THE VEHICLE RIDE COMFORT INCREASE AT THE EXPENSE OF SEMIACTIVE SUSPENSION SYSTEM

Alex Podzorov

Volgograd State Technical University, Faculty of Auto-tractor
Zholudeva Street 38-91, 400006 Volgograd, Russia
tel.: +7 8442 291435, fax: +7 8442 248182
e-mail: legionerpav@gmail.com

Vyacheslav Prytkov

Volgograd State Technical University, Faculty of Auto-tractor
Lenin Avenue 50, 400005 Volgograd, Russia
tel.: +7 8442 3355422, fax: +7 8442 248182
e-mail: Prytkov34@gmail.com

Elena Cherkashina

Volgograd State Technical University, Faculty of Auto-tractor
Lgovskaya Street 17, 400065 Volgograd, Russia
tel.: +7 8442 751629, fax: +7 8442 248182
e-mail: vilena400065@inbox.ru

Michail Liashenko

Volgograd State Technical University, Faculty of Auto-tractor
Parhomenko Street 15-14, 400087 Volgograd, Russia
tel.: +7 8442 248171, fax: +7 8442 248182
e-mail: tslmv@vstu.ru

Abstract

The problem of control by a semi-active suspension for increase a ride comfort of a vehicle is considered. On an example of multidimensional mathematical model of the bus, an efficiency of a semi-active suspension with controllable shock absorbers and pneumatic springs is analyzed. Results of computer simulation of movement of the bus with a semi-active suspension on road with a stochastic profile are presented. Calculations have been executed in the program of modelling of dynamics of systems solid and elastic bodies - FRUND. As conclusions it is possible to notice that the semiactive suspension with controlled under offered laws elastic and damper elements allows to depress considerably dynamic ability to loading spring vehicle parts, thereby, raising its stroke ride comfort, without deterioration of parameters influencing stability and movement roadability. Results of numerical experiment and offered laws of controlling can form a basis for creation structurally concerning simple semiactive suspension systems a not demanding for work of a significant amount of power. The semiactive suspension with controlled under offered laws elastic and damper elements allows depressing considerably dynamic ability to loading spring vehicle parts, thereby, raising its stroke ride comfort, without deterioration of parameters influencing stability and movement roadability.

***Keywords**: vehicle, semi-active suspension, ride comfort, shock absorber, pneumatic spring, multidimensional model*

Passive suspension systems with no controllable standard characteristics are the most widespread on producing vehicles. Firstly, it is caused by an enough simple design, concerning high reliability, absence of necessity of power supply. However, potential possibilities of such systems in satisfaction of growing requirements from ride comfort are rather limited and have reached the maximum. In this connection leading foreign motor-car manufacturers, such as Mercedes-Benz, Volkswagen, Citroen etc. engage the cars equipped with various versions of controllable suspension systems. These systems make expensive a total cost, and also complicate a design in the lineup, but during too time allows improving characteristics of stroke ride comfort essentially.

Engineers should come across this number of problems at suspension designing. One of them consists in the conflict of requirements to a suspension from stroke ride comfort, stability and roadability. On the one hand, the suspender should have "softer" options to increase comfort of the car for the driver and passengers, to provide safety of transported cargoes. On the other hand, the suspender should have more "harder" options for stability and roadability increase namely that there were no dangerous body heel in a cross-section plane by turning manoeuvre, and also longitudinal body pecks by acceleration and braking. One more problem is that to vehicle traffic conditions, such as speed regime, spring weight size, a way covering, a movement mode are peculiar to change. Therefore characteristic of passive suspension system cars get out proceeding from average service conditions. It is necessary to search for the compromise between inconsistent requirements of a suspension. However optimized on all conditions series of car suspension exploitation, it does not appear optimum in each of concrete current road situations. Controllable suspension systems allow resolving existing problems or reducing them to a minimum.

From the analysis of producing controllable suspension systems it is possible to conclude that semiactive systems are optimal than passive suspension systems from the point of view of improvement of stroke ride comfort, roadability, stability. And they optimal than active suspension systems because it is enough low power consumption. Semiactive suspension systems are such systems in which external energy is spent only for change of passive elements parameters, for example, changing shock-absorber resistance or changing ruggedness of elastic elements.

