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INVESTIGATION OF THE SAW CYLINDER OF A LINTER MACHINE WITH DISTRIBUTED PARAMETERS

ИССЛЕДОВАНИЕ ПИЛЬНОГО ЦИЛИНДРА ЛИНТЕРНОЙ МАШИНЫ С РАСПРЕДЕЛЕННЫМИ ПАРАМЕТРАМИ

D.M. MUKHAMMADIEV, F.KH. IBRAGIMOV, O.Kh. ABZOIROV

Д.М. МУХАММАДИЕВ, Ф.Х. ИБРАГИМОВ, О.Х. АБЗОИРОВ

(Institute of Mechanics and Seismic Stability of Structures named after M.T. Urazbaev of the Academy of Sciences of the Republic of Uzbekistan)

(Институт механики и сейсмостойкости сооружений им. М.Т. Уразбаева Академии наук Республики Узбекистан)

E-mail: davlat_mm@mail.ru

The article presents the results of research on the machine unit of the saw cylinder of a linter machine with distributed parameters. For this, subsystems were used with concentrated parameters (Lagrange equations of the second kind) and distributed parameters (Laplace's equation in cylindrical coordinates). As a result, the study of the machine unit of the saw cylinder of a linter machine with concentrated parameters according to the characteristic of the asynchronous electric motor proposed by A.E. Levin showed that the critical driving moment of the electric motor is 340.92 N·m, the transient process lasts for 0.8 s, and the maximum value of the angular acceleration of the saw cylinder of the linter machine reaches 675.05 rad/s^2 at t=0.223 s. An asynchronous electric motor 4A200M8U3 with a power of 18.5 kW and a rotation speed of 735 rpm, and a rated torque of 240 N·m was installed on the rotor shaft. Calculations have established an increase in the starting torque by 340.92/240=1.42 times. In addition, the maximum values of the angle of relative rotation and the angle of rotation of the saw cylinder of the linter machine under torsion were set; they are 0.729 °/m and 0.523 °(at t=0.519 s), respectively. In general, the study of machines in the form of machine units made it possible to establish the dynamics of starting the electric motor and torsional vibrations of the saw cylinder of a linter machine with distributed parameters.

В статье приведены результаты исследований машинного агрегата пильного цилиндра линтерной машины с распределенными параметрами. Для этого были использованы подсистемы как с сосредоточенными параметрами (уравнения Лагранжа II рода), так и с распределенными параметрами (уравнение Лапласа в цилиндрических координатах). В результа-

те изучение машинного агрегата пильного цилиндра линтерной машины с сосредоточенными параметрами по предложенной характеристике асинхронного электродвигателя А.Е. Левина показало, что критический движущий момент электродвигателя составляет 340.92 Н·м, переходный процесс протекает в течение 0,8 с, а максимальное значение углового ускорения пильного цилиндра линтерной машины достигает 675,05 рад/с^2 при t=0,223 с. При этом использован асинхронный электродвигатель 4А200М8У3 с мощностью 18,5 кВт, частотой вращения 735 об/мин и номинальным крутящим моментом 240 Н·м. Расчетами установлено увеличение пускового момента в 340,92/240=1,42 раза. Кроме того, определены максимальные значения угла относительного поворота и угла поворота пильного цилиндра линтерной машины при кручении, которые соответственно равны 0,729°/м и 0,523° (при t=0,519 с). Получена математическая модель крутильных колебаний пильного цилиндра линтерной машины с распределенными параметрами.

Keywords: linter machine, saw cylinder, machine unit with lumped and distributed parameters, electric motor, torsional vibrations, mathematical model, angle of shaft rotation.

Ключевые слова: линтерные машины, пильный цилиндр, машинный агрегат с сосредоточенными и распределенными параметрами, электродвигатель, крутильные колебания, математическая модель, угол поворота вала.

Introduction

I.I. Artobolevsky proposed to study machines in the form of machine units; this makes it possible to more accurately assess the dynamic processes occurring in the “drive-transmission-actuator” system under the influence of technological loads [1].

I.I. Vulfson [2, 3] used an idealized calculation scheme in the form of a subsystem with distributed parameters to reduce the complexity of the machine unit calculation.

N.S. Piskunov [4] considered the equation of torsional vibrations of a homogeneous cylindrical rod in the form of the Laplace equation.

In [5], to determine the pattern of change in the frequency and rotation irregularities of the rotor in the electric motor and saw cylinder, depending on the elastic-dissipative parameters of the coupling, the moment of inertia of the electric motor, the moment of inertia and the resistance of the saw cylinder, the equation of motion of the machine unit of the 156-saw cylinder of the gin were used.

