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VIBRATION-PROTECTIVE PROPERTIES OF THE SUSPENSION WITH STEPWISE REGULATION OF INELASTIC RESISTANCE IN THE VIBRATION CYCLE

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A car suspension is a set of mechanisms. It creates an elastic connection between the undercarriage and the wheels of a motor vehicle. It adjusts the position of the body while the car is moving. The car suspension also reduces the dynamic loads on the motor vehicle undercarriage and wheels. Dynamic loads are the cause of body vibrations. They are formed when the car wheels meet into contact with the road surface. The dynamics of the motor vehicle movement largely depends on the vibration-protective properties of the body suspension. It is the most important parameters of the motor vehicle suspension when driving on uneven roads. Vibration-protective properties have a significant impact on the operational properties of the motor vehicle. These include smoothness, average speed, fuel efficiency, etc. To improve the vibration-protective properties of the car suspension, its elastic and damping characteristics should change depending on different conditions of the motor vehicle implementation. The article investigates the vibration-protective properties of the suspension with stepwise regulation of inelastic resistance in the vibration cycle. The article presents a mathematical model of a single-mass single-support oscillatory system with an adjustable damper. A theoretical comparative analysis of vibration-protective properties of a single-mass single-support suspension with unregulated and instantly adjustable damping in the oscillation cycle is carried out. It was found that, in comparison with unregulated damping, its instantaneous regulation in the oscillation cycle provides a low level and approximate constancy of the vertical acceleration range of the sprung mass in the resonant oscillation zone, but causes an abrupt change in acceleration at the moments when the damping is turned off when the suspension passes its middle position.

Keywords: car suspension, vibration-protective properties, single-mass single-support suspension, unregulated, instantly adjustable damping

1. Introduction

In the works [1, 2] it is shown that in the single-mass model of the vehicle suspension (fig. 1) with harmonic kinematic disturbance in the oscillation cycle, there are zones of ineffective shock absorber operation, in which the presence of a conventional unregulated shock absorber increases the kinetic energy of the sprung mass, which reduces the vibration-protective properties of the suspension and increases the loss of energy in it.

The authors of [1–20] proved that to maximize the reduction of vertical displacements of the sprung mass, an instant (stepwise) shutdown of damping in ineffective zones is necessary, which, in contrast to unregulated damping, allows obtaining a nonresonant amplitude-frequency characteristic (AFC) of vertical displacements, which is characteristic only of active suspension systems. Such regulation of damping in the oscillation cycle will further be called the optimal instantaneous regulation of damping (fig. 1).

Mathematically, the description of the dynamics of a single-mass oscillatory system (fig. 1b) with instantaneous optimal damping control can be presented in the following form:

$$m\ddot{z} + uk(\dot{z} - \dot{q}) + c(z - q) = 0, \quad (1)$$

where m – is the sprung mass; k – is the damping factor; c – suspension stiffness; q – is the coordinate of the disturbing effect; z – movement of the sprung mass; u – parameter for controlling the shock absorber resistance:

$$u = \begin{cases} 0 & at \;\; \dot{z} \cdot (\dot{z} - \dot{q}) < 0, \\ 1 & at \;\; \dot{z} \cdot (\dot{z} - \dot{q}) > 0. \end{cases} \quad (2)$$

Thus, the minimum amplitudes of displacement of the sprung mass is achieved with an instantaneous deactivation of damping in the zones of ineffective operation of the shock absorber, and the minimum minimorum is achieved provided that zero acceleration is achieved $\ddot{z}(t) = 0$ as a result of switching off the damping (it is this condition that determines the optimal control of the damping during instantaneous switching).

