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Statically Analysis of the Stress State of Saw Gins Consisting of 90, 100, 110, 120, 130 Saws

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Abstract

In the primary processing of cotton, it is important to increase the productivity of ginning and reduce the wear of working bodies. The working body of the genie is the working chamber, the saw cylinder and the rib grate. The main wear is on the saw cylinder shaft and saw teeth. The wear of these parts leads to additional material costs, as well as to a decrease in the quality of the fiber. The wear of the shaft is affected by the number of saws and the mass of raw cotton in the working chamber of the gin. To prevent wear of the saw cylinder, the article determines the optimal static load on the shaft by calculating saw gins consisting of 90, 100, 110, 120, 130 saws. An analysis of tables shows that maximum value to bending shows 120 and 130 saws cylinder, because shaft bending angles along the length appear. This leads to a 2% - 3% reduction in the distance between the saws, serves for the premature wear of the saw, the exit of short fibers.

Keywords

Raw, Cotton, Shaft, Statically, Analysis, Saw, Gin

1. Introduction

Many primary cotton processing enterprises are faced with the problem of obtaining high-quality fiber with less trash and Comparison of fiber properties processed in different cotton processing plants, with different models of fiber separators. Cotton fiber was obtained as a result of ginning with such factory gins 4DP-130.5 DP-130, 3HDD and Lummus gins. The fiber properties of all samples were measured with the Advanced HVI Fiber Information System, to determine the yield of short fibers and UHML. The study is to determine which model of sawing gin produces more short fibers.

Also many studies were carried out to determine the static and dynamic loads on the saw cylinder shaft with gins 4DP-130 and 5DP-130. Nevertheless, do not have statically analysis of the stress state of saw gins consisting of 90, 100, 110, 120, 130 saws. But, this is research necessary to reduce shaft deflection and increase productivity.

Start our research by weighing the parts of the saw cylinder. Determining the own weight of the shaft, taking into account the number of saws and saw gasket for one saw cylinder [1] [2].

The Saw cylinder shaft material steel, steel density— $\rho_{steel} = 7800\ \text{kg}/\text{m}^3$.

Saw cylinder shaft diameter— $d_{axis} = 0.06(\text{m})$.

The weight of one saw is $G_{saw} = 0.5(\text{kg})$ **Figure 1**.

Gasket weight $G_p = 0.3(\text{kg})$ **Figure 2**.

Distance between saws—Δ.

Mainly depends on the number of saws.

The length of the saw cylinder determined [3] by the formula:

$$l = n \cdot \Delta + 2 \cdot \Delta; (\text{m}) \tag{1}$$

The volume of the saw cylinder determined [4] by the formula:

$$V = S \cdot l = \frac{\pi \cdot d_{axis}^2}{4} \cdot l; (\text{m}^3) \tag{2}$$

Figure 1. The weight of one saw

Figure 2. Gasket weight

The weight of the saw cylinder determined [5] [6] [7] by the formula:

$$G_{axis} = \rho_{steel} \cdot V; (\text{kg}) \tag{3}$$

2. Materials and Methods

The total weight of the saws of the saw cylinder determined [8] by the formula

$$G_{\Sigma saw} = n \cdot G_{saw}; (\text{kg}) \tag{4}$$

The total weight between the saw gaskets determined [9] by the formula

$$G_{\Sigma p} = n \cdot G_p; (\text{kg}) \tag{5}$$

The total weight of the saw cylinder determined [10] by the formula

$$G_{\Sigma} = G_{axis} + G_{saw} + G_p; (\text{kg}) \tag{6}$$

In the calculations, the own weight of the shaft and saw is determined depending on the number of saws and saw blades, as well as the length of the shaft showed in **Table 1**.

Determination of the weight of raw cotton in the working chamber of the saw cylinder [11] [12].

When determining the action of raw cotton pressure on the saw cylinder, the following assumptions made:

- The force of cotton pressure along the length of the shaft is constant;
- The coefficient of friction between the teeth of the saw and the cotton also assumed contact.

