Increasing efficiency of vibration protection system by using pneumatic rubber cord devices

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Abstract. The article is devoted to investigation of the vibration protection system based on pneumatic rubber-cord devices. The device and principle of operation of the combined vibration protection system with the use of rubber-cord shells as power elements are presented. The analysis of the developed mathematical model is carried out. New methods of construction and algorithms for controlling the active vibration protection system are investigated on the created experimental complex. The results obtained at the experimental complex are presented and analyzed. The study of the developed mathematical model of combined vibration protection system with throttle control of gas pressure in power cells has shown its efficiency in the low-frequency range. The evaluation of the quality of the combined vibration protection system using integral criteria has shown its effectiveness up to 40% in comparison with the passive vibration protection system with harmonic excitation. The obtained test results have a discrepancy in the low-frequency region of the operating range with mathematical modeling data at the level of 10%.

Key words: active suspension, control strategy, vibration protection, rubber cord device

1. Introduction
Modern machinery production is aimed at implementing high-precision equipment and manufacturing processes. The process of achieving high precision of the component dimensions is, however, related to significant challenges due to negative vibration impact of natural and production-induced character [1].
The analysis of natural and production-induced vibration sources indicates their frequency range lying in the region of 0.1 Hz to 10 kHz [8–11]. Modern vibration protection systems, however, have frequency range bottom margin ≈2 Hz with maximal oscillation suppression ratio of 35–45 dB achievable at ≈10 Hz frequency [6, 12]. With due consideration of high-precision equipment cost and the importance of the research and production challenges solved with this equipment [3, 6, 12], it is unmistakable that vibration protection systems development is significant for such equipment.
At present time passive vibration protection systems are wide spread, but they have a number of drawbacks. Vibration protection systems with active elements connected to the external power source are in fact automatic control systems, and they nearly always work together with passive systems.
As passive vibration protection elements one uses pneumatic rubber-cord shells (RCS) having high lifting capacity and reliability [1, 2, 5]. Their usage as active systems actuating elements is, however, challenging, since they may create force only in one direction. Reverse movement at pressure relief is carried out either due to protected equipment gravity or additional spring element action. As the spring element one should use RCS incorporated into the system via lever similar to that of a tensile testing machine for compressing specimen.

2. Problem statement
The present work is concerned with the development of theoretical and practical approaches to produce combined vibration protection support based on RCS including both active and passive components (figure 1).

![Figure 1. Scheme of combined vibration protection system with RCS: 1 – table; 2 – vibration-proof platform; 3 – RCS of the passive vibration protection system; 4, 5 – upper and lower RCS of active vibration protection system; 6 – lever; 7 – speed sensor; 8 – control unit; 9 – compressor.](image)

Vibration protection support bearing element is passive system RCS 3, whose pressure is not controlled during operation. Pressure in active system RCS 4 and 5 is altered by control unit 8 in antiphase to platform 2 oscillations. When it is moving downwards, the pressure is applied to RCS 4, and when it is moving upwards then to RCS 5 hindering this displacement by lever 6. Control unit activates according to the signals of speed sensor 7 installed on platform 2.

3. Theory
Calculation scheme of the combined vibration protection system is given in Figure 2.

![Figure 2. Calculation scheme: m – protected platform mass; J – the moment of inertia of the protected platform relative to the center of mass; l₁, l₂ – distance from the center of mass to the suspension points; Δl – distance from the center of mass to the corresponding axis of the protected platform](image)

Platform oscillations are examined in two generalized coordinates \( z \) and \( \varphi \), as well as subsidiary coordinates \( q \) (foundation displacements). The system has two degrees of freedom, a vibration-proof platform with mass \( m \) and inertia moment \( J \).

