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Dynamic Analysis of Machine Aggregates with Compound Belt Transmissions

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Abstract: This article examines the dynamic characteristics of the PD-type feeding system in the 4DP-130 cotton cleaning machine when utilizing a composite belt drive. The study focuses on analyzing angular velocity oscillations and technological resistances by developing and solving a mathematical model using Mathcad software. Key parameters investigated include belt tension force (M_c), deformation of the composite pulley's elastic element (M_0), stiffness (c), and dissipation (b).
The findings indicate that the structural features of the composite belt drive contribute to partially absorbing dynamic loads, thereby reducing stresses on the shafts within the drive system. Additionally, an increase in the shafts' moments of inertia was found to prolong the machine assembly's startup time. Based on these results, the study provides recommendations on the interrelation of parameters such as M_c=100 Nm, M_0=4 Nm, c=20 Nm/rad, and b=8 Nms/rad, and their impact on the machine's dynamic performance.

Key words: belt transmissions, dynamic analysis, machine aggregates, feeder machine, fiber separation machine, moments of inertia, transmission ratios.

Introduction

The efficient operation of a working shaft in a technological process primarily depends on the dynamic parameters of the machine unit, including the moments of inertia of the masses, the dissipation properties of the elastic elements, and the influence of technological loads. To accurately determine these parameters, it is essential to study the dynamics of the machine unit with a belt transmission in both transient (static) and steady-state operating modes.

When conducting a dynamic analysis of a machine unit with a belt transmission, several key issues can be addressed:

- investigating external technological resistance: this involves examining how external factors-such as the weight force of the links, frictional forces, and the inertia force of the mechanism-affect the motion characteristics of both the driving and driven pulleys;
- reducing dynamic loads: identifying methods to minimize dynamic loads on the driving and driven pulleys while maintaining the required motion modes. This includes analyzing and regulating the relationships between motion regime parameters and the dynamic parameters of the machine unit under the influence of both external and internal forces.

Dynamic analysis of machine aggregates is crucial in studying the working mechanisms of technological machines, as it enables the assessment of how variations in external parameters affect the operation of the machine aggregate.

1. **Analysis of scientific works and literature**

A number of scientists, in their scientific research, do not consider dynamic factors based on static stresses and their extended models based on the accepted calculated models of reliability of belt drive machine units in failure or "stop" mode. In this case, dynamic equations for the system, taking into account dynamic loads and material properties, were proposed and studied depending on time. This takes into account the dynamic dependence of the machine unit during the "stopping" process and improper maintenance [1-3].

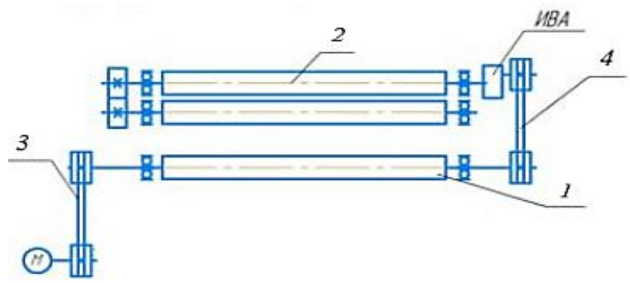
During operation in belt transmission, forces can be axial, transverse, and rotational. The analysis of forced and free oscillations occurring in a belt transmission was studied by comparing existing models. The influence of initial tension, the angular velocity of the shafts, bending stiffness, and reaction forces on the supports was discussed. Oscillations arising in belt drives were also considered, and the complex influence of factors was studied [2-7].

Material itself and the attenuation of vibration energy through the external tension system were taken into account. Using a closed system of equations of eigenvalues of the system, the dynamic problem of the hybrid model was solved. In this case, the numerical modeling method was used to account for the effectiveness of calculations. For each oscillatory mode, the optimal attenuation value has been determined. Analysis of the obtained results did not show that it is at this value that the oscillation is damped most rapidly [4,9,10].

In his research, M. I. Belov, based on experimental studies, studied the dependence of the values of the tractive forces arising in a belt drive consisting of driving and driven pulleys and a flat belt on the relative slippage of the belt on the pulley and the efficiency of the belt drive. A mathematical model has been developed that allows for a theoretical assessment of the dependence of the belt's traction capacity on the operating mode in the zone of elastic slippage of the belt along the pulley in the belt drive [5, 8, 10].

2. Main theoretical part

As an example of a machine aggregate with a belt transmission and composite pulleys, we consider the kinematic diagram of the 4DP-130 fiber separator machine, one of the key technological machines used in cotton processing plants. This machine features a PD-type composite feeder and serves as an illustrative model for the study. The kinematic diagram of the machine is presented in Figure 1 [11].



1 - spiked drum, 2 - feeding shaft, 3,4 - belt transmission

Fig. 1. - Kinematic diagram of the PD feeder machine unit of the 4DP-130 fiber separation machine

When the recommended composite driving pulley belt transmission is applied in the PD-type feeder of the 4DP-130 saw gin machine, this machine aggregate consists of a five-mass system (Figure 2).

