

COMBINED CONTROL OF HYDROPNEUMATIC SPRINGS OF THE VEHICLE

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Abstract

The article is devoted to research of a vehicle ride comfort, equipped the suspension system with combined control of springing and damping elements. The spatial mathematical model of a vehicle which meets the requirements of adequacy to real object was developed. The results of the computer simulation of movement of the vehicle with controllable and passive suspension systems on stochastic microprofile are presented.

It can be concluded that the spatial multi-mass model is satisfactory close to a real object and can be used for further calculations.

Having analyzed simulation results we can state significant increase of ride comfort of a vehicle equipped with a controlled suspension system. Decrease of vertical acceleration of a cabin in a place where a driver sits amounted to 40% in average that can be considered as a limit value for this type of controlled suspension systems.

It can be noted that a suspension system with combined control of springing and damping elements can be considered as rather effective in terms of decreasing load of vehicle spring-mounted parts that along with low power consumption makes it rather competitive. The shown research results and control laws for springing and damping elements can be used as a basis for development of relatively simple controlled suspension systems.

Keywords: *controlled suspension system, ride comfort, hydropneumatic spring, road tests, spatial mathematical model, vehicle*

1. Introduction

Passive suspension systems with springing and damping characteristics remaining invariable in the course of operation except for the changes related to wear and tear, warming up and breakdown are commonly used in existing transportation vehicles. Mainly it is due to comparatively simple and reliable design, as well as to lack of necessity in external power source. However performance potentials of such systems in terms of meeting the demanding requirements for ride comfort are rather restricted and have more or less reached their limit. Hence it is necessary to find new ways of increasing ride comfort of transportation vehicles; one of them is to use various kinds of controllable suspension systems [7] which allow changing springing and damping characteristics as per a specified algorithm in the course of moving. Although these systems make the total price for a transportation vehicle higher, as well as make the design more complicated but allow improving ride comfort.

Among the known kinds of controllable suspension systems we can distinguish the semiactive systems [8] which are considered to be the most optimal in terms of increasing ride comfort (advantage over passive suspension systems), power consumption and design complexity (advantage over active suspension systems). The semiactive suspension system is such a system where external energy is consumed only to change parameters of passive springing and damping elements it includes, namely to change their springing and damping characteristics.

The article contains the research of the operational efficiency of the suspension system with the combined control of springing and damping elements (hydropneumatic springs) when a vehicle moves on stochastic microprofile.

2. Algorithm of the combined control

Analytically, the used algorithm of control of damping element of hydropneumatic spring can be represented through the following equation system:

$$r_{si} = \begin{cases} r_1, & (\dot{z}_i \cdot (\dot{z}_i - \dot{\zeta}_i) > 0) \ \& \ (-\dot{\zeta}_i \cdot (\dot{z}_i - \dot{\zeta}_i) > 0) , \\ r_2, & (\dot{z}_i \cdot (\dot{z}_i - \dot{\zeta}_i) > 0) \ \& \ (-\dot{\zeta}_i \cdot (\dot{z}_i - \dot{\zeta}_i) \leq 0) , \\ r_3, & (\dot{z}_i \cdot (\dot{z}_i - \dot{\zeta}_i) \leq 0) \ \& \ (-\dot{\zeta}_i \cdot (\dot{z}_i - \dot{\zeta}_i) > 0) . \end{cases} \quad (1)$$

where:

$\dot{z}_i, \dot{\zeta}_i$ - vertical velocity of sprung and unsprung mass,

r_{si} - current resistance factor of hydropneumatic spring,

r_1, r_2, r_3 - factors of variable resistance of hydropneumatic spring ($r_1 > r_2 > r_3$),

i - hydropneumatic spring number.

As control law for springing element we suggest the algorithm the essence of which is to change springing and damping characteristics of hydropneumatic spring by interconnecting two volumes of gas cavity depending on current values of vertical speeds of sprung and unsprung mass. The efficiency was also investigated on a bus model with controlled springing elements of low pressure in the form of rubber-cord pneumatic springs [3-6].

