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Numerical simulation of tasks of vertical viscoelastic oscillation of suspension systems of frame, engine, cabin and seat

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Abstract. The analysis of the structures of the springing systems of modern vehicles showed that polymer composites are used in vehicles for which the decrease in curb weight is a critical indicator. The use of springs (suspensions) from polymer composites makes it possible to reduce the weight of the elastic element and increase durability while improving the smoothness of movement, reducing noise, and increasing traffic safety. When replacing the steel suspension with a polymer composite suspension, the mass of unsprung parts of the car decreases, and the economic performance of wheeled vehicles improves.
The article discusses the problem of vertical viscoelastic oscillations of the suspension systems of the skeleton, engine, cabin, and seat. When considering the suspension's rheological properties, the Boltzmann-Volterra integral model is used. The Koltunov-Rzhanitsyn nucleus is used as a nucleus, which has weakly singular features of the Abel type. The system of integro-differential equations describing the studied process is solved by a method based on the use of quadrature formulas. The effect of the parameters of the hereditary-deformable properties of the suspension on the vibration shape is shown.

1 Introduction

The power and speed of tractors constantly increase, which leads to an increase in the dynamic load on the parts of the chassis and transmissions and an increase in the level of vibrations generated by them. Vibration loads adversely affect tractor assemblies, parts, environment, and operator [1-3].

The degree of versatility of modern tractors is constantly increasing. Each modern tractor must be adapted to more diverse traction, transport, and other works, so their designs are becoming more complex. To increase labor productivity, the power ratio of tractors is constantly increasing, and the speed of movement of tractor units is increasing. But this inevitably leads to an increase in the dynamic load of the parts of the chassis and transmissions and an increase in the level of vibrations generated simultaneously, as a result of which the vibration load of the operator's workplace increases [4].

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Dynamic and vibration loads adversely affect the units and parts of the tractor itself, the environment, and the operator. In the transmission and chassis, they cause constant violations of the spatial location and laws of movement of the parts, resulting in fatigue damage accumulating in their material. Engine vibrations on its suspension lead to deterioration of fuel efficiency indicators. The vibration of the undercarriage parts hurts the structure of the treated soil and acts oppressingly on the cultivated crops. To a particularly important extent, vibration loads impact the operator's operability and health. The constant long-term action of vibrations leads to increased fatigue and an increase in the number of control errors, which ultimately affects the performance of the tractor unit [5].

Many tasks of assessing the dynamic properties of technical objects under the action of vibration loads on them are solved when using design schemes in the form of mechanical vibrational systems with several degrees of freedom, which gives certain opportunities to assess the forms of dynamic interactions and determining the requirements for appropriate structural solutions, which are predetermined by the properties of the constituent elements [6-8].

Springs as viscoelastic elements are used, usually on independent suspensions. The most widespread are cylindrical coiled springs made of a round steel bar. Since the spring design features allow for a wider range of movements of the sprung and unsprung tractor masses, the spring suspensions can provide a better smoothness than the spring suspension.

Since the spring is a critical element of the suspension, the operation of which depends not only on comfort but also on the safety of movement, special steels and technologies are used to manufacture springs.

1.1 Statement of problem

Following the above, the work aimed at reducing the vibration load of the tractor operator's workplace is relevant. A flat model for design studies of the joint operation of the systems for springing the skeleton, engine, cabin, and operator's seat, a dynamic model, was developed, as shown in Figure 1. During its development, it was assumed that the transverse-angular vibrations of the sprung masses of the tractor during movement are significantly less significant than the vertical and longitudinal-angular ones. Therefore, the dynamic model is represented by a flat diagram. Here the mass and vertical oscillations of the skeleton, engine, cabin, and seat are $m_1,\ m_2,\ m_3,\ m_4\ and\ x_1,\ x_2,\ x_3,\ x_4$ respectively; $k_1,\ k_2,\ k_3,\ k_4,\ k_5,\ k_6,\ k_7$ are coupling stiffness coefficients and p (t) is force excited by wheel imbalances.

