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DYNAMIC ANALYSIS OF THE MACHINE UNIT IN THE DRIVE OF A TRANSMISSION REDUCER WALTZ MACHINE WITH A COMPOSITE, FLEXIBLE ELEMENT GEAR

Abstract: *Objective.The article describes the kinematic scheme and principle of operation of the Waltz machine, which adjusts the height of the shock absorbers. The magnitude of the solution of the problem of the dynamics of the machine unit with a gear element with a gear element with a flexible element in the two-speed reducer in the machine is given. The laws of motion of gears, the working drum, and the electric drive rotor are derived. Connection graphs were constructed and, based on their analysis, optimal values of machine unit movements were recommended.*

Methods. In the process of research, higher mathematics, theory of machines and mechanisms, theory of oscillations, dynamics of machines, test methods of mechanical engineering and technological machines were used.

Results. The main requirement for these rubber pads is that they are evenly distributed over the surfaces of the pads. For this purpose, rubber drums and tapes are passed through special drums on Waltz machines to the required thickness.

Conclusions: The generalized formula for determining the degree of excitability of flexible gear mechanisms was proposed. At the same time, a method of eliminating redundant connections in flat mechanisms was developed.

Key words: *Shock absorber, virginity, composite, flexible element, dissipation, gear wheel, angular velocity, inertia, work efficiency, vibration, regularity, moment of inertia, working drum, reducer.*

Language: *English*

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Introduction

Methods. Scheme and principle of operation of the machine "Waltz", which adjusts the height of the shock absorbers. It is known that a number of technological machines use special shock-absorbing rubber pads to reduce vibrations, transmission torques and vibration amplitudes, ie to dampen [1]. In particular, in the automotive industry, in general, the internal combustion engines of mobile cars are mounted on the body through such pads. They are also mounted on the foundation by means of rubber pads of the required thickness to dampen the vibrations of metal cutting machines and a number of technological machines. The main requirement for these rubber pads is that they are evenly distributed over the surfaces of the pads. For this purpose, rubber drums and tapes are passed through special drums on Waltz machines to the required thickness.

According to the kinematic scheme shown in Figure 1, the Waltz machine works as follows.

Results.

The driving power is 18 KW, n = 1000 rpm, from 1 electric conductor through the coupling 2 to 3 reducers. The gearbox is a two-speed gearbox with a 5-component flexible element via a 4-speed gear. In this case, the 5 wheels are transmitted to the 10 and 11 working drums from 8, 9 internal gear transmission through 6 clutches 7, respectively. The drums are compressed with a rubber band of a specified thickness between 10 and 11, having the desired density, i.e., virginity. The flatter the angular velocities of the drums 10 and 11 in the machine, i.e. the smaller the coefficient of unevenness of the angular velocity, the higher the quality of the output product, i.e., according to [2;3]:

$$\delta = \frac{\omega_{\sigma max} - \omega_{\sigma min}}{\omega_{бyp}} \qquad (1)$$

Where, are the maximum, minimum, and average angular velocities of drums 10 and 11.

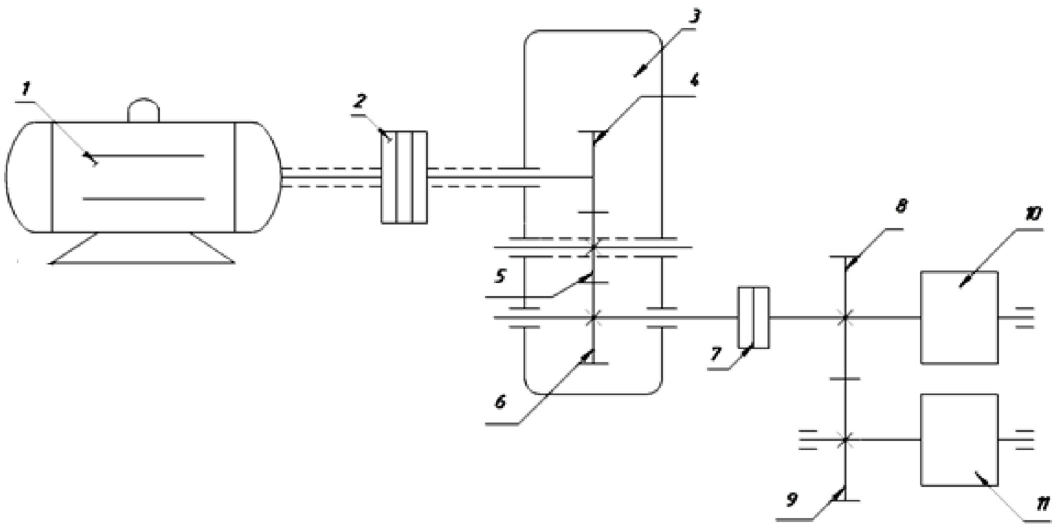


Figure 1. Kinematic scheme of the Waltz machine, which adjusts the height of the shock absorbers.
1 Electric drive, 2, 7 clutches, 3 reducers, 4, 5, 6 gears, 8, 9 external contact gears, 10, 11 working drums.