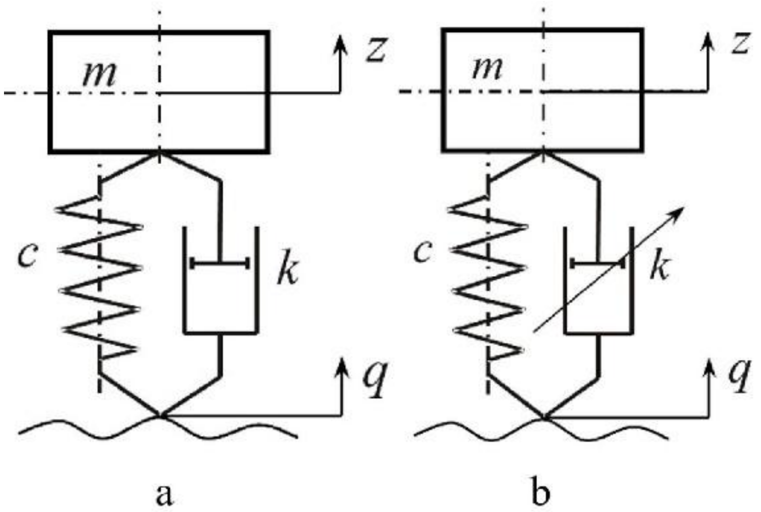


Fig. 1. Design diagrams of a single-mass single-support oscillatory system with unregulated and adjustable dampers

Zero vertical acceleration of the sprung mass $\ddot{z}(t) = 0$ as a result of disabling damping, it is achieved at a certain (optimal) value of the maximum damping level at a given disturbance frequency $\psi(\iota) = \frac{k}{2\sqrt{mc}}$ (for example, at the resonant frequency of the disturbance $\iota = 1$ relative attenuation coefficient $\psi \approx 0{,}92$, and at a resonant relative frequency $\iota = 2$ optimal maximum value of the relative attenuation coefficient $\psi \approx 0{,}49$). Here ψ – relative attenuation coefficient; $\iota = \omega/\omega_0$ – forced vibration relative frequency, ω_0 – natural frequency of the sprung mass, ω – frequency of forced vibrations of the sprung mass.

Thus, the optimal value of the shock absorber resistance when adjusting the damping in the vibration cycle depends not only on the phase (amplitude and direction), but also on the vibration frequency.

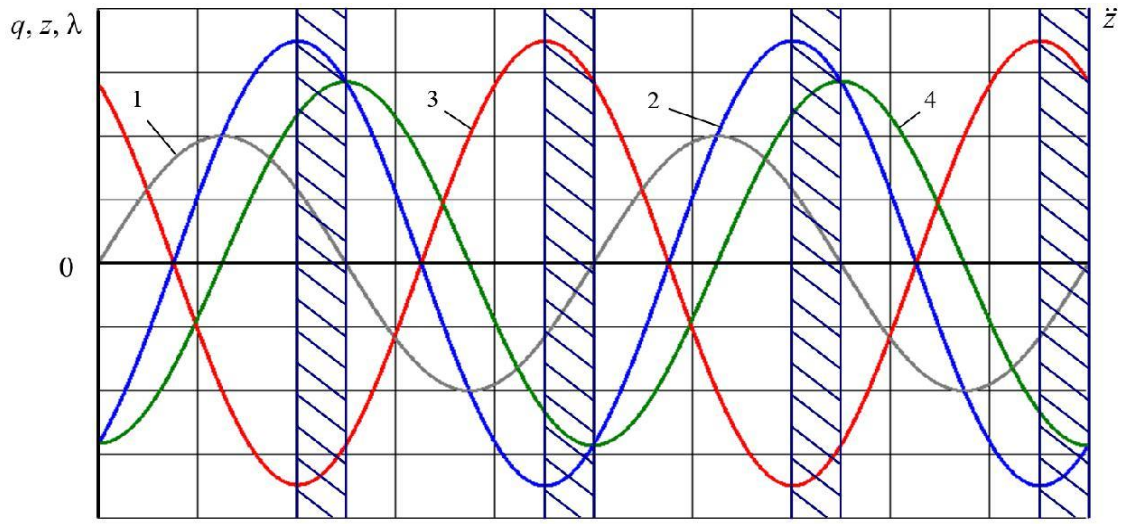
2. Analysis of the results of theoretical studies of vibration-protective properties of a single-mass single-support suspension with unregulated and instantly adjustable damping in the oscillation cycle

Fig. 2 shows oscillograms of the suspension at the resonant frequency $\iota = 1$ without damping regulation at $\psi = 0{,}35$ and with optimal instantaneous damping control at $\psi = 0{,}92$.

