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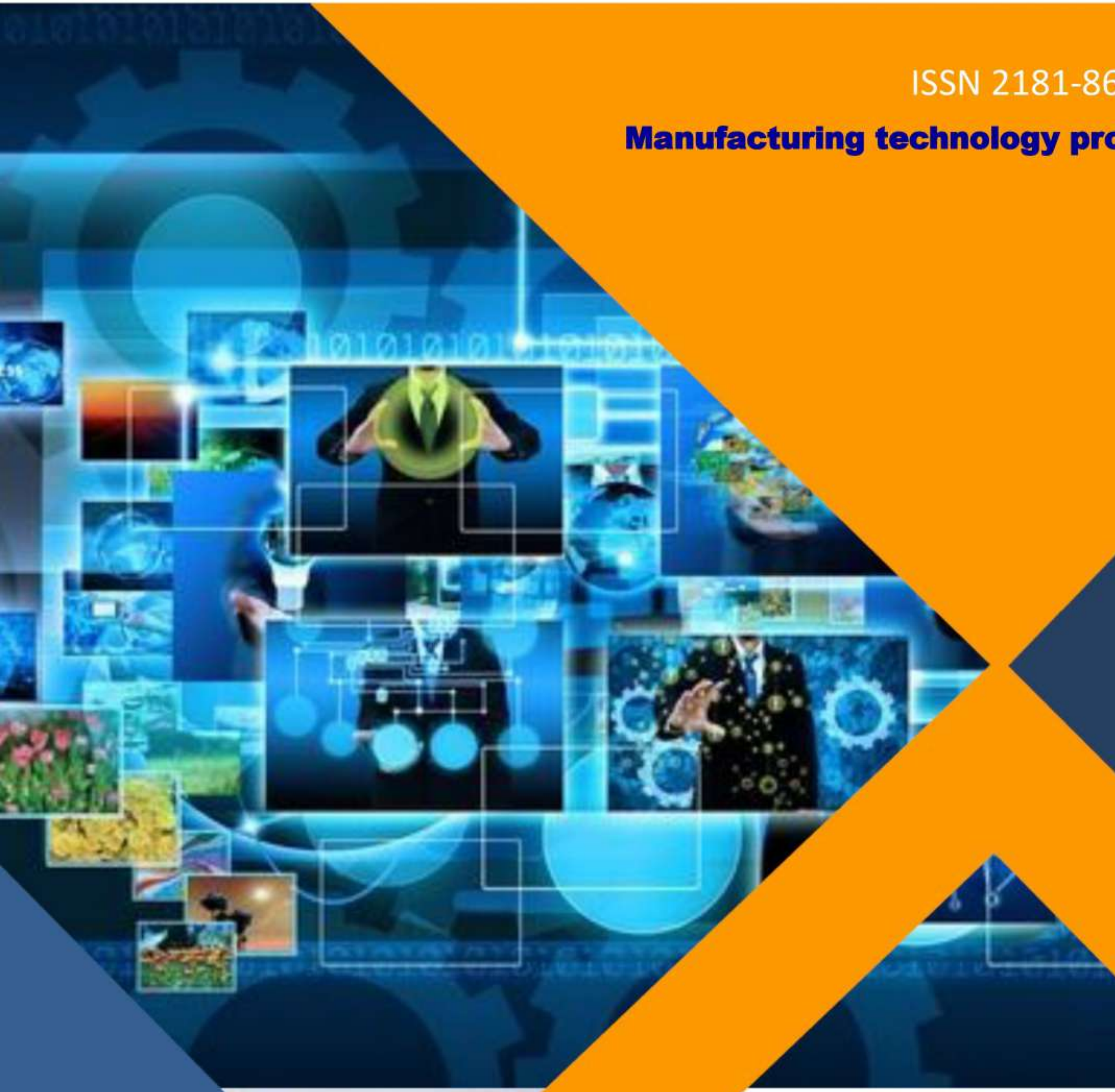

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MATHEMATICAL MODEL OF MACHINE AGGREGATE OF TILLAGE EQUIPMENT PROCESS

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Abstract: In the article, the mathematical model of the Tillage-cultivating unit was obtained. Based on the numerical solution of the system of motion equations of the machine unit, the laws of speed, acceleration and moment changes of the driving sprocket of the pile drum and chain transmission were determined depending on the resistance moment coming from the cutting. The laws of dependence of the average values of the vibration interval of the angular speed, acceleration and torques of the pile drum and the driven sprocket on the resistance moment were studied. Mechanical characteristics of pile drum and chain drive sprocket are determined.

Keywords: Machine, aggregate, drum with pile, cutting, chain drive, driven sprocket, shaft, dynamics, technological resistance, moment, moment of inertia, potential energy, mechanical characteristic.

Introduction. The rapid development of the engineering field places a high demand on the processes of production of technological machines and their operation. As the main parts of modern technical systems, machine units can perform important tasks such as efficient use of energy sources, optimization of production processes, and creation of machine structures capable of withstanding the loads that occur during operation. Therefore, the dynamic analysis of machine assemblies is important in improving their quality and ensuring their efficiency. Dynamic analysis of machine assemblies is a research direction aimed at in-depth study of their movement, their condition under the influence of loads, and dynamic processes that occur. Through this analysis, it is possible to determine such factors as load distribution, vibration and imbalance in the elements of aggregates. The main goal of such an analysis is to optimize the operation of the system, increase its reliability and prevent malfunctions [1-5].