The main unevenness and noise in machine operation is generated by the reducer transmission. Therefore, it can be ensured by increasing the drum masses to reduce the “ δ”. However, this will dramatically increase power consumption. Belt and chain extensions could also be used. However, due to the large power transmission, the operating time of these extensions is greatly reduced. Therefore, in order to reduce the torque, velocity oscillations, and noise in the gearbox transmission, we put a flexible element in the middle gear of the two-speed gearbox. By selecting the flexible bushing at the desired height, it is possible to reduce the amplitudes of vibration of speed and torque, as well as noise.

Dynamic and mathematical models of machine units with gears in the drive of the Waltz machine. According to the kinematic scheme of the machine “Waltz” for the production of shock-absorbing tires, we see a dynamic model of the machine unit. This allows you to increase the mass of the flexible elements in the system. The first mass includes the electric rotor and half-clutch mass, the second mass includes the half-clutch drive gear, the third mass includes the composite gear, the drive outgoing gear and the second half-clutch mass, the fourth mass includes the half-clutch, external clutch and working drum masses . Hence, the system consists of a four-mass machine unit.

This model is shown in Figure 2.

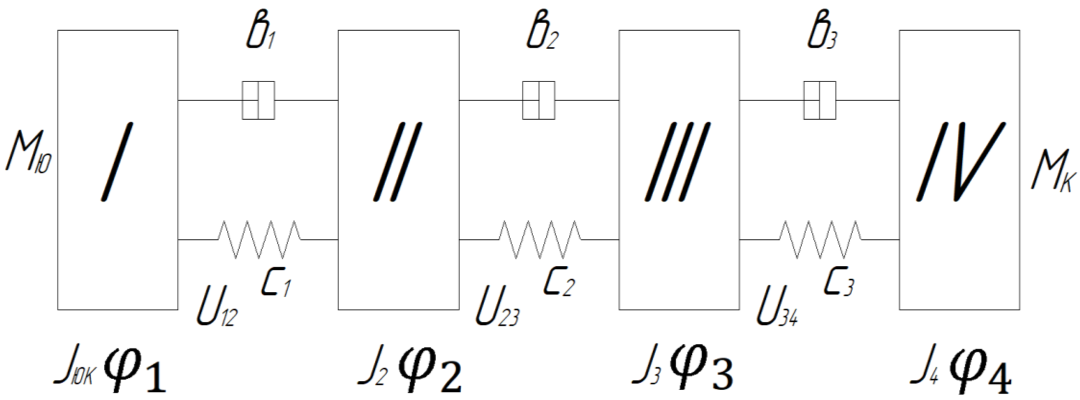


Figure 2. Machine aggregate dynamic model.

According to the research, it is taken into account through the mechanical dynamic characteristics of the electric drive. For a more detailed analysis of the tensile processes in the machine unit under consideration, the mechanical dynamic characteristics of the electric drive were expressed from the system of differential equations proposed by A.E. Levin.

$$\frac{\partial M_{ю}}{\partial t} = \left(\omega_c - P\frac{d\varphi_1}{\partial t}\right)\psi - \frac{M_{ю}}{T_э};$$

$$\frac{\partial\psi}{\partial t} = \frac{2M_к}{T_э} - \frac{\psi}{T_э} - \left(\omega_c - P\frac{d\varphi_1}{\partial t}\right) - M_{ю}; \qquad (2)$$

$$T_э = (\omega_c \cdot S_k)^{-1};\ \psi = \frac{S_k}{S}\left(M_{ю} + T_э\frac{\partial M_{ю}}{\partial t}\right)$$

Where, are the driving moments of the electric drive and its critical value;

Number of R-pairs;

- driver slip and its critical value; - frequency of rotation;

- driver electromagnetic constant time;

- variable.

We determine the equations of motion of the masses of a machine aggregate for each generalized coordinate separately using Lagrangian second-order differential equations [4]:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot q}\right) - \frac{\partial T}{\partial q} + \frac{\partial П}{\partial q} + \frac{\partial Ф}{\partial \dot q} = Q(q) \qquad (3)$$

Where -generalized displacement and velocity, -time,

K, kinetic and potential energies of P-system, F-Relay dissipation function;

φ, $\dot\varphi$ – generalized coordinate and velocity,

Q (q) - is the total power.