Overall performance of controlled suspension system in many respects depends on the chosen control algorithm. In this work, algorithms of damper controlling (hydraulic shock absorber) and elastic controlling (air spring) have been offered by vehicle suspension elements. Them efficiency has been analysed by means of computer modelling. The throat resistance of hydraulic absorber and ruggedness of an air spring accordingly were used as controlling parameters of damper and elastic elements. In practical realization of suspension system a change of resistance of the hydraulic shock absorber can be reached by changing of throttle section or a working liquid viscosity, and change of elastic characteristic ruggedness by means of variation its air spring working volume.

The algorithm based on step variants of controlling algorithms has been developed for controlling of the absorber «sky hook» [3-6] and «ground hook» [7] and allowing to find balance between oscillations reduction cushioned and uncushioned masses.

The analytical expression of the offered algorithm, with reference to one-basic two-mass vehicle model (quarter vehicle model), represented on Fig. 1, represents the following system:

$$r = \begin{cases} r(\psi = 0{,}5),\ \dot{z}(\dot{z} - \dot{\zeta}) > 0 \ and \ -\dot{\zeta}(\dot{z} - \dot{\zeta}) > 0, \\ r(\psi = 0{,}1),\ \dot{z}(\dot{z} - \dot{\zeta}) > 0 \ and \ -\dot{\zeta}(\dot{z} - \dot{\zeta}) \le 0, \\ r(\psi = 0{,}2),\ \dot{z}(\dot{z} - \dot{\zeta}) \le 0 \ and \ -\dot{\zeta}(\dot{z} - \dot{\zeta}) > 0. \end{cases} \quad (1)$$

where:
ψ - factor of system aperiodicity.

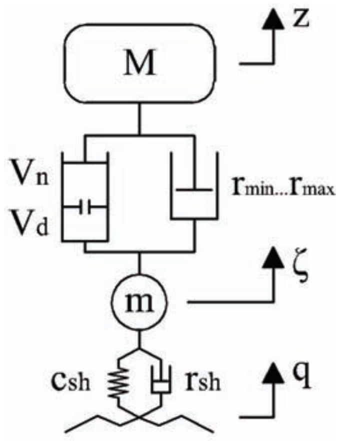

Fig. 1. One-basic two-mass model of a suspension with a controllable damper and a controllable pneumatic spring: M, m – cushioned and uncushioned masses, Z, ζ – vertical displacement of cushioned and uncushioned masses; q - kinematic fluctuation, c_{sh}, r_{sh} – rigidity and coefficient of resistance of the tyre, V_0, V_d - the basic and additional volumes of the pneumatic spring, r_{min}, r_{max} - the min and max values of damper resistance coefficient

As the law of controlling the air spring authors had been offered the algorithm, which essence consists in change of air spring ruggedness depending on current values of vertical speeds cushioned and uncushioned masses. Using the equation of the elastic characteristic of passive pneumatic element [2] it is possible to receive the elastic characteristic of an air spring with changeable ruggedness, which will pay off under the following formula:

$$F_u = \left[p_a \left(\frac{V_n}{V_n + F_e \lambda_i} \right)^k \left(\frac{V_n + V_d + F_e \lambda_i}{V_n + V_d + F_e (z - \zeta)} \right)^k - p_v \right] F_e , \tag{2}$$

where:
F_u – elastic force of a pneumatic spring;
p_a – absolute pressure in a pneumatic spring;
p_v – atmospheric pressure;
k – an indicator of a polytrope;
F_e – the effective area of a pneumatic spring;
V_o , V_d – the basic and additional volumes of a pneumatic spring;
λ_i – suspension deformation corresponding to the moment of switching-on (switching-off) of additional volume of a pneumatic spring.

Ruggedness of an air spring at the expense of working volume change with reference to one-basic two-mass model of the vehicle (quarter vehicle model) it is possible to present controlling algorithm of analytically following system:

$$V = V_n + F_e (z - \zeta) + \begin{cases} V_d, \ \dot{z}(\dot{z} - \dot{\zeta}) \leq 0; \\ V_d, \ \dot{z}(\dot{z} - \dot{\zeta}) > 0 \ and \ abs(z - \zeta) > abs(\lambda_i); \\ 0, \ \ \dot{z}(\dot{z} - \dot{\zeta}) > 0 \ and \ abs(z - \zeta) \leq abs(\lambda_i). \end{cases} \tag{3}$$

Probe of overall performance of vehicle semiactive suspension system was spent by means of computer system of modelling FRUND [1]. By means of that program complex, the spatial multi-mass vehicle model which general view is presented on Fig. 2 has been generated.

