POSSIBILITIES FOR CONTROL OF SEMI-ACTIVE SHOCK ABSORBERS IN ORDER TO REDUCE CASES OF SUSPENSION JOUNCES WHEN BRAKING

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Abstract: *In this work a model of a truck when braking is created. The model takes into account the deflections in the suspension and vibrations of the sprung mass. The differential equations of the model are worked out. Numerical experiments are performed and some results are given. It is shown that if the shock absorber damping ratio is increased when vehicle braking, it is possible to reduce the pitch angle and the jounces in the suspension.*

Keywords: SEMI-ACTIVE SUSPENSION, SHOCK ABSORBER, JOUNCE STOP, MODELING, SUSPENSION

1. Introduction

When the ground vehicles brakes, the wheel suspension travel may be spend and shocks may occur as a result of the inclusion of the jounce stops at maximum suspension deflection. The phenomenon is known as a suspension "slam" or "jounce" in the suspension, an amalgamation of the words jump and bounce. In suspension terminology, it means the most compressed condition of a spring. For instance, many suspensions use jounce stop blocks to prevent frame-to-axle contact (slam). In addition, significant accelerations can cause extreme strains in the vehicle's construction, leading to dangerous cracks and breakdown of the chassis and suspension components. Avoiding frequent shocks is usually associated with the requirement to increase the dynamic stroke of the suspension. However, when designing the suspension, the designer is limited by the size of the individual units, the requirements for the low frame layout above the road surface, the overall planning of the vehicle, etc. The process of braking the trucks and buses, changing the dynamic stroke of the suspension and the load distribution of individual axles and wheels has been numerically and experimentally studied by a number of authors [1, 2, 3 and 4]. Experimental studies of the jounce phenomenon of trucks on typical road surfaces are published in [5].

The purpose of this work is to study numerically the possibility of reducing the "jounces" in the suspension of a truck by changing the characteristics of adjustable semi-active shock absorbers. The test will be carried out in the event of an impact corresponding to the inertia force when the vehicle braking.

2. Dynamic Model

The model used in this study is based on the model shown in the work [6]. Fig. 1 shows a scheme of the used model. It takes into account the mass of the vehicle, its moment of inertia around the transverse axis, the elasticity of the front and rear suspension and the damping of the shock absorbers.

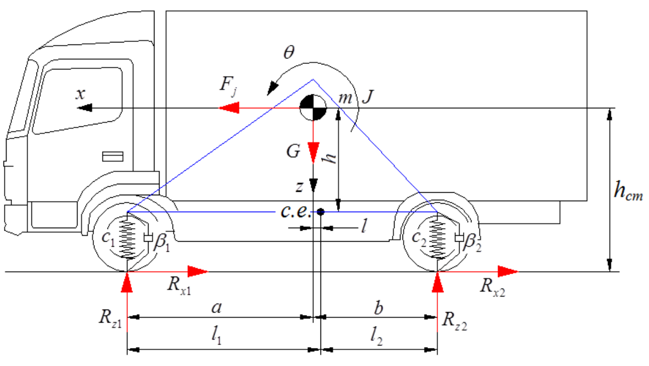

Fig. 1 *Dynamic model scheme*

The vector of the generalized coordinates is:

$$q = [x \quad z \quad \theta]^T$$

where *x* is the longitudinal displacement of the center of mass in the direction of movement of the truck, *z* - vertical displacement of the center of mass, θ - angular displacement of the body of the vehicle.

The following assumptions have been adopted:

- the characteristics of the elastic and damping elements are linear;
- the vehicle moves horizontally;
- ignoring the aerodynamic drag;
- the rolling resistance forces are neglected;
- the influence of the inertia moments of the rotating parts is neglected;
- it is assumed that the body angle is small (up to 15 °) and $\sin\theta = \theta$, $\cos\theta = 1$ is applied;

The differential equations describing the rectilinear motion of the vehicle are as follows:

$$m\ddot{x} = -R_x$$

$$m\ddot{z} = -F_{s1} - F_{s2}$$

$$J\ddot{\theta} = -aF_{s1} + bF_{s2} + M$$

where $R_x = R_{x1} + R_{x2}$, and R_{x1} R_{x2} the longitudinal reactions in the contact between the wheels and the road when the vehicle is braking.

$$F_{s1} = F_{sp1} + F_{sh1} = c_1(z + a\theta) + \beta_1(\dot{z} + a\dot{\theta})$$

$$F_{s2} = F_{sp2} + F_{sh2} = c_2(z - b\theta) + \beta_2(\dot{z} - b\dot{\theta})$$

are forces in the front and rear suspension that arise when the system is pulled out of its static equilibrium, such as:

$$F_{sp1} = c_1(z + a\theta)$$

$$F_{sp2} = c_2(z - b\theta)$$

are the elastic forces of the front and rear suspension, and:

$$F_{sh1} = \beta_1(\dot{z} + a\dot{\theta})$$

$$F_{sh2} = \beta_2(\dot{z} - b\dot{\theta})$$

are the forces of resistance created by the shock absorbers respectively from the front and rear suspension of the vehicle. Through them we can find the dynamic vertical reactions on the road:

$$R_{z1} = G_{w1} + F_{s1}$$

$$R_{z2} = G_{w2} + F_{s2}$$

where G_{w1} and G_{w2} are the static loads of the front and rear wheels.