The torsion angles of the four masses of the machine aggregate,, and s were taken as the generalized coordinates. In this case, we construct a separate Lagrangian equation for each generalized coordinate. In this case, the kinetic energy of the system:

$$T=\frac{1}{2}\{(J_p+J_M)\dot\varphi_1^2+(J_M+J_{z1})\dot\varphi_2^2+[J_{z2}+U_{z23}^2(J_{z3}+J_M)]\dot\varphi_3^2+ +(J_n+J_{z4}+J_{z5}+J_{\delta1}+J_{\delta2})\dot\varphi_4^2\}$$

System potential energy:

$$П = \frac{1}{2}C_1(\varphi_1 - U_{12}\varphi_2)^2 + +\frac{1}{2}C_2(\varphi_2 - U_{23}\varphi_3)^2 + \frac{1}{2}C_3(\varphi_3 - U_{34}\varphi_4)^2$$

The dissipation function of the relay:

$$Ф = \frac{1}{2}B_1(\varphi_1 - U_{12}\varphi_2)^2 + +\frac{1}{2}B_2(\varphi_2 - U_{23}\varphi_3)^2 + \frac{1}{2}B_3(\varphi_3 - U_{34}\varphi_4)^2$$

Where J_p, J_M- are the moments of inertia of the electric drive rotor and the half - clutch;

$J_{z1}J_{z2}J_{z3}J_{z4}J_{z5,}$- moments of inertia of gears;

$U_{12,}U_{23,}U_{34,}$- transmission ratios;

$J_{\delta1}J_{\delta2}$ - moments of inertia of working drums;

C_1, C_2, C_3- coefficients of rotation of flexible elements;

b_1,b_2, b_3- dissipation coefficients.

The Lagrangian equations define the additions.

Derivatives of kinetic energy:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial\dot\varphi_1}\right) = (J_p + J_м)\,\ddot\varphi_1$$

$$\frac{d}{dt}\left(\frac{\partial T}{\partial\dot\varphi_2}\right) = (J_м + J_{z1})\,\ddot\varphi_2$$

$$\frac{d}{dt}\left(\frac{\partial T}{\partial\dot\varphi_3}\right) = [J_{z2} + U_{z23}^2(J_{z3} + J_м)]\ddot\varphi_3$$

$$\frac{d}{dt}\left(\frac{\partial T}{\partial\dot\varphi_4}\right) = (J_{z3} + J_{z4} + J_{z5} + J_{\delta1} + J_{\delta2})\ddot\varphi_4$$

Derivatives of potential energy:

$$\frac{\partial П}{\partial\varphi_1} = C_1(\varphi_1 - U_{12}\varphi_2);$$

$$\frac{\partial П}{\partial\varphi_2} = -U_{12}C_1(\varphi_1 - U_{12}\varphi_2) + C_2(\varphi_2 - U_{23}\varphi_3);$$

$$\frac{\partial П}{\partial\varphi_3} = -U_{23}C_2(\varphi_2 - U_{23}\varphi_3) + C_3(\varphi_3 - U_{34}\varphi_4);$$

$$\frac{\partial П}{\partial\varphi_4} = -U_{34}C_3(\varphi_3 - U_{34}\varphi_4).$$

Derivatives from the dissipation function:

$$\frac{\partial \Phi}{\partial \dot{\varphi}_1} = b_1(\dot{\varphi}_1 - U_{12}\dot{\varphi}_2);$$

$$\frac{\partial \Phi}{\partial \dot{\varphi}_2} = -U_{12}b_1(\dot{\varphi}_1 - U_{12}\dot{\varphi}_2) + b_2(\dot{\varphi}_2 - U_{23}\dot{\varphi}_3);$$

$$\frac{\partial \Phi}{\partial \dot{\varphi}_3} = -U_{23}b_2(\dot{\varphi}_2 - U_{23}\dot{\varphi}_3) + b_3(\dot{\varphi}_3 - U_{34}\dot{\varphi}_4);$$

$$\frac{\partial \Phi}{\partial \dot{\varphi}_4} = -U_{34}b_3(\dot{\varphi}_3 - U_{34}\dot{\varphi}_4)$$

Moments of generalized forces in Lagrange's equations:

$$M(\varphi_1) = M_{ю};$$
$$M(\varphi_2) = M_{и2};$$
$$M(\varphi_3) = M_{и3};$$
$$M(\varphi_4) = M_{и4} + M_{к};$$
$$M_{к} = M_{ко} = \delta M(M_{ю})$$

The masses of the machine unit and their constituent loads were determined experimentally. In this case, loads of weights b1 and b2 were hung on the pulleys (wrapping the rope around the shafts). The time it took for the loads to fall to the ground was measured three times. Accelerations a1 and a2 were calculated based on them. They were then calculated by putting them in the following expression:

$$J = [(1-\frac{a_1}{g})G_1 - (1-\frac{a_2}{g})G_2]\frac{D}{2(a_1-a_2)} \qquad (4)$$

where pulley diameter;
free fall acceleration.

The moments of inertia of the masses are as follows:

$J_p = 0{,}212$ кгм^2; $J_м = 0{,}303$ кгм^2;
$J_{z1} = 0{,}261$ кгм^2; $J_{z2} = 0{,}41$ кгм^2;
$J_{z3} = 0{,}643$ кгм^2; $J_{z4} = 1{,}03$ кгм^2;
$J_{z5} = 1{,}03$ кгм^2; $J_{\delta1} = J_{\delta2} = 3{,}469$ кгм^2;

The rotational coefficients of elastic elements were determined using the following formula [5,6,7]:

$$C = \frac{R^2 EFa}{l_p} \qquad (5)$$

Where the average radius of the elastic element;
E – elastic element elastic modulus;
F – cutting surface;
l_p – elastic element length;
a – coefficient taking into account the deformation of the elastic element.

