Machine Unit Dynamics Cotton Cleaner Section With New Drive Circuit

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ABSTRACT

The article presents a modernized drive scheme for a cotton-ginning unit of installation cotton-ginning-complex (ICGC). The computational and mathematical models of machine units of the cleaner with mechanisms of sawing, peeling, and transporting drums have been calculated and completed. Based on the numerical solution of the problems of the dynamics of machine units, the regularities of the movement, graphic dependences and parameters of the working bodies obtained considering on the given calculations. Analyzes of the constructed graphical dependencies with justified and recommended specific values of the parameters of the working bodies and belt drives in the drive of the cotton cleaner with the recommended drive layout.

Keywords: Raw cotton, fine and coarse litter, saw peg, conveyor, drum, belt drive, law of motion, angular velocity, dissipation, cleaning effect.

1. Introduction

Cotton ginners have the purpose of cleaning cotton, both coarse and fine litter [1,2]. Most cotton ginners are driven, in which the rotational movement is transmitting from the electric motor through the belt drive, and then the movement is transmitting through the belt drives to the saw and peg drums. At the same time, both on the rotor shaft and on the shafts of the saw cylinder, peg drum and brush, the torques have a complex character due to the variability of the load from the corresponding loads from the cleaned cotton [3,4]. At the same time, drive mechanisms were use in cotton ginning units, including belt drives and electric motors. The first section of the unit uses four electric motors. The process of cleaning raw cotton from fine and coarse litter carried out simultaneously in the ICGC unit. At the same time, various loads are experiencing on the working bodies (pegs, brush drums, feed rollers, and saw cylinders). In addition, the moments of inertia of each working element are also different, for example, the most massive is the saw cylinder. Moreover, the number of belts and their elastic-dissipative characteristics are also different for the working bodies of the ICGC unit. In this case, the change in load and angular velocities on the shafts of the working bodies depends on the above conditions. The loading and angular velocities of the working bodies directly affect the nature of the movement of the drive motor. All this leads to the fact that due to the absence of a kinematic connection between the working bodies, they will rotate with angular velocities significantly different from the calculated ones. This leads to a violation of the cyclist and flow of cleaning of raw cotton in the ginning unit of the ICGC, the efficiency of cleaning significantly decreases, and damage to seeds and cotton fibers will increase. Therefore, a new and efficient unit drive arrangement is recommending.

2. Research Methodology

Effective drive circuit of the cotton-ginning unit. To ensure the continuity of the working mechanism of the cleaning unit, instead of four electric motors one electric motor with a power of 11.0 kW, n=1000 min^{-1} and an electric motor for the feed rollers (P=1.5 kW, n=(0÷20) min^{-1}) are used. Also a motor for a removable brush drum (P=2.2 kW, n=930 min^{-1}). The recommended diagram of the drive mechanisms of the unit shown in Fig.1. A feature of the proposed drive scheme of the unit is the kinematic connection between the main working bodies, saw cylinders 4, 5 pegs 7, 8, 11, 12 and brush drums 6, 10 with a debris auger 13. At the same time, the work of the working bodies were interconnecting, as a single flow chain. This allows decrease in slaughter, increase in cleaning effect, and decrease in cotton damage.

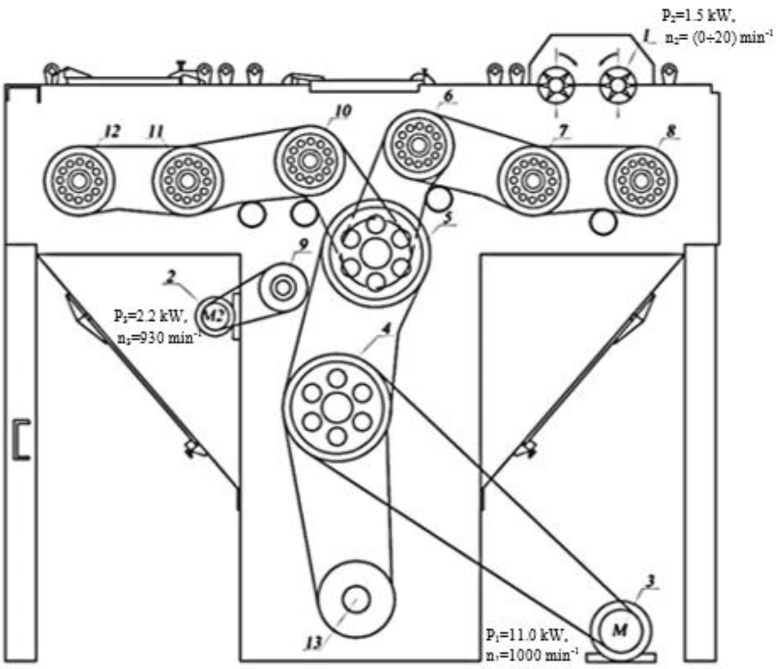
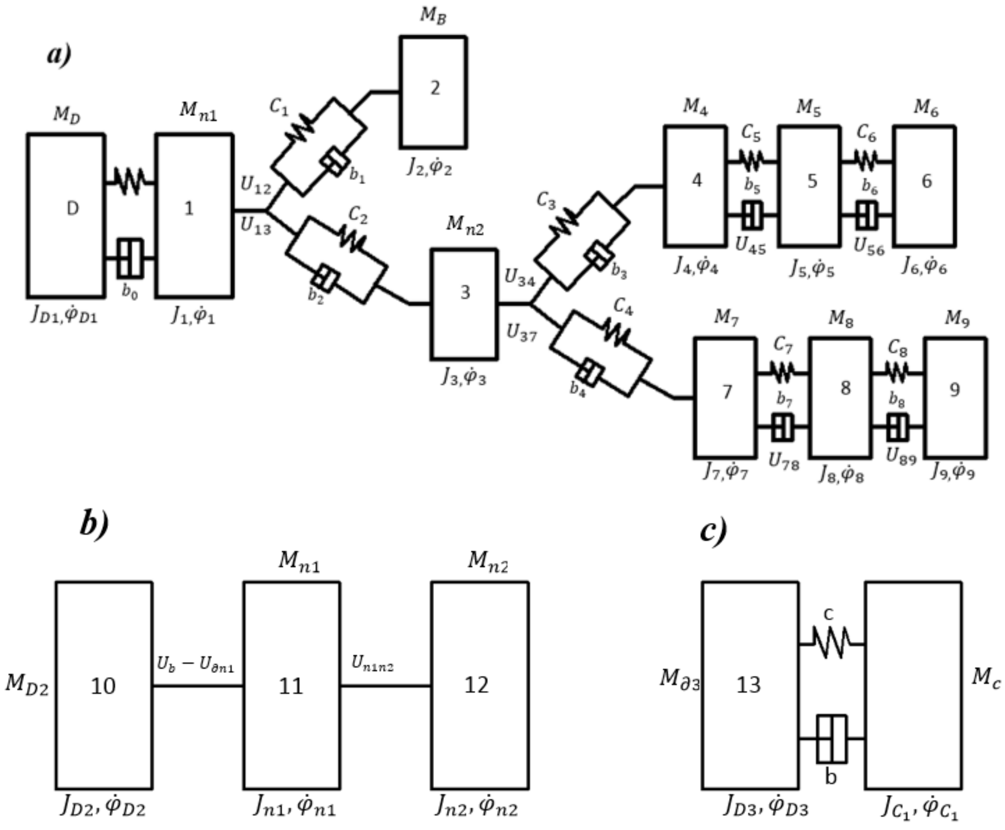


