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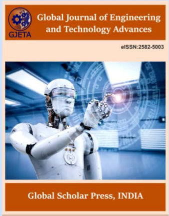

(Research Article)

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Numerical experiments of the action of power mechanisms and ways of determining the torque on the final actuators of mobile technological machines

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Abstract

A numerical experiment of angular displacement and angular velocity of power mechanisms is carried out on the example of potato harvesting machines. The ways of determining the torque on the final actuators such as hydraulic motors, propellers of machines and cardan gears of mobile technological machines using kinematics parameters are indicated.

Keywords: Motion Parameter; Angular Displacement; Angular Velocity; Angular Acceleration; Torque; Final Actuator; Power Parameter; Potato Harvester.

1. Introduction

The combination of modern means of mechanics, hydraulics, pneumatics and automation allows you to synchronize the speed and position of the working mechanisms so accurately that it is possible to implement high speeds in one machine, minimum time for technological operation and accuracy of performing a given instruction. Butthis, inturn, leads to increased requirements for the dynamics of the drive and the positioning accuracy of the actuators [1].

Mobile machines have relatively wide technological capabilities, which determine the scope of its application [2], being in its main purpose, a means of its high mobility and maneuverability during operation. And the theories of all such actions are not sufficiently modeled.

Currently defined parameters in the power mechanisms of the dismembered and separate form. A single and interrelated mathematical model was missing. Our research has found answers to such problematic questions [3]. On this scientific material, we will show one of the ways to solve this issue.

To conduct numerical experiments of angular displacement and angular velocities of power mechanisms of mobile technological machines, we take as an example the developed technological scheme of a potato harvesting machine. And in it we will find the angular displacement from the energy source to the final actuators.

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2. Material and methods

One of the promising directions of agricultural development is to increase production, improve quality and reduce the cost of potato cultivation [4,5].

Potato harvesters have a complex layout and consist of several separate units responsible for the processes of digging and separating the potato heap, transporting and unloading potatoes from the storage hopper [6]. Each individual block has its own specific requirements, and often, when considering each of the blocks separately, they meet these requirements. However, in the process of sequential inclusion in the work of all organs of the machine, there is a deviation of agrotechnical indicators from the required ones, which is most clearly manifested when working in conditions other than optimal.

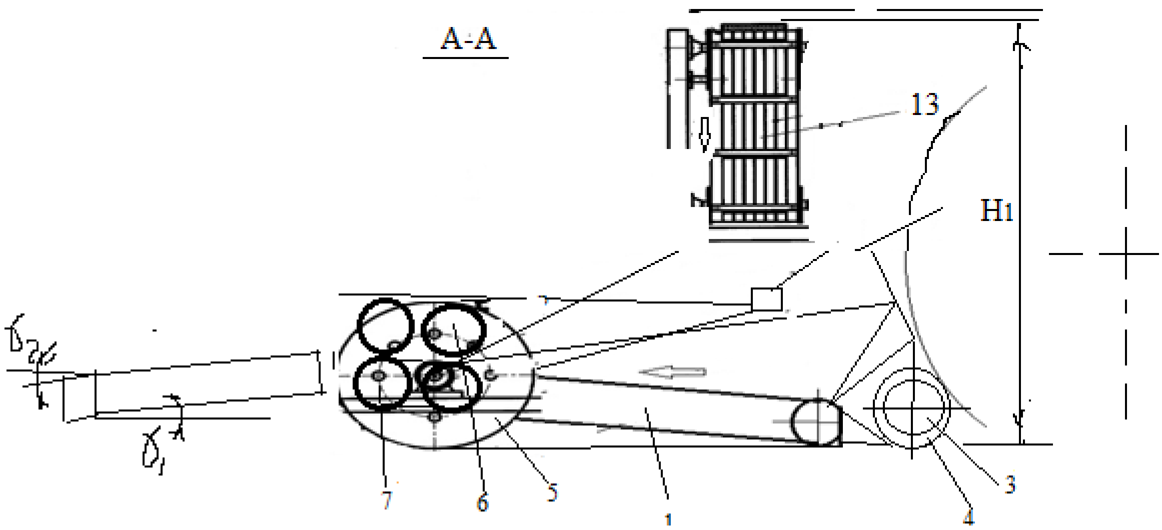
It is well known that potatoes are one of the main crop products consumed on the planet. Potatoes are cultivated in more than 100 countries around the world on an area of 19.3 million hectares, from which more than 376.5 million tons of tubers are harvested annually. The share of Russia accounts for approximately 11...14% of total production. The resulting potatoes are spent on food, livestock feed, technical purposes and seed stock [7].