Using the expression (5), the coefficients of rotation of the flexible elements from the machine unit were calculated:

$C_1 = (450 \div 500)$ км/рад;
$C_2 = (400 \div 420)$км/рад;
$C_3 = (450 \div 500)$км/рад.

The dissipation coefficients of the elastic elements were calculated using the existing expression [8,9,10]:

$$b = \frac{\psi_y C}{2\pi(\frac{2\pi}{T_g})} \qquad (6)$$

Where ψ_y –is the coefficient representing the transmission of rotational motion; T_g –oscillation period.

The following values were obtained for the elastic elements under consideration:

$b_1 = (6{,}8 \div 7{,}2)$ кмс/рад;
$b_2 = (5{,}5 \div 6{,}0)$ кмс/рад;
$b_3 = (6{,}5 \div 7{,}0)$ кмс/рад;

Taking into account the additions of the obtained Lagrange equations, we create a system of differential equations representing the motion of the unit of the machine “Waltz” with a gear with a flexible element:

$$\frac{\partial M_{ю}}{\partial t} = \left(\omega_c - P\frac{d\varphi_1}{\partial t}\right)\psi - \frac{M_{ю}}{T_э};$$

$$\frac{\partial \psi}{\partial t} = \frac{2M_к}{T_э} - \frac{\psi}{T_э} - \left(\omega_c - P\frac{d\dot{\varphi}_1}{\partial t}\right) - M_{ю};$$

$$T_э = (\omega_c \cdot S_k)^{-1}; \ \psi = \frac{S_k}{S}(M_{ю} + T_э\frac{\partial M_{ю}}{\partial t});$$

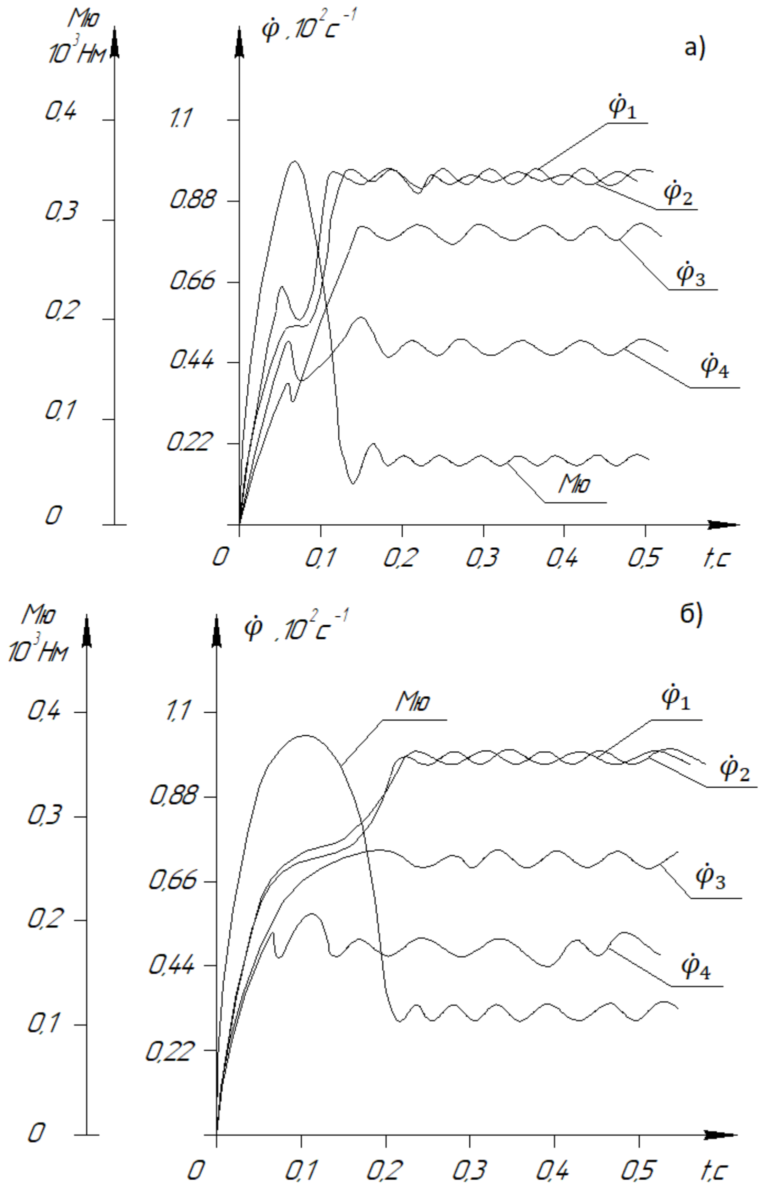
$$\begin{aligned}
&(J_p + J_m)\varphi_1 = M_{10} - b_1(\varphi_1 - u_{12}\varphi_2) - c_1(\varphi_1 - u_{12}\varphi_2);\\
&(J_m + J_{z1})\varphi_2 = u_{12}b_1(\varphi_1 - u_{12}\varphi_2) + u_{12}c_1(\varphi_1 - u_{12}\varphi_2) -\\
&- b_2(\varphi_2 - u_{23}\varphi_3) - c_2(\varphi_2 - u_{23}\varphi_3) - M_{k2};\\
&[J_{z2} + u_{z23^2}(J_{z3} + J_m)]\varphi_3 = u_{23}b_2(\varphi_2 - u_{23}\varphi_3) +\\
&u_{23}c_2(\varphi_2 - u_{23}\varphi_3) - b_3(\varphi_3 - u_{34}\varphi_4) - c_3(\varphi_3 - u_{34}\varphi_4) - M_{k3};\\
&(J_m + J_{z4} + J_{z5} + J_{z1} + J_{z2})\varphi_4 = u_{34}b_2(\varphi_3 - u_{34}\varphi_4) +\\
&+ u_{34}c_3(\varphi_3 - u_{34}\varphi_4) - M_{r4}[M_{k0} \pm \delta M(M_{k0})];
\end{aligned} \qquad (7)$$

