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The Influence of Changes in Technological Loads on the Deflection of the Saw Cylinder Shaft of a Linting Machine

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Abstract. In the article the calculation of the deflection of the saw cylinder shaft of the linter machine is given taking into account the technological resistance revealed due to the mass of the seed roll. For theoretical calculation of the influence of raw material roll density (mass of raw material roll, machine productivity) on the process of deformation of the saw cylinder shaft, the bending calculation of the saw cylinder shaft without taking into account the masses of saws and gaskets, with taking into account saw discs and gaskets and with taking into account the raw material roll (in static position) is made. By reducing the bending of the saw cylinder shaft, the angular speed and the productivity of the process can be increased.

Keywords. Shaft bending, saw cylinder, seed roller, shaft deformation, linter machine, transverse force, bending moment, equations of equilibrium, support reaction.

Introduction.

Currently, there are more than 100 primary cotton processing plants in the country. The cotton plants' machinery and technology are gradually being upgraded. The main objective of modernisation of the plants is to increase the productivity of machines and to produce high-grade cotton fibre and lint that meets the requirements of the world market.

In the primary processing of cotton, the main process is the separation of the spinning fibre from the seeds, which produces its main products fibre and seeds. Along with the development of cotton cultivation, technology and techniques of cotton harvesting and processing there appeared the necessity to carry out other processes: preparation, storage, drying and cleaning of both raw material and its products. In addition, the range of products produced by cotton processing plants has expanded [1,2].

The history of the emergence of proginised seed processing processes is inextricably linked to the development of other sectors of the national economy. Until the second half of the 19th century, prodiginised seeds were used only as sowing material, which required only a small part of the produced seeds.

The cotton ginning industry produces three types of lint. The first type of lint contains fibres with a mass-length of 13/14 mm or more, the second type - from 7-8 to 12-13 mm, and the third type - 6-7 mm or less. In addition, the lint is also characterised by its grade, which is determined according to the grade of the seeds to be processed [3-5].

The process of linting of cotton seeds originated as a preparatory process necessary to maximise the compression of oil from the cotton seeds. The resulting lint had no industrial value. Seed linting was performed on lintering machines developed similar to saw gins in the United States of America [6,7].

