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# Efficiency of Shock Absorber in Vehicle Suspension

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**Abstract**

This article presents the analytical proof of the presence of two inefficient areas in vehicle suspension oscillation cycle in linear shock absorber work. It derived on the basis of analysis of the dynamics equations for the linear single-support single-mass vibrating system with fixed elastic and damping characteristics at harmonic kinematic disturbance. Expressions of the full, efficient and inefficient work of the linear damper in one oscillation cycle are derived. Thereunder the equations for calculation of the effective shock absorber work ratio for various types of damping characteristics are determined. The graphs of the effective shock absorber work ratio on the relative frequency of the disturbance and the relative damping factor are implemented. The evaluation of vehicle suspension efficiency based on the values of inefficient force pulse ratio for known efficient control algorithms of elastic-damping characteristic of pneumatic shock absorber.

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*Keywords:* vehicle; suspension; shock absorber; sprung mass; deformation; oscillation cycle; inefficient work area.

## 1. Introduction

The damper work in the oscillation cycle is characterized by the presence of two areas in which its action is aimed at increasing velocity and displacement amplitude of the sprung mass [1-11]. These areas are characterized by the coincidence of the force action direction from the damper to the sprung mass with the mass movement direction. This is possible when the direction of sprung mass movement and the direction of disturbing base are the same but the speed of the disturbing base is more than the absolute value of the sprung mass velocity. The beginning of each area corresponds to a moment of changing the direction of the mass movement, and the end corresponds to a moment of changing the direction of suspension deformation.

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## 2. Mathematical modelling

For evaluation the efficiency of the shock absorber work in the vehicle suspension it is necessary to consider the linear single-mass single-support vibrating system with fixed elastic and damping characteristics at harmonic kinematic disturbance (Fig. 1).

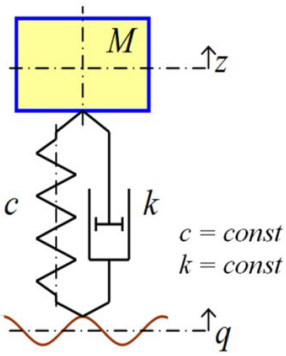


Fig. 1. Design scheme of linear vibrating system with fixed elastic and damping characteristics:
M – sprung mass; z – displacement of sprung mass; q – kinematic disturbance; c – stiffness of elastic element (spring);
k – coefficient of inelastic resistance of the damping element (shock absorber).

First we need to analyze for this the dynamic equations describing the given vibrating system under harmonic kinematic disturbance, and their solutions for the absolute and relative oscillations of the sprung mass.

This linear vibrating system is described by the following differential equation:

$$M\ddot{z} + k\left(\dot{z} - \dot{q}\right) + c\left(z - q\right) = 0 \quad \text{either} \quad \ddot{z} + 2h\left(\dot{z} - \dot{q}\right) + \omega_0^2\left(z - q\right) = 0\,, \tag{1}$$

where $M$ is a sprung mass; $c$ is a stiffness of elastic element (spring); $k$ is coefficient of inelastic resistance of the damping element (shock absorber); $h$ is coefficient of relative resistance of shock absorber ($h = k/2M$); $\omega_0$ is natural frequency of suspension oscillations ($\omega_0 = \sqrt{c/M}$); $z$ – displacement of sprung mass; $q$ – kinematic disturbance.

For calculating the absolute and relative oscillations it is more convenient to introduce equation (1) to the following forms:

$$\ddot{z} + 2h\dot{z} + \omega_0^2 z = 2h\dot{q} + \omega_0^2 q = Q_z\,, \tag{2}$$

$$\ddot{x} + 2h\dot{x} + \omega_0^2 x = \ddot{q} = Q_x\,, \tag{3}$$

where $x = q - z$ is suspension deformation; $\dot{x} = \dot{q} - \dot{z}$ is a velocity of suspension deformation; $\ddot{x} = \ddot{q} - \ddot{z}$ is an acceleration of suspension deformation; $\ddot{q}$ is disturbing kinematic impact; $Q_z$ and $Q_x$ are disturbing functions determining steady forced oscillations of suspension by coordinates $z$ and $x$.

Let the kinematic disturbance is described by harmonic law:

$$q = q_0 \sin \omega t\,, \tag{4}$$

where $q_0$ is disturbance amplitude; $\omega$ is disturbance frequency.

In this case, according to the linear suspension theory in steady mode absolute and relative oscillations of the sprung mass are also committed in harmonic law in conformity with according to the equations:

$$z = z_0 \sin\left(\omega t + \beta_z\right), \tag{5}$$

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