In work the pressure of the raw roller on the saw cylinder was studied and the total values of the pressure of raw cotton on the section AB of the saw showed in **Figure 3** depending on the angle of inclination of the grate—φ are given. In this work, we will use the results and consider the value $P = 82$ Newton [12].

$$\begin{cases} P_n^0 = P \cdot \cos\varphi \\ P_\tau^0 = P \cdot \sin\varphi \\ F_t^0 = f \cdot P_n = f \cdot P \cdot \cos\varphi \end{cases} \tag{7}$$

Table 1. Weight of the shaft depending on the number of saw blades.

№	N	Δl [m]	L [m]	V [m^3]	G_{axis} [kg]	$G_{\Sigma saw} + G_{\Sigma p}$ [kg]	G_{Σ} [kg]
1	90	0.018	1.656	0.00468	36.5	52.2	88.7
2	100	0.017	1.734	0.0049	38.2	58	96.2
3	110	0.016	1.792	0.0051	39.5	63.8	103.3
4	120	0.016	1.952	0.0055	43.03	69.6	112.6
5	130	0.016	2.112	0.0060	46.55	75.4	122
6	120	0.022	2.684	0.0076	59.16	69.6	128.76
7	130	0.022	2.904	0.0082	64.01	75.4	139.41

AB—rib grate;

M—Point of application of the resulting cotton pressure;

P_n—is the projection of *P* onto the normal;

P_τ—projection of *P* onto the tangent;

φ—the angle of inclination of the grate to the horizontal;

F_t—is the friction force of the cotton on the saw cylinder;

f—is the coefficient of dry friction.

The pressure of raw cotton on the saw cylinder along the length of the shaft, depending on the number of saws—*n*.

$$\begin{cases} P_n = P_n^0 \cdot n \\ P_\tau = P_\tau^0 \cdot n \\ F_t = F_t^0 \cdot n \end{cases} \tag{8}$$

Values formula 8 a given in **Table 2**.

In the plane of the cross sections of the shaft, the pressures of raw cotton a calculated according to the norms and in the tangential direction, depending on the amount of saw [13].

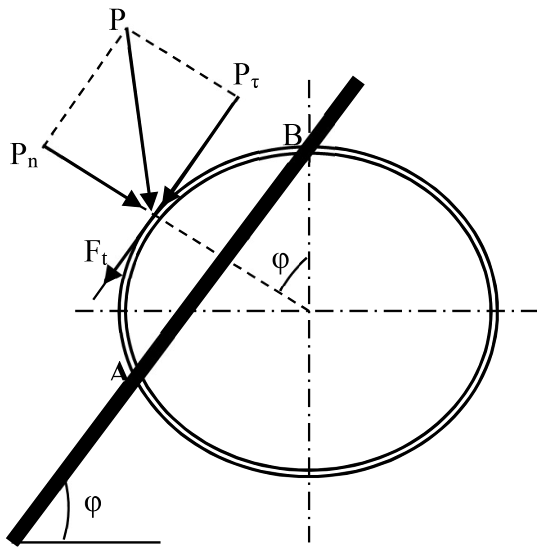

Figure 3. The pressure of raw cotton on the section AB

Table 2. The pressures of raw cotton on the saw cylinder along the length of the shaft a given, depending on the number of saws—*n*.

№	*N*	P_{max} [N]	P_n [N]	P_τ [N]	F_t [N]	φ [grad]
1	90	82	58	57.9	11.6	45˚
2	100	82	58	57.9	11.6	45˚
3	110	82	58	57.9	11.6	45˚
4	120	82	58	57.9	11.6	45˚
5	130	82	58	57.9	11.6	45˚

With simultaneous deformation by bending with torsion, the internal force in the cross section of the shaft leads to five components:

1) Torque $M_X = M_K$—relative to the geometric axis—X.

2) Bending moment M_Y—relative to the main central axes of inertia of the sections—JY.

3) M_Z—relative to the main central axes of inertia of the sections—JZ.

4) Shear forces Q_Y relative to the geometric axis—Y.

5) Shear forces Q_Z about the geometric axis—Z.

Since the shaft has, a circular cross section with a diameter d_0, the shear stresses determined by the forces Q_Y and Q_Z are of secondary importance and can ignored in the calculations.