The solution of problems occurs using the Boltzmann-Volterra principle: the solution of the linear viscoelasticity problem can be obtained from the solution of the corresponding problem of linear elasticity theory by replacing elasticity constants in it with integral operators [9]. Then the mathematical model, that is, the system of integro-differential equations describing the vertical viscoelastic oscillation of sprung masses of the dynamic model as a result of the action of operational kinematic and force disturbances when the tractor moves on surfaces with different roughness profiles has the following form [10-12]:

$$\begin{cases} m_1\ddot{x}_1 + 2k_1(1-R_1^*)x_1 + 2k_2(1-R_2^*)x_1 - 2k_3(1-R_3^*)(x_2-x_1) - 2k_4(1-R_4^*)(x_2-x_1) - \\ -2k_5(1-R_5^*)(x_3-x_1) - 2k_6(1-R_6^*)(x_3-x_1) = p(t); \\ m_2\ddot{x}_2 + 2k_3(1-R_3^*)(x_2-x_1) + 2k_4(1-R_4^*)(x_2-x_1) = 0; \\ m_3\ddot{x}_3 + 2k_5(1-R_5^*)(x_3-x_1) + 2k_6(1-R_6^*)(x_3-x_1) - 2k_7(1-R_7^*)(x_4-x_3) = 0; \\ m_4\ddot{x}_4 + 2k_7(1-R_7^*)(x_4-x_3) = 0; \end{cases}$$

$R_j^* w(t) = \int_0^t R_j(t-\tau) w(\tau) d\tau -$ integral operators with relaxation cores $R_j(t) = \varepsilon_j t^{\alpha_j - 1} e^{-\beta_j t}$; $j = \overline{1,7}$.

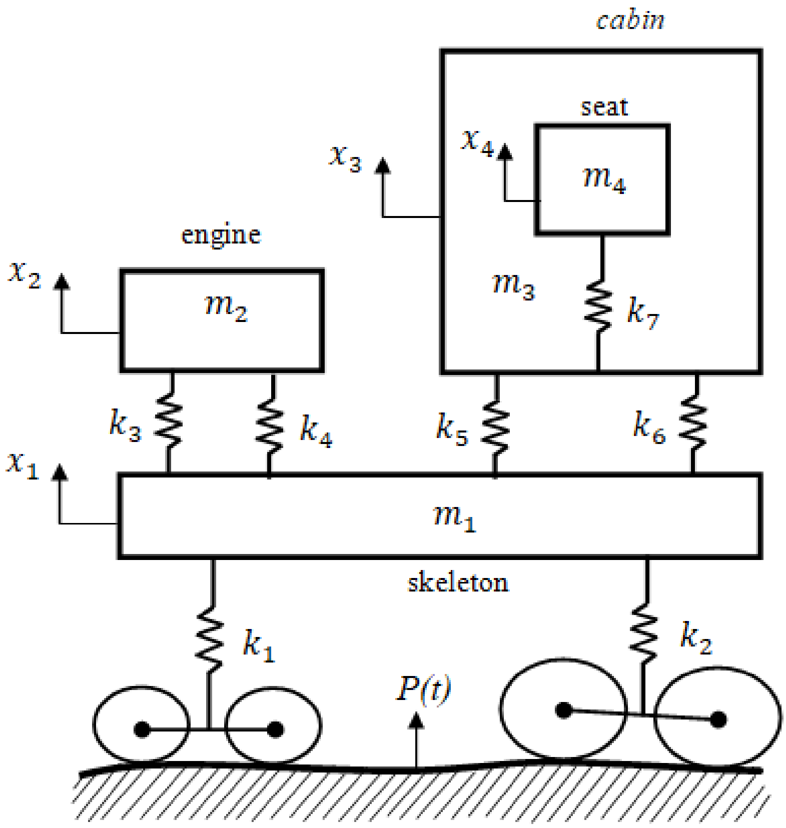

Fig.1. Dynamic model of the suspension of the skeleton, engine, cabin, and seat.

