Results of computer modeling of a modular independent tubular suspension of a logging truck

V O Nikonov¹, V I Posmetev and V V Posmetev
Department of production, repair and operation of machines, Voronezh State University of Forestry and Technologies named after G F Morozov, 8 Timiryazev Street, 394087, Voronezh, Russian Federation
¹E-mail: prem@vglta.vrn.ru

Abstract. The necessity of increasing the efficiency of timber trucks by improving the designs of their suspensions has been substantiated. The main disadvantages of traditional suspension structures used in modern logging trucks have been described. A promising design of a modular independent tubular torsion bar suspension for a logging truck has been proposed. The aim of the research consisted of two stages. At the first stage, a mathematical model for the functioning of the proposed torsion bar suspension has been developed. At the second stage, a computer program for a preliminary assessment of the effectiveness of the functioning of this suspension (when driving on an insufficiently equipped forest road) has been made. Time dependences of the angular positions of suspension arm fork and ends of the tubular torsion bars were obtained. The regularities of the influence of the movement speed, average height of irregularities, coefficient of dry friction on the relative spread of vertical loads and suspension indicators of vibration dispersions were revealed.

1. Introduction
Improving the efficiency of logging (timber) truck using is one of the most important scientific and technical problems. In the general case, the efficiency of a logging truck is understood as a generalized characteristic of its quality. It is determined by the ratio of the results from its use to the total costs of developing, manufacturing and maintaining a timber truck in working condition. The operation of a logging truck is characterized by difficult climatic conditions, as well as its low adaptation to traffic on insufficiently equipped logging roads. In addition, structural elements of timber truck are the subjects of high dynamic loads. Their impact leads to breakdowns; and idle time in the repair and waiting of a timber truck it makes up to 30% of the working time. All this adversely affects its service life before decommissioning, which becomes significantly less than the established standard period. In this regard, a promising direction for increasing the level of operational properties of timber trucks is finding the optimal suspension parameters, as well as development of promising suspension structures. The use of such suspensions reduces the amount of costs associated with the transportation of timber to the consumer, which will significantly affect the efficiency of logging enterprises as a whole. At present Russian and foreign scientists offer various designs of different types of suspension for trucks. In addition, there are many results of theoretical and practical studies showing the main indicators of the proposed suspensions effectiveness.

Nikonov V O [1] carried out a functional study of the proposed design of a timber truck hydropneumatic suspension on the basis of the developed simulation model. It showed that such a
suspension is highly effective in overcoming irregularities and obstacles on the supporting surface of a timber road. In addition, the proposed suspension provides more favorable amplitude-frequency response in comparison with the traditional design of a spring-damping suspension. Anubi O M [2] proposed and investigated a promising progressive-rate suspension. The scientific results (on the basis of theoretical and experimental study) showed that this suspension has better elastic characteristics in comparison with the traditional one. Ding F [3] proposed a promising hydraulic suspension design developed for a three-axle truck. The method of simulation modeling was used in the research to study the functioning of the proposed suspension. It had better damping properties in comparison with the traditional design. This makes it possible to smooth out vibrations when overcoming various obstacles more effectively. Moheyeldein M M [4], based on the method of computer simulation, showed a comparison between the characteristics of tire grip and driving performance of trucks equipped with leaf spring and air suspension. It was found that a truck equipped with an air suspension, in comparison with a truck with a leaf spring suspension, had a 10% improved ride and a 20% reduction in dynamic loads on tires. These advantages make it possible to increase the driver’s comfort when driving a truck. Gokul P [5] proposed a promising air suspension design, equipped with a linear-quadratic control regulator. It improved controllability of a truck moving on insufficiently equipped roads. Again, the research was based on the method of simulation modeling. Vilas A M study [6] provided a detailed description of the design and operation of a modern torsion bar suspension of a truck. A comparison of several torsion shafts made of different materials was shown using the method of computer simulation. It was found that a torsion shaft made of carbon composite material had 3 times lower torsion stress and 20% less deformation (in comparison with a traditional torsion shaft). Posmetev V I [7] carried out a study on the basis of mathematical and simulation modeling of multi-shaft torsion bar suspension functioning. The use of such a compact suspension under normal operating conditions was not inferior to a single-shaft torsion bar suspension of a vehicle. Nabaglo T [8] proposed an improved model of a hyperbolic torsional spring used in a vehicle suspension, based on the finite element method. This model took into account the interaction forces between the spring-loaded suspension arms. It became possible to simulate its operation even in the case of a large torsion angle. The proposed suspension design was compared with the traditional torsion bar for statistical suspension deflection and range of its mobility. Tavares R [9] developed a simulation model to assess the driving comfort conditions of a vehicle equipped with an active torsion bar suspension. The model also enabled as to assess energy consumption and potential energy collection when driving on uneven roads. It was possible to get 50-60 W of energy from shock absorbers under comfortable driving conditions of a vehicle at a speed of 50 km/h. Geonea I [10] gave the results of the study carried out on the basis of numerical modeling and full-scale bench tests for fatigue. Characteristics of angular deformations and stiffness of torsion springs used in vehicle suspensions were determined. The work of Zhou S-T [11] presented a study of sensitivity characteristics effect on the parameters of a double-wishbone torsion bar suspension of an electric vehicle. The basic size of the torsion bar was optimized. It enabled to increase the rigidity of the double-wishbone torsion bar suspension, reduce shock loads and increase the comfort of driving on the vehicle.