Also, it is assumed that: \( c_{11} = c_{12} = c_{p1}, c_{21} = c_{22} = c_{p2}, q_1 = q_2 = q, l_1 = l_2. \)

Allowing for the assumptions and the fact that protected platform gravity is balanced by the passive system RCS string forces, and the values of generalized coordinates are measured from the
equilibrium states, and neglecting the friction forces in suspension as well as RCS damping properties, the differential equations system takes the form of:

\[
\begin{align*}
\ddot{z} + 2c_1 (z - q + l_1 \phi) + 2c_2 (z - q + l_2 \phi) \\
= 2S_{sp} P_1 + 2S_{sp} P_2 - 2c_{p1} (z - q + l_1 \phi) - 2c_{p2} (z - q + l_2 \phi); \\
J \ddot{\phi} + 2c_1 l_1 (z - q + l_1 \phi) + 2c_2 l_2 (z - q + l_2 \phi) \\
= -2l_1 S_{sp} P_1 + 2l_2 S_{sp} P_2 - 2c_{p1} l_1 (z - q + l_1 \phi) - 2c_{p2} l_2 (z - q + l_2 \phi).
\end{align*}
\]

where \(z, \phi\) are linear and angular displacements of protected mass; \(S_{sp}\) is RCS effective area; \(P_{l,2}(\phi, t)\) is pressure in RCS.

When analyzing the system configuration and considering that its protected part mass has significant value and exceeds the supported equipment mass, the center of mass is practically coincides with the elastic centre. This configuration feature lets us assume that vertical protected mass oscillations are independent of longitudinal-angular ones because of the suspension symmetry, and the differential equations system takes the form of:

\[
\begin{align*}
\ddot{z} + 2c_1 (z - q) + 2c_2 (z - q) &= 2S_{sp} P_1 + 2S_{sp} P_2 - 2c_{p1} (z - q) - 2c_{p2} (z - q); \\
J \ddot{\phi} + 2(c_1 l_1^2 + c_2 l_2^2) \phi &= -2l_1 S_{sp} P_1 + 2l_2 S_{sp} P_2 - 2(c_{p1} l_1^2 + c_{p2} l_2^2) \phi.
\end{align*}
\]

Structural diagram of gas pressure regulating system in active system RCS is given in figure 3.

When constructing mathematical model of distributing pneumatic devices (DPD), the equation for gas consumption via cylindrical throttle is of importance.

\[P_{1,2}(\phi, t) = \text{pressure in RCS}.\]

Figure 3. Structural diagram of the system for regulating the gas pressure in the RCS of the active system: 1 – speed sensor; 2 – sum block; 3 – amplifier; 4 – electromechanical converter; 5 – control cascade slide valve; 6 – gas supply line; 7 – fluid supply line; 8 – main slide valve; 9 – rubber-cord shells; 10 – control object

Passive vibration protection system is constructed on the basis of I-09 type RCS. The pressure in these RCS is set according to the load on the protected equipment plate and does not change in operation. The filling time of I-09 type RCS in active vibration protection system in isothermal process can be determined by the expression:

\[t_u = \frac{(P_{\text{max}} - P_{\text{min}}) \cdot V_0}{10 \alpha \cdot f \cdot P_s \cdot \sqrt{T \cdot N}},\]

where \(\alpha = \sqrt{gRTN} \approx 20\sqrt{T}\) m/s is elastic wave propagation speed in medium; \(f\) is internal cross section area of the main distributor slide valve; \(T\) is absolute temperature; \(N\) is heat capacity ratio; \(V_0\) is RCS volume in middle position; \(R\) is gas constant.

RCS drainage time under the same conditions is determined by the expression:

\[t_0 = -\ln \frac{P_{\text{min}}}{P_{\text{max}}} \cdot \frac{V_0}{0.58 \cdot \alpha^2 \cdot f}.
\]
Allowing for \( V_0 = 1 \cdot 10^{-3} \text{ m}^3 \); \( T = 298 \text{ K} \); \( N = 1.4 \); \( f = 3.14 \cdot 10^4 \text{ m}^2 \), RCS filling and drainage process control at small deviations is described by the following equations:

\[
\Delta Q_2 = K_{op} \Delta x_2 - K_{op} \Delta p_u; \\
\Delta p_u S_{op} \cdot c \cdot \Delta z + c \cdot \Delta z_m = 0; \\
m \frac{d^2(\Delta z_m)}{dt^2} + k_p \frac{d(\Delta z_m)}{dt} + (c + c_u) \Delta z_m = c \cdot \Delta z; \\
\Delta Q_2 = S_{op} \frac{d(\Delta z)}{dt} + \frac{V_0}{2E'} \frac{d(\Delta p_u)}{dt}; \\
\Delta x_2 = K_{ib} \Delta h - K_{w} \Delta z.
\]

