Mathematical Modeling of the Under Dressor Forward Axle, Movement and Kinematic of the Suspension Bracket of the Skeleton, Engine, Cabin and Seat Which of the Wheel Tractor and Numerical Determination of the Operator'S Seat Vibration

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Abstract. In article is given mathematically modeling of fluctuation, movement and kinematic forward axle, suspension bracket of a skeleton, engine, cabin and seat of the wheel tractor. It is given an analytical type of the equation, the describing vertical fluctuations and longitudinally angular fluctuations of a cabin and the main parts of the tractor. Using this model will be found as far as distances are displaced the main parts of the tractor at vibration. The equations of fluctuation of linear system with one degree of freedom of the under cabin damper which that shock-absorber of the tractor are described.

1. Introduction

To ensure the comfort of the driver and passengers, protect the transported goods from vibration and good grip of the wheels with the road. At low vibrational frequencies (nII≈ 30 min-1), a person's susceptibility to vibrations and their speed are 80% lower than when using previously rigid suspensions with frequencies of 100 min-1 .

The decrease in suspension stiffness is limited by the ability to move the wheels from the middle position during the rebound t1+t2, as well as by a large roll of the body at bends with soft suspension. To reduce the roll, stabilizers should be used. Measurements carried out on a large number of passenger car models showed that Daimler - Benz, Renault, Auto Union and Chrysler cars considered to be comfortable (and equipped with steel elastic elements) have a vibration frequency of the sprung masses of the front suspension nIIV=55…65 min-1, and the total travel of the suspension is about 200 mm.

The creation of an effective vibration protection system for a wheeled tractor driver is an urgent task [1,2] aimed at improving the driver's work and ensuring safety. Seat acceleration and noise levels in the driver's cab of a wheeled vehicle are strictly regulated by international standards. Therefore, global manufacturers of wheeled vehicles are constantly improving vibration protection systems and introducing new noise-absorbing materials in the design of driver's cabins. Thus, studies [3] have determined the frequency ranges of natural vibrations of various human organs, and also established the distribution of vibration frequencies over the structural elements of a wheeled tractor.

2. Research Methods

The material contains mathematical models of wheeled vehicles with similar kinematic connections, but they are theoretically investigated at a new level. In this study, methods of mathematical modeling of each part of the tractor, such as the front axle, suspension of the skeleton, engine, cab and operator's seat, were applied to the sprouting process. Using the analysis method, we studied the oscillations of a linear system with one degree of freedom under the cabin shock absorber of a tractor

Currently, the problem of vibration protection of the driver is being solved by creating effective suspension systems for the driver's seat and cab. In modern wheeled tractors, to improve the vibration protection of the driver of a wheeled tractor, a method of secondary suspension of the rms acceleration value (suspension of the driver's cab) is widely used.

Based on the analysis of the results of modeling the oscillations of the masses of a wheeled tractor, a cushioning scheme for the driver's seat is proposed, including an additional damping element.

The simulation results showed that this suspension system significantly reduces the root-mean-square acceleration on the driver's seat when the tractor is operating on field surfaces with low-frequency exposure from 1 to 8 Hz, as well as on paved roads with high-frequency exposure from 10 to 30 Hz.

3. Simulation results

The stiffness of the elastic elements, which is included in all calculations, leads to only one wheel. For the body, the equation of the oscillation frequency [in min^{-1}] without taking into account damping, as well as the influence of the axis and tires has the form / 3.4 / :

$$n_{IIV,h} = 9.55\sqrt{C_{2V,h}/m_{2V,h}} \quad (1)$$

and for oscillations of unsprung masses [in min^{-1}] associated with one wheel:

$$n_{IV,h} = 9.55\sqrt{\left(K_F C_{IV,h} + C_{2V,h}\right)/m_{IV,h}} \quad (2)$$

where $C_{IV,h}$- is the stiffness of the tire, K_F- is the coefficient of increase in stiffness, which takes into account its growth with increasing speed. After substituting the mass of the m_I axis and the stiffness C_I the tires of the equation for the frequency of body oscillation in min^{-1} will take the following form:

$$n_{II} = 9.55\sqrt{C_2/[m_2 + C_2 / K_F C_I (m_1 + m_2)]} \quad (3)$$

From this equation with omitted indices v and h for the front and rear axles, respectively, it can be noted that the oscillation frequency in comparison with that calculated using equation (1) will be the smaller, the greater the ratio c_2/c_1. This will be the case with rigid body suspension (large value c_2) and soft tires (small value c_1).

