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ROPE-CHOKER TRAILED IMPLEMENTS: DETERMINATION OF PARAMETERS

The article describes the mathematical models of a wheel skidding machine movement for various rope-choker trailed implements design solutions. The dynamic load is assessed for a wheel skidding machine. The weight and geometry parameters of trailed implements are determined.

Introduction. Forest vehicle operation conditions can be classified as highly variable dynamic processes, with their origin and time behavior depending on external and internal factors. Now, the great number of design alternatives is common for this stage in the design of integrated wheel machines, providing opportunities to carry out various processing operations and techniques applied to the materials to be processed. During the design stage, the choice of basic parameters for the forest machinery is based on dynamic phenomena in the vehicle's units and assemblies [1].

To solve this problem, the mathematical apparatus can be designed for the synthesis of the system's dynamic components. The factors, predominantly affecting the transportation system's behavior dynamics, can be analyzed in order to choose the calculated kinematic parameters and the weight parameters for the systems under comparison [2]. In this article we have considered, as these factors, the statistical values of vertical accelerations in centers of gravity of a trailed module, a driver and a seat.

The machines different in their design principles were used to choose the vehicle design model for the analysis of operational and ergonomic characteristics in the centers of gravity of rope-choker trailed implements (RCTI), a driver and a seat. As a basic machine, MTZ-82.1 utility tractor was used, carrying the standard skidding process equipment with attached trunnions.

Mathematical models describing movement of a wheel skidding machine with trailed implements. The principles of the design model construction and the assumptions used to develop the mathematical models describing movement of a wheel skidding machine with RCTI are similar to the mathematical apparatus described in [3]: the independent time-varying coordinates (degrees of freedom), determining the positions of all machine's masses in transient and steady-state movement modes, shall be found. The generalized coordinates describing the design dynamic models are as follows: the towing tractor's center of gravity vertical, angular and longitudinal movement (y_1, y_2, y_3); the towing tractor front axle's center of gravity vertical movement (y_4); the trailed implements' center of gravity vertical, angular and longitudinal movement (y_5, y_6, y_7); the motor crankshaft rotation angle (y_8); the towing tractor wheels rotation angle (y_9 and y_{10}); the vertical and longitudinal movements of discrete masses in the bundle of trees (y_{11}, y_{12}, y_{13}); the driver's and seat's center of gravity vertical movement (y_{14}); the trailed implements axles' balance trolley angular movement (y_{15}).

