Computer simulation results of modular hydropneumatic suspension of a logging vehicle

V O Nikonov*, 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, Voronezh 394087, Russian Federation

*E-mail: af_nikonovvo@vgltu.ru

Abstract. The necessity of increasing the efficiency of timber transportation to consumers by timber trucks equipped with traditional dependent suspensions has been substantiated. The analysis of articles made by foreign scientists in the field of research of characteristics of hydropneumatic suspensions of vehicles is presented. A promising design of a modular-type hydropneumatic suspension is proposed. The goal of the work is formulated to develop mathematical and simulation models for the functioning of the proposed construction of a hydropneumatic suspension to assess the influence of the main dependences of the nature of oscillations of a timber truck in time on the values of volume, pressure and temperature of air in the cavity of the pneumatic cylinder of the suspension structure. Developed and investigated mathematical and on its basis simulation models of the functioning of the hydropneumatic suspension. The dependences of the vibration frequencies of the body of a timber truck equipped with a hydropneumatic suspension on the vibration spectrum, as well as the time dependence of the volume, pressure and temperature of the air in the pneumatic cylinder when driving over a random uneven surface and single irregularities, are obtained.

1. Introduction

The territory of the Russian Federation contains 20% of all forests on the planet. According to estimates, as of 2019, the total forested area in our country was 809 million hectares, and timber reserves reached 80.7 billion m³. Currently, the timber industry of the Russian Federation is not functioning efficiently enough. This is primarily due to the low level of development and use of the existing reserves of mature forest, which, according to average estimates, is about 30% of the possible use. The reasons for this low level of utilization are the remote, difficult-to-develop location of mature forest reserves, which is complicated by the lack of road infrastructure. In addition, many Russian logging enterprises, unable to purchase modern technological equipment, are forced to use outdated technological processes of timber processing, which leads to the production of wood with a low degree of processing, which is significantly inferior in cost to wood with a high degree of processing, produced by logging enterprises of developed foreign countries.

Being the most important element of the production process of logging, affecting the final cost of products, the efficiency of the functioning and interaction of logging enterprises with timber consumers, timber transportation is of great economic importance. For transportation of harvested timber to consumers, various types of transport can be used, among which the most widespread is timber road transport. The efficiency of timber transportation by road is influenced by a large number of factors related to the design perfection of timber trucks, operating conditions, used maintenance and repair technologies,
qualifications of drivers and repair personnel, rational formation of rolling stock, etc.

Timber road transport currently used in the forestry industry, in most cases, meets the basic requirements for it by consumers. In spite of this, the timber trucks, equipped with traditional dependent suspensions, operated on low-quality forest roads, significantly worsen their operational properties. Insufficient efficiency of such suspensions, as applied to timber trucks, consists in: limited ground clearance and steering angles; in reducing reliability, cross-country ability in difficult road conditions due to the contact of protruding nodes of a timber truck with a timber road; in the impossibility of using the underframe space located between the wheels; in a significant height of the center of gravity, leading to a deterioration in the lateral stability of a loaded logging vehicle; in the absence of mechanisms to adjust the ride height; in the impossibility of using suspensions made in the form of wheel modules due to the large overall dimensions and the complexity of the design of the timber truck; in the absence of the functionality of hanging the wheels of a logging vehicle when it is moving in an empty state; in the inability to provide an individual turn of each of the wheels of the logging vehicle. Improving the operational properties of the suspension is of particular importance for the timber truck used for hauling timber in the Russian Federation. The accuracy of the calculation of the design parameters of the suspension, as well as the selection of the suspension design that is optimal for the timber truck, makes it possible to reduce the amount of money spent on the hauling of timber.