Fig.1. Kinematic diagram of a cotton ginner with a recommended arrangement of drive mechanisms

Calculation scheme and mathematical model of the dynamics of the machine unit of the cotton cleaner ICGC. For the main working bodies of the raw cotton cleaner from coarse and fine litter, sawing, splitting, brush - transporting drums, the design scheme consists of a ten-mass system. The working bodies receive movement from the electric motor P_1=11.0 kW, n_1=1000 min^{-1} (see Fig. 2,*a*). In the second diagram, a three-mass system includes feed rollers and a motor, P_2=1.5 kW, n_2 = (0÷20) min^{-1}. Accordingly, the third scheme includes two masses: the rotor of the motor and the brush removable drum (see Fig. 2,*b*). In this case, P_3=2.2 kW, n_3=930 min^{-1}. A feature of the recommended cotton ginner drive is the use of belt drives with variable gear ratios that have split pulleys and eccentric idler pulleys with rubber bushings.



a) for pegs, feed-transporting and serrated drums; b) for feed rollers; c) for the brush drum.

Fig.2. Dynamic models of the cleaning unit drive mechanisms

In this case, from the kinematic analysis of the belt drive with eccentric tension rollers [5,6,7] it is known that

$$\omega_2 = \frac{\omega_1 r_1}{r_2}\cdot\frac{e\cdot\cos\varphi_3' + \sqrt{r_3^2 - e^2 sin^2\varphi_3'}}{e\cdot\cos\varphi_3 + \sqrt{r_3^2 - e^2 sin^2\varphi_3}}; \quad \varphi_3 = arctg\frac{r_3 sin\frac{r_1\varphi_1}{r_3}}{e + r_3 cos\frac{r_1\varphi_1}{r_3}};$$

$$\varphi_3' = arctg\frac{r_3 sin\frac{r_2\varphi_2}{r_3}}{e + r_3 cos\frac{r_2\varphi_2}{r_3}}; \quad U_{12} = \frac{r_2}{r_1}\cdot\frac{e\cdot\cos\varphi_3' + \sqrt{r_3^2 - e^2 sin^2\varphi_3}}{e\cdot\cos\varphi_3 + \sqrt{r_3^2 - e^2 sin^2\varphi_3'}} \tag{1}$$

where, r_1, r_2, r_3- respectively, the radii of the driving and driven pulleys, the tension roller; φ_1, φ_2- respectively, the angular displacement of the driving and driven pulleys; φ_3, φ_3'- the angles of the belt of the tension roller in the branches of the driving and driven pulleys; e-eccentricity of the tension roller; U_{12}- transmission ratio.

It should be noted that the asynchronous electric motor is taken into account in the form of mechanical dynamic characteristics, continued by A.E. Levin, and elastic-dissipative characteristics of belt drives through the coefficients of circular stiffness and dissipation, technological resistance through the moments of resistance on each shaft of the working bodies.