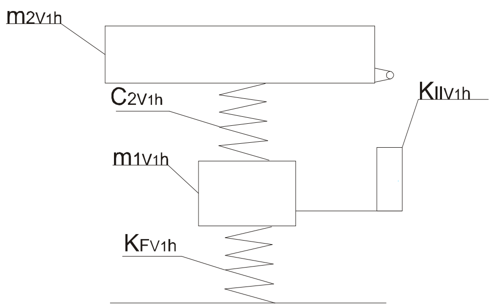

Fig. 1.

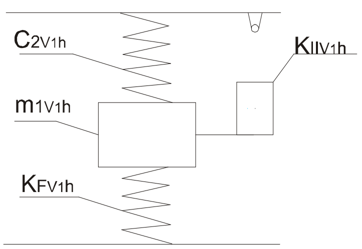

Fig. 2.

The natural vibration frequency of the wheel $n_{IV,h}$ is a function of the mass of the axis $m_{IV,h}$, the stiffness of the suspension $C_{2V,h}$ (рис.1, 2), the stiffness of the tire $C_{1V,h}$ and the coefficient of resistance of the shock absorber $K_{IIV,h}$ reduced to the wheel. Additionally influence movement speed, which is accounted for by the coefficient K_F.

The body vibration frequency $n_{IIV,h}$ depends not only on the sprung mass $m_{2V,h}$ and suspension stiffness $C_{2V,h}$ but also on the mass of the $m_{1V,h}$, axis, tire stiffness $C_{1V,h}$, K_F coefficient and shock absorber resistance coefficient $K_{IIV,h}$.

The oscillation frequency of sprung masses in mini tractors with steel elastic elements is n_{IIV} = 55 ... 80 min^{-1} for the front axle and n_{IIh} = 68 ... 100 min^{-1} for the rear axle.

To ensure comfort, one should strive for what is achievable even for suspensions of relatively light cars (“Renault - 4”, “Renault - 6”). However, for the rear suspension, this is only possible if the mobile device is equipped with a body level control system.

The reduction of dynamic loads in the elastic ties of the tractor can be achieved by the installation of damping shock absorbers and its mounting in rational places.

We investigate the oscillations of a linear system with one degree of freedom of the under-head damper of a tractor

In the simplest case, the depreciable object and the shock absorbers supporting it form the system /1/, schematically shown in Figure 3.

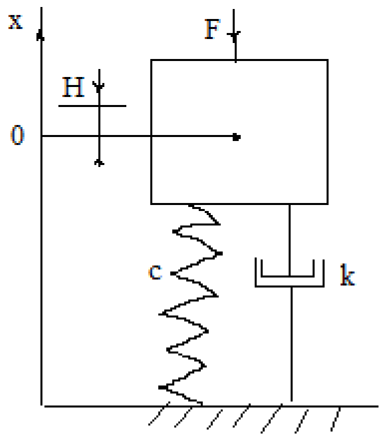

Fig. 3. Oscillating system with an inertial element m, stiffness c, damping coefficient k, loading the force disturbance F cab

The differential equation of mass motion under the influence of force for a damped system

$$m\ddot{x} + k + cx = F(t),$$

where is the damping coefficient, $c = G/\delta_{st}$ - is the static shock absorber stiffness coefficient, δ_{st} - is the static deformation (deflection) of the shock absorber, mm .G- cab weight.

The solution of the equation (in the case of free oscillation without damping k = 0) allows us to obtain the value of the natural resonance frequency

$$\omega_0 = \sqrt{\frac{c}{m}} \text{or } f_0 = \frac{1}{2\pi}\sqrt{\frac{c}{m}} = \frac{1}{2\pi}\sqrt{\frac{g}{\delta_{st}}} = \frac{1}{2\pi}\sqrt{\frac{cg}{G}}.$$

In the case of damping, the solution to the equation depends on the relative attenuation coefficient

$$D = \frac{k}{k_{kr}},$$

where $k_{kr} = 2\sqrt{cm}$ - is the critical damping coefficient.

Given this, we can determine the resonant frequency of the shock absorber with damping as

$$\omega_k = \sqrt{\frac{\omega_0^2 - k^2}{4m^2}}.$$

The quality of depreciation is characterized by a vibration isolation coefficient η which shows how many times the amplitude of the object’s movement x is greater or less than the amplitude A of the movement of the base on which the depreciation object is located:

$$\eta = x/A.$$

Another indicator characterizing the quality of depreciation is the coefficient of effectiveness of vibration isolation

$$E = (1 - \eta)100\%.$$

Note the following shock absorber operating conditions:

-when the frequency ratio is $\gamma = f/f_0$close to unity, i.e. the condition is met $f_0 \approx f$, then resonance occurs;

-when $f < f_0$vibration isolation is absent$(\eta = 1)$.