In this case, the shaft works on a complex resistance. In the YZ plane, torsion occurs; in the XY and XZ planes, the shaft bends. Their values a given in **Table 4**. The values formula (9), (10) of the bending moments of inertia with different numbers of saws and between of saw distance showed in **Table 4**.

$$M_{kr} = \frac{P \cdot D \cdot \sin\varphi}{2} + \frac{F_{tr} \cdot D}{2}; \tag{9}$$

$$\begin{cases} M_y = \dfrac{q_y \cdot l}{2} \cdot x - \dfrac{q_y \cdot x^2}{2}; \\ M_z = \dfrac{q_z \cdot l}{2} \cdot x - \dfrac{q_z \cdot x^2}{2}; \end{cases} \tag{10}$$

$$q_y = q_{oy} + q_{cvy} + q_{try} \cdot q_z = q_{trz}; \tag{11}$$

q_{oy}—Intensity of pressure of the own weight of the shaft, taking into account saws and saw spacers along the Y axis;

q_{cvy}—Intensity of raw roller pressure along the Y axis;

q_{try}—The intensity of the friction force of the raw roller along the axis—Y;

q_{trz}—The intensity of the friction force of the raw roller along the axis—Z;

Values formula 11 a given in **Table 3**.

According to the model of static calculation of the saw cylinder showed in **Figure 4**, internal force factors are determined: torque—M_{kr}.

Bending moments M_z, M_y—respectively, relative to the main axes of inertia of the sections iY, iZ.

In the case under consideration, the shaft works for complex resistance, that is, in the YZ plane—for torsion in the XZ plane and XY for bending.

In the YZ plane, torsion occurs; the shaft bends in the XY and XZ planes showed in **Figure 4**.

Plots of twisting and bending moments from the pressure of raw cotton and from the pressure of the own weight of the shaft of saws and inter saw spacers are constructed [14].

From graphics $M_{kr}^{\max}$, $M_z^{\max}$, $M_y^{\max}$—it can be seen that the dangerous section of the shaft will be the section in the middle along the length of the shaft. The maximum normal bending stresses and shear stresses are determined based