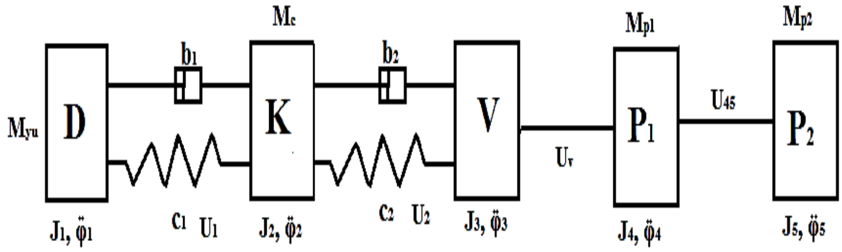


Fig. 2. - Dynamic model of the PD feeder machine unit of the 4DP-130 fiber separation machine with the recommended belt drive

In this case, the first mass represents the electric motor, the second mass corresponds to the spiked drum, the third mass denotes the IVA-type variator, while the fourth and fifth masses represent the PD-type feeder shafts. We assume that the specific elasticity of the belt remains constant in this scenario.

The external forces acting on the machine aggregate's mechanical system vary in nature and play different roles in the torsional-vibrational process. Initially, before the influence of external forces, the system is in equilibrium. However, when external forces are applied, they disturb this equilibrium, causing restoring forces to emerge as the system attempts to return to its original state. These restoring forces arise due to deviations from equilibrium and drive the system back toward its initial condition, leading to oscillations within the system.

Given that the pulleys in our case are composite (incorporating an elastic element), we assume that torsional-vibrational movements occur in the belt-driven machine. Here, restoring forces play a crucial role, acting as the primary factor in these oscillations. These forces originate from the elastic element of the pulley and exhibit an elastic nature.

In a linearly deformable system, the restoring force due to elasticity is proportional to the torsional deformation of the system. The connections between elastic elements are defined by a single parameter-the stiffness coefficient, which characterizes the rigidity of the elastic element within the pulley.

It should be noted that, in addition to restoring forces, frictional forces also arise within the elastic element. These frictional forces contribute to the dissipation (or dispersion) of mechanical energy, and are referred to as dissipative forces. In this context, the dissipative force in the elastic element of the compound pulley absorbs part of the system’s energy and reduces torsional vibrations in the shaft.

The characteristics of both restoring and dissipative forces depend entirely on the material properties of the elastic element within the belt drive pulley.

Furthermore, another category of forces, known as excitation forces, emerges during the operation of a belt-driven machine with composite pulleys. These forces are represented by explicit time-dependent functions, meaning they do not directly depend on the system's movement but actively influence it. For example, the unbalanced rotation of the driving shaft transmits a force to the conveyor belt, which serves as an excitation force.

As the mass of material transported by the conveyor belt periodically fluctuates, unbalanced rotation occurs in the driving shaft. In this scenario, external forces include the torque acting on the motor shaft and the loads exerted on the driving shaft, which are set into rotational motion through the belt drive.

I.S. Pinchuk, in his research on asynchronous motors, proposed treating these motors as dynamic mechanical systems [12,13].

$$\dot{M}_d = 2\dot{M}_k\omega_c - 2M_kP\dot{\varphi}_d - \omega_cS_kM_d \quad (1)$$

where M_d and M_k represent the torque of the electric motor and its critical value, respectively;

S_k is the critical slip in the electric motor; ω_c is the rotational frequency of the motor shaft.

The differential equations of motion for the belt-driven machine aggregate in the given system were derived using the second-order Lagrange equation [14].

$$\frac{d}{dt}\left[\frac{\partial T}{\partial \dot{\varphi}_i}\right] - \frac{\partial T}{\partial \varphi_i} + \frac{\partial \Pi}{\partial \varphi_i} + \frac{\partial \Phi}{\partial \dot{\varphi}_i} = M_i(\varphi_i) \quad (2)$$

where T – the kinetic energy of the system;

Π – the potential energy of the system;

Φ – Rayleigh's dissipative function;

φ_i – generalized coordinate;

$\dot{\varphi}_i$ – generalized velocity;

$M_i(\varphi_i)$ – generalized force.