The following tasks can be cited as the importance of dynamic analysis:

reduce energy consumption, optimize transmission systems and increase overall system performance through proper analysis of dynamic processes in machine aggregates;

analysis prevents accidents that may occur during operation of the system, as well as reducing vibration and ensuring balance in high-speed units;

it is possible to extend their service life and reduce repair costs by predetermining damage caused by loads and vibrations in machine aggregates.

The following modern methods and technologies are used in the process of dynamic analysis:

mathematical modeling allows to develop dynamic models of machine units and to correctly analyze movement processes [6-11];

computer simulation - it is possible to save time and costs by checking dynamic processes in a virtual environment with the help of computer technologies;

experimental tests - as a result of experiments conducted in real conditions, the accuracy of modeling is confirmed and additional data is collected.

This study aims to perform a dynamic analysis of the machine aggregate machine of the tillage aggregate equipment.

Methodology & empirical analysis. The normal and continuous operation of the earth-moving unit mainly depends on the reduction and smoothing of the maximum values of the impact force by grinding the cutting on the bearing supports, the guide, and the elements of the transmission mechanism [7-8]. It is also important to reduce the average value of the power consumption and its fluctuation range. Therefore, in order to increase the level of accuracy of theoretical research, it is appropriate to consider the dynamic analysis of the mechanism of the tillage unit in the machine unit [1-3].

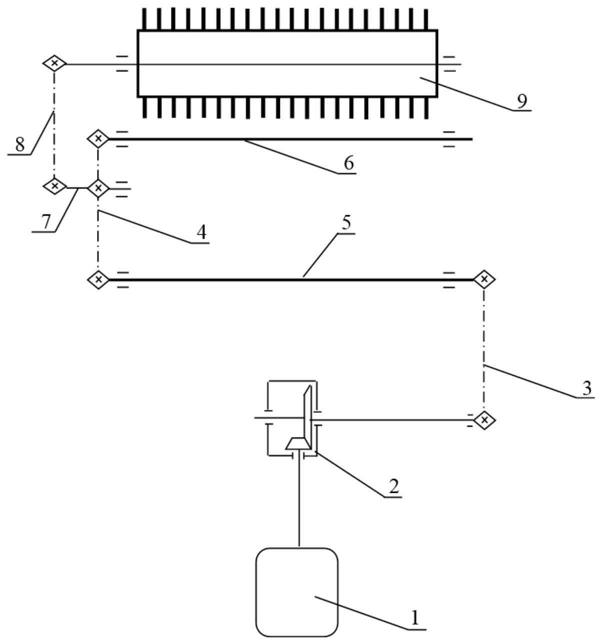


Figure 1. Kinematic scheme of the tillage equipment

The transmission mechanisms of the tillage unit consist of chain gears, and the rotary motion is taken from the tractor's power take-off shaft. Figure 1 is a ground-working unit the kinematic scheme of transmission and working mechanisms is presented. The movement is transmitted from the power take-off shaft 1, bevel gear reducer 2, chain sprockets 3, 4 to conveyors 5 and 6. Through the additional star shaft 7 in the chain drive 4, the movement is transmitted to the next chain drive 8 and then the working body to the pile drum 9, which grinds the cuttings. That is, two elements, conveyors 5, 6 and piled drum 9, are mainly involved in the cutting technology. Therefore, in theoretical research, the resistance to these working bodies was obtained as a result of experimental research.

It is important to determine the laws of movement velocity, taking into account the resistance coming from the cut. As can be seen from the scheme in Fig. 1, the sufficiently large amount of transmitted power and the number of revolutions per minute of the pile drum of the working body are around 300 *min* $^{-1}$, which justify the use of chain gears instead of belt gears in the transmission mechanism of the equipment. Also, the rotary motion is in two branches, one drives the conveyors and the other drives the pile drum.

As a result of experimental research, the moment of resistance on the shaft of the working body in the grinding of the piece was carried out in laboratory conditions, in a stationary mode, that is, the vibrations of the body due to ground irregularities were not taken into account. Therefore, in research, we considered that the body of the chipping equipment does not move. This allows for an autonomous assessment of the grinding process. Because the additional vibration of the equipment body velocity up the process of grinding the piece, but the resistance to the conveyor increases somewhat. By making the dynamic model of the unit as simple as possible, reducing the number of masses, it is desirable to obtain [6-8] .

The calculation scheme of the unit is 6 masses was obtained (Fig. 2).

In this case, I is the total mass of the power shaft to the output shaft of the reducer, II is the mass of the first conveyor shaft, III is the mass of the second conveyor output shaft, IV is the mass of the branching shaft to the transmission mechanism of the chip crushing equipment pile drum, V is the recommended transmission drive sprocket shaft. mass, VI is the reference mass of the star base with the driving structure, the pile drum and the shaft. Therefore, we consider the power take-off shaft in the machine unit as a source of motion and bring it to the output shaft of the reducer. The remaining masses were chosen to match the shafts separated by chain drives.