The design models (Fig. 1) include the parameters as follows: the motor torque (T_M); the moment of inertia of rotating masses in the motor and in the driving parts of the clutch (I_M); the moments of inertia of the transmission elements and the towing tractor wheels (I_{K1} and I_{K2}); the moments of inertia of the towing tractor and the trailed skidding implements (I_T и I_S); the moment of inertia of the trailed implements axles' balance trolley (I_B); the towing tractor weight (W_T); the sprung mass of the towing tractor's front axle (m_A); the trailed skidding implements weight (m_S); the discrete masses within the timber bunch (m_1, m_2 and m_3); the driver's and seat's sprung mass (m_D); the towing tractor's front and rear axle tire vertical stiffness and resistance (c_2, k_2 and c_3, k_3); the vertical and horizontal stiffness and resistance of the coupling between the towing tractor and the trailer (c_{41}, k_{41} and c_{42}, k_{42}); the trailed implements' axle tire vertical stiffness and resistance (c_{51}, k_{51}, c_{52}, k_{52}, c_{53} and k_{53}); the tire and soil longitudinal stiffness and resistance calculated for the points of contact between the tractor's front and rear axle wheels and the bearing surface (c_{61}, k_{61} and c_{62}, k_{62}); the towing tractor's front and rear axle drive shaft angular stiffness and resistance (c_7, k_7 and c_8, k_8); the longitudinal and vertical stiffness and resistance of the link between the bundle and the tractor (c_9, k_9 and c_{10}, k_{10}); the timber bundle vertical stiffness and resistance (c_{11} and k_{11}); the driver seat vertical stiffness and resistance (c_{12} and k_{12}); the towing tractor's front and rear axle drive gear ratios (i_1 and i_2); the towing tractor's center of gravity coordinates (a, b and h_T); the driver's center of gravity horizontal coordinate (l_D); the trailed skidding implements' center of gravity coordinates (l_S, h_1); the coordinates of the points of the coupling between the towing tractor and the trailer (l_C, h_C, h_2); the timber bundle length (L_B); the timber bundle's center of gravity coordinates (l_1, l_2); the distance between the trailed skidding implements' axle and the horizontal coordinate of the point where the bundle's butt contacts the shield (l_3); the distance between the axle's horizontal coordi-nate and the trailed skidding implements' center of gravity (l_4); the distances between the axles' horizontal coordinates and the trailed skidding implements' center of gravity (l_6, l_7); the distance between the trailed skidding implements' center of gravity and the horizontal coordinate of the point where the bundle's butt contacts the shield (l_8); the distance between the horizontal coordinate of the balance trolley's center and the trailed implements (l_9); the trailed implements axles' balance trolley arms (l_{10} и l_{11}); the distance between the 3rd axle's horizontal coordinate and the trailed implements' center of gravity (l_{12}); the distance between the bearing surface and the arch rope-driving roller (h_3); the distance between bearing surface and the point where the bundle's butt contacts the trailed skidding implements' shield (h_4); the towing tractor wheels and the trailed skidding implements axle tires rolling radii (r_1, r_2 и r_3); the tractor front and rear wheels' tangential towing forces (P_{K1} and P_{K2}); the forces resisting the towing tractor wheels and the trailed skidding implements' axle tires rolling (P_{F1}, P_{F2}, P_{F3}, P_{F4}, and P_{F5}); the force resisting the bundle drugging (P_V); the current roughness under the towing tractor wheels and the trailed skidding implements' axles (q_1, q_2, q_3, q_4 and q_5).

The mass and geometry parameters, the moments of inertia, the resistance forces and the tangential towing forces similar to that in [3] were used to describe the dynamic systems under consideration.

See Fig. 1 for the design models describing the various design solutions for the RCTI-equipped wheel skidding machine's dynamic system. Several assumptions were taken into consideration, the dynamic system design and its components' movement kinematic characteristics were analyzed resulting in alternatives as follows:

a – two-axle RCTI (14 degrees of freedom);

b – RCTI mounted on a two-axle balance trolley (15 degrees of freedom);

c – three-axle RCTI (15 degrees of freedom).

Due to the designed mathematical apparatus, the calculations as follows became possible in *MatLab R*2006*a* high-level programming environment: the models' DoF deviation matrix (the numerical values of deviations), the first derivatives of these deviations and the respective times in the process course; due to these results, all parameters were calculated that are necessary to assess the dynamic loads affecting the RCTI-equipped wheel skidding machines. The results were obtained for various RCTI design solutions (for the single-axle RCTI, see [3] for the mathematical apparatus description).