The transmission ratio of the machine aggregate, including the drive mechanism of the feeding shafts, is determined as follows:

$$U_1 = \frac{\dot{\varphi}_1}{\dot{\varphi}_2};\ U_2 = \frac{D_2EF}{D_1(EF - S_0(e^{f(\beta\pm\Delta\beta)} - 1))}\sin\alpha;\ U_3 = U_v = \frac{\dot{\varphi}_3}{\dot{\varphi}_4};\ U_{45} = \frac{\dot{\varphi}_4}{\dot{\varphi}_5}; \quad (3)$$

where u_1, u_2, u_3, u_{45} – transmission ratios between the corresponding masses; $\dot{\varphi}_1$ – angular velocity of the electric motor shaft, $\dot{\varphi}_2$ – angular velocity of the spiked drum, $\dot{\varphi}_3$ – angular velocity of the IVA-type variator shaft, $\dot{\varphi}_4$ and $\dot{\varphi}_5$ – angular velocities of the feeder shafts; D_1 – diameter of the driving pulley in the belt transmission with a composite pulley, D_2 – diameter of the driven pulley in the belt transmission with a composite pulley; S_0 – initial tension force of the belt; F – cross-sectional area of the belt; E – modulus of elasticity; β – belt wrap angle around the pulley; $\Delta\beta$ – variable wrap angle of the composite driving pulley; f – coefficient of friction.

We accept the angular displacements of the rotating masses of the machine aggregate for generalized coordinates as φ_1, φ_2, φ_3, φ_4, and φ_5. The kinetic energy of the considered system has the following form:

$$T = \frac{J_1\dot{\varphi}_1^2}{2} + \frac{J_2\dot{\varphi}_2^2}{2} + \frac{J_3\dot{\varphi}_3^2}{2} + \frac{J_4\dot{\varphi}_4^2}{2} + \frac{J_5\dot{\varphi}_5^2}{2} \quad (4)$$

where J_1, J_2, J_3, J_4, J_5 – moments of inertia of the electric motor and working parts.

The potential energy of the system is a homogeneous quadratic form of generalized coordinates and is written as follows:

$$\Pi_1 = \frac{1}{2}[c_1 \cdot (\varphi_1 - U_1 \cdot \varphi_2)^2];\Pi_2 = \frac{1}{2}[c_2 \cdot (\varphi_2 - U_2 \cdot \varphi_3)^2] \quad (5)$$

For the given system, the Rayleigh dissipative function is expressed as follows:

$$\Phi_1 = \frac{1}{2}[b_1 \cdot (\dot{\varphi}_1 - U_1 \cdot \dot{\varphi}_2)^2];\ \Phi_2 = \frac{1}{2}[b_2 \cdot (\dot{\varphi}_2 - U_2 \cdot \dot{\varphi}_3)^2] \quad (6)$$

where c_1, c_2, b_1, and b_2 are the stiffness and dissipations of the belt drive.

We determine the conditions of the Lagrange equations:
a) derivatives of the kinetic energy of the machine aggregate

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}_1}\right) = J_1\ddot{\varphi}_1;\ \frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}_2}\right) = J_2\ddot{\varphi}_2;\ \frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}_3}\right) = J_3\ddot{\varphi}_3;\ \frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}_4}\right) = J_4\ddot{\varphi}_4;\ \frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}_5}\right) = J_5\ddot{\varphi}_5 \quad (7)$$

b) partial derivatives of displacements from the potential energy

$$\frac{\partial \Pi_1}{\partial \varphi_1} = c_1 \cdot (\varphi_1 - U_1\varphi_2);\ \frac{\partial \Pi_1}{\partial \varphi_2} = -c_1U_1 \cdot (\varphi_1 - U_1\varphi_2);$$
$$\frac{\partial \Pi_2}{\partial \varphi_2} = c_2 \cdot (\varphi_2 - U_2\varphi_3);\ \frac{\partial \Pi_2}{\partial \varphi_3} = -c_2U_2 \cdot (\varphi_2 - U_2\varphi_3) \quad (8)$$

c) partial derivatives of the dissipative function with respect to velocities

$$\frac{\partial \Phi_1}{\partial \dot{\varphi}_1} = b_1 \cdot (\dot{\varphi}_1 - U_1 \cdot \dot{\varphi}_2); \ \frac{\partial \Phi_1}{\partial \dot{\varphi}_2} = -b_1 U_1 \cdot (\dot{\varphi}_1 - U_1 \cdot \dot{\varphi}_2)$$
$$\frac{\partial \Phi_2}{\partial \dot{\varphi}_2} = b_2 \cdot (\dot{\varphi}_2 - U_2 \cdot \dot{\varphi}_3); \ \frac{\partial \Phi_2}{\partial \dot{\varphi}_3} = -b_2 U_2 \cdot (\dot{\varphi}_2 - U_2 \cdot \dot{\varphi}_3) \quad (9)$$

d) moment:

$$M(\varphi_1) = M_d; \ M(\varphi_2) = M_c; \ M(\varphi_3) = M_c + M_0 Sin(at);$$
$$M(\varphi_4) = M_{34} - M_{p1}; \ M(\varphi_5) = M_{45} - M_{p2} \quad (10)$$

where M_c – the tension force moment of the belt in the belt drive;

M_0 - the moments of the deformation forces of the flexible element of the composite pulley,

M_{34} and M_{45} are the fourth and fifth moments of a five-mass machine unit, representing the moments generated on the feeder shafts due to technological resistance.

