Improvement of the Saw Cylinder of the Saw Gin Stand

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ABSTRACT

The article examines the problem of increasing the rigidity of the saw cylinder of a saw gin by changing the cross-section of its shaft. It has been proposed to use a hexagonal shaft instead of a round shaft. Calculations were carried out using the SolidWorks Simulation program. Performed static, frequency, fatigue, as well as equilibrium and SolidWorks – Design Insight analysis of two types of saw cylinders. The influence of the shaft shape on the static performance of saw blades has been studied. The results made it possible to use a hexagonal shaft for the saw cylinder of the saw gin.

Keywords: Shaft, Stress, Displacement, Deformation, Safety Factor

The saw gin stand is the main technological machine for the primary processing of cotton. The quality of the produced fibre and seeds, machine productivity, downtime of associated technological machines, the service life of the main working parts, energy consumption, etc. depend on its correct operation. Therefore, improving the saw gin, in particular the saw cylinder, is one of the pressing issues in the primary processing of cotton.

Scientists and engineers have studied the problems of increasing strength and rigidity, reducing weight, reducing vibration of the saw cylinder, etc. [1-4], but on the saw cylinders in use, there are problems with the bending of its shaft above the permissible level. [5, 6]

The design of a gin saw cylinder [7] is known, containing a shaft, saw blades installed on it, spacers between the saws, washers and clamping nuts. The shaft is splined with transitional curves at the spline bases and grooves, and the saw blades are equipped with tongues located with the possibility of contact with the grooves on the shaft, and the tongues and grooves are made symmetrically on both sides. However, the known design is characterized by the complexity of design and manufacture.

The closest in technological essence to the proposed one is a saw cylinder [8], containing a shaft, saw blades with “fingers” installed on it, which fit into the groove of the shaft, spacers, washers and clamping nuts. The disadvantage of the known design is a significant deflection of the shaft, leading to a change in technological distances between saws and gaps, a large power requirement due to the massiveness of the saw cylinder, which leads to damage.

The gin saw cylinder was chosen as the prototype [8].

The existing designs of gin saw cylinders are very massive, which causes deflections beyond the permissible limits (0.36-0.40 mm). As a result, there is a change in the position of the saws in the slot gap between the grates, leading to damage to the fibres when they are pulled by the teeth of the saws through the grates, a reduction in the service life of the saws and grates, as well as an increase in energy consumption due to friction of the saws on the grates. Therefore, the development of a new gin saw cylinder is of great importance.

The objective of the study is to increase the reliability of the gin saw cylinder, save resources and increase productivity.

The problem is solved by the fact that the saw cylinder contains a shaft, saw blades mounted on it, spacers between the saws, washers and clamping nuts, the shaft is made hexagonal, and the saw blades have a hexagonal inner surface located with the possibility of contact with the edges of the hexagonal shaft.

Making the shaft hexagonal can significantly reduce weight while maintaining the bending rigidity of the shaft, leading to resource-saving, increased reliability and production of fibre with the required quality indicators.

The gin saw cylinder (Fig. 1), contains a shaft 1, saw blades 2 mounted on it, spacers 3 between the saws, washers 4 and clamping nuts 5, the shaft is made hexagonal, and the saw blades have a hexagonal inner surface, located with the possibility of contact with the edges of the hexagonal shaft.

The design works as follows. During operation, when feeding seed cotton, saw blades 2 capture strands of fibres and drag them behind the grate bars (not shown in the figure), and the strands of fibres are torn off from the cotton seeds. Reducing the mass of the saw cylinder of the gin by making shaft 1 hexagonal ensures the bending of shaft 1 within acceptable limits, allows the required process of cotton fibre separation, and reduces the required power of the gin. The manufacture of saw blade 2 with a hexagonal inner surface during operation leads to a kind of balancing of the masses of the system relative to the axis of rotation.

The recommended design makes it possible to increase reliability, reduce the required power of the gin, and obtain high-quality fibre with high productivity.

Fig. 1, shows a general view of the saw cylinder of the gin, containing shaft 1, saw blades 2 installed on it, spacers 3 between the saws, washers 4 and clamping nuts 5, the shaft is made hexagonal, and the saw blades have a hexagonal inner surface located with the possibility of contact with the edges hexagonal shaft, in Fig. 1, b – section A-A of the saw cylinder shown in Fig. 1, but with a shaft, a saw blade with a hexagonal inner surface and an inter-saw spacer.

To further study the proposed hex shaft, mechanical characteristics were calculated and compared with the existing shaft.

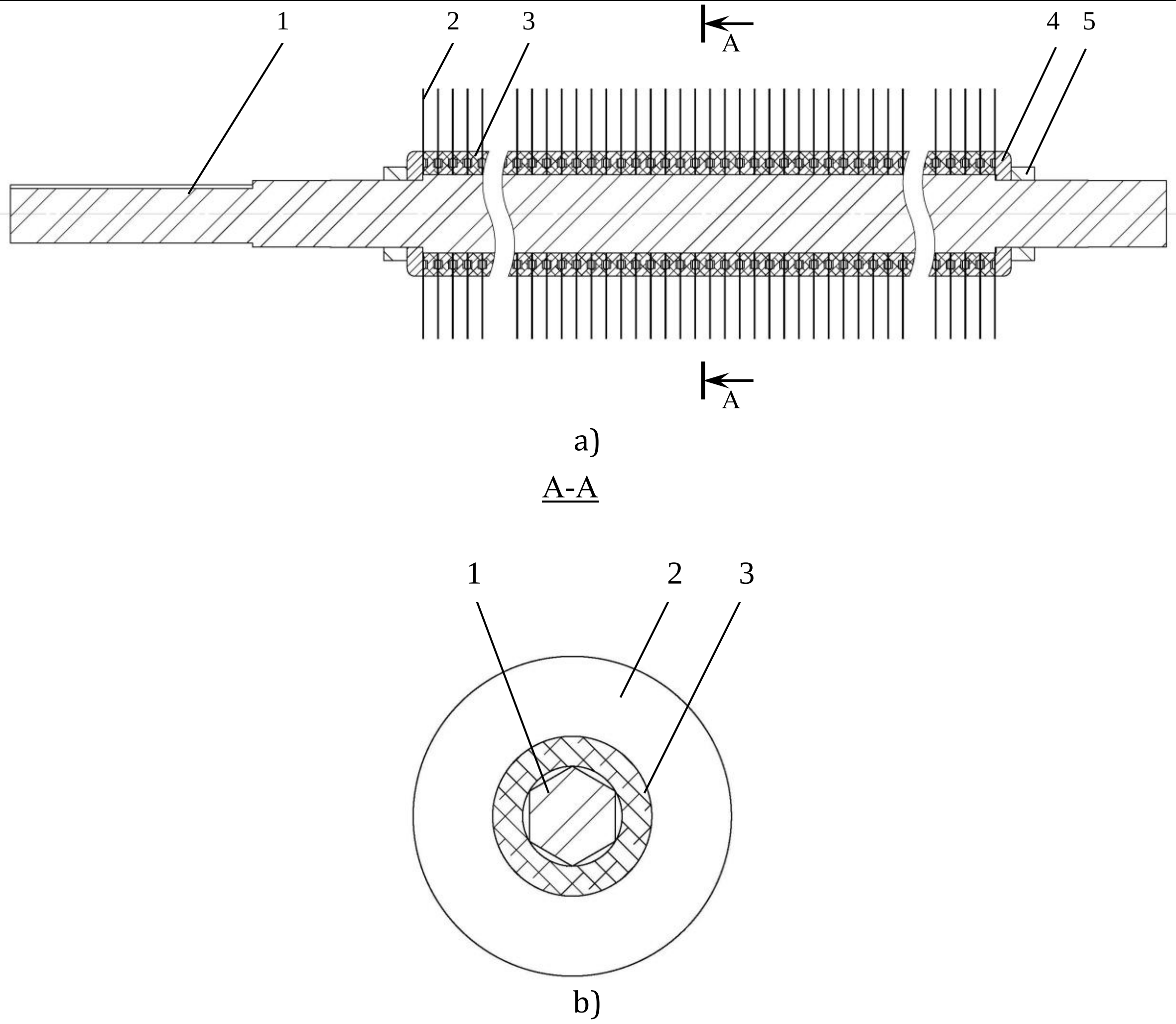


Fig.1. Improved gin saw cylinder

Static calculation of the hexagonal shaft of the saw cylinder. The saw cylinder of the saw gin is the most loaded working part. Insufficient strength and rigidity, as well as vibrations of the saw cylinder parts, negatively affect the ginning process and deteriorate the quality of fibre and seeds. Therefore, it is important to perform power calculations of the shaft, saw blades, and spacers between the saws.

Figure 2 shows the design diagram of the saw cylinder and the loads affecting it. External loads affecting the shaft include the following:

- evenly distributed load on the shaft between the supports (we do not take into account the weight of the seed roll due to its insignificance)

$$q_1 = \frac{G_v + G_{ar} + G_q + G_{q.sh.}}{l_0},$$

Here: G_v – shaft weight between supports, kg;

G_{ar} – saw weight, kg;

G_q – weight of inter-saw spacers, kg;

$G_{q.sh.}$ – weight of clamping nuts, kg;

- uniformly distributed load from the weight of the cantilever part of the shaft q_2;

- the axial tightening force of saws A_0 ($A_0 = 20 \cdot 10^3$ N);

- coupling weight G_m;

- torque from electric motor drive M_b.