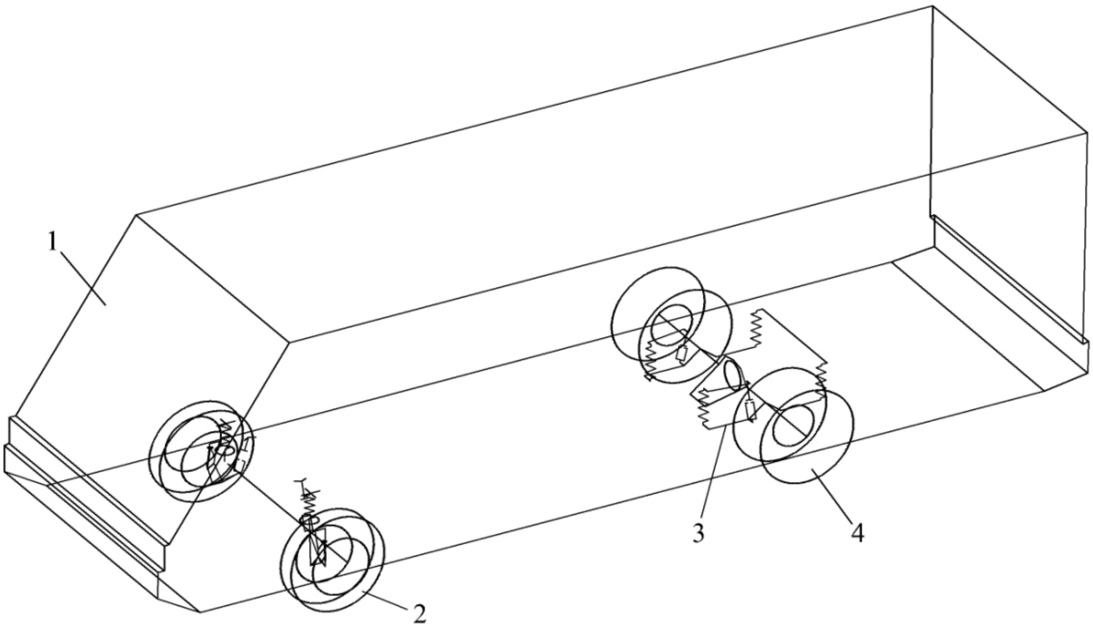

Fig. 2. The General view of the settlement scheme of the bus: 1 – a body, 2 – a forward wheel, 3 – a rear axle, 4 – a back wheel

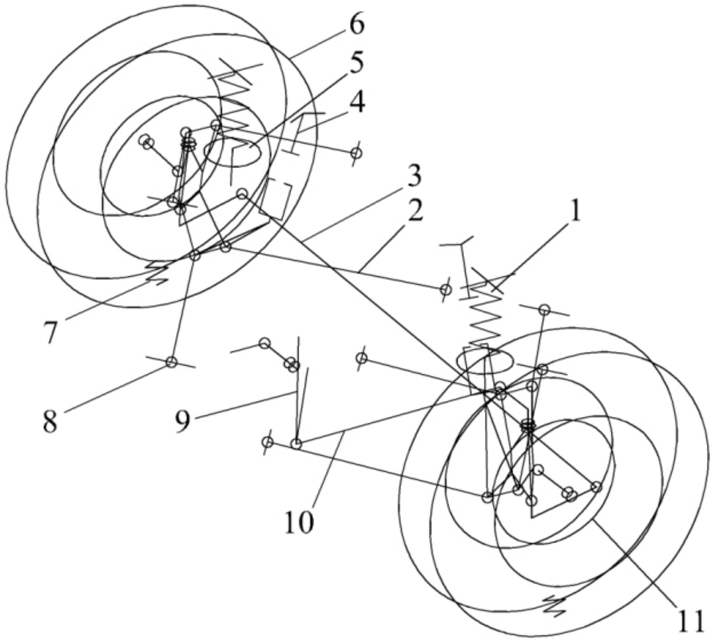

Fig. 3. The front suspension: 1 - an air spring, 2 - the lever, 3 - a cross-section track rod, 4 - the hydraulic shock absorber, 5 - a swivel member, 6 - a front wheel, 7 - the tire, 8 - the hinge; 9 - drop arm, 10 - drag link, 11 - a nave