The intensity of separation of soil lumps depends on the initial velocity of the potato heap components and their distribution over the surface of the bar elevator. To ensure the relative movement of the components of the potato heap along the grain elevator web, agitators are used [8].

To compile systems of equations, we use lagrangeequations of the second kind. The compiled systems of equations are solved by the Runge-Kutta method.

3. Results and discussion

In Fig. 1, we present a schematic diagram of a newly developed original potato harvesting machine, which performs a multi-time separation of heap impurities from the composition of potato tubers during harvesting.



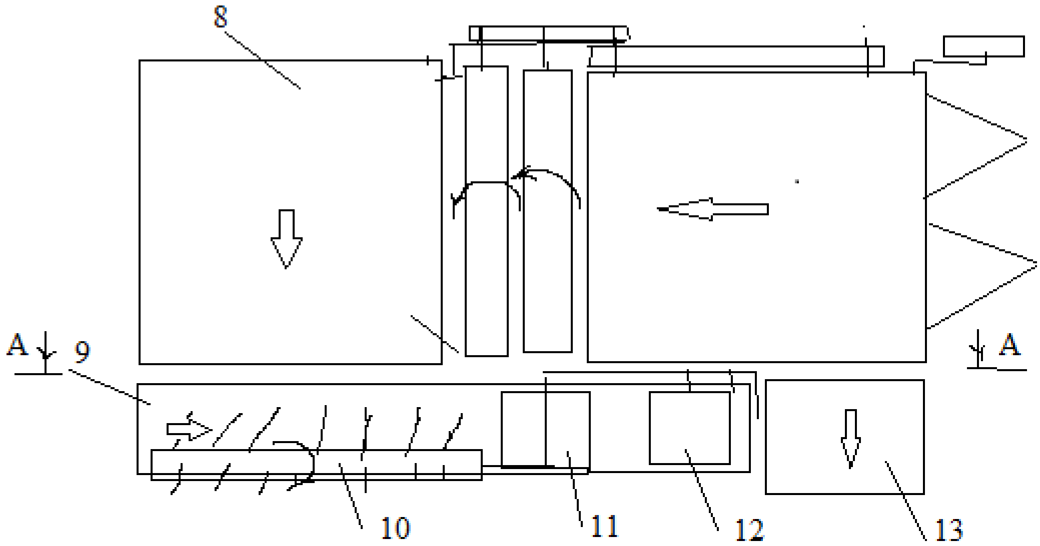


Figure 1 schematic diagram of the potato harvesting machine being developed

1– lifting Elevator-separator conveyor, 3 - pumping discs, 4 - right and left support rollers, 5 - wheels, 6 - pneumatic cylinder, 7 - arm conveyor, 8 –inclined Elevator-separator conveyor, 9 - Elevator-separator conveyor, 10 - topper remover, 11 – pneumatic (hydro) separation chamber, 12-automatic (or remote - controlled) mechanical) Elevator-separator, 13-lifting Elevator-separation conveyor.

The design is carried out that all the conveyor, Elevator-separator, conveyor is reinforced with a change in its overall size, have several reinforcement place to the frame. Length 2.5-3 m; height 2.5-3 m (withoutliftingconveyorheight) [9,10].

We turn to mathematical modeling and numerical experiment of the driving mechanisms of the potato harvester.

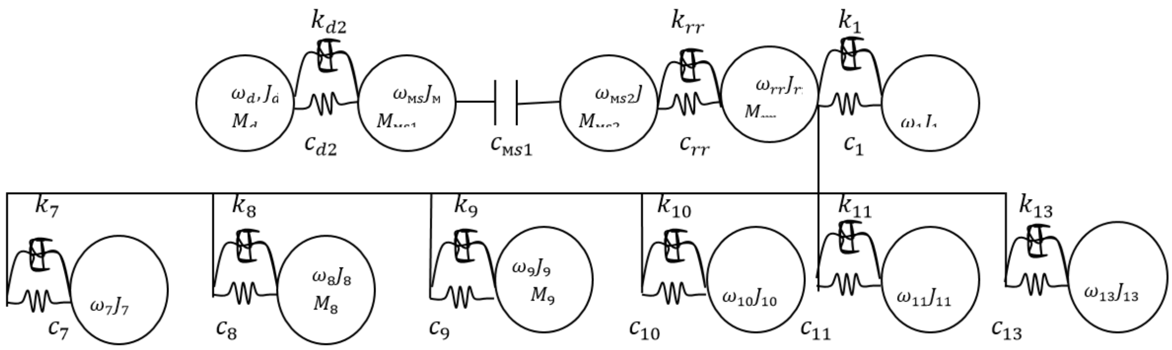


Figure 2 Equivalent design diagram of the drive mechanisms of the developed potato harvester

To facilitate modeling, we accept the following assumptions: the energy division in the transfer gearbox is considered to be evenly divided between seven mechanisms; we assume that the automatic Elevator-separator does not take energy from the transfer gearbox; we neglect the parameters of oscillation and stability, etc.