In the article by D.M. Mukhammadiev et al. [6], the dynamic characteristics of the gin saw cylinder were considered in the form of the Laplace equation, as a subsystem with concentrated and distributed parameters using the characteristics of an asynchronous electric motor proposed by M.M. Sokolov.

It is known that when an electric motor is connected to the network, an increase in the starting torque by 1.5–6 times relative to the nominal one is observed [7]. This process takes place at the maximum load on the electric motor at the time of starting, which indicates the need to study the dynamic processes occurring in machine units using various characteristics of asynchronous electric motors.

The saw cylinder of a linter machine consists of an electric motor and the saw cylinder with a coupling.

Therefore, the objectives of the research are to find the pattern of change in the angular acceleration of the rotation of the rotor of electric motor and the saw cylinder as a function of time, taking into account the elastic-dissipative parameters of the coupling, the moment of inertia and the moment of resistance of the electric motor and the saw cylinder using the equation of motion of the machine unit of the saw cylinder, which provides the normal mode of linter machine operation. To do this, it is necessary to take into account the elasticity of the links and damping factors (dissipation) of the drive, while the elasticity and dissipation of supports due to the generalization of the system coordinates are not taken into account.

To study the dynamic parameters of the saw cylinder of a linter machine, we consider the machine unit as a system consisting of subsystems with concentrated and distributed parameters. In accordance with data obtained in [1, 5], a mathematical model of the first subsystem with concentrated parameters was developed; the model for the subsystems with distributed parameters, was developed according to the data obtained in [2–4, 6].

Materials and methods

I. Saw cylinder subsystem of linter machine with concentrated parameters

As follows from the dynamic model of the saw cylinder (Fig. 1a), the angular displacement of the electric motor (D) through the coupling is transmitted to the long saw cylinder (PS), the torsional vibrations of which can be quite significant. In the dynamic model of the saw cylinder accepted (shown in Fig. 1a) the following symbols are used: $\Im_d$, $\Im_{ps}$ are the concentrated moments of inertia of the electric motor and saw cylinder, kg·m^2; $\Im$ is the distributed moment of inertia of the saw cylinder and parts rigidly connected with it, kg·m^2; c, в are the coefficients of rigidity (N·m/rad) and dissipation (N·m·f/rad) of the coupling; φ_d, $\varphi_{ps}(x)$ are the absolute coordinates of the corresponding sections, rad; Q(x) is the distributed generalized force applied to the saw cylinder.

Let us take φ_d and $\varphi_{ps}(x)$ as generalized coordinates. Section x=0 divides the dynamic model of the saw cylinder (Fig. 1a) into subsystems with concentrated and distributed parameters, where two reactive moments M– and M+ are applied (Fig. 1b); these moments are equal in magnitude and opposite in direction (M+ =-M–). The reactive moment in the "output" of the element (on the right) is taken as the positive direction of the angles count $\varphi_{ps}(x)$, and for the reactive moment in the "input" of the element (on the left), it is negative.

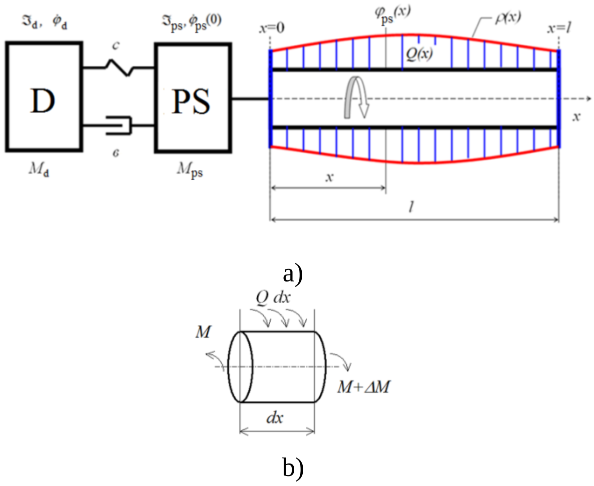


Fig. 1

When deriving differential equations for the saw cylinder of a linter machine, the Lagrange equation of the second kind is used:

$$\frac{d}{dt}\left[\frac{\partial T}{\partial \dot{\phi}_i}\right]-\frac{\partial T}{\partial \phi_i}+\frac{\partial L}{\partial \phi_i}+\frac{\partial \Gamma}{\partial \phi_i}=Q\left[\phi_i\right]. \quad (1)$$

The dynamic model of the machine unit and the kinematic diagram of the saw cylinder are shown in Fig. 1a, where $\Im_d$, $\Im_{ps}$ are the moments of inertia of the electric motor and the saw cylinder of the linter machine, respectively, kg·m^2; M_d, M_{ps} are the moments of loads acting on the rotating shaft of the electric motor and the saw cylinder of the linter machine, respectively, N·m; c is the stiffness of the coupling, N·m/rad; в is the dissipation coefficient of the coupling, N·m·f/rad; $\dot{\phi}_d$, $\dot{\phi}_{ps}$ are the angular velocities of the rotor of the electric motor and the saw cylinder of the linter machine, s^{-1}; i is the gear ratio of the coupling.

