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Dynamic Analysis of the Transfer Mechanism of the Soil Processing Unit

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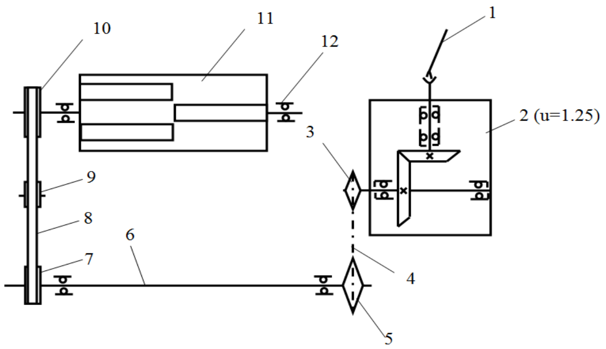
ABSTRACT: One section of the seedbed preparation machine was considered as a four-mass machine unit. Using the Lagrange equations of the second kind, an equation of motion of the masses of a machine unit is compiled. As a result, the regularities of the change in the turnover and torque of the milling drum shaft from the resistance of the soil, the change in the coefficient of unevenness from the moment of inertia and the additional angle of rotation of the driven composite gear pulley were determined.

KEYWORDS. Torque, resistance, soil, lump, stiffness, pulley, belt, moment of inertia, cutter, drum, reducer.

I. INTRODUCTION

Among agricultural machinery, tillage machines play an important role. This is because a large amount of energy is expended in the process of cultivating the soil. Therefore, it is important to study the transmission mechanisms of active working bodies that process the soil before planting.

In the dynamic analysis of a gear belt milling machine with a drive pulley in the transmission mechanism, which is processed before planting in the soil, one section of the machine (Fig. 1) was considered as a four-mass machine unit (Fig. 2). Where I is the total mass of the power take-off shaft applied to the gearbox output shaft, II is the mass of the intermediate shafts, and III is the mass of the driven gear pulley flange; IV-driven gear pulley base, milling drum and shaft bearing mass. In this case, we consider the power shaft in the machine unit as a source of motion, and bring it to the output shaft of the reducer. The remaining masses were selected to fit the shafts separated by chain and gear belt extensions.



1-cardan shaft, 2-single-taper conical reducer, 3-drive sprocket, 4-chain, 5-drive sprocket, 6-intermediate shaft, 7-drive gear pulley, 8-gear belt, 9-tension roller, 10-component driven gear pulley, 11-milling drum, 12-bearing

Fig. 1. Kinematic diagram of a milling working body with a gear belt drive with a drive pulley in the transmission mechanism

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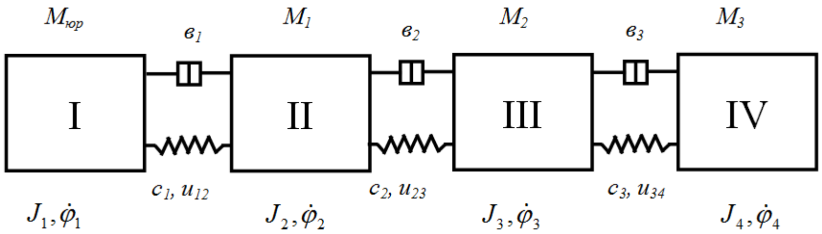

Fig.2. Calculation scheme of a four-mass machine unit

II. LITERATURE SURVEY

It is known from the calculation scheme shown in Figure 2 that four masses rotate, so that 4 generalized coordinates can be determined. We use Langrage's type II equation to derive the equation of motion of a four-mass machine unit from a single section of a machine [1; 2; 3]

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

where φ_i is the generalized coordinates of the masses for a system of mass i, i.e. the angles of rotation; T is the total kinetic energy of a system of mass i, Nm; P is the total potential energy of the system, Nm; The dissipative function of the relay in Φ-flexible and flexible joints, Nm; M_i (φ_i) is the moment of the total force acting on the masses of the system, Nm.

