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ANALYSING THE RESULTS OF SOLVING THE PROBLEM OF A MACHINE UNIT WITH ACCELERATOR MECHANISM

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Abstract: *Solutions of the obtained nonlinear differential equations of the machine unit are presented using the Maple mathematical program, determining the dependencies of changes in the parameters of the working body in the steady state as a function of the technological load on cotton for different values of the eccentricity of the tension roller of the belt drive. The article presents the changes in the angular velocities of the masses and torque on the engine shaft of the system as a function of time, the dependence of the change in the amplitude of oscillations of the accelerator shaft of the raw chamber and the amplitude of the motor load as a function of the resistance of raw cotton, the dependence of the change in the unevenness coefficients of the rotating shafts of the machine unit are shown as a function of the viscous friction coefficient of the elastic gear. By solving a system of nonlinear and differential equations of the movement of a machine unit, the laws of motion of the accelerator shaft and the rotor of the electric motor were determined. It was found that with increasing load on the flap, the fluctuation range of the angular velocity of the accelerator roller increases, as well as the fluctuation range of the torque on the motor shaft.*

Keywords: *machine unit, options,working body, process load, tension roller, roller accelerator, eccentricity, differential equations, not linear equations*

Annotatsiya: *Maple matematik dasturi yordamida mashina blokining olingan nochiziqli differensial tenglamalari yechimlari keltirilgan, bunda barqaror holatdagi ishchi organ parametrlarining o‘zgarishining paxtadan texnologik yuklamaga bog‘liqligi kamarni taranglovchi rolikining eksantrikligi qiymatlari turli qiymatlari uchun olingan. Tizimning dvigatel validagi massalar va momentning burchak tezligining vaqtga bog‘liq o‘zgarishi, xomashyo kameraning tezlatgich valining tebranishlari amplitudasining o‘zgarishiga va dvigatel yuklanishining amplitudasiga bog‘liqligi. paxta xomashyosiga qarshilik funksiyasi, elastik uzatmaning yopishqoq ishqalanish koeffitsientiga bog‘liqligi, mashina agregatining aylanadigan vallari notekislik koeffitsientlarining o‘zgarishiga bog‘liqligi keltirilgan. Mashina agregati harakatining chiziqli bo‘lmagan va differentsial tenglamalari tizimini yechish orqali tezlatgich vali va elektr motor rotorining harakat qonunlari olingan. Aniqlanishicha, paxtadan yuklanish ortishi bilan tezlashtiruvchi rolikning burchak tezligidagi tebranishlar diapazoni, shuningdek, dvigatel validagi momentning tebranishlari oralig‘i ortadi.*

Tayanch iboralar: *mashina agregati, parametrlar, ishchi organ, texnologik yuklanish, taranglovchi rolik, valik tezlatkichi, ekstsentriklik, tebranish diapazoni, paxta xomashyosidan qarshilik, aylanma notekislik koeffitsienti.*

Аннотация: *Приведены решения полученных нелинейных дифференциальных уравнений машинного агрегата с помощью математической программы maple, где получены зависимости изменения параметров рабочего органа в установившемся режиме в функции технологической нагрузки от хлопка при различных значениях эксцентриситета натяжного ролика ремённой передачи. Приведены изменения угловых скоростей масс и момента на валу двигателя системы в зависимости от времени, зависимости изменения размаха колебаний вала ускорителя сырцовой камеры и размаха нагруженности двигателя в функции сопротивления от хлопка - сырца, зависимости изменения коэффициентов неравномерностей вращающихся валов машинного агрегата в функции коэффициента вязкого трения упругой передачи. Решением системы нелинейных и дифференциальных уравнений движения машинного агрегата получены законы движения вала ускорителя и ротора электродвигателя. Установлено, что с увеличением нагрузки от хлопка возрастает размах колебаний угловой скорости ускоряющего валика, а также размах колебаний момента на валу двигателя.*

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Ключевые слова: *машинный агрегат, параметры, рабочий орган, технологическая нагрузка, натяжной ролик, ускоритель валика, эксцентриситет, размах колебаний, сопротивление от хлопка - сырца, коэффициент неравномерности вращения.*

Introduction

Due to the high density of the raw material roll, high dynamic loads occur in the working chamber of the saw gin. This leads to increased seed damage and increased defects in the fibre. Some researches in this direction have been carried out.