The resulting system of differential equations describing the dynamics of the movement of the rotor of the engine, saw, peg, transporting drums and debris auger has the form (Fig. 2,a):

$$\frac{dM_{D1}}{dt} = \left(\omega_{c_1} - P_1\frac{d\varphi_{D1}}{dt}\right)\psi_1 - \frac{M_{D1}}{T_{e1}}; \quad \frac{d\psi_1}{dt} = \frac{2M_{k1} - \psi_1}{T_{e1}} - \left(\omega_{c_1} - P_1\frac{d\varphi_{D1}}{dt}\right) - M_{D1};$$

$$T_{e1} = (\omega_{c_1}\cdot S_{k1})^{-1}; \quad \psi_1 = \frac{S_{k1}}{S_1}\left(M_{D1} + T_{e1}\frac{dM_{D1}}{dt}\right);$$

$$I_{D1}\frac{d^2\varphi_{D1}}{dt^2} = M_{D1} - C_0(\varphi_{D1} - U_{D1}\psi_1) - \text{в}_0\left(\frac{d\psi_1}{dt} - U_{D1}\frac{d\psi_1}{dt}\right); \tag{2}$$

$$I_1\frac{d^2\psi_1}{dt^2}=U_{D1}C_0(\varphi_{D1}-U_{D1}\psi_1)+U_{D1}в_0\left(\frac{d\varphi_{D1}}{dt}-U_{D1}\frac{d\psi_1}{dt}\right)-C_1(\psi-U_{12}\psi_2)-в_1\left(\frac{d\psi_1}{dt}-U_{12}\frac{d\psi_2}{dt}\right)-$$
$$C_2(\psi_1-U_{13}\psi_3)-в_2\left(\frac{d\psi_1}{dt}-U_{13}\frac{d\psi_3}{dt}\right)-M_{n1};$$
$$I_2\frac{d^2\psi_2}{dt^2}=U_{12}C_1(\psi_1-U_{12}\psi_2)-U_{12}в_1\left(\frac{d\psi_1}{dt}-U_{12}\frac{d\psi_2}{dt}\right)-M_B;$$
$$I_3\frac{d^2\varphi_1}{dt^2}=U_{13}C_2(\varphi_1-U_{13}\varphi_3)+U_{13}$$
$$в_2\left(\frac{d\varphi_1}{dt}-U_{13}\frac{d\varphi_3}{dt}\right)-C_3\left(\varphi_3-\widetilde{U}_{34}\varphi_4\right)\left(1-\frac{\partial\widetilde{U}_{34}}{\partial\varphi_3}\right)-в_3\left(\frac{d\varphi_3}{dt}-U_{34}\frac{d\varphi_4}{dt}\right)\left(1-\frac{d\varphi_3}{dt}\frac{\partial\widetilde{U}_{34}}{\partial\dot{\varphi}_3}\right)-$$
$$C_4\left(\varphi_3-\widetilde{U}_{35}\varphi_7\right)\left(1-\frac{\partial U_{37}}{\partial\varphi_3}\right)-в_4\left(\frac{d\varphi_3}{dt}-\widetilde{U}_{35}\frac{\partial\varphi_7}{\partial t}\right)\left(1-\frac{d\varphi_3}{dt}\frac{d\widetilde{U}_{35}}{d\dot{\varphi}_3}\right)-M_{n2};$$
$$I_4\frac{d^2\varphi_4}{dt^2}=\widetilde{U}_{34}C_3\left(\varphi_3-\widetilde{U}_{34}\varphi_4\right)+\widetilde{U}_{35}в_3\left(\frac{d\varphi_3}{dt}-\widetilde{U}_{34}\frac{d\varphi_4}{dt}\right)-C_5(\varphi_4-U_{45}\varphi_5)-в_5\left(\frac{d\varphi_4}{dt}-U_{45}\frac{d\varphi_5}{dt}\right)$$
$$-M_4;$$
$$I_5\frac{d^2\varphi_5}{dt^2}=U_{45}C_5(\varphi_4-U_{45}\varphi_5)+U_{45}в_5\left(\frac{d\varphi_4}{dt}-U_{45}\frac{d\varphi_5}{dt}\right)-C_6\left(\varphi_5-\widetilde{U}_{56}\varphi_6\right)\left(1-\frac{dU_{56}}{d\varphi_5}\right)$$
$$-в_6\left(\frac{d\varphi_5}{dt}-\widetilde{U}_{56}\frac{d\varphi_6}{dt}\right)\left(1-\frac{d\varphi_5}{dt}\frac{d\widetilde{U}_{56}}{d\dot{\varphi}_5}\right)-M_5;$$
$$I_6\frac{d^2\varphi_6}{dt^2}=U_{56}C_6\left(\varphi_5-\widetilde{U}_{56}\varphi_6\right)+U_{56}в_6\left(\frac{d\varphi_5}{dt}-\widetilde{U}_{56}\frac{d\varphi_6}{dt}\right)-M_6;$$
$$I_7\frac{d^2\varphi_7}{dt^2}=\widetilde{U}_{35}C_9\left(\varphi_3-\widetilde{U}_{35}\varphi_7\right)+\widetilde{U}_{35}в_4\left(\frac{d\varphi_3}{dt}-\widetilde{U}_{35}\frac{d\varphi_7}{dt}\right)-C_7\left(\varphi_7-\widetilde{U}_{78}\varphi_8\right)\left(1-\frac{d\widetilde{U}_{78}}{d\varphi_7}\right)$$
$$-в_7\left(\frac{d\varphi_7}{dt}-\widetilde{U}_{78}\frac{d\varphi_8}{dt}\right)\left(1-\frac{d\varphi_7}{dt}\frac{d\widetilde{U}_{78}}{d\dot{\varphi}_7}\right)-M_7;$$
$$I_8\frac{d^2\varphi_8}{dt^2}=\widetilde{U}_{78}C_7\left(\varphi_7-\widetilde{U}_{78}\varphi_8\right)+\widetilde{U}_{78}в_7\left(\frac{d\varphi_7}{dt}-\widetilde{U}_{78}\frac{d\varphi_8}{dt}\right)-C_8(\varphi_8-U_{89}\varphi_9)$$
$$-в_8\left(\frac{d\varphi_8}{dt}-U_{89}\frac{d\varphi_8}{dt}\right)-M_8;$$
$$I_9\frac{d^2\varphi_9}{dt^2}=U_{89}C_8(\varphi_8-U_{89}\varphi_9)+U_{89}в_8\left(\frac{d\varphi_8}{dt}-U_{89}\frac{d\varphi_9}{dt}\right)-M_9;$$