The analysis of scientific works of Russian and foreign scientists showed that dependent suspensions of various types were used in the chassis of timber trucks. Such suspensions have the following disadvantages. The first is a decrease in lateral resistance to rollover due to the high center of gravity of the logging vehicle. The second is high location of the loading platform. The third is the lack of the possibility of useful use of the underbody and interwheel space in a logging vehicle. The fourth is an unjustifiably reduced ground clearance. Fifth is a set of structural parts of the undercarriage assemblies protruding in the lower part of the timber truck. Such details contribute to a decrease in the reliability of a timber truck and deterioration in its passability in driving conditions on insufficiently equipped logging roads. In this regard, a promising direction is development of modular independent suspensions. It enables to significantly increase the efficiency of a timber truck. The design eliminates the identified shortcomings and implements a modular method of timber truck completing.

Previously, the authors analyzed numerous traditional and promising truck suspension designs using various available literary sources, as well as patent bases. Also, the results of many studies aimed at
determining the dependencies affecting the performance indicators and optimal parameters of such suspensions were studied. The analysis of these works showed that compact modular independent tubular torsion suspensions for timber trucks have not been developed yet. It simultaneously eliminated the disadvantages discussed above, as well as performed the following three functions. The first was to use the inter-wheel space more efficiently. The second was to increase the safety of the timber truck by lowering its center of gravity. The third was high protection against external adverse influences when logging vehicle moved on an insufficiently equipped logging road.

Also, the influence of the change in time of the angular positions of the lever fork and the ends of the tubular torsion bars was not taken into account in the study of the main characteristics of torsion suspensions in the studies considered. The absence of such dependencies will not provide comparable torsion stiffness coefficients of tubular torsion bars, which are characterized by the dependence of the wall thickness of the tubular torsion bars on the radius. In addition, the considered works did not study the effect of the speed of movement of a timber truck, the average height of irregularities and the coefficient of dry friction on the spread of loads and indicators of energy dissipation in a tubular torsion bar suspension. Knowledge of these dependencies will make it possible to determine the values of the studied indicators at which the most effective damping of oscillations is provided.

The aim of the work consisted of two stages. The first stage was development of a mathematical model for the functioning of a modular independent tubular torsion bar suspension of a timber truck. The second stage was creation of a computer program for preliminary assessment of the performance indicators of the proposed suspension design. Also it was necessary to find the dependence of time changes of the angular positions of the fork and ends of the tubular torsion bars. It was made taking into account the analyzed results of scientific research of other authors. In addition, it was required to determine the impact of changes in the movement speed of timber truck, average height of irregularities and coefficient of dry friction on the indicators of vibration damping efficiency.