In the article [1] the calculation scheme and mathematical model of the machine unit with the accelerator mechanism of the raw material chamber of the saw fibre separator, taking into account the inertial, elastic-dissipative properties of the elements, as well as the variability of the transmission ratio of the elastic gear and the technological load from raw cotton, are drawn up.

In the article [2, 3] the results of the experimental study of the developed profile of the working chamber of the saw gin with a throwing drum are given. The influence of the performance of the saw gin with a pick-up drum and a double drum feeder on the power consumption of the electric motor, the density of the raw cotton roll, fibre content and raw cotton roll speed using experimental studies is studied.

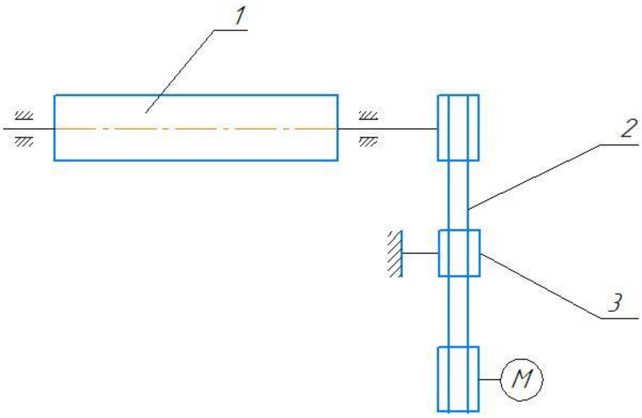

Fig. 1. Kinematic diagram of the saw gin accelerator drive: 1 - accelerator, 2 - belt transmission of the accelerator drive, 3 - belt tensioner.

The paper [4] presents an efficient resource-saving design solution of the elastic bearing support of the saw gin. Expressions for calculating the amplitude and frequency of oscillations of the saw cylinder on the elastic support as a two-mass system of the machine unit are obtained. Formulas for determining the laws of motion of the electric motor rotor and the saw cylinder shaft are obtained analytically.

By solving the differential equations, taking into account the dynamic characteristics of the asynchronous electric motor, the formula for the transmission ratio between the driving and driven shafts of the accelerator drive of the raw material roller with an eccentric idler is derived.

In the drive of the raw material roller accelerator there is a tension roller (Fig. 1), which periodically changes the belt sliding, due to which the accelerator rotates with variable frequency, giving the raw material roller impulsive rotation. This results in, firstly, shaking of the raw material roller, secondly, cyclic change of the rarefaction in the seed comb area, and thirdly, due to the impulsive rotation of the raw material roller, a variable centrifugal force is generated. The raw cotton is periodically thrown onto the saw blade teeth and the bare seeds are intensively released.

The cyclic changes in rarefaction lead to a more complete discharge of the denuded seeds from the raw material chamber [5]. The impulsive force in the raw material roller, resulting from the uneven rotation of the accelerator, favours better capture of raw cotton slices by the saw teeth and their better retention on the saw teeth. This leads to a more complete separation of the fibre from the seed.

Statement of the Problem. The solution method is based on solving the obtained nonlinear differential equations of the machine unit using the maple mathematical programme [6-10].

The equation describing the motion of two mass elastic system [1, 11, 12]:

$$I_1\omega_1 = \frac{M_1 - e_p(\omega_1 - i_{12}\omega_2)\cdot\left\{1 - \frac{D_2}{D_1}EF\left[\left(e^{\mu\alpha}-1\right)\cdot S_k\cdot\left(a\sin\varphi_1\cdot i_b + \frac{a^2\sin\varphi_1\cdot i_b}{\sqrt{r^2a^2\sin^2\varphi_1\cdot i_b} - r}\right)\right]\right\}}{\left[EF - \left(e^{\mu\alpha}-1\right)\cdot\left(S_0 + \cos\varphi_1\cdot i_b + \sqrt{r^2a^2\sin\varphi_1}\cdot i_b - r\right)\cdot S_k\right] - \beta_{BT}(\omega_1 - i_{12}\omega_2)}; \tag{1}$$

$$I_2\omega_2 = e_p(\omega_1 - i_{12}\omega_2)i_{12} + \beta_{BT}(\omega_1 - i_{12}\omega_2)i_{12} - M_2.$$