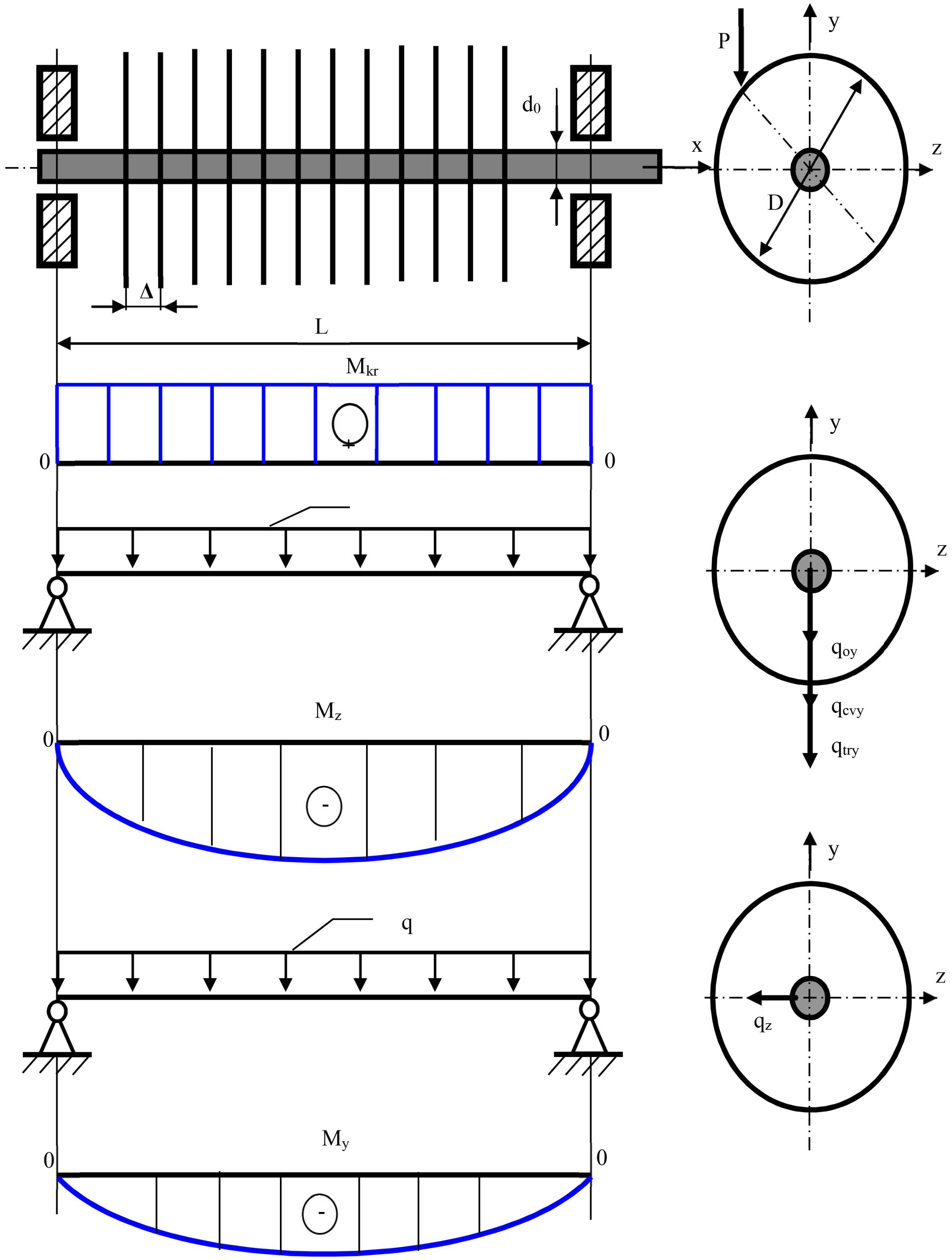

Figure 4. Saw cylinder static calculation model.

Table 3. The pressure intensity of the saw cylinder, raw roller is given, taking into account the friction force relative to the Y and Z axes.

№	n	Δl (m)	q (N/m)	q_n (N/m)	$q_{t\text{-}y}$ (N/m)	q_y (N/m)	q_z (N/m)
1	90	0.018	535	48.2	4.82	548.02	8.2
2	100	0.017	555	47.10	4.71	566.81	8.2
3	110	0.016	576	45.60	4.56	586.16	8.2
4	120	0.016	577	42.50	4.25	623.75	8.2
5	120	0.022	479	30	4.25	513	8.2
6	130	0.016	578	40.30	4.03	622.33	8.2
7	130	0.022	480	27	4.03	511	8.2

Table 4. Shows the values of the bending moments of inertia with different numbers of saws and between of saw distance.

№	n	Δl (m)	$M_{kr}^{\max}$ (N·m)	$M_y^{\max}$ (N·m)	$M_z^{\max}$ (N·m)	$M_{ekv}^{\max}$ (N·m)
1	90	0.018	11.13	187.85	3.98	187.89
2	100	0.017	11.13	212.73	4.36	212.77
3	110	0.016	11.13	235.23	4.66	235.27
4	120	0.016	11.13	296.73	5.52	296.77
5	120	0.022	11.13	461.95	10.44	462.06
6	130	0.016	11.13	346.81	6.47 /	346.87
7	130	0.022	11.13	538.67	12.23	538.81

on the maximum values of the moments according to the formula:

$$\sigma_{\max} = \frac{M_{ekv}}{W}$$
$$\tau_{\max} = \frac{M_{kr}}{W} \tag{12}$$

W—static section modulus of the shaft

$$W = 0.1 \cdot d_0^3 = 21.6 \times 10^{-6}\ \text{m}^3$$

The shear stress, which determines the torque, reaches the highest value at all points of the section contour, and the largest normal stresses, which determine the moments M_y and M_z, are obtained at the points of the edge of the section contour lying at the ends of the diameter perpendicular to the vector of the resulting bending moment, showed in Figure 4.

According to the third hypothesis of strength, we determine the calculated stress values σ_p by the formula

$$\sigma_p = \sqrt{\sigma_{\max}^2 + 4\tau_{\max}^2} \tag{13}$$

According to the diagrams of torque and bending moments, the maximum values of normal stresses and tangential stresses a found. Their values, depending on the number of saws, a given in Table 5. Increasing the angle of the shaft and increasing the deflection of the shaft leads to vibration of the shaft which reduces the performance of the fiber showed Figure 5.

All calculations will be similar to those for a single cylinder gin. The solution results formula (14) and (15) are deflection and angle of inclination of the shaft shown in Table 6.