Torque M_b transmitted to the saw cylinder shaft

$$M_b = 9550 \cdot \frac{N}{n} \qquad Newton \cdot m,$$

Here: N – power transmitted to the saw cylinder shaft (for gin 4DP-130, 5DP-130 – $N = 75\ kW$, for gins DPZ, 7DP, 8DP – $N = 45\ kW$); (1)

n – saw cylinder rotation speed ($n = 730\ rpm$).

$$M_b = 9550 \cdot \frac{75}{730} = 981 \qquad Newton \cdot m.$$

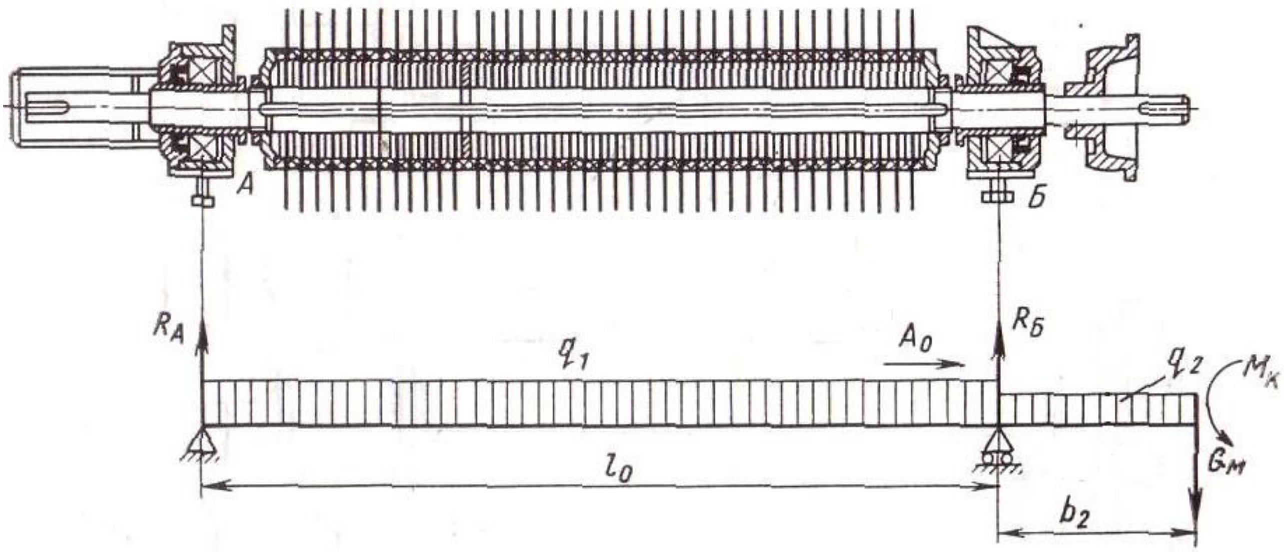
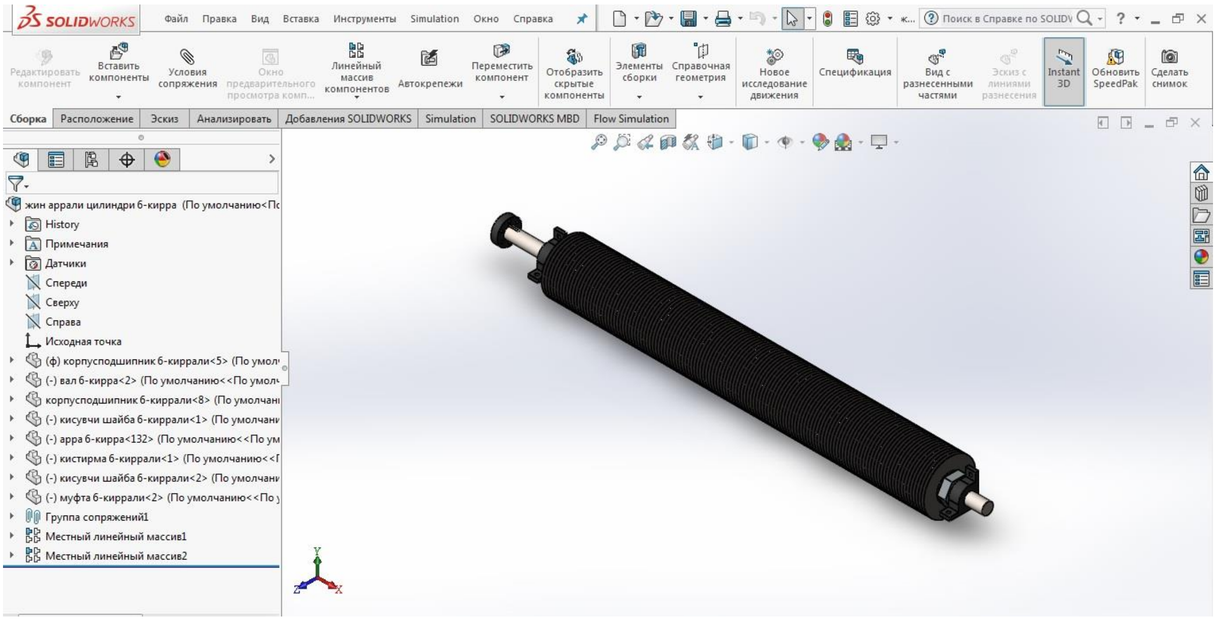


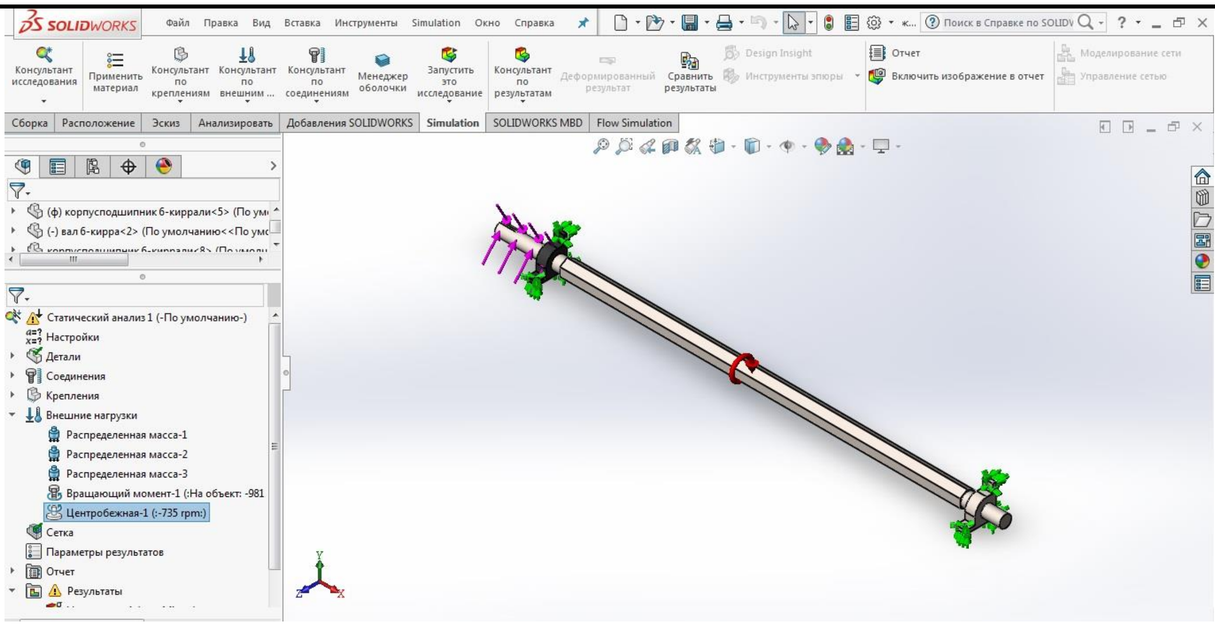
Fig. 2. Design diagram of the saw cylinder shaft and the forces acting on it

We will calculate the hexagonal shaft using the Simulation package of the SolidWorks program. For this purpose, parts of the saw cylinder were designed according to dimensions in 3D, materials were selected and assembly was performed (Fig. 3, a). To simplify the calculation, the influencing loads are shown according to the above diagram (Fig. 3, b).