2. Methods and materials
To eliminate the identified shortcomings, the authors previously developed a promising design of a modular independent tubular torsion bar suspension for a timber truck based on the analysis of existing structures of torsion suspensions used in vehicles. It included tubular torsion bars 1-5, spacing tubes 6-9 and connecting rings 10 (figure 1).

Modern methods of modeling make it possible to reproduce mechanisms with high physical adequacy and are widely used in mechanical engineering. Therefore, in this work, a mathematical model of movement of a timber truck with a modular independent tubular torsion bar suspension has been developed. It was necessary to reproduce a coordinated operation of coaxial tubular torsion bars, movement of the wheel of the modular independent suspension of the timber truck, as well as interaction of the wheel of the modular independent suspension with the supporting surface of the model timber road. The developed mathematical model was based on the methods of classical mechanics. The model consisted of three subsystems. The first was a subsystem of the translational vertical motion of wheel of a modular independent torsion bar suspension connected to it by a lever. The second was a subsystem of coaxial rotational motion of torsion bar suspension bodies. The third was a subsystem for specifying the geometric configuration of irregularities on the supporting surface of a timber road. It was assumed that a wheel of a modular independent suspension moved only in vertical direction along the vertical Z axis (figure 2).

Three forces acted on the wheel: gravity, from the side of the supporting surface of the logging road, and from the side of the suspension arm. Therefore, the equation of wheel motion of a modular independent torsion bar suspension was as follows:

$$m_w \frac{d^2 z_u}{dt^2} = -m_w g + \frac{M_w}{L_s \cos \phi_i} + \begin{cases} -c_v (z_u - K - z_d) - d_f \left(v_z - \frac{dz_u}{dt}\right), & z_u - R_w < z_d; \\ 0, & z_u - R_w < z_d, \end{cases}$$

(1)
Figure 1. Modular independent tubular torsion bar suspension of a logging truck.

Figure 2. Representation of the wheel of a modular independent tubular torsion bar suspension (timber truck) in the mathematical model.

where $m_w$ – wheel mass of modular independent suspension; $z_w$ – vertical coordinate of the wheel axis of the tubular torsion bar suspension; $t$ – time; $g$ – acceleration of gravity; $M_w$ – moment on the arm of the tubular torsion bar suspension relative to its axis; $L_a$ – arm length of independent tubular torsion bar suspension (distance $BC$); $\varphi_c$ – angle of arm deflection of the tubular suspension from the horizontal
axis; $c_r$ – lateral rate; $d_f$ – damping factor; $R_w$ – wheel radius; $z_A$ – vertical coordinate of the point of the support road surface interacting with the lower point of the tubular torsion bar suspension wheel at the moment in time $t$.

The given differential equation of the second order was solved numerically in a system with equations for the dynamics of the rotational motion of torsion suspension bodies. The tubular torsion bar suspension was represented in the model by a set of four rotary bodies. The moments of inertia of the tubular torsions $J_1, \ldots, J_4$ were get to this bodies (figure 3). For ease of perception, the tubular torsion bars in the diagram were arranged sequentially in a row, while in reality they were coaxially inside each other. Geometrically conditional bodies were the ends of tubular torsions, to which the moment of inertia of the nearest halves of the tubular torsions was get. Adjacent rotary bodies interacted with each other by viscous-elastic interaction (interaction between the torsion stiffness coefficients $c_i$ and viscosity $d_i$). Another type of interaction was dry and viscous friction, acting between bodies through one body $(J_1 \cdot J_2, J_2 \cdot J_4, J_3 \cdot J_4)$. Interaction was given by the torsional viscosity $d_i$ and dry friction coefficients $\mu$. All bodies in the model rotated in the same direction.

