The New Method of the sprung mass oscillations Damping in the suspension for Vehicle.

(The Substitution for the Shock Absorber? - May be ...)

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Summary

The article presents the setting and solving of the task of optimum quick-acting demping the Vehicle sprung mass oscillations without an ordinary hydraulic Shock Absorber.

Thus the possibility to decrease the oscillations amplitude at the frequency range from *0* to the infinity becomes by reality.

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1.1. INTRODUCTION

In spite of large number and diversity of suspension constructions all of them come to the same simplest scheme. It is shown in Fig. 1.

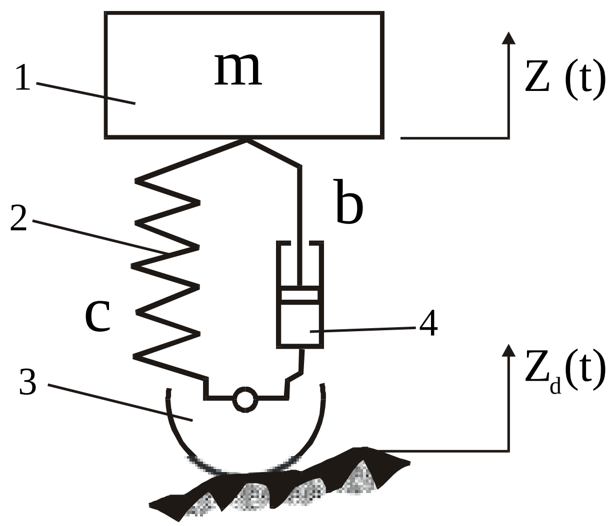

Fig.1 Simplest single-supporting scheme of transportation means for research suspension:
1 - sprung body or frame of vehicle;
2 - spring of suspension;
3 - wheel of a vehicle;
4 - shock absorber of suspension.
Z (t) - vertical coordinate of sprung body;
Zd(t) - the external disturbance (microprofile of the road).

The sprung mass 1 equal m designates a sprung body or frame of vehicle. Member 2 denotes the springy element of suspension. It is connected with sprung body 1 and unsprung mass, that is to say, wheel 3. The spring 2 is characterized by specific rigidity or stiffness factor equal c. The damping element or shock absorber 4 is installed parallel to the spring 2. It is characterized by the viscous resistance coefficient equal b. In Fig. 1 the guide device of suspension is conditionally not shown.

The technical perfection of vehicle suspension can be judged by an appearance of amplitude-frequency characteristics (AFC) of the sprung body oscillations. The amplitude-frequency characteristics (AFC) of the sprung body (Fig. 1) vertical accelerations in dimensionless form are shown in Fig. 2, where the next designations are used:

-A- relative Amplitude;

-ϖ - relative frequency, of oscillations.

In Fig. 2 line 1 corresponds to the springy suspensions with the relative damping factor approximately equal 0,1. Such suspensions are conditionally considered low damped. Line 2 also conforms to the elastic suspensions with the relative damping factor approximately equal 0,3. Such suspensions are conditionally considered powerfully damped.

These two lines demonstrate the main contradiction or main defect of the modem hydraulic shock absorber:

- it decreases the sprung body oscillation amplitude A at the relative frequency ϖ range from 0 to $\sqrt{2}$, that's the most important positive quality of the shock absorber;
- it increases the oscillation amplitude A at the relative frequency ϖ range from $\sqrt{2}$ to an infinity, that's the very unfavorable effect of the shock absorber to the ride smoothness.

Under existing conditions the above-mentioned hydraulic shock absorber contradiction is unsolved.

In our opinion the aforesaid contradiction of the hydraulic shock absorber may be solved only without resistance forces depending on the rate coordinates in time.

<u>PROPOUNDING OF THE PROBLEM:</u>

To find the control algorithm by the stiffness factor of the suspension spring with purpose optimal quick-acting damping of the vehicle sprung body without ordinary Shock Absorber.

1.2. THE MATHEMATICAL PROPOUNDING OF ABOVE-MENTIONED PROBLEM

Consider that the motion of the body (Fig.l) to be vibro-isolated is described on the phase plane by the following set of the differential equations in dimensionless form:

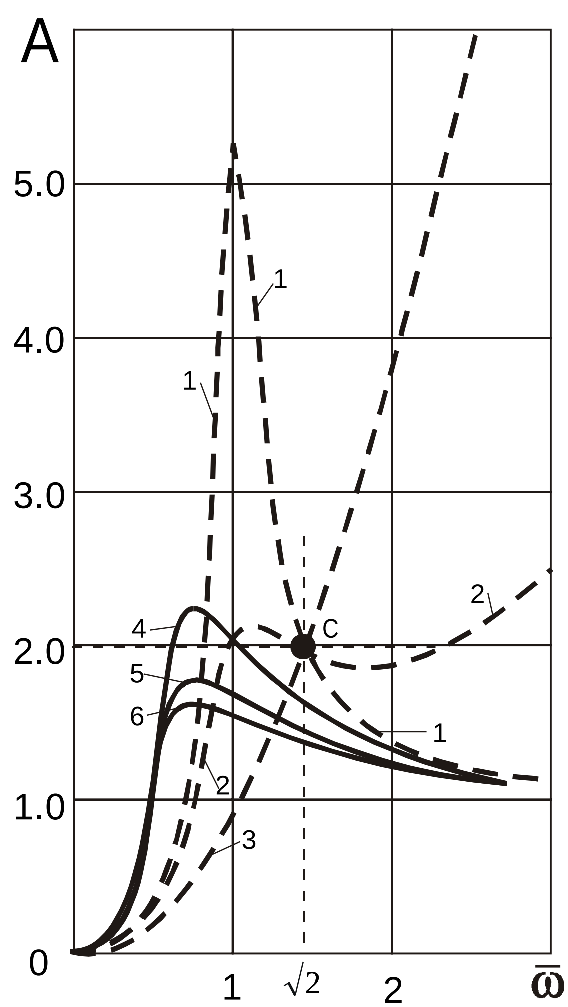

Fig.2.