The transmission ratio between the drive and driven shafts with eccentric idler is expressed by the equation:

$$i_{12} = \frac{D_2 EF}{D_1 \left\{ EF - \left(e^{\mu\alpha} - 1\right) \left[S_0 + \alpha \cos\left(\varphi_1 \cdot \frac{D_1}{D_2}\right) + \sqrt{\left(\frac{D_b}{2}\right) - \alpha^2 \sin^2\left(\varphi_1 \cdot \frac{D_1}{D_b}\right)} - \frac{D_b}{2} \right] \cdot S_k \right\}}, \quad (2)$$

here : D_1 - pulley diameter on the motor shaft; D_2 - idler pulley diameter; D_b - idler pulley diameter; E - belt elasticity modulus; F - belt cross-sectional area; S_0 - initial belt tension; α - belt girth angle equal to the sliding arc; e_p - eccentricity of the idler; i_b - idler shaft ratio; S_k - belt tension coefficient; β_{cf} - viscous friction coefficient; e - base of natural logarithms; μ - belt friction coefficient;

For information processing and calculations it is necessary to set initial conditions and initial parameters of the system. To solve the problem we apply the following initial conditions of the system: t = 0; angular displacements of the motor rotor φ1 and accelerator shaft φ_2, $\varphi_1 = i_{12}\varphi_2$; M_1 - moment on the motor shaft, M_2 - moment of forces of technological resistance of fibrous material acting on the accelerator of the raw roll, $M_1 = 0$; $M_2=0$.

Let's define the initial parameters of the system.

Drive accelerator raw material roller is carried out by asynchronous electric motor brand A02-42-8 with the parameters [13]: motor power N = 3,0 kW; nominal revolutions n = 720 r/min; ratio of starting torque to nominal $M_{start}/M_{nom} = 1,2$; ratio of maximum torque to nominal (overload capacity) $\lambda = M_{max}/M_{nom} = 1,7$; network frequency f = 50 Hz; number of poles P = 4; flywheel motor torque $GD^2 = 1,7$ Nm.

Let's calculate the necessary parameters of the motor.

Nominal angular speed

$$\omega_H = \frac{\pi \rightleftarrows n}{30} = 75{,}36\ rad/\text{сe}k. \quad (3)$$

Angular speed of idle speed of the motor rotor

$$\omega_0 = \frac{2\pi \rightleftarrows f_e}{P} = 78{,}5 rad/\text{сe}k \quad (4)$$

Moment of inertia of the electric motor rotor

$$J_1 = \frac{GD^2}{4g} = 0{,}04332 n\text{мс}^2. \quad (5)$$

Rated motor torque:

$$M_N = \frac{9550 \cdot N_H}{n_H} = 39{,}79\,\text{нм}. \quad (6)$$

The critical torque of an electric motor:

$$M_k = \lambda M_H = 1{,}7 \cdot 39{,}79 = 67{,}5\ \text{Нм}. \quad (7)$$

Nominal value of slip:

$$S_H = \frac{\omega_0 - \omega_H}{\omega_0} = 0{,}0394\,. \quad (8)$$

The critical value of slip:

$$S_K = \lambda S_H \left(1 + \sqrt{1 - \frac{1}{\lambda^2}}\right) = 0{,}121\,. \quad (9)$$

The machine unit was calculated with the following machine parameters:

$J_1 = 0{,}04332$ Нмс^2;
$J_2 = 1{,}57$ Нмс^2.

For maintenance of necessary non-uniformity of angular velocities of masses for rotary currents of technological machines, we take a variant of parameters of eccentricity ep, coefficient of viscous friction of elastic transmission βBT, moment of inertia of accelerator shaft J2, moment of resistance from technological load Me.

J2 = 0.5...3 Nms^2.
Stiffness coefficient of the elastic transmission

$$C = \frac{EF}{e_p}. \quad (10)$$

For Type A and B belts that can be used to drive a gin accelerator,

$$C = 1515\ldots2580\ \text{нм}/\text{рад}.$$

Viscous friction coefficient of elastic transmission

$$\beta_{BT} = \frac{M}{\Delta\varphi_1} = 2...8\ нмс/рад. \qquad (11)$$

These values of stiffness and viscous friction coefficient are established according to known calculations [14, 15].