For this machine aggregate, the mathematical model recommended by A.E. Levin [15] was used to determine the dynamic mechanical characteristics of the electric drive, based on the second-order Lagrange equation. The system of differential equations describing the motion of the machine aggregate is as follows:

$$\dot{M}_d = 2\dot{M}_k \omega_c - 2M_k P \dot{\varphi}_d - \omega_c S_k M_d;$$
$$J_1 \ddot{\varphi}_1 = M_d - c_1(\varphi_1 - u_1\varphi_2) - b_1(\dot{\varphi}_1 - u_1\dot{\varphi}_2);$$
$$J_2 \ddot{\varphi}_2 = u_1 c_1(\varphi_1 - u_1\varphi_2) + u_1 b_1(\dot{\varphi}_1 - u_1\dot{\varphi}_2) - c_2(\varphi_2 - u_2\varphi_3) - b_2(\dot{\varphi}_2 - u_2\dot{\varphi}_3) - M_c;$$
$$J_3 \ddot{\varphi}_3 = u_2 c_2(\varphi_2 - u_2\varphi_3) + u_2 b_2(\dot{\varphi}_2 - u_2\dot{\varphi}_3) - M_c + M_0 Sin(at);$$
$$J_4 \ddot{\varphi}_4 = u_3 M_c - M_{34} - M_{p1}; \ J_5 \ddot{\varphi}_5 = u_{45} M_{45} - M_{p2} \quad (11)$$

During the operation of a belt-driven machine aggregate with composite pulleys (incorporating an elastic element), the entire system is subjected to periodic loading. This influences the reduction of oscillations in the working shaft and causes variations in the transmission ratio of the belt drive due to the impact of external technological resistance (working force).

In our study, we analyze the dynamics of a belt-driven machine aggregate with composite pulleys under two key operating modes:

- the startup mode of the system;
- the steady-state operation of the system.

3. Main practical part

Particular attention is given to angular velocity oscillations and technological resistance in the machine aggregate's drive mechanism. The primary objective of this dynamic analysis is to determine the motion laws governing the system's components and the associated resistances while considering the acting forces. Next, let us examine the forces acting on the belt-driven machine aggregate with composite pulleys.

Investigation of the motion modes of the parameters of the 4DP-130 cotton cleaning machine drive system with a composite driving pulley belt transmission and the forces acting on the system using a computer. The dynamic problem of the proposed flat belt drive composite machine aggregate was implemented in the *"Mathcad"* software using modern computer technologies and a numerical method for solving the system of linear differential equations.

To solve the mathematical model of the flat belt drive composite machine aggregate in computer programs, we introduce some notations and define the following designations:

$$M_g(t) = y_1(t); \ \dot{M}_g(t) = \dot{y}_1(t);$$
$$\varphi_1(t) = y_2(t); \ \dot{\varphi}_1(t) = \dot{y}_2(t) = y_3(t); \ddot{\varphi}_1(t) = \dot{y}_3(t);$$
$$\varphi_2(t) = y_4(t); \ \dot{\varphi}_2(t) = \dot{y}_4(t) = y_5(t); \ \ddot{\varphi}_2(t) = \dot{y}_5(t);$$
$$\varphi_3(t) = y_6(t); \dot{\varphi}_3(t) = \dot{y}_6(t) = y_7(t); \ \ddot{\varphi}_3(t) = \dot{y}_7(t);$$
$$\varphi_4(t) = y_8(t); \ \dot{\varphi}_4(t) = \dot{y}_8(t) = y_9(t); \ \ddot{\varphi}_4(t) = \dot{y}_9(t);$$
$$\varphi_5(t) = y_{10}(t); \ \dot{\varphi}_5(t) = \dot{y}_{10}(t) = y_{11}(t); \ \ddot{\varphi}_5(t) = \dot{y}_{11}(t)$$

After determining the parameters, calculations for the equation of motion of the composite machine unit with a flat belt transmission were performed using the "Mathcad" software.

The selection of the electric motor was based on the research results provided in [11]. The movement of the working part of the flat belt drive composite machine aggregate mechanism is carried out using a 4A132S4Y3 asynchronous electric motor, which has the following parameters:

P=2,2 kVt- nominal power of the motor; n=1456,5 min^{-1} - nominal rotational speed of the motor shaft; $M_к=M_н\cdot 3=43,36$ Nm, $M_к$ – critical (maximum) torque of the motor shaft; $M_n=9550\cdot(P/n)=9550\cdot(2,2/1456,5)=14,46$ Nm, M_n – nominal torque of the motor shaft; $f_c=50$, Gs- frequency; P=2- number of pole pairs.