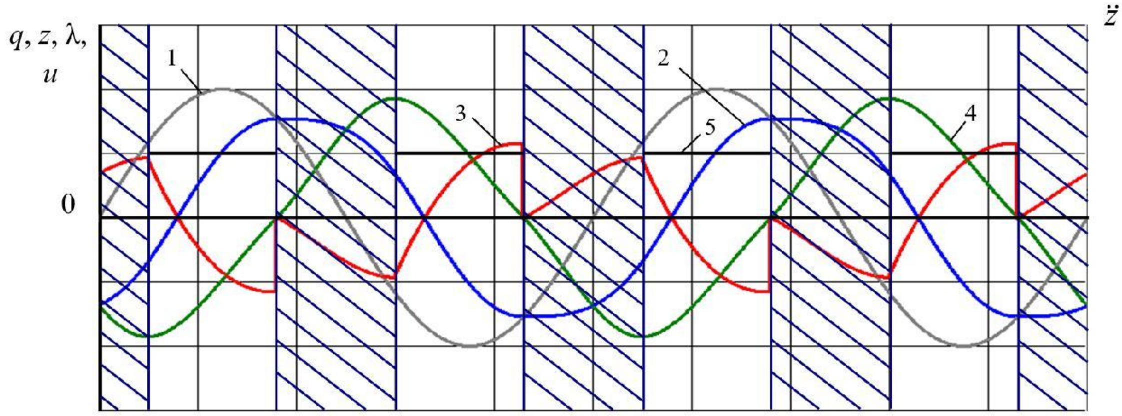
Fig. 3 shows suspension operating diagrams with optimal instantaneous damping control at resonance frequency $\iota = 1$ and $\psi = 0{,}92$ and at a resonant frequency $\iota = 2$ and $\psi = 0{,}49$. From a comparison of the diagrams in fig. 3 and 1, it can be seen that the introduction of optimal damping control by the vibration phase not only reduces the deformation and maximum suspension force, but also significantly reduces the heat release determined by the area of the working diagram.

However, the optimal instantaneous damping control is accompanied by an abrupt change in acceleration when the damping is turned off (fig. 2b), which significantly disrupts the ride and causes jolts and shocks in the suspension. In addition, the use of an unregulated shock absorber or the absence of a shock absorber makes it possible to obtain substantially lower amplitudes of vertical accelerations of the sprung mass in a wide frequency range than when using the optimal instantaneous regulation.

Fig. 4 shows oscillograms of steady-state vibrations of the suspension with infinitesimal damping and with optimal instantaneous damping control at the resonant frequency, from a comparison of which it can be seen that the optimal instantaneous damping control at the resonant disturbance frequency in comparison with an oscillatory system without a damper leads to a decrease in displacements, but causes growth of accelerations.