To find q_1, q_2, G_m, the "Mass Characteristics" function of the SolidWorks program was used. The data obtained for shafts of different profiles are shown in Table 1. In Figure 4 shows diagrams of calculations of the shaft "Stress", "Displacement", "Deformation" and "Safety Factor".



a)



b)

Fig. 3. 3D view of a saw cylinder with a hexagonal shaft (a) and the reduction of external loads (b)

Table 1: Mechanical characteristics data obtained for different shaft sections

Shaft type	Weight of saw blades, kg	Shaft weight, kg	Saw cylinder weight, kg	Mass applied to the shaft, kg	Maximum shaft stress, MPa	Maximum shaft movement, mm	Maximum shaft deformation	Shaft safety factor
Round shaft								
Option 1	0,54212	161,76	319,82	296,32	24,0	0,33	0,00017	7,4
Option 2					24,0	0,32	0,00018	7,7
Octahedral								
Option 1	0,54759	148,37	307,14	283,64	60,2	0,22	0,00062	3,7
Option 2					72,4	0,21	0,00065	3,7
Hexagonal								
Option 1	0,55189	137,86	297,52	274,02	23,4	0,040	0,00014	11,6
Option 2					23,3	0,039	0,00014	11,4
Difference (between round and hexagonal shafts)	0,00977	-23,9	-22,3	-22,3	-0,7	-0,28	-0,00003	3,7
Difference, %	1,8	-14,8	-7,0	-7,5	-2,9	-84,8	-16,7	48,1

*Option 1 – position of the shaft with the groove (edge) at the top;

*Option 2 – shaft position when rotated 90°;
*The "-" sign means a decrease, the "+" sign means an increase.

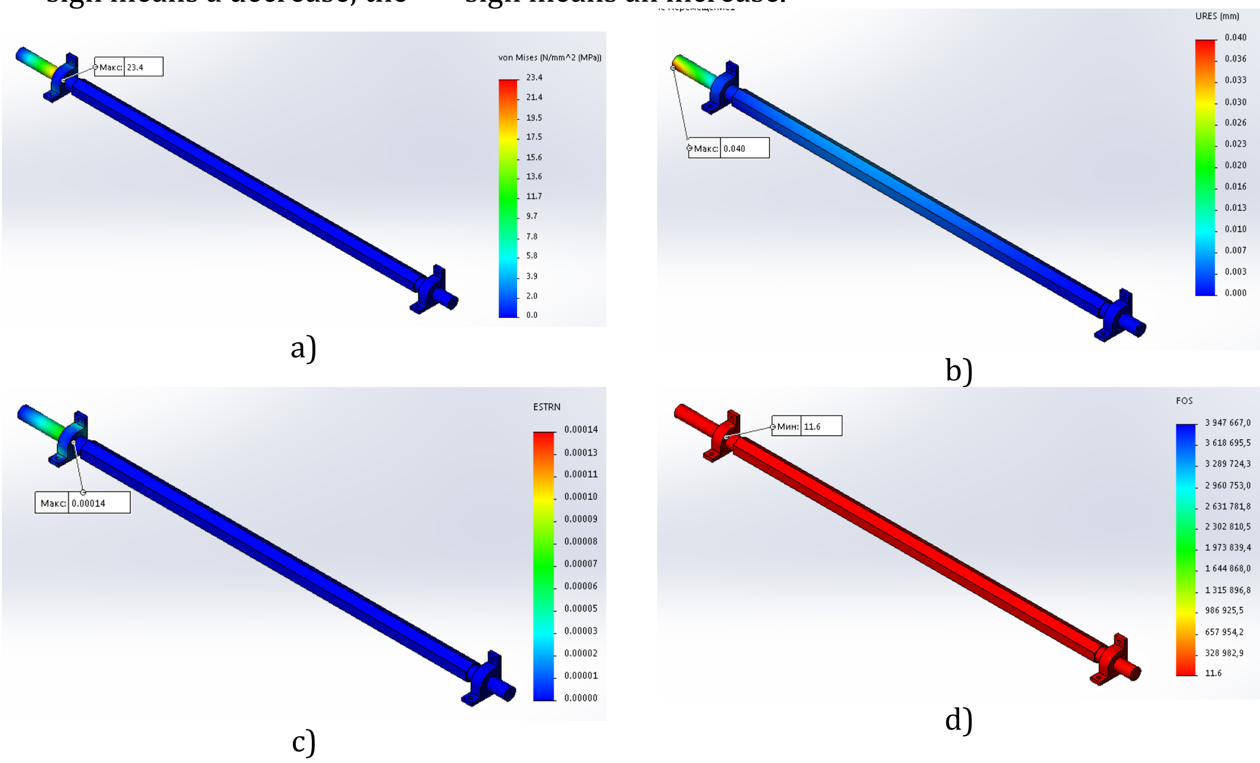


Fig. 4. Stress (a), displacement (b), deformation (c) and safety factor (d) of the hexagonal shaft of the saw cylinder

As can be seen from Table 1, the maximum movement of the hexagonal shaft was 0.04 mm, the minimum safety factor was 11.6. For the shafts under study, the displacement should not be more than 0.36-0.40 mm, and the safety factor should not be less than $[n] \geq 1{,}5 \div 2{,}5$ [8]. From which we can conclude that the proposed hexagonal shaft meets the requirements.

Determination of the natural frequency of the hexagonal shaft of the saw cylinder. In the available gins, the saw blade diameter is 320 mm and the rotation speed is 730 rpm. In order to increase productivity, it was proposed to increase the rotation speed of the saw cylinder, which led to machine vibrations. [8]

The operating speeds of the shafts of primary cotton processing machines are up to the first critical speed, so it is enough for us to determine the first critical speed. Such shafts are called “rigid”, and those exceeding the first critical speed are called “flexible”. Based on practice, for rigid shafts, condition $n_{1kr} \geq 1{,}3 n_{ish}$ can be used between operating n_{ish} and critical speed n_{1kr}.

As is known, the critical speed of a part in the general case depends on the rigidity K, i.e. material, and its mass M : $\omega_{kr} = \sqrt{K/M}$.

This means that increasing the size of a part in order to increase strength will lead to an increase in its mass. As a result, its critical speed decreases, i.e. it is necessary to rotate the part at a lower speed. The higher the critical speed, the more we can increase the rotation speed. When calculating the critical speed of the saw cylinder, we can use the existing diagram (Fig. 5).

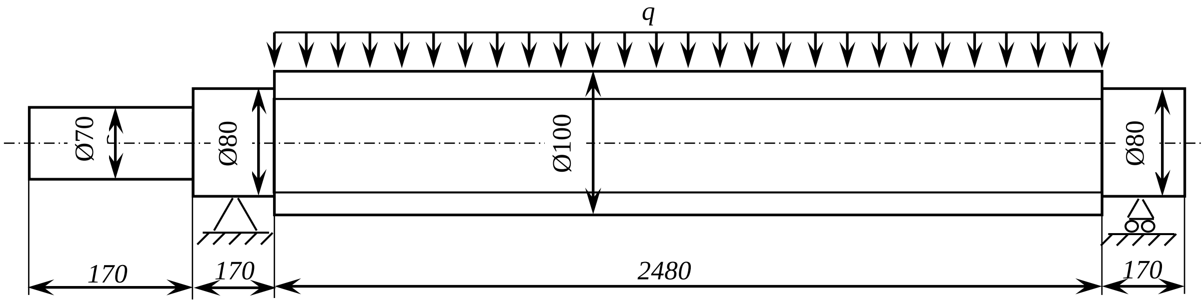


Fig. 5. Calculation diagram for determining the critical speed of a saw cylinder with a hexagonal shaft

The calculation was carried out using the SolidWorks Simulation program. Having entered all the necessary data and performed the required operations, we find the critical amplitude of shaft oscillations (Fig. 6).