The saw cylinder drive of the linter machine consists of a coupling. The following kinematic relations are valid:

$$i=1. \quad (2)$$

The angular velocities of the rotating masses of the electric motor and the saw cylin-der of the linter machine $\dot{\phi}_d$, $\dot{\phi}_{ps}$ are tak-en as the generalized coordinates.

The kinetic energy of the saw cylinder of a linter machine is:

$$T = \frac{\Im_d \dot{\phi}_d^2}{2} + \frac{\Im_{ps} \dot{\phi}_{ps}^2}{2}. \qquad (3)$$

The potential energy of the saw cylinder of a linter machine is a homogeneous quadrat-ic form of generalized coordinates and is writ-ten in the following form:

$$L = \frac{1}{2}\left[c(\phi_d - i\phi_{ps})^2\right]. \qquad (4)$$

The dissipative function of the system is expressed as:

$$\Gamma = \frac{1}{2}\left[в(\dot{\phi}_d - i \cdot \dot{\phi}_{ps})^2\right]. \qquad (5)$$

We define the terms of the Lagrangian equations:

a) partial displacement derivatives of po-tential energy –

$$\left.\begin{aligned} \frac{\partial L}{\partial \phi_d} &= c(\phi_d - i \cdot \phi_{ps}) \\ \frac{\partial L}{\partial \phi_{ps}} &= -ci(\phi_d - i \cdot \phi_{ps}) \end{aligned}\right\}, \qquad (6)$$

b) partial displacement derivatives of the dissipative function –

$$\left.\begin{aligned} \frac{\partial \Gamma}{\partial \dot{\phi}_d} &= в(\dot{\phi}_d - i \cdot \dot{\phi}_{ps}) \\ \frac{\partial \Gamma}{\partial \dot{\phi}_{ps}} &= -вi(\dot{\phi}_d - i \cdot \dot{\phi}_{ps}) \end{aligned}\right\}, \qquad (7)$$

c) partial derivatives with respect to veloc-ities of generalized coordinates –

$$\frac{\partial T}{\partial \dot{\phi}_d} = \Im_d \dot{\phi}_d, \quad \frac{\partial T}{\partial \dot{\phi}_{ps}} = \Im_{ps} \dot{\phi}_{ps}; \qquad (8)$$

d) differentiation in time –

$$\left.\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\phi}_d}\right) = \Im_d \cdot \ddot{\phi}_d, \quad \frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\phi}_{ps}}\right) = \Im_{ps} \ddot{\phi}_{ps}\right\}, \qquad (9)$$

e) generalized forces –

$$Q_d(\phi_d) = M_d, \quad Q_{ps}(\phi_{ps}) = -M_{ps}. \qquad (10)$$

Substituting certain terms (6–10) into equation (1), we obtain a system of differen-tial equations for the machine unit movement of the saw cylinder of the linter machine in a general form:

$$\left.\begin{aligned} \Im_d \ddot{\phi}_d &= M_d - c(\phi_d - i\phi_{ps}) - в(\dot{\phi}_d - i\dot{\phi}_{ps}) \\ \Im_{ps} \ddot{\phi}_{ps} &= ci \cdot \phi_d - i\phi_{ps}) + \\ &+ вi(\dot{\phi}_d - i\dot{\phi}_{ps}) - M_{ps} \end{aligned}\right\}. \qquad (11)$$

The asynchronous motor is taken into ac-count in the form of a dynamic characteristic proposed by A.E. Levin [8]:

$$\begin{aligned} \dot{M}_d &= (\omega_c - P\dot{\phi}_d) \cdot \psi - \frac{M_d}{T_E}; \\ \dot{\psi} &= \frac{(2M_k - \psi)}{T_Э} - (\omega_c - P\dot{\phi}_d) M_d. \end{aligned} \qquad (12)$$

Here $T_E = (\omega_c S_k)^{-1}$ is the electromagnetic time constant of the engine, s; ω_c is the circu-lar frequency of the network supplying the electric motor, s^{-1}; P is the number of pairs of poles; S, S_k are the slip of the rotor of the en-gine and its critical value, respectively; $\psi = S_k \frac{(M_D + T_E \frac{dM_D}{dt})}{S}$ is the auxiliary variable, N·m.