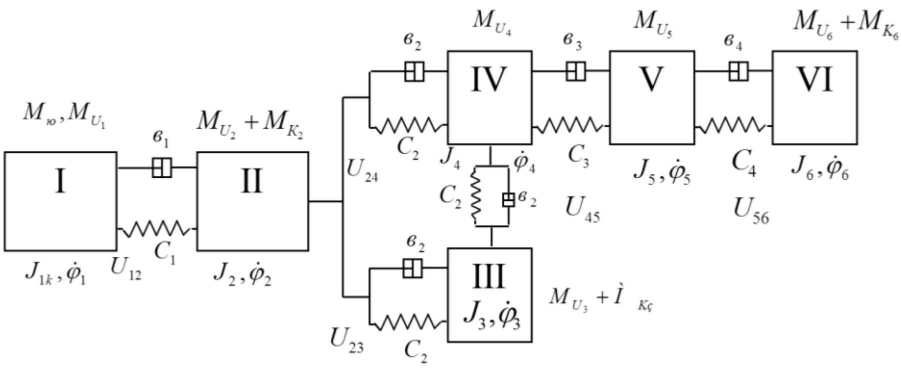


Figure 2. Scheme for calculating the machine aggregate's tillage equipment

Mathematical model of the machine assembly of the tillage device. According to the calculation scheme (Fig. 2), it is known that 6 masses move in rotation, so 6 generalized coordinates can be defined. In order to derive the equation of motion of the 6-mass machine unit of the chip grinding equipment, the known [1,2,6] second-order Langrage equation, taking into account the conservative and dissipative forces, was used:

$$\frac{d}{dt}\left[\frac{\partial T}{\partial \dot{\varphi}_i}\right]-\frac{\partial T}{d\varphi_i}+\frac{\partial \Pi}{\partial \varphi_i}+\frac{\partial \Phi}{\partial \dot{\varphi}_i}=M_i(\varphi_i), \tag{1}$$

here φ_i – i turning angles of masses for a mass system, that is, generalized coordinates; $T - i$ total kinetic energy of the system with mass ta; Π – total potential energy of the system; Φ – Dissipative function of Reley in strap and flexible joints; $M_i(\varphi_i)$ – i torques of external forces acting on the masses of the mass system . According to the calculation scheme of the machine aggregate, the expression for determining the total kinetic energy is as follows:

$$T = \frac{1}{2}\left[J_1 \cdot \left(\frac{d\varphi_1}{dt}\right)^2 + J_2\left(\frac{d\varphi_2}{dt}\right)^2 + J_3\left(\frac{d\varphi_3}{dt}\right)^2 + J_4\left(\frac{d\varphi_4}{dt}\right)^2 + J_5\left(\frac{d\varphi_5}{dt}\right)^2 + J_6\left(\frac{d\varphi_6}{dt}\right)^2\right], \quad (2)$$

here $\varphi_1, \varphi_2, \varphi_3, \varphi_4, \varphi_5, \varphi_6$ - the generalized coordinates of the rotating masses, that is, the angles of rotation, which are included in the machine assembly part of the earth-moving aggregate equipment; $J_1, J_2, J_3, J_4, J_5, J_6$ – moments of inertia of masses. The moment of inertia of the mass is determined by the following expressions:

$$J_1 = \left[\left(J_{в_2} + J_{k_2} + J_{ю1}\right) + \left(J_{в_1} + J_{k_1}\right)U_{1p}^2\right]; \quad J_2 = J_{в_5} + J_{ю_2} + J_{ю_3} + J_{T_1} + J'_{T_1} \cdot U_{T_1}^2;$$

$$J_3 = J_{в_6} + J_{ю_4} + J_{T_2} + J'_{T_2} \cdot U_{T_2}^2; \quad J_4 = J_{в_7} + J_{ю_5} + J_{ю_6}; \quad J_{ю_6} = J'_{ю_6} + J''_{ю};$$

$$J_5 = J''_{ю_6}; \quad J_6 = J_{в_9} + J_{б} + J''_{ю_6}, \quad (3)$$

where $J_{в_1}, J_{в_2}, J_{в_5}, J_{в_6}, J_{в_9}$ - moments of inertia of rotating shafts, respectively; J_{k_1}, J_{k_2} – moments of inertia of the bevel gears of the reducer; $J_{ю1}, J_{ю_2}, J_{ю_3} J_{ю_4}, J_{ю_5}, J_{ю_6}$ - moments of inertia of chain drive suitable sprocket $J_{б}$ - moment of inertia of pile drum; $J_{T_1}, J_{T_2}, J'_{T_2}, U_{T_1}, U_{T_2}$ - the moments of inertia of the first and second transporters and the inertia moments and transfer functions of the driving sprockets of the transporter, respectively; U_{1p} - the number of transmissions between the power take-off shaft 1 and the bevel gear output shaft. 3 chain drives are used in the considered six-mass machine unit. Taking them into account, the expression for determining the total potential energy for the machine assembly is:

$$\Pi = \frac{1}{2}\left[C_1(\varphi_1 - U_{12}\varphi_2)^2 + C_2(\varphi_2 - U_{23}\varphi_3)^2 + C_2(\varphi_2 - U_{24}\varphi_4)^2 + \right.$$
$$\left. + C_2(\varphi_3 - U_{34}\varphi_4)^2 + C_3(\varphi_4 - U_{45}\varphi_5)^2 + C_4(\varphi_5 - U_{56}\varphi_6)^2\right], \quad (4)$$

where C_1, C_2, C_3, C_4 – the coefficients of elasticity of chain drives; $U_{12}, U_{23}, U_{24}, U_{34}, U_{45}, U_{56}$ – rotating masses respectively transmission ratios between.