$M = F_j.h = m.j.h$ is disturbing moment when braking;

$F_j = R_x = R_{x1} + R_{x2}$ is force of inertia;

j is braking deceleration;

h is distance from the center of elasticity to the mass center of the truck in the vertical direction (Fig. 1).

To approximate calculation of the moment of inertia, [7, 8] is used:

$$J = m.a.b$$

where J is the moment of inertia around the transverse axis of the vehicle with full load.

The recommended damping ratio of the shock absorbers for roads in good condition is [7, 9, 10]:

$$\psi_{0i} = \frac{\beta_i}{2m_i\omega_{0i}} = 0{,}2 \div 0{,}3$$

where ψ_{0i} is a damping aperiodic coefficient, m_i - the mass of the front or rear axle, ω_{0i} - its natural frequency which is:

$$\omega_{0i} = \sqrt{\frac{c_i}{m_i}}\ , \ \text{rad/s}$$

Then for the overall damping ratio of the two shock absorbers from the front axle of the vehicle can be written:

$$\beta_1 = 2.\psi_{01}m_1\omega_{01} = 2.0{,}2.3400.7{,}0 = 9520 N.s/m$$

and for the overall resistance coefficient of the two dampers at the rear axle:

$$\beta_2 = 2.\psi_{02}m_2\omega_{02} = 2.0{,}2.4100.7{,}5 = 12300 N.s/m$$

When driving on uneven roads it is recommended that the damping aperiodic coefficient to be $\psi_{0i} = 0{,}6 \div 0{,}8$ [10, 11 and 12] and the critical aperiodic coefficient at which oscillations are aperiodic is $\psi_{0i} = 1$ [13] and is determined by the formulas:

$$\beta_1 = 2.m_1\omega_{01} = 2.3400.7{,}0 = 47600 N.s/m$$

$$\beta_2 = 2.m_2\omega_{02} = 2.4100.7{,}5 = 61500 N.s/m$$

3. Numerical Simulation

The numerical simulation is performed in program field of MATLAB. The values of the parameters are given in Table 1.

Table 1: *Parameters of the dynamic model*

Parameter	Symbol	Value	Dimension
Truck mass	m_0	5000	kg
Load mass	m_l	2500	kg
Full mass	m	7500	kg
Moment of inertia	J	33582	kg.m^2
Front suspension stiffness	c_1	166600	N/m
Rear suspension stiffness	c_2	230625	N/m
Front suspension damping	b_1	var	Ns/m
Rear suspension damping	b_1	var	Ns/m
Distance	h_{ec}	1,6	m
Distance	h	1,0	m
Distance	a	2,32	m
Distance	b	1,93	m
Static load – front axle	G_{w1}	33,355	kN
Static load – rear axle	G_{w2}	40,221	kN

The total braking force is described by trapezoidal law, analogous to the theoretical law of variation of braking deceleration in the braking diagram. Fig. 2 shows the variation of the front suspension deflection with low $\beta_1 = 9520 N.s/m$ and high damping ratio $\beta_1 = 30000 N.s/m$ of the front shock absorbers. The braking deceleration increase from 0 to 8 m/s in time interval 0,1 s.

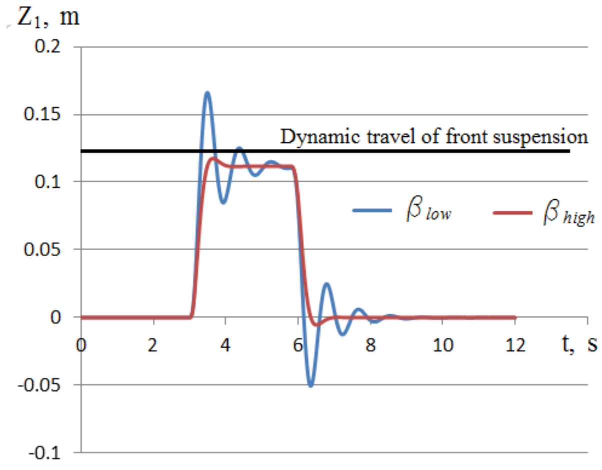

Fig. 2 *Deflection of front suspension with low(blue line) and high damping (red line)*

When the truck brakes with maximum brake deceleration there is a risk of a slam or jounce in the suspension. If the damping ratio of the shock absorbers increased at the beginning of braking (similar to the brake assistant control logic) the slams can be avoided (Fig. 2).

4. Conclusion

The considered model enables to study the vehicle pitch angle and jounces in suspension when vehicle brake. There are possibilities for reducing the suspension jounce if appropriate logic is used to control the semi-active suspension. With fast performance system can achieve good results and improve the comfort and reliability of the vehicle.

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