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Control efficiency evaluation of the optimal design of hitched system of cotton harvesters

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ABSTRACT: Based on the obtained equations of motion, the models and optimal control algorithm for the hitching system of cotton harvesting machine (CHM) MX-1.8 are developed in the paper. The necessary conditions for optimal control of the CHM MX-1.8 motion are investigated using the Mayer problem and applying the Pontryagin maximum principle. The operation parameters of the CHM MX-1.8 hitching system under vertical vibrations are determined. The optimal angle of rotation of the rockshaft which predetermines the uniform oscillation of the picking unit of the harvester hitching system is determined.

KEY WORDS: cotton picker, hitching mechanism, modeling, optimal control, shaft, diameter, fatigue strength.

I.INTRODUCTION

The control efficiency of cotton harvesters with attached arrangements is due to the technical features related to the issue of how fully they meet the established design requirements. For its maximum performance, the action of the control system should occur according to a certain pattern described by the control algorithm, which accounts for all the acting factors and all known dependencies. At that, one of the most important roles is played by the characteristics of the cotton harvester as an object of control.

The enumeration and exact mathematical description of these characteristics is a rather difficult task. However, from the point of view of creating effective, that is, optimal from a practical point of view, control systems, a complete and accurate solution to this problem is not required. It is enough to emphasize those characteristics that have the greatest impact on a specific type of control. However, it turns out that here it is necessary to take into account many relationships and interactions inherent in the operation of the cotton harvester.

As practice shows, when a cotton harvester with hitched systems moves, random uncontrollable disturbances cause natural oscillations. As a result, steady-state oscillations can be established in the system.

The system under consideration, like any other self-oscillating system, is nonlinear. The amplitudes, frequencies and modes of oscillations, as well as the existence of a stationary oscillatory mode, can be determined only from the solution of nonlinear equations, associated with great difficulties for a system with many degrees of freedom. At that, an important factor for assessing controllability is the response to a rapid change in the position of the control unit (i.e. the rocker shaft) of the hitched system of the cotton harvester

II. LITERATURE SURVEY

In the process of research and analysis of scientific publications, the specific features were established in the development of object-oriented mathematical models to assess their application in the operation of machine-tractor units and cotton harvesters under various driving conditions.

In [2], the technological process of a horizontal-spindle apparatus was considered, and experimental results obtained at test stations were given.

In [7], the results of field tests of a semi-hitched horizontal-spindle cotton-harvesting machine were presented. The possibility of its use with a zoned selected cotton variety with high quality indices of harvested cotton was established.

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The equations of motion of the machine unit of cotton harvesters and cotton stalks pullers were derived in [8], their analytical solutions were obtained.

In [9], well-known and new models of productivity of cotton harvesters were presented, as well as the results of calculations of the productivity of various tractors.

In [12], some aspects regarding the design and working process of the pressing wheel of agricultural machinery for planting and sowing were presented. In addition, theoretical studies of the working bodies of agricultural machinery were presented regarding resistance to rolling, the level of compaction and the assessment of the compaction depth caused by the pressing wheel. Finite element analysis was presented for the pressing wheel and the maintenance of its stability during the working process, for the normal nature of the wheel motion on the ground.

III. PROBLEM STATEMENT AND SOLUTION METHODS

Based on the analysis of the design scheme of a cotton harvester with a hitched system, a generalized mathematical model of vertical oscillation of cotton harvesting machine (CHM) MX-1.8 in the process of moving over the roughness on the headland of a cotton field was obtained in the form of the Lagrange equations of the second kind [3-14]:

$$\left.\begin{aligned}
& m_{\text{м}}\ddot{y}_{\text{м}} = F_y - b_1(\dot{y}_{\text{м}} - \dot{y}_{\kappa_1}) - c_1(y_{\text{м}} - y_{\kappa_1}) - b_2(\dot{y}_{\text{м}} - \dot{y}_{\kappa_2}) - c_2(y_{\text{м}} - y_{\kappa_2}) \\
& m_1\ddot{y}_{\kappa_1} = b_1(\dot{y}_{\text{м}} - \dot{y}_{\kappa_1}) + c_1(y_{\text{м}} - y_{\kappa_1}) - m_1\frac{2\pi^2 V_{\kappa_1}^2}{l_5^2} h_n\left(1-\cos\frac{2\pi V_{\kappa_1}}{l_5}t\right) \\
& (m_2-m_3)\ddot{y}_{\kappa_2} = b_2(\dot{y}_{\text{м}} - \dot{y}_{\kappa_2}) + c_2(y_{\text{м}} - y_{\kappa_2}) - (m_2-m_3)\frac{2\pi^2 V_{\kappa_2}^2}{l_5^2} h_n\left(1-\cos\frac{2\pi V_{\kappa_2}}{l_5}t\right) \\
& j_{\text{гц}}\ddot{\varphi}_{\text{гц}} = F_{\text{гц}}\cdot l_6 - b_3(\dot{\varphi}_{\text{гц}} - \dot{\varphi}_{\text{вк}}) - c_3(\varphi_{\text{гц}} - \varphi_{\text{вк}}) - l_7\cdot m_a\ddot{y}_{\text{м}} \\
& j_{\text{вк}}\ddot{\varphi}_{\text{вк}} = b_3(\dot{\varphi}_{\text{гц}} - \dot{\varphi}_{\text{вк}}) + c_3(\varphi_{\text{гц}} - \varphi_{\text{вк}}) - l_7\cdot m_a\ddot{y}_{\text{м}} \\
& m_{\text{гц}}\ddot{y}_{\text{гц}} = \frac{j_{\text{гц}}\ddot{\varphi}_{\text{гц}}}{l_7 - l_6} \\
& m_{\text{вк}}\ddot{y}_{\text{вк}} = \frac{j_{\text{вк}}\ddot{\varphi}_{\text{вк}}}{l_7}
\end{aligned}\right\}, \quad (1)$$

Where $\dot{y}_{\text{м}}$ *and* $\ddot{y}_{\text{м}}$– are the linear speed and acceleration of machine; $\dot{y}_{\kappa_1}$ and $\ddot{y}_{\kappa_1}$ - are the linear speed and front wheels acceleration; $\dot{y}_{\kappa_2}$ and $\ddot{y}_{\kappa_2}$ - are the linear speed and rear wheels acceleration; $\dot{\varphi}_{\text{гц}}$ *and* $\ddot{\varphi}_{\text{гц}}$-are the rotational speed and acceleration of torsional vibrations of the hydraulic cylinder lever; $\dot{\varphi}_{\text{вк}}$ *and* $\ddot{\varphi}_{\text{вк}}$- are the rotational speed and acceleration of torsional vibrations of the rockshaft lever; b_3 is the coefficient of viscous resistance of the rockshaft of the hitching mechanism of the harvesting unit; c_3 –is the stiffness coefficient of the rockshaft of the hitching mechanism of the harvesting unit; m_a is the distributed mass of the harvesting apparatus; $m_{\text{гц}}$ is the distributed mass along the hydraulic cylinder; $m_{\text{вк}}$is the distributed mass along the rockshaft; $F_{\text{гц}}$is the force in the hydraulic cylinder of the harvester hitching mechanism; l_1, l_2, l_3,l_4 and l_5 are the distances between supports and roughness; $l_6$is the lever length of the hydraulic cylinder; l_7 is the lever length of the hitching mechanism of the harvester; $j_{\text{гц}}$and $j_a$are the moments of inertia of the linking levers of hydraulic cylinder and the harvester hitching mechanism.

It is determined that the left and right harvesting devices oscillate unevenly under vertical vibration of the machine and the main reason for the uneven oscillation of the harvesting devices is the installation of a lever for connecting the hydraulic cylinder on the left edge, and not in the middle of the rocker shaft.

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In the problem to be solved, a necessary condition is an effective (optimal from a practical point of view) estimation of the angle of rotation of the rocker shaft, which can provide the least oscillations of the control system.

In general, the equation of motion of an object in a vector form is:

$$\dot{\bar{x}}(t) = A\bar{x}(t) + \bar{B}\,u + \bar{q} \tag{2}$$

Boundary conditions

$$\bar{x}(0) = 0 \tag{3}$$

Control conditions

$$0 \le F_{гц} \le 2225, t \in [o, T],\ T = 10.$$

Quality criterion

$$J\,(u) = x_8(T) \to min,$$

here $u=F_{гц}$, T is the set time of the end of rotation. It is required to choose the control $u(t)$ so that at time T the angular velocity of the hydraulic cylinder is zero, and the angle of rotation is maximal. The control problem was reduced to the Mayer problem.