Similarly , we write the expression of Rayleigh's dissipation function for the system as follows[3]:

$$\Phi = \frac{1}{2}\left\{в_1\left(\frac{d\varphi_1}{dt} - U_{12}\frac{d\varphi_2}{dt}\right)^2 + в_2\left(\frac{d\varphi_2}{dt} - U_{23}\frac{d\varphi_3}{dt}\right)^2 + в_2\left(\frac{d\varphi_2}{dt} - U_{24}\frac{d\varphi_4}{dt}\right)^2 + \right.,$$
$$\left. + в_2\left(\frac{d\varphi_3}{dt} - U_{34}\frac{d\varphi_4}{dt}\right)^2 + в_3\left(\frac{d\varphi_4}{dt} - U_{45}\frac{d\varphi_5}{dt}\right)^2 + в_4\left(\frac{d\varphi_5}{dt} - U_{56}\frac{d\varphi_6}{dt}\right)^2\right\} \quad (5)$$

where are the dissipation $в_1, в_2, в_3, в_4$ – coefficients of chain drives.

Generalized moments of force in the machine assembly:

$$M_1(\varphi_1) = M_{ю1} - M_{u1}; \; M_2(\varphi_2) = -(M_{u2} + M_{к2}); \; M_3(\varphi_3) = -(M_{u3} + M_{к3});$$

$$M_4(\varphi_4) = -M_{u4};\ M_5(\varphi_5) = -M_{u5};\ M_6(\varphi_6) = -(M_{u6} + M_{\kappa 6}), \quad (6)$$

where $M_{u1}, M_{u2}, M_{u3}, M_{u4}, M_{u5}, M_{u6}$ - moments of frictional forces on the shafts, respectively; $M_{\kappa 2}, M_{\kappa 3}, M_{\kappa 6}$ - torques of the resistance forces generated by the lump being crushed in the tractor and pile drum.

Mechanical characteristics of the first shaft:

$$M_{ю_1} = M_{юб} - K_ю \dot{\varphi}, \quad (7)$$

where $M_{юб}$ is the value of the initial torque on the shaft, Nm ; $K_ю$ – the slope coefficient of the mechanical characteristic, i.e. $tg\alpha_ю = M_{ю_1} / \Delta\dot{\varphi}_1; \dot{\varphi}_1$ – the angular velocity of the first shaft.

Now for each mass, we determine the integrals of Lagrange's structured equations. The specific derivatives of the kinetic energy with respect to each generalized coordinate velocity and the derivative with respect to additional time are:

$$\begin{aligned}
&\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}_1}\right) = \left[\left(J_{в_2} + J_{k_2} + J_{ю_1}\right) + \left(J_{в_1} + J_{k_1}\right)\cdot U_{1p}^2\right]\cdot \frac{d^2\varphi_1}{dt^2};\\
&\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}_2}\right) = \left(J_{в_5} + J_{ю_2} + J_{ю_3} + J_{T_1} + J_{T_2} + J'_{T_1}\cdot U_{T_1}^2\right)\cdot \frac{d^2\varphi_2}{dt^2};\\
&\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}_3}\right) = \left(J_{в_6} + J_{ю_4} + J_{T_2} + J'_{T_2}\cdot U_{T_2}^2\right)\cdot \frac{d^2\varphi_3}{dt^2};\\
&\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}_4}\right) = \left(J_{в_7} + J_{ю_5} + J_{ю_6}\right)\cdot \frac{d^2\varphi_4}{dt^2}; \qquad \frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}_5}\right) = J_{ю_7}\cdot \frac{d^2\varphi_5}{dt^2};\\
&\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}_6}\right) = \left(J_{в_3} + J_{б} + J_{ю_7}\right)\cdot \frac{d^2\varphi_6}{dt^2}.
\end{aligned} \quad (8)$$

The specific derivatives of the total potential energy of the system in terms of generalized coordinates are as follows:

$$\begin{aligned}
&\frac{\partial \Pi}{\partial \varphi_1} = C_1(\varphi_1 - U_{12}\varphi_2); \quad \frac{\partial \Pi}{\partial \varphi_2} = -U_{12}C_1(\varphi_1 - U_{12}\varphi_2) + C_2(\varphi_2 - U_{23}\varphi_3) + C_2(\varphi_2 - U_{24}\varphi_4);\\
&\frac{\partial \Pi}{\partial \varphi_3} = -U_{23}C_2(\varphi_2 - U_{23}\varphi_3) + C_2(\varphi_3 - U_{34}\varphi_4);\\
&\frac{\partial \Pi}{\partial \varphi_4} = -U_{24}C_2(\varphi_2 - U_{24}\varphi_4) - U_{34}C_2(\varphi_3 - U_{34}\varphi_4) + C_3(\varphi_4 - U_{45}\varphi_5);\\
&\frac{\partial \Pi}{\partial \varphi_5} = -U_{45}C_3(\varphi_4 - U_{45}\varphi_5) + C_4(\varphi_5 - U_{56}\varphi_6); \frac{\partial \Pi}{\partial \varphi_6} = -U_{56}C_4(\varphi_5 - U_{56}\varphi_6).
\end{aligned} \quad (9)$$