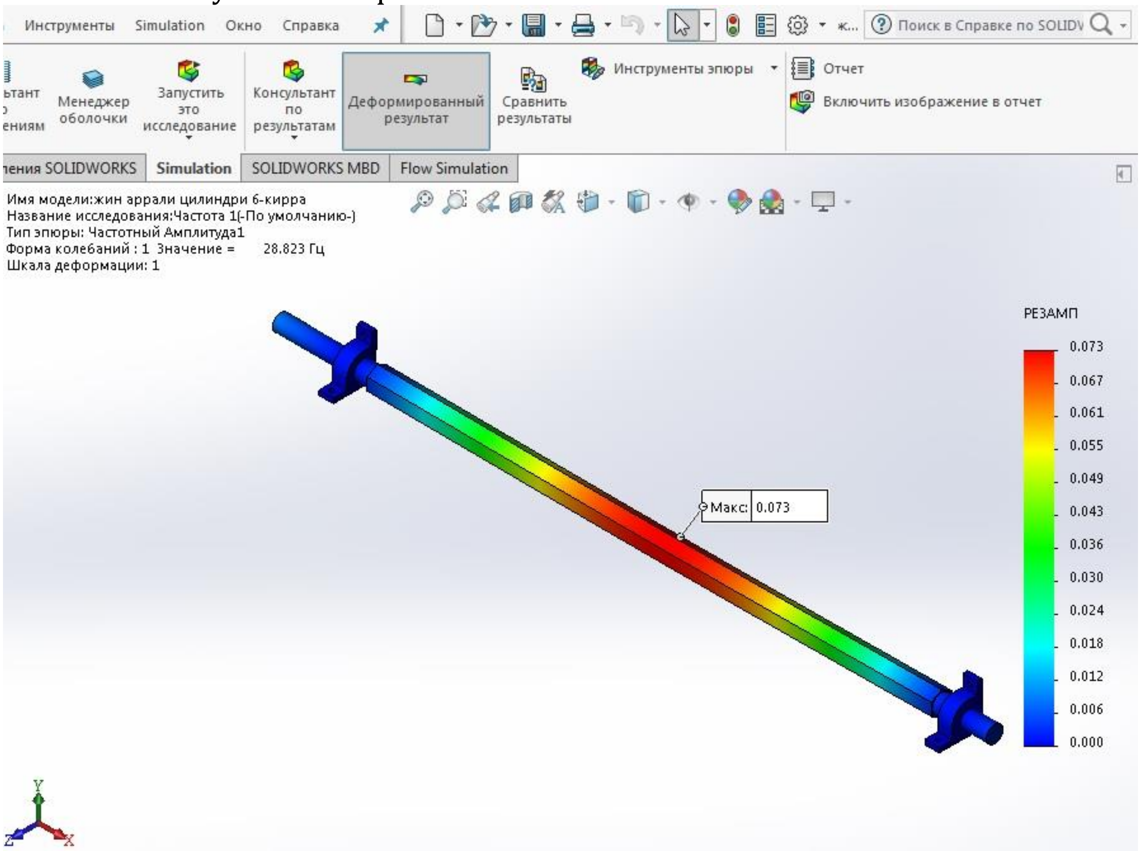
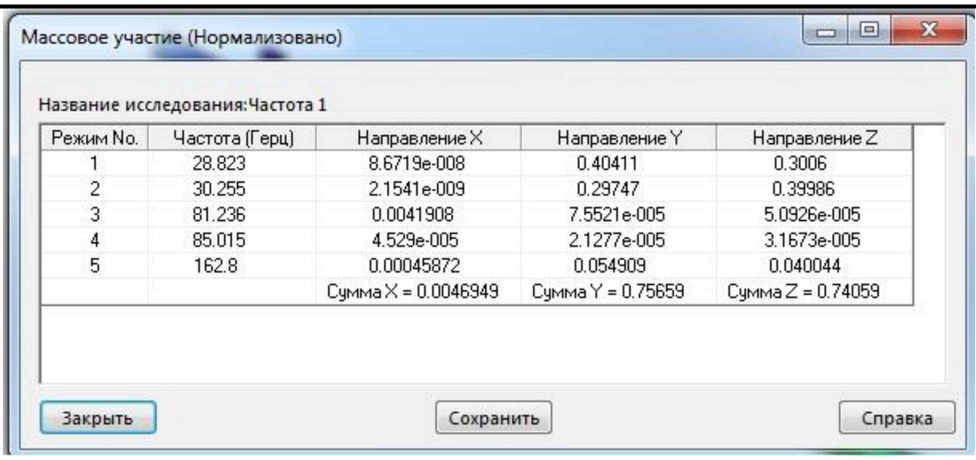


Fig. 6. First critical amplitude of the hexagonal shaft

As can be seen from the diagram, the greatest amplitude, represented in red, occurs in the middle of the shaft. When finding the natural frequency of the shaft, first of all, it is necessary to determine the total effective mass (Fig. 7).



Режим No.	Частота (Герц)	Направление X	Направление Y	Направление Z
1	28.823	8.6719e-008	0.40411	0.3006
2	30.255	2.1541e-009	0.29747	0.39986
3	81.236	0.0041908	7.5521e-005	5.0926e-005
4	85.015	4.529e-005	2.1277e-005	3.1673e-005
5	162.8	0.00045872	0.054909	0.040044
		Сумма X = 0.0046949	Сумма Y = 0.75659	Сумма Z = 0.74059

Fig. 7. Total effective mass of the hexagonal shaft during vibration

As can be seen from the table (Fig. 7), the first critical frequency was 28.823 Hz, where a maximum of 40.4% of the shaft mass along the Y axis is involved. To convert to rotational speed, we use equality $1\,Hz = 60\,rpm$. As a result, the first critical shaft speed was $28{,}823\ Hz = 60 \cdot 28{,}823 = 1729{,}38$ rpm.

From condition $n_{1кр} \geq 1{,}3 n_{иш}$ we find the operating speed $n_{иш}$: $n_{1кр}/1{,}3 \geq n_{иш}$; $n_{1кр}/1{,}3 \geq n_{иш}$; $1729{,}38/1{,}3 \geq n_{иш}$; $1330\ rpm \geq n_{иш}$.

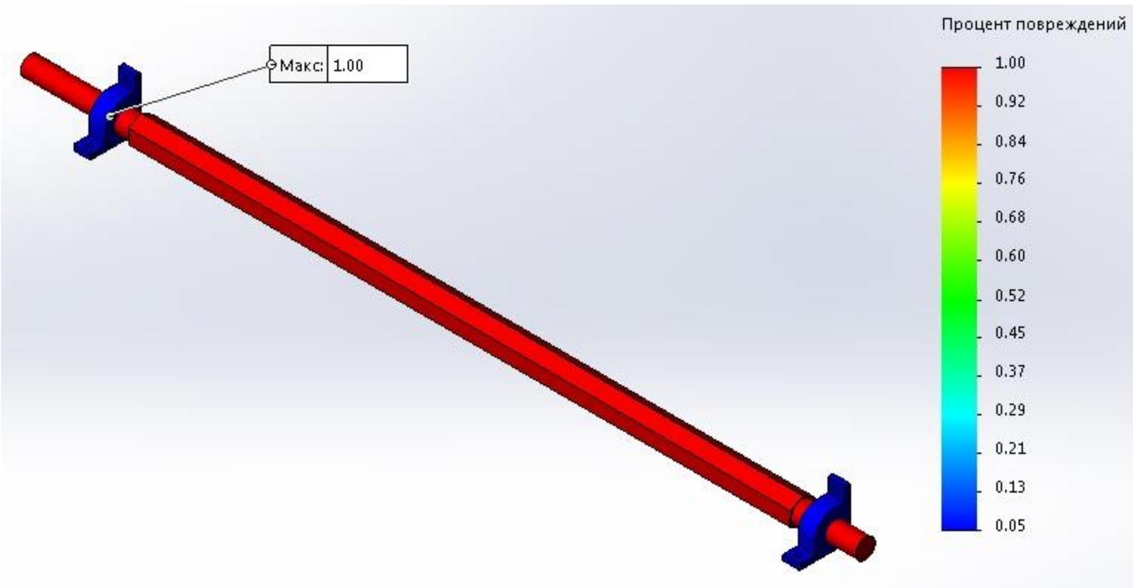
As a result, we can conclude that the hexagonal shaft of the saw gin at the above loads can rotate up to 1330 rpm, which is significantly higher than the existing speed.

Fatigue analysis of saw cylinder hexagonal shaft. Material fatigue is the gradual deterioration of the mechanical properties of a material under the influence of repeated (cyclic) stresses with alternating directions. At the same time, microcracks appear and develop, which eventually leads to destruction. This failure is called fatigue failure.

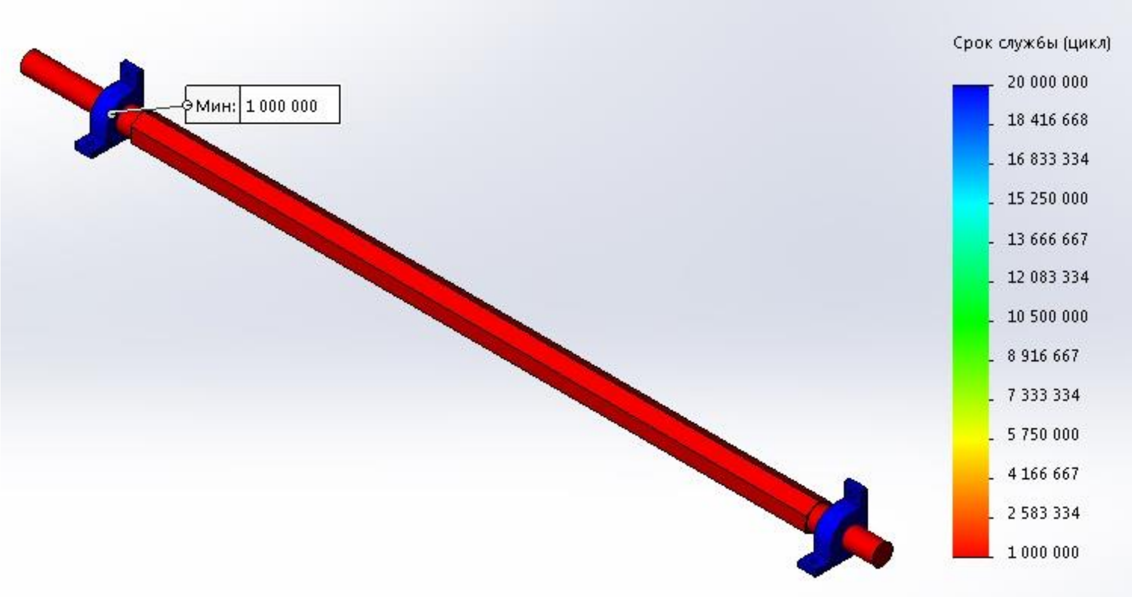
For fatigue analysis of a hexagonal shaft, we use the SolidWorks program. Fatigue analysis is carried out after the static analysis of the shaft. Therefore, we will use the results of the static calculation given above (Fig. 4).