The interaction of torsions was considered backlash-free. Therefore, for rotary bodies $I$ and $j$, the arising moment of elastic-viscous force $M_{ij}$ was determined by the formula:

$$M_{ij} = c_{ij} (\varphi_i - \varphi_j) + d_{ij} (\omega_i - \omega_j),$$

where $c_{ij}$ and $d_{ij}$ – the coefficients of torsional stiffness and damping at the interaction of rotation suspension bodies; $\varphi_i$ and $\varphi_j$ – angular positions of rotary bodies; $\omega_i$ and $\omega_j$ – angular velocities of rotary bodies.

![Figure 3. Representation of the design scheme in the mathematical model (used to formulate a system of equations for the dynamics of rotational motion of five rotary bodies in a tubular torsion bar suspension.](image)

Torsional stiffness of a tubular torsion bar depended on the torsion bar diameter, tube wall thickness, torsion bar length and material. The biggest difference between the five tubular torsion bars was their
diameter. In this case, wall thickness of the tubes could be either the same or decrease with an increase in the diameter, so that the torsional stiffnesses of the torsion bars were approximately the same. Using the developed model, various variants of the torsional stiffness changes $c_{i,i+1}$ with an increase in the radius of the tubular torsion bars $R_i$ were tested: no dependence $c_{i,i+1} = \text{const}(R_i)$, linear dependence $c_{i,i+1} = kR_i$, quadratic dependence $c_{i,i+1} = kR_i^2$ (figure 4).

For the first option (when the wall thickness of the tubular torsion bar increased with increasing radius) the angular displacement with load application was the same for adjacent pairs of rotary bodies due to the same torsional stiffnesses (figure 4a). If the wall thickness of the tubular torsion bar was the same, then the dependence of $c_{i,i+1}$ on $R_i$ was approximately quadratic one. When a load was applied, angular displacement was mainly experienced by tubular torsion bars of a small radius. The outer tubular torsion bars practically did not twist relative to each other (figure 4c). Therefore, it was necessary to decrease wall thickness with an increase in the radius of the tubular torsion bar along the linear or quadratic dependence. It was necessary to make all tubular torsion bars to work in the same type.

![Figure 4](image_url)

**Figure 4.** The nature of tubular torsion bars twisting with different dependences of torsional stiffness $c_{i,i+1}$ on the radius $R_i$: a – the stiffnesses were the same $R_i$; b – linear dependence $c_{i,i+1} = kR_i$; c – quadratic dependence $c_{i,i+1} = kR_i^2$.

The equations of rotational motion dynamics of five rotary bodies can be written as follows.

$$
\begin{align*}
J_1 \frac{d^2 \phi_1}{dt^2} &= M_{1w}; \\
J_2 \frac{d^2 \phi_2}{dt^2} &= M_{1w} - M_{12}; \\
J_3 \frac{d^2 \phi_3}{dt^2} &= M_{12} - M_{23}; \\
J_4 \frac{d^2 \phi_4}{dt^2} &= M_{23} - M_{34}; \\
J_5 \frac{d^2 \phi_5}{dt^2} &= M_{34} - M_{45}.
\end{align*}
$$

where $J_n, J_1 \ldots J_5$ – moments of inertia of the considered rotary bodies, $\phi_w, \phi_1 \ldots \phi_5$ – angular positions of the considered rotary bodies, $W$ – a suspension arm with a wheel as a body of rotation; $F$ – point of rigid termination of the inner shaft of the suspension; $M_{ij}$ – moments of contact of rotary bodies $I$ and $j$, which can be described as follows:

$$
\begin{align*}
M_{1w} &= F_w L_a \cos \phi_w + c_{w1} (\phi_w - \phi_1) + d_{w1} (\omega_w - \omega_1), \\
M_{12} &= c_{12} (\phi_1 - \phi_2) + d_{12} (\omega_1 - \omega_2), \\
M_{23} &= c_{23} (\phi_2 - \phi_3) + d_{23} (\omega_2 - \omega_3), \\
M_{34} &= c_{34} (\phi_3 - \phi_4) + d_{34} (\omega_3 - \omega_4), \\
M_{45} &= c_{45} (\phi_4 - \phi_5) + d_{45} (\omega_4 - 0).
\end{align*}
$$
where $\omega_W, \omega_1 \ldots \omega_4$ – angular velocities of the considered rotary bodies; $F_W$ – force acting from the wheel on the suspension arm; $L_a$ – suspension arm length.