The vehicle model is developed on the basis of bus design data and contains some mobile bodies: a body 1, forward 2 and back 4 wheels, the rear axle 3, and as guiding elements of suspension system. The model of suspension system contains basic elements of forward and back suspensions, and also completely reproduces their kinematics. The forward suspension, which scheme is represented on Fig. 3, represents an independent lever suspension on cross-section levers 2.

It contains elastic 1 and damper 4 elements in the form of air springs and hydraulic shock absorbers accordingly. On Fig. 4, the scheme of a rear axle suspension of the bus, which consists of the pull-rods 7, is presented. They unload air springs 1 from longitudinal and cross-section forces. Also, hydraulic shock absorbers 3 and the antiroll bar 8 are presented. The characteristic of tires of front and rear wheels considers their separation from road. The elastic characteristic of front and rear suspensions has the sites describing work of dynamic stroke arresters – elastic elements of the big ruggedness.

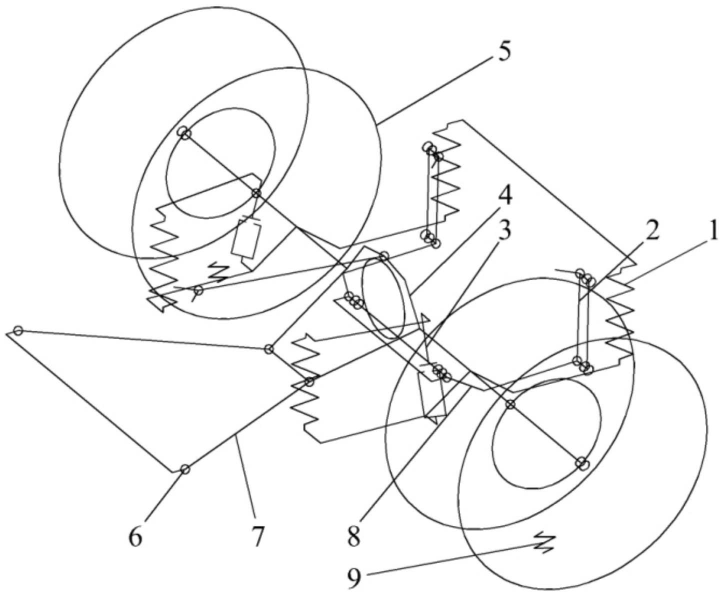

Fig. 4. The rear suspension: 1 - an air spring, 2 - an earring, 3 - the hydraulic shock absorber, 4 - the rear axle, 5 - a rear wheel, 6 - the hinge, 7 - pull-rod, 8 - the antiroll bar, 9 - the tire

For check of overall performance of controlled suspension system were modelling of an in-line motion of the bus with a speed of 60 km/h on road with the casual profile, road of autoproving ground representing experimental realization by NAMI (Central scientific research automobile and automotive engine institute), type «a smooth cobble-stone» that corresponds to road of satisfactory quality has been spent. Two variants of suspension systems have been considered: 1 – a suspender with regular absorbers (ψ=0,3) and regular pneumosprings ($Vn=0.011m^3$, Vd=0), 2 – a semiactive suspender with steered absorbers (ψ=0.1-0,5) and steered pneumosprings ($Vn=0.011m^3$, $Vd=0\text{-}0.01m^3$).