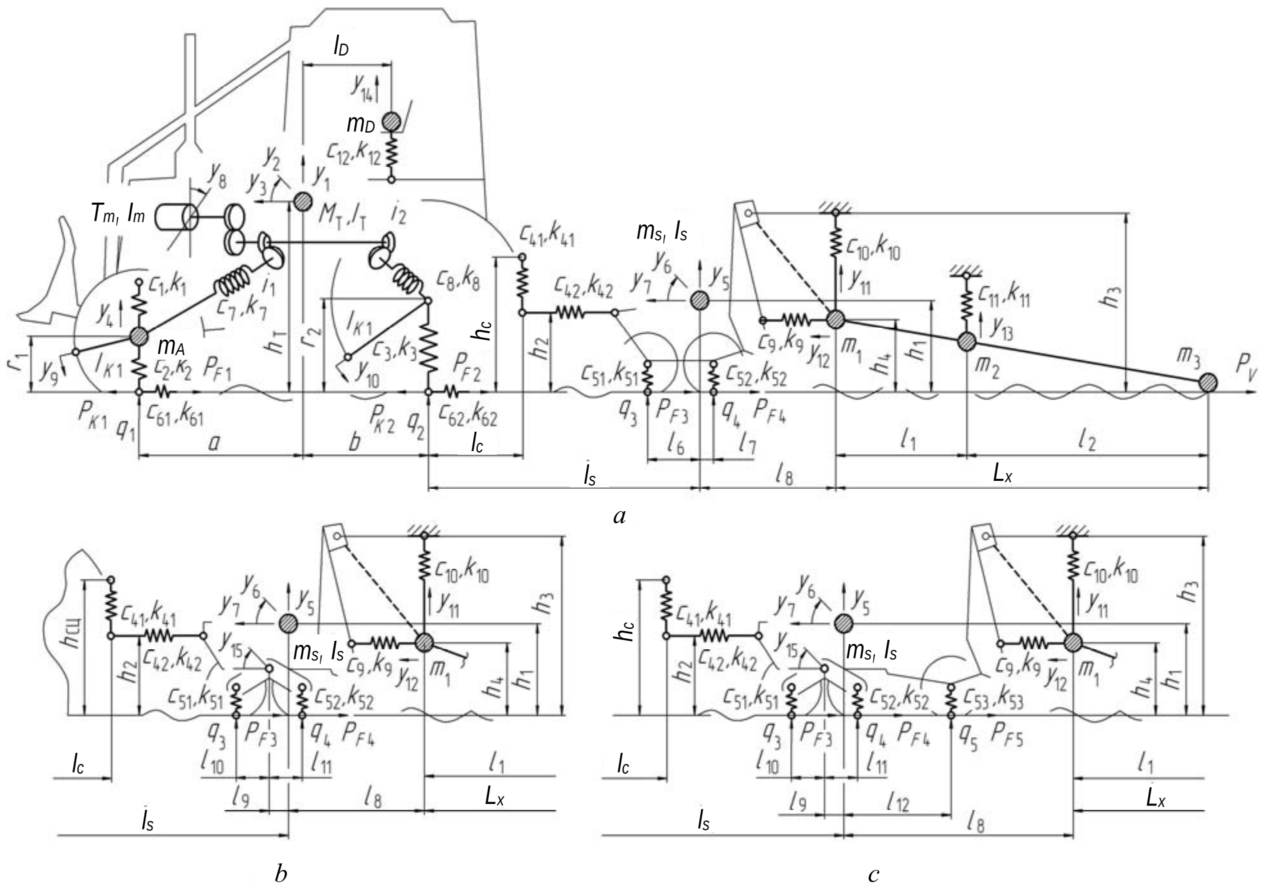

Fig. 1. The dynamic system of a wheel skidding machine with a rope-choker trailed implements: the design models

The equations as follows were used to calculate the vertical acceleration variations, as a function of time, in the RCTI center of gravity:

a) for the single-axle RCTI:

$$\begin{aligned}\ddot{Y}_5 = \Big[& c_{41}\big(Y_1-(b+l_C)Y_2-Y_5-(l_S-l_C)Y_6\big)+\\ & + k_{41}\big(\dot{Y}_1-(b+l_C)\dot{Y}_2-\dot{Y}_5-(l_S-l_C)\dot{Y}_6\big)-\\ & - c_5(Y_5-l_4Y_6-Q_3)-k_5\big(\dot{Y}_5-l_4\dot{Y}_6-\dot{Q}_3\big)-\\ & - c_{10}(Y_5-(l_4+l_3)Y_6-Y_{11})-\\ & - k_{10}\big(\dot{Y}_5-(l_4+l_3)\dot{Y}_6-\dot{Y}_{11}\big)\Big]/m_S;\end{aligned}$$

b) for the two-axle RCTI:

$$\begin{aligned}\ddot{Y}_5 = \Big[& c_{41}\big(Y_1-(b+l_C)Y_2-Y_5-(l_S-l_C)Y_6\big)+\\ & + k_{41}\big(\dot{Y}_1-(b+l_C)\dot{Y}_2-\dot{Y}_5-(l_S-l_C)\dot{Y}_6\big)-\\ & - c_{51}(Y_5+l_6Y_6-Q_3)-c_{52}(Y_5-l_7Y_6-Q_4)-\\ & - k_{51}(\dot{Y}_5+l_6\dot{Y}_6-\dot{Q}_3)-k_{52}(\dot{Y}_5-l_7\dot{Y}_6-\dot{Q}_4)-\\ & - c_{10}(Y_5-l_8Y_6-Y_{11})-k_{10}\big(\dot{Y}_5-l_8\dot{Y}_6-\dot{Y}_{11}\big)\Big]/m_S;\end{aligned}$$

