}{8 \cdot \left(\frac{C_1 + C_n}{\infty}\right)}} = \frac{1}{\sqrt{2}}
\]

\[
b = \frac{2}{\sqrt{2}} \cdot \left(m_1 + m_{sp}\right)
\]

If the base has elastic properties, then the infinity sign must be replaced with the equivalent complex rigidity of the base, determined experimentally by the method of test action.
3. The results of numerical simulation

The result of choosing the optimal damping with the constancy of the damping parameters of the geometry is presented in Figure 2. Figure 2 shows the three transfer function curves. T1 describes the mode of optimal damping. T2 describes the mode of operation of a classical vibration isolator with optimal damping. T3 describes the mode of operation of a realistic achievable damping with the use of metallic elastic elements without taking into account losses in the channel.

![Figure 2. Vibration protection system transmission characteristics with inertial converter with different damping.](image)

The introduction of clarifying dependencies in the calculation system allows you to increase the depth of system configuration. The effectiveness of the solutions is shown in Graph 3, where the construction of the T3 curve does not take into account the specific features of hydraulic local resistances, and the T4 curve uses additional criteria taking into account the hydrodynamic characteristics of the vibration protection system.

![Figure 3. Comparison of standard and refined models.](image)

Curve T4 allows to estimate the effect of inertial properties in the channel. The weak consideration of the inertial properties in the channel leads to large errors when choosing the optimal damping and leads to an increase in resonance phenomena. Interestingly, adding a damping converter without changing the damping in a metal spring that absorbs static load may cause the tuning frequency to shift. Picture 4 shows this variation of the transfer functions. In the simulation, the frequency offset is 5 Hz. (From 133 to 137 Hz). If we analyze the differential equation of motion and the equation defining the creation of relative inertia, it is theoretically possible to create a system in which the inertia in relative motion can exceed the inertia of the object itself. In practice, this contradicts the theory of optimal quenching, according to which the dynamic mass, working to stabilize an object, should be ten times less than the mass of an object.
An increase in the inertial properties of the system above the real inertia of the object leads to a decrease in the vibration-protective properties at high frequencies. The use of vibration protection systems with inertia above 0.2 leads to a limitation on the minimum vibration protection. It was established experimentally that the optimal creation of inertia in relative motion is 10% of the mass of the object.

An increase in inertia leads to a change in the tuning frequency and a deterioration in the damping efficiency at high frequencies, and a narrowing of the effective damping region, as compared to the passive vibration isolator modeled by the 2 T2 curve. With an increase in the ratio between the inertia of 0.31, there is a significant deterioration in the efficiency of quenching already at a frequency of 200 Hz. Therefore, the recommended limit on the increase in inertia should be considered the ratio

\[ m_{sp} = 0.1 \cdot m_i \]

The minimum possible reduction in rigidity when using systems with inertial motion converters while.

4. Results of a full-scale experiment
For the experiment was used the methodology described in [2]. For practical proof of the impossibility of an infinite increase in the length of the channel was used a model with a channel length of 150 mm (Figure 5). The results of the experiment presented in Figure 6 confirm the disintegration of fluid operation in the channel into two coherently oscillating masses, thus the impossibility of creating super-large inertial effects was proved, even with optimization of the damping properties.
Figure 6. The transfer function of a system with an extra large channel. Disintegration of fluid flow into two coupled vibrations.

1 - the tube is not fixed on intermediate supports
2 - the tube is fixed on elastic supports

Figure 7. System with a motion converter tuned to 125 Hz.

The result of the work of the same system with the optimization of inertia and damping (Figure 7) made it possible to obtain broadband quenching with high efficiency.

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