Thus, the dynamic behavior of a wheel module with a tubular torsion bar suspension was described by a system of differential equations (1)-(8).

Realistic functioning of the timber truck suspension was essential to be reproduced in the developed model. It was necessary to create an intensive movement of the wheel in the vertical direction, corresponding to the actual relief of the supporting surface of the timber road. The accepted assumptions of the model must be taking into account. So, the model used the function of a random uneven surface $z(x)$.

The principle of creating random irregularities in the model was based on the real operating conditions of the logging truck. A significant part of the logging vehicle path fell on insufficiently equipped access logging roads, characterized by the presence of obstacles, irregularities and a large number of defects. Gaussian functions were used to reproduce a wide range of highs and depressions of the supporting surface when creating the relief of the logging surface in the model.

In this regard, the perturbing function of the wheel of an independent tubular torsion bar suspension was described by the superposition of Gaussian functions. The main parameters of Gaussian peaks (length and height) were randomly set in the model and could vary over a wide range (the length of obstacles varies from 0.1 to 0.5 m, and the length of hills varies from 2 to 5 m) (figure 5).

**Figure 5.** An example of the supporting surface of a timber road simulated when the wheel of a modular independent tubular torsion bar suspension moves.

The supporting surface of the timber road was represented as a function of the surface height from the coordinate of the corresponding contact point $z(x)$. This function was characterized by a superposition of Gaussian peak with the parameters of obstacle position $x_i$, its height $H_i$, and the width $\sigma_i$:

$$z(x) = \sum_{i=1}^{N_0} H_i \exp \left\{ -\frac{(x-x_i)^2}{\sigma_i^2} \right\}, \quad (9)$$

where $H_i$ – height of the obstacle passed by the wheel; $N_0$ – number of obstacles in the control area; $x_i$ – coordinate characterizing the center point of the obstacle; $\sigma_i$ – obstacle width.

Wheel motion of a modular independent tubular torsion bar suspension was described by a system of differential equations of the second order. In most cases, these systems of differential equations did not have an analytical solution due to the presence of complex perturbing functions. And in this case, the function of the topography of the timber road surface was presented in the form of a linear combination of analytically non-integrable Gaussian functions. It did not allow finding an analytical solution. Therefore, such systems of equations of motion were usually solved by numerical methods. A fairly universal second-order Runge-Kutta method was chosen for the numerical solution of differential equations. In particular, for a rotary body $i$, the numerical integration of the equations of motion was performed by the formula:

$$\begin{align*}
\phi_{i+1} &= \phi_i + \omega_i \cdot \Delta t + \frac{M_{ii}}{J_i} \left( \frac{\Delta t}{2} \right)^2,
\omega_{i+1} &= \omega_i + \frac{M_{ii}}{m} \cdot \Delta t,
\end{align*} \quad (10)$$
where \( \phi_i \) and \( \omega_i \) – angular position and angular velocity of the body with the moment of inertia \( J_i \) at the previous step of integration over time \( \tau \); \( \phi_{i+1} \) and \( \omega_{i+1} \) – the same, at the next step of integration over time \( \tau + 1 \). Similarly, the equations of motion of other rotary bodies and wheel motion of independent tubular torsion bar suspension in the vertical direction were integrated.