Moment of resistance from technological load (from the result of experiment)

Mc = 0...45 Nm.

At the software solution of the problem the following parameters were varied: amplitude of oscillations of the transmission ratio; technological load from raw cotton; moments of inertia of the machine unit masses; stiffness and dissipation coefficients of the elastic transmission; eccentricity of the tension roller.

On the basis of the problem solution the laws of motion of rotating masses of the machine unit are obtained. It is revealed that due to the variable transmission ratio of the belt transmission the start-up process of the system is delayed up to 0.1s. Fluctuations of angular velocities and moments on the shafts of the machine unit are mainly caused by the variability of the transmission ratio and non-uniformity of the technological load of raw cotton.

The technological load is selected within the range of (3.5-2.5) Nm. Fig. 2 shows the variation of angular velocities of masses and torque on the motor shaft of the system. The range of torque fluctuations is (15... 45) Nm, and the non-uniformity of the angular velocity on the raw material roller is δ2 = 0,05...0,15. Unevenness of rotation of the driven shaft arises from the technological resistance of cotton. The unevenness of the idler shaft rotation affects the rotation of the driving shaft and the motor torque.

Fig. 3 shows the graphical dependences of the change in the oscillation spread ΔMg, Δφ2 in steady-state mode as a function of the technological load from the cotton at different values of the eccentricity of the belt transmission idler roller. The figure shows that with the increase in the saw gin productivity, the range of oscillation of the angular velocity of the accelerating roller of the raw material chamber increases, not linearly. At the eccentricity of the idler roller e = 3,0 mm and resistance from cotton Mx = 12 Nm, the increase in the range of oscillations of the accelerating roller speed reaches from 4,5 rad/s to 18 rad/s., and at e = 1,5 mm the value of Δφ2 increases from 3,4 rad/s to 16 rad/s at Mx = 15 Nm. In the first case, the average variation of the accelerating roller speed is 6.75 rad/s, and in the second case, 6.3 rad/s.

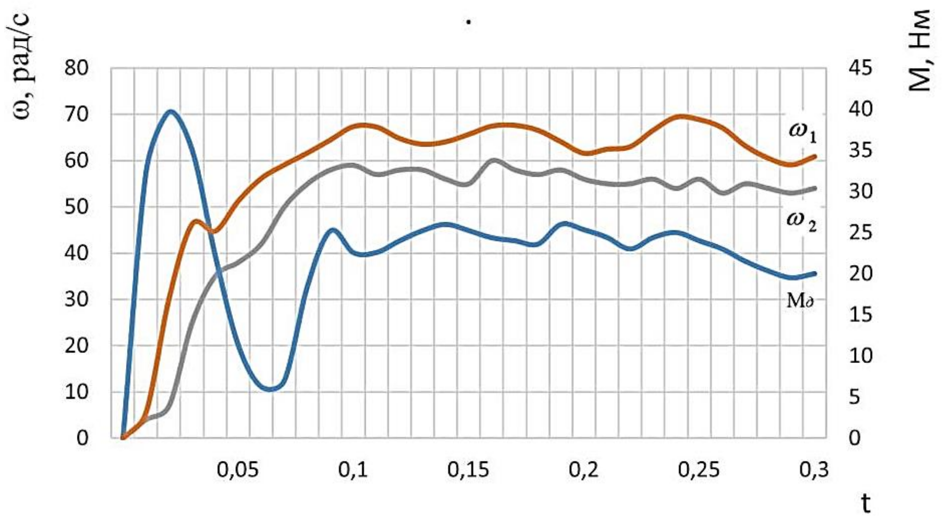

Fig 2. Time variation of the accelerator shaft angular velocity and motor shaft torque.