The expression for determining the total kinetic energy according to the calculation scheme of the machine unit shown in Figure 2 is as follows

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

where φ_1, φ_2, φ_3, φ_4 are the generalized coordinates of the rotating masses of the machine unit, ie the angles of rotation; J_1, J_2, J_3, J_4 - moments of inertia of masses, kgm^2.

The moment of inertia of the mass is determined by the following expressions

$$J_1 = (J_{к1} + \frac{J_{к2}}{u_p^2}) + J_{ю1}; \quad J_2 = J_{о.в} + J_{ю2} + J_{ш1};$$

$$J_3 = J_{шг2}; \quad J_4 = J_{ф} + J_{ф.в} + J_{ш2}, \quad (3)$$

where $J_{о.в}$, $J_{ф.в}$ - moments of inertia of rotating shafts, respectively, kgm^2; J_{k1}, J_{k2} - moments of inertia of reducer conical gears, kgm^2; $J_{ю1}$, $J_{ю2}$ - corresponding moments of inertia of chain extension stars, kgm^2; $J_{ф}$- the moment of inertia of the milling drum, kgm^2; $J_{ш1}$ – moments of inertia of the drive gear pulley, kgm^2; $J_{ш2}$ - moments of inertia of the drive gear pulley flange, kgm^2; $J_{ш2}$ – moments of inertia of the drive gear pulley, kgm^2; u_p –the number of transmissions of the conical reducer.

The four-mass machine unit in question used 1 chain drive and 1 gear belt drive. Taking them into account, the expression for determining the total potential energy for a machine unit is as follows

$$\Pi = \frac{1}{2}\left[C_1(\varphi_1 - U_{12}\varphi_2)^2 + C_2(\varphi_2 - U_{23}\varphi_3)^2 + C_3(\varphi_3 - U_{34}\varphi_4)^2, \quad (3)$$

where C_1, C_2 – the coefficients of elasticity of the chain and the belt, respectively, Nm / rad; C_3 – the coefficient of elasticity of the flexible element in the drive gear pulley, Nm / rad; U_{12}, U_{23}, U_{34} – transmission ratios between rotating masses, respectively.

For the system, we write the expression for the dissipation function of the relay as follows

$$\Phi = \frac{1}{2}\left\{в_1\left(\frac{d\varphi_1}{dt} - U_{12}\frac{d\varphi_2}{dt}\right)^2 + в_2\left(\frac{d\varphi_2}{dt} - U_{23}\frac{d\varphi_3}{dt}\right)^2 + в_3\left(\frac{d\varphi_3}{dt} - U_{34}\frac{d\varphi_4}{dt}\right)^2, \quad (4)$$

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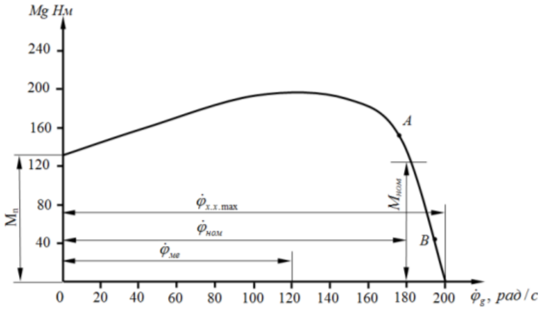
where $в_1$, $в_3$ – the dissipation coefficients of the chain and the belt, respectively; $в_3$ – the dissipation coefficient of the elastic element in the structure of the driven gear.
Generalized torques in the machine unit

$$M_1(\varphi_1) = M_{ю1} - M_{u1};\ M_2(\varphi_2) = -M_{u2};\ M_3(\varphi_3) = -M_{u3};$$
$$M_4(\varphi_4) = -(M_{u4} + M_\phi). \qquad (5)$$

where M_{u1}, M_{u2}, M_{u3}, M_{u4} – the moments of friction forces on the shafts, respectively, Nm; M_ϕ – the torque of the resistance forces coming from the ground and the ground in the milling drums, Nm.
If we take into account the number of blades in the milling drum and the random values of the moment of resistance, M_ϕ equal to