Table-defined functions of the time dependence of the vertical coordinate (wheel center) and the angular positions of the rotary bodies were found after solving the system of differential equations. Further, efficiency indicators of the tubular torsion bar suspension of a timber truck were calculated according to the table-set functions. Thus, a mathematical model of the tubular torsion bar suspension of a timber truck was developed. It enabled to study the influence of suspension parameters, supporting surface of the timber road and working conditions on the efficiency of its functioning.

Efficiency of tubular torsion bar suspension was estimated by the following indicators: \( \Delta F_s/F_z \) – the ratio of the average spread of force \( \Delta F_s \) (force acted from the side of the suspension on the car body) to the weight of the car body \( F_z \) falling to the suspension:

\[
\frac{\Delta F_s}{F_z} = \frac{\Delta t}{t_2 - t_1} \sum_{t_1}^{t_2} \left[ F_s' - F_z \right],
\]

where \( \Delta t \) – number and value of integration step of the motion equations over time; \( t_1 \) and \( t_2 \) – the time of the beginning and the end of the computer experiment on the movement of the investigated mechanical system on an uneven supporting surface. Square brackets rounded a number to an integer value. Straight brackets indicated the absolute value of a number.

The lower \( \Delta F_s/F_z \) index, the more effective the suspension was in terms of reducing vertical vibrations of the car body (in increasing car smoothness).

Indicator \( \eta \) (dissipation of unfavorable vibration energy in suspension) was the inverse of the suspension efficiency. It was calculated by the formula:

\[
\eta = \frac{\Delta t}{t_2 - t_1} \sum_{t_1}^{t_2} \left[ \frac{A_{Fr}'}{E_s' + A_{Fr}'} \right],
\]

where \( A_{Fr} \) – work of forces of dry and viscous friction between the bodies of tubular torsion bar suspension; \( E_s \) – potential energy of elastic interaction accumulated in the suspension due to torsional deformation of tubular torsion bars. The higher the \( \eta \) value, the more efficient the suspension was, the more the vehicle's vertical vibration energy was dissipated in the suspension. At the same time, the higher the \( \eta \) index, the higher the heat generation in the torsion bar suspension was (due to friction).

It was usually necessary to dissipate the energy of vertical vibrations in the suspension as much as possible (unlike many classes of mechanisms, where it was necessary to achieve high efficiency, in the suspension, on the contrary). Subsequent research consisted in changing the most important input parameters and studying (on the basis of computer experiments) the influence of parameters on performance indicators \( \Delta F_s/F_z \) and \( \eta \).

3. Results and discussion
A computer program "Program for modeling a modular independent tubular torsion bar suspension of a timber truck" was developed for further research of the obtained mathematical model. It enabled to simulate functioning of a modular independent tubular torsion bar suspension during the movement of a timber truck along the supporting surface of a timber road in order to assess its efficiency (smoothness, dynamic loading, and friction losses).

The developed model had sufficiently high physical and geometric adequacy and enabled to change a large number of suspension parameters. The first group of input variables included the following parameters of the tubular torsion bar suspension: \( L_a \) was the length of the arm of the tubular torsion bar suspension; \( \phi_a \) was the angle of the equilibrium position of the suspension arm; \( c_i \) and \( d_i \) were coefficients of torsional and damping stiffness of the interaction of contacting conventional rotary bodies of revolution (ends of tubular torsion bars); \( \mu \) – dry friction coefficients (reduced to polar coordinates) of the interaction of contacting conventional rotary bodies; \( J_i \) were the moments of inertia of rotary...
bodies. The second group of variables included the parameters of the support surface relief: \(n\) was the linear density of defects, obstacles and irregularities; \(h_{\text{min}}\) and \(h_{\text{max}}\) were the minimum and maximum values of defect obstacles and irregularities heights. They were the boundaries of the range for selection according to the uniform law of probability distribution. \(\sigma_{\text{min}}\) and \(\sigma_{\text{max}}\) were the minimum and maximum lengths of irregularities. The third group of input variables included conditions the operation of the timber truck: \(v\) was horizontal speed of the timber truck; \(F_z\) was the wheel load of the modular tubular torsion bar suspension of the timber truck.