The calculation of the required parameters and coefficients of the motor was carried out using the following formulas [11]

System rotational frequency: $\omega_c=2\pi f_c=314$ rad/s. Nominal angular velocity of the electric motor shaft: $\omega_n=\pi n/30=3,14\cdot 1456,5/30=152,447$ rad/s. Angular velocity of the electric motor rotor in ideal no-load operation:

$\omega_o=2\pi f_c/P=2\cdot3{,}14\cdot50/2=157$ rad/s. Moment of inertia of the electric motor rotor: $J_1=0{,}028$ kg·m^2. Nominal slip value: $S_n=$ 2,9%. Critical slip value: $S_k=19{,}5\%$. Mine the moments of inertia of the shafts of the PD type feeding device of a belt-driven saw machine by calculating them, where:

spike drum: d = 400 mm; L = 4000 mm; thickness S = 4 mm; GOST 10704-91.

$$V = d \cdot L \cdot S = 0{,}4 \cdot 4 \cdot 0{,}004 = 0{,}0064\, m^3;$$

When the steel density is $\rho=7700 \div 7900$ kg/m^3,

$$m = \rho \cdot V = 7850 \cdot 0{,}0064 = 50{,}24\, kg$$

$$J = m \cdot r^2 = 50{,}24 \cdot (0{,}2)^2 = 2{,}0096\, kg \cdot m^2$$

- feeder drums: d=140mm; L=4000mm; S=3mm.

$$V = d \cdot L \cdot S = 0{,}14 \cdot 4 \cdot 0{,}003 = 0{,}00168\, m^3;$$

$$m = \rho \cdot V = 7850 \cdot 0{,}00168 = 13{,}188\, kg$$

$$J = m \cdot r^2 = 13{,}188 \cdot (0{,}07)^2 = 0{,}065\, kg \cdot m^2$$

IVA-type variator shaft: d=40mm; r=20mm=0,02m; L=0,5m; m=4,9325kg; F=12,566 sm^2 , GOST 2590-2006

$$J = \frac{1}{2} m \cdot r^2 = \frac{4{,}9325 \cdot (0{,}02)^2}{2} = 0{,}00099\, kg \cdot m^2$$

Moments of inertia of all shafts of the machine aggregate: J_1 = 0,028 kgm^2; J_2 = 2,0096 kgm^2; J_3 = 0,099 kgm^2; J_4 = 0,065 kgm^2; J_5 = 0,065 kgm^2

To find the relationship between the dynamic parameters of the machine aggregate and its motion modes, we apply the following variations to the values:

- rigidity coefficient of the flexible element: c=20÷40 Nm/rad
- dissipation coefficient of the flexible element: b=8÷16 Nms/rad

The range of variation of the coefficient values characterizing the flexible element was determined based on experimental results for different types of rubber [16].

When studying the dynamics of the PD feeder machine aggregate of the 4DP-130 fiber-separating machine with the recommended composite belt drive, the influence of the composite pulley parameters on the operation of the feeder rollers and the variator shaft was analyzed [7]. In this study, the transmission ratio between the electric motor and the pinned drum was taken as $U_1 = 2$, the average transmission ratio of the variators as $U_3 = 10$, and the transmission ratio between the feeder shafts as $U_{45} = 1$.

From the kinematic scheme of the PD-type feeder of the saw gin and the corresponding dynamic model of the aggregate, it can be observed that the proposed belt drive is installed between φ_2 and φ_3. If φ_4 and φ_5 are considered as the working shafts of the feeder, then the technological resistances from seed and cotton can be taken into account through M_{p1} and M_{p2}. The values of M_{p1} and M_{p2} depend on the loads exerted by seed and cotton.

It is known from the technological process that the feeder shafts rotate at the same angular velocity. Therefore, it can be assumed that their loads are also identical. The main issue of interest is the effect of the loads generated by the technological resistances in the feeder rollers on the angular accelerations of the variator shaft. In studying the dynamics of the considered machine aggregate, to account for the random technological resistances generated by seed due to cotton, we used the mathematical expression $M_c + M_0 \sin(\alpha t)$ in the calculations.

4.Results and discussions

In the analysis of the given dynamic model (Figure 2), the transition process of the machine aggregate under the influence of external loads is of interest, particularly the establishment of oscillations in the angular velocities of the working shaft masses. The main parameter in this case is the time spent on the transition process (Δt), which depends on the parameters of the machine aggregate. When the technological resistance changes, it affects the masses $M_d, \ddot{\varphi}_1, \ddot{\varphi}_2$,, and $\ddot{\varphi}_3$, causing a sharp decrease in their values. After a certain period, these oscillations subside, and this exact time Δt is considered as the time required for the velocity transition process.

In analyzing the data obtained from studying the dynamics of the flat belt drive composite machine aggregate, the variation and interrelation of the belt tension force (M_c), the deformation (M_0) and stiffness (c) of the flexible element of the composite pulley, as well as dissipation (b), are taken into account. The variated values are presented in Table 1 below.