Next, we determine the passport parame-ters and coefficients of the asynchronous mo-tor, for the purpose of unifying production, we accept 4A200M8U3 [7]: N=18.5 kW is the rated power of the motor; n=735 rpm is the rated number of revolutions of the motor rotor; M_N=240 N·m is the rated torque on the shaft of the motor rotor; $M_K = M_N \cdot 2.7 = 648$ is the critical moment on the shaft of the motor rotor; $M_П = M_N \cdot 2 = 480$ N·m is the starting torque on the shaft of the motor rotor; $\Im_p$=0.41 kg·m^2 is the dynamic moment of in-ertia of the electric motor rotor; f_c = 50 Hz is the network frequency; $\omega_c = 2 \cdot \pi \cdot f_c = 314.15$ s^{-1} is the circular frequency of the network supplying the electric motor; η=0.9 is the engine efficiency; cosφ=0.76 is the rated motor power factor; ω_o=78.53982 s^{-1} is the synchronous frequency of rotation of the motor rotor; ω_n=76.96902 s^{-1} is the rated frequency of rotation of the motor rotor; $S_n=(\omega_o-\omega_n)/\omega_o$=0.02 is the rated motor slip value; S_K=0.07464086 is the critical value of motor slip; P=4 is the number of pairs of poles; $I_{n.f.}$= 41.1A is the rated phase current; $I_{п.f.}$= 6.4 A is the starting phase current.

The moments of inertia of the saw cylinder of the linter machine were determined by the acceleration method used to determine the moment of inertia of bodies of revolution.

The saw cylinder under study is mounted on bearings; therefore, the experiments were conducted directly on the linter machine. To do this, a thread was wound on a pulley, with weights G_1 and G_2 suspended at its end. These loads were lifted to a height of h. During two experiments with different loads, the fall times t_1 and t_2 were recorded using video recording and the accelerations W_1 and W_2 were determined. Then, the sought-for moment of inertia of the saw cylinder of the linter machine was determined from the following equation:

$$\Im=\left(G_1\left(1-\frac{W_1}{g}\right)-G_2\left(1-\frac{W_2}{g}\right)\right)\frac{r^2}{(W_1-W_2)} \quad (13)$$

where: G=m·g is the gravity force, N; m is the weight of the load, kg; g=9.81 m/s^2 is the free fall acceleration; r is the pulley radius, m; t is the time of lowering the load, s; h is the height of lowering the load, m.

Accelerations of falling weights were set in the following form

$$W_1=\frac{2h}{t_1^2}, \qquad W_2=\frac{2h}{t_2^2}. \quad (14)$$

The results of the experiments are presented in Tables 1 and 2.

The moments of inertia of the motor rotor with a half-coupling $\Im_d$=0.4373 kg·m^2 (Table 1) and the saw cylinder of the linter machine with a half-coupling $\Im_{ps}$=0.7033kg·m^2(Table 2) were obtained in the experiments.

To study the machine unit of the saw cylinder of the linter machine, the technological load acting on the rotating shaft of the saw cylinder $M=M_{cp}+M_0\sin(\pi\omega_{ps}t+\varphi_{ps0})$ was experimentally determined (here M_{cp}=207,8 N·m; M_0=19,41 N·m; $\omega_{ps}=\pi\cdot735/30$ rad/s; t – time; φ_{ps0} is the initial phase) and then, by calculation, the stiffness of the coupling c=9581 N·m/rad and the dissipation coefficient в=55.39 N·m·f/rad of the coupling were determined.

The implementation of the equations of motion of the saw cylinder of the machine unit of the linter machine (11) with the characteristic of the drive motor (12) made it possible to establish the pattern of change in the angular acceleration of the saw cylinder of the linter machine using A.E. Levin's characteristic (Fig. 2, 3) as a function of time.

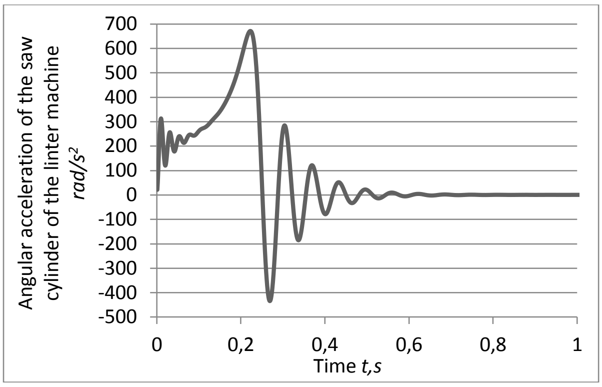


Fig. 2

The following parameters of the system were used: the technological load acting on the rotating saw cylinder of the linter machine $M_c=M_{cp}+M_0\cos(\pi\omega_{ps}t+\varphi_{ps0})$ (hereM_{cp}=207.8 N·m; M_0=19.41 N·m; $\omega_{ps}=\pi\cdot735/30$ rad/s; t – time; φ_{ps0} – initial phase); elastic-dissipative parameters (c=9581 H·m/rad andв=55.39 N·m·f/rad) of the coupling, moment of inertia of the electric motor ($\Im_d$=0.4373 kg·m^2) of saw cylinder($\Im_{ps}$=0.7033 kg·m^2).