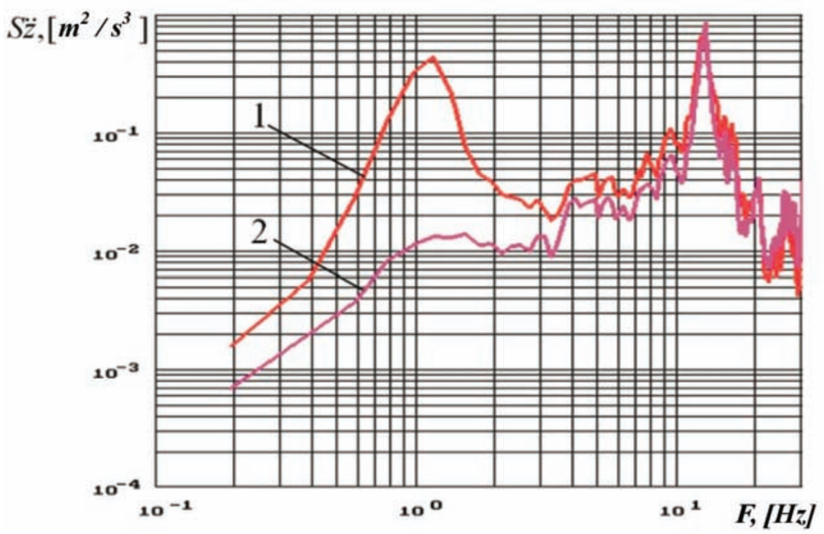

Fig. 5. Spectra of vertical speedups of a body over a front suspension:
1 - a regular suspension, 2 - a semiactive suspension

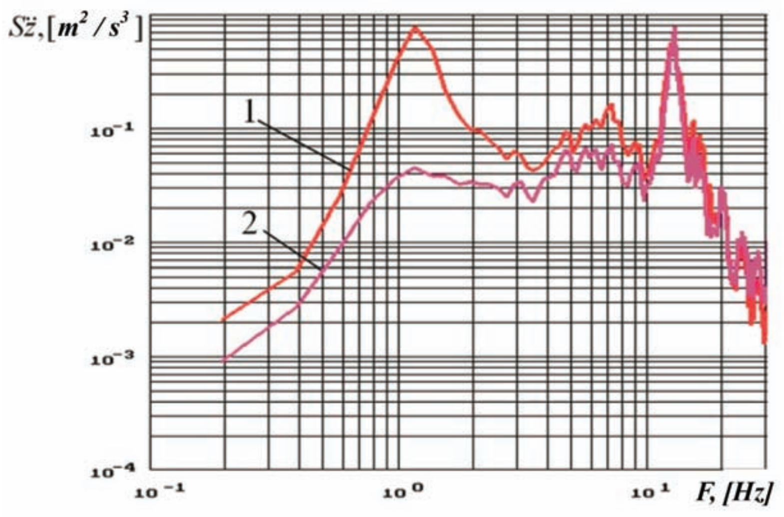

Fig 6. Spectra of vertical speedups of a body over a rear suspension:
1 - a regular suspension, 2 - a semiactive suspension

The spectra of vertical speedups of a body in various points (Fig. 5, 6) as criterion of vehicle stroke ride comfort were used. For descriptive reasons in Tables 1, 2 are presented mean square values (MSV) of vertical speedups of a body in two points for regular and semiactive suspension systems.

Tab. 1. MSV vertical speedups of a body over a front suspension in 1/3-octave frequencies strips, $m/s^2\ 10^{-3}$ (1 – a regular suspension, 2 – a semiactive suspension)

Compound frequencies f [Hz]	**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**
1	248	271	120	114	104	118	167	193	227	286	458	1070	538
2	46	51	53	63	66	82	134	155	182	235	343	959	426

Tab. 2. MSV vertical speedups of a body over a rear suspension in 1/3-octave frequencies strips, $m/s^2\ 10^{-3}$ (1 – a regular suspension, 2 – a semiactive suspension)

Compound frequencies f [Hz]	**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**
1	276	396	203	193	167	179	207	281	418	362	343	976	483
2	84	92	86	112	108	134	167	230	297	252	297	860	389

The spectra of vertical speedups of uncushioned elements of front and rear suspenders were used as criteria of an estimation of roadability and stability of vehicle movement (Fig. 7, 8). The spectra of relative conveyances of front and rear wheel tires were used as spectra of relative conveyances of air springs of front and rear suspensions (Fig. 9, 10).

From the analysis of schedules represented on Fig. 5, 6 and Tables 1, 2 it is possible to conclude that the semiactive suspension system allows to raise essentially stroke ride comfort the vehicle in comparison with passive system, especially in the field of a low-frequency resonance of a body and in between resonances area of body oscillations and uncushioned elements of suspension system. In turn, the values of parameters influencing roadability and movement stability of the vehicle (Fig. 7-10), for semiactive and passive suspension systems are quite comparable that allows.