Equations describing rotationally moving mechanical mechanisms from an energy source (Fig. 2), for ease of understanding, the angular velocity ωis replaced by the angle φ.

$$(J_d + J_{MS1})\ddot{\varphi}_d + k_{d2}(\dot{\varphi}_d - \dot{\varphi}_{MS1}) + c_{d2}(\varphi_d + \varphi_{MS1}) = M_d - \frac{M_{MS1}}{i_2}, \varphi_d = \varphi_{MS1}$$

$(J_{MS1} + J_{MS2})\ddot{\varphi}_{MS2} + c_{MS1}(\varphi_d + \varphi_{MS2}) = -M_{MS2} sign(\dot{\varphi}_{MS2}) i_{MS2},$

$J_{rr}\ddot{\varphi}_{rr} + k_{34}(\dot{\varphi}_{мs2} - \dot{\varphi}_{rr}) + c_{rr}(\varphi_{мs2} + \varphi_{rr}) = -M_{rr}sign(\dot{\varphi}_{rr})i_{rr},$

$J_1\ddot{\varphi}_1 + k_1(1/7\ \dot{\varphi}_{rr} - \dot{\varphi}_1) + c_1(1/7\varphi_{rr} + \varphi_1) = -M_1 sign(\dot{\varphi}_1)i_1,$

$J_7\ddot{\varphi}_7 + k_7(1/7\ \dot{\varphi}_{rr} - \dot{\varphi}_7) + c_7(1/7\varphi_{rr} + \varphi_7) = -M_7 sign(\dot{\varphi}_7)i_7,$

$J_8\ddot{\varphi}_8 + k_8(1/7\ \dot{\varphi}_{rr} - \dot{\varphi}_8) + c_8(1/7\varphi_{rr} + \varphi_8) = -M_8 sign(\dot{\varphi}_8)i_8,(1)$

$J_9\ddot{\varphi}_{dm} + k_9(1/7\ \dot{\varphi}_{rr} - \dot{\varphi}_9) + c_9(1/7\varphi_{rr} + \varphi_9) = -M_9 sign(\dot{\varphi}_9)i_9,$

$J_{10}\ddot{\varphi}_{10} + k_{10}(1/7\ \dot{\varphi}_{rr} - \dot{\varphi}_{10}) + c_{10}(1/7\varphi_{rr} + \varphi_{10}) = -M_{10} sign(\dot{\varphi}_{10})i_{10},$

$J_{11}\ddot{\varphi}_{11} + k_{11}(1/7\ \dot{\varphi}_{rr} - \dot{\varphi}_{11}) + c_{11}(1/7\varphi_{rr} + \varphi_{11}) = -M_{11} sign(\dot{\varphi}_{11})i_{11},$

$J_{13}\ddot{\varphi}_{13} + k_{13}(1/7\ \dot{\varphi}_{rr} - \dot{\varphi}_{13}) + c_{13}(1/7\varphi_{rr} + \varphi_{13}) = -M_{13} sign(\dot{\varphi}_{13})i_{13},$

гдеJ_d, $J_{мs1}$, ω_d, $\omega_{мс1}$, M_d, $M_{мs1}$–moment of inertia, angular velocities and torques of the engine and the left side of the clutch;$J_{мs2}$,$\omega_{мс2}$, $M_{мs2}$– moment of inertia, angular velocity and torque of the right side of the clutch;J_{rr},ω_{rr},M_{rr}– moment of inertia, angular velocity and torque of the transfer gearbox;J_1,ω_1, M_1 – moment of inertia, angular velocity and torque of the separating mechanism;J_7,ω_7, M_7– moment of inertia, angular velocity and torque of the pillow conveyor;J_8,ω_8,M_8 – moment of inertia, angular velocity and torque of the inclined elevator-separator conveyor;J_9,ω_9,M_9– moment of inertia, angular velocity and torque of the elevator-separator conveyor;J_{10}, ω_{10}, M_{10}– moment of inertia, angular velocity and torque of the topper;J_{11},ω_{11},M_{11}– moment of inertia, angular velocity and torque of the pneumatic (hydro) separation chamber;J_{13},ω_{13},M_{13}– moment of inertia, angular velocity and torque of the lifting elevator-separation conveyor;k_iиc_i- coefficient of damping and stiffness of the corresponding drive and separation mechanisms, respectively.1/7– values that on the transfer gear movement is divided into seven mechanisms.