Els P S [1] based on the developed models of heat transfer between the gas and the environment, and between the damper oil and gas, an analysis of the effect of temperature change on the spring characteristics of hydropneumatic suspension systems was carried out, which showed that hydropneumatic suspension systems have a significant degree of intrinsic damping due to for heat transfer. In the article by Sujatha C [2], as a result of the study of the operation of two high-carrying capacity vehicles in different conditions, equipped with the first - spring, the second – hydropneumatic suspension, it was revealed that the best performance indicators of the movement of a vehicle with a hydropneumatic suspension are achieved at 2/3 of full load, and with a spring without load. In the study by Baldi M [3], a promising design of a hydropneumatic suspension is presented, which allows you to change the height of the vehicle by changing the pressure of the working fluid of the hydraulic drive. An experimental check of the developed mathematical model with methods for calculating the parameters of the hydropneumatic suspension has been carried out. Wu Z-C [4] considered the issue of the influence of the volume of the gas accumulator and the ambient temperature on the change in the stiffness of the hydropneumatic springs, which led to the conclusion that by changing the stiffness of the hydropneumatic spring it is possible to influence the comfortable driving conditions of the vehicle and its adaptation to the ambient temperature. Sun T [5] in his work presents the results of modeling the influence of the intensity of heavy vehicles operation on the reliability of their hydropneumatic suspensions with an active control system, which show that an active hydropneumatic suspension with a system link handling has better reliability and stable stability when using a reliable regulator in given uncertainty intervals. Van Der Westhuizen S F [6] in his work on the basis of simulation and laboratory tests performed a comparison of the characteristics of three variations of the ideal gas law and two approaches with real gas, changing during the functioning of the hydropneumatic suspension of the vehicle. Zhang P [7] in his work describes the results of a study carried out on the basis of modeling the influence of various parameters of a hydropneumatic vehicle suspension on nonlinear characteristics of stiffness and damping. The results showed that the hydropneumatic suspension has a better damping effect with a high-frequency vibration signal, the rigidity of the hydropneumatic suspension decreases with an increase in the initial air volume in the cylinder, the rigidity and damping of the hydropneumatic suspension increases with an increase in the cross-sectional area of the piston rod. Ali D and Frimpong S [8] in a joint study attempted to develop artificial intelligence models for modeling and predicting the characteristics of hydropneumatic struts in large dump trucks, which allowed in real time to improve the safety of the driver's workplace, his health and overall system efficiency. Lin D [9] in his work studied the dynamic characteristics of a hydropneumatic suspension strut of a vehicle suspension. For this, a technique was proposed for performing the experiment, which makes it possible to identify the patterns of change in friction and displacement forces depending on the working pressure in the rack. In addition,
a mathematical model has been developed for the functioning of the shock absorber, which takes into account the processes of polytropic gas, turbulent flows through the piston holes, hysteresis and nonlinear friction force, and also allows with high accuracy to predict the main investigated characteristics of the suspension strut during the model movement of the vehicle. Yesilyrt H Y [10] in his article performed, on the basis of simulation modeling, taking into account the damping coefficient, the adjustment of the design parameters of the hydropneumatic suspension of a 10 ton vehicle during its model movement. It was found that the coincidence of the cruising speed and wheelbase of the vehicle at low speeds with the natural frequency of movement of the vehicle body leads to violation of the maximum value of the amplitude of the vertical movement of the driver's seat. It was revealed that with an increase in the average speed of a vehicle on an uneven surface, the amplitude of the vertical acceleration of the driver's seat increases, which negatively affects his working conditions, well-being and fatigue. In his study, Dong X [11], based on a three-layer neural network model, considers the issue of a possible rollover of a vehicle depending on the design parameters of its hydropneumatic suspension, on the lateral and longitudinal tilt angles of the supporting surface, as well as on the algorithm of actions when using the steering. It was revealed that the results obtained using the model have a high convergence with field studies of the vehicle.