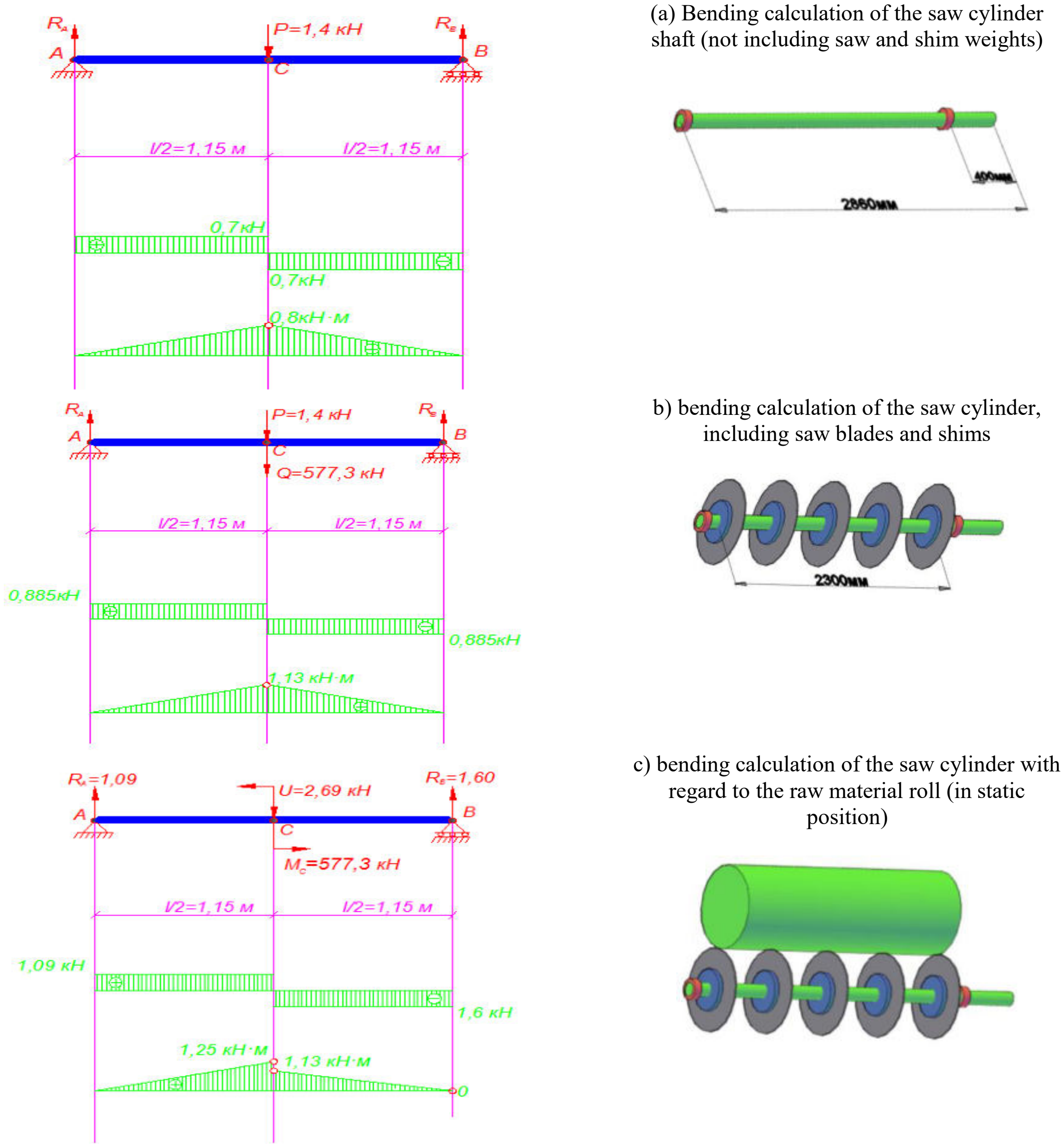
In the creation of machines for cotton ginning industry with high parameters can be realised only on the basis of deep knowledge of physical processes occurring in machines in different loading modes and development of new, more perfect methods of calculation of acting loads, which are the basis for calculation of machine parts and units for strength and endurance. Especially urgent are the issues of development and refinement of methods of calculation of drives and shafts of machines of large capacity or importance for production. This would make it possible to carry out production tests to refine the adopted parameters.

One such machine in the cotton processing industry is the linter machine. The lintering process is carried out on linter machines, where the main working organ is the saw cylinder. It is the drive of the saw cylinder of linter machines 18.5 kW that consumes a significant amount of power in the machine.

1. Research methodology

For theoretical calculation of the influence of raw roll density (raw roll mass, machine productivity) on the process of deformation of the saw cylinder shaft, a calculation is made, which consists of several steps, which are presented in Fig. 1.

Given that the saw cylinder shafts are made from a whole shaft and are more than 2 metres long, there is an inherent bending of the shaft. Let's consider this with the help of calculations. The following data are required for the calculation: shaft length l=2300 mm*;* gravity force P=m·g*,* where m-mass *of* the shaft. The theoretical weight of steel relative to the diameter is [8,9]. Ø61,8=23,56 kg/m, Ø100=61,65 kg/m. Taking into account the shaft length m_{sh} = t·l = 61,65·2,3 = 141,2 kg. These are the values in newtons P=m_{sh}·g = 141,2 · 9,8 = 1389,6 N. (Fig.1.a)



(a) Bending calculation of the saw cylinder shaft (not including saw and shim weights)

b) bending calculation of the saw cylinder, including saw blades and shims

c) bending calculation of the saw cylinder with regard to the raw material roll (in static position)

Fig. 1. - Saw cylinder shaft and calculation diagram

2. Results and discussion

Composing the equations of equilibrium for this system, we find the reaction forces on the bearings (supports). The reactions of the supports occur along the directions along which the rod (shaft) will not be able to move. Table 1 summarises the calculation data for all variants.

To plot the transverse forces Q_y and bending moments $M_{из}$ the method of sections is used. The transverse force in the section of the rod is equal to the sum of projections of external forces on the Y axis acting on the rest of the rod (shaft). The bending moment in the section of the rod (shaft) is equal to the sum of the moments of external forces acting on the remaining part of the rod (shaft), relative to the centre of gravity of the section. The obtained design data for all variants are summarised in Table 2.