Table 1. Data obtained from studying the dynamics of the flat belt drive

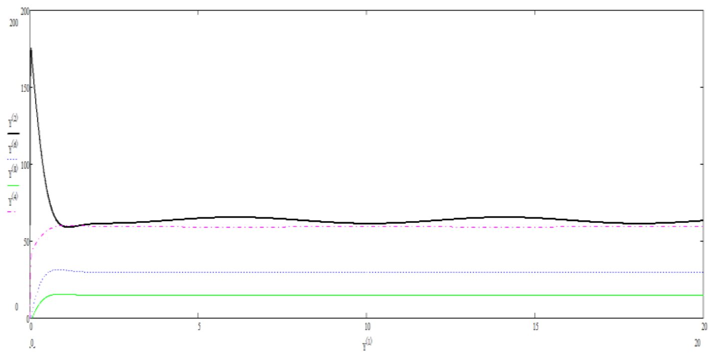
M_c (Nm)	80	100	120
M_0 (Nm)	4	8	12
c (Nm/rad)	20	30	40
α (grad)	π/2	π/4	π/6

In the belt drive system with a composite pulley containing a flexible element, the variation in belt tension (M_c) is considered as a factor affecting the load changes on the working shaft. It is known that changes in belt tension are primarily influenced by variations in the torsional moment on the shafts. The fluctuation in tension depends on the coefficient of friction generated between the belt and the pulleys. The belt tension force (M_c) formed during operation can be analyzed based on a harmonic (sinusoidal) law, reflecting the nature and magnitude of load variations.

The dynamic problem of the proposed flat belt drive composite machine aggregate was solved using a system of linear differential equations by employing numerical methods in the *"Mathcad"* software, utilizing modern computer technologies.

Based on the obtained solution, the oscillation range of the velocities and rotational irregularities for each shaft ($\dot{\varphi}_1, \dot{\varphi}_2, \dot{\varphi}_3, \dot{\varphi}_4, \dot{\varphi}_5$) were determined, along with the variation graphs of the angular velocities of the electric motor (M_d). In Figure 3, the belt drive system with a composite transmission is illustrated, showing the torque on the electric motor shaft and the variation graphs of the angular velocities of the shafts.

The operation process of machine units is divided into three stages: "Startup," "Operation," and "Shutdown." It is well known that the highest load occurs during the "Startup" phase, during which the torque (M_d) on the motor shaft reaches its maximum value. At this stage, the inertia moments of the shafts are the primary influencing factor, while technological loads must also be considered.



$Y^{(1)}$-time (s); $Y^{(2)}$-torgue on the shaft of the electric motor (Nm); $Y^{(4)}$, $Y^{(6)}$,$Y^{(8)}$-oscilliation rangers of the angular velocities of the first, second and third masses, respectively (*m/s²)*

Fig 3. - Graphs of the variation of the torsional moment on the electric motor shaft and the angular velocities of the shafts in the belt-driven transmission system

We analyze the theoretical research based on the variable values presented in Table 1, considering their interdependencies. The solution was obtained for all variation cases, and to study the results, the oscillation range of the driving motor shaft torque (M_d) and the angular accelerations of the shafts ($\ddot{\varphi}_1, \ddot{\varphi}_2, \ddot{\varphi}_3, \ddot{\varphi}_4, \ddot{\varphi}_5$) during the "Startup" process was examined. The calculated values are summarized in Table 2, where only the values for Mc = 80 are presented. Similarly, the calculated values were also obtained for Mc = 100 and Mc = 120.

Table 2.Oscillation range of the driving motor shaft torque (Md) and the angular accelerations of the shafts

ΔMd		C2=20	C2=30	C2=40
M₀=4	*p/2*	116,43	116,42	116,42
	Δt	1,115	1,15	1,115
	p/4	116,09	116,40	116,08
	Δt	1,13	1,115	1,13
	p/6	115,88	116,07	115,87
	Δt	1,13	1,13	1,125
M₀=8	*p/2*	117,47	117,48	117,49
	Δt	1,125	1,13	1,155
	p/4	116,81	116,82	116,83
	Δt	1,16	1,165	1,155
	p/6	116,37	116,38	116,38
	Δt	1,155	1,155	1,155
M₀=12	*p/2*	118,55	118,56	117,59
	Δt	1,12	1,12	1,16
	p/4	116,90	117,56	117,58
	Δt	1,15	1,16	1,16
	p/6	116,91	116,92	116,92
	Δt	1,15	1,15	1,15