Angle of inclination of the shaft we find by (14) formula

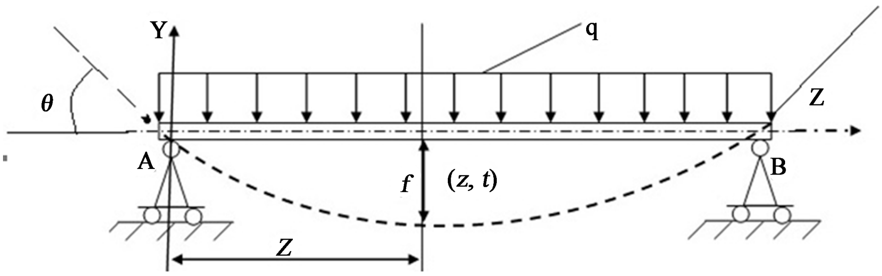

Figure 5. Shaft deflection.

Table 5. The values of all voltages are given depending on the number of saws in the saw cylinder.

№	n	$\tau_{max} \cdot 10^5$ [N/m²]	$\sigma_{max} \cdot 10^5$ [N/m²]	$\sigma_p \cdot 10^5$ [N/m²]	Δl [N/m]
1	90	5.13	8.7	8.71	0.018
2	100	5.13	9.85	9.86	0.017
3	110	5.13	10.89	10.90	0.016
4	120	5.13	13.74	13.75	0.016
5	130	5.13	16.06	16.07	0.016
6	120	5.13	21.39	21.40	0.022
7	130	5.13	24.94	24.95	0.022

Table 6. Deflection and angle of inclination of the shaft.

№	n	f_{max} [m]	θ_{max} [rad]
1	90	0.00033	$0.1 * 10^{-4/2}$
2	100	0.00040	$0.11 * 10^{-4/2}$
3	110	0.00045	$0.13 * 10^{-4/2}$
4	120	0.00064 *	$0.16 * 10^{-4/2}$
5	130	0.00085	$0.21 * 10^{-4/2}$
6	120	0.002.04	$0.40 * 10^{-2}$
7	130	0.00239	$0.50 * 10^{-2}$

$$\theta = \frac{q \cdot l^2}{24EI_x}; \tag{14}$$

The deflection of inclination of the shaft we find by (15) formula

$$f = \frac{5ql^4}{384EI_x}; \tag{15}$$

The results of the calculation of a single-chamber two-cylinder gin for a stress-strain state are shown in **Table 6**. Where the self-weight of the shaft, saw and saw blades is determined.

An analysis of **Table 5** and **Table 6** shows that due to bending, shaft bending angles along the length appear. This leads to a 2% - 3% reduction in the distance between the saws, serves for the premature wear of the saw, the exit of short fibers.

3. Conclusions

The calculations determined the own weight of the shaft and saw, depending on the number of saws and saw blades, as well as the length of the shaft. The heaviest was the saw cylinder with 130 saw blades.

In the plane of the cross sections of the shaft, the pressures of raw cotton were calculated according to the norms and along the tangential direction, depending on the amount of saw.

According to the model of static calculation of the saw cylinder internal force factors were determined: torque—M_{kr}, Bending moments M_z, M_y—respectively, relative to the main axes of inertia of the sections JY, JZ; in the case under consideration, the shaft works for complex resistance, that is, in the YZ plane—for torsion in the XZ plane and XY for bending.

Plots of twisting and bending moments from the pressure of raw cotton and from the pressure of the own weight of the shaft of saws and between saw spacers are constructed. According to the diagrams of torque and bending moments, the maximum values of normal stresses and tangential stresses a found. It has been established that with an increase in the number of saws, the maximum value of normal stresses increases. As in classical calculations, the dangerous section of the shaft is the middle of the shaft along the length; raw cotton pressure on the shaft; maximum, the value of torque and bending moments for dangerous sections of the shaft. A comparison was made of the values of the moments depending on the width of the saw blades. The values of tangential and normal stress an obtained depending on the number of saws and the distance between the saws. An analysis of tables shows that maximum value to bending shows 120 and 130 saws cylinder, because shaft bending angles along the length appear, reduced fiber removal from saw teeth, also this bending leads to a 2% - 3% reduction in the distance between the saws, serves for the premature wear of the saw, the exit of short fibers. It is necessary to reduce the number of saws to 90 - 80 pieces and reduce the weight of the shaft to increase the effective removal of fiber from the saw teeth.

Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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