c) for the RCTI mounted on a balance trolley:

$$\begin{aligned}\ddot{Y}_5 = \Big[& c_{41}\big(Y_1-(b+l_C)Y_2-Y_5-(l_S-l_C)Y_6\big)+\\ & + k_{41}\big(\dot{Y}_1-(b+l_C)\dot{Y}_2-\dot{Y}_5-(l_S-l_C)\dot{Y}_6\big)-\\ & - c_{51}(Y_5+l_{10}Y_{15}+(l_9+l_{10})Y_6-Q_3)-\\ & - k_{51}(\dot{Y}_5+l_{10}\dot{Y}_{15}+(l_9+l_{10})\dot{Y}_6-\dot{Q}_3)-\\ & - c_{52}(Y_5-l_{11}Y_{15}+(l_9-l_{11})Y_6-Q_4)-\\ & - k_{52}(\dot{Y}_5-l_{11}\dot{Y}_{15}+(l_9-l_{11})\dot{Y}_6-\dot{Q}_4)-\\ & - c_{10}(Y_5-l_8Y_6-Y_{11})-k_{10}\big(\dot{Y}_5-l_8\dot{Y}_6-\dot{Y}_{11}\big)\Big]/m_S;\end{aligned}$$

d) for the three-axle RCTI:

$$\begin{aligned}\ddot{Y}_5 = \Big[& c_{41}\big(Y_1-(b+l_C)Y_2-Y_5-(l_S-l_C)Y_6\big)+\\ & + k_{41}\big(\dot{Y}_1-(b+l_C)\dot{Y}_2-\dot{Y}_5-(l_S-l_C)\dot{Y}_6\big)-\\ & - c_{51}(Y_5+l_{10}Y_{15}+(l_9+l_{10})Y_6-Q_3)-\\ & - k_{51}(\dot{Y}_5+l_{10}\dot{Y}_{15}+(l_9+l_{10})\dot{Y}_6-\dot{Q}_3)-\\ & - c_{52}(Y_5-l_{11}Y_{15}+(l_9-l_{11})Y_6-Q_4)-\\ & - k_{52}(\dot{Y}_5-l_{11}\dot{Y}_{15}+(l_9-l_{11})\dot{Y}_6-\dot{Q}_4)-\\ & - c_{53}(Y_5-l_{12}Y_6-Q_5)-k_{53}(\dot{Y}_5-l_{12}\dot{Y}_6-\dot{Q}_5)-\\ & - c_{10}(Y_5-l_8Y_6-Y_{11})-k_{10}\big(\dot{Y}_5-l_8\dot{Y}_6-\dot{Y}_{11}\big)\Big]/m_S.\end{aligned}$$

The equation as follows was used to calculate the vertical acceleration variations, as a function of time, in the driver's and seat's center of gravity:

$$\begin{aligned}\ddot{Y}_{14} = [& -c_{12}\big(Y_{14}-Y_1+l_BY_2\big)-\\ & -k_{12}\big(\dot{Y}_{14}-\dot{Y}_1+l_D\dot{Y}_2\big)]/m_D.\end{aligned}$$

In these equations, capital letters for the degrees of freedom mean the matrices resulting from the modeling procedure. These matrices were processed to plot the normalized spectral densities for the accelerations affecting the centers of gravity of the RCTI, the driver and the seat. Also, the spectral density variations versus the process equipment parameters were plotted (see Fig. 2, 3).