Enter the number of cycles – 10000, the second condition – “Completely reversed”. We perform the necessary operations and get the following results.

The maximum “Percentage of damage” was 1.0, and is located in the area indicated on the diagram (Fig. 8, a). The diagram “Service life” shows the minimum cycle, which amounted to 1 million cycles (Fig. 8, b).



a)



b)

Fig. 8. Diagrams of the percentage of damage (a) and service life (b) of the hexagonal shaft

Determination of the equilibrium of a saw cylinder with a hexagonal shaft. When designing a saw cylinder shaft, its position must be statically and dynamically balanced. As a result of inaccurate manufacturing and assembly, as well as the heterogeneity of materials, in most cases the saw cylinder is unbalanced. This causes the occurrence of additional negative loads with increasing mass and rotation speed, as well as vibrations, leading to premature breakdowns of the machine.

To fully balance the saw cylinder, it is necessary to ensure the condition of static balance, i.e. the centre of gravity of the saw cylinder must be on the axis of its rotation, and the condition of dynamic balance, i.e. The axis of rotation of the shaft must coincide with the central axis of its inertia.

In Figure 9 shows a diagram of a saw cylinder, which on one side has an unbalanced mass $G_н/g$, which causes an unbalanced centrifugal force during rotation

$$C_н = \frac{G_н}{g} r_н \omega^2$$

Here: $G_н/g$ – unbalanced mass;

$r_н$ – radius of rotation of the unbalanced mass;

ω – angular velocity of the saw cylinder.

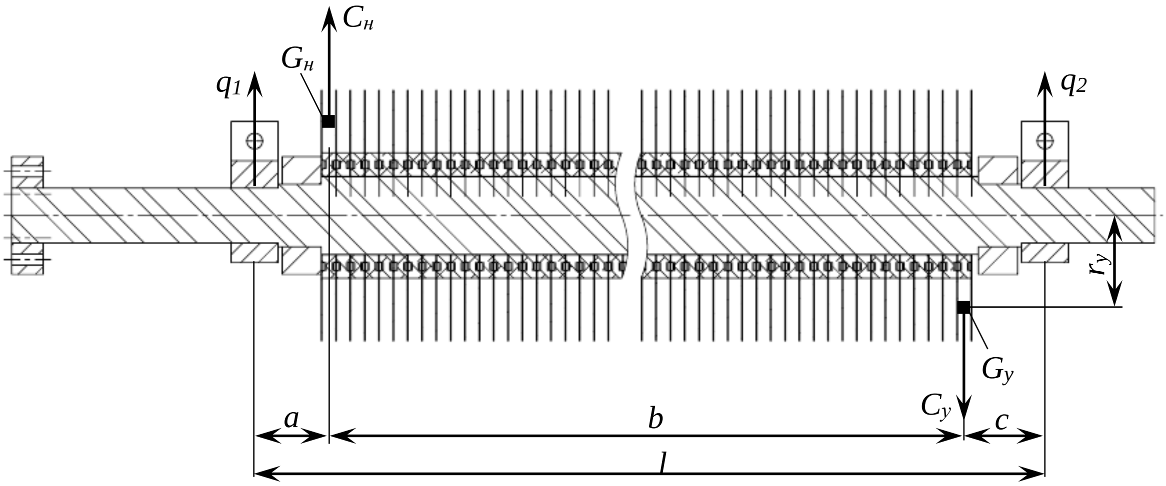


Fig. 9. Diagram of an unbalanced saw cylinder

Unbalanced centrifugal force causes additional harmful loads on bearings

$$\left.\begin{aligned} q_1 &= C_{\text{н}} \frac{b+c}{l}; \\ q_2 &= C_{\text{н}} \frac{a}{l}; \end{aligned}\right\} \tag{4}$$

$$q_1 + q_2 = C_{\text{н}}. \tag{5}$$

By attaching a balancing weight G_y to the other end at a distance r_y from the axis of rotation, which is selected so that the centre of gravity of the saw cylinder coincides with the axis of its rotation, i.e. the condition of static balancing of the saw cylinder is met. In this case, the moments of the loads $G_{\text{н}}$ and G_y relative to the axis of rotation are equal to each other, and we can write

$$G_{\text{н}} r_{\text{н}} = G_y r_y, \tag{6}$$

from here

$$G_y = \frac{G_{\text{н}} r_{\text{н}}}{r_y}. \tag{7}$$

If such a saw cylinder is rotated on supports that have an insignificant coefficient of friction, then it will stop in any position.

But even when the condition of static balancing is met, when the saw cylinder rotates, a pair of unbalanced centrifugal forces arise from loads $G_{\text{н}}$ and G_y, which causes additional harmful loads on the bearings of the saw cylinder, determined by the formula

$$q = C_{\text{н}} \frac{b}{l}. \tag{8}$$

Of the variety of imbalances in saw cylinders, three typical cases can be distinguished:

1. The center of gravity of the saw cylinder is at some distance from the axis of rotation. This imbalance is called static and can be detected by a static test when the drum is mounted on strictly horizontal supports.

2. The center of gravity is on the axis of rotation and the saw cylinder is statically balanced, but during rotation, a pair of centrifugal forces arises, which rotates with the drum and causes vibrations of the supports. Such imbalance is called dynamic since the position and magnitude of the unbalanced masses can only be determined when the drum rotates.

3. There is static and dynamic imbalance. Such imbalance is called mixed. For saw cylinders with mixed imbalance in a static state, only the static part of the imbalance can be identified and eliminated.

To study this issue, we will use the SolidWorks program. Using the “Mass characteristics” function, we obtain data on the mass, center of gravity and moment of inertia of the existing (Fig. 10) and proposed (Fig. 11) saw cylinder (Table 2).

From Table 2 we can see that the center of gravity of the proposed design of the hexagonal shaft of the saw cylinder is shifted to the axis of its rotation, due to the removal of the groove for installing the saw blades. As a result, the main moments of inertia ($kg \cdot m^2$) are reduced from P_{x1}=1.56; P_{y1}=207.24; P_{z1}=207.26 to P_{x2}=1.51; P_{y2}=197.12; P_{z2}=197.14.

The moments of inertia ($kg \cdot m^2$), determined at the center of gravity and aligned with the output coordinate system, are shifted accordingly with L_{xx1}=1.56; L_{yy1}=207.25; L_{zz1}=207.25 in L_{xx2}=1.51; L_{yy2}=197.13; L_{zz2}=197.13.

Table 2: Mass characteristics of saw cylinders with round and hexagonal shaft sections

Round shaft	**Hex Shaft**
Weight=319.82 kg Volume=0.06 m^3 Center of gravity: (m) X=-1,21 Y=0,07 Z=0,07	Weight=297.52 kg Volume=0.05 m^3 Center of gravity: (m) X=-1,21 Y=0,08 Z=0,07
Main axes of inertia and main moments of inertia: ($kg \cdot m^2$) Center of gravity	

I_x=(1.00; 0.00; 0.00) P_x=1.56 I_y=(0.00; 0.67; -0.74) P_y=207.24 I_z=(0.00; 0.74; 0.67) P_z=207.26	I_x=(1.00; 0.00; 0.00) P_x=1.51 I_y=(0.00; 0.72; -0.70) P_y=197.12 I_z=(0.00; 0.70; 0.72) P_z=197.14
Moments of inertia: ($kg \cdot m^2$) Defined at the center of gravity and aligned with the output coordinate system.	
L_{xx}=1.56 L_{xy}=0.00 L_{xz}=0.00 L_{yx}=0.00 L_{yy}=207.25 L_{yz}=-0.01 L_{zx}=0.00 L_{zy}=-0.01 L_{zz}=207.25	L_{xx}=1.51 L_{xy}=0.00 L_{xz}=0.00 L_{yx}=0.00 L_{yy}=197.13 L_{yz}=-0.01 L_{zx}=0.00 L_{zy}=-0.01 L_{zz}=197.13
Moments of inertia: ($kg \cdot m^2$) Calculated using the output coordinate system.	
I_{xx}=5.04 I_{xy}=-28.80 I_{xz}=-28.22 I_{yx}=-28.80 I_{yy}=676.23 I_{yz}=1.73 I_{zx}=-28.22 I_{zy}=1.73 I_{zz}=676.30	I_{xx}=4.80 I_{xy}=-27.25 I_{xz}=-26.41 I_{yx}=-27.25 I_{yy}=636.31 I_{yz}=1.64 I_{zx}=-26.41 I_{zy}=1.64 I_{zz}=636.41

Fig. 10. Calculation of balancing the shaft of a circular saw cylinder

Fig. 11. Calculation of balancing the shaft of a saw cylinder with a hexagonal cross-section

The moment of inertia ($kg \cdot m^2$) calculated using the output coordinate system decreases accordingly with I_{xx1}=5.04; I_{yy1}=676.23; I_{zz1}=676.30 to I_{xx2}=4.80; I_{yy2}=636.31; I_{zz2}=636.41, which shows the static balance of the saw cylinder.