We accept that $x_1(0) = \dot{x}_1(0) = x_2(0) = \dot{x}_2(0) = x_3(0) = \dot{x}_3(0) = x_4(0) = \dot{x}_4(0) = 0$.

2 Method of solution

Entering dimensionless parameters $\frac{t}{\tau}$; $\frac{R_i}{\tau}$; $\frac{x_1}{l}$; $\frac{x_2}{l}$; $\frac{x_3}{l}$; $\frac{x_4}{l}$ and while maintaining the previous designations, we have:

$$\begin{cases} \ddot{x}_1 + c_1(1-R_1^*)x_1 + c_2(1-R_2^*)x_1 - c_3(1-R_3^*)(x_2-x_1) - c_4(1-R_4^*)(x_2-x_1) - \\ \quad -c_5(1-R_5^*)(x_3-x_1) - c_6(1-R_6^*)(x_3-x_1) = q(t); \\ \ddot{x}_2 + c_7(1-R_3^*)(x_2-x_1) + c_8(1-R_4^*)(x_2-x_1) = 0; \\ \ddot{x}_3 + c_9(1-R_5^*)(x_3-x_1) + c_{10}(1-R_6^*)(x_3-x_1) - c_{11}(1-R_7^*)(x_4-x_3) = 0; \\ \ddot{x}_4 + c_{12}(1-R_7^*)(x_4-x_3) = 0, \end{cases} \qquad (1)$$

where $c_1 = \frac{2k_1\tau^2}{m_1}$; $c_2 = \frac{2k_2\tau^2}{m_1}$; $c_3 = \frac{2k_3\tau^2}{m_1}$; $c_4 = \frac{2k_4\tau^2}{m_1}$; $c_5 = \frac{2k_5\tau^2}{m_1}$; $c_6 =$

$\frac{2k_6\tau^2}{m_1}$; $c_7 = \frac{2k_3\tau^2}{m_2}$;
$c_8 = \frac{2k_4\tau^2}{m_2}$; $c_9 = \frac{2k_5\tau^2}{m_3}$; $c_{10} = \frac{2k_6\tau^2}{m_3}$; $c_{11} = \frac{2k_7\tau^2}{m_3}$; $c_{12} = \frac{2k_7\tau^2}{m_4}$; $q(t) = \frac{\tau^2}{l}p(t)$.

The system of integro-differential equations (1) is solved by the method based on the use of the quadrature formula [13-14]. Integrating twice by t system (1) in the interval [0; t] and assuming $t = t_n = n \cdot \Delta t$, $n = 0,1,2,3,...$ (Δt -step in time) we have:

$$\begin{cases} x_{1n} + c_1 \int_0^{t_n} G_1(t_n - s)x_1(s)\,ds + c_2 \int_0^{t_n} G_2(t_n - s)x_1(s)ds - \\ -c_3 \int_0^{t_n} G_3(t_n - s)[x_2(s) - x_1(s)]ds - c_4 \int_0^{t_n} G_4(t_n - s)[x_2(s) - x_1(s)]ds - \\ -c_5 \int_0^{t_n} G_5(t_n - s)[x_3(s) - x_1(s)]ds - c_6 \int_0^{t_n} G_6(t_n - s)[x_3(s) - x_1(s)]ds = \int_0^{t_n} (t_n - s)\,q(s)ds; \\ x_{2n} + c_7 \int_0^{t_n} G_1(t_n - s)[x_2(s) - x_1(s)]\,ds + c_8 \int_0^{t_n} G_4(t_n - s)[x_2(s) - x_1(s)]ds = 0; \\ x_{3n} + c_9 \int_0^{t_n} G_5(t_n - s)[x_3(s) - x_1(s)]\,ds + c_{10} \int_0^{t_n} G_6(t_n - s)[x_3(s) - x_1(s)]ds - \\ -c_{11} \int_0^{t_n} G_7(t_n - s)[x_4(s) - x_3(s)]ds = 0; \\ x_{4n} + c_{12} \int_0^{t_n} G_7(t_n - s)[x_4(s) - x_3(s)]\,ds = 0; \end{cases} \quad (2)$$

here

$$G_j(t_n - s) = t_n - s - \int_0^{t_n - s} (t_n - s - \tau)R_j(\tau)\,d\tau; \quad j = \overline{1,7}.$$