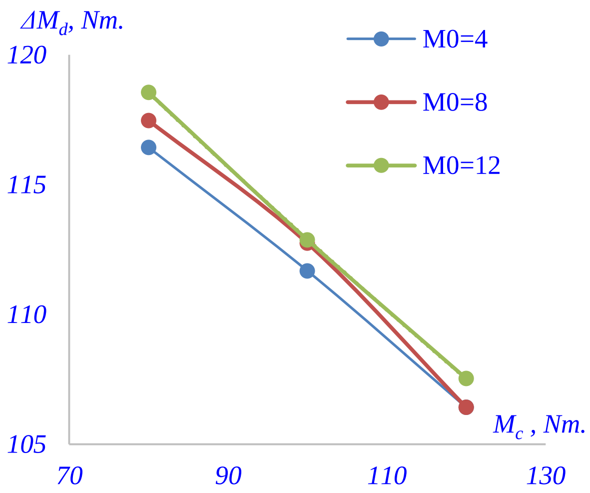
$\Delta\varphi_1$		C2=20	C2=30	C2=40
M₀=4	*p/2*	0,7967	0,7958	0,7958
	Δt	3,07	3,07	3,07
	p/4	0,7219	0,7262	0,7334
	Δt	6,06	6,06	6,06
	p/6	0,6983	0,6975	0,7075
	Δt	9,055	9,055	9,055
M₀=8	*p /2*	1,129	1,131	1,1305
	Δt	3,085	3,085	3,08
	p /4	1,0066	1,0075	1,009
	Δt	6,06	6,06	6,065
	p /6	0,934	0,936	0,935
	Δt	9,07	9,065	9,055
M₀=12	*p /2*	1,4655	1,4661	1,47
	Δt	3,08	3,08	3,08
	p /4	1,2852	1,283	1,2845
	Δt	6,06	6,06	6,06
	p /6	1,1752	1,176	1,176
	Δt	9,055	9,055	9,055

Continuation of table 2

$\Delta\varphi_2$		C2=20	C2=30	C2=40
$M_0=4$	*p/2*	1,9711	1,9694	1,9695
	Δt	2,685	2,695	2,685
	p/4	1,8836	1,8869	1,882
	Δt	5,63	5,63	5,63
	p/6	1,8525	1,8508	1,8574
	Δt	8,575	8,575	8,575
$M_0=8$	*p/2*	2,214	2,216	2,18
	Δt	2,76	2,76	2,76
	p/4	2,046	2,049	2,05
	Δt	5,64	5,64	5,635
	p/6	1,984	1,986	1,986
	Δt	8,57	8,57	8,57
$M_0=12$	*p/2*	2,4638	2,4653	2,4672
	Δt	2,775	2,775	2,775
	p/4	2,2132	2,2157	2,2173
	Δt	5,63	5,63	5,63
	p/6	2,1222	2,1239	2,1239
	Δt	8,575	8,575	8,575

$\Delta\varphi_3$		C2=20	C2=30	C2=40
$M_0=4$	*p/2*	1,0178	0,9748	0,9748
	Δt	2,455	2,45	2,45
	p/4	0,9609	0,9297	0,8996
	Δt	5,245	5,17	5,17
	p/6	0,9254	0,8734	0,8702
	Δt	8,09	7,97	7,97
$M_0=8$	*p/2*	1,0505	1,1627	1,1869
	Δt	2,515	2,515	2,525
	p/4	0,882	1,0219	1,0622
	Δt	5,085	5,17	5,24
	p/6	0,8084	0,9631	0,9631
	Δt	7,795	7,97	7,97
$M_0=12$	*p/2*	1,3502	1,3569	1,3599
	Δt	2,535	2,555	2,555
	p/4	1,0257	1,1445	1,1737
	Δt	5,09	5,17	5,24
	p/6	1,0561	1,0972	1,0972
	Δt	7,965	8,085	8,085

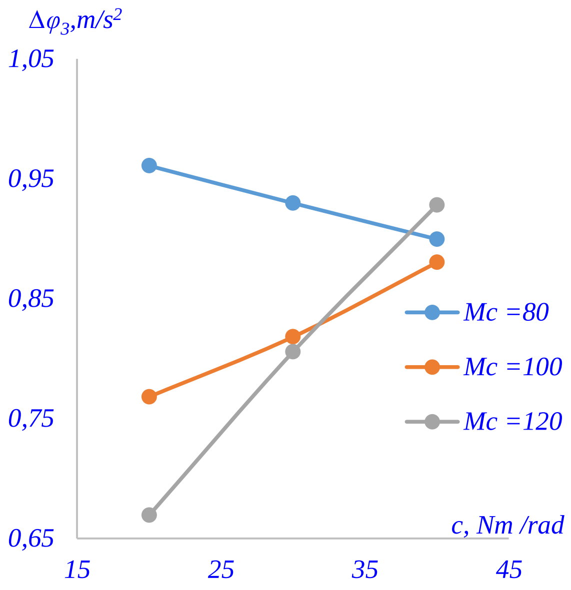
Analysis based on tabulated values shows that the designated startup time for motors ranges from 0,1 ÷ 0.15 seconds. However, considering additional rotating masses, including the parameters of the composite pulley belt drive, this value (Δt) increases to the range of 1,1 ÷ 1,2 seconds. Naturally, to reduce the startup time of the flat belt-driven machine unit, it is possible to decrease the moment of inertia of the rotating masses within the machine unit.