Similarly, we obtain derivatives of the dissipation function with respect to the generalized coordinate velocities:

$$\begin{aligned}
&\frac{\partial \Phi}{\partial \dot{\varphi}_1} = в_1\left(\frac{d\varphi_1}{dt} - U_{12}\frac{d\varphi_2}{dt}\right);\\
&\frac{\partial \Phi}{\partial \dot{\varphi}_2} = -в_1 U_{12}\left(\frac{d\varphi_1}{dt} - U_{12}\frac{d\varphi_2}{dt}\right) + в_2\left(\frac{d\varphi_2}{dt} - U_{23}\frac{d\varphi_3}{dt}\right) + в_2\left(\frac{d\varphi_2}{dt} - U_{24}\frac{d\varphi_4}{dt}\right);\\
&\frac{\partial \Phi}{\partial \dot{\varphi}_3} = -U_{23}в_2\left(\frac{d\varphi_2}{dt} - U_{23}\frac{d\varphi_3}{dt}\right) + в_2\left(\frac{d\varphi_3}{dt} - U_{34}\frac{d\varphi_4}{dt}\right);\\
&\frac{\partial \Phi}{\partial \dot{\varphi}_4} = -U_{24}в_2\left(\frac{d\varphi_2}{dt} - U_{24}\frac{d\varphi_4}{dt}\right) - U_{34}в_2\left(\frac{d\varphi_3}{dt} - U_{34}\frac{d\varphi_4}{dt}\right) + U_{45}в_3\left(\frac{d\varphi_4}{dt} - U_{45}\frac{d\varphi_5}{dt}\right);\\
&\frac{\partial \Phi}{\partial \dot{\varphi}_5} = -U_{45}в_3\left(\frac{d\varphi_4}{dt} - U_{45}\frac{d\varphi_5}{dt}\right) + в_4\left(\frac{d\varphi_5}{dt} - U_{56}\frac{d\varphi_6}{dt}\right); \quad \frac{\partial \Phi}{\partial \dot{\varphi}_6} = -U_{56}в_4\left(\frac{d\varphi_5}{dt} - U_{56}\frac{d\varphi_6}{dt}\right).
\end{aligned} \quad (10)$$

The obtained (8), (9), (10) and (6) and (7) into the equation (1) for each mass, we create a system of differential equations representing the machine assembly of the chip grinding equipment with a belt element leading star as follows:

$$M_{ю_1} = M_{юб} - K_{IO}\frac{d\varphi_1}{dt}\cdot$$

$$\left[\left(J_{в_2} + J_{к_2} + J_{ю_1}\right) + \left(J_{в_1} + J_{к1}\right)U_{1p}^2\right]\cdot\frac{d^2\varphi_1}{dt^2} = M_{ю_1} - M_{U1} - C_1(\varphi_1 - U_{12}\varphi_2) - в_1\left(\frac{d\varphi_1}{dt} - U_{12}\frac{d\varphi_2}{dt}\right);$$

$$\left(J_{в_5} + J_{ю_2} + J_{ю_3} + J_{T_1} + J'_{T_1}\cdot U_{T_1}^2\right)\frac{d^2\varphi_2}{dt^2} = U_{12}C_1(\varphi_1 - U_{12}\varphi_2) + в_1 U_{12}\left(\frac{d\varphi_1}{dt} - U_{12}\frac{d\varphi_2}{dt}\right) -$$

$$- в_2\left(\frac{d\varphi_2}{dt} - U_{23}\frac{d\varphi_3}{dt}\right) - в_2\left(\frac{d\varphi_2}{dt} - U_{24}\frac{d\varphi_4}{dt}\right) - C_2\left[(\varphi_2 - U_{23}\varphi_3) + (\varphi_2 - U_{24}\varphi_4)\right] - \left(M_{U_2} + M_{K_2}\right);$$

$$\left(J_{в_6} + J_{ю_4} + J_{T_2} + J'_{T_2}\cdot U_{T_2}^2\right)\frac{d^2\varphi_3}{dt^2} = U_{23}C_2(\varphi_2 - U_{23}\varphi_3) - C_2(\varphi_3 - U_{34}\varphi_4) +$$

$$+ U_{23}в_2\left(\frac{d\varphi_2}{dt} - U_{23}\frac{d\varphi_3}{dt}\right) - в_2\left(\frac{d\varphi_3}{dt} - U_{34}\frac{d\varphi_4}{dt}\right) - \left(M_{U_3} + M_{K_3}\right);$$

$$\left(J_{â_7} + J_{þ_5} + J_{þ_6}\right)\frac{d^2\varphi_4}{dt^2} = U_{24}C_2(\varphi_2 - U_{24}\varphi_4) + U_{34}C_2(\varphi_3 - U_{34}\varphi_4) - C_3(\varphi_4 - U_{45}\varphi_5) - M_{u4};$$

$$J'_{ю_7}\frac{d^2\varphi_5}{dt^2} = U_{45}C_3(\varphi_4 - U_{45}\varphi_5) - C_4(\varphi_5 - U_{56}\varphi_6) + U_{45}в_3\left(\frac{d\varphi_4}{dt} - U_{45}\frac{d\varphi_5}{dt}\right) - в_4\left(\frac{d\varphi_5}{dt} - U_{56}\frac{d\varphi_6}{dt}\right) +$$

$$+ U_{24}в_2\left(\frac{d\varphi_2}{dt} - U_{24}\frac{d\varphi_4}{dt}\right) + U_{34}в_2\left(\frac{d\varphi_3}{dt} - U_{34}\frac{d\varphi_4}{dt}\right) - U_{45}в_3\left(\frac{d\varphi_4}{dt} - U_{45}\frac{d\varphi_5}{dt}\right) - M_{U_5};$$

$$\left(J_{в_9} + J_6 + J''_{ю_7}\right)\frac{d^2\varphi_6}{dt^2} = U_{56}C_4(\varphi_5 - U_{56}\varphi_6) + U_{56}в_4\cdot\left(\frac{d\varphi_5}{dt} - U_{56}\frac{d\varphi_6}{dt}\right) - \left(M_{U6} + M_{к_6}\right). \quad (11)$$

