Improving the processing accuracy of the puller shaft of the cotton harvesting machine controlling the dynamic system parameters

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Abstract. The article is dedicated to improving the accuracy of shaft turning by controlling the elastically deformed state of the part during processing. This study is conducted on the example of a puller shaft of a cotton harvesting machine, which belongs to the class of non-rigid parts such as bodies of rotation. To evaluate the possibilities of the method and establish theoretical patterns of the behavior of the part during transverse-longitudinal bending, the equation of the elastic line of a non-rigid shaft is solved. An integral assessment is adopted as a criterion of the accuracy in controlling elastic deformations of a technological system.

1 Introduction

Solving practical problems in engineering is closely related to the methods of mathematical modeling. To build a mathematical model of a technical object, it is necessary to transform the studied object into a calculation scheme.

The cotton harvesting machine used to collect the spindle of raw cotton consists of a device that includes a shaft which is 0.95 meters long and 0.026 meters in diameter. The necessary sections of removable brushes are installed on the shaft, which perform the function of removing raw cotton wound by the spindle for its further transportation to the hopper of the machine. The efficiency of the puller mainly depends on the quality of its parts, especially on the precision of the shaft manufacturing. This shaft belongs to the class of non-rigid parts. Turning is one of the main methods of puller shafts processing [1]. Shafts with $L / d \geq 20$ length-to-diameter ratio are considered to be non-rigid parts.

High demands are made on the accuracy parameters of geometric shapes and surface quality of non-rigid parts. The main method of processing is external longitudinal turning. At the same time, the manufacturing accuracy during turning operations should correspond to 8-11 accuracy standards with a roughness of Rα=0.63-2.5 μm. In some cases turning operations are the final type of processing [2].

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2 Methods of research

Mathematical models (MM) are of great importance in the processing technological system control of elastically deformated non-rigid shafts. They serve as the basis for optimizing processes and increasing the efficiency of this system.

The issue of choosing a specific type of model can be resolved only after a final constructive decision on the method and construction of the fastening model, parametric identification and analysis of the proximity of the results obtained on the simulation model with experimental data [3].

When controlling the processing technological system of elastically deformed non-rigid

When bending a non-rigid shaft in the transverse-longitudinal way, in the theory of elasticity and mechanics of materials, the method of constructing a description of an elastic line using a system of fourth-order differential equations is often used. For this description, deformation equations and equilibrium equations are used, which allows to establish a relationship between bending forces, moments and deformations of the part [3- 5].

3 Modeling of the technological turning system of the puller shaft and determination of the parameters of the elastically deformed state during longitudinal and transverse bending

To study the effect of the tensile force on the static stiffness of the described type of parts, the following mathematical model can be applied [6].

The part bedding with the application of a central tensile force can be carried out in collet mode. This method of bedding-in is interpreted as a rigid restraint with the possibility of axial movement (Figure 1). In order to minimize elastic deflection, it is also possible to control the angle of rotation of the part at the attachment point by applying an off-center tension [3, 7].

To evaluate the possibilities of the method and establish theoretical patterns of the behavior of the part during transverse-longitudinal bending, the equation of the elastic line of a non-rigid shaft is solved.

The description of the elastic line of a non-rigid shaft with transverse-longitudinal bending [8] can be represented in the form of fourth-order differential equations with constant coefficients

$$y_i^4 - k^2 y_i'' = 0 \,. \tag{1}$$

For a beam that is stretchable and carries an arbitrary transverse load, the solution of this equation gives the general equation of the elastic line

$$y = y_0 + y_0' kx + y_0''(1-\cos kx) + y_0'''(kx - \sin kx) + f(x)\,,$$

where $y_0 = \dfrac{P_y \cdot l^3}{48EJ}$, y_0', y_0'', y_0''' – is respectively the deflection, the angle of rotation, the second and the third derivatives at the origin; $k = \sqrt{P_{x1} / EJ}$; P_{x1} – tensile force, EJ – bending stiffness; *f(x)* – the function of the influence of transverse loads.