The obtained system of differential equations (2.19) was carried out on the basis of the initial conditions at $t = 0$ and $\dot{\varphi}_1 = \dot{\varphi}_2 = \dot{\varphi}_3 = \dot{\varphi}_4 = 0$ ва $M_{u2} = M_{u3} = M_{u4} = 0$ as well as at the following calculated values of the parameters:

$N_{ю} = 18$ кВт;
$n_{ю} =$ айл/мин;
$\dot{\varphi}_p = 104{,}6$ с$^{-1}$;
$\dot{\varphi}_2 = 86{,}8$ с$^{-1}$;
$\dot{\varphi}_3 = 57{,}8$ с$^{-1}$;
$\dot{\varphi}_{\delta1} = \dot{\varphi}_{\delta2} = 57{,}8$ с$^{-1}$;

$$U_{12} = 1{,}0; U_{23} = 1{,}2; U_{34} = 1{,}5; Z_1 = 30;$$
$$Z_2 = 36; Z_3 = 48; Z_4 = 56; Z_5 = 56;\ fc = 50 \mathit{Гц};$$
$$cos\varphi = 0{,}86; h = 0{,}85; S = 0{,}061; S_k = 0{,}187; M_K = 109{,}7 \mathit{Нм};\ p = 2{,}0;$$
$$F = 0{,}152 \cdot 10^{-4} m^2; a = 2{,}2; l_p = 0{,}18m; F = 44 \cdot 10^6\ н/м^2; M_{ko} = 156 \mathit{Нм};$$
$$\delta M(M_{k0}) = (0{,}1 \div 0{,}12) M_{K0;}; M_{K2} = 10{,}5 \mathit{Нм}; M_{K3} = 12{,}1 \mathit{Нм};$$
$$M_{K4} = 19{,}7 \mathit{Нм}; D\delta_1 = D\delta_2 = 0{,}2 м.$$

Numerical solution of the problem and the results of the analysis. In the “Waltz” machine, the rubber shock absorbers are squeezed through the working drums to adjust the density and stiffness over the entire surface. This will ensure that the drums rotate evenly. Also, due to friction and load changes in the transmission of motion in a two-speed gearbox, the gear wheel is quickly worn out and out of order, the noise increases, the working life of the gearbox is reduced.



а –Mю=120 Нм; δ – $M_к$ = 170 Нм

Figure 3. Angular velocities on the rotor, gears, and drums of the Waltz machine and the laws of change of drive load.

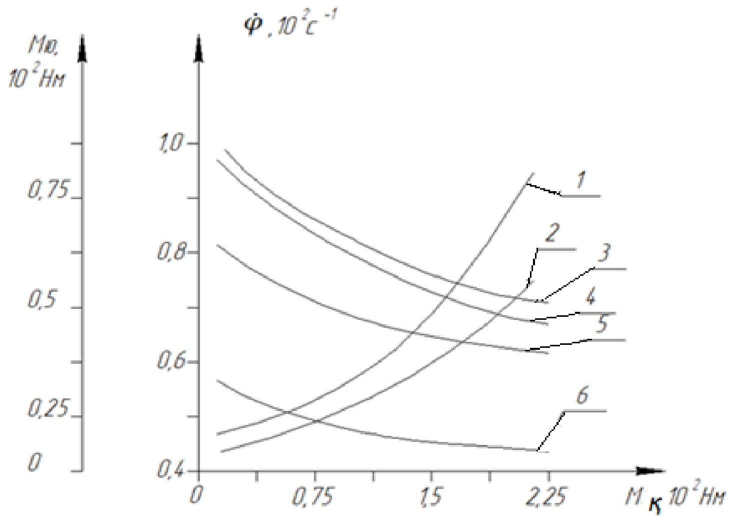
After the installation of the flexible bushing, which includes the middle gear of the gearbox, the impact on the joints is reduced, the amplitudes of vibrations of the loads are reduced, the movement is stabilized. As a result, the workforce increases. Therefore, it is important to determine the parameters of the proposed flexible element, the required values of the extension operating modes, to substantiate the parameters of virginity-dissipation.