(11) to obtain the numerical solution of the system of differential equations, the Adams method of the MathCAD 14 program was used. In this case, the values of the parameters included in (11) were determined by calculating the corresponding values for the chip grinding equipment. The constant parameters in the mechanical characteristics were taken as follows:

$$M_{ю.б} = 559{,}77\ Nm;\ K_{ю} = tg\alpha_{ю} = 147{,}3.$$

We determined the calculation of moments of inertia of 6 rotating masses using the existing method [1-5]. According to this method, the moments of inertia of each shaft were divided into parts of constant diameter. The following expression was used for each part:

$$J = \frac{\gamma\pi}{32}ld^4, \quad (12)$$

here γ - material density; kg/m^3; g – gravitational acceleration, 9.81 m/s²; l and d – length and diameter of the part.

Complex shaped elements calculated using the data table given in [3] are presented in Table 1.

Table 1. The values of the moment of inertia of the rotating masses, $kg \cdot m^2$

$J_{в_1}$	$J_{в_2}$	$J_{в_5}$	$J_{в_6}$	$J_{в_7}$	$J_{в_9}$	J_{k_1}	J_{k_2}	$J_{ю_1}$
0.0088	0.0064	0.0081	0.0081	0.0051	0.0062	0.0064	0.0073	0.0034
$J_{ю_2}$	$J_{ю_3}$	$J_{ю_4}$	$J_{ю_5}$	$J_{ю_6}$	$J'_{ю_7}$	$J''_{ю_7}$	$J_б$	J_{T_1}
0.0034	0.0034	0.0034	0.0034	0.0034	0.0021	0.0013	0.0025	0.0079
J'_{T_1}	J_{T_2}	J_{1k}	J_1	J_2	J_3	J_4	J_5	J_6
0.0085	0.0026	0.0086	0.0388	0.0232	0.0233	0.0127	0.0085	0.054

The transmission ratios in the transmission mechanisms of the ground handling unit were taken at the following values:

$$U_{юв} = 3{,}35;\ U_{1pc} = 1{,}7;\ U_{12} = 1{,}0;\ U_{23} = 1{,}0;\ U_{T1} = 1{,}0;$$
$$U_{T2} = 1{,}0;\ U_{24} = 1{,}0;\ U_{34} = 1{,}0;\ U_{45} = 1{,}2;\ U_{56} = 1{,}0.$$

The given uniformity of the chain in the chain drive was calculated by the following formula [10]:

$$C_з = \frac{1}{l_3} = \frac{R^2 \cdot t \cdot F}{K_з \cdot l_з}, Nm/rad, \tag{13}$$

here is l_3 – the coefficient of tendency to deformation (a quantity opposite to unity); R – initial circle radius of the sprocket on the driven shaft, *m;* F –chain roller cross-sectional surface, *mm²;* $F = d \cdot l;$ d – diameter of the shaft of the chain link, *m;* l – joint length, *m;* $K_з$ - coefficient of tendency to deformation of the chain; t - chain pitch, *mm*.

The calculated length of chain links was determined by the following formula [4]:

$$l_3 = L_3 \cdot t = 2 \cdot \frac{A_з}{t} + 0{,}5(z_1 + z_2) + \frac{(z_1 + z_2)^2}{4\pi^2 (A_з / t)} \cdot t, m, \tag{14}$$

$z_1 = z_2$ if, $l_ц = 2l'_z + z_1 t,\ m$ here is $A_з$ – the distance between the axes, m ; z_1, z_2 – number of sprocket teeth. Damping coefficient according to [3]. Calculated by the following expression:

$$в_3 = \frac{\psi \cdot C_3}{2\pi (2\pi / T)}. \tag{15}$$

The values of the parameters of the machine unit elements were determined by calculating:

$$z_1 = z_2 = z_3 = z_4 = z_5 = 15;\ z_6 = 21;\ z_7 = 25;\ в_1 = в_2 = в_3 = в_4 = 5{,}12\, Nms/rad;$$
$$C_1 = C_2 = C_3 = C_4 = 643\, Nm/rad;\ M_{U_1} = 10{,}7Nm;\ M_{U_2} = 14{,}3Nm;\ M_{U_3} = 14{,}3Nm; M_{U_4} = 6{,}2Nm;$$
$$M_{U_5} = 2{,}4Nm;\ M_{U_6} = 16{,}2Nm; M_{K_2} = 22{,}8Nm; M_{K_3} = 22{,}8Nm.$$