With the help of the equation of springing characteristics of passive hydropneumatic element [2] we can get springing characteristics of hydropneumatic element with the variable springing and damping properties which will be calculated as per the formula below:

$$F_{ui} = p_{iz} \cdot \left(\frac{V_o}{V_o + F_p \cdot \lambda_{ji}} \right)^k \cdot \left(\frac{V_o + V_d + F_p \cdot \lambda_{ji}}{V_o + V_d + F_p \cdot (z_i - \zeta_i)} \right)^k \cdot F_p, \quad (2)$$

where:

- F_{ui} - springing force of hydropneumatic spring,
- p_{iz} - initial excessive pressure in hydropneumatic spring,
- k - polytrophic coefficient,
- F_p - hydropneumatic spring piston area,
- V_o, V_d - primary and additional volumes of gas cavity of hydropneumatic spring,
- λ_{ji} - hydropneumatic spring deformation coinciding with the moment of additional volume supply/cutting off.

Analytically the algorithm of control of a pneumatic springing element due to change of its operating volume can be represented through the equation system:

$$V_i = V_o + F_p \cdot (z_i - \zeta_i) + \begin{cases} V_d, & \dot{z}_i \cdot (\dot{z}_i - \dot{\zeta}_i) \leq 0, \\ V_d, & (\dot{z}_i \cdot (\dot{z}_i - \dot{\zeta}_i) > 0) \ \& \ (|z_i - \zeta_i| > |\lambda_{ji}|), \\ 0, & (\dot{z}_i \cdot (\dot{z}_i - \dot{\zeta}_i) > 0) \ \& \ (|z_i - \zeta_i| \leq |\lambda_{ji}|). \end{cases} \quad (3)$$

where:

V_i - current volume of gas cavity of hydropneumatic spring.

3. Spatial mathematical model of the vehicle

The research of operational efficiency of the suspension system with combined control of springing and damping elements was performed using proven software (FRUND) [1]. This software system was used when constructing ZIL car, KAMAZ sport car, VAZ 2123 motor car etc; it is also used when designing and developing different mechanical systems in the subdivisions of some Russian mechanical engineering plants. This software system has allowed generating a spatial multi-mass model of a vehicle, general design diagram of which is shown on Fig. 1.