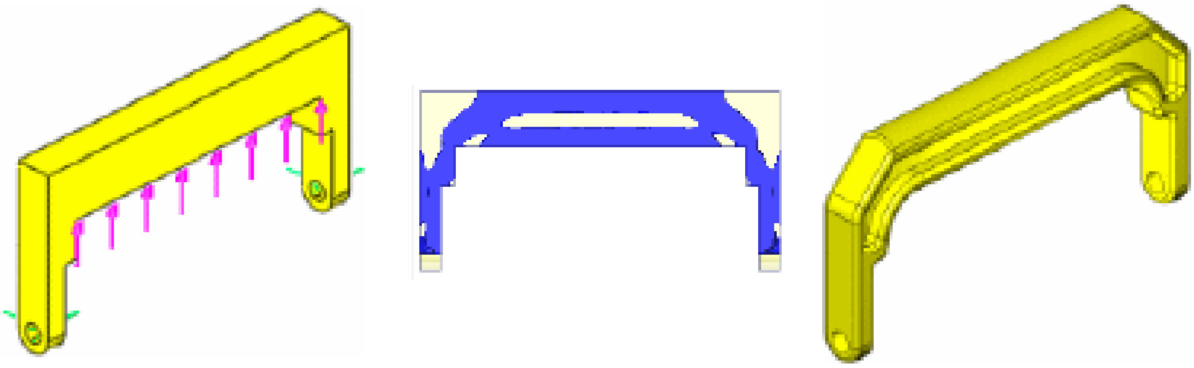
Analysis of a hex shaft using the SolidWorks – Design Insight manager. There are many known advantages of the SolidWorks program in engineering calculations, among which Design Insight has a special place.

The Design Insight plot shows the areas of the model that carry loads most efficiently. Some users may recognize in this diagram the diagram of the “trajectories of loads”. You can use this information to reduce the amount of material in your model. [9]

To use this manager, first of all, you need to conduct a static analysis of the part in SolidWorks. When the slider is set to "Most Loaded", the areas of the model that carry the most loads are drawn in blue. Usually, this is only a small section of the model. The semitransparent areas of the plot indicate the boundary of the original model.

Design Insight diagrams do not suggest where to add material. You can make inferences from the actual information provided by the plots, but that is not the purpose. These plots are most effective when you analyze the model that makes the most sense and remove material to optimize volume.

The following example highlights the benefits of Design Insight plots. (Fig. 12).



The original model shows the loads (pink) and constraints (green)

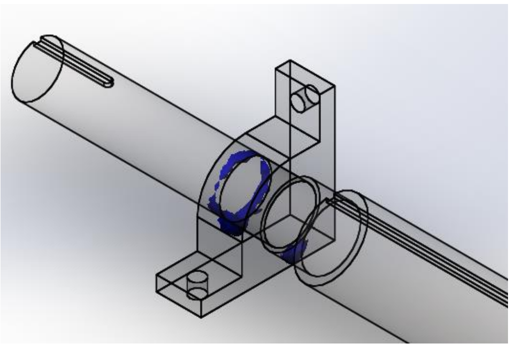
Design Insight plot with continuous path between loads and constraints

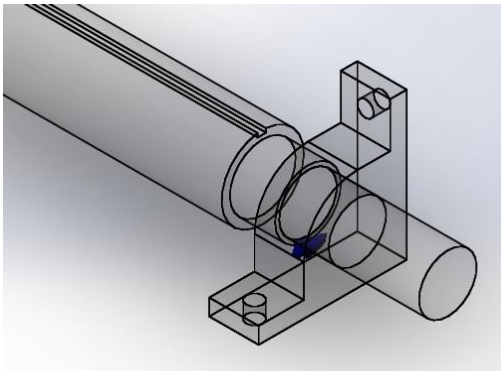
Model after material has been manually removed by the designer in areas that do not bear much load

Fig. 12. Design Insight diagram

In the same order, using the Design Insight manager, we will study the limitations of round and hexagonal shafts.

When analyzing a circular shaft, the following was obtained. First, stress began in the area of the shaft under the bearings (blue color). Secondly, by changing the position of the slider, we can see unevenly distributed stresses (Fig. 13)





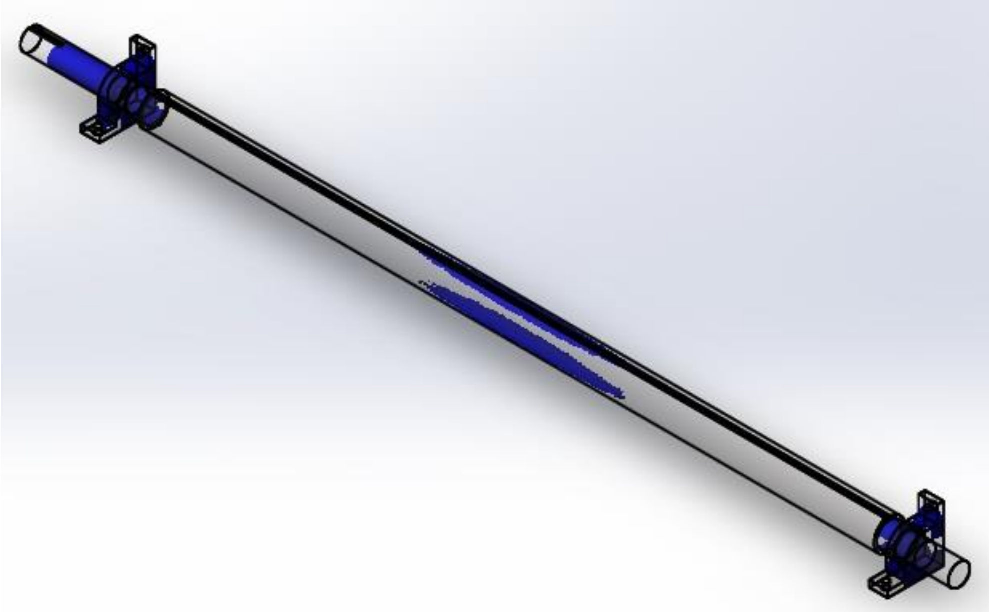


Fig. 13. Design Insight diagram of a round shaft

And for a hex shaft, stresses begin on the shaft under the left bearing. By changing the position of the slider, we can see evenly distributed stress in the middle of the shaft (Fig. 14).

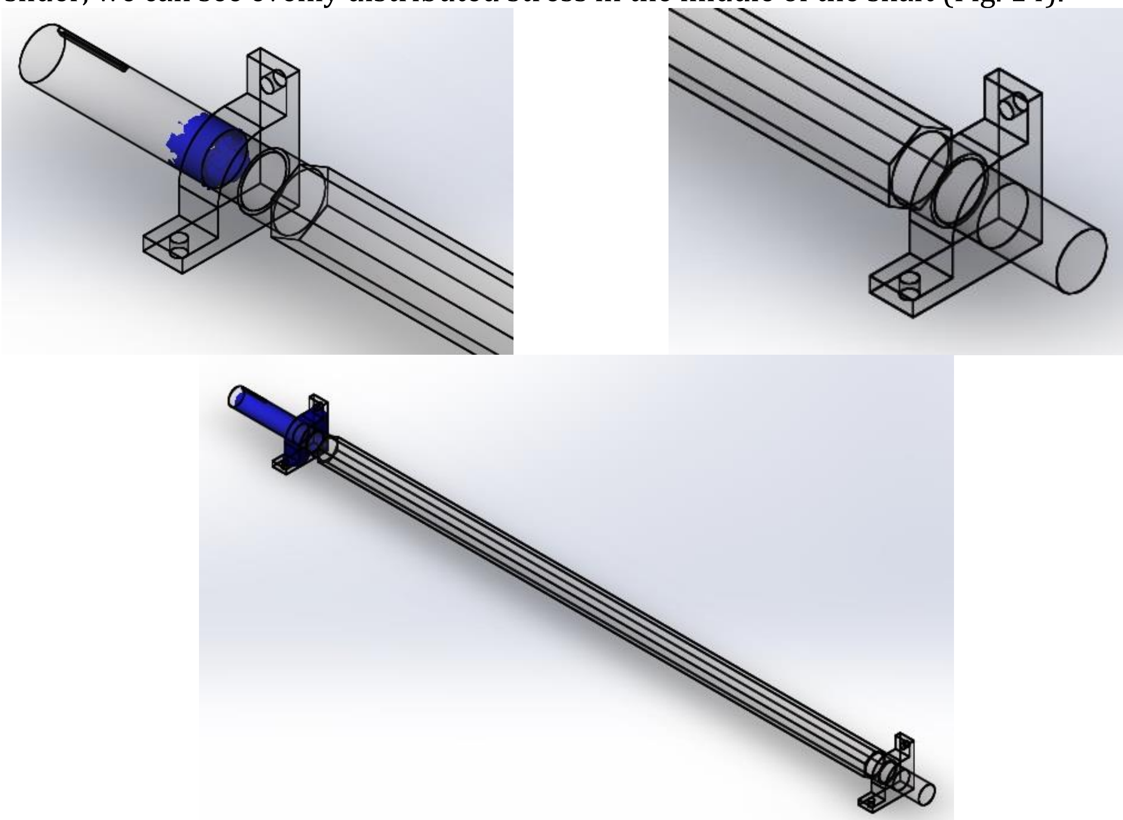


Fig. 14. Design Insight diagram of a hex shaft

By applying the obtained data, it is possible to optimize the geometry of the saw cylinder shaft of the saw gin. This makes it possible to lighten the weight of the saw cylinder, reduce energy consumption, reduce deflection and vibration of the shaft, etc.