Table 1. Reactions of the supports occur along the directions along which the rod (shaft)

$\sum M_{ia} = 0$	$\sum M_{ib} = 0$	Verification: $\sum Z_i = 0$
Saw cylinder shaft		
$P \cdot {}^{l}/_{2} - R_B \cdot l = 0$	$R_A \cdot l - P\,{}^{l}/_{2} = 0$	$R_A - P + R_B = 0$
$R_B = \dfrac{1400 \cdot 1{,}15}{2{,}3} = 700\ \text{H}$	$R_A = \dfrac{1400 \cdot 1{,}15}{2{,}3} = 700\ N$	$700 - 1400 - 700 = 0$
Saw cylinder, including saws and gaskets.		
$Q \cdot {}^{l}/_{2} + P \cdot {}^{l}/_{2} - R_B \cdot l = 0$	$R_A \cdot l - P\,{}^{l}/_{2} - Q \cdot {}^{l}/_{2} = 0$	$R_A - P - Q + R_B = 0$
$R_B = \dfrac{1977 \cdot 1{,}15}{2{,}3} = 988{,}5\ N$	$R_A = \dfrac{1977 \cdot 1{,}15}{2{,}3} = 988{,}5\ N$	$988{,}5 - 1400 - 577 - 988{,}5 = 0$

In order to make a mathematical description of the object of calculation and to solve the problem as simply as possible, real structures are replaced by idealised models or calculation schemes. In this case the calculation becomes approximate, with the help of this method we have carried out the bending calculation of the saw cylinder. Fig.1.b. shows the general view of the saw cylinder in the assembled version and the calculation diagram. In this step, the bending calculation of the saw cylinder shaft is carried out, taking into account the masses of saws and shims.

Fig. 1.b shows the scheme of the saw cylinder including saws and shims in the equally distributed variant in the calculation of which the transverse force is distributed along the length of the shaft ascending, and the bending moment in the flat form, this occurs directly under the influence of the equally distributed force (g - saw blades and shims (mass)). It follows that changing the mass of the spacer leads to a reduction in the bending moment of the shaft itself, as changing the mass of the saw blade is a difficult problem to solve. Then we apply the method of approximations for this case of the first approximation. And for the shaft itself we apply the second approximation and build the calculation diagram, which is shown in Fig. 1.b.

$$g_{sh} = \frac{G_s + G_d}{l} = \frac{g(m_s + m_d)}{l} \tag{1}$$

where, m_s, m_d - is the total mass of saw blades and gaskets mounted on the shaft, l-shaft length.

From reference data [10,11], the masses of which are respectively *0.3kg* and *0.15kg*. Taking into account their number on the saw cylinder, we calculate the total value of the equidistributed force.

$$g_s = \frac{9{,}8(39 + 20)}{2{,}3} = 251\ {}^{\text{H}}/_{\text{м}}$$

$$Q = g_s \cdot l \cdot {}^{l}/_{2} = 251 \cdot \frac{2{,}3^2}{2} = 577\ \text{H}$$

Calculations to determine the reaction forces on the supports were carried out using the above methodology and the results obtained are summarised in Table 2.

Table 2. The reaction forces

Formula	1-section	2-section
Saw cylinder shaft		
$Q_y = \Sigma P_y$	$Q_I = R_A = 700\ N$	$Q_{II} = R_A - P = -700\ N$
$M_{be} = \Sigma M_c$	$M_I = R_A x_1$ At, $x_1 = 0$; $M_I = 0$; $x_1 = 1.15$; $M_I = 0.8\ kN\text{-}m$	$M_{II} = R_A x_2 - P(x_2 - {}^{l}/_{2})$; At, $x_2 = 1.15$; $M_{II} = 0.8$; $x_2 = 2.3\ M_I = 0.7\text{-}2.3 - 1.4(2.3\text{-}1.15) = 0$
Saw cylinder, including saws and gaskets		
$Q_y = \Sigma P_y$	$Q_I = R_A = 988.5\ N$	$Q_{II} = R_A - P - Q = -988.5\ N$
$M_{be} = \Sigma M_c$	$M_I = R_A x_1$ At, $x_1 = 0$; $M_I = 0$; $x_1 = 1.15$; $M_I = 1136\ N\text{-}m$	$M = R_{IIA} x_2 - P(x_2 - {}^{l}/_{2}) - Q(x_2 - {}^{l}/_{2})$; At, $x_2 = 1.15$; $M_{II} = 1136\ N\text{-}m$; $x_2 = 2.3\ M = 0_{II}$

