Methods of simplification and reduction of the dynamic system «engine-transmission tractor pack of wood» of tracked skidding tractors

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Abstract. The article presents a method for simplifying the dynamic model «engine-transmission-tractor-bundle of wood», which makes it possible to minimize labor costs in the study of resonant vibrations. Only a limited spectrum of natural frequencies is determined, including only those frequencies at which resonant oscillations can occur.

1. Introduction
Skidding tractors with various technical parameters and solutions are widely used in the logging industry and in forestry [1-5].

The most important condition for the effective functioning of skidding tractors is to equip them with a modern transmission design. The design and type of transmission largely determines the traction and dynamic properties, performance and durability of the machine, its ergonomic properties, as well as the environmental compatibility of the engine with the forest soil. The specific operating conditions of timber tractors impose special requirements on the design and characteristics of their transmission [5-10]. When improving the design, there is also such a question as improving the reliability and durability of the transmission elements. Their complexity and the variety of loads acting on the transmission parts, as well as the peculiarity of the operating conditions of tractors in the cutting area, lead to frequent failures of transmission parts and large downtime of the tractor [11-13]. Downtime of the tractor in repair affects the reduction of the annual operating time, the increase in the cost of maintenance and repair, and as a result-a significant reduction in the economic efficiency of the tractor. Therefore, improving the reliability and performance of the crawler skidder in general and transmission parts in particular is an urgent task [5, 13-16].

2. Methods and Materials
Methods for simplifying and reducing the parameters to a single speed of rotation or movement are used when replacing a real dynamic system with an equivalent design scheme [5, 11, 17]. Such equivalent schemes serve as the basis for the development and research of mathematical models in order to measure the dynamic properties of the system and improve its potential properties.

Foreign companies [4, 5, 10, 15, 18, 19] and domestic researchers argue that there are no uniform methods for simplifying a real dynamic system with replacing it with an equivalent dynamic scheme. Such a replacement of a real dynamic system with an equivalent calculation scheme is accompanied
by reasonable assumptions, simplifications and reduction of the system parameters [5, 6, 7, 12, 20]. When replacing a real dynamic system with an equivalent calculation scheme, the researcher's skills and skill are shown. It is proved that even for the analysis of the natural frequencies of the system, the degree of its simplification is also determined by the research tasks, the machine operating modes and the dynamic properties of the system [2, 15, 20]. For example, to analyze the dynamic loads that occur in the mechanisms and systems of the machine under the influence of single power peaks (the beginning of the movement of the machine from a place, overcoming single obstacles, etc.), the dynamic system can be represented in the form of three or even two mass schemes [5, 14, 16, 21]. It is proved that a multi-mass dynamical system behaves like a two-and, more rarely, a three-mass system in transient modes [5, 9, 14]. Experimental studies of tracked skidding tractors have shown that resonant processes occur in the transmission from the engine at the third natural frequency [4, 5, 13, 20], therefore, the equivalent design scheme of the dynamic system «engine-transmission-tractor-pack» should have no more than four masses [16, 22, 23]. The real dynamic system "engine-transmission-tractor-bundle of wood", which has about twenty discrete masses connected by elastic inertia-free bonds, was replaced for such studies by a four-mass scheme [1, 7, 19, 22]. It should be noted that any simplification of the design scheme should be accompanied by an analysis of the natural frequencies of the system, with the establishment of the degree of distortion of the studied natural frequencies under the influence of simplification of the design scheme [2, 5, 12, 19].

Determining and calculating the parameters of the oscillatory system "engine-transmission-tractor-bundle of wood" is quite time-consuming, primarily experimental work, so analytical studies on industrial and agricultural tractors are practically not found [15, 18, 22].

Of the many known experimental methods for determining the moment of inertia, the most accurate is the torsional vibration method [6], in which the part or assembly under study is suspended from a steel wire so that the wire is a continuation of the axis of symmetry of the rotation of the part, relative to which the moment of inertia is determined [11].

For any moment of time, the dynamic equilibrium equation of the «part – wire» system can be written [9]. When the part is twisted by an angle $\varphi$, stresses occur in the wire material, which, relative to the axis, will create a moment of force [2, 14, 20, 21]:

$$M = \frac{G I_p}{L} \varphi,$$

where, $G$ - is the elastic modulus of the wire under shear; $I_p$ - is the polar moment of inertia of the cross-sectional area; $L$ - is the length of the wire subject to torsion deformation; $\varphi$ - is the angle of torsion relative to the equilibrium position of the part [3, 12, 19].

From the expression (1):

$$M = C \varphi,$$

where, $C$ - is the torsional stiffness coefficient of the wire, $C = G I_p / L$.

To determine the moment of inertia, a reference part of a simple geometric shape is suspended on the same wire [7, 10, 15], most often a disk, the moment of inertia is calculated simply and the oscillation period is determined, after which the oscillation period of the test part is determined by the same method (node) [16, 18]. The solution of the simplest linear homogeneous differential equation allows us to obtain the transition formula [6, 19]:

$$J_g = J_e \frac{T_e^2}{T_o^2},$$

where, $T_o, T_e$ - the period of one oscillation of the part and the reference on the wire, respectively; $J_o, J_e$ - moments of inertia of the part and the reference on the wire, respectively [12, 23].

The moment of inertia of the standard with clear geometric dimensions:

$$J_e = \frac{g e^2}{2 g},$$
where, $G_e$ - weight of the reference part; $r$ - the radius of the cylinder (disk).