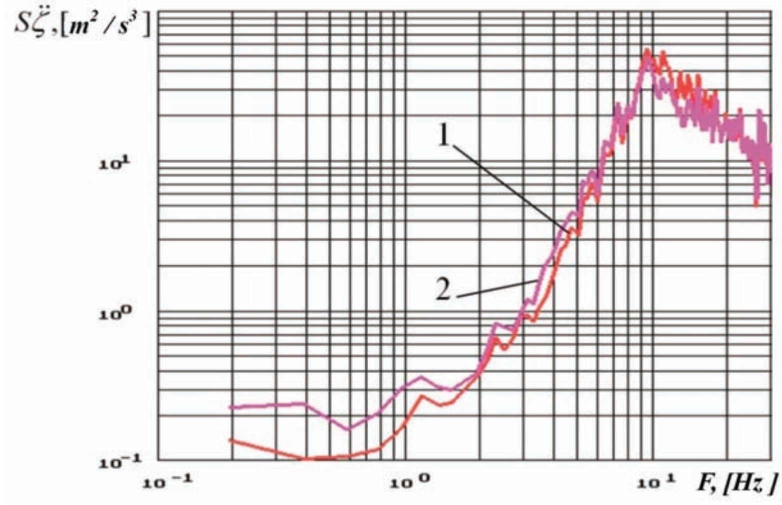

Fig. 7. Spectra of vertical speedups of the left swivel member of a front suspension:
1 - a regular suspension MSV =29.41 m/s^2, 2 - a semiactive suspension MSV =29.38 m/s^2

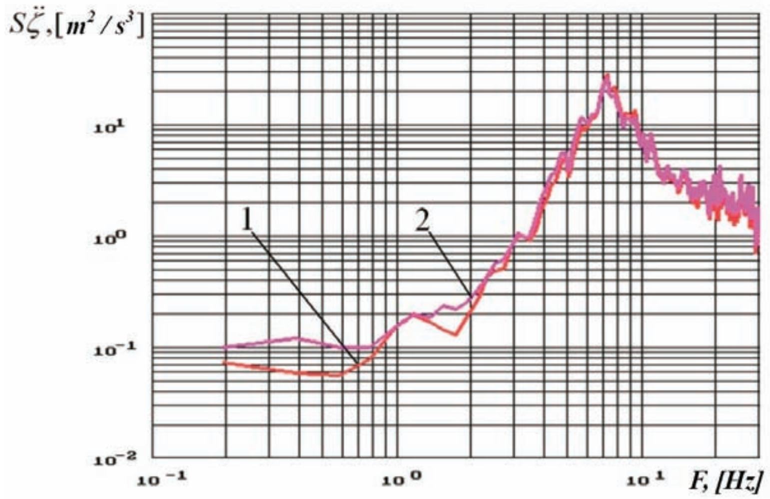

Fig. 8. Spectra of vertical speedups of the rear bridge:
1 - a regular suspension MSV =12.71 m/s^2, 2 - a semiactive suspension MSV =13.16 m/s^2

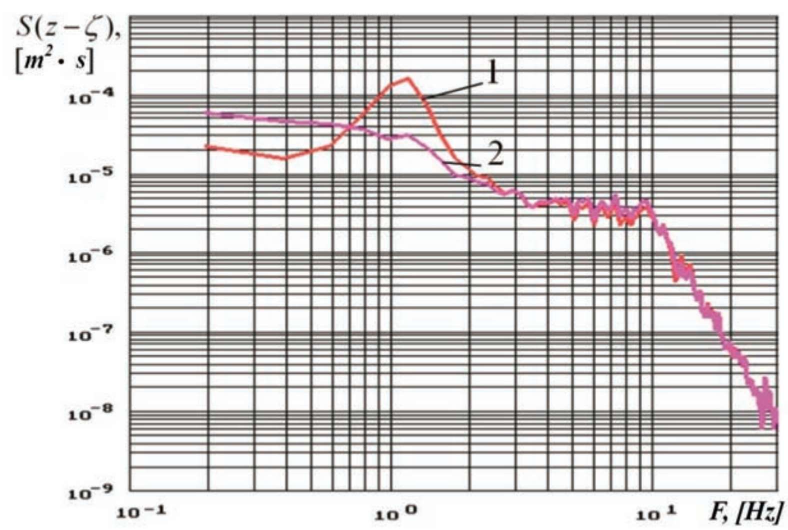

Fig. 9. Spectra of relative conveyances of a front left pneumospring:
1 - a regular suspension MSV =0.0131 m, 2 - a semiactive suspension MSV =0.0106 m