The equations of the elastic line in sections I and II (calculating scheme Figure 1) have the next form

$$\left.\begin{aligned} y_I &= y_0' kx + y_0''(1-\cos kx) + y_0'''(kx - \sin kx) \\ y_{II} &= y_0' kx + y_0''(1-\cos kx) + y_0'''(kx - \sin kx) + f(x) \end{aligned}\right\} \tag{2}$$

The initial parameters are determined by:

$$y_0' = -\frac{P_y}{kP_{x1}}\left\{\frac{[kl(\alpha-1+\cos\alpha kl)-\sin kl](1-\cos kl)}{kl\cdot\cos kl-\sin kl}-(1-\cos\alpha kl)\right\}+ \\ +\frac{M}{P_{x1}}\left[\frac{(kl\sin kl+\cos kl-1)\cdot(1-\cos kl)}{kl\cdot\cos kl-\sin kl}+\sin kl\right], \tag{3}$$

$$y_0''' = -\frac{P_y}{kP_{x1}}\left[\frac{kl(\alpha-1+\cos\alpha kl)-\sin\alpha kl}{kl\cdot\cos kl-\sin kl}\right]-\frac{M}{P_{x1}}\left[\frac{kl\sin kl+\cos kl-1}{kl\cdot\cos kl-\sin kl}\right], \tag{4}$$

where P_y – is the transverse load (cutting force); $\alpha=\frac{l-a}{l}$; l – is the length of the part; a – is the coordinate of the transverse load application.

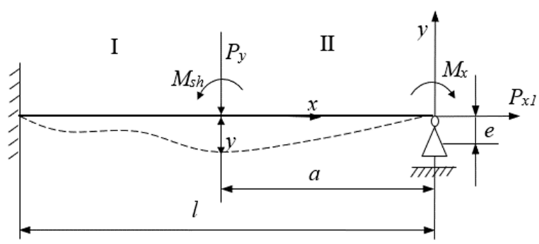


Fig. 1. Calculating scheme of loads and elastic line of the shaft under tension: e – eccentricity of tensile force application; M_{sh} – is the cutting forces moment; $M_x=P_{x1}\cdot e$ Nm – is the moment of the tensile force.

Given that a bending moment is applied to the support at the origin, it is necessary to determine the initial parameter y_0''. Differentiating (2), we find:

$$y_I'' = -y_0''\cdot k^2\cos kx - y_0'''\cdot k^2\sin kx\,. \tag{5}$$

After multiplying (5) by the bending stiffness EJ, the equation of the bending moment in the I section is obtained:

$$EJ\cdot y_I'' = -EJ(y_0''k^2\cos kx + y_0'''k^2\sin kx),$$

or taking into account $EJy_I'' = M_I(x)$; $EJ = P_{x1}/k^2$ we get:

$$M_I(x) = -\frac{P_{x1}}{k^2}(y_0''k^2\cos kx + y_0'''k^2\sin kx)\,. \tag{6}$$

If we consider that when $x=0$, $M_I(0)=M_x$, then from (6) follows $M_x = P_{x1}/k^2$ and accordingly

$$y_0'' = -\frac{M_x}{P_{x1}}\,. \tag{7}$$

Taking into account the function of the influence of the transverse load

$$f(x) = -\frac{P_y(x-a)}{P_{x1}} + \frac{P_y}{kP_{x1}}\sin k(x-a)\,, \tag{8}$$

finally, the deflection equations for the sections take the form

$$\left.\begin{aligned} y_I(x) = -\frac{P_y}{P_{x1}}\left[A(1-\cos kl)-(1-\cos\alpha kl)\right]x + \frac{M}{P_{x1}}[B(1-\cos kl)-\sin kl]kx - \\ -\frac{M}{P_{x1}}(1-\cos kx) + \left(\frac{P_y\cdot A}{P_{x1}} - \frac{M\cdot B}{P_{x1}}\right)(kx-\sin kx), \\ y_{II}(x) = y_I(x) - \frac{P_y}{P_{x1}}(x-a) + \frac{P_y}{k\cdot P_{x1}}\sin k(x-a). \end{aligned}\right\}, \quad (9)$$

where $A = \dfrac{kl(\alpha - 1 + \cos\alpha kl) - \sin\alpha kl}{kl\cos kl - \sin kl}$; $B = \dfrac{kl\sin kl + \cos kl - 1}{kl\cos kl - \sin kl}$.

Based on the study, a computational experiment was conducted, the main parameters were derived: α=0.5, P_y=431.351 N, P_{x1}=115229.35 N, k=4.63, l=0.6 m, EJ=5376 N·m^2, E=2.1·10^{10} N/ m^2, J=0.1d^4=256·10^{-6} m^4, d=0.04 m, $r_в$=0.02 m, M_x= P_{x1}·e=34.57 N·m, e=0.0003 m, x=0.15 m, a=0.3 m, $f(x)$=-0.0000425 m.

The results are presented in the Table and in Figure 2.

Table 1. Calculation results.