In the running system, there are rotating parts with different axes (drive wheel, tension wheel, support roller, etc.) [5]. The task is reduced to determining the moment of inertia of a complex configuration unit, such as a tension wheel, and bringing the driven shaft or drive wheel to the axis of rotation [7]. For this purpose, the equality of the kinetic energy of the tensioning wheel $T_k$ and the kinetic energy of the tensioning wheel brought to the axis of the driven shaft of the on-board transmission $T_k^p$ is considered:

$$T_k = T_k^p,$$

$$J_k \omega_k^2 = J_k^p \omega_0^2,$$

(5)

where, $J_k$ - tension wheel moment of inertia; $J_k^p$ - the reduced moment of inertia of the wheel to the axis of the driven shaft of the on-board transmission; $\omega_k$ - wheel speed; $\omega_0$ - the speed of the driven shaft of the on-board transmission or the drive wheel.

After some transformations, the final link formula can be written as:

$$J_k^p = J_k \frac{r_z^2 v}{r_k^2},$$

(6)

where, $r_k$ - tension wheel radius; $r_z$ - the radius of the drive wheel (drive sprocket).

The track chain makes a complex movement. The sagging part of the chain moves translationally relative to the tractor and together with the tractor [4, 15, 16]. The parts of the track chain located on the drive and guide wheels perform a rotational movement. In the absence of losses on skidding, the part of the track chain that is in contact with the drag is equal to zero:

$$V - \omega_0 r_z v = 0,$$

(7)

Tractor speed $V$:

$$V = \omega_0 r_z v,$$

(8)

The kinetic energy of the track chain $T_i$ in this mode is determined on the basis of Koenig's theorem [10, 21, 23], according to which the kinetic energy of a system of material points is equal to the sum of the kinetic energies of all the masses of the system, mentally concentrated in its center of mass and moving at the speed of the center of mass and the kinetic energy of the system in its relative motion with respect to a translationally moving reference frame with the beginning in the center of mass, that is:

$$T = \frac{1}{2} \sum_{i=1}^{n} m_i \left(\nu_i^{(r)}\right)^2,$$

(9)

where, $\nu_i^{(r)}$ - the value of the mass $m_i$ in relation to the system moving translationally with the center of mass [2, 22]. Based on this theorem, the kinetic energy of the track chain $T_i$ in the absence of skidding is found as the sum of the energies in motion and with respect to the center of mass:

$$T_g = T_g' + T_g'' = \frac{1}{2} m_0 + \frac{1}{2} m_0 V^2 = m_0 \omega_0^2 r_z v^2,$$

(10)

where, $m_0$ - mass of the moving part of the track chain; $T_g'$, $T_g''$ - accordingly, the energy of movement of the track chain together with its center of mass and relative to the center of mass.

When the track chain is completely skidding relative to the support surface [15], that is, when the tractor has $V = 0$:

$$T_g = \frac{1}{2} m V_z v^2 = \frac{1}{2} m \omega_0^2 r_z v^2,$$

(11)

where, $m$ - track weight.

When bringing the moment of inertia of the crawler chain to the axis of the driven shaft of the onboard transmission, we equate the kinetic energies:
\[ T_g = T^n_g, \]  
(12)

where, \( T^n_g \)- the kinetic energy of the track chain, brought to the axis of the driven shaft.

The mode of full skidding of the crawler engine is not typical for the operation of a skidder tractor, it is extremely rare, so we exclude it from further consideration [3]. According to the existing standards, the permissible skidding should not exceed 5%. Hence, the kinetic energy can be considered as in the absence of skidding [12].

In the absence of skidding:

\[ m_o \omega^2 r^2 = J'' \omega_o^2, \]  
(13)

Then:

\[ J'' = m_o r^2, \]  
(14)

where \( J'' \)- the reduced moment of inertia of the moving part of the track chain.

The given moment of inertia of the translational-moving masses of the skidding system \( m_o \) (skidding tractor with a bundle of wood) is determined from the equality of the kinetic energies of the \( T_c \) and \( T^n_c \), we get:

\[ \frac{m_v \omega^2}{2} = \frac{I^2 \omega_o^2}{2}. \]  
(15)

After converting the formula (5), we get:

\[ J^n_c = \frac{m_v r^2 \omega_o^2}{\omega_o^2} = m_c r^2, \]  
(16)

where, \( T_c \) and \( T^n_c \)- the kinetic energy of the skidding system and the kinetic energy of the skidding system, brought to the axis of the driven shaft; \( J^n_c \)- the moment of inertia of the translational-moving mass of the skidding system, brought to the axis of the driven shaft.

3. Results and Discussion

If the skidding significantly exceeds the established GOST 5%, then the given moment of inertia of the translational-moving mass of the skidding system can be determined [3, 5-7, 9, 12, 14, 15, 17-19, 21]. To do this, in formula (5), we replace the theoretical speed of the skidder \( v \) with the actual \( v_g \) and remember that \( v_g = (1 - \delta) v \), we write formula (16):

\[ J^n_c = \frac{m_v r^2 (1-\delta)^2 \omega_o^2}{\omega_o^2} = m_c r^2 (1 - \delta)^2, \]  
(17)

where, \( \delta \)- slip rate.

When evaluating the dynamic properties of the system, the stiffness coefficient is estimated according to the known formulas for the resistance of materials [5, 7, 16, 18].

4. Conclusions

When studying the dynamic systems of forest machines, it is customary to bring the moments of inertia and the shaft stiffness coefficients to a single rotation speed. For example, in the study of the system «engine-transmission-tractor-bundle of wood» lead to the speed of rotation of the crankshaft [5, 7, 10, 12, 15, 16, 21]. For the given moments of inertia and stiffness coefficients to any speed, their values must be divided by the square of the transmission ratio [5].

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