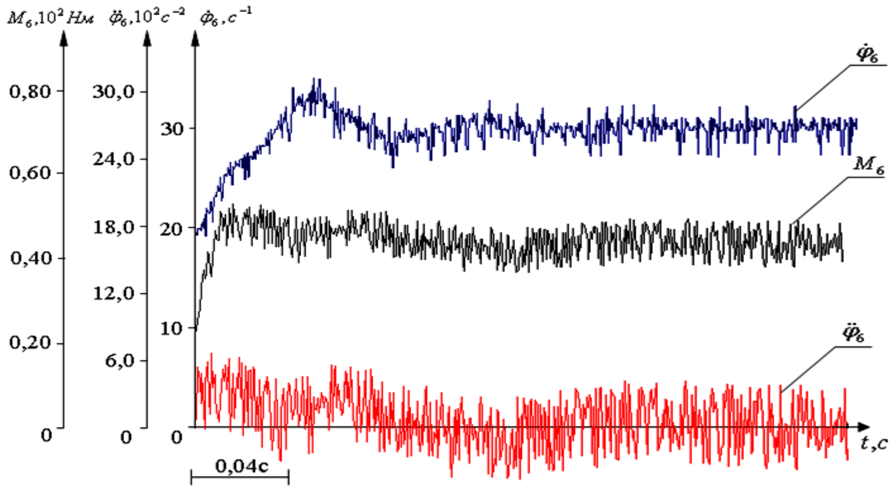
The moment of resistance coming from the cuttings to the piled drum, based on the experimental results, is determined by the following expression:

$$M_{K_6} = M_y \pm \delta M_y, \tag{16}$$

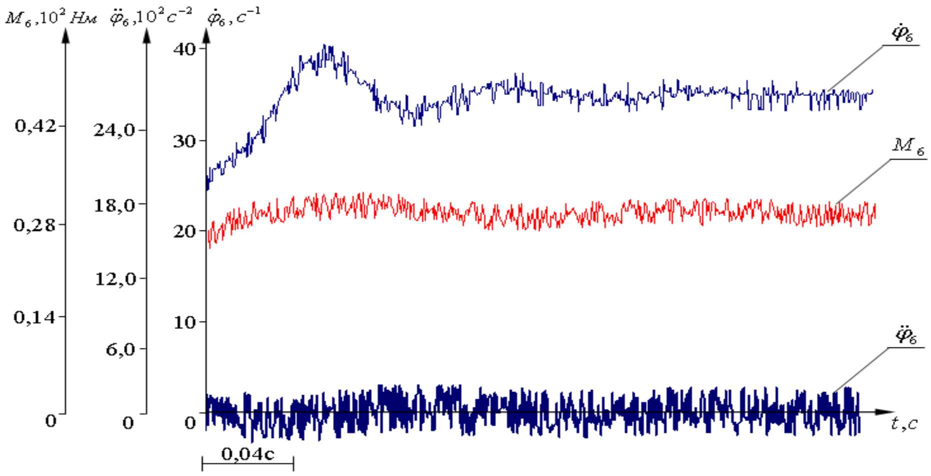
where $M_{\dot{y}}$ – the average value of the moment of resistance coming from the cut (respectively, it was obtained based on the results of the experiment based on the performance); $\delta M_{\dot{y}}$ –random values of the average values of the moment of resistance.

II. Results. The solution of the system of differential equations (11) obtained for the machine unit was carried out numerically using the MathCAD program. In this case , it was done by putting the experimentally obtained values of n in (16). δM_y In the solution of the problem, the angular acceleration angular velocity of each of the six rotating masses and the laws of change of the moment in them were obtained over time. In the solution of the problem, the initial conditions were obtained as follows: $t = 0$ when $\dot{\varphi}_1 = 0{,}31c^{-1}$; $M_{ю_1} = 1{,}32 \cdot 10^2\, Nm.$

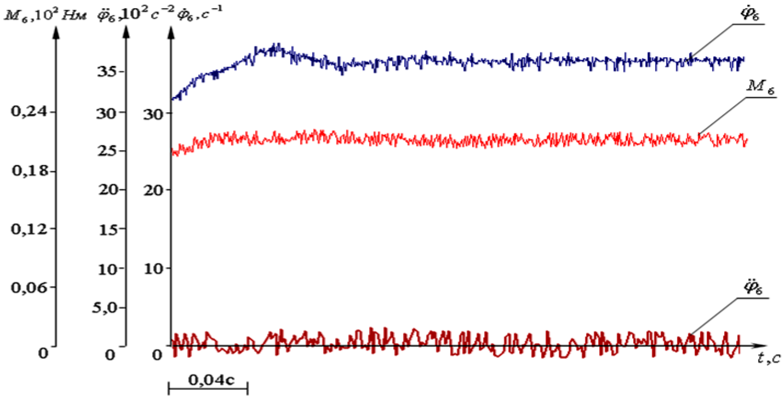
Random components of acceleration, velocity and torque values were also taken into account in the solution of the problem in the above-mentioned method. Figure 3 shows the patterns of angular acceleration, angular velocity and torque on the shaft of aggregate pile drum with respect to time.