To study the performance of the written matmodel (1), we conduct a numerical experiment. In it, we conditionally accept all parameters as single numbers, within the range of 0.7-1.9, and the mass is 30-50. Using the Mepple program, we obtain numerical data and construct corresponding graphs of angular displacements of the structure to be torn apart (Fig. 3).

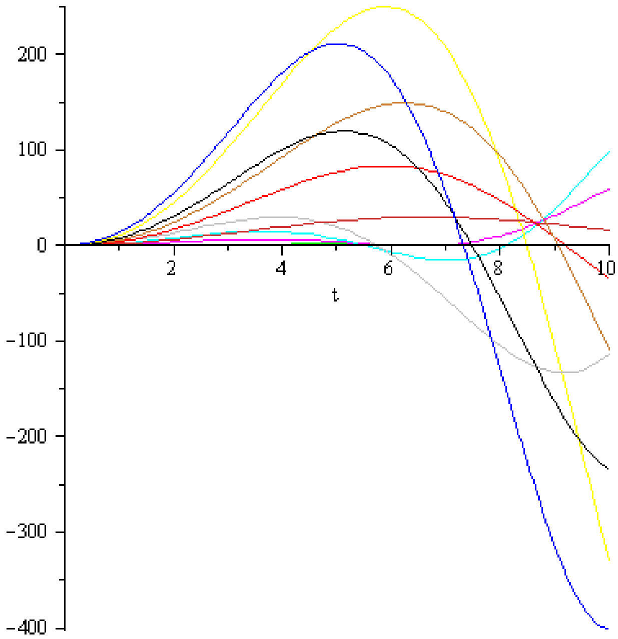


Figure 3 Angular displacement of power mechanisms when transmitting motion from an energy source through the main parts of a potato harvester

The color of the pointer we describe; φ_3-green: the angular displacement of the motor shaft; φ_4-angular displacement of the shaft left side clutch is equal $\varphi_4 = \varphi_3$;φ_5- orange: angular displacement of the right side of the shaft clutch; φ_{rr} –

grey: angular displacement of the shaft of the gearbox; φ_1–purple: angular displacement of the separating mechanism; φ_7–red: angular displacement poduzetnika conveyor; φ_8–gold: the angular displacement of the inclined elevator-separator conveyor; φ_9– black: angular displacement of the elevator-conveyor separator; φ_{10}–Cyan:angular movement of the topper; φ_{11} - yellow: angular movement of the pneumatic (hydro) separating chamber; φ_{13} -blue: angular movement of the lifting elevator-separating conveyor.

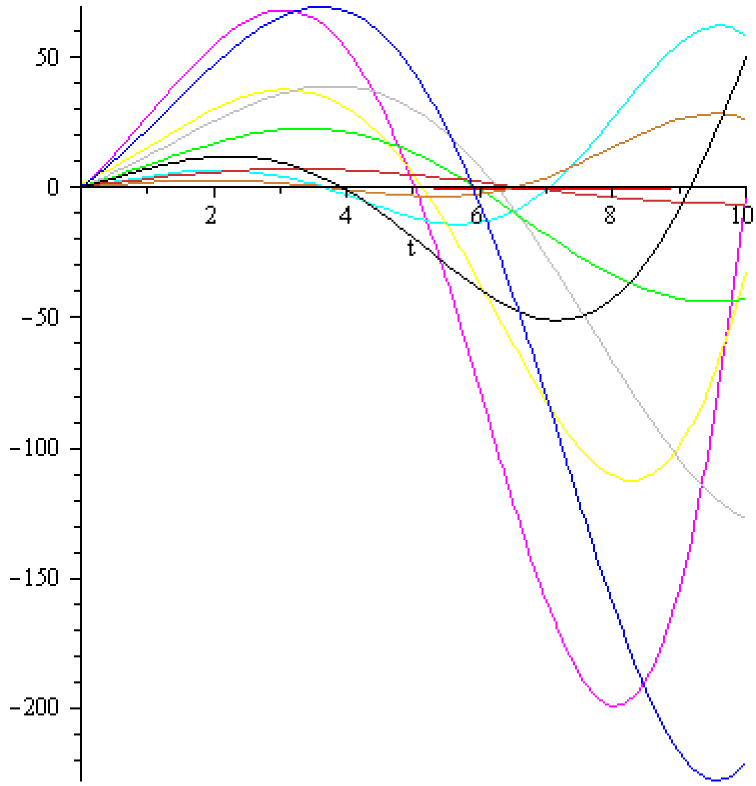


Figure 4 Angular velocities of power mechanisms when transmitting motion from an energy source through the main parts of a potato harvester