The performed analysis of works in the field of existing structures of hydropneumatic suspensions, as well as studies aimed at determining dependencies that affect the efficiency indicators and optimal parameters of such suspensions, showed that at present there are no modular-type hydropneumatic suspensions structures for timber trucks that simultaneously provide recuperation of hydraulic energy when overcoming irregularities and obstacles of a logging road, compensation for leaks of working fluid, remote control from the cab of a logging vehicle by the amount of ground clearance, increasing cross-country ability, lateral stability, reliability, lowering the center of gravity, metal consumption, using the inter-wheel space of a logging vehicle and improving working conditions for drivers. Also, when studying the main characteristics of hydropneumatic suspensions, the conditions of movement of vehicles are not sufficiently taken into account, in particular, the condition of roads, the presence of irregularities, obstacles, steep descents with ascents, as well as various defects are not taken into account. In addition, the considered works did not compare the vibration spectra of vehicles on roads of different quality with different designs of the studied hydropneumatic suspensions.

The aim of the work is to develop mathematical and, on its basis, simulation models for the functioning of the proposed design of a hydropneumatic suspension of a modular type for a timber truck. Also in the work it is necessary in the process of modeling the movement of a timber truck equipped with the proposed suspension design to reveal the dependence of the influence of the nature of the oscillations in time on the values of volume, pressure and air temperature in the cavity of the pneumatic cylinder of the hydropneumatic suspension. In addition, it is required to compare the vibration spectra of the bodies of timber trucks equipped with a traditional suspension with a hydraulic shock absorber, a proposed hydropneumatic modular suspension with and without a shock absorber.

2. Material and methods
To eliminate the shortcomings identified in the course of studying the results of studies of foreign scientists in the field of increasing the efficiency of vehicles based on the use of various hydropneumatic suspensions in their design, as well as analyzing the patterns of changes in efficiency indicators obtained by them in various operating conditions, the authors have designed a promising scheme of hydropneumatic suspension of modular type, applied to the timber truck (figure 1).

To study the advantages of a hydropneumatic suspension of a timber truck in comparison with a widespread suspension consisting of an elastic element and a shock absorber, a sufficiently adequate mathematical model of a timber truck moving along the supporting surface of a low-quality model timber road is required. To achieve this goal, a two-mass model of a hydropneumatic suspension with a perturbing function that sets the nature of the surface relief was used. This model allows you to convey the main aspects of the work of the hydropneumatic suspension of the timber truck.
Figure 1. Hydropneumatic suspension of a forest car: (a) kinematic diagram, side view; (b) circuit diagram; $V_1, V_2$ – working cavities of hydraulic and pneumatic cylinders; $P_1, P_2$ – pressure in gas and hydraulic cylinders; $l_1, l_2, L_n$ – positions of the pistons of the pneumatic and hydraulic cylinders and the movement of the hydraulic cylinder, respectively; 1 – wheel; 2 – brake drum hub; 3, 5, 6 – brackets; 4 – pneumatic hydraulic cylinder; 7 – shock absorber; 8 – propeller shaft; 9 – hydraulic motor with gear; 10 – frame; 11 – facing panel; 12 – wheel stepping hydraulic motor.

The body and wheel of a timber truck within the framework of the developed mathematical model are reduced to point masses $M$ and $m$, respectively (figure 2). Their coordinates are designated as $z_M$ and $z_m$. In this mathematical model, material points characterizing the body and wheel of a timber truck can move both along the vertical axis and along the $OZ$ axis. The wheel of a timber truck interacts with the supporting surface of a model timber road through a conventional tire, represented in accordance with the visco-elastic approximation by a spring having a stiffness $c_{tire}$ and a damper characterized by a damping coefficient $d_{air}$.

Figure 2. Calculation diagram of a hydropneumatic suspension for modeling the process of vibrations of a timber truck while driving.
The point of interaction with the reference surface of the model timber road is designated by the coordinate \(z_{\text{sup}}\). This coordinate is a function of \(z_{\text{sup}}(t)\) and changes over time. The appearance in the model of visco-elastic forces between the supporting surface of the model timber road and the center of the wheel of the timber truck contributes to the appearance of the movement of the center of mass of the wheel \(m\).