It should be noted that as the cotton load increases, the torque on the motor shaft also increases. We were interested in how the load on the motor shaft fluctuates with increasing machine output. Thus, when the eccentricity of the idler roller e = 3.0 mm, ΔMg increases from 1.8 Nm to 8.5 Nm at Mx = 20.5Nm, and when e = 1.5 mm, ΔMg increases from 1.2 Nm to 4.9 Nm at Mx = 20.5Nm. This is justified by the fact that with the increase in the eccentricity of the idler roller the constant component of the change in Δφ2 and ΔMg increases proportionally, and with the increase in productivity due to the variability of the load from the cotton additionally increases the angular velocity spread of the accelerating roller and the torque spread on the motor shaft.

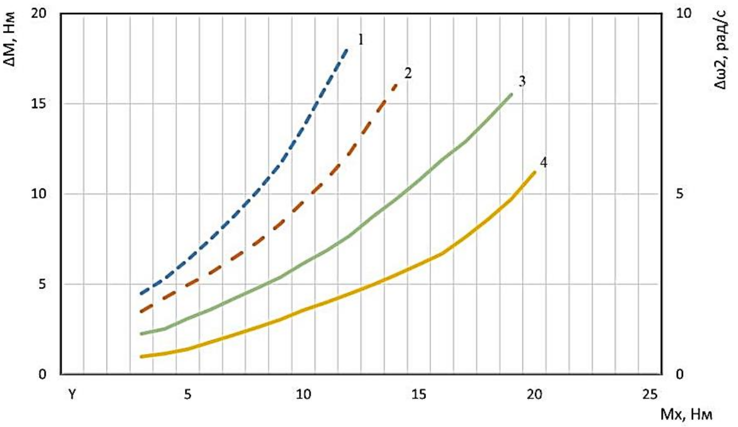

Fig 3. Dependences of the change of vibration spread of the accelerator shaft of the raw chamber Δω2 and motor load spread Δ_{Mg} in the function of resistance from cotton - raw material

1 - $\Delta_{\omega 2}$ at eccentricity e = 3 mm; 2 - $\Delta_{\omega 2}$ at e = 1.5 mm; 3 - Δ_{Mg} at e = 3 mm; 4 - Δ_{Mg} at e = 1.5 mm.

Important are the researches of the machine unit with the mechanism of the accelerating shaft of the raw material chamber of the saw gin at variation of parameters of the elastic transmission: stiffness and viscous friction coefficients and eccentricity of the idler roller of the transmission [15]. It is necessary to consider the recommended parameters of the elastic transmission, at which the necessary fluctuations of the angular velocity of the accelerating shaft, intensification of bare seed separation and fibre separation process are provided.

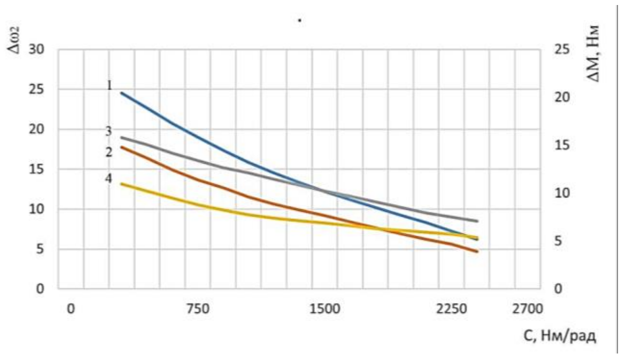

Fig 4. Graphical dependences of the change of the oscillation range of the angular velocity of the accelerating roller $\Delta\omega_2$ and the load range on the motor shaft ΔM as a function of the stiffness coefficient of the elastic transmission C: 1- $\Delta\omega_2$, 3 - ΔMg at the eccentricity of the idler roller e = 3 mm; 2 - $\Delta\omega_2$ and 4 - ΔMg at e = 1.5 mm.

Fig. 4 shows the graphical dependences of the change in the range of oscillations of the angular velocity of the accelerating roller and the range of load on the motor shaft ΔMg as a function of the stiffness coefficient of the elastic transmission C. With increasing stiffness coefficient, the range of oscillations of the angular velocity Δω2 and the range of load on the motor shaft ΔMg decreases. Thus, at e = 3.0 mm, Δω2 decreases from 23.5 rad/s at C = 250 Nm/rad, to 8.5 rad/s at C = 2250 Nm/rad. The decrease in the sweep of the accelerating roller angular velocity oscillations with increasing stiffness of the belt transmission is explained by the fact that the system becomes more rigid, close to unity. In the same way it is possible to explain the decrease of the torque spread on the motor shaft with increasing stiffness of the belt transmission C (see Fig. 2.5, curves 3, 4). In addition, it should be noted that the decrease of Δω2 and ΔMg are different for different eccentricities e = 3.0 mm and e = 1.5 mm (see Fig. 4, curves 1,2,3,4).