The computational experiment was conducted for the following parameter values:
c_1=1254208.5 N/m; b_1=105634.2 Nf/m; c_2=637650 N/m; b_2=53705.3 Nf/m; c_3=263377.3 Nm/rad; b_3=22182.643 Nmf/m; $m_м$=7714 kg;m_1=5114 kg; m_2=2600 kg;m_3=1262 kg; m_a=675 kg; $j_{гц}$=552.96 Nmf^2; $j_{вк}$=276.48 Nmf^2; r_1=0.785 m; r_2=0.43 m; $h_п$=0.07 m; $h_ш$=0.03 m; $V_м$=1.21 m/s; $F_м = 18050\ N$, $F_y = F_м \sin\alpha = 18050 \sin 45^0 = 12763.277\ H$; $F_{гц} = 2225\ N$

The results of solving Mayer's problem showed that a change in the angle of rotation of the rocker shaft by 15.9 degrees predetermines the steady-state motion of the cotton harvester and uniform oscillation of the hitched system of the harvester, which shows the correctness of the developed model.

During the operation of the shafts, the load value and direction are constant or variable. The strength of the shafts is determined by the magnitude and nature of the voltages arising in them under the action of loads.
Shafts generally experience cyclically varying voltages. Hence, the main criterion for the appropriate performance of shafts and axles is fatigue strength. Static damage is very rare. It occurs under the influence of random short-term overloads. The calculation of fatigue resistance in shafts is considered basic. The calculation for static strength is performed as a verification.

The fatigue strength (endurance) of the shafts is estimated by the safety factor. The main design force factors for shafts are bending $M_и$ and torsional moments $M_к$. The influence of tensile and compressive forces is insignificant, therefore, they are not taken into account in the calculations.

The method for assessing the strength of shafts is the comparison of the calculated voltages with the allowable ones under the following strength conditions:

$$\sigma_и \le [\sigma]_и ; \tau_к \le [\tau]_к , \tag{4}$$

where $\sigma_и$, $\tau_к$ are the arising (calculated) bending and torsional voltages in the dangerous section of the shaft; $[\sigma]_и$ and $[\tau]_к$ — are the allowable bending and torsional voltages.

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Shafts calculated to ensure static or fatigue strength sometimes fail due to their insufficient rigidity or high vibrations. In addition, low rigidity interferes with the normal operation of gears and bearings. The shafts are additionally calculated on stiffness and vibrations.

To calculate the shafts for strength and stiffness, a design scheme is drawn up. When calculating for bending, rotating shafts and axles are considered as beams on pivot bearings. In the calculation diagrams, forces and rotating moments are conventionally taken as concentrated ones.

An approximate calculation is performed as a design one; the diameters of the characteristic sections of the shaft are tentatively established on its basis with the subsequent refinement of the safety factors for endurance.

IV. RESULTS OF COMPUTATIONAL EXPERIMENT

We calculate the CHM MX-1.8 rocker shaft for torsion with a tire deflection $h_ш = 0.04$ *m*. In this calculation, the diameter of the output end of the shaft or the diameter of the shaft under the bearing (under the support), which undergoes only torsion, is usually determined.

1. Determine the maximum shear voltage by static methods:

$$\tau_{вк} = \frac{M_{вк}}{W_p} = \frac{2 \cdot m_a l_7}{0.2 \cdot d_{вк}^3} = \frac{2 \cdot 675 \cdot 0.64}{0.2 \cdot (0.075)^3} = \frac{864}{0.2 \cdot (0.075)^3} = 1.024 \cdot 10^7 \ \frac{H}{м^2}.$$

2. Determine the maximum shear voltage based on the simulation results:

$$\tau_{вк} = \frac{M_{вк}}{W_p} = \frac{M_{вк}}{0.2 \cdot d_{вк}^3} = \frac{645.068}{0.2 \cdot (0.075)^3} = 0.7645 \cdot 10^7 \ \frac{H}{м^2}.;$$

3. Calculation of the diameter of the shafts based on shear voltages by static methods:

$$d_{вк} = \sqrt[3]{\frac{M_{вк}}{0.2\tau_{вк}}} = \sqrt[3]{\frac{864}{0.2 \cdot 1.024 \cdot 10^7}} = 0.077 \ м;$$

4. Shaft diameter calculation based on shear voltages using the simulation results:

$$d_{вк} = \sqrt[3]{\frac{M_{вк}}{0.2\tau_{вк}}} = \sqrt[3]{\frac{645.068}{0.2 \cdot 0.7645 \cdot 10^7}} = 0.077 \ м,$$

where$M_{вк}$ is the torque in the dangerous section of the shafts; c_3 is the shaft stiffness, c_3 = 263377.3 Nm/rad; τ_i and $[\tau]_к$ are calculated and allowable torsional voltages in the dangerous section of the shafts (for steels 45 and St5 $[\tau]_к$ = 25 ÷ 35 MPa = (25 ÷ 35)·10^{10} N/m^2 [84].