where, $\varphi_{D1},\varphi_1,\varphi_1\dots,\varphi_9$ are the angular displacements of the motor rotor, saw, peg and transport drums, waste auger; M_{D1}, $M_{к1},M_1,M_2\dots,M_9$ - moments on the rotor shafts, rotating masses of the system; $I_{D1},I_1,I_2,\dots,I_9$ - moments of inertia of the engine rotor and the corresponding working parts of the cleaner; $C_0,C_1,C_2,\dots,C_9$, $в_0,в_1,в_2,\dots,в_8$- coefficients of circular stiffness and dissipation of belt drives; $U_{D1},U_{12},\dots,U_9$- gear ratios of belt drives between the masses of the system. The system of differential equations for the section of the feed rollers is (Fig. 2,b):

$$\frac{dM_{D2}}{dt}=\left(\omega_{c_2}-P_2\frac{d\varphi_{D2}}{dt}\right)\psi_2-\frac{M_{D2}}{\mathrm{T}_{e2}};\ \frac{d\psi_2}{dt}=\frac{2M_{k2}-\psi_2}{\mathrm{T}_{e2}}-\left(\omega_{c_2}-P_2\frac{d\varphi_{D2}}{dt}\right)-M_{D2};\quad(3)$$
$$\mathrm{T}_{e2}=(\omega_{c_2}\cdot S_{k2})^{-1};\quad \psi_2=\frac{S_{k2}}{S_2}\left(M_{D2}+\mathrm{T}_{e2}\frac{dM_{D2}}{dt}\right);$$
$$I_{D2}\frac{d^2\varphi_{D2}}{dt^2}=M_{D2}-M_{Dn1};$$
$$I_{n1}\frac{d^2\varphi_{n1}}{dt^2}=U_{в}U_{Dn1}-M_{n1n2}-M_{n1};$$
$$I_{n2}\frac{d^2\varphi_{n2}}{dt^2}=U_{n1n2}M_{n1n2}-M_{n2};$$

where, φ_{D2}, φ_{n1}, φ_{n2} - angular displacement of the motor rotor and feed shafts; M_{D2}, $M_{к2}$ - driving moment and its practical significance; I_{D2}, I_{n1}, I_{n2} - moments of inertia of the rotor and feed rollers; M_{Dn1}, M_{n1n2}, M_{n1}, M_{n2} - interacting moments between masses and technological loads on the feed rollers; $U_{в},U_{Dn1},U_{n1n2}$ - gear ratios between the corresponding masses.

Differential equations for the removable drum section (Fig. 2,c):

$$\frac{dM_{D3}}{dt}=\left(\omega_{c_3}-P_3\frac{d\varphi_{D3}}{dt}\right)\psi_3-\frac{M_{D3}}{\mathrm{T}_{e3}};\quad \frac{d\psi_3}{dt}=\frac{2M_{k3}-\psi_3}{\mathrm{T}_{e3}}-\left(\omega_{c_3}-P_3\frac{d\varphi_{D3}}{dt}\right)-M_{D3};$$

$$\mathrm{T}_{e3}=(\omega_{c_3}\cdot S_{k3})^{-1};\quad \psi_3=\frac{S_{k3}}{S_3}\left(M_{D3}+\mathrm{T}_{e3}\frac{dM_{D3}}{dt}\right); \tag{4}$$

$$I_{D3}\frac{d^2\varphi_{D3}}{dt^2}=M_{D3}-\mathrm{C}(\varphi_{D3}-U\varphi_c)-\mathrm{в}\left(\frac{d\varphi_3}{dt}-U\frac{d\varphi_c}{dt}\right);$$

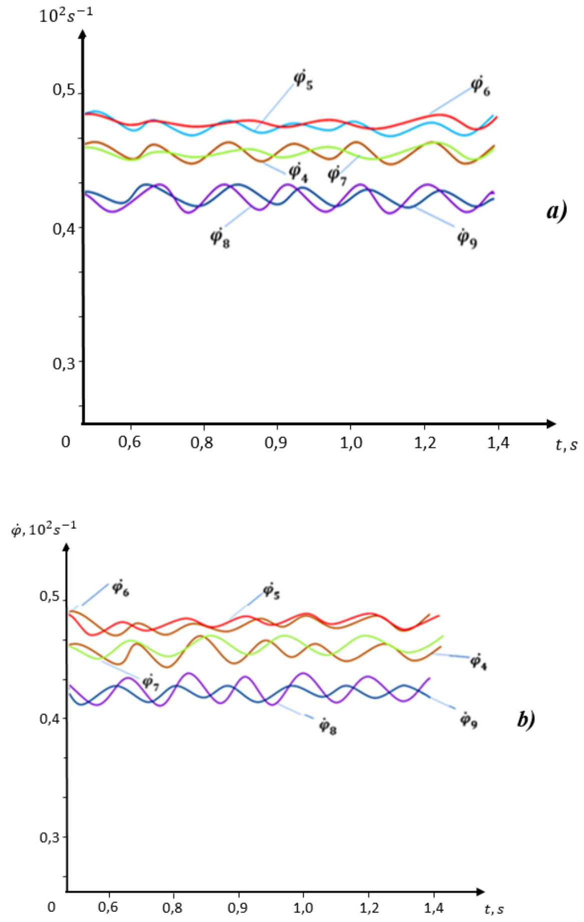
$$I_c\frac{d^2\varphi_c}{dt^2}=UC(\varphi_{D3}-U\varphi_{\mathrm{c}})+U\mathrm{в}\left(\frac{d\varphi_{D3}}{dt}-U\frac{d\varphi_{\mathrm{c}}}{dt}\right)-M_{\mathrm{c}};$$