By replacing integrals with quadrature trapezoidal formulas in the system (2), we have the following recompetitive relations to determine the vertical movements of the $x_{1n} = x_1(t_n)$; $x_{2n} = x_2(t_n)$; $x_{3n} = x_3(t_n)$ where $x_{4n} = x_4(t_n)$ corresponding to the frame, engine, cabin, and seat weight.

$$\begin{cases} x_{1n} = -c_1 \sum_{i=0}^{n-1} A_i G_1(t_n - t_i) x_{1i} - c_2 \sum_{i=0}^{n-1} A_i G_2(t_n - t_i) x_{1i} + \\ +c_3 \sum_{i=0}^{n-1} A_i G_3(t_n - t_i)(x_{2i} - x_{1i}) + c_4 \sum_{i=0}^{n-1} A_i G_4(t_n - t_i)(x_{2i} - x_{1i}) + \\ +c_5 \sum_{i=0}^{n-1} A_i G_5(t_n - t_i)(x_{3i} - x_{1i}) + c_6 \sum_{i=0}^{n-1} A_i G_6(t_n - t_i)(x_{3i} - x_{1i}) + \sum_{i=0}^{n-1} A_i (t_n - t_i) q(t_i); \\ x_{2n} = -c_7 \sum_{i=0}^{n-1} A_i G_1(t_n - t_i)(x_{2i} - x_{1i}) - c_8 \sum_{i=0}^{n-1} A_i G_4(t_n - t_i)(x_{2i} - x_{1i}); \\ x_{3n} = -c_9 \sum_{i=0}^{n-1} A_i G_5(t_n - t_i)(x_{3i} - x_{1i}) - c_{10} \sum_{i=0}^{n-1} A_i G_6(t_n - t_i)(x_{3i} - x_{1i}) + \\ +c_{11} \sum_{i=0}^{n-1} A_i G_7(t_n - t_i)(x_{4i} - x_{3i}); \\ x_{4n} = -c_{12} \sum_{i=0}^{n-1} A_i G_7(t_n - t_i)(x_{4i} - x_{3i}). \end{cases} \quad (3)$$

3 Results and Discussion

For conducting a computational experiment, a computer program has been developed that reflects the results in the form of graphs. The following initial data were used in the calculation: $c_1 = 182{,}19$; $c_2 = 182{,}19$; $c_3 = 87{,}01$; $c_4 = 174{,}02$; $c_5 = 208{,}33$; $c_6 = 208{,}33$; $c_7 = 139{,}22$; $c_8 = 278{,}44$; $c_9 = c_{10} = 500$; $c_{11} = 306{,}2$; $c_{12} = 612{,}4$; $\alpha_1 = \alpha_2 = \alpha_3 = \alpha_4 = \alpha_5 = \alpha_6 = \alpha_7 = 0{,}25$; $\beta_1 = \beta_2 = \beta_3 = \beta_4 = \beta_5 = \beta_6 = \beta_7 = 0{,}05$; $\varepsilon_1 = \varepsilon_2 = \varepsilon_3 = \varepsilon_4 = \varepsilon_5 = \varepsilon_6 = \varepsilon_7 = 0{,}05$. The force excited by wheel imbalances is given as $q(t) = 8e^{-0{,}02t} sin6\pi t$

Figures 2-5 show graphs of the time changes in the vertical movements of the skeleton, cab, engine, and seat, where solid and dashed lines indicate the elastic and viscoelastic tasks, respectively. It can be seen from the graph that taking into account the viscosity of the spring leads to a decrease in the amplitude of the skeleton, cab, engine, and seat from the position of static equilibrium. Due to the viscosity of the spring, the frequency of oscillation decreases. Over time, the influence of viscoelastic properties of the suspension on amplitudes and frequencies is significant.