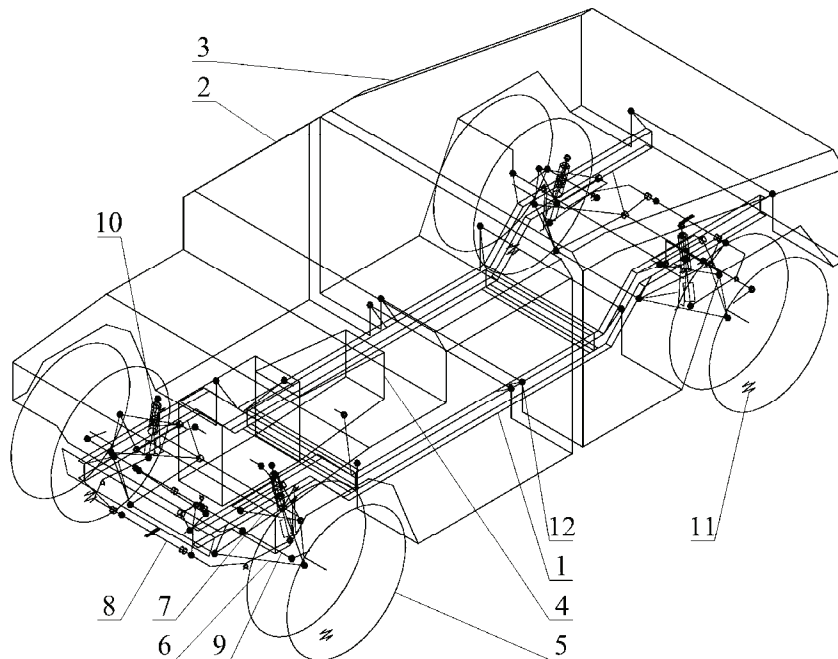


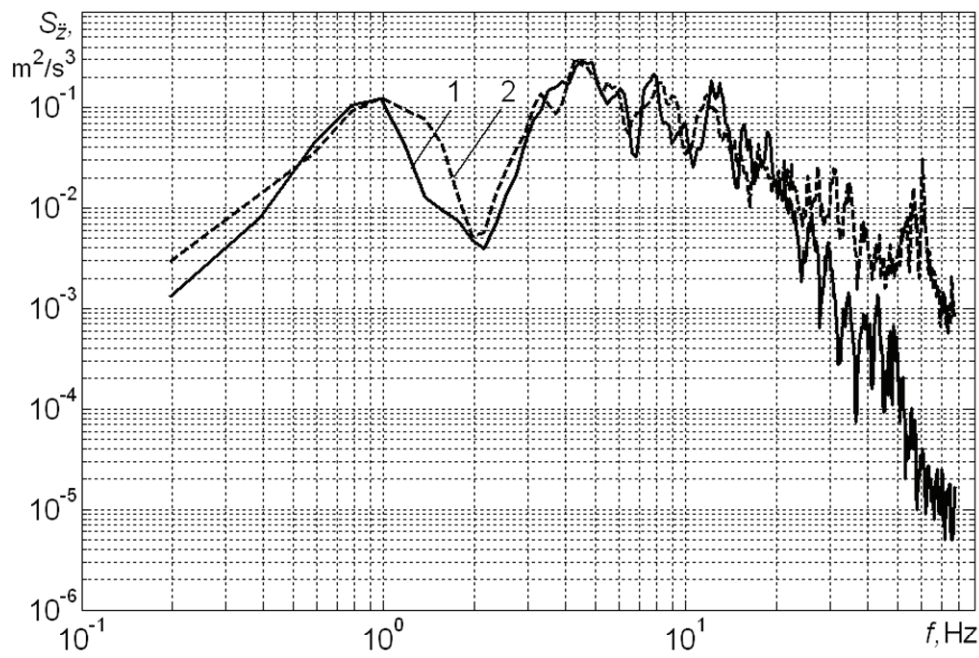
Fig. 1. General design diagram of a vehicle

The vehicle is a 4x4 two-axle off-road conventional vehicle with independent front and rear suspensions. The model consists of rigid bodies, namely frame 1, cabin 2, van 3, power plant 4, wheels 5, lateral arms 6 and 7, stabilizer bar 8, rods 9. The rigid bodies are connected to each other and to the road through the springing and damping elements: hydropneumatic springs 10, tires 11, as well as through different kinematic pairs 12. The model of the suspension system faithfully reproduces front and rear suspensions geometry of a real object with variances and connections of all degrees.

In order to verify the developed model for validity we compared the design data with the experimental data acquired in the course of the road tests of vehicles with a passive (non-controlled) suspension system. The experimental data in the form of spectral density and RMS of vertical acceleration of a cabin in a place where a driver sits was acquired by measuring when moving directly along a special cobblestone road of the satisfactory quality that correlates with roads of category 2 (OST 37.001.275-84 – «Motor vehicles. Testing procedures for ride comfort».) with steady speed 55 km/h.

The calculation method was built up similarly to the experimental measurements method at road tests. Cobblestone road of satisfactory quality was selected as drive function.

Figure 2 shows the diagrams of design 1 and experimental 2 spectral densities of vertical acceleration of a cabin in a place where a driver sits when a vehicle with a passive suspension system moves directly on a cobblestone road of satisfactory quality with steady speed of 55 km/h.



1 – design (RMS=1.27 m/s²), 2 –experimental (RMS=1.37 m/s²)

Fig. 2. Design and experimental spectral densities of vertical acceleration of a cabin in a place where a driver sits

Table 1 shows design and experimental RMS of vertical acceleration of a cabin in a place where a driver sits in 1/3 of octave frequency band.

Comparison of the vertical acceleration spectral densities curves, as well as RMS of vertical acceleration in 1/3 of octave frequency band derived through design and experimental ways shows acceptable convergence in the frequency range of 0.7-22.4 Hz for the vehicle movement mode under review. Average deviation of the design acceleration from the experimental one does not exceed 15%, which is allowable for random nature of the process. Based on that, it can be concluded that the spatial multi-mass model is satisfactory close to a real object and can be used for further calculations.