where, φ_{D3}, φ_{c}- angular displacement of the engine rotor and removable drum; I_{D3}, I_c- moments of inertia of the motor rotor and removable drum; *c, b* -coefficients of circular stiffness and dissipation of the belt drive; U- gear ratio of the belt drive; M_{c}- moment of resistance from cotton; ω_{c_1}, ω_{c_2}, ω_{c_3}- circular frequencies of motor networks; P_1, P_2, P_3- the number of pole pairs of the corresponding motors; S_1,S_2, S_3, S_{k1}, S_{k2}, S_{k3}- respectively slip and their critical values in engines; ψ_1, ψ_2, ψ_3- auxiliary variables; T_{e1}, T_{e2}, T_{e3}- electromagnetic time constants.

3. SOLVING THE PROBLEM AND ANALYZING THE RESULTS

The solution of the system of differential equations (2) describing the movement of the machine unit with the main working bodies of the cotton ginner with the recommended drives was carried out numerically on a PC using known programs, taking into account the following calculated values of the ginners parameters: $\dot{\varphi}_{D1}=104{,}6s^{-1}$; $\dot{\varphi}_1=31{,}4s^{-1}$; $\dot{\varphi}_3=31{,}4s^{-1}$; $\dot{\varphi}_2=13{,}65s^{-1}$; $\dot{\varphi}_4=50{,}24s^{-1}$; $\dot{\varphi}_5=50{,}24s^{-1}$; $\dot{\varphi}_6=50{,}24s^{-1}$; $U_{D1}=3{,}33$; $U_{12}=2{,}3$; $U_{13}=1{,}0$; $\widetilde{U}_{34}=0{,}625$; $U_{45}=1{,}0$; $\widetilde{U}_{56}=1{,}0$; $\dot{\varphi}_7=50{,}24s^{-1}$; $\dot{\varphi}_8=50{,}24s^{-1}$; $\dot{\varphi}_9=50{,}24s^{-1}$; $\widetilde{U}_{37}=0{,}625$; $\widetilde{U}_{78}=1{,}0$; $U_{89}=1{,}0$; $n_{D1}=1000\,min^{-1}$; $f_c=50\,Hz$; $\cos\varphi=0{,}84$; $\omega_0=157{,}1\,s^{-1}$;η=0,82; $\omega_{\mathrm{c}}=101{,}48\,s^{-1}$; $S_{\mathrm{H}}=0{,}056$; $S_{\mathrm{K}}^{`}=0{,}192$; $\mathrm{P}=2$; $\mathrm{C}_3=\mathrm{C}_4=\frac{(255\div325)\boldsymbol{Nm}}{\boldsymbol{rad}}$; $\mathrm{C}_5=\mathrm{C}_7=\frac{(255\div265)\boldsymbol{Nm}}{\boldsymbol{rad}}$; $\mathrm{C}_6=\mathrm{C}_8=\frac{(200\div220)\boldsymbol{Nm}}{\boldsymbol{rad}}$; $\mathrm{в}_3=\mathrm{в}_4=\frac{(4{,}5\div5{,}0)\boldsymbol{Nms}}{\boldsymbol{rad}}$; $\mathrm{в}_5=\mathrm{в}_7=\frac{(4{,}0\div4{,}5)\boldsymbol{Nms}}{\boldsymbol{rad}}$; $\mathrm{в}_6=\mathrm{в}_6=\frac{(3{,}5\div4{,}0)\boldsymbol{Nms}}{\boldsymbol{rad}}$; $\mathrm{в}_0=\frac{(7{,}0\div7{,}5)\boldsymbol{Nms}}{\boldsymbol{rad}}$; $\mathrm{в}_1=\mathrm{в}_2=\frac{(5{,}5\div6{,}5)\boldsymbol{Nms}}{\boldsymbol{rad}}$; $\mathrm{C}_0=\frac{(350\div385)\boldsymbol{Nm}}{\boldsymbol{rad}}$; $\mathrm{C}_1=\mathrm{C}_2=\frac{(340\div350)\boldsymbol{Nm}}{\boldsymbol{rad}}$; $I_{D1}=1{,}06\,kgm^2$; $I_1=I_3=3{,}31\,kgm^2$; $I_2=3{,}74\,kgm^2$; $I_4=I_7=2{,}34\,kgm^2$; $I_5=I_6=I_7=I_8=(1{,}85\div2{,}15)kgm^2$; $\mathrm{M}_{n1}=\mathrm{M}_{n2}=(1{,}8\mp0{,}3\mathrm{M}_{n0})\cdot10^2Nm$; $\mathrm{M}_8=\mathrm{M}_9=(0{,}6+0{,}05\sin\omega_4t)\cdot10^2Nm$; $\mathrm{M}_4=\mathrm{M}_7=(0{,}8+0{,}06\sin\omega_5t)\cdot10^2Nm$; $\mathrm{M}_6=\mathrm{M}_5=(1{,}1+0{,}08\sin\omega_6t)\cdot10^2Nm$; $\mathrm{M}_B=(1{,}2+0{,}1\sin\omega_2t)\cdot10^2Nm$;