$a - M_{K6} = 32 \pm 3{,}0 Nm$



$b - M_{K6} = 20 \pm 2{,}3 Nm$

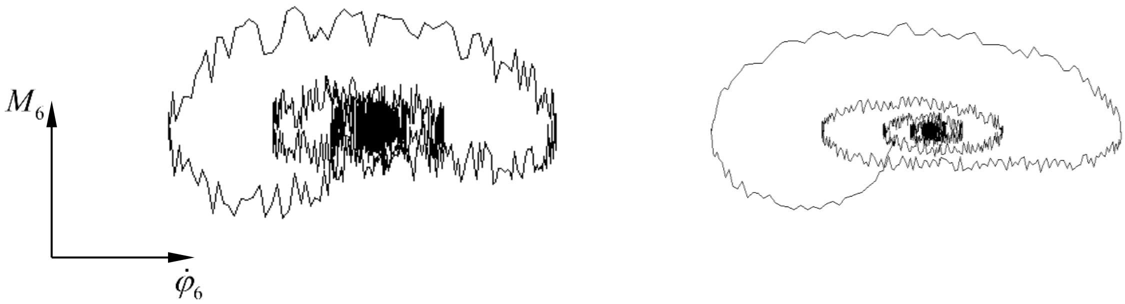


$c - M_{K6} = 12 \pm 1{,}5Nm$

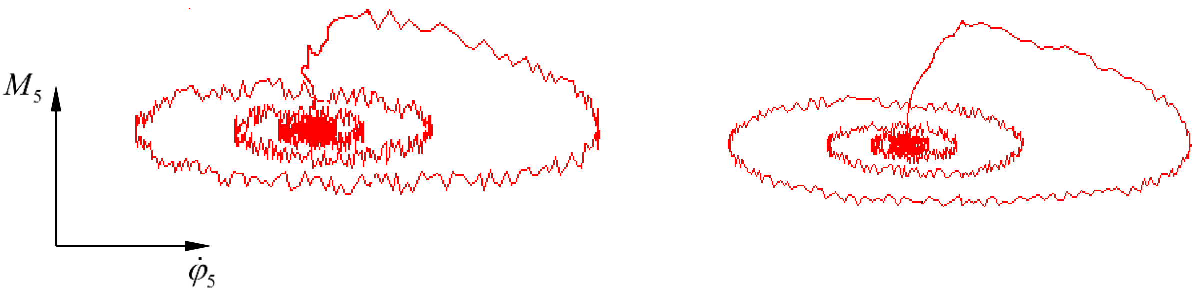
Figure 3. Angular velocity and acceleration of the piled drum of the tillage aggregat, the laws of change of torques on the shaft under the influence of the moment of resistance coming from the tillage.

The laws of change of the results are presented when $\dot{\varphi}_6, \ddot{\varphi}_6, M_6$ the moment of resistance coming from the grinding piece is on average 12 Nm and a random component $\pm 1{,}5Nm$(Fig. 3. c), as well as for the variants $20Nm \pm 2{,}3Nm$with (Fig. 3. b) and $32Nm \pm 3{,}0$ (Fig. 3. a)From the obtained laws, it can be seen that with the increase of the resistance moment coming from the cutting, the angular speed of the pile drum decreases in a non-linear law, therefore, the value of the torque on the shaft increases. For example, $M_{\kappa 6} = 15Nm$ if the average is the angular speed $36{,}1s^{-1}$, when the average value of the resistance moment $60Nm$is, the angle $32{,}6s^{-1}$is reduced to the speed. In this case, the torque on the pile drum shaft $22{,}3Nm$increases from to. $64{,}7Nm$ So, in order for the unit to work in normal mode ($\dot{\varphi}_6 = 35s^{-1}$ and $M_6 = 42{,}5Nm$) $38 - 45Nm$ is required not to exceed the moment of resistance coming from the cut.

The mechanical characteristics of each shaft of the tiller unit determine the sufficient priority of its movement. Due to the mechanical characteristics of the power take-off shaft, the drive is not $0{,}31s^{-1}$taken from the ting point. Therefore, the mechanical characteristics of subsequent masses also start from a certain angular velocity. That is, the process of taking action is not fully reflected. The mechanical characteristics of the drive star with pile drum and chain transmission are presented in Fig. 4. The analysis of the mechanical characteristics shows that the actions of the pile drum and the material guide sprocket in the working mode are the priority, and the shape changes depending on the values of the resistance moment coming from cutting the piece. In particular, $12 \pm 1{,}5Nm$ when there is a resistance moment, the vibration values in the characteristics (see Fig. 4. *a, b) are small.* But $32 \pm 3{,}0Nm$when there is a resistance moment, the vibration patterns are significant. When the resistance torque is increased, the forward movement is disturbed and the system can stop.



$a - M_{K6} = 32 \pm 3{,}0 Nm$ $\qquad$ $M_{K6} = 12 \pm 1{,}5 Nm$



$b - M_{K6} = 32 \pm 3{,}0 Nm$ $\qquad$ $M_{K6} = 12 \pm 1{,}5 Nm$

a – pile drum mechanical characteristics; b - mechanical characteristics of the component driven star of the chain transmission.

4 . Mechanical characteristics of pile drum and chain drive sprocket.

Conclusions. Based on the numerical solution of the system of motion equations of the earth-cultivating aggregate equipment, the laws of speed, acceleration and moment changes of the pile drum and chain drive star were obtained depending on the resistance moment coming from the cutting. Plots of resistance torque dependence of the average values of the pile drum and the driven star angular speed, acceleration and torque range of oscillation were constructed.

Recommended parameters:

$J_5 = 0{,}015 - 0{,}018\, kgm^2;\ J_6 = 0{,}055 - 0{,}064\, kgm^2; C_4 = 450 - 480\, Nm/rad; в_4 = 3{,}2 - 3{,}8 Nms/rad.$

The mechanical characteristics of the pile drum, chain transmission sprockets and distributor shafts were determined in the regimes of the change limits of the resistance moment, and the extreme limit regimes were determined.

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