where \( z \) is RCS displacement; \( z_m \) is protected plate displacement; \( \Delta p_u \) is pressure drop in RCS; \( x_2 = x_0 + \Delta x_2 \); \( z_m = z_{m0} + \Delta z_m \); \( \Delta Q_2 \) is main cascade slide valve consumption alteration; \( x_2 \) is main cascade slide valve displacement; \( K_{op} \) is transmission gains; \( m \) is protected plate mass; \( K_p \) is friction coefficient; \( c_i \) is contact stiffness; \( K_{fb} \) is feedback gain.

Assuming that forces created in active system RCS and acting on the protected platform are controlled by a two-stage electrohydraulic pneumatic distributor with main slide valve position feedback system, the schematic diagram of the electrohydraulic system can be represented as in figure 4.

![Figure 4. Schematic diagram of the pneumatic drive unit.](image-url)

**Figure 4.** Schematic diagram of the pneumatic drive unit: 1 – upper RCS of active vibration protection system; 2 – lower RCS of active vibration protection system; 3 – RCS of passive vibration protection system; 4 – the distributor of the main (second) cascade; 5 – the distributor of the control (first) cascade; 6 – electromechanical converter (EMC) of the control valve cascade; 7 – feedback sensor of the slide valve position of the main cascade; \( x_1, x_2 \) – slide valve movement of the control and main cascade; \( P_u, P_a \) – pressure in the power line and atmospheric pressure.

Перемещения управляющего золотника (золотника первого каскада) \( x_1 \) определяются уравнениями электромагнитного привода (ЭМП).

\[
\tau \frac{m}{c} \frac{d^3x_1}{dt^3} + \left( \frac{m}{c} + \frac{h}{c} \right) \frac{d^2x_1}{dt^2} + \left( \frac{h}{c} \right) \frac{dx_1}{dt} + x_1 = K_U U_1; \\
U_1 = K_{amp} \left[ U_{in}(t) - K_{fb} x_2 \right].
\]

Where \( K_1 \) is EMD transmission gain; \( \tau \) is EMD time constant; \( m \) is transducer moving members mass; \( h \) is viscous resistance coefficient; \( c \) is EMD anchor suspension stiffness; \( U_1 \) is voltage applied to EMD; \( K_{amp} \) is amplifier gain; \( K_{fb} \) is feedback gain.

The equation for the main slide valve motion takes the form of: \( x_2 = K_{ny} x_1 \), where \( K_{ny} \) is gain ratio by slide valve displacement.

In adiabatic process, at heat capacity ratio \( N = 1.4 \), \( \varepsilon_{cp} = 0.528 \). Setting \( P_u = 0.17 \text{ MPa} \), \( P_a = 0.1 \text{ MPa} \), one may assume that maximum working pressure in RCS (\( P_{max} \)) should not exceed 0.15 MPa, while minimal one (\( P_{min} \)) should not exceed 0.1 MPa.

Passive system RCS stiffness is determined by the expression:
\[ c_0 = \frac{TP_0 S_d^2}{V} + \frac{\partial S_d}{\partial z} P_0. \]

Assuming in the first approximation that value \( \frac{\partial S_d}{\partial z} \) is small and RCS volume \( V \) is a linear function of displacement, RCS stiffness ratio can be estimated using the following expression:

\[ c_0 = \frac{NP_0 S^2}{h_0}, \]

where \( h_0 \) is RCS height in middle position; \( c_0 \) is passive system RCS stiffness ratio; \( P_0 \) is pressure in passive system RCS.

Therefore, in terms of the previously given assumptions, the above-mentioned equations determine longitudinal movement dynamics of a pneumatic-mechanical system with passive and active vibration protection system.

To analyze protected platform oscillations (acting on one support), let us take \( m_1 = 80 \) kg. I-09 type RCS was also chosen as active system spring elements.