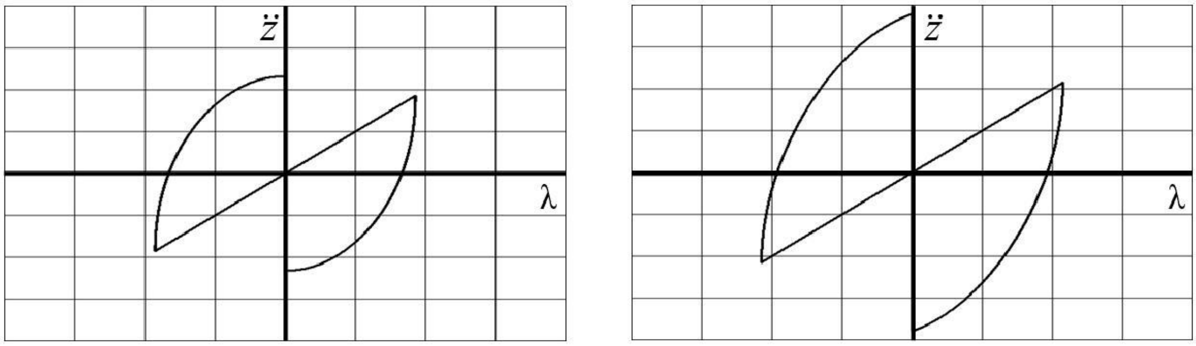
a



b

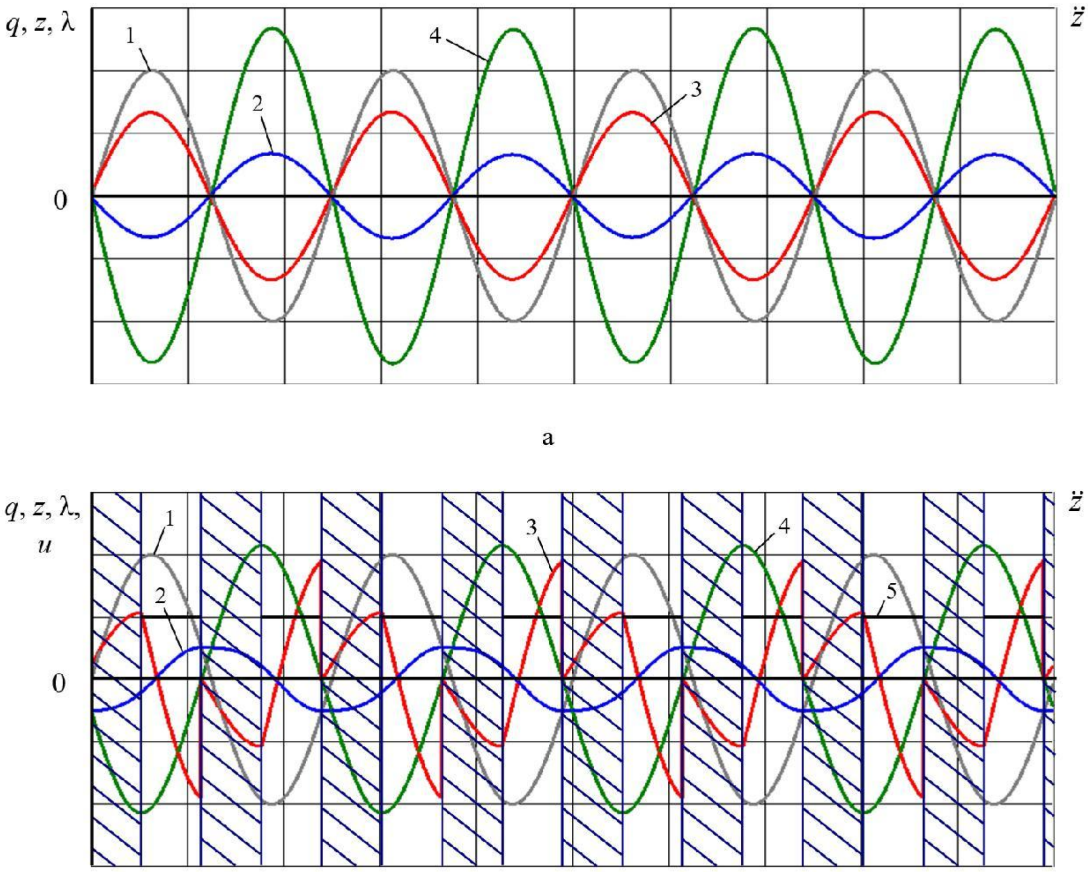
Fig. 2. Oscillograms of the suspension on the resonant ι = 1: a – without regulation at ψ = 0,35; b – with optimal instantaneous damping control at ψ = 0,92; 1 – kinematic disturbance; 2 – absolute movement of the sprung mass; 3 – sprung acceleration; 4 – relative displacement of the sprung mass (suspension deformation $\lambda = z - q$); 5 – control parameter *u*

Fig. 5 shows the frequency response of the movements of the sprung mass on the suspension without damping (curve 1), with constant damping (curve 2) and with optimal instantaneous damping control (curve 3), as well as the phase-frequency characteristics (PFC) of the absolute oscillations of the sprung mass (curve 4) , Phase-frequency characteristic of the magnitude of the section of operation of the oscillatory system with disabled damping (curve 5), dependence with optimal damping control (curve 6). Here z_0 – is the amplitude of the absolute oscillations of the sprung mass, z_0 – is the amplitude of the kinematic disturbance.