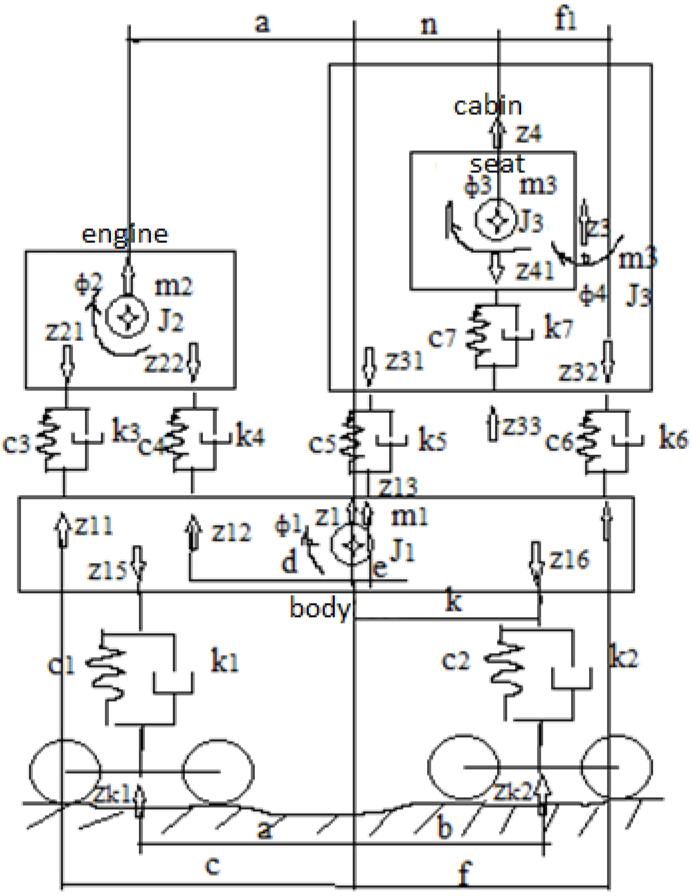

Fig. 4. D dynamic equivalent design of the main parts of the tractor

Mathematical modeling of the vibration, displacement and kinematics of the suspension of the skeleton, engine, cab and tractor seat (Fig. 4). The energy load and tractor speed are constantly increasing, which leads to an increase in the dynamic load of the main parts and an increase in the level of oscillations generated by them. Vibration loads adversely affect the components and components of the tractor, the environment and cabs with the operator.

The main parts of the tractor are described created at the first stage, a flat, at the next stage, a spatial model of suspension systems /2,5/

a) equations describing vertical vibrations

$$\begin{cases} m_1\ddot{z}_1 - [2c_1(z_{15} - z_{k1}) + 2k_1(\dot{z}_{15} - \dot{z}_{k1})] + [2c_2(z_{16} - z_{k2}) + 2k_2(\dot{z}_{16} - \dot{z}_{k2})] - \\ [2c_3(z_{21} - z_{11}) + 2k_3(\dot{z}_{21} - \dot{z}_{11})] - [2c_4(z_{22} - z_{12}) + 2k_4(\dot{z}_{22} - \dot{z}_{12})] + \\ [2c_5(z_{31} - z_{13}) + 2k_5(\dot{z}_{31} - \dot{z}_{13})] - [2c_6(z_{32} - z_{14}) + 2k_6(\dot{z}_{32} - \dot{z}_{14})] = m_1 g, \\ m_2\ddot{z}_2 + [2c_3(z_{21} - z_{11}) + 2k_3(\dot{z}_{21} - \dot{z}_{11})] + [2c_4(z_{22} - z_{12}) + 2k_4(\dot{z}_{22} - \dot{z}_{12})] = m_2 g, \\ m_3\ddot{z}_3 + [2c_5(z_{31} - z_{13}) + 2k_5(\dot{z}_{31} - \dot{z}_{13})] + [2c_6(z_{32} - z_{14}) + 2k_6(\dot{z}_{32} - \dot{z}_{14})] - \\ [2c_7(z_{41} - z_{32}) + 2k_7(\dot{z}_{41} - \dot{z}_{32})] = m_3 g, \\ m_4\ddot{z}_4 + [2c_7(z_{41} - z_{32}) + 2k_7(\dot{z}_{41} - \dot{z}_{32})] = m_4 g, \end{cases}$$