The numerical solution of the resulting (9) system was performed on a PC. In the calculated values of the initial conditions and parameters, the law of change of angular velocities of the drive, gears and working drums, as well as the load on the drive was obtained. The resulting laws are shown in the figure. While the loaded resistance in Figure 3 a is 120 Nm, the resistance in Figure 3 b is taken as 170 Nm. Analysis of the results obtained shows that the average value of the angular velocity on the electric drive shaft at a load of 120 nm was between 98.1 s-1 reducer first gear shaft almost the same 97.2 s-1

It can be seen that the angular velocity of the second gear wheel shaft of the composite flexible element is 82.3 s-1, while the angular velocity of the output gear shaft, the movement of the working drums is 51.7 s-1. In this case, the torque on the rotor shaft is in the range of 87.3 nm. As the corresponding distributed resistance increases, that is, as the thickness of the shock absorber raw material passing between the working drums increases, or when the density, high-density raw material is used, the load on the drive also increases. That is, when Mq = 170 nm, it was found that Myu = 183.2 nm. In this case, the resistances due to additional friction were calculated taking into account the calculated values.

As a result of processing the obtained laws, graphs linking the parameters were constructed. Figure 4 shows the angular velocities of the electric drive rotor, drive gear, composite gear, drive gear and working drum shafts, the change in load on the drive, the dependence of the change in technological resistance

Based on the analysis of the graphs, it can be noted that as the distributed resistance increases from 0.37102 km to 2.25 102 km, the angular velocity of the rotor shaft decreases in a nonlinear pattern from 98.7 s-1 to 78.1 s-1, the angular velocity in the flexible element gear shaft from 82.4 s-1 to 68.3 s A decrease to -1 can be observed.

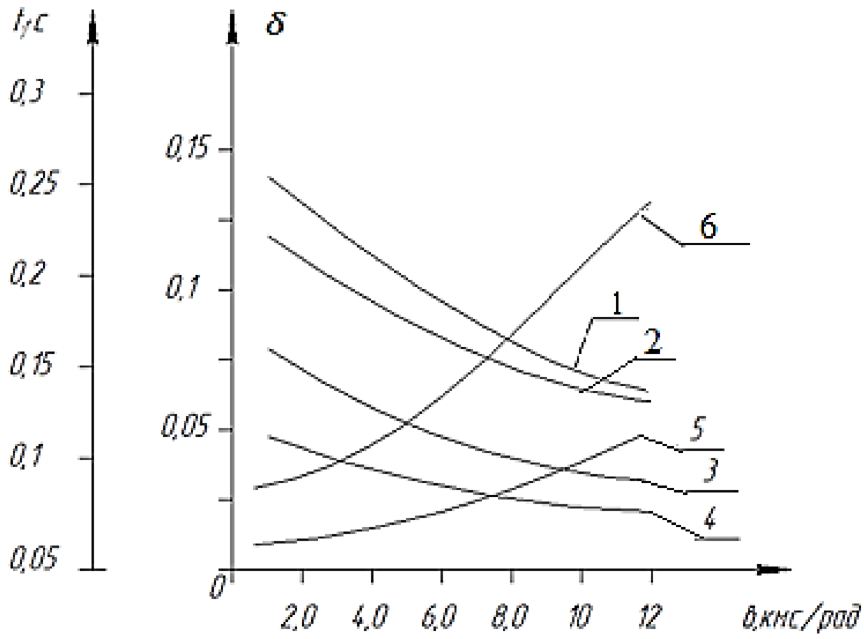


1-$M_ю = f(M_к) - J_ю = 0{,}25$ кгм^2; 2-$M_ю = f(M_к) - J_ю = 0{,}212$ кгм^2;
3-$\dot{\varphi}_1 = f(M_к)$; 4-$\dot{\varphi}_2 = f(M_к)$; 5-$\dot{\varphi}_3 = f(M_к)$; 6-$\dot{\varphi}_4 = f(M_к)$;

Figure 4. Graphs of the dependence of the angular velocities on the shafts of the electric drive rotor, drive gear, compound gear, drive gear and working drums, the change in load on the drive, the change in technological resistance.

The angular velocities of the working drums decreased in a nonlinear pattern from 55.2 s-1 to 48.1 s-1. Correspondingly, it was found that the load on the drive increases in a nonlinear pattern from 0.09102 km to 0.72102 km when the load = 0.212 kgm2. When the moment of inertia of the first mass of the machine unit was observed to 0.25 kgm2, the load on the drive was observed to increase to 1.09102 km. It is therefore recommended that the technological resistance Mq (150) nm not exceed 102 km to ensure that the load on the drive (0.9) does not exceed 102 km.