Now let us calculate the rocker shaft for the combined action of torsion and bending. The section of the shaft between the supports is calculated for the combined action of torsion and bending in equivalent moment $M_{экв}$.

The equivalent moment is calculated by the following formula (when calculating according to the theory of maximum shear voltages):

$$M_{экв} = \sqrt{M_и^2 + M_{вк}^2}$$

where $M_и$ and $M_{вк}$ are the bending and torque moments.

1. Calculation of bending moments by the static method:

$$M_и^I = 1.1 \cdot 0.5 \cdot P_{ок} b = 1.1 \cdot 0.5 \cdot \frac{M_{вк}}{l_6} b = 1.1 \cdot 0.5 \cdot \frac{864}{0.350} 0.463 = 628.62 \ Нм;$$

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$$M_u^{II}=1.1\cdot 0.5\cdot P_{ok}b=1.1\cdot 0.5\cdot \frac{M_{\text{вк}}}{l_6}b=1.1\cdot 0.5\cdot \frac{864}{0.350}0.463=628.62\ \textit{Нм};$$

2. Calculation of the shaft diameter of the rocker in bending moments by the static method:

$$\sigma_u^{I}=\frac{M_u^{I}}{0.1d_{\text{вк}}^3}=\frac{628.62}{0.1\cdot 0.077}=1.377\cdot 10^7\le[\sigma]_{\text{и}};$$

$$d_{\text{вк}}^{I}=\sqrt[3]{\frac{M_u^{I}}{0.1\sigma_u^{I}}}=\frac{628.62}{0.1\cdot 1.377\cdot 10^7}=0.079\ \textit{м};$$

$$\sigma_u^{II}=\frac{M_u^{II}}{0.1d_{\text{вк}}^3}=\frac{628.62}{0.1\cdot 0.077}=1.377\cdot 10^7\le[\sigma]_{\text{и}};$$

$$d_{\text{вк}}^{I}=\sqrt[3]{\frac{M_u^{II}}{0.1\sigma_u^{II}}}=\frac{628.62}{0.1\cdot 1.377\cdot 10^7}=0.079\ \textit{м};$$

3. Calculation of bending moments based on the simulation results:

$$M_u^{I}=1.1\cdot 0.5\cdot P_{ok}b=1.1\cdot 0.5\cdot \frac{M_{\text{вк}}}{l_6}b=1.1\cdot 0.5\cdot \frac{645.068}{0.350}0.463=469.333\ \textit{Нм};$$

$$M_u^{II}=1.1\cdot 0.5\cdot P_{ok}b=1.1\cdot 0.5\cdot \frac{M_{\text{вк}}}{l_6}b=1.1\cdot 0.5\cdot \frac{645.068}{0.350}0.463=469.333\ \textit{Нм};$$

4. Checking calculation of the rocker shaft diameters based on the simulation results:

$$\sigma_u^{I}=\frac{M_u^{I}}{0.1d_{\text{вк}}^3}=\frac{469.333}{0.1\cdot 0.077}=1.028\cdot 10^7\le[\sigma]_{\text{и}};$$

$$d_{\text{вк}}^{I}=\sqrt[3]{\frac{M_u^{I}}{0.1\sigma_u^{I}}}=\frac{469.333}{0.1\cdot 1.028\cdot 10^7}=0.079\ \textit{м};$$

$$\sigma_u^{II}=\frac{M_u^{II}}{0.1d_{\text{вк}}^3}=\frac{469.333}{0.1\cdot 0.077}=1.028\cdot 10^7\le[\sigma]_{\text{и}};$$

$$d_{\text{вк}}^{I}=\sqrt[3]{\frac{M_u^{II}}{0.1\sigma_u^{II}}}=\frac{469.333}{0.1\cdot 1.028\cdot 10^7}=0.079\ \textit{м};$$

A design scheme is built (Figure 1) using the drawing of the node.

In the scheme, $M_{\text{к}}$ is the torque arising in the cross-sections of the shaft; P_i are the forces acting on the shaft in the vertical plane.

After preliminary calculations and structural design of the shafts of shaped structures having a number of steps, holes, annular and keyway grooves, etc., in critical cases, a revised (verification) calculation of the shafts for fatigue strength (endurance) is performed.