Amplitude-frequency characteristics of sprung body acceleration for some simplest suspension schemes in dimensionless form:

1 - ordinary passive suspension with minimum stiffness factor and relative damping coefficient equal 0,1;
2 - ordinary passive suspension with minimum stiffness factor and relative damping coefficient equal 0,3;
3 - *passive suspension with absolute rigid springy element and infinitely great damping coefficient:;*
4,5,6 – see below.

$$\begin{cases} \dot{Z}_1 = Z_2 \\ \dot{Z}_2 = -UZ_1 \end{cases}$$

where:
Z_1= Z - coordinate of the body;

$$Z_2 = \frac{dZ_1}{dt}$$

The control U is determined by the following expression:

$$\begin{cases} c_1 \le U \le c_2 \\ 0 \le c_1 \le 1 \le c_2 \end{cases}$$

It is necessary to find the control U at area (2) as such in order to the object is described by the set of the differential equations (1) from any point of the phase plane long to go to the coordinates origin.

The solution of this problem with the help of the Pontriaguin's Maximum principle was published in [1,2]. The unknown control U in (1) is determinted by the following algebraic expression:

$$U = \begin{cases} c_2, & when \quad Z_1 \cdot Z_2 \ge 0 \\ c_1, & when \quad Z_1 \cdot Z_2 < 0 \end{cases}$$

Thus the optimum trajectory of the image point motion from any point of the phase plane long to go to the coordinates origins shown in Fig.3.

The stiffness factor of suspension spring is equal conditionally maximum value **C2** (3) in the first and in the third quadrants of the phase plane.

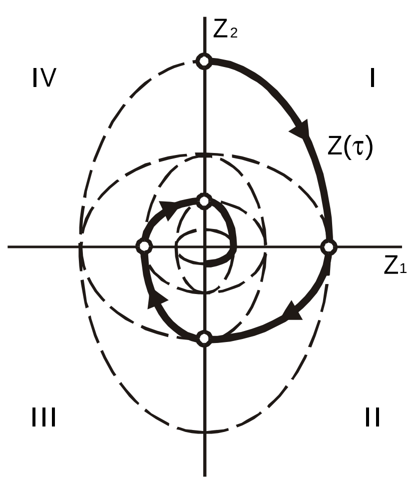

Fig. 3.
The optimal quick-acting Trajectory of the image point motion to the Coordinates Origin.
1,2,3... - the points where the stiffness factor of suspension spring is step changed on the law (3).

And the stuffiness factor of suspension spring is equal minimum C_1 (3) in the second and in the fourth quadrants of the phase plane. Thus, the control U supposes the step change of the stiffness factor of the suspension spring.

1.3. THE TECHNICAL REALIZATION OF OPTIMAL CONTROL WITHOUT RESISTANCE FORCES DEPENDING ON THE RATE COORDINATE IN TIME.

Consider that the stiffness factor of ordinary suspension spring is equal

conditionally the maximum value **C2,** that is to say, the object to be vibro-isolated moves without some controlling influence in the first and the third quadrants of the phase plane, where it deviates from equilibrium position. Here only elastic and inertional forces act to the sprung body.

The return of the above-mentioned object to the equilibrium position takes place in the second and the fourth quadrants of the phase plane.

It should be born in mind that the return of the object to the equilibrium position starts after on the deformation maximum reaching of the spring. At the given moment it is possible to release the strain energy very slow and this process may be to go on plenty of time.

In such away the stiffness factor of the suspension spring is step changed to

the minimum value C_1 (3) by the special device, which has the outward appearance and overall dimensions analogous to the ordinary telescopic Shock Absorber. What is more almost the large and vital parts, structure members, packing parts, fastenings of this devices are the same like in the ordinary shock absorber. Besides the aforesaid device is mounted like the telescopic Shock Absorber.

This special device with its control system creates the force only during the sprung mass return to the equilibrium position. This device doesn't forms the hydraulic viscous resistance, but it creates the new force, which partly compensates the elastic restoring force. The dependence of the new force on the elastic restoring force prevents the wheel separation from the roadbed.

The energy supply from an additional source is not needful, not counting the certain energy consumption by the members of the automatic control system.

The theoretical fundamentals of the new damping method are described in [1,2] and the constructive suspension schemes are expounded in monograph [1] and in the Invention descriptions [5,8].

The dependences of the sprung body vertical accelerations Amplitude on the frequency in dimensionless form are shown in Fig.2 by curves 4,5,6, which are distinguished by the different amplification factor of me automatic control system.

Thus we offer the new method of the sprung mass oscillation Damping. This damping method solves the main contradiction of the our times

hydraulic shock absorber.

The possibility to decrease the oscillation amplitude at the frequency range

from 0 to the infinity becomes by reality.

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