***T*, s**	y_0 **, μm**	y_I **, μm**	y_{II} **, μm**	y_{def} **, μm**
0	0	-97.22	32300	32300
0.1	0.1	0.018	-6.38	-6.38
0.2	0.2	0.0585	-7.71	-7.71
0.3	0.3	-0.06295	-3.67	-3.67
0.4	0.4	0.09134	-8.8	-8.8
0.5	0.5	0.00263	-5.853	-5.853
0.6	0.6	0.4888	-7.392	-7.392
0.7	0.7	0.0628	-7.856	-7.856
0.8	0.8	0.04888	-7.392	-7.392
0.9	0.9	-0.1133	-1.994	-1.994
1	1	-0.21978	1.5483	1.5483

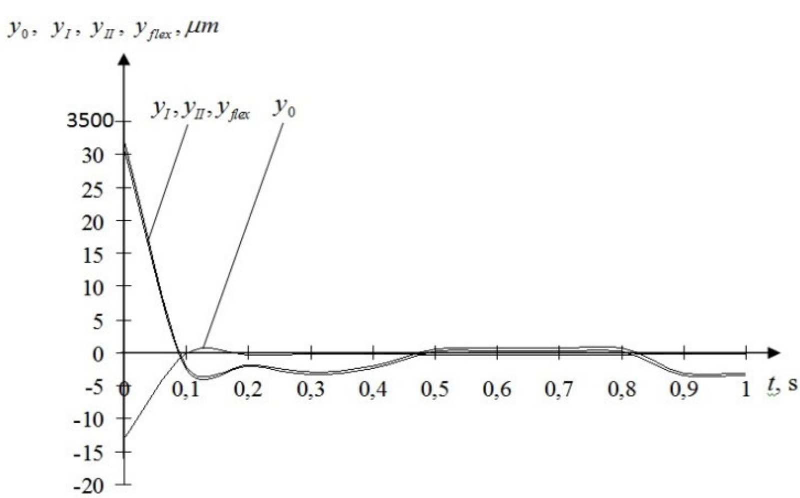


Fig. 2. The graph of the accuracy changes for the technological process of turning the puller shaft.

From the analysis of the developed mathematical models and analytically constructed dependences of changes in elastic deflections of parts when controlling technological systems in an elastically deformed state, using the application of tensile forces as well as transversive-longitudinal bending as regulatory influences, it follows that the magnitude of elastic deflections can be significantly reduced and practically stabilized along the entire length [9].

Thus, the solution of the general equation of an elastic line under the influence of tensile forces and bending moments is obtained. The values of tensile forces, bending moments and deflections of the shaft in sections during processing are determined, which allows to increase the accuracy of structural dimensions during turning of non-rigid shafts.

4 Integral quality criteria implemented by the control system

As a criterion characterizing the rate of attenuation of forced oscillations of elastic deformations and their deviation from the steady state, it is advisable to use an integral quadratic estimate [10-11]:

$$I = \int_0^\infty y^2(t)\,dt \,. \tag{10}$$

The criterion I is a function of the chip formation T_{sk} time constant and the cutting stiffness coefficient K_{sk}.

The integral estimate (10) can be accepted as a criterion of accuracy in controlling elastic deformations of the dynamic system, since the smaller the value of *I*, the smaller the oscillations $y(t)$ will be under the influence of a disturbing effect.

For the above calculations, transient processes in the control system were modeled, which are shown in Figure 3, a, where *Curve* 1 characterizes forced oscillations of elastic deformations for the values of the cutting mode parameters V = 60 m/min, t=1.5 mm, S=0.7 mm/rev. The analysis of the transient process indicates a fairly rapid attenuation of forced vibrations of elastic deformations, however, significant oscillations of this transient process may in some cases, for example, during finishing, be undesirable, since the roughness of the treated surface deteriorates.

This disadvantage can be eliminated by using an improved integrated assessment management system as a work accuracy criterion, the expression for which can be written as:

$$I_0 = \int_0^\infty \left[y^2(t) + \tau_1^2 y^2(t) \right] dt \,. \tag{11}$$

Considering the rate of change of elastic deformations in a dynamic system, taking into account weight, gives the machine control system qualitatively new properties.