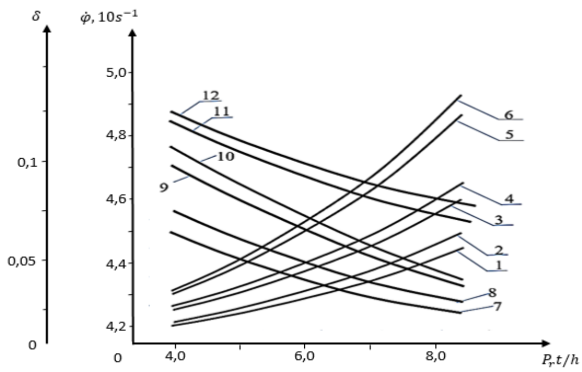
We carry out the numerical solution of the problem on a PC using a number of well-known standard programs. The research results obtained the regularities of changes in the angular velocities of the peg, transporting drums at different values of the purifier productivity, which seen in Fig.3. At the same time, changes in $\dot{\varphi}_4,\dot{\varphi}_5,\dot{\varphi}_6,\dot{\varphi}_7,\dot{\varphi}_8,\dot{\varphi}_9$ are presenting at a steady mode of the purifier movement. It should be noted that during the operation of the cleaner, the raw cotton is transported from the side of the splitter drum $\dot{\varphi}_9$. Therefore the greatest load falls on drums $\dot{\varphi}_8$ and $\dot{\varphi}_9$. In this case, the decrease in the angular velocities of these peg drums will be greater by $(4{,}5\div6{,}2)s^{-1}$ than the angular velocities $\dot{\varphi}_7$. In the output part of the machine unit, the drum drums rotate with an angular speed $\dot{\varphi}_5=\dot{\varphi}_6=(48{,}2\div49{,}9)s^{-1}$ due to less load on these drums.



a) P_r=5.5 t/h b) 8.5 t/h

Fig.3. Regularities of changes in the angular velocities of the working bodies of the modernized raw cotton cleaner ICGC

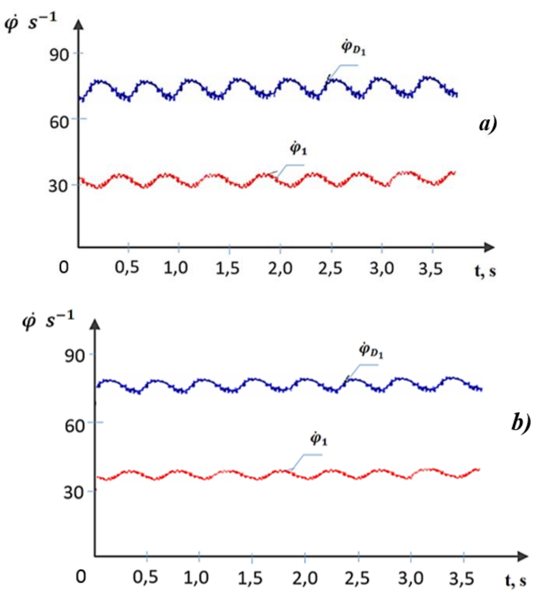
Based on the processing of the obtained regularities in Fig.3 graphical dependences of the change in the average values of angular velocities and their unevenness coefficients of the peg, transporting drums on the performance of the ICGC cleaner with a new drive arrangement built, which shown in Fig.4. Analysis of the graphs shows that an increase in the productivity of the cleaner from 3.5 t/h to 8.0 t/h, the angular velocity of the first peel drum $\dot{\varphi}_6$ decreases minimally, from 45.61 s^{-1} to 42.54 s^{-1}. Accordingly, the angular velocity $\dot{\varphi}_5$ of the second drum decreases from 46.18 s^{-1} to 43.52 s^{-1}. This is because of in the initial zone, raw cotton will be the less loosened and the load on these peg drums will be the greatest, respectively M_5 and M_6.



where, 7-$\dot{\varphi}_6(P_r)$; 8-$\dot{\varphi}_5(P_r)$; 9-$\dot{\varphi}_4(P_r)$; 10-$\dot{\varphi}_7(P_r)$; 11-$\dot{\varphi}_8(P_r)$; 12-$\dot{\varphi}_9(P_r)$;
2-$\delta_8(P_r)$; 1-$\delta_9(P_r)$; 4-$\delta_4(P_r)$; 3-$\delta_7(P_r)$; 6-$\delta_5(P_r)$; 5-$\delta_6(P_r)$;

Fig.4. Dependences of the change in the average values of the angular velocities and their unevenness coefficients of the picking, transporting drums on the performance of the ICGC cleaner with a new drive circuit