a b

Fig. 3. Working diagrams of the suspension with the optimal instantaneous damping control according to the algorithm (13): a – at $\iota = 1$ and ψ = 0,92; b – at $\iota = 2$ and ψ = 0,49



b

Fig. 4. Oscillograms of the suspension at the resonant frequency ι = 2: a – with infinitesimal damping; b – with optimal instantaneous damping control at ψ = 0,49; 1 – kinematic disturbance; 2 – absolute movement of the sprung mass; 3 – sprung acceleration; 4 – relative displacement of the sprung mass (suspension deformation $\lambda = z - q$); 5 – control parameter u

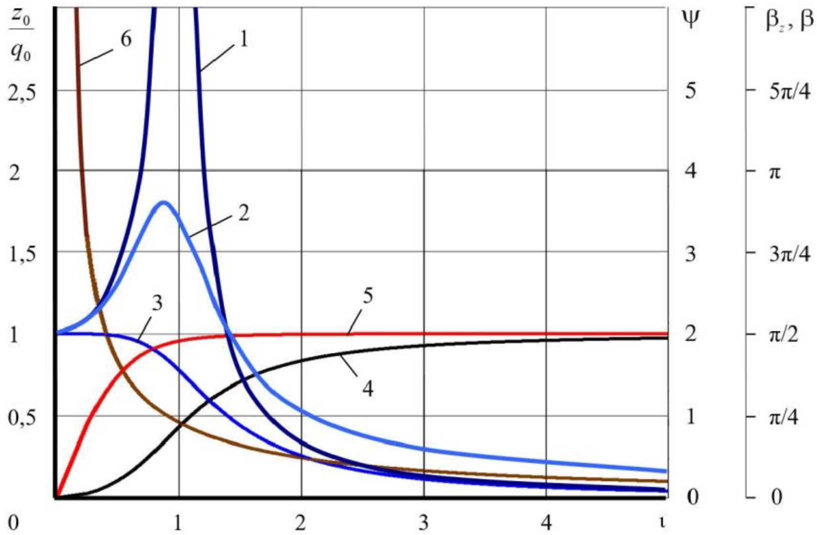


Fig. 5. Characteristics of a single-mass oscillatory system with harmonic kinematic disturbance: 1, 2 and 3 – frequency response of the displacements of the sprung mass on the suspension without damping, with constant damping at ψ = 0,35 and with optimal instantaneous damping control according to the algorithm (13); 4 – Phase-frequency characteristic of absolute oscillations of the sprung mass; 5 – Phase-frequency characteristic of the magnitude of the section of operation of the oscillatory system with disabled damping; 6 – dependence $\psi(\iota)$ with optimal instantaneous damping control