b)equations describing longitudinal-angular oscillations

$$\begin{cases} J_1\ddot{\varphi}_1 + [2c_1(z_{15} - z_{k1}) + 2k_1(\dot{z}_{15} - \dot{z}_{k1})]a - [2c_2(z_{16} - z_{k2}) + 2k_2(\dot{z}_{16} - \dot{z}_{k2})] \\ b - [2c_3(z_{21} - z_{11}) + 2k_3(\dot{z}_{21} - \dot{z}_{11})]c - [2c_4(z_{22} - z_{12}) + 2k_4(\dot{z}_{22} - \dot{z}_{12})]d + \\ [2c_5(z_{31} - z_{13}) + 2k_5(\dot{z}_{31} - \dot{z}_{13})]e - [2c_6(z_{32} - z_{14}) + 2k_6(\dot{z}_{32} - \dot{z}_{14})]f = 0, \\ J_2\ddot{\varphi}_2 + [2c_3(z_{21} - z_{11}) + 2k_3(\dot{z}_{21} - \dot{z}_{11})](c - n) + [2c_4(z_{22} - z_{12}) + 2k_4(\dot{z}_{22} - \dot{z}_{12})] \\ (n - d) = 0, \\ J_3\ddot{\varphi}_3 + [2c_5(z_{31} - z_{13}) + 2k_5(\dot{z}_{31} - \dot{z}_{13})](h - e) + [2c_6(z_{32} - z_{14}) + 2k_6(\dot{z}_{32} - \dot{z}_{14})] \\ (f - h) = 0, \\ J_4\ddot{\varphi}_4 + [2c_7(z_{41} - z_{33}) + 2k_7(\dot{z}_{41} - \dot{z}_{33})](f - f_1) = 0, \end{cases} \varphi_1$$

$$\begin{cases} z_{11} = z_1 - \varphi_1 c, \\ z_{12} = z_1 - \varphi_1 d, \\ z_{13} = z_1 + \varphi_1 e, \\ z_{14} = z_1 + \varphi_1 f, \\ z_{15} = z_1 - \varphi_1 a, \\ z_{17} = z_1 - \varphi_1 b, \end{cases} \begin{cases} z_{21} = z_2 + \varphi_2(c - n), \\ z_{22} = z_2 - \varphi_2(n - d), \\ z_{31} = z_3 + \varphi_3(h - e), \\ z_{32} = z_3 - \varphi_3(f - h), \\ z_{33} = z_3 + \varphi_3(h - k), \\ dz_{11} = dz_1 - d\varphi_1 c, \\ dz_{12} = dz_1 - d\varphi_1 d, \end{cases} \begin{cases} dz_{13} = dz_1 + d\varphi_1 e, \\ dz_{14} = dz_1 + d\varphi_1 f, \\ dz_{15} = dz_1 + d\varphi_1 a, \\ dz_{16} = dz_1 - d\varphi_1 b, \\ dz_{21} = dz_2 + d\varphi_2(c - n), \\ dz_{22} = dz_2 - d\varphi_2(n - d), \\ dz_{31} = dz_3 + d\varphi_3(h - e), \\ dz_{32} = dz_3 - d\varphi_3(f - h), \\ dz_{33} = dz_3 + d\varphi_3(h - k). \end{cases}$$

Here m_i – i- th sprung mass; J_i-moment of inertia of their-th sprung mass; z_i, $\dot{z}_i$, $\ddot{z}_i$- accordingly, vertical displacement, speed, acceleration, i-th sprung mass; dz_i - corresponding speeds of vertical movements ; $\varphi_i, \dot{\varphi}_i, \ddot{\varphi}_i$ – angular displacement, speed, acceleration, i-th sprung mass; c_i i- th connection stiffness; k_i i - th damping coefficient; a,b,c,d,e,f,h,k,n- geometric parameters of the elements; g- acceleration of gravity.

4. Discussions

Based on a flat model, using an algorithmic program, it is possible to obtain the oscillatory systems of the skeleton, cab and seat, as well as the graphs of movements, speeds and accelerations of these parts of the tractor.

Based on the conducted analytical study of the wheel tractor seat suspension, we consider it appropriate to use the proposed model when designing active and semi-active elastic elements for the seat suspension, which will reduce the harmful effects of transport vibration on the operator and increase the operational speed of the wheeled tractor.

Improving the traction and dynamic properties of the tractor can be achieved by optimizing and reducing vibrations of the main parts of the wheeled tractor.

So, the equations describing the suspension and vertical oscillations of the equation, also describing the longitudinal-angular oscillations of a much hierarchical level allow us to find the oscillation values of each individual part of the tractor.

Now we are conducting a numerical experiment to determine the vertical acceleration of the operator's cabin.