Three series of computer experiments were carried out in the process of theoretical research. Each series consisted of 5-7 computer experiments with different values of the parameter investigated in this series. In each experiment, a model timber truck was driven on an uneven supporting surface of a timber road at a speed of 40 km/h for 10 seconds. The dependences of the angles of rotary bodies \(\varphi_i\) on time \(t\) were recorded in files during the computer experiment (figure 6).

Figure 6. Influence of the time \(t\) of the arm-fork angular positions \(\varphi_W\) on the ends of the tubular torsion bars \(\varphi_1 \ldots \varphi_4\).

Transient processes occurred in the initial 0.5 seconds, and the tubular torsion bar suspension came to a state of dynamic equilibrium. It can be seen from the output of the \(\varphi(t)\) graphs to saturation (figure 6). The magnitude of the peaks in the graphs increased approximately linearly from body 4 to body 1 and the "W" arm. Linearity was due to the inverse quadratic dependence of the tubular torsion wall thickness on the radius, which provided comparable torsional stiffness coefficients of tubular torsions.

Efficiency indicators calculated by averaging over the time of computer experiment were \(\Delta F_s/F_z = 0.12\) and \(\eta = 0.22\) (for the most typical (basic) parameter values). The indicators were interpreted as follows. When a timber truck was moving on an insufficiently equipped forest road surface, tubular torsion bar suspension provided a low average vertical acceleration of \(-0.12\) g. 22% of the vibration energy of this suspension was dissipated as heat in the friction pairs between the individual tubular torsion bars.

Frequency of irregularities of supporting surface on the timber road (mechanical system "wheel – suspension – body of the timber truck") depended on the movement speed of the logging truck. A series of six computer experiments was carried out to study the functioning of the developed modular independent tubular torsion bar suspension in wide ranges of speed variation. The speed of the timber truck was increased with a step of 10 km/h in the range from 10-60 km/h during the computer experiments. At the same time, the average height of irregularities was the same and amounted to 60 mm, which corresponded to an insufficiently equipped logging road.

Both relative load spread \(\Delta F_s/F_z\) and energy dissipation index in the tubular torsion bar suspension \(\eta\) initially grew slowly with an increase in the speed of a timber truck from 10 to 30 km/h (figure 7). Both indicators increased approximately linearly at the speeds of 30-50 km/h. The growth of the graphs slowed down with a further increase in the speed \(v\) above 50 km/h. It was due to the fact that the wheel did not have time to copy the complex topography of the timber road and was not in contact with the supporting surface for an increasing proportion of the time. It reduced the transmission of vibrations to the suspension and the body of the logging vehicle.

The following conclusion can be made on the basis of the diagrams. The proposed design of the
tubular torsion bar suspension provides effective vibration damping in a wide range of travel speeds from 10 to 60 km/h on a substantially uneven forest road with an average height of irregularities of 60 mm. The load spread on the body does not exceed 0.18 from the weight. Suspension absorbs from 8 to 38% of vibration energy.

Figure 7. Influence of the movement speed of a timber truck \( v \) on the relative spread of vertical loads \( \Delta F_z/F_z \) (a) and indicator of dissipation of vibration energy in the suspension \( \eta \) (b).

The intensity of vibrations in the tubular torsion bar suspension depends on the height of the irregularities of the supporting surface of the timber road. A series of six computer experiments was performed to study the effect of the average height of irregularities \( h_i \), in which \( h_i \) was changed from 0 to 100 m with a step of 20 mm.

The spread of the vertical force on the body increases approximately according to a quadratic law with an increase in the height of the irregularities (figure 8a). The indicator of energy dissipation in the suspension increases according to a law close to linear (figure 8b). Therefore, with an increase in the height of irregularities, the relative absorption of the energy of unfavorable vibrations by the suspension improves.

Figure 8. Influence of the average height of irregularities \( h_i \) of the supporting surface of the timber road on the relative spread of vertical loads \( \Delta F_z/F_z \) (a) and suspension vibration energy dissipation \( \eta \) (b).