The action of the forces from the hydropneumatic suspension of the timber truck is carried out between the point of its attachment to the body and the center of mass of the wheel. These forces depend on the distance \(z_M - z_m\) between the attachment points to the body \(M\) and the center of mass of the wheel \(m\), as well as on time \(t\) and the time derivative \(dz_M(t)/dt\) from this distance \(z_M - z_m\). The shock absorber investigated in this work is connected parallel to the hydropneumatic suspension of the timber truck and is represented in the mathematical model by a conditional damper characterized by the damping coefficient \(d_{\text{dam}}\).

Using Newton's second law, the motion of the body and the wheel of a timber truck can be written with the following system of equations:

\[
\begin{align*}
    \frac{d^2 z_M}{dt^2} &= -Mg + P_{pn}(t)S_{pn} - d_{\text{dam}} \left( \frac{dz_M}{dt} - \frac{dz_m}{dt} \right), \\
    \frac{d^2 z_m}{dt^2} &= -mg - P_{pn}(t)S_{pn} + d_{\text{dam}} \left( \frac{dz_M}{dt} - \frac{dz_m}{dt} \right) + \\
    &+ c_{\text{w0}} (z_m - z_{\text{sup}}(t)) - d_{\text{w0}} \frac{dz_m}{dt},
\end{align*}
\]

where \(z_m\) – the coordinate of the center of mass of the wheel of the timber truck; \(z_M\) – the coordinate of the attachment point of the hydropneumatic suspension to the body of the logging vehicle; \(t\) – the time; \(M\) and \(m\) – the masses of the body and the wheel; \(g\) – acceleration of gravity; \(P_{pn}\) – the pressure in the pneumatic cylinder of the hydropneumatic suspension of the timber truck; \(S_{pn}\) – area of the piston of the pneumatic suspension cavity; \(c_{\text{w0}}\) – the effective coefficient of tire stiffness; \(d_{\text{w0}}\) – the coefficient of damping of a timber truck tire; \(d_{\text{dam}}\) – the damping coefficient of the shock absorber of the hydropneumatic suspension of the timber truck; \(z_{\text{sup}}\) – the coordinate of the point of interaction of the wheel of the timber truck with the supporting surface of the model timber road; \(z_{\text{eq}}\) – the equilibrium coordinate of the center of mass of the wheel of the timber truck.

The solution of the above system of differential equations is carried out by using the numerical Runge-Kutta method, which consists in discretizing at equal time steps \(\Delta t\), denoted by the variable \(\tau\) with a step interval \(\Delta t\). The calculation of the forces \(F_{mt}\) and \(F_{Mr}\) acting on the wheel and body of the timber car is carried out at each step of integration. Further, according to the given coordinates and velocities of the wheel and body of the timber truck at the current step of integration, the coordinates and velocities of the wheel and body are determined at the next step of integration:

\[
\begin{align*}
    z_{m\tau + 1} &= z_{mt} + v_{mt} \cdot \Delta t + \frac{F_{mt}}{m} \cdot \frac{(\Delta t)^2}{2}; \\
    z_{Mr + 1} &= z_{Mr} + v_{Mr} \cdot \Delta t + \frac{F_{Mr}}{M} \cdot \frac{(\Delta t)^2}{2}; \\
    v_{mt + 1} &= v_{mt} + \frac{F_{mt}}{m} \cdot \Delta t; \\
    v_{Mr + 1} &= v_{Mr} + \frac{F_{Mr}}{M} \cdot \Delta t,
\end{align*}
\]

where \(z_{mr}, z_{mt + 1}, z_{Mr + 1}\) – the coordinates of the upward movement of the wheel \(m\) and the body \(M\) of the logging vehicle at the previous \(\tau\) and subsequent steps \(\tau + 1\) of time integration; \(v_{mt}, v_{mt + 1}, v_{Mr}, v_{Mr + 1}\) – the speeds of upward movement of the wheel \(m\) and the body \(M\) of the logging vehicle at the previous \(\tau\) and subsequent steps \(\tau + 1\) of integration over time.