The results of the analysis of Fig. 2 show that the maximum values of the angular acceleration of the saw cylinder of the linter machine are 675.05 rad/s^2 at t=0.223 s, and the transient process lasts for 0.8 s; the maximum value of the power consumption of the electric motor reaches up to 25.21 kW at t=0.227 s and the pattern of change in the angular acceleration of the saw cylinder of the linter machine (Fig. 2) can be determined by dividing into three parts in time: I - $t\in[0;0.1]$; II - $t\in[0.1;0.23]$ and III - $t\in[0.23;1.0]$.

Taking into account the pattern of change in the angular acceleration of the saw cylinder of the linter machine (Fig. 2), it can be expressed as the following function:

$$\begin{cases} \dfrac{\partial^2\phi_{ps}}{\partial t^2} = \ddot{\phi}_{ps}(t) = 190 + 660t - 210\, e^{-40t}\, \cos(285t) \\ \qquad \text{if } t\in[0-0.1] \\ \dfrac{\partial^2\phi_{ps}}{\partial t^2} = \ddot{\phi}_{ps}(t) = -155\,677\,919.5\cdot^{5} + 121\,644\,370.7t^4 - \\ \quad -37\,268\,055.1t^3 + 5\,619\,112.97t^2 - \\ \quad -416\,168.8t + 12\,357{,}1 \text{ if } t\in[0.1-0.23] \\ \dfrac{\partial^2\phi_{ps}}{\partial t^2} = \ddot{\phi}_{ps}(t) = 7000\, e^{-10.5t}\, \sin(88t) \text{ if } t\in[0.23-1.0] \end{cases} \tag{15}$$

Table 1

No. of repetition	Load mass m, kg	Gravity of loads G, N	Load lowering time t, s	Acceleration of falling loads W, m/s^2	Moment of inertia of an electric motor with a coupling $\Im_d$, kg·m^2
1	5	49.03	10.540	0.018	0.4331
	6	58.836	7.905	0.032	
2	6	58.836	7.905	0.032	0.4294
	7	68.642	6.588	0.046	
3	7	68.642	6.588	0.046	0.4493
	8	78.448	5.797	0.059	
Average					0.4373

Table 2

No. of repetition	Load mass m, kg	Force of gravity of loads G, N	Load lowering time t, s	Acceleration of falling loads W, m/s^2	Moment of inertia of saw cylinder with a coupling $\Im_{ps}$, kg·m^2
1	5	49.03	13.333	0.0112	0.6959
	6	58.836	10.000	0.02	
2	6	58.836	10.000	0.02	0.6906
	7	68.642	8.333	0.0288	
3	7	68.642	8.333	0.0288	0.7233
	8	78.448	7.333	0.0372	
Average					0.7033

II. Saw Cylinder Subsystem of a Linter Machine with Distributed Parameters

Consider a subsystem with distributed parameters [2–4, 6]. Let us designate an elementary section with length dx on the saw cylinder of the linter machine 2 (Fig. 1b), then the moment of inertia of the section is $\rho = \frac{\partial \Im}{\partial x} dx$. In this case, $\Im$ is a variable reduced moment of inertia non-uniformly distributed along the x–axis, then $\rho=\rho(x,t)$; for $\Im$=const we have $\rho=\rho(x)$; under a uniform distribution of masses $\rho=\Im/l=0.7033$ kg·m^2/2.15m=0.327 kg·m^2=const, where l is the length of the saw cylinder of the linter machine.

Using the theorems of changing the angular momentum, we determine the derivative of the angular momentum with respect to time

$$\frac{\partial}{\partial t}\left(\rho \frac{\partial \phi_{ps}}{dt}\right) dx = -M + (M + dM) + Q dx \,. \tag{16}$$

where $dM = \frac{\partial M}{\partial x} dx$ is the increment of the moment M in section dx.