The \( \eta \) indicator turns out to be undefined when driving on a flat supporting surface of a timber road \((h_i = 0 \text{ mm})\), since there are no oscillations in the suspension. Therefore the first point is missing on the graph (figure 8b).

Thus, the developed tubular torsion bar suspension remains operational in a wide range of heights of irregularities in the supporting surface of the timber road. Tubular torsion bar suspension provides a spread of vertical loads on the body of no more than 0.32 of the weight even for the case of a significant average height of irregularities of 100 mm. At least 33% of the vibration energy is dissipated in the suspension. Mechanical behavior of a tubular torsion bar should substantially depend on the coefficients of dry and viscous friction between the tubular torsion bars. A series of six computer experiments was carried out in which the coefficients \( \mu \) of dry friction (simultaneously four coefficients \( \mu_{K2}, \mu_{13}, \mu_{24}, \mu_{3F} \))
were changed from 0.05 to 0.30 with a step of 0.05.

In rotary motion, the dry friction coefficient is the proportionality coefficient between the forces of dry friction forces $F_{fr.tang}$ arising in the tangential direction at the points of friction between the tubular torsion bars and the average circumferential force $N$ of the tubular torsion bars contact in the normal direction: $F_{fr.tang} = \mu N$.

![Figure 9](image)

**Figure 9.** Influence of the dry friction coefficient $\mu = \mu_i$ on the relative spread of vertical loads $\Delta F_s/F_z$ (a) and the coefficient of vibration energy dissipation in the suspension $\eta$ (b).

The indicator of energy dissipation in the suspension $\eta$ in its physical meaning is directly related to the coefficient of friction $\mu$, the graph $\eta(\mu)$ and turned out to be linear one (figure 9b). With an increase in friction coefficients, the absorption of unfavorable vibrations in the suspension improves, and spread of forces $\Delta F_s$ acting on the car body decreases (figure 9a).

On the other hand, an increase in the coefficient of friction can make it difficult for the suspension bodies to rotate relative to each other. Therefore, the optimal value of the dry friction coefficient is 0.15-0.20. Thus, (in the places of contact of rotary bodies of the tubular torsion bar suspension) it is advisable to use materials with a sufficiently high coefficient of friction to effectively damp unfavorable vibrations in the suspension. But the coefficient should be low enough so that friction does not impede the rotation of the tubular torsion bars of the suspension. The spread of force from the side of the suspension to the body of the logging vehicle does not exceed 14% of the weight of the logging vehicle falling on the suspension with an optimal friction coefficient of 0.15-0.20. Dissipation of vibration energy in the suspension is not less than 18%.

4. Conclusion

A preliminary assessment of the performance indicators for the proposed design of a modular independent tubular torsion bar suspension for a timber truck was carried out on the basis of computer modeling. The assessment made it possible to formulate the following conclusions:

- The linear dependence of the angular positions of the fork-arm and the ends of the tubular torsion bars on time is characterized by inverse square-law dependence of the wall thickness of the tubular torsion bars on the radius;
- In a wide range of travel speeds from 10 to 60 km/h on an insufficiently equipped forest road with an average height of irregularities of 60 mm, the studied suspension provides effective vibration damping;
- The developed tubular torsion bar suspension remains operational in a wide range of heights of irregularities in the supporting surface of the timber road;
- Optimum value of dry friction coefficient, provided effective damping of unfavorable vibrations in the suspension, was 0.15-0.2;
- The use of a modular independent tubular torsion bar suspension in the structure of a logging truck will have several advantages. Vehicle safety increases by lowering the center of gravity of the logging vehicle. Protection of the suspension from external adverse influences improves. Then, the location of the timber platform becomes lower and makes it possible to load the underbody and interwheel space....
with timber. Protruding parts of the chassis of the logging vehicle can be eliminated and cross-country ability and reliability of the logging vehicle will be increased.

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