Next, we determine the internal forces of the saw cylinder shaft, taking into account the seed roller shaft (in static position). Fig. 1.c. shows the general view of the process and the calculation diagram. The calculation of which is important in the assembly and operation of the machines. Where to take into account the seed roller in the calculations consider in the following form. Considering the mass of the seed roller, we will calculate the mass force, and using the approximation method we will consider the distributed forces as concentrated forces (Fig. 1.b). Let's take the mass of the seed shaft in the following interval. $m_{rm} = 60 \div 90kg$ for this case, $U = 588 \div 882N$. Then in this case the moment of resistance arising from the load of the seed roller on the saw cylinder $M_{rm} = F_{tr} R$, where, $F_{tr} = f{\cdot}U$.

The next question of interest is the influence of the seed shaft on the deformation process of the saw cylinder shaft. Let's consider the proposed calculation scheme in the following form, where the shaft gravity force P and equally distributed force g will be considered as accepted in the variant of the calculation scheme of the second approximation. Forces P and Q by substituting the geometric centre of the shaft in our case - it is point (C) Fig. 1.c. In order to take into account the seed shaft (external resistance) we will insert the force U, here from the seed shaft arises torque or the so-called moment of resistance of the system M_c. In Fig. 1.c, the calculation scheme in cross section is shown. The calculation is carried out according to the above mentioned method. Table 3 summarises the calculation data on the influence of the seed shaft mass on the deformation process of the saw cylinder shaft, taking into account the seed shaft.

Table 3. The influence of the seed shaft mass on the deformation process of the saw cylinder shaft

M_{rm} (kg)	U (N)	F_{fr} (H)	M_{rm} (N-m)	P(H)	Q(H)	R_A (H)	R_B (H)
60	588	705,6	112,90	1400	577	1331,59	1233,41
70	686	823,2	131,71	1400	577	1388,77	1274,23
80	784	940,8	150,53	1400	577	1445,95	1315,05
90	82	1058,4	169,34	400	77	1503,13	1355,87

The shaft diameter can now be determined using the allowable stress theory [6]:

$$\sigma_{max} = \frac{M_{max}}{W_y} \leq [\sigma] \tag{2}$$

Axial moment of resistance of a cross section:

$$W_z = \frac{\pi d^3}{32} \tag{3}$$

From this we can calculate the shaft diameter for the permissible load:

$$d \geq \sqrt[3]{\frac{32 \cdot M_{max}}{\pi \cdot [\sigma]}} \tag{4}$$

Next, we determine the transverse internal forces and bending moments for each case separately using the section method (Table 4).

Table 4. The transverse internal forces and bending moments for each case

M_{rm} (kg)	Q_1 (H)	Q_2 (H)	M_1 (N-m)		M_2 (N-m)	
			x=0	x=1,15	x=1,15	x=2,3
60	1331,59	-1233,41	0	1531,323	1418,427	0
70	1388,77	-1274,23	0	1597,081	1465,369	0
80	1445,95	-1315,05	0	1662,839	1512,311	0
90	1503,13	-1355,87	0	1728,597	1559,253	0

To determine the deflections Y_1, Y_2, Y_3 in the direction of mass oscillations we apply unit forces; the epuples from these unit forces M_1 , M_2 , and M_3 are multiplied with the epuple of bending moments M from forces Q by the Vereshchagin method. The moments of inertia of the cross sections are calculated according to [12,13]:

$$J = \frac{\pi d^4}{64}$$

On the basis of the obtained solutions with variations of technological resistance (seed shaft mass) (m_{rm} , kg) the graphs characterising the influence of the seed shaft mass on the forces of the cross section (Q, N) and on the bending moment of the shaft (M_{be} , N/m) are plotted in Fig.2.

The results of processing of the obtained solutions with variations of technological resistance (seed shaft mass) show that with the increase of technological resistance linearly increases the reaction forces on the bearings, the growth of which directly affects the forces of cross section and on the bending moment of the tedder shaft. The character of the obtained curves of influence of the seed roller mass on the reaction forces in the bearings corresponds to the character of the curves of influence of the seed roller mass on the cross-sectional forces (Q, N) and on the bending moment of the shaft (M_{be} , N/m). So if at the mass of the seed shaft m_{rm} =60kg force of the cross section Q=1331 N and bending moment of the shaft M_{be} =1531 N/m, then at the mass of the seed shaft m_{rm} =90kg force of the cross section and bending moment of the shaft respectively was Q=1503 N, M_{be} =1728N/m. It should be noted that with the increase of the seed shaft mass from 60kg to 90kg also increases the value of the cross-sectional forces from 2565N to 2869N and the bending moment of the shaft from 112N/m to 170N/m respectively Fig. 2.