Static calculation of the saw blade of the saw blade. The saw blade is one of the important working parts of the saw cylinder of the saw gin. The main dimensions of saw blades are shown in Figure 15. [10]

To ensure joint rotation of the saws with the shaft, there is a groove on the shaft, and a corresponding protrusion on the saw blades. In order to increase the rigidity of the saw cylinder, saw blades and inter-saw spacers are clamped on the shaft with nuts. The rigidity of the saw cylinder depends on the axial tightening force of the saws, equal to $A_0 = 20 \cdot 10^3\ N$. As is known from research [11], at the moment the fibers are separated from the seeds in contact with the seed roll, 1/4 of the number of saw blade teeth is located in the roll box and pressure acts on these teeth, equal to $P = 20\ MPa$.

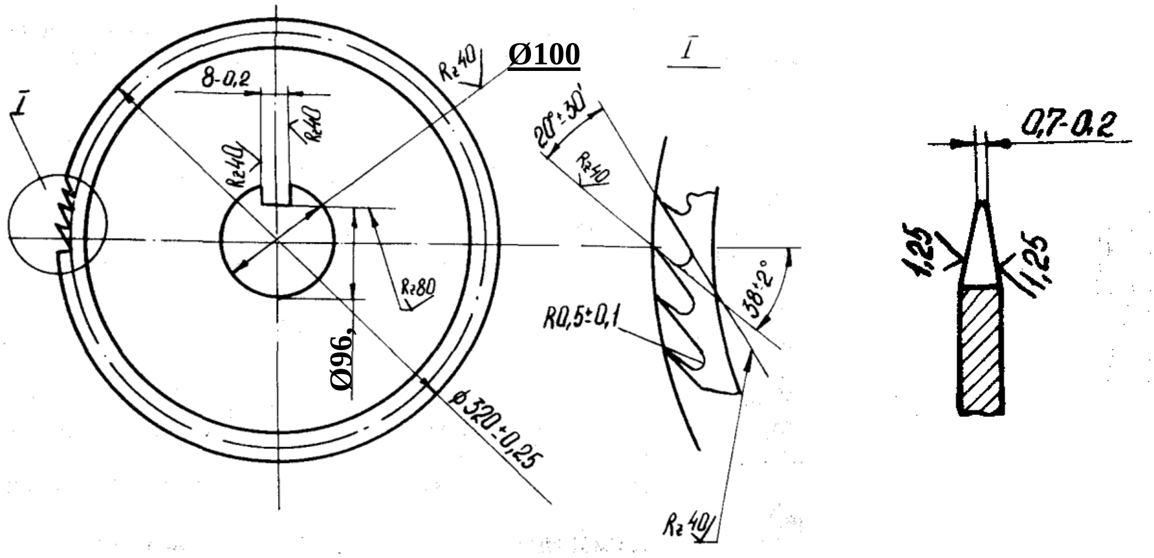


Fig. 15. Saw blade of saw gin stand

In order to increase the rigidity of the saw cylinder, a hexagonal shaft was proposed instead of a round shaft. To determine the mechanical characteristics of saw blades for round and hexagonal shafts, the following studies were carried out.