As a result of calculating the coordinates and speeds of the wheel \(m\) and the body \(M\) of the timber
truck at the previous and subsequent integration steps, the functions $z_m(t), z_M(t)$ are obtained, which are used to further evaluate the nature, amplitude and frequency of oscillations of the wheel and body of the timber truck.

During the operation of the hydropneumatic suspension of the timber truck, the state of the gas in the cavity of the pneumatic cylinder changes significantly. In the mathematical model, only the pneumatic subsystem of the hydropneumatic suspension is considered, since the efficiency of the suspension depends to a greater extent on it. It is advisable to take into account the hydraulic subsystem of the hydropneumatic suspension when simulating the movement of a timber truck as a whole on a substantially uneven surface, when the wheels of a timber truck make significant vertical movements and the hydraulic subsystem must ensure the flow of working fluid between several wheel suspensions.

The solution to the equation for changing the state of the gas is carried out in the same way as the equations of motion of the wheel and body presented above. The following dependences of the description of the gas state are presented in finite differences to combine with a mathematical model of the mechanical suspension subsystem.

At the initial step of time integration $\tau_0 = 0$, the air pressure in the pneumatic cylinder is taken equal to the operating pressure $P_0$, and the temperature, respectively, equal to the ambient temperature $T_0$.

\[
P_{pn}^0 = P_0; \quad T_{pn}^0 = T_0; \quad (3)
\]

The initial amount of gas $\nu_{pn}^0$ in the cavity of the pneumatic cylinder, determined on the basis of the ideal gas state equation, depends on the maximum volume $V_{pn}^0$ of the cavity of the pneumatic cylinder, and also takes into account the universal gas constant $R$ in the calculation:

\[
\nu_{pn}^0 = \frac{P_0^0 V_{pn}^0}{RT_{pn}^0}; \quad (5)
\]

The algorithm of the actions performed from the beginning to the end of the numerical integration process is carried out at each time step $\tau_i$ according to the following sequence:

1. The current position of the piston $x_{pn}^\tau$ in the pneumatic cylinder of the hydropneumatic suspension is established based on the results of mathematical modeling of the mechanical subsystem.
2. Further, according to the set position of the piston $x_{pn}^\tau$ in the pneumatic cylinder of the hydropneumatic suspension and the known value of the diameter $D_{pn}$ of the pneumatic cylinder, the current volume $V_{pn}^\tau$ of its cavity is determined:

\[
V_{pn}^\tau = x_{pn}^\tau \pi \frac{D_{pn}}{4}; \quad (6)
\]

3. Then, using the equation of the adiabatic approximation for air, the change in gas temperature in the pneumatic cylinder is calculated at the previous $\tau - 1$ and current $\tau$ steps of integration over time:

\[
T_{pn}^\tau = T_{pn}^{\tau - 1} \left( \frac{V_{pn}^\tau}{V_{pn}^{\tau - 1}} \right)^{\frac{\gamma}{\gamma - 1}}; \quad (7)
\]

4. Then, using the Newton-Richman law, the change in gas temperature at each step of numerical integration is calculated, which occurs in the process of heat exchange of gas with the surrounding space through the elements of the pneumatic cylinder:

\[
T_{pn}^\tau = T_{pn}^\tau - \alpha_{pn} \left( T_{pn}^\tau - T_0 \right) \Delta \tau, \quad (8)
\]

where $\alpha_{pn}$ – the heat transfer coefficient.
5. Following this, using the equation of state of an ideal gas, the current gas pressure \( P_{pn}^{t} \) in the cavity of the pneumatic cylinder is determined from the known values of temperature \( T_{pn}^{t} \) and volume \( V_{pn}^{t} \):

\[
P_{pn}^{t} = V_{pn}^{t} \frac{T_{pn}^{t}}{V_{pn}^{t}}. \tag{9}
\]