$$M_\phi = M_\phi^1 \pm \Delta M_\phi \quad sin(4\omega t). \qquad (6)$$

where M_ϕ^1 – the value of Nm, not taking into account the random values of the moments of resistance forces coming from the grinding blocks in the milling drum; ΔM_ϕ – the random value of the moments of the resistance forces, Nm.
To obtain the driving torque of the base tractor on the power shaft, we obtain the mechanical characteristics of the shaft, i.e. the law of dependence of the driving torque $M_ю$ on the angular velocity of the shaft MTZ-80 for a tractor with an internal combustion engine capacity of 500 kVt and a rotational frequency of 1800 min^{-1}.
Tractor drive operating modes mainly include stable start, steady motion and stop processes. In the tractor, the movement from the drive is done through the clutch. Therefore, the movement of the power take-off shaft operates mainly in the steady-state mode. In this case, the AB on the graph of the mechanical and static characteristics of the drive (Figure 3) the operating range is expected to be in the range of points.



$\dot{\varphi}_{ю.з}$ - angular velocity in the unloaded state; $\dot{\varphi}_н$ - nominal speed; $M_{ном}$ - rated torque of the drive

Fig. 3. Graph of mechanical-static characteristics of the internal combustion engine of the tractor

In this case, the mechanical characteristics of the power take-off shaft are expected to change along a straight line $M_{юк} = f(\dot{\varphi}_1)$ for the operating condition. In this case, if the rotational frequency of the crankshaft of the working drive is 1800 min^{-1}, the rotational frequency of the power take-off shaft is 540 min^{-1}, so the number of their mutual transmissions is 3.15 [4].
Given the above, the mechanical characteristics of the power take-off shaft were seen for the operating mode (Fig. 4). It was taken as a straight line (when brought to the 1st shaft). In this mechanical characteristic, zone A_1B_1 is the main operating mode zone, and it corresponds to the *AB* operating mode zone of the drive mechanical characteristic in Figure 4. In this case, the mechanical characteristics of the first shaft:

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

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

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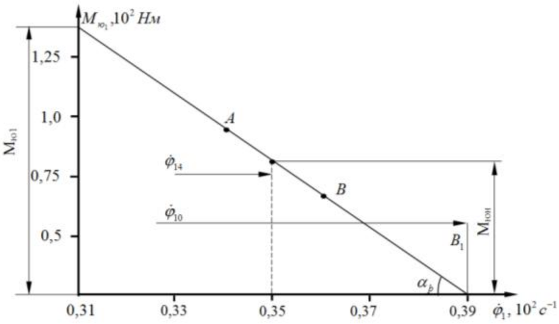

Fig. 4. Mechanical characteristics of the power take-off shaft for the operating mode

The torque is determined as follows

$$M_1=\frac{M_{\max}+M_{ном}}{2} \tag{8}$$

where M_{max}–the maximum torque of the engine, Nm; $M_{ном}$ –the rated torque of the engine, Nm.

$$M_{\max}=kM_{ном} \text{ and } M_{ном}=\frac{N_{ном}}{\omega_{ном}} \tag{9}$$

where $N_{ном}$ –the nominal power of the power take-off shaft, kW; $\omega_{ном}$ – nominal angular velocity of the power take-off shaft, rad / s.

$$M_1=\frac{kM_{ном}+M_{ном}}{2}=\frac{M_{ном}(k+1)}{2} \tag{10}$$

Since the nominal power of the MTZ-80 tractor engine $P_{ном.ю}$=60 kW and the nominal number of revolutions of the power shaft is $n_{ном.ю}$=540 ayl/min, the torque of the power shaft is equal to

$$M_{ном.ю}=\frac{30P_{ном.ю}}{\pi n_{ном.ю}}=\frac{30\cdot 60\cdot 10^3}{3{,}14\cdot 540}=1061{,}6 \text{ Nm.}$$