The integral (11) can be represented as:

$$I_0 = \int_0^\infty \left[y(t) + \tau_1 y(t) \right]^2 dt - 2\tau_1 \int_0^\infty y(t)dy(t) \,, \tag{12}$$

since $y(\infty) = 0$, then

$$I_0 = \int_0^\infty \left[y(t) + \tau_1 y(t) \right]^2 dt + \tau_1 y^2(0) \,. \tag{13}$$

From the last expression follows that the minimum of the integral quadratic estimate is determined by the expression:

$$I_{0\min} = \tau_1 y^2(0) \,. \tag{14}$$

Given that $y(0) = K_{sk}K_y / (1 + K_{sk}K_y)$, it is possible to write:

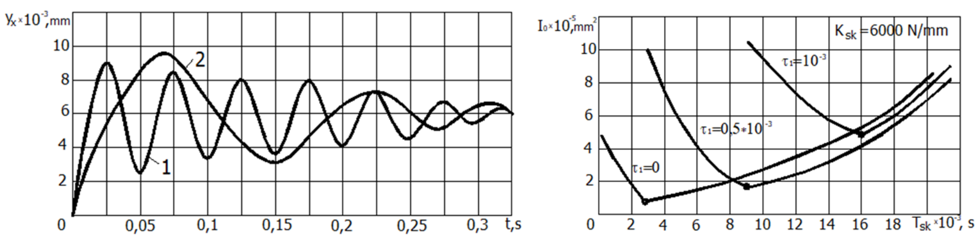
$$I_{0\min} = \tau_1 \left(\frac{K_y K_{sk}}{1 + K_y K_{sk}} \right)^2 = \tau_1 \left(\frac{K \cdot t}{C_y + K \cdot t} \right)^2 . \tag{15}$$

Expression (15) is of interest due to the fact that it allows, for known values of the rigidity of the elastic system of the machine tool C_y and the specific cutting force, to assign a cutting depth t at which the required precision of processing parts is ensured.

The numerical value of the weight coefficient τ_1 determines the system rapidity and ensures the smoothness behavior of transients.

5 Experimental results

The dependences of the criterion I_0 on the chip formation time constant T_{sk} with constant cutting stiffness $K_{sk} = 6000$ N/mm and various values of the weight coefficient τ_1 are shown in Figure 3, b. The analysis of the obtained dependencies shows that the desire to reduce the oscillation of transients in a dynamic system by increasing the weight coefficient τ_1 leads to a deterioration in the accuracy of parts processing, therefore, when choosing optimal cutting modes, a compromise solution to the problem that takes into account both the requirements for accuracy and quality of transients, is necessary.



a *b*

Fig. 3. Transients in the control system: curve 1 using integral quadratic estimation; curve 2 – improved integral quadratic estimation - a; criterion $I_0 = f(T_{sk})$ dependencies at different values of the weighting coefficient τ_1 – b.

In addition, it is possible to "smooth out" the oscillations of elastic deformations only at high values of the chip formation time constant [12]. This, in turn, leads to the fact that the cutting process must be carried out at sufficiently high values of longitudinal feed and low cutting speeds. Curve 2 in Figure 3, and the transient process of forced oscillations of elastic deformations in a dynamic system is obtained at $\tau_1 = 0,5 \cdot 10^{-3}$ and values of cutting mode parameters v = 20 m/min, t = 1.5 mm, = 0.9 mm/rev.

6 Software simulation of the turning process of the puller shaft of a cotton harvesting machine

A graphical model of the part and a simulation of its processing is formed in the Solid works software product (Figure 4).

When modeling a part and processing it, the following tasks were solved: the geometry of the model was created; physical and mechanical properties were determined; boundary conditions were set; a grid of finite elements was built; calculations of the shaft were carried out: for torsion, bending, tension and deviation from the cylindricity; interpretation of the results was performed.