The obtained numerical results show that the suspensions from polymer composites have the lowest possible specific weight indicators while providing the required stiffness indicators. As a result, the loaded weight of the tractor decreases, and the decrease in the weight of the suspension guide also reduces the weight of the sprung and unsprung parts, which positively affects the smoothness indicators.

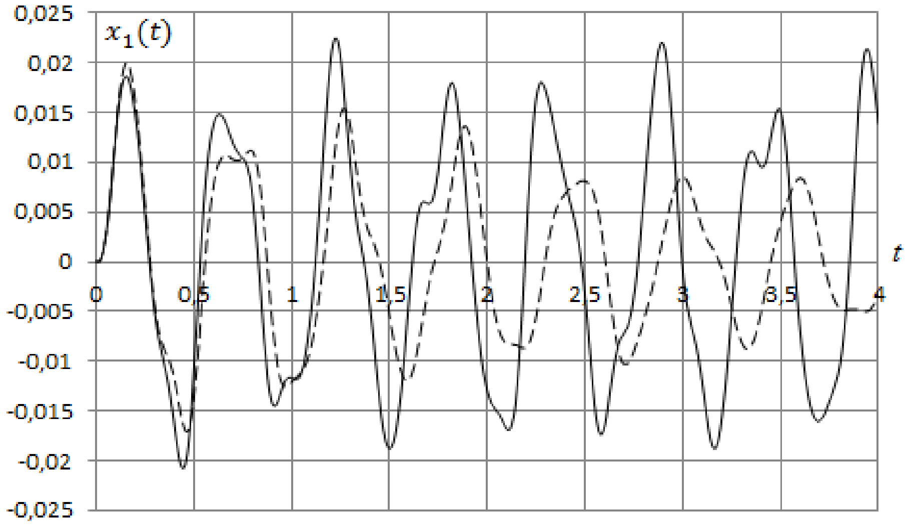

Fig.2. The oscillations shape of skeleton.

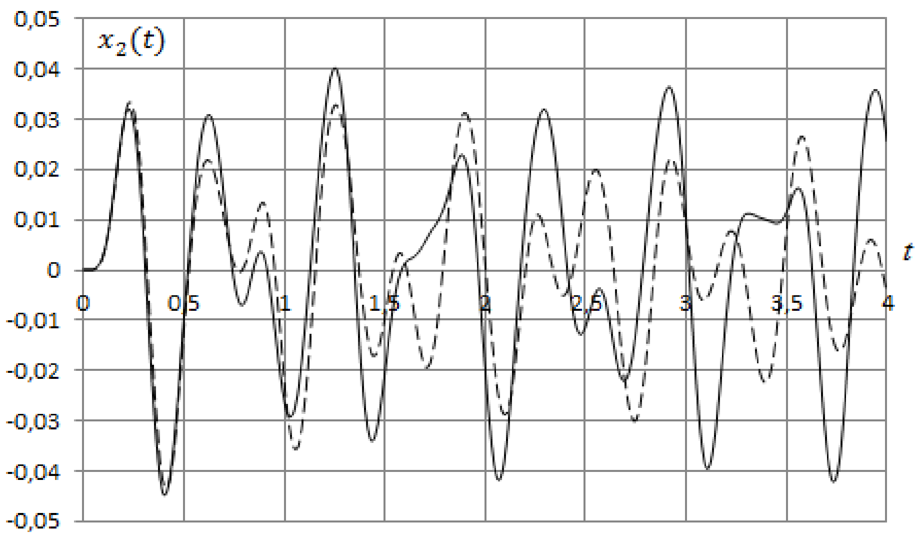

Fig.3. Oscillations shape of cabin.