Moment of resistance in the milling drum $M_ф$=76,9 Nm.
We now define the additions in Lagrange's type II equation for each mass. The specific products obtained from the kinetic energy for each generalized coordinate velocity and the products obtained for the additional time are as follows:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial\dot{\varphi}_1}\right)=\left[(J_{\kappa1}+\frac{J_{\kappa2}}{u_p^2})+J_{ю1}\right]\cdot\frac{d^2\varphi_1}{dt^2};\quad \frac{d}{dt}\left(\frac{\partial T}{\partial\dot{\varphi}_2}\right)=(J_{в2}+J_{ю2}+J_{ш1})\cdot\frac{d^2\varphi_2}{dt^2};$$
$$\frac{d}{dt}\left(\frac{\partial T}{\partial\dot{\varphi}_3}\right)=\frac{J_{шг2}d^2\varphi_3}{dt^2};\quad \frac{d}{dt}\left(\frac{\partial T}{\partial\dot{\varphi}_4}\right)=\left(J_ф+J_{в3}+J_{ш2}\right)\cdot\frac{d^2\varphi_4}{dt^2}. \tag{11}$$

The specific derivatives from the total potential energy of the system on the generalized coordinates are as follows:

$$\frac{\partial П}{\partial\varphi_1}=C_1(\varphi_1-U_{12}\varphi_2);\quad \frac{\partial П}{\partial\varphi_2}=-U_{12}C_1(\varphi_1-U_{12}\varphi_2)+C_2(\varphi_2-U_{23}\varphi_3);$$
$$\frac{\partial П}{\partial\varphi_3}=-U_{23}C_2(\varphi_2-U_{23}\varphi_3)+C_3(\varphi_3-U_{34}\varphi_4);\quad \frac{\partial П}{\partial\varphi_4}=-U_{34}C_3(\varphi_3-U_{34}\varphi_4). \tag{12}$$

Similarly, from the dissipation function we obtain the products of the generalized coordinate velocities:

$$\frac{\partial\Phi}{\partial\dot{\varphi}_1}=в_1\left(\frac{d\varphi_1}{dt}-U_{12}\frac{d\varphi_2}{dt}\right);\quad \frac{\partial\Phi}{\partial\dot{\varphi}_2}=-в_1U_{12}\left(\frac{d\varphi_1}{dt}-U_{12}\frac{d\varphi_2}{dt}\right)+в_2\left(\frac{d\varphi_2}{dt}-U_{23}\frac{d\varphi_3}{dt}\right);$$

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$$\frac{\partial \Phi}{\partial \dot{\varphi}_3} = -U_{23} в_2 \left(\frac{d\varphi_2}{dt} - U_{23} \frac{d\varphi_3}{dt} \right) + в_3 \left(\frac{d\varphi_3}{dt} - U_{34} \frac{d\varphi_4}{dt} \right); \quad \frac{\partial \Phi}{\partial \dot{\varphi}_4} = -U_{34} в_3 \left(\frac{d\varphi_3}{dt} - U_{34} \frac{d\varphi_4}{dt} \right). \quad (13)$$

Substituting the resulting expressions (7), (11), (12) and (13) and expressions (4) and (5) for each mass into equation (1), We create a system of differential equations as follows:

$$M_{юр} = M_{юб} - K_{Ю} \frac{d\varphi_1}{dt};$$

$$\left[(J_{к1} + \frac{J_{к2}}{u_p^2}) + J_{ю1} \right] \cdot \frac{d^2\varphi_1}{dt^2} = M_{ю_1} - M_{и1} - C_1(\varphi_1 - U_{12}\varphi_2) - в_1 \left(\frac{d\varphi_1}{dt} - U_{12} \frac{d\varphi_2}{dt} \right);$$

$$(J_{о.в} + J_{ю2} + J_{ш1}) \cdot \frac{d^2\varphi_2}{dt^2} = U_{12} C_1 (\varphi_1 - U_{12}\varphi_2) - C_2(\varphi_2 - U_{23}\varphi_3) +$$