$M_0=4$ ΔMd	$M_0=8$ ΔMd	$M_0=12$ ΔMd
116,432	117,471	118,5524
111,6794	112,7448	112,8725
106,421	106,421	107,5332

Fig. 4. - Graph of the dependence of the engine torque on the belt tension force

The main factor of interest is the effect of the recommended belt drive parameter variations on the dynamics of the machine unit. Based on this, our further analysis is focused on this aspect. In this study, the variation in belt tension (voltage) (M_c) and the deformation of the elastic element due to load (M_0) were considered. Additionally, to account for the stiffness of the elastic element in the composite pulley, the stiffness coefficient (c) and the damping (dissipation) factor (b) were also analyzed (Figure 4). For example, when the value of (M_c) changes from 80 Nm to *120 Nm* and $M_0 = 4$, the difference between the peak and nominal values of torque ΔM_d during the "Startup" process decreases from *116,4 Nm* to *106,0 Nm*. However, the time required for the transition (Δt_1) increases from 1,11 seconds to 1,15 seconds.

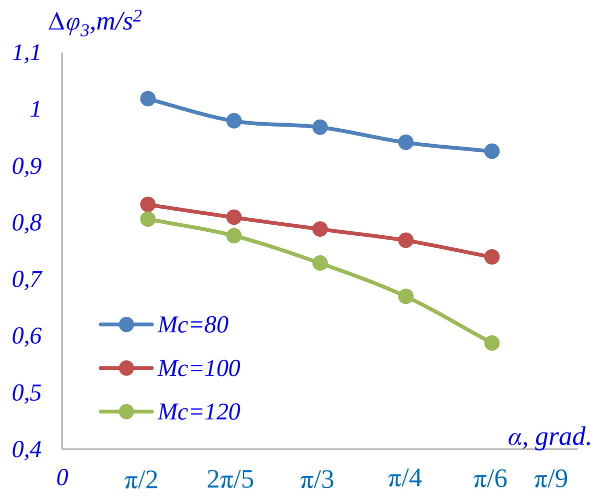


M_c =80	M_c =100	M_c=120
0,9609	0,7681	0,6692
0,9297	0,8181	0,8056
0,8996	0,8802	0,9281

Fig. 5. - The graph of the dependence of $\ddot{\varphi}_3$ on the stiffness coefficient of the elastic element when M_0 = 4 and p/4 is given

The relationship between the parameters of the elastic element, specifically the stiffness "c," was analyzed in the range of 20 Nm/rad to 40 Nm/rad depending on (M_c). The effect is most noticeable in ($\ddot{\varphi}_3$) (Figure 5). For example, when M_c = 80 Nm, the changes in ($\ddot{\varphi}_1$) and ($\ddot{\varphi}_2$) are insignificant, but the value of $\Delta\ddot{\varphi}_3$ decreases from 0,96 m/s² to 0,897 m/s². Conversely, when M_c=120 Nm, the value of $\Delta\ddot{\varphi}_3$ increases from 0,66 m/s² to 0,92 m/s².

If we analyze the variation of M_c based on the harmonic law (Figure 6), i.e., when α changes from π/2 to π/6, with M_c = 80 Nm and M_0 = 4 Nm, ($\Delta\ddot{\varphi}_3$) decreases from 1,01 to 0,92. Similarly, when M_c = 120 Nm and M_0 = 4 Nm, ($\Delta\ddot{\varphi}_3$) decreases from 0,80 to 0,58.



α	M_c=80	M_c =100	M_c =120
π/2	1,0178	0,8313	0,8058
2π/5	0,9789	0,8087	0,7764
π/3	0,9679	0,7881	0,7281
π/4	0,9409	0,7681	0,6692
π/6	0,9254	0,7387	0,587

Fig. 6. - The graph of the dependence of $\ddot{\varphi}_3$ on the harmonic variation of the load M_c.

Conclusion

By utilizing the constructive features of the recommended belt transmission, it is possible to partially absorb (dampen) the dynamic loads generated by the rotating masses within the machine aggregate. This, in turn, helps reduce the loads on the shafts within the drive system. The analysis of the research results indicates that an increase in the inertia moments of the shafts leads to an increase in the "Startup" time of the machine aggregate. In this case, reducing the equivalent inertia moment of the motor shaft has a stronger effect compared to reducing the inertia moments of the subsequent shafts.

Within the scope of studying the dynamics of the above-mentioned flat belt transmission machine aggregate, the conducted research and analysis of the results suggest that it is possible to recommend considering the variation and interrelation of the belt tension force (M_c*=100Nm*), the deformation of the compliant element of the composite pulley (M_0*=4*), stiffness (*c=20* Nm/rad), and dissipation (*b=8* Nms/rad) within the given range.

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