Taking into account the requirements of seed separation and ginning technology, the recommended values of stiffness coefficient of elastic transmission are 1000...2000 Nm/rad at eccentricity e = 1.5 mm and 1700...2000 Nm/rad at e = 3.0 mm. At these values the angular velocity spread Δω2 = 10...12 rad/s is provided.

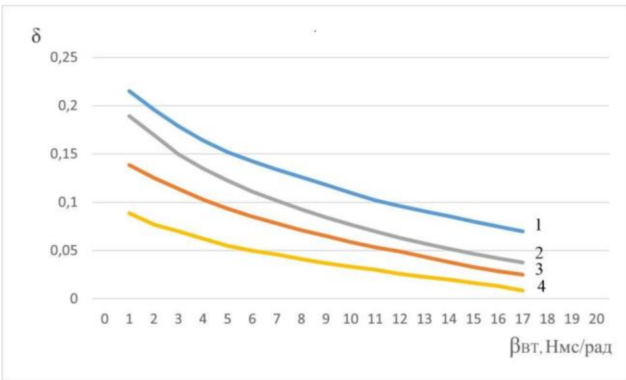

Fig 5. Graphical dependences of changes in the irregularity coefficients of the rotating shafts of the machine unit δ as a function of the viscous friction coefficient of the elastic transmission βVT:

1 - δ1, 3 - δ2 at eccentricity e = 3 mm; 2 - δ1, 4 - δ2 at e = 1.5 mm.

The influence of the viscous friction coefficient of the elastic transmission on the character of motion of the machine unit is evaluated highly from the point of view of the dynamics of the investigated jin machine unit. Fig. 5 shows the obtained curves of dependence of change of the coefficients of non-uniformity of angular velocities δ1 and δ2 of the accelerator shaft of the raw chamber and the rotor of the electric motor as a function of the coefficient of internal viscous friction of the elastic kinematic coupling βVT (belt transmission with variable ratio) [16].

Increase of viscous friction coefficient βVT leads to decrease of coefficients of non-uniformity of angular velocities δ1 and δ2. So, at e = 3,0 mm δ1 decreases from 0,22 to 0,09, and at e = 1,5 mm from 0,15 to 0,03 at increase of βVT from 1,2 Nms/rad to 17,5 Nms/rad (see fig.5, curves 1, 3). The decrease in the motor rotor irregularity coefficient is significant at e=3.0mm, from 0.2 to 0.04, and at e=1.5mm is insignificant (from 0.1 to 0.025). Acceptable values of the viscous friction coefficient are 3.0 and 5 Nms/rad, at e = 1.5mm and 10...12 Nms/rad at e = 3.0mm. At these values of viscous friction coefficient βVT the necessary values of δ1 (0.13...0.15) and δ2 (0.03...0.08) are provided.

Conclusion

1. By solving the system of nonlinear and differential equations of motion of the machine unit, the laws of motion of the accelerator shaft and the electric motor rotor are obtained. It is established that with the increase of the cotton load, the range of fluctuations of the angular velocity of the accelerating roller increases, as well as the range of fluctuations of the torque on the motor shaft.
2. Taking into account the requirements of the technology of seed separation and ginning recommended values of the stiffness coefficient of the elastic transmission is 1000 ... 2000 $N_m/_{rad}$ at the eccentricity of the tension roller e = 1.5 mm and 1700 ... 2000 Nm/rad at e = 3.0 mm. At these values the angular speed of the driven shaft ω_2 = 10...12 rad/s is provided.
3. Increase in the viscous friction coefficient of the elastic transmission leads to a decrease in the coefficients of unevenness of rotation of shafts δ_1 and δ_2. The applied values β_{VT} = 3,0...5,0 Nm s/rad at e = 1,5 mm, β_{VT} = 10...12 Nm s/rad at e = 3, 0 mm.

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