6. The calculation of gas leakage from the pneumatic cylinder when its pressure in the cavity of the pneumatic cylinder of the hydropneumatic suspension exceeds atmospheric pressure, taking into account the throttling coefficient \( d_{thr} \), is carried out according to the following formula:

\[
V_{pn}^{t} = V_{pn}^{t-1} - d_{thr} \sqrt{P_{pn}^{t} - P_{atm}^{t}}, \tag{10}
\]

7. The difference in gas temperature with a change in the amount of substance in the process of air leaks from the pneumatic cavity of the hydropneumatic suspension is described by the following relationship:

\[
T_{pn}^{t} = T_{pn}^{t-1} + \left( T_{0} - T_{pn}^{t-1} \right) \frac{V_{pn}^{t-1} - V_{pn}^{t}}{V_{pn}^{t-1}}. \tag{11}
\]

The described algorithm of actions, which includes seven stages, is repeated many times during computer experiments. In the process of data processing at any step of integration, the gas pressure in the cavity of the pneumatic cylinder \( P_{pn}(t) \) makes it possible to determine the force acting between the wheel and the body of the timber truck. To study the influence of vibrations of a timber truck with a hydropneumatic shock absorber on the performance indicators of the suspension, the function of the relief of the supporting surface of the model timber road \( z(x) \) was set.

The article considered the movement of a timber truck over two types of irregularities. A single trapezoidal roughness was given by the formula:

\[
z(t) = \begin{cases} 
0, & \text{if } t < t_{un} \text{ or } t > t_{un} + \frac{B_{un} - B_{fr}}{v} + 2t_{fr}; \\
(t - t_{un})H_{un}, & \text{if } t \geq t_{un} \text{ and } t < t_{un} + t_{fr}; \\
H_{un}, & \text{if } t \geq t_{un} + t_{fr} \text{ and } t < t_{un} + t_{fr} + \frac{B_{un}}{v}; \\
H_{un} - \left( t - \left( t_{un} + t_{fr} + \frac{B_{un}}{v} \right) \right)H_{un}, & \text{if } t \geq t_{un} + t_{fr} + \frac{B_{un}}{v} \text{ and } t < t_{un} + 2t_{fr} + \frac{B_{un}}{v},
\end{cases} \tag{12}
\]

where \( t_{un} \) – the time of the beginning of the contact of the wheel with the trapezoidal unevenness of the supporting surface of the timber road; \( B_{un} \) – the width of the unevenness of the supporting surface; \( H_{un} \) – maximum height of unevenness of the supporting surface; \( t_{fr} \) – the time during which the height of the unevenness of the supporting surface increases from 0 to \( H_{un} \).

A random uneven support surface on a timber road was given by the superposition of a set of Gaussian peaks:

\[
z(t) = \sum_{i=1}^{N_{gp}} H_{i} \exp \left( -\frac{(v \cdot t - x_{i})^2}{2\sigma_{i}^2} \right), \tag{13}
\]

where \( N_{gp} \) – the number of Gaussian peaks set on a section of a timber road, length \( L_{x} \); \( H_{i}, \sigma_{i}, x_{i} \) – height, width and center coordinate of the \( i \)-th roughness.

A detailed study of the influence of vibrations of a timber truck in the vertical direction was carried out according to the amplitude-frequency characteristics, in particular, from the vibration spectra \( A_{i}(f) \), using the Fourier transform of the function \( z_{a}(t) \):
\[ A_z(f) = k_{nor} \left( \int_0^t z_M(t) \sin(2\pi f t) \, dt \right)^2 + \left( \int_0^t z_M(t) \cos(2\pi f t) \, dt \right)^2, \]  

(14)

where \( f \) – the vibration frequency of the timber truck; \( k_{nor} \) – coefficient characterizing the degree of fluctuations of the timber truck. Since the function \( z_M(t) \) is given in a table, the integrals in the formula for the spectrum are calculated numerically – the rectangle method.