Elementary angular deformation $d\varphi_{ps}$ is:

$$d\phi_{ps} = \frac{M}{G\ I(x)} dx \tag{17}$$

Where $G=8\cdot10^{10}$ N/m^2–is the shear modulus for a steel shaft; I(x) is the polar moment of inertia of the shaft, which in the general case can vary along the x-axis.

The moment is determined from dependence (17)

$$M = G\ I(x)\frac{d\phi_{ps}}{dx} \qquad (18)$$

hence its differential is

$$dM = G\frac{\partial}{\partial x}\left(I(x)\frac{\partial \phi_{ps}}{\partial x}\right)dx.$$

Substituting (18) into equation (16) and reducing by dx, we get

$$\frac{\partial}{\partial t}\left(\rho\frac{\partial \phi_{ps}}{\partial t}\right) - G\frac{\partial}{\partial x}\left(I(x)\frac{\partial \phi_{ps}}{\partial x}\right) = Q(x)\ . \qquad (19)$$

If ρ=const and I(x)=const=$7.42\cdot10^{-6}$ m^4, equation (19) has the following form

$$\rho\frac{\partial^2 \phi_{ps}}{\partial t^2} - G\ I\frac{\partial^2 \phi_{ps}}{\partial x^2} = Q(x) \qquad (20)$$

in this case, the generalized distributed force along the length x∈[0; l], applied to the saw cylinder of the linter machine has the following form

$$Q(x) = \frac{M_{cp} + M_0\ \cos(\pi\omega_{ps}\ t + \phi_{ps0})}{\pi R\ l}x\ . \qquad (21)$$

where M_{cp}=35.67 N·m; M_0=3.33 N·m; ω_{ps}=π·735/30 rad/s; t – time; φ_{ps0}=0 – is the initial phase; l=2.15 m is the saw cylinder length; R=0.1109 m is the radius in the center of the saw cylinder gasket of the linter machine. Then equation (20) for $\frac{\partial^2 \phi_{ps}}{\partial x^2} = \ddot{\phi}_{пшх}$ takes the following form:

$$\begin{cases} \ddot{\phi}_{psx} = \frac{1}{GI}\left(\begin{array}{l}\rho(190 + 660t - 210\ e^{-40t}\ \cos(285t)) - \\ \qquad -\frac{M_{cp} + M_0\ \cos(\pi\omega_{ps}\ t)}{\pi R\ l}x\end{array}\right) \\ \qquad\qquad \text{if } t\in[0-0.1] \\ \ddot{\phi}_{psx} = \frac{1}{GI}\left(\begin{array}{l}\rho(-155\ 677\ 919.5t^5\ +\ 121\ 644\ 370.72{\cdot}^{4}\ - \\ -\ 37\ 268\ 055.1t^3\ +\ 5\ 619\ 112.97t^2\ - \\ -416\ 168.82t\ +\ 12\ 357{,}15) - \\ \qquad -\frac{M_{cp} + M_0\ \cos(\pi\omega_{ps}\ t)}{\pi R\ l}x\end{array}\right) \\ \qquad\qquad \text{if } t\in[0.1-0.23] \\ \ddot{\phi}_{psx} = \frac{1}{GI}\left(\begin{array}{l}\rho(7000\ e^{-10.5t}\ \sin(88t)) - \\ -\frac{M_{cp} + M_0\ \cos(\pi\omega_{ps}\ t)}{\pi R\ l}x\end{array}\right)\ \text{if } t\in[0.23-1.0] \end{cases} \qquad (22)$$

The pattern of change in the angular velocity of the saw cylinder of the linter machine for x=0 is $\dot{\phi}_{psx} = 0,$ then C_1=0, C_2=0, C_3=0, and the equations have the following form