It is known from the theory of machines and mechanisms that [11,12] is achieved by increasing their moments of inertia to smooth the motion of rotating working bodies. But excessive increase of the moment of inertia increases the load, power consumption, the working resource of the machine also decreases. Figure 5 shows graphs of the coefficients of unevenness of the angular velocities of the gears, working drums in the drive of the technological machine "Waltz" and the moments of inertia of the drive load.



1-$\delta_1 = f(J_p + J_m)$; 2-$\delta_2 = f(J_u + J_{z1})$; 3-$\delta_3 = f[J_{z2} + U_{z2}^2(J_{z3} + J_m)]$;
4-$\delta_4 = f(J_m + J_{z4} + J_{z5} + J_{\delta 1} + J_{\delta 2})$; 5-$M_{ю} = f(J_{ки})$ $-M_к$=120 нм
6-$M_{ю} = f(J_{ки})$ $-M_к$=170 нм.

Figure 5. Graphs of the dependence of the coefficients of unevenness of angular velocities of gears, working drums in the drive of the technological machine "Waltz" and the moments of inertia of the drive load.

For each mass in the graphs, the effect of the moment of inertia of that mass is considered. In this case, the moments of inertia of the masses are equal in the calculated values.

In particular, when the moments of inertia of working drums increase from 0.5 kgm2 to 3.0 kgm2, the coefficient of unevenness of the angular velocities of their shafts decreases from 0.165 to 0.064. The value of the coefficient of roughness of the angular velocity of the gear element with a flexible element decreases in a nonlinear law from 0.123 to 0.05.

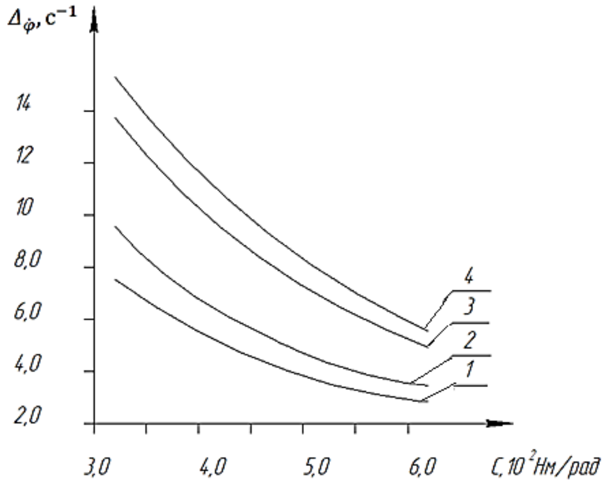
A decrease in the value of the electric drive rotor shaft was found to be from 0.042 to 0.018. It should be noted that according to the results of experimental research, the angular velocities of the working drums do not exceed the coefficients of unevenness (0.05), it is possible to carry out the technological process on the basis of requirements. Therefore, based on the analysis of the graphs, the recommended values of the moments of inertia of the masses are $(J_p + J_m) \geq$ (0,6÷ 0,8) кгм^2; $(J_m + J_{z1}) \geq$ (0,9÷ 1,0) кгм^2;

$[J_{z2} + U_{z23}^2(J_{z3} + J_m)] \geq$ (1,2÷ 1,4) кгм^2;

$(J_m + J_{z4} + J_{z5} + J_{\delta 1} + J_{\delta 2}) \geq$ (3,7÷ 5,2) кгм^2.

Based on the recommended parameters, it should be noted that while it is recommended that the machine unit obtain more moments of inertia than the calculated values of the first three masses, it is advisable to reduce the moments of inertia of the working drums relative to the calculated values.

Figure 6 shows graphs of the dependence of the angular velocities of the machine shaft drive shaft, gears and working drum shafts on the rotational coefficients of rotation of the corresponding flexible elements of the vibration coverage. The values of the oscillation coverage of the angular velocities decrease in the nonlinear law as the elastic elements of the machine unit increase in rotation. In particular, when the coefficient of rotation of the coupling elastic element increases from 3.2102 nm / rad to 6.9102 nm / rad, it decreases in a nonlinear pattern from 1.6 s-1 to 3.2 s-1, respectively, the values from 9.8 s-1. A decrease of 4.3 s-1 was detected. Accordingly, as the coefficient of rotation of the flexible ring of a compound gear increases, its values also decrease in the nonlinear law. At the same time, the value decreases from 13.6 s-1 to 4.9 s-1. Hence, it is expedient to drastically increase the coefficient of rotation of the composite gear wheel elastic element.