A numerical experiment will be carried out in stationary spatial oscillations of the dynamic system of a wheeled tractor, which is interpreted between four main components, was used to optimize the structure and vibration protection parameters of a 110 kW wheeled tractor. Its design is made according to an integral layout, has a frame, a cabin, seats and involves the use of dependent wheel suspension.

When calculating the optimization of the parameters of the vibration protection system, as a function of the target, we chose the implementation of the recommended and permissible parameters of the rms values on the seat of a person - the operator in the first four octave frequency ranges / 9,10 /. The level of these accelerations ensures that the operator can work for an 8-hour working day without damaging his health and reducing the set productivity.

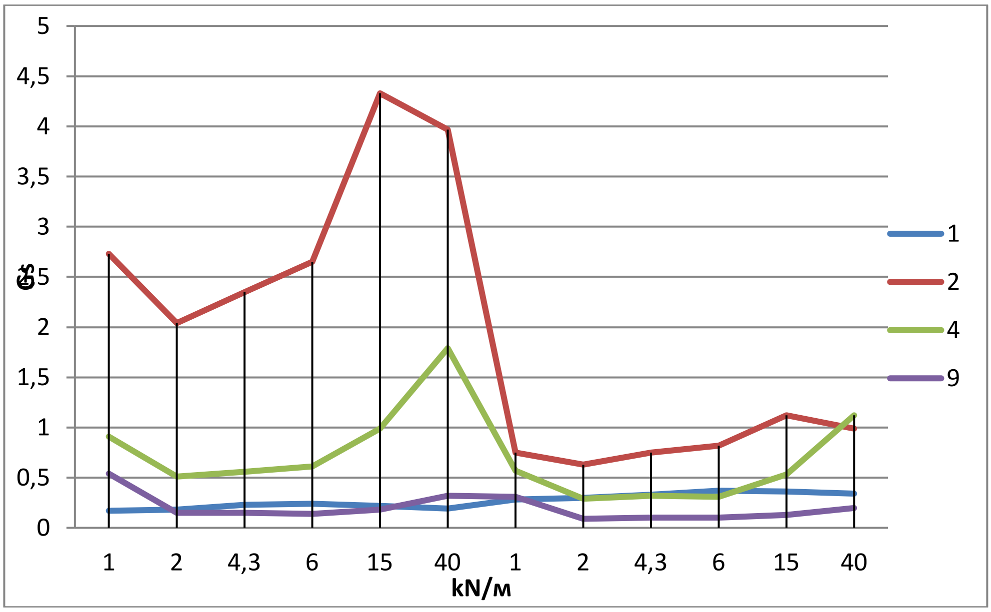

Figure 5 - Vertical acceleration on the driver's seat, k = 0.8 kNs / m; frequency range, values at values 1,2,4,9, Hz; seat suspension rigidity: the initial part of the wheel suspension is absent and the next part is an adjustable wheel suspension with optimal parameters, from 1 to 40, kN / m

Studies of the potential possibilities of seat cushioning to reduce vibration were carried out for variants with locked suspensions of the tractor frame, tires and cab suspension on rubber shock absorbers. The graphs in Fig. 2 indicate that the seat with the lowest stiffness of 2 kN / m (corresponding to the operator's natural frequency of vertical oscillations of 0.88 Hz) has the greatest efficiency. In the most dangerous for a person (in vertical vibration), the effect of this option in comparison with a hard seat is 1.33 and 1.79 times, respectively. For the AU-31.00000 seat (rigidity 4.1 kN / m), the efficiency is less pronounced - 1.15 and 1.59 times. However, the level of vertical acceleration on the operator's seat is still very high and exceeds the standard for a seat with a rigidity of 2.1 kN / m by 2.5 times and for a seat with a little damped mechanism by 3 times.
From the calculation, it can be said that the use of seats with very low stiffness on an unsprung tractor cannot significantly improve the working conditions of the operator.

5. Conclusions

The above equations suspension front axle of the tractor will find relevant parameters oscillations.

Using the definition of the oscillation of a linear system with one degree of freedom of the under-head damper of the tractor, we can find the vibration isolation efficiency coefficient, which notifies the presence or absence of an operator’s cab selection.

The analytical form of the equations describing vertical vibrations and longitudinal-angular vibrations of the cab and the main parts of the tractor is given. By numerically solving these equations, we can find the angular and vertical displacements and the angular velocities of the cab and the main parts of the tractor. This means that how far the distances are shifted to the main parts during selection will be found.

A numerical experiment revealed that the use in a seat with very low stiffness on an unsprung tractor cannot significantly improve the working conditions of the operator, only with the use of a combined and relaxation dampers can arrogant vibrations of the operator's seat be reduced.

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