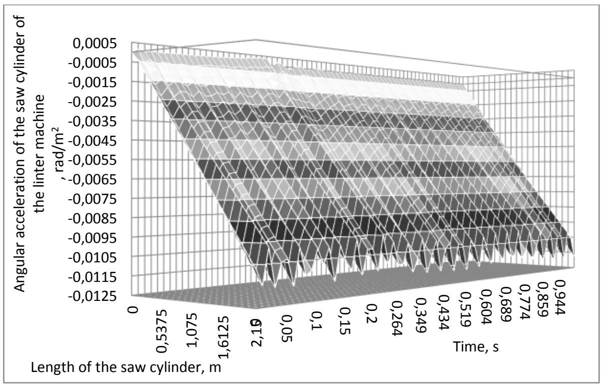
$$\begin{cases} \dot{\phi}_{psx} = \frac{1}{GI}\left(\begin{array}{l}\rho(190 + 660t - 210\ e^{-40t}\ \cos(285t))x - \\ \qquad -\frac{M_{cp} + M_0\ \cos(\pi\omega_{ps}\ t)}{2\pi R\ l}x^2\end{array}\right) \\ \qquad\qquad \text{if } t\in[0-0.1] \\ \dot{\phi}_{psx} = \frac{1}{GI}\left(\begin{array}{l}\rho(-155\ 677\ 919.5{\cdot}^{5}\ +\ 121\ 644\ 370.72{\cdot}^{4}\ - \\ -\ 37\ 268\ 055.1t^3\ +\ 5\ 619\ 112.97t^2\ - \\ -416\ 168.82t\ +\ 12\ 357{,}15)x - \\ \qquad -\frac{M_{cp} + M_0\ \cos(\pi\omega_{ps}\ t)}{2\pi R\ l}x^2\end{array}\right) \\ \qquad\qquad \text{if } t\in[0.1-0.23] \\ \dot{\phi}_{psx} = \frac{1}{GI}\left(\begin{array}{l}\rho(7000\ e^{-10.5t}\ \sin(88t))x - \\ -\frac{M_{cp} + M_0\ \cos(\pi\omega_{ps}\ t)}{2\pi R\ l}x^2\end{array}\right) \\ \qquad\qquad \text{if } t\in[0.23-1.0] \end{cases} \qquad (23)$$

The pattern of changing the angular rotation of the saw cylinder of the linter machine for x=0 is φ_{psx}=0, then C_4=0, C_5=0, C_6=0, and the equations have the following form

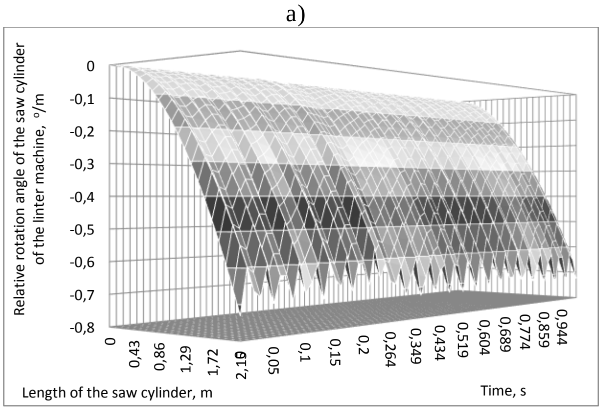
$$\begin{cases} \phi_{psx} = \frac{1}{GI}\left(\begin{array}{l}\rho(190 + 660t - 210\ e^{-40t}\ \cos(285t))\frac{x^2}{2} - \\ \qquad -\frac{M_{cp} + M_0\ \cos(\pi\omega_{ps}\ t)}{6\pi R\ l}x^3\end{array}\right) \\ \qquad\qquad \text{if } t\in[0-0.1] \\ \phi_{psx} = \frac{1}{GI}\left(\begin{array}{l}\rho(-155\ 677\ 919.5t^5\ +\ 121\ 644\ 370.7t^4\ - \\ -37\ 268\ 055.1t^3\ +\ 5\ 619\ 112.97t^2\ - \\ -416\ 168.82t\ +\ 12\ 357{,}15)\frac{x^2}{2} - \\ -\frac{M_{cp} + M_0\ \cos(\pi\omega_{ps}\ t)}{6\pi R\ l}x^3\end{array}\right) \\ \qquad\qquad \text{if } t\in[0.1-0.23] \\ \phi_{psx} = \frac{1}{GI}\left(\begin{array}{l}\rho(7000\ e^{-10.5t}\ \sin(88t))\frac{x^2}{2} - \\ -\frac{M_{cp} + M_0\ \cos(\pi\omega_{ps}\ t)}{6\pi R\ l}x^3\end{array}\right)\ \text{if } t\in[0.23-1.0] \end{cases} \qquad (24)$$

Results and discussions

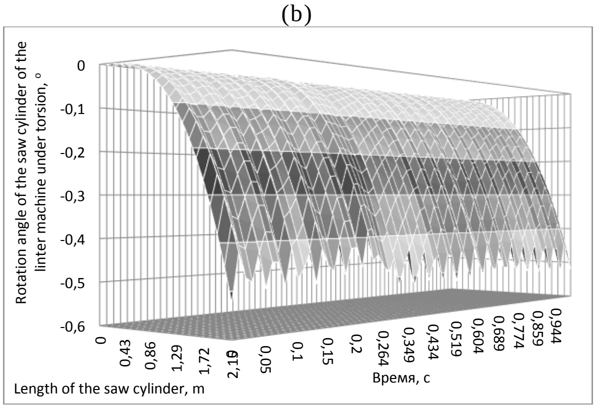
The solution to equations (22-24) made it possible to study the dynamics of torsional vibrations of the saw cylinder of a linter machine with distributed parameters (Figs. 3, 4).



a)



(b)



c)

Fig. 3

The graphs plotted (Fig. 3) made it possible to determine the maximum values of the angle of relative rotation and the angle of rotation of the saw cylinder of the linter machine under torsion; they are 0.729°/m and 0.523° (at*t*=0.519 s), respectively.