$$+ в_1 U_{12} \left(\frac{d\varphi_1}{dt} - U_{12} \frac{d\varphi_2}{dt} \right) - в_2 \left(\frac{d\varphi_2}{dt} - U_{23} \frac{d\varphi_3}{dt} \right) - M_{и2};$$

$$\frac{J_{ше2} d^2\varphi_3}{dt^2} = U_{23} C_2 (\varphi_2 - U_{23}\varphi_3) - C_3(\varphi_3 - U_{34}\varphi_4) +$$

$$+ U_{23} в_2 \left(\frac{d\varphi_2}{dt} - U_{23} \frac{d\varphi_3}{dt} \right) - в_3 \left(\frac{d\varphi_3}{dt} - U_{34} \frac{d\varphi_4}{dt} \right) - M_{и_3};$$

$$\left(J_ф + J_{ф.в} + J_{ш2} \right) \cdot \frac{d^2\varphi_4}{dt^2} = U_{34} C_3 (\varphi_3 - U_{34}\varphi_4) +$$

$$+ U_{34} в_3 \left(\frac{d\varphi_3}{dt} - U_{34} \frac{d\varphi_4}{dt} \right) - (M_{и4} + M_ф) \quad (14)$$

III. RESULTS

The constant parameters of the mechanical characteristic were accepted as follows [5]:

$$M_{ю.б} = 1061{,}6 \ \ Нм; \ K_ю = tg\alpha_ю = 147{,}3 \ .$$

We determine the moments of inertia of rotationally moving masses using the existing method [6].

Asterisk on the gearbox output shaft: mass $m_{ю1}$ = 0.8 kg; outer circle radius $R_{ю1}$= 0.0545 m; inner circle radius $r_{ю1}$= 0.015 m;

$$J_{ю1} = \frac{1}{4} m_{ю1} \left(R_{ю1}^2 + r_{ю1}^2 \right) = \frac{1}{4} \cdot 0{,}8 \cdot \left(0{,}0545^2 + 0{,}015^2 \right) = 0{,}0006 \ кгм^2.$$

Asterisk on the intermediate shaft: mass $m_{ю2}$= 1.1 kg; outer circle radius $R_{ю2}$= 0.063 m; inner circle radius $r_{ю2}$= 0.0175 m;

$$J_{ю2} = \frac{1}{4} m_{ю1} \left(R_{ю2}^2 + r_{ю2}^2 \right) = \frac{1}{4} \cdot 1{,}1 \cdot \left(0{,}063^2 + 0{,}0175^2 \right) = 0{,}00117 \ кгм^2.$$

Gear pulley on the drive shaft of the milling drum: mass $m_{ш1}$= 1.05 kg; outer circle radius $R_{ш1}$= 0.063 m; inner circle radius $r_{ш1}$= 0.01 m;

$$J_{ш1} = \frac{1}{4} m_{ш1} \left(R_{ш1}^2 + r_{ш1}^2 \right) = \frac{1}{4} \cdot 1{,}05 \cdot \left(0{,}063^2 + 0{,}01^2 \right) = 0{,}001068 \, кгм^2.$$

Asterisk on the milling drum shaft: mass $m_{ш2}$= 0.9 kg; outer circle radius $R_{ш2}$= 0.063 m; inner circle radius $r_{ш2}$= 0.01 m;

$$J_{ш2} = \frac{1}{4} m_{ш2} \left(R_{ш2}^2 + r_{ш2}^2 \right) = \frac{1}{4} \cdot 0{,}9 \cdot \left(0{,}063^2 + 0{,}01^2 \right) = 0{,}0009155 \, кгм^2.$$

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Moments of inertia of shafts:
Intermediate shafts: mass $m_{о.в}$ = 8.0 kg; radius $r_{о.в}$ = 0.02 m;

$$J_{о.в} = \frac{1}{2} m_{о.в} r_{о.в}^2 = \frac{1}{2} \cdot 8 \cdot 0{,}02^2 = 0{,}0016 \ кгм^2.$$