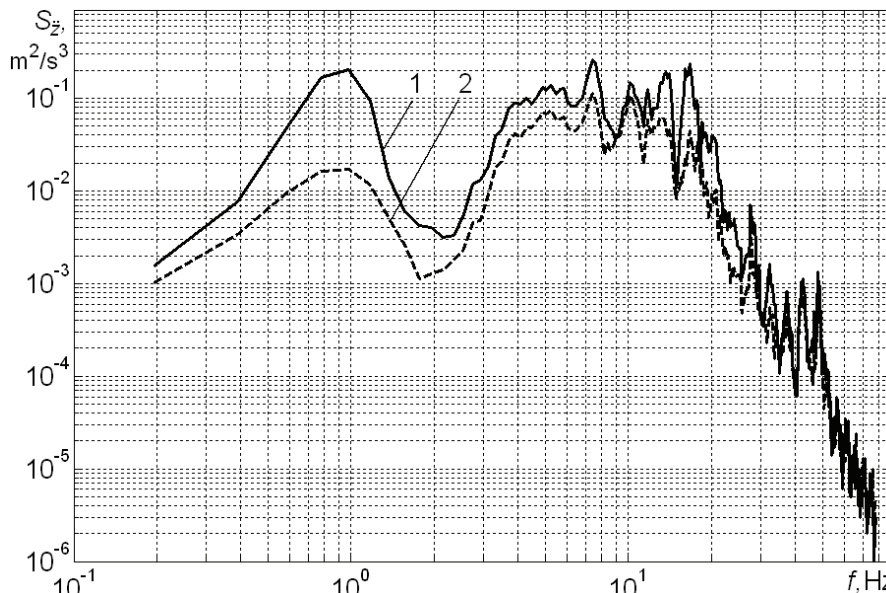
Tab. 1. Design and experimental RMS of vertical acceleration of a cabin in a place where a driver sits, m/s^2 (1 – design, 2 – experiment)

f_r , Hz	0.80	1.00	1.25	1.60	2.00	2.50	3.15	4.00	5.00	6.30	8.00	10.00	12.50	16.00	20.00
1	0.144	0.154	0.091	0.043	0.041	0.064	0.203	0.384	0.434	0.380	0.459	0.316	0.556	0.350	0.318
2	0.133	0.157	0.136	0.091	0.046	0.086	0.239	0.371	0.432	0.397	0.463	0.377	0.478	0.316	0.300
δ , %	8.3	1.9	33.1	52.7	10.9	25.6	15.1	3.5	0.5	4.3	0.9	16.2	16.3	10.8	6.0

f_r – geometric mean frequency, $\delta = (\text{experiment} - \text{design} / \text{experiment}) \cdot 100\%$.

4. Simulation of movement of the vehicle with controllable suspension system

Then we simulated movement of a vehicle equipped with a suspension system with combined control of hydropneumatic spring under similar movement conditions. The simulation results are shown on Fig. 3 and summarized in Tab. 2.



1 – passive suspension system ($RMS=1.27 m/s^2$), 2 – controlled suspension system ($RMS=0.81 m/s^2$)

Fig. 3. Design spectral densities of vertical acceleration of a cabin in a place where a driver sits

Tab. 2. Design RMS of vertical acceleration of a cabin in a place where a driver sits, m/s^2 (1 – passive suspension system, 2 – controlled suspension system)

f_r , Hz	0.80	1.00	1.25	1.60	2.00	2.50	3.15	4.00	5.00	6.30	8.00	10.00	12.50	16.00	20.00
1	0.144	0.154	0.091	0.043	0.041	0.064	0.203	0.384	0.434	0.380	0.459	0.316	0.556	0.350	0.318
2	0.050	0.046	0.038	0.022	0.026	0.043	0.125	0.243	0.326	0.307	0.258	0.236	0.253	0.245	0.203
δ , %	65.3	70.1	58.2	48.8	36.6	32.8	38.4	38.0	24.9	19.2	43.8	25.3	54.5	30.0	36.2

$\delta = (\text{passive} - \text{controlled} / \text{passive}) \cdot 100\%$.

Having analyzed simulation results we can state significant increase of ride comfort of a vehicle equipped with a controlled suspension system. Decrease of vertical acceleration of a cabin in a place where a driver sits amounted to 40% in average that can be considered as a limit value for this type of controlled suspension systems.

5. Conclusions

To summarize, it can be noted that a suspension system with combined control of springing and damping elements can be considered as rather effective in terms of decreasing load of vehicle spring-mounted parts that along with low power consumption makes it rather competitive. The shown research results and control laws for springing and damping elements can be used as a basis for development of relatively simple controlled suspension systems.

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