CONCLUSIONS

In general, the study of machines in the form of machine units made it possible to establish the dynamics of starting the electric motor and torsional vibrations of the saw cylinder of a linter machine with distributed parameters. For this, subsystems were used with concentrated parameters (Lagrange equations of the second kind) and distributed parameters (Laplace's equation in cylindrical coordinates).

The study of the machine unit of the saw cylinder of a linter machine with concentrated parameters according to the characteristic proposed by A.E. Levin showed that the critical driving moment of the electric motor is 340.92 N·m, the transient process lasts for 0.8 s, and the maximum value of the angular acceleration of the saw cylinder of the linter machine reaches 675.05 rad/s^2 at t=0.223 s.

The asynchronous electric motor 4A200M8U3with a power of 18.5 kW, a rotation speed of 735 rpm, and a rated torque of 240 N·m, installed on the rotor shaft leads to an increase in the starting torque according to the characteristic proposed by A.E. Levin by 340.92/240=1.42 times.

The result of calculations to determine the torsion of the saw cylinder of a linter machine using the characteristic proposed by A.E. Levin is 0.523°.

ЛИТЕРАТУРА

1. *Артоболевский И.И.* Теория механизмов и машин: учеб. для втузов. 4 изд., перераб. и доп. М.: Наука, Гл. ред. физ.-мат. лит., 1988. 640 с.

2. *Вульфсон И.И.* Динамика цикловых машин. СПб.: Политехника, 2013. 425 с.

3. *Вульфсон И.И.* К проблеме снижения виброактивности приводов цикловых машин при учете динамических характеристик электродвигателя // Проблемы машиностроения и надежности машин. 2017. №4. С.12–19.

4. *Пискунов Н.С.* Дифференциальное и интегральное исчисления: учебное пособие для втузов. Изд. 13. М.: Наука, Главная редакция физико-математической литературы, 1985. Т. 2. 560 с.

5. *Мухаммадиев Д.М.* Исследование неравномерности вращения пильного цилиндра джина 5ДП-156 при различных характеристиках асинхронного электродвигателя // Вестник КрасГАУ. 2008. №1. С. 236…241.

6. *Mukhammadiev D.M. Ibragimov F.Kh., Mukhammad T.D.* Modeling the Motion of a Saw Ginning Machine // Journal of Machinery Manufacture and Reliability. 2020, Vol. 49, No. 3, pp. 255–261.

7. *Кравчик А.Э. и др.* Асинхронные двигатели серии 4А. М.: Энергоиздат, 1982. 504 с.

8. *Левин А.Е.* Математическое моделирование приводов машин-орудий // Сб. науч. тр. по теории механизмов и машин. Алматы: Наука Казахстана, 1977. С. 259…260.

R E F E R E N C E S

1. *Artobolevsky I.I.* Theory of mechanisms and machines / Textbook for technical colleges. M.: Nauka. Phys.-Math. Lit. 1988. 640 p.

2. *Vulfson I.I.* Dynamics of cycle machines. St. Petersburg: Polytechnic. 2013. 425 p.

3. *Vulfson I.I.* On the problem of reducing the vibration activity of drives of cyclic machines considering the dynamic characteristics of the electric motor // Problems of mechanical engineering and reliability of machines. 2017. No. 4. P.12–19.

4. *Piskunov N.S.* Differential integral calculus: Textbook for higher educational institutions. M.: Nauka. Phys. and Math. Lit. 1985. V. 2. 560 p.

5. *Mukhammadiev D.M.* Study of uneven rotation of the saw cylinder gin 5DP-156 under different characteristics of an asynchronous electric motor // Bulletin of KrasGAU. 2008. No. 1. P.236–241.

6. *Mukhammadiev D.M., Ibragimov F.Kh., Mukhammadiev T.D.* Modeling the Motion of a Saw Ginning Machine // Journal of Machinery Manufacture and Reliability. 2020. Vol. 49. No. 3. pp. 255–261.

7. *Kravchik A.E. et al.* Series 4A asynchronous motors. M.: Energoizdat. 1982. 504 p.

8. *Levin A.E.* Mathematical modeling of machine tool drives // Scientific articles on the theory of mechanisms and machines. Almaty: Science of Kazakhstan. 1977. P. 259-260.

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