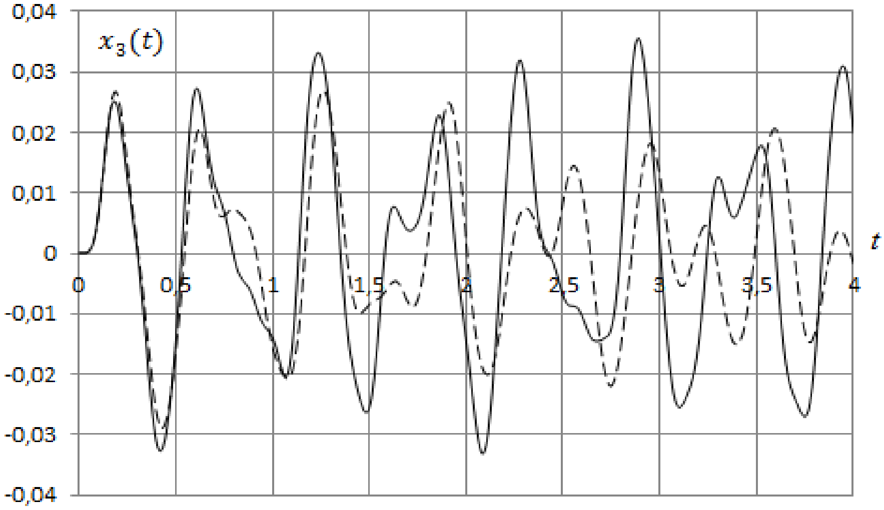

Fig.4. Oscillations shape of engine.

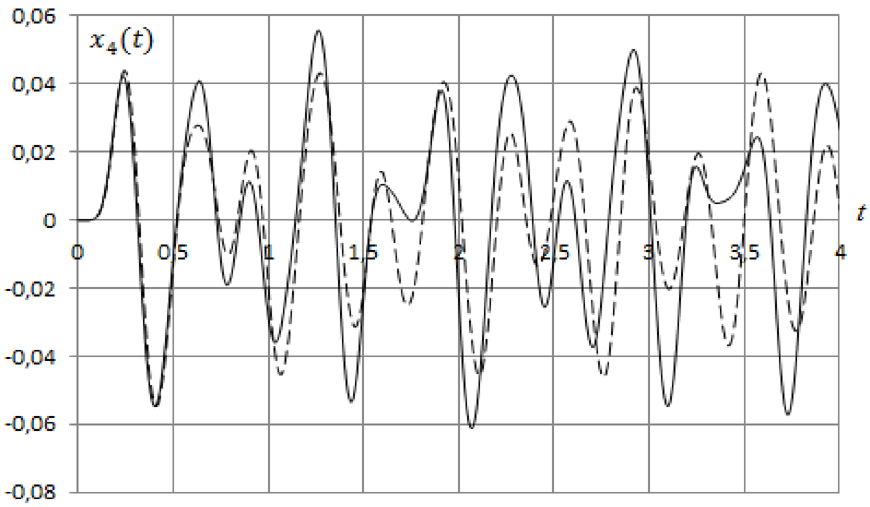

Fig.5. Oscillations shape of seat.

4 Conclusions

Methods have been developed that allow predicting viscoelastic properties at the design stage and simulating the mechanical behavior of a suspension made of a composition of the polymer and structural materials, taking into account the design features and energy characteristics of the materials.

The use of schemes capable of solving the problem in question in a closed form or algorithms of type (3) is of great interest. The obtained results allow us to conclude that it is advisable to consider the hereditary-deformable properties of the suspension to reduce the amplitude of oscillations of the skeleton, engine, cabin, and seat. Based on the calculation studies performed using the developed mathematical models, it is possible to optimize the characteristics of viscoelastic suspension elements, in which the level of dynamic impact at the operator's workplace is reduced.

Based on the results of the studies performed for practical use, a set of programs for a personal computer was created, designed to study the viscoelastic properties of suspensions at the design stage without creating expensive single samples. The developed mathematical model, which is the basis of a complex of programs, differ in versatility and make it possible to predict the properties of suspension systems of various shapes, consisting of a composition of several materials.

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