Mathematical model was studied in two stages, and the investigation was made of:

- vertical oscillations of the platform with active system speed control;
- vertical oscillations of the platform with active system displacement control.

In simulation it was assumed that, with ideal measuring element for protected platform vertical displacements speed, gas pressure application to active system RCS is performed by a two-stage slide valve, it being immediately connected with the platform in displacement control.

Structural diagram of vibration protection system with linear servo system for the first stage (in MATLAB environment with Simulink) is given in figure 5.

![Figure 5. Structural diagram of vibration control system with speed control](image)

Step-function response type at out-of-operation active system of protected platform vibration protection system (line 2) and at putting the active system in operation (line 1) is shown in figure 6.
The simulation results of protected platform vertical oscillations depending on the disturbing oscillation frequency are presented in figure 7.

The obtained graphs allow for the following conclusions to be made. First, realization of the combined vibration protection system with displacements speed control decreases the amplitude of the protected platform forced oscillations. Second, and most importantly, this occurs at low frequencies, where standard and modified passive damping systems are inefficient. Realization of the combined vibration protection system with displacements value control has almost no effect on the forced vertical oscillations amplitude of the protected platform.

4. Experimental results
The experimental results were obtained with the use of the physical model, that is experimental complex for investigating combined vibration protection systems, allowing for the proposed design of the combined vibration protection system to be realized and various control algorithms to be studied. Field tests were aimed at evaluating operation efficiency of the combined vibration protection system. When experiments were conducted, the variable was the actuating signal frequency of the vibrations initiator. Moreover, linear displacements and table and platform displacements speed were measured.
5. Discussion of the results
The graphs of platform displacements gain frequency characteristic, obtained on the workbench and in the mathematical model, were compared, and the results are given in figure 8. The received results are comparable. The analysis of the oscillations amplitude changes by frequency demonstrates that the graphs type is similar except the region in the range of 3...4 Hz, where in full-scale study one can notice the resonance caused by RCS frequencies overlapping.

Then the operation of combined vibration protection system with working active and passive systems was analyzed.
The diagrams in figure 9 indicate the efficiency of platform oscillations suppression by combined vibration protection system in the whole given frequency range. However, the maximum is observed in the region of 2.15...2.3 Hz just as in case of tests only with the passive system. In addition resonance phenomena are not explicit.

![Figure 8. Comparison of gain frequency characteristic of the movements of the upper beam only with the passive system obtained on the experimental complex (curve 1) and gain frequency characteristic of the movements obtained on the mathematical model (curve 2); curve 3 and 4 are trend lines](image)

![Figure 9. Gain frequency characteristic of movements of the upper beam with a combined vibration protection system (curve 1) and gain frequency characteristic of movements of the lower beam (curve 2)](image)

To evaluate the adequacy of the mathematical model, the results of full-scale experiment and mathematical simulation were used. As the diagrams (figure 10) show, dynamic characteristics of combined vibration protection system and mathematical model correlate satisfactorily.
Figure 10. Gain frequency characteristic of oscillations of the upper beam with a combined vibration protection system obtained at the experimental complex (curve 1) and the results of mathematical modeling (curve 2)

By using linear integral estimation it was specified that results have a discrepancy of 7.4%. This proves that the experiment and mathematical simulation results are adequate.

6. Conclusions
The study of the developed mathematical model demonstrated that the combined damping system is efficient in the frequency range of 0.5...2.5 Hz following the effect of the pneumatic system operation speed. Consequently, active system must be switched off in the range higher than 2.5 Hz.

The examinations of the combined vibration protection system conducted on the experimental complex demonstrated that equipment vibration load level decreases by 40%. Amplitudes of the oscillations speed caused by single impact are reduced by up to 21%. At periodical disturbance the combined vibration protection system efficiency reaches 47% compared to the passive system. The operation efficiency of the combined vibration protection system with pneumatic drive unit control by the plate displacement is no more than 14% in comparison with the passive system.

The obtained results provide us with new directions for investigating new control algorithms of active vibration protection systems and optimizing the existing ones.

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