Color index will describe; φ_3-red:angular velocity of the motor shaft; φ_4- angular velocity of the shaft of the left part of the clutch, is equal to $\varphi_4 = \varphi_3$;φ_5- orange: angular velocity of the right part of the shaft of the clutch; φ_r –black:angular velocity of the shaft of the transfer gearbox; φ_1 –gold:angular velocity of the separating mechanism;φ_7 –green:angular velocity of the –separator conveyor; φ_8 –gray:the angular velocity of the inclined elevator-separator conveyor;φ_9 –yellow:the angular velocity of the elevator separation line;φ_{10} –cyan:the angular velocity of the defoliator;φ_{11} -blue:angular velocity of the pneumatic (hydro) separation chamber;φ_{13} -magenta:angular velocity of the lifting elevator-separation conveyor.

It can be seen that the developed model is workable, continuing the numerical experiment we can also find the acceleration parameters on the transition process.

We are looking for what moment is transformed on the final actuators during the transfer of energy from the energy source to the final mechanism. In our considered model we can find the final motion: angle of displacement, angular velocity and acceleration. Using these parameters of these movements it will be possible to find the torque of one of the final mechanism. Let's assume that the final actuator is a hydraulic motor. Having the ability to find the angular velocity $\varphi_0$and the working fluid pressure p_0, we can find the torque of the hydraulic motor shaft M_{4gm}.

Maximum load on the axis of the hydraulic motor

$$p_{4gm} \approx p_0 + J_{4gm}\dot{\varphi}_0^2(2p_0e)^{-1}; M_{4gm} \approx \left(V_{4gm}/2\pi\right)p_{4gm}$$

If the actuator is a mechanical body with it, the torque through the angular velocity is determined as follows. In this case, the acceleration created on the wheel axis $\ddot{\varphi}_5$ is considered as the final movement found from the previous transmitted movements. The moment of coupling of the movers M_φ with the ground is determined taking into account the vibrodynamic effect of disturbing loads and the variable speed of rotation of the wheel according to the formula

$$\mathrm{M}_\varphi = \left[\mathrm{mq}\varphi_\mathrm{p} + (1-\mathrm{m})(\mathrm{c} + \mathrm{q}\ \mathrm{tg}\varphi)\right]\mathrm{F}_0\mathrm{r}_\mathrm{k}\sqrt[\alpha]{\frac{\delta}{\delta_\mathrm{max}}}\sum_{\mathrm{i}=1}^{\mathrm{n}}\sqrt[\alpha]{\mathrm{i}}\exp[-\alpha_\tau(|\ddot{\varphi}_5 - \ddot{\varphi}_\mathrm{c}|\mathrm{r}_\mathrm{k})]$$

here, m is the coefficient of saturation of the tread pattern of the tyre; q – normal pressure of the tire on the ground; φ_p - walking angle; c – cohesion of soil; φ is the angle of internal friction of soil; F_0-contact area; r_k is the radius of the wheel; α is the exponent dependent on the ground; δ , δ_{max} - shear characteristics; α_τ - constant coefficient, characterizing the physico-mechanical properties of soils; $\ddot{\varphi}_5$ acceleration generated on the axis of the wheels; $\ddot{\varphi}_c$ is the acceleration of the fluctuations in the load on the ground;i - quantitatively, the lugs being in engagement.

Now consider the definitions of torque $M_в$, having to find the possibility of angular velocity and acceleration. We present the equations characterized by the rotation of the shaft of the technological machine

$J_в\ddot{\varphi}_в = \mathrm{M}_с - M_в sign(\dot{\varphi}_в)i_в$,

where$J_в$ - given the moment of inertia of the rotating mechanisms of the machine; $\dot{\varphi}_в$, $\ddot{\varphi}_в$ is angular velocity and acceleration of rotating machinery unit; $M_с\mathrm{M}_в$ –torque resistance Assembly and a rotating mechanism (for propeller shaft); $i_в$– ratio rotating mechanism (propeller shaft).

4. Conclusion

Thus, using the mathematical model created by us, we can find the motion parameters of final actuators, such as hydraulic motors, machine propellers and cardan gears. And with the use of these motion parameters: angular displacement, angular velocity and angular acceleration, we will determine the generated torques of the final actuators. Such a mathematical apparatus allows you to find the power parameters, mass ratio, overall dimensions of the main parts of ground vehicles during design. The final result is a promising resource-saving design of the technological machine.

Compliance with ethical standards

Disclosure of conflict of interest

All authors declare that they have no conflict of interest.

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