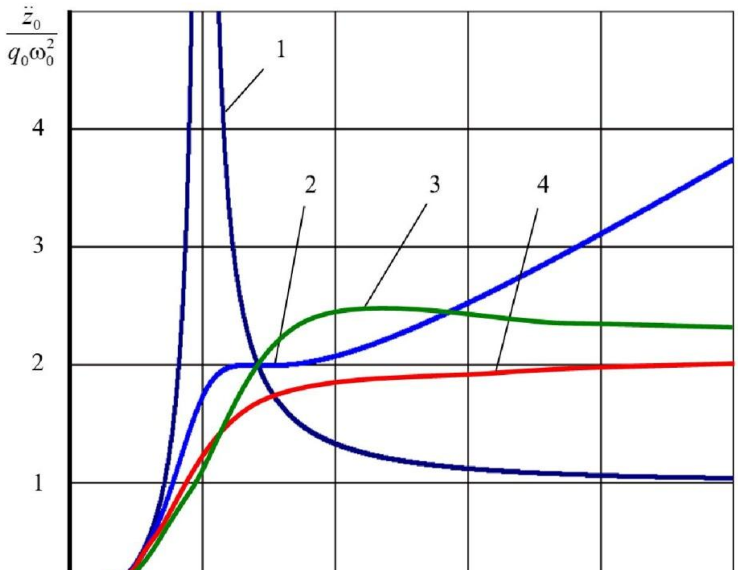


Fig. 6. Frequency response of the sprung mass accelerations on the suspension: 1 – without damping; 2 – with constant damping at ψ = 0,35; 3 – with damping, variable according to the dependence determined by curve 6 in Figure 8, but without instant cyclic regulation; 4 – with optimal instantaneous damping regulation according to the algorithm (13)

Fig. 6 shows the frequency response of the acceleration of the sprung mass on the suspension without damping (curve 1), with constant damping at ψ = 0,35 (curve 2), with frequency-controlled damping without instantaneous cyclic control according to algorithm (13) (curve 3) and with optimal instantaneous regulation of damping in frequency and in the oscillation cycle (curve 4).

From the analysis of the graphs in fig. 5 and 6, it can be seen that in comparison with constant damping at ψ = 0,35 (curves 2 in fig. 5 and 6) with optimal instantaneous regulation of damping, the maximum values of displacement of the sprung mass (curve 3 in fig. 5) and its accelerations (curve 4 in fig. 6) are significantly reduced in the entire frequency range. When damping is regulated only by frequency and there is no instant cyclic control of damping, the acceleration of the sprung mass becomes slightly larger (up to 20 %) (curve 3 in fig. 6).

3. Conclusions

1. In comparison with constant damping, its instant (stepwise) regulation in the oscillation cycle provides a low level and constancy of the vertical acceleration range of the sprung mass in the resonant oscillation zone.

2. Optimal instantaneous (stepwise) control of damping in the oscillation cycle provides minimum amplitudes of displacement of the sprung mass with a dynamic coefficient in resonance less than 1, but causes a jump in acceleration at the moments of damping off when the suspension passes its middle position.

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ВИБРОЗАЩИТНЫЕ СВОЙСТВА ПОДВЕСКИ ПРИ СТУПЕНЧАТОМ РЕГУЛИРОВАНИИ НЕУПРУГОГО СОПРОТИВЛЕНИЯ В ЦИКЛЕ КОЛЕБАНИЙ

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Подвеска автомобиля – это совокупность механизмов, создающих упругую связь между несущей системой и колесами автотранспортного средства (АТС). Она регулирует положение кузова во время движения автомобиля. Подвеска также обеспечивает уменьшение динамических нагрузок на несущую систему АТС и колеса. Динамические нагрузки, которые являются причиной возникновения колебаний кузова, формируются при контакте колес автомобиля с дорожным покрытием. Динамика движения АТС в большей степени зависит от виброзащитных свойств подвески кузова, которые являются наиболее важными показателями подвески АТС при движении по неровным дорогам. Виброзащитные свойства оказывают существенное влияние на эксплуатационные свойства АТС: плавность хода, средняя скорость движения, топливная экономичность и др. Для улучшения виброзащитных свойств подвески ее упругие и демпфирующие характеристики должны меняться в зависимости от различных условий реализации АТС. В статье исследуются виброзащитные свойства подвески автомобиля при ступенчатом регулировании неупругого сопротивления в цикле колебаний. В статье приведена математическая модель одномассовой одноопорной колебательной системы с регулируемым демпфером. Проведен теоретический сравнительный анализ виброзащитных свойств одномассовой одноопорной подвески с нерегулируемым и мгновенно регулируемым в цикле колебаний демпфированием. Установлено, что по сравнению с нерегулируемым демпфированием его мгновенное регулирование в цикле колебаний обеспечивает низкий уровень и примерное постоянство размахов вертикальных ускорений подрессоренной массы в зарезонансной зоне колебаний, но вызывает скачкообразное изменение ускорений в моменты отключения демпфирования при прохождении подвеской своего среднего положения.

Ключевые слова: подвеска автомобиля, виброзащитные свойства, одномассовая одноопорная подвеска, мгновенно-регулируемое демпфирование

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