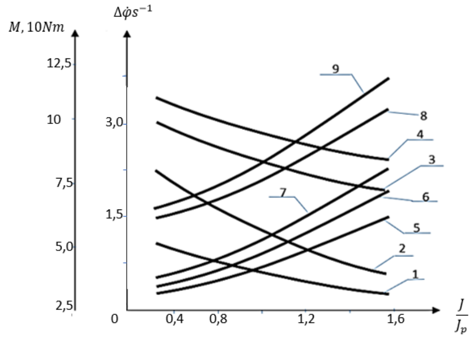
In addition, the difference in the values of the variability of the gear ratio $\widetilde{U}_{56}$ (variable, tension by the eccentric roller) of the $\dot{\varphi}_5$value obtained $(1,0 \div 1,5)s^{-1}$ less than the $\dot{\varphi}_6$values. This difference is obtained between the angular velocities $\dot{\varphi}_4$ and $\dot{\varphi}_7$. It should be noted that the decrease in the angular velocities $\dot{\varphi}_8$ and $\dot{\varphi}_9$will be insignificant; since these peeling drums are affected by the more loosened and pre-cleaned raw cotton. In this case, $\dot{\varphi}_8$ decreases to 46.11 s^{-1}, $\dot{\varphi}_9$ to 47.22 s^{-1}. At the same time, it is important to clean and move the cotton with an increasing angular speed of the peg drums, which ensured the movement of cotton without bottoming. Changes in the amplitude of angular velocities in the operating mode are special. Moreover, the greater the unevenness of the angular velocities of the drums, the greater the effect of loosening and cleaning the cotton. This is due to the fact that, changes in angular velocities lead to angular accelerations of the peg drum, that is, to the appearance of additional impulsive forces. It is important to provide sufficient impulsive forces in the first two peg drums to allow the raw cotton to loosen as needed. Therefore, to obtain sufficient values in the initial zone, belt drives with gear ratios $\widetilde{U}_{34}$ and $\widetilde{U}_{56}$ with eccentric tensioning rollers are used. In the general theory of machines and mechanisms, the moments of inertia of the rotating elements of the system usually increased to reduce fluctuations in angular velocities. Figure 5 shows the patterns of change in the angular velocities of the motor rotor and saw drums of the ICGC cleaner with the recommended drive. The analysis of patterns according to Fig.5 shows that with an increase in the productivity of the cleaner from 5.5 t/h to 8.5 t/h, both high frequency and low-frequency oscillations $\dot{\varphi}_D, \dot{\varphi}_1, \dot{\varphi}_3$ significantly increase.



a) at P_r-5.5 t/h; b) at P_r-8.5 t/h

Fig. 5. Regularities of changes in the angular velocities of the engine rotor and saw drums of the modernized cotton cleaner ICGC

Fig.6 shows the plotted graphical dependencies of changes in the amplitude of oscillations of angular velocities and the load of the rotor shafts of the engine saw and peg drums of the machine unit on their moments of inertia. An increase in the moments of inertia of the saw and peg drums leads to a decrease in the values of the oscillation range of angular velocities, but also to an increase in the torque on the shafts (see Fig.5, curves 5,6,7,8,9).



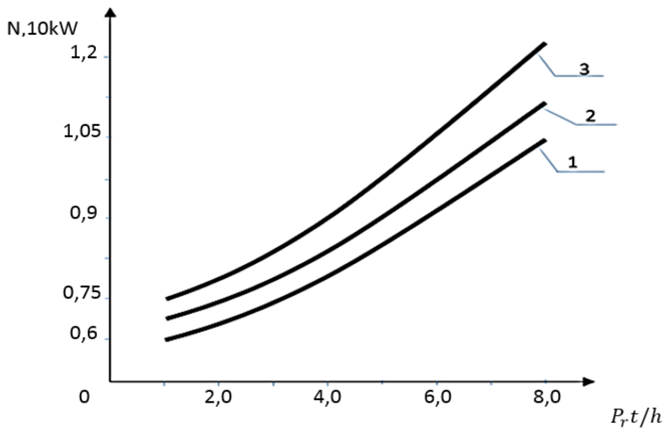
where, $1\text{-}\Delta\dot{\varphi}_{D1} = f\left(\frac{\mathcal{J}}{\mathcal{J}_p}\right)$; $2\text{-}\Delta\dot{\varphi}_8 = f\left(\frac{\mathcal{J}}{\mathcal{J}_p}\right)$; $3\text{-}\Delta\dot{\varphi}_3 = f\left(\frac{\mathcal{J}}{\mathcal{J}_p}\right)$; $4\text{-}\Delta\dot{\varphi}_1 = f\left(\frac{\mathcal{J}}{\mathcal{J}_p}\right)$;

$5\text{-}M_{D1} = f\left(\frac{\mathcal{J}}{\mathcal{J}_p}\right)$; $6\text{-}M_5 = f\left(\frac{\mathcal{J}}{\mathcal{J}_p}\right)$; $7\text{-}M_8 = f\left(\frac{\mathcal{J}}{\mathcal{J}_p}\right)$; $8\text{-}M_1 = f\left(\frac{\mathcal{J}}{\mathcal{J}_p}\right)$;

$9\text{-}M_3 = f\left(\frac{\mathcal{J}}{\mathcal{J}_p}\right)$;

Fig.6. Graphical dependences of changes in the amplitude of oscillations of angular velocities and torques on the shafts of the rotor of the engine saw and peg drums from changes in their moments of inertia