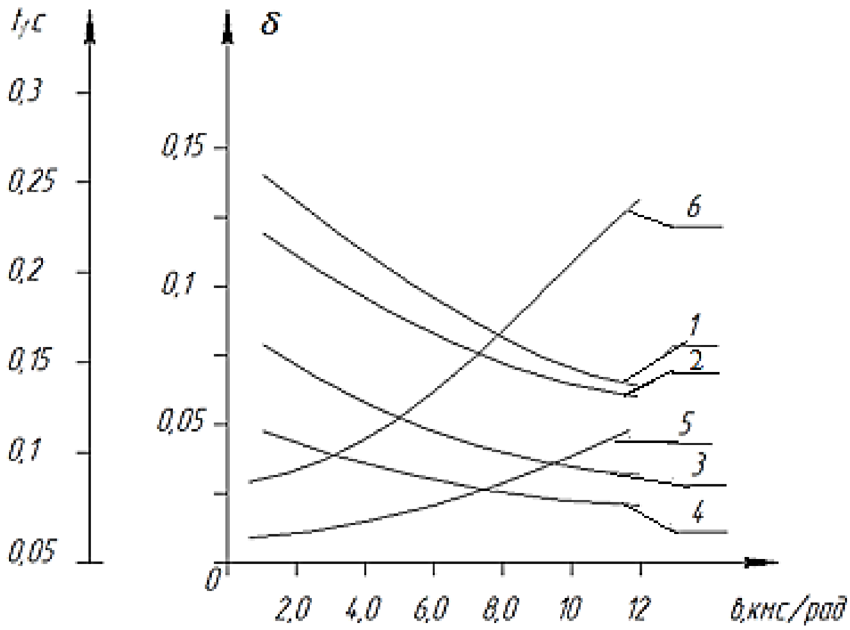


4-$\Delta\dot\varphi_4 = f(c_3)$; 3-$\Delta\dot\varphi_3 = f(c_2)$; 2-$\Delta\dot\varphi_2 = f(c_1)$; 1-$\Delta\dot\varphi_1 = f(c_1)$;

Figure 6. Graphs of the dependence of the angular velocities of the drive shaft, gears and working drum shafts of the machine unit on the rotational coefficients of rotation of the flexible elements corresponding to the vibration coverage.

It is recommended to select the flexible element ring in the range of rotational coefficients of rotation (6.5) 102 Nm / rad to ensure that the values do not exceed (4.0) s-1, taking into account the pitch of the composite gear teeth, i.e. to minimize impact when interlocking teeth. Accordingly, in order to reduce the oscillation of the angular velocities of the working drums, it is also recommended to obtain the coupling coefficient of the recommended values in the range of recommended values (7.5) 102 Nm / rad. The coefficient of virginity of the elastic element of the first coupling (5.0) was recommended in the range of 102 Nm / rad.

It should be noted that the dissipation coefficient of the elastic element also reduces the angular velocity oscillations. But this increases the loading value. Figure 7 shows the graph of the dependence of the angular velocities on the shafts of the electric drive rotor, gears and working drums of the machine unit on the coefficients of unevenness of the angular velocities on the shafts and the dissipation coefficients of the flexible elements.



1-$\delta_4 = f(b_3)$; 2-$\delta_3 = f(b_2)$; $3 - \delta_2 = f(b_1)$; $4 - \delta_1 = f(b_1)$;
$5 - t_{ў} = f(b_к) - М_к = 120$ нм; $6 - t_{ў} = f(b_к) - М_к = 170$ нм;

Figure 7. Graphs of the angular velocities on the shafts of the electric drive rotor, gears and working drums of the machine unit, the angular velocities on the shafts, the coefficients of unevenness and the dissipation coefficients of the flexible elements.

When the technological coupling increases the dissipation coefficient of the second coupling elastic element from 2.0 kms / rad to 12.0 kms / rad at 120 Nm, the steady-state motion of the system increases from 0.02 s to 0.12 s when the load is Mq = 170 nm the time increases from 0.1 s to 0.27 s. This can cause the electric motor to overheat and malfunction. Due to the change in the dissipation coefficient of the elastic element corresponding to each mass, it will be possible to adjust the values of, and (Fig. 7, Figures 1,2,3,4). In particular, when the value of b2 increases from 1.8 kms / rad to 12.0 kms / rad, the values of decrease in a nonlinear pattern from 0.075 to 0.031. It is therefore recommended that the dissociation coefficient of the composite gear wheel elastic element be in the range of (10.0) nms / rad to ensure that ni is in the range of (4.0) s-1, respectively.

Respectively, the dissociation coefficients of the flexible elements of the couplings are recommended to be in the range of b_1 = (5.5) nms/rad and b_3 = (9.5) nms/rad.

Conclusions

1. Dynamic and mathematical models of the technological machine "Waltz" with a flexible element gear in the drive were built taking into account the dynamic mechanical characteristics of the electric drive, technological parameters, flexible elements, virginity-dissipative properties.
2. Based on the numerical solution of the problem of dynamics of a four-mass machine unit, the recommended values of the parameters of the angular velocities of the drive, gears and drums, as well as the parameters of the change in drive load were determined.

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