To study in a mathematical model the efficiency indicators of a timber truck equipped with a hydro pneumatic suspension of a modular design, a computer program was developed that additionally allows setting the parameters of possible irregularities and defects existing on a low-quality timber road. Roughness can be set in the program text by various functions, in particular, a single trapezoidal irregularity and a random uneven surface are investigated. In the process of modeling the movement of a timber truck, the program displays graphs of oscillations of the wheel and body of a timber truck, as well as the time dependences of the main indicators. The program provides the ability to set various types of suspension (spring-damping, hydropneumatic with and without a shock absorber) and change the parameters of the suspension over a wide range. Also, the program allows you to set the physical and geometric parameters of the mechanical system: the stiffness and damping coefficient of the tire, the mass of the wheel and body, the diameter of the wheel. To study the influence of the type of unevenness on the nature of the operation of the hydropneumatic suspension, two series of experiments were carried out in the developed program: the first series – on the movement of a model timber truck through a trapezoidal unevenness, the second series – when driving on a random uneven surface. A comparative study of the smooth running of a timber truck with a conventional spring-shock-absorbing suspension, a hydropneumatic suspension without a shock absorber and with an additional shock absorber was carried out.

3. Results and discussion

It was revealed that when moving a trapezoidal irregularity with a height of 10 cm and bases of 10 and 15 cm, the volume of the pneumatic cavity decreases for a short time by 7 l, after which it increases for a short time by 7 l from the average value of 27 l (figure 3a). In this case, the pressure correspondingly increases by 0.6 atm, after which it decreases by 0.6 atm from the average value of 2.6 atm (figure 3b). The air temperature in the cavity when crossing the trapezoidal unevenness experiences a slight surge with an amplitude of less than 1 K (figure 3c).

It was found that when a model logging vehicle moves along a low-quality logging road with random irregularities and surface defects corresponding to an access logging dirt road, the volume of the pneumatic cavity changes from 22 to 32 l (figure 4a), the pressure changes from 2 to 3 atm (figure 4b), the temperature ranges from 272.5 to 273.4 K (at an ambient temperature of 273 K) (figure 4c). Thus, in the mode of movement on a substantially uneven surface, the hydropneumatic suspension is in a stable operational state, the volume, pressure and temperature are within predictable limits, the pressure does not exceed 3 atm, which allows the use of conventional, widespread elements of the pneumatic system.

![Figure 3](image)

**Figure 3.** Depending on the time when driving a single roughness: (a) the volume of the cavity of the pneumatic cylinder; (b) air pressure in the pneumatic cylinder; (c) air temperature in the pneumatic cylinder.
Figure 4. Dependence on time when moving on a random uneven surface: (a) the volume of the cavity of the pneumatic cylinder; (b) air pressure in the pneumatic cylinder; (c) air temperature in the pneumatic cylinder.

An analysis of the vibration spectra of the timber car body showed that the vibration amplitude in the mid-frequency range from 1 to 2.5 Hz is more important when using a conventional suspension with a hydraulic shock absorber in comparison with a hydropneumatic suspension without a shock absorber. It was also found that the greatest amplitude of oscillations in the low frequency interval not exceeding 0.8 Hz is observed when using a hydropneumatic suspension without a shock absorber in comparison with a traditional suspension and a hydropneumatic suspension with a shock absorber (figure 5, curves 1-3).

Figure 5. Oscillation spectra of a forest car body when driving on a random uneven surface at a distance of 1000 m at a speed of 10 m/s with a conventional suspension with hydraulic shock absorber 1, hydropneumatic suspension without shock absorber 2 and with shock absorber 3.