Milling drum shaft: mass $m_{ф.в}$ = 5.0 kg; radius $r_{ф.в}$ = 0.015 m;

$$J_{ф.в} = \frac{1}{2} m_{ф.в} r_{ф.в}^2 = \frac{1}{2} \cdot 5 \cdot 0{,}015^2 = 0{,}00056 \ кгм^2.$$

Moment of inertia of the milling drum disk: mass $m_{ф.д}$ = 3.5 kg;
Outer circle radius $R_{ф.д}$ = 0.12 m; inner circle radius $r_{ф.д}$ = 0.015 m;

$$J_{ф.д} = \frac{1}{4} m_{ф.д} \left(R_{ф.д}^2 + r_{ф.д}^2\right) = \frac{1}{4} \cdot 3{,}5 \cdot \left(0{,}12^2 + 0{,}015^2\right) = 0{,}0128 \ кгм^2.$$

Since the milling drum has two discs, we multiply the output by two to get the following

$$2J_{ф.д} = 2 \cdot 0{,}0128 = 0{,}0256 \ кгм^2.$$

The total mass of the milling drum blade is $m_{ф.п}$ = 0.93 kg. To simplify finding the moment of inertia, we divide the blade into two parts. The first part is the cutting part, ie the blade is 0.22 m, the mass is $m_{п.к}$ = 0.6 kg, the second part is the disk-mounted part, i.e. the blade is 0.12 m, the mass is $m_{п.у}$= 0.33 kg .

Moment of inertia of the cutting part of the blade: mass $m_{п.к}$ = 0.6 kg; distance to the axis of rotation $r_{п.к}$ = 0.17 m;

$$J_{п.к} = \frac{1}{2} m_{п.к} r_{п.к}^2 = \frac{1}{2} \cdot 0{,}6 \cdot 0{,}17^2 = 0{,}00867 \ кгм^2.$$

Moment of inertia of the blade on the disk: mass $m_{п.у}$=0.33 kg; distance to the axis of rotation $r_{п.у}$= 0.11 m;

$$J_{п.у} = \frac{1}{2} m_{п.у} r_{п.у}^2 = \frac{1}{2} \cdot 0{,}33 \cdot 0{,}11^2 = 0{,}00199 \ кгм^2.$$

The total moment of inertia of the blade

$$J_{пич} = J_{п.к} + J_{п.у} = 0{,}00867 + 0{,}00199 = \frac{1}{2} \cdot 0{,}33 \cdot 0{,}11^2 = 0{,}01066 \ кгм^2.$$

Considering that 4 blades are mounted on the milling drum disk, the moment of inertia of the blades in the milling drum in general

$$J_{у.пич} = 8J_{пич} = 8 \cdot 0{,}01066 = 0{,}08528 \ кгм^2.$$

The total moment of inertia of the milling drum is as follows

$$J_{ф} = J_{у.пич} + J_{ф.д} = 0{,}08528 + 0{,}0256 = 0{,}11088 \ кгм^2.$$

The determined values of the moments of inertia of the masses are given in Table 1.

Table 1.
Moments of inertia of rotating moving masses

$J_{ю1}$	$J_{ю2}$	$J_{ш1}$
0,0006	0,00117	0,001068
$J_{ш2}$	$J_{о.в}$	$J_{ф.в}$
0,0009155	0,0016	0,00056
$J_{к1}$	$J_{к2}$	$J_{ф}$
0,0064	0,0073	0,11088

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IV. CONCLUSION.

In multi-mass systems, if it is necessary to reduce the coefficient of inequality of the number of revolutions of a shaft, it is desirable to reduce the stiffness of the elastic element (extension) connected to that shaft in order to increase the moment of inertia of that shaft. Taking into account the above data, taking into account that the coefficients of unevenness in the milling drum shaft of the combined unit do not exceed 0.015-0.03 and in the power take-off shaft does not exceed 0.015, the recommended parameters for the system are as follows: J_{ϕ}== 0.9-0.12 kgm^2, c = 200- 300 Nm / rad. At these values, the drums rotate sufficiently flat and the power consumption is not high. It is also recommended that the gear belt drive traction used in driving the machine be satisfactory.

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