When $\frac{I}{I_p}$ changes for the saw and peg drums under the consideration from 0.5 to 1.5 on each rotating shaft, the torque on the rotor shaft increases from 27.5 Nm to 53.9 Nm, and on the saw drum shaft, respectively M_1 from 59.8 Nm to 98.9 Nm, M_3=(61.7÷109.8) Nm. At the same time, on the shafts of the peg drums, M_8 increases from 33.1 Nm to 76.4 Nm, and M_5 from 31.4 Nm to 72.6 Nm. Accordingly, with an increase in $\frac{I}{I_p}$, the range of angular velocities of the working bodies decreases according to nonlinear laws (see Fig.6 unit 1,2,3,4). According to the results of experiments [8,9,10], the recommended values are $\frac{I}{I_p} = (1{,}2 \div 1{,}25)$ to ensure the required values of the oscillation amplitudes of the angular velocities of the saw and peg drums, which provide a high cleaning effect of raw cotton from coarse and fine litter. Noted that an increase in the moments of inertia of the working bodies and the productivity of the cleaner leads to an increase in the required power of the electric drive. Figure 7 shows the plotted graphical dependences of the change in the required power for the electric motors of the ICGC cleaner on the change in the productivity and rigidity parameters of the belt drives.

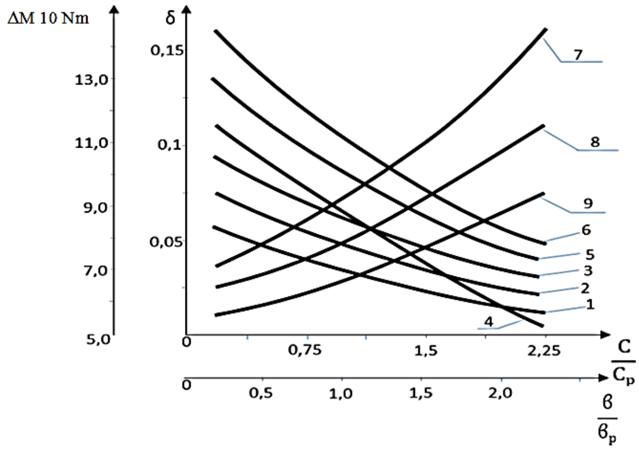


where, 1-at $\frac{C}{C_p} = 1{,}0$; 2-at $\frac{C}{C_p} = 1{,}5$; 1,2-with the recommended drive layout; 3-with existing drive

Fig.7. Graphical dependences of the change in the required power by the electric motor of the ICGC cleaner on the change in the productivity and elastic parameters of belt drives

With an increase in the productivity of the purifier from 2.0 t/h to 8.0 t/h, the required power increases from 7.1 kW to 10.56 kW at $\frac{c}{c_p} = 1{,}0$. When $\frac{c}{c_p} = 1{,}5$ is selected, the power reaches 11.36 kW.

Generally, the required power in the existing version of the ICGC machine drive increases from 0.762 kW to 13.21 kW. The analysis of the obtained graphs shows that the required power in the recommended version of the cotton-cleaning unit drive is reducing on average by (1.4÷2.0) kW in comparison with the existing version of the ICGC machine drive. At the same time, in order to reduce the detailed power and ensure the required values of uneven angular velocities of the peg and serrated drums of the circular stiffness of the belt drives for each working body, it is recommended to select within $\frac{C}{C_p} = (1{,}05 \div 1{,}22)$, and the eccentricities of the tension rollers $e = (1{,}6 \div 2{,}19) \cdot 10^{-3}$M. Fig.8 shows the dependences of the change in the unevenness of the angular velocities to the torque on the shafts of the saw, peg, transporting drums on the change in the elastic - dissipative parameters of the belt drives of the cleaner. As noted above, an increase in the circular rigidity of the belt drive for each working body leads to a decrease in the uneven rotation and an increase in the load. Therefore, with an increase in $\frac{C}{C_p}$ from 0.5 to 2.0, the load on the saw drum increases from 72 Nm to 134 Nm, respectively, M_6 increases to 117 Nm, and M_4 reaches 93.1 Nm. It should be noted that an increase in dissipation coefficients belt drives allow a reduction in ratios irregularities of angular velocities, so δ_3 decreases to 0.03, and δ_4 decreases to 0.035, respectively, δ_6 decreases to 0.049. To ensure the necessary δ values of the working bodies of the cleaner, allowing an increase in the effect of cleaning from fine and large litter, the recommended parameter values are:



where, 1-$\delta_3 = f(\frac{\beta}{\beta_p})$; 2-$\delta_4 = f(\frac{\beta}{\beta_p})$; 3-$\delta_6 = f(\frac{\beta}{\beta_p})$; $4 - \delta_3 = f(\frac{c}{c_p})$; 5-$\delta_4 = f(\frac{c}{c_p})$; 6-$\delta_6 = f(\frac{c}{c_p})$; 7-$M_3 = f(\frac{c}{c_p})$; 8-$M_6 = f(\frac{c}{c_p})$; 9-$M_4 = f(\frac{c}{c_p})$;

Fig.8. Graphical dependences of the change in the unevenness of angular velocities and torques on the shafts of the saw, peg, transporting drums from the change in the elastic-dissipative parameters of the elastic transmissions of the cotton cleaner ICGC with a new drive arrangement

4. Conclusion

The drive of the ICGC machine with an efficient layout recommended using. Based on solving the problem of the dynamics of the machine unit of the purifier, taking into account the elastic-dissipative and inertial parameters, as well as technological loads. In addition, the laws of motion of the saw peg and transport drums and the best parameters of the purifier have been obtained for good to use.

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