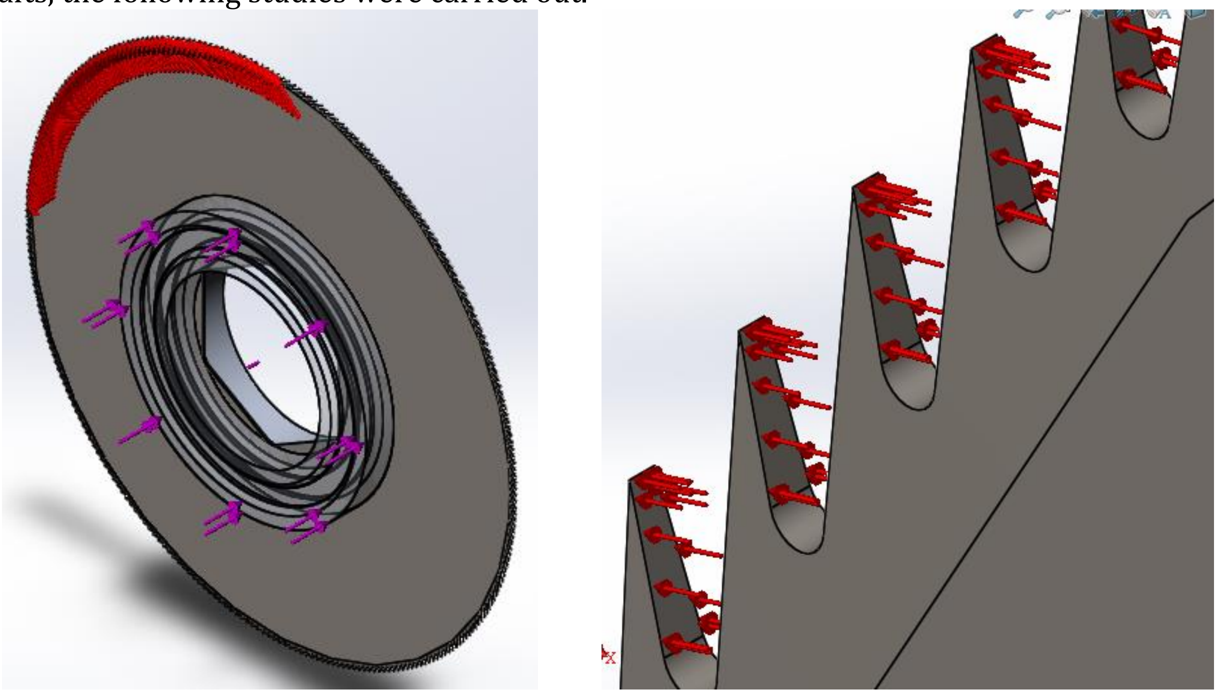
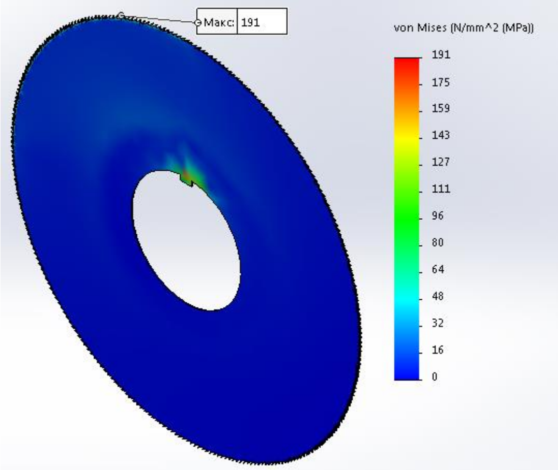
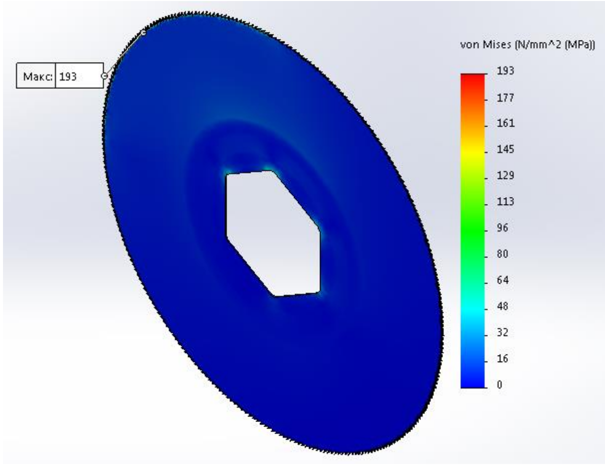
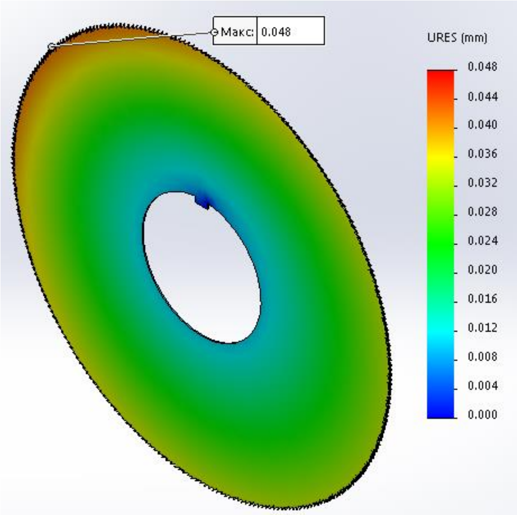
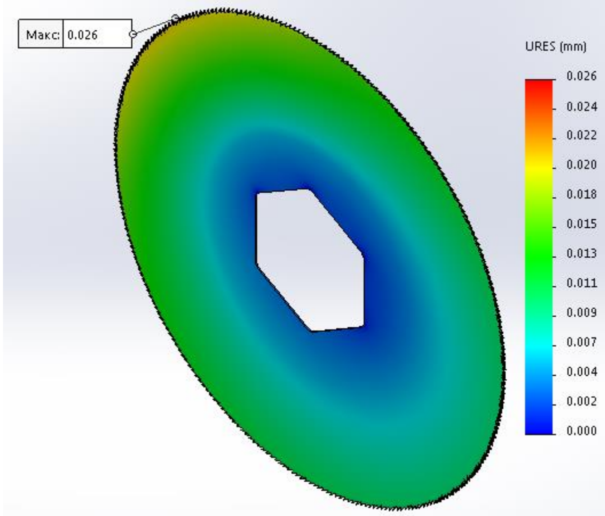
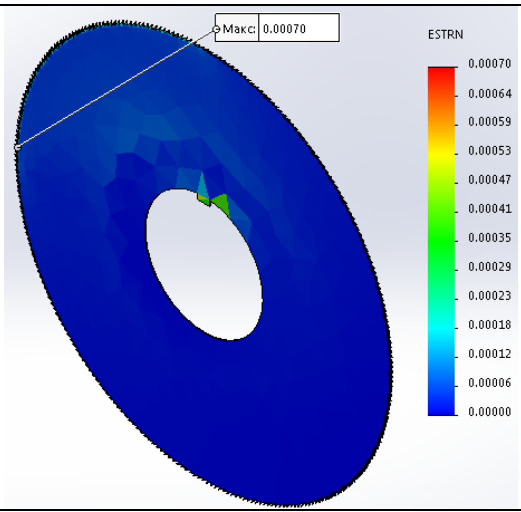
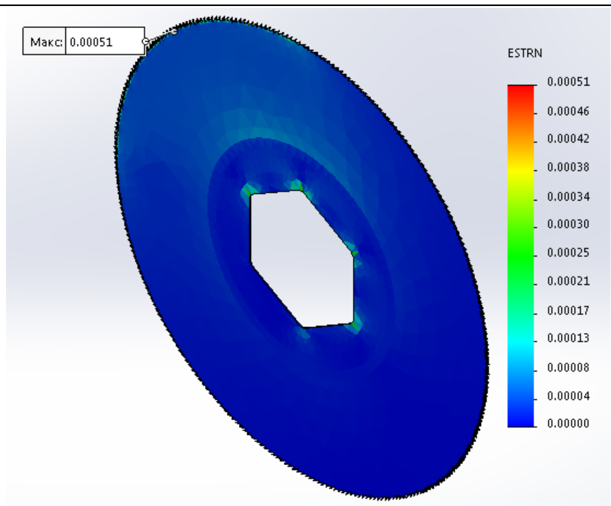
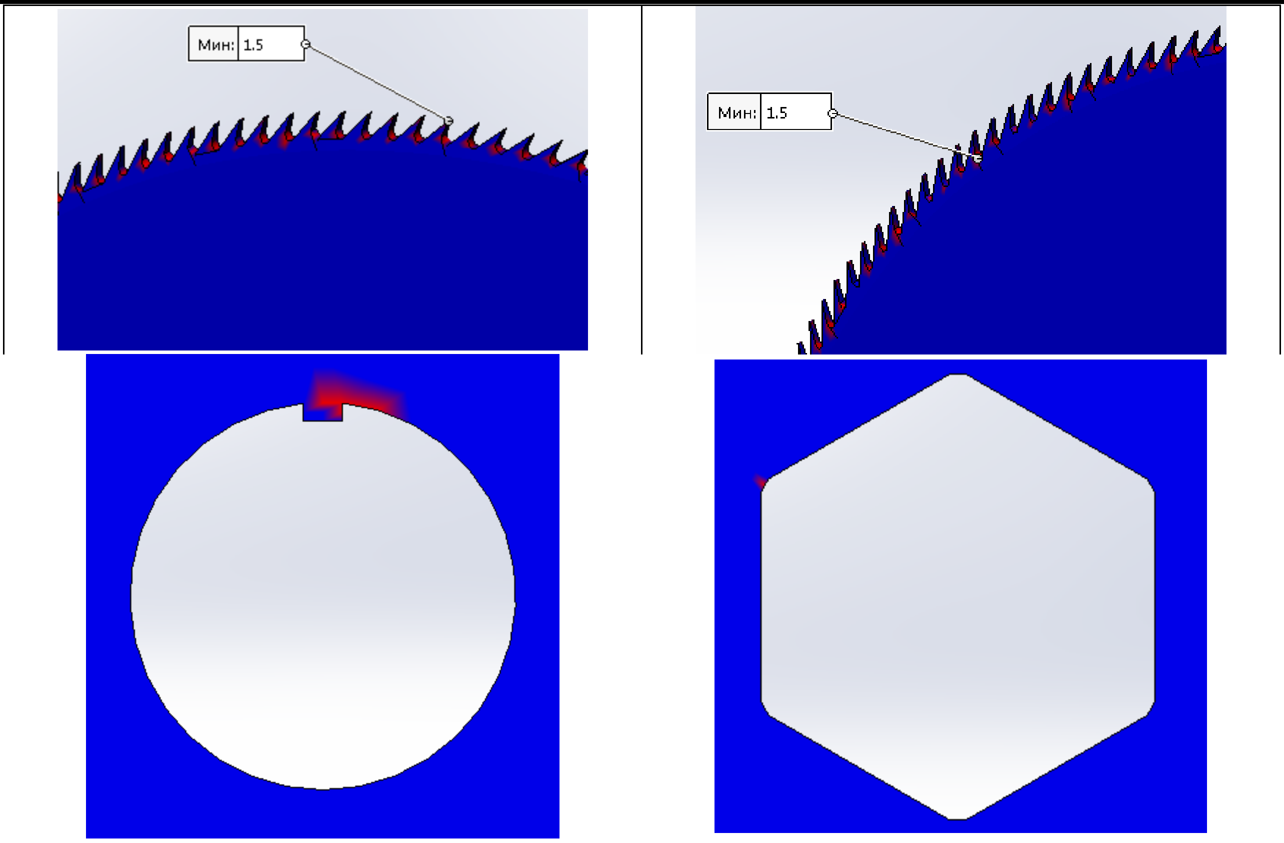


Fig. 16. Forces acting on the saw blade

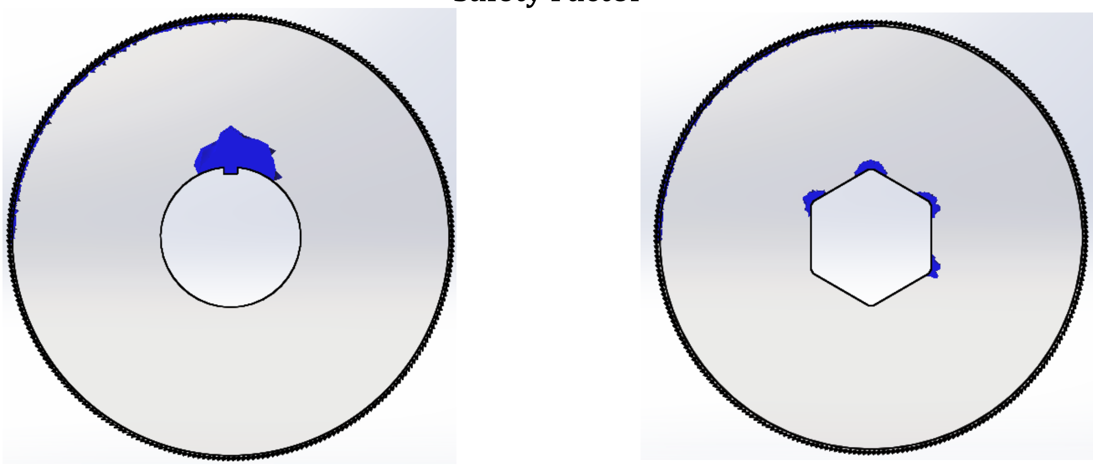
Using the data given above, we will conduct a static analysis of the saw blade during ginning. To do this, we use the Simulation package of the SolidWorks program.

Table 3: Mechanical characteristics of saw blades for round and hexagonal shafts

Round	Hexagonal
	
Stress	
	
Displacement	
	
Deformation	



Safety Factor



The area where destruction begins

First of all, we will 3D design the saw blade and the inter-saw spacer according to dimensions, and also apply the loads given above (Fig. 16). The results in the form of diagrams are shown in Table 3.

The table shows that when comparing saw blades for shafts with round and hexagonal cross-sections, the maximum stress was 191 MPa and 193 MPa, the maximum displacement was 0.048 mm and 0.026 mm, the maximum deformation was 0.00070 and 0.00051, respectively, and the minimum Safety Factor is 1.5, which occurs on the saw teeth.

The greatest stress in the area of contact of the saw blades with the shaft (red area) and the area of the beginning of destruction (blue area) for a round shaft is concentrated in the protrusion of the saw blade, and for a hexagonal shaft this area is dispersed in several places.

Using the data obtained, we can conclude that some mechanical characteristics of saw blades for a hexagonal shaft are superior to those of blades for a round shaft.

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