The fatigue strength of the shaft is ensured if condition $K_p\ge[K_{\text{зп}}]$ is met, where K_p and $[K_{\text{зп}}]$ are the actual (calculated) and allowable safety factors for the dangerous section; (usually $[K_{\text{зп}}]$ = 1.5 ÷ 2.5; for gear shafts $[K_{\text{зп}}]$>(1.7 ÷ 3).

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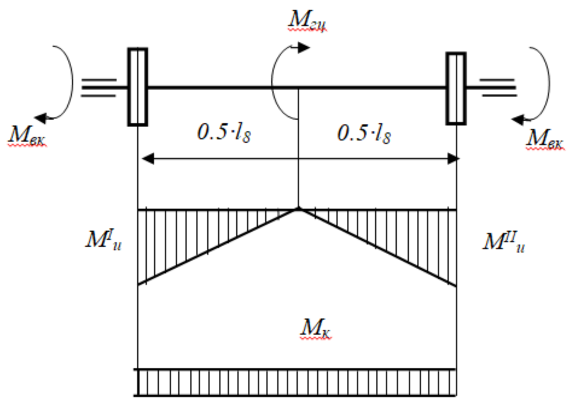
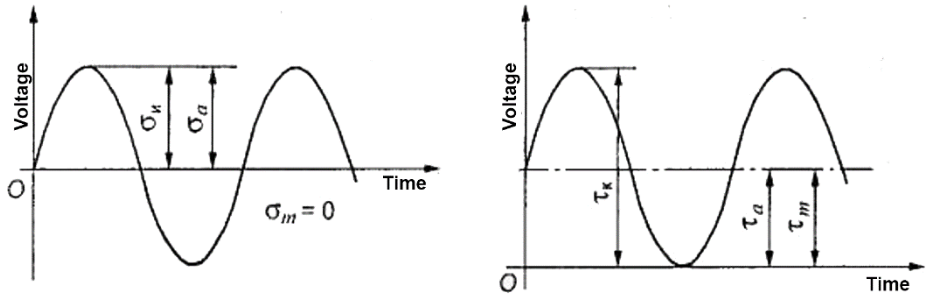


Fig 1. Diagram of bending and torque moments of the rocker shaft

When calculating the fatigue strength, it is necessary to establish the pattern of the cycle of voltage changes. In most cases, the actual load cycle of machines under operating conditions is difficult to establish. When calculating shafts for fatigue strength, it is assumed that bending voltages change in a symmetric cycle (Fig. 2, *a*), and torsional voltages change in a pulsating (from zero) cycle (Fig. 2, *b*).



a) *b)*

Fig 2. Cycles of voltage changes in the shaft sections:
a - symmetric cycle (bending voltage);
b - from zero cycle (torsional voltage)

For dangerous sections, the safety factors of fatigue resistance are determined and compared with the allowable ones. With the simultaneous action of bending and torsional voltages, the safety factor of fatigue resistance is determined by the following formula.

1. By static method:

$$K_p^I = \frac{K_\sigma K_\tau}{\sqrt{K_\sigma^2 + K_\tau^2}} = \frac{24.88 \cdot 17.6}{\sqrt{24.88^2 + 17.6^2}} = 14.368;$$

$$K_p^{II} = \frac{K_\sigma K_\tau}{\sqrt{K_\sigma^2 + K_\tau^2}} = \frac{24.88 \cdot 17.6}{\sqrt{24.88^2 + 17.6^2}} = 14.368;$$

2. Based on the simulation results:

$$K_p^I = \frac{K_\sigma K_\tau}{\sqrt{K_\sigma^2 + K_\tau^2}} = \frac{33.15 \cdot 23.68}{\sqrt{33.15^2 + 23.68^2}} = 19.268;$$

$$K_p^{II} = \frac{K_\sigma K_\tau}{\sqrt{K_\sigma^2 + K_\tau^2}} = \frac{33.15 \cdot 23.68}{\sqrt{33.15^2 + 23.68^2}} = 19.268;$$

where K_σ is the safety factor of fatigue resistance for normal bending voltages;

K_τ is the safety factor of fatigue resistance in shear torsional voltages.

V.CONCLUSION

Thus, the condition for fatigue strength in sections is satisfied. Analysis of the results shows that for $d_{вк}$= *0.079m* static and fatigue strengths are ensured. The initially obtained safety factors for fatigue strength in sections were found to be excessive. The calculations show that the diameter of the rocker shaft can be changed $d_{вк}$= *0,080 m*. This, in turn, will reduce the weight of the rocker shaft

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