The high peak in the frequency range 0-0.8 Hz in the spectrum for hydropneumatic suspension can be reduced by using one or two shock absorbers connected in parallel with the suspension. Then the specified peak decreases from 0.41 to 0.26 relative units, which corresponds to the amplitude of oscillations when using a conventional spring-shock-absorbing suspension (figure 5, curve 3). However, in this case, the energy is redistributed to the frequency range 1.8-3.8 Hz, where the spectrum increases by approximately 0.5 relative units.

Compared to a conventional spring-damping suspension, the hydropneumatic suspension provides a more favorable amplitude-frequency response by reducing the intense peak in the 1-2 Hz region from 0.25 to 0.14 relative units by increasing the less intense and more extended spectral region in the 1.5-4.0 Hz at 0.2-0.5 rel. units.

4. Conclusion
Thus, the developed mathematical and simulation models of movement on a timber road of various qualities of a timber truck equipped with the proposed construction of a hydropneumatic suspension of a modular type, and the dependencies revealed in the study using them, allow us to conclude that the proposed hydropneumatic suspension allows you to effectively overcome single trapezoidal and random irregularities, being in a stable operating condition, volume, pressure and temperature are within predictable limits, compared to a conventional spring-damping suspension, a hydropneumatic suspension provides a more favorable amplitude-technical characteristic.
References

[1] Els P S and Grobbelaar B 1999 Heat transfer effect on hydropneumatic suspension systems. *Journal of Terramechanics* **36**(4) 197 DOI: 10.1016/S0022-4898(99)00012-9

[2] Sujatha C and Tejesu P 2002 Heavy vehicle dynamics-comparison between leaf spring and hydropneumatic suspensions. *Proceedings of SPIE – The International Society for Optical Engineering* **4753** 311

[3] Baldi M and Melrelles P S 2003 Hydropneumatic suspension design. *Proceedings of the ASME Design Engineering Technical Conference* **5** 2555

[4] Wu Z C, Chen S Z, Yang L and Meng X 2005 Stiffness analysis of controllable stiffness hydropneumatic springs for vehicles. *Transaction of Beijing Institute of Technology* **25**(12) 1035

[5] Sun T, Yu F, Zhang Z D and Shen X M 2007 Uncertainty analysis for a hydro-pneumatic suspension system of heavy duty vehicles and H controller design. *Journal of Vibration and Shock* **26**(9) 51

[6] Van Der Westhuizen S F and Schalk Els P 2015 Comparison of different gas models to calculate the spring force of a hydropneumatic suspension. *Journal of Terramechanics* **57** 41 DOI: 10.1016/j.jterra.2014.11.002

[7] Zhang P, Li Y and Li P 2016 Dynamic characteristics analysis of hydro-pneumatic suspension system. *Journal of Computational and Theoretical Nano science* **13**(9) 5794 DOI: 10.1166/jctn.2016.5490

[8] Ali D, Frimpong S 2018 Artificial intelligence models for predicting the performance of hydropneumatic suspension struts in large capacity dump trucks. *International Journal of Industrial Ergonomics* **67** 283 DOI: 10.1016/j.ergon.2018.06.005

[9] Lin D, Yang F, Gong D, Rakheja S 2020 Design and experimental modeling of a compact hydropneumatic suspension strut. *Nonlinear Dynamics* **100**(4) 3307 DOI: 10.1007/s11071-020-05714-3

[10] Yesilyurt H Y, Alkalin O 2018 Ride dynamics of an off road vehicle with hydropneumatic suspension system. 9th *International Automotive Technologies Congress OTECJN 2018* (Bursa) p 9

[11] Dong X, Jiang Y, Zhong Z, Zeng W, Liu W 2018 An improved rollover index based on BP neural network for hydropneumatic suspension vehicles. *Mathematical Problems in Engineering*, 7859521 DOI: 10.1155/2018/7859521