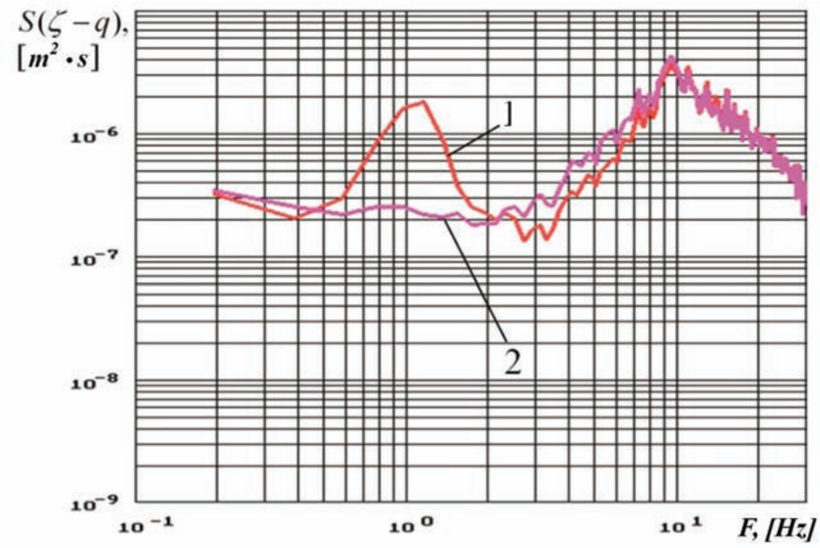

Fig. 10. Spectra of relative conveyances of the front left tire:
1 - a regular suspension MSV =0.00637 m, 2 - a semiactive suspension MSV =0.00634 m

to assume preservation of roadability and movement stability of the vehicle at level at least corresponding to passive system.

As conclusions it is possible to notice that the semiactive suspension with controlled under offered laws elastic and damper elements allows depressing considerably dynamic ability to loading spring vehicle parts, thereby, raising its stroke ride comfort, without deterioration of parameters influencing stability and movement roadability. Results of numerical experiment and offered laws of controlling can form a basis for creation structurally concerning simple semiactive suspension systems a not demanding for work of a significant amount of power.

References

[1] Gorbtsov, A. S., *Algoritmy chislennogo integrirovaniya uravneniy dvizheniya sistemy tel s mnozhitelyami Lagranzha*, Mashinostoryenie i inzhenernoye obrozovaniye, No 3, s. 20-27, 2005.

[2] Pevzner, Ya. M., *Pnevmaticheskiye i gidropnevmaticheskiye podveski*, L.: [b. i.], 1963.

[3] Podzorov, A. V. *Issledovaniye upravlyayemoy sistemy podressorivaniya avtobusa posredstvom prostranstvennoy matematicheskoy modeli,* Izv. VolgGTU. Seriya "Nazemnyye transpornyye sistemy": mezhvuz. sb. nauch. st., VolgGTU, Vyp. 3, No 10, s. 67-73, Volgograd 2010.

[4] Podzorov, A. V., Gorbtsov, A. S., Lyashenko, M.V., *Matematicheskaya model` upravlyayemoy sistemy podressorivaniya ATS,* Avtomobil`naya promychlennost`, No 9/2010, s. 18-21.

[5] Podzorov, A. V., *The optimal control of the multidimensional system at stochastic excitation,* Journal of KONES. Powertrain and Transport (Poland), Vol. 16, No. 1, pp. 411-418, 2009.

[6] Karnopp, D. C., *Vibration control using semi-active force generators,* Transactions of the ASME. Journal of Engineering for Industry, Vol. 96, pp. 619-626, 1974.

[7] Valášek, M., *Extended ground-hook – new concept of semi-active control of truck's suspension*, Vehicle system dynamics, Vol. 27, No. 5-6, pp. 289-303, 1997.