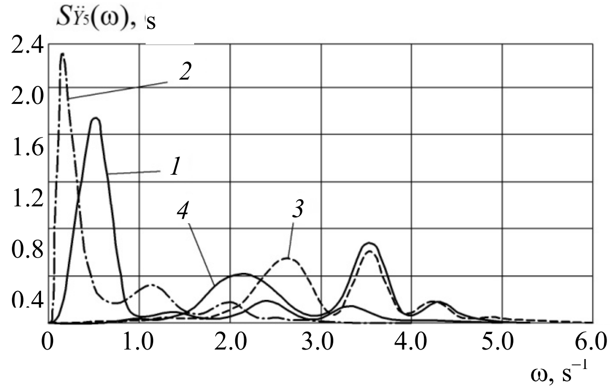

Fig. 2. Normalized spectral densities for the RCTI center of gravity vertical accelerations, for various modes of movement on the skidding road:
1 – single-axle; *2* – two-axle; *3* – on a balance trolley; *4* – three-axle

The dynamic process modeling results for various designs of the trailed component were obtained for the timber bundle transportation on the skidding road, with the bundle volume 1.0 m^3, the transportation speed 4.26 km/hour. The mathematical statistics methods were used to calculate the skidding road microprofile parameters.

See Fig. 2 for the normalized spectral densities for the accelerations affecting the RCTI center of gravity for various design solutions.

For the single-axle design, the maximum normalized spectral density, 1.73 s, was recorded when the frequency was 0.5 s^{-1}. For the two-axle design, the spectral density maximum was recorded at 0.22 s^{-1}. The statistical parameters of the accelerations decay more intensively for the single-axle design.

If the balance trolley is used in the trailed module, the spectral density maximums were recorded twice, at 2.68 s^{-1} and 3.51 s^{-1}. For the three-axle trailed implements design, the frequency range is wider, with the maximums recorded at 2.10 s^{-1} and 3.53 s^{-1}.

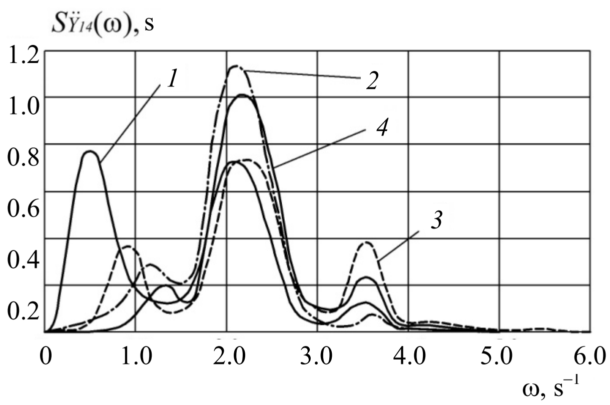

Fig. 3. Normalized spectral densities for the driver's and the seat's center of gravity vertical accelerations, for various modes of movement on the skidding road: *1* – single-axle; *2* – two-axle; *3* – on a balance trolley; *4* – three-axle

The normalized spectral density plots for the driver's and the seat's center of gravity vertical accelerations (see Fig. 3) demonstrate that there are three maximums recorded within the frequency range 0.5 s^{-1} … 3.5 s^{-1}.

For the driver's and the seat's center of gravity accelerations, the frequency ranges where the spectral density maximums are recorded and the absolute values of these maximums depend on the number of RCTI bearing axles and the type of their joint. If, instead of the single bearing axle, two axles are used, the frequency range is shifted to higher frequencies, from 0.5 s^{-1} to 2.2 s^{-1}, with the absolute maximum becoming 1.51 times higher. If a balance trolley is used instead of one axle, the frequency range is shifted to higher frequencies, up to 2.3 s^{-1}, where the peak value is observed, with the absolute maximum becoming 1.08 times lower. If three axles are used in the RCTI, the spectral density maximums become 1.43 times higher.

The dynamic system parameters affecting the wheel skidding machine travelling mode on a skidding road were varied to determine the RCTI weights and geometric sizes. Spectral density maximums were considered as optimization criteria.

Conclusion. As a result of the mathematical modeling, the variation ranges were determined for the rope-choker trailed implements: weight, 0.5…0.9 tons; center of gravity height, 0.7…1.1 m; distance from the towing vehicle's rear wheel to the center of gravity, 0.8…1.1 m; timber bundle bracket length, 0.4….0.6 m; distance from the center of gravity to the balance trolley center, 0.3…0.1 m; balance trolley arms distance, 0.65…0.75 m.

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