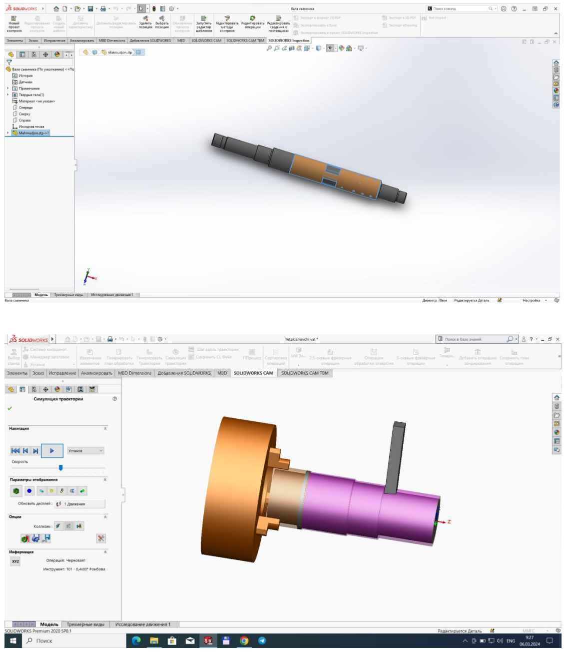
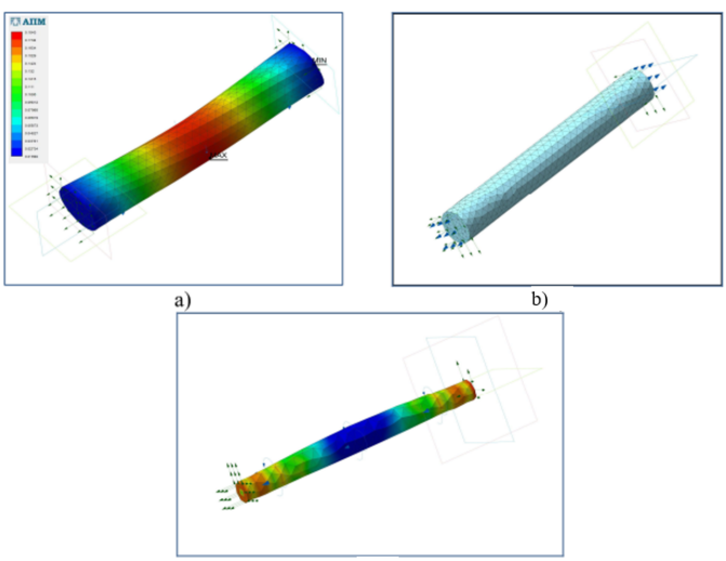


Fig. 4. A graphical 3D model of the part and a simulation of its processing.

The stress distribution along the modeling trajectory for various types of deformations is illustrated with contour lines in Figure 5, a,b,c and Figure 6 (bending strength analysis).



a) b)

c)

Fig. 5. Stress distribution: a − torsion, b– tension, c – deviation from the cylindrical.

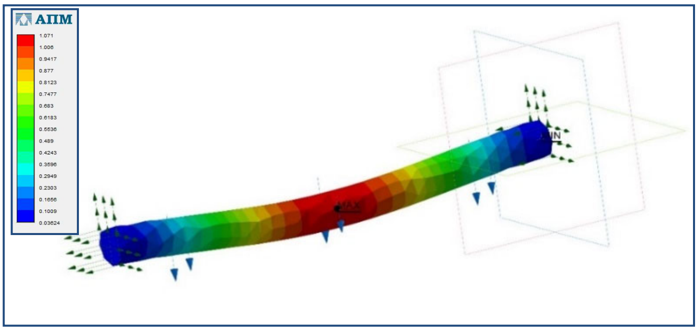


Fig. 6. Distribution of surface stresses when modeling the "shaft" part for tool positions at reference points.

The change in stress and deformation during the process is clearly presented. The deeper point, marked "MAX" in Figure 6, shows where the cutting tool interacts with the workpiece during modeling and includes elastic and plastic deformation together, while the non-integrated area, considered as "MIN", includes only plastic deformation. When the stresses reach their ultimate strength, stresses can form. Stresses due to elastic deformation will disappear from the workpiece after passing the tool and only residual stresses due to plastic deformation remain on the workpiece.

The model has been tested for adequacy using well-known formulas of resistance of materials for calculating the beam for strength and stiffness in oblique bending, torsion and transverse bending [13]. The results of the calculations show that the error does not exceed 3-5% for different tool positions and force application options. In this case, the maximum deformations are observed in the area closer to the center of the part [14], the point is determined by the pointer "Max". As a result of the conducted research, it was revealed that the maximum value of the amplitudes of x, y, z movements can be both at the point of application of force and outside it.

The analysis of elastic deformations from an external load showed that the maximum values are observed in the radial direction (see Figure 6), then – tangentially and the smallest – axially, with the largest values corresponding to the middle of the part.

7 Conclusion

An increase in the speed and quality of adaptive control systems can be ensured by using integral quadratic estimation and improved integral quadratic estimation as a criterion of accuracy in controlling elastic displacements and the elastically deformed state. The analysis of the transient curves obtained by numerical modeling in the control system indicates a fairly rapid attenuation of forced oscillations of elastic displacements in the technological system and high accuracy of parts processing.

The principles of MM construction and control systems obtained in this work can be adapted to other technological processes of turning elastically deformed parts, grinding, drilling, characterized by a wide range of variations in the paramete1rs of the dynamic system. Improving the processing accuracy and reliability of the turning process is achieved by constructing and using more accurate mathematical models, taking into account the features of phenomena occurring in the processing zone and elastic deflections in the technological system, as well as using the obtained MM in the development of optimal control algorithms.

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