Next, we calculate the shaft deflection:

$$Z = Z_0 + \Theta_0 x + \frac{1}{EJy}\left[\sum M\frac{(x-a)^2}{2} + \sum P\frac{(x-b)^3}{6} + \sum q\frac{(x-c)^4}{24}\right] \quad (5)$$

Initial condition: x=0; $\begin{cases} Z_0 \neq 0 \\ \Theta_0 \neq 0 \end{cases}$, $a=b=c=l/2$

Taking into account the shaft arrangement and the arrangement of the masses, it can be determined as followsΘ_0 as follows, if $x = 2a$, то $Z_b = 0$ hence:

$$\Theta_0 = \left\{ \frac{\frac{1}{EJ_y}\left[\frac{R_A 2a^3}{6} - \frac{M(2a-a)^2}{2} - \frac{P(2a-a)^3}{6} + \frac{R_b(2a-2a)^3}{6}\right]}{2a} \right\} \quad (6)$$

After determining Θ_0 the shaft deflection can be calculated for each section or for each x value. The calculations are summarised in Table 5.

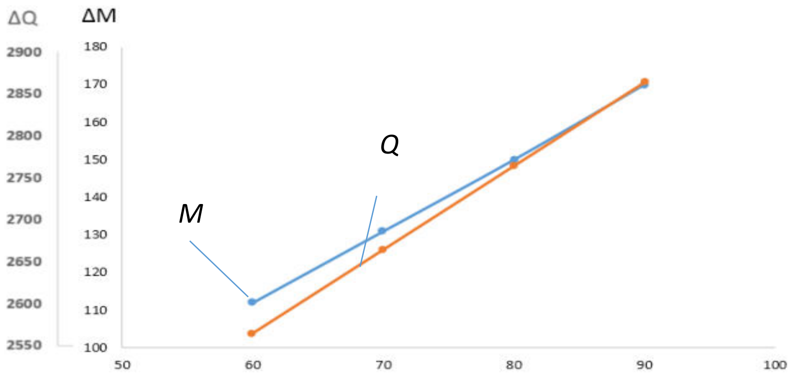


Fig. 2. - Effects of seed roll mass on the cross-sectional force differences (Q·15, N)and on the shaft bending moment (M_{be} , N/m)

For each case, the effect of changing the density or mass of the raw shaft on the saw cylinder shaft deflection was considered.

Table 5. Effect of changing the density or mass of the raw shaft

M_{rm}, kg	Shaft deflection *Z, (mm)*			
	x = 0,58	x = 1,15	x = 1,73	x = 2,3
60	-0,46413	-0,66344	-0,5526	0
70	-0,48247	-0,68879	-0,5756	0
80	-0,50082	-0,71414	-0,59859	0
90	-0,51917	-0,73949	-0,62158	0

The cross-section of the shaft and the weight of the saw cylinder in the existing design is unreasonably large (*d=100mm)*. If the shaft diameter is reduced, a more favourable variant in terms of energy saving can be achieved. This large mass of the saw cylinder (mainly due to the mass of the shaft) leads to the fact that the machine requires a high power consumption (17-17.5 kW) when starting the machine.

It is known [14-18] that the power consumption in the set mode of the saw cylinder is 12-12.5 kW. Inertia mass of the saw cylinder is large, in the stop mode it takes 7-12 seconds to completely stop the saw cylinder. This adversely affects the product quality.

Conclusion

After the calculation works to identify the influence of the technological load on the saw cylinder shaft, it is suggested to reduce the inertia moment (mass) of the saw cylinder and to adjust the density of the seed shaft. The main method of mass reduction is to lighten the shaft itself or to reduce the number of saws and shims, which will reduce the length of the shaft. The adjustment of the seed roll density depends on many parameters but the most effective is the development of a new design of seed tedder for the working chamber of the linter machine.

The angular speed can be increased by reducing the bending of the saw cylinder shaft. In this way, the machine performance is not actually reduced.

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