DOI: 10.65322/j.srd.05192026.58

DEVELOPMENT, STATIC ANALYSIS, AND PRACTICAL IMPLEMENTATION OF A RESOURCE-EFFICIENT LINTER SHAFT DESIGN

Abdurasulov A., *PhD,*
Faculty of Information Technology and Artificial Intelligence,
E-mail: ahror19982026@gmail.com,
Namangan State Technical University

Abstract*. Increasing energy efficiency in the cotton industry is one of the pressing issues. This article presents the results of research aimed at reducing energy consumption during the operation of a cotton 5LP linter machine. The issues of creating and refining methods for calculating high-performance machine drives and shafts are of particular importance in creating machines for the cotton ginning industry. The article shows that the task of reducing the mass of the main working body of the linter, the sawn cylinder shaft, was performed by grooving and slotting, and the design features of the resource-saving shaft are determined.*

Annotatsiya. *Paxta sanoatida energiya samaradorligini oshirish dolzarb masalalardan biridir. Ushbu maqolada 5LP rusumli paxta linteri mashinasining ishlashi davomida energiya sarfini kamaytirishga qaratilgan tadqiqot natijalari keltirilgan. Paxta tozalash sanoati uchun mashinalar yaratishda yuqori unumdorlikka ega mashina yuritmalari va vallarini hisoblash usullarini ishlab chiqish hamda takomillashtirish masalalari alohida ahamiyatga ega. Maqolada linter mashinasining asosiy ishchi organi – arrali silindr valining massasini ariqchalar va o‘yiqlar hosil qilish orqali kamaytirish masalasi ko‘rib chiqilgan hamda resurstejamkor valning konstruktiv xususiyatlari aniqlangan.*

Аннотация. *Повышение энергоэффективности в хлопковой промышленности является одной из актуальных задач. В данной статье представлены результаты исследований, направленных на снижение энергопотребления при эксплуатации хлопкового линтера марки 5ЛП. Особое значение при создании машин для хлопкоочистительной промышленности имеют разработка и совершенствование методов расчёта высокопроизводительных машинных приводов и валов. В статье рассматривается вопрос снижения массы вала пильного цилиндра – основного рабочего органа линтерной машины – путём выполнения продольных канавок и выемок. Определены конструктивные особенности ресурсосберегающего вала.*

Keywords: *Cotton, deformation, sawn cylinder, shaft, bending, layer, displacement, support, circle*

Kalit so‘zlar: *Paxta, deformatsiya, arrali silindr, val, egilish, qatlam, siljish, tayanch, aylana.*

Ключевые слова: *хлопок, деформация, пильный цилиндр, вал, изгиб, слой, смещение, опора, окружность.*

Introduction. The main drawbacks of the 5LP linter machine include high electricity consumption during the lintering process, rapid dulling of the saw teeth, increased seed damage when certain operations are stopped to reduce energy consumption, and low productivity in terms of seed and lint output.

The material composition of the shafts was studied. The St45 steel contains 0.42–0.50% carbon, 0.50–0.80% manganese, 0.17–0.37% silicon, up to 0.04% sulfur, and up to 0.035% phosphorus. This steel possesses high strength and ductility and provides good resistance to impact loads. It is used in the manufacture of heavily loaded components, as well as shafts, rotors, and turbines operating under conditions of significant variable loading.

DOI: 10.65322/j.srd.05192026.58

Considering the shaft's resistance to bending during operation, its calculations must be performed based on accurate and reliable data. Figure 1 illustrates the shaft design incorporating longitudinal grooves, and the calculations were carried out using these same design parameters.

When the stiffness of the saw cylinder shaft of the 5LP linter machine is insufficient, excessive shaft deflection may occur, adversely affecting the lintering process. In particular, the fiber separation process occurring in the clearances between the grid bars may be disrupted.

In this study, the bending and torsional strength of the saw cylinder shaft of the linter machine is calculated using the methodology presented in Reference [1-3].

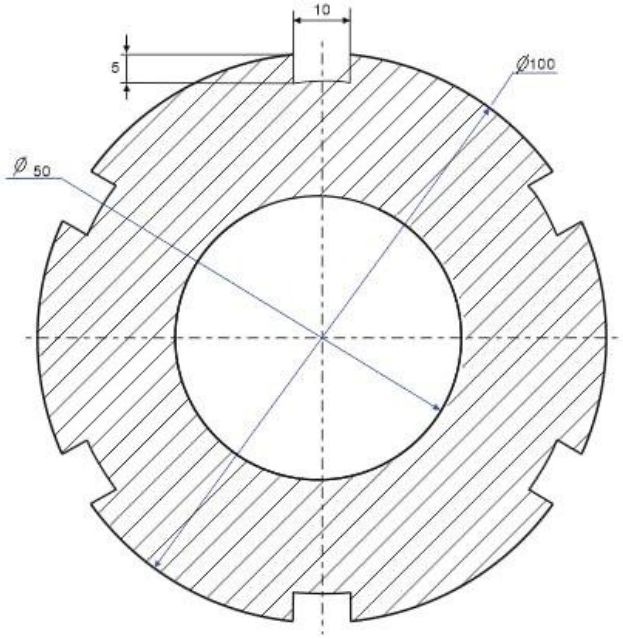

Figure 1. Cross-sectional diagram of the improved linter shaft

The section modulus of the shaft is determined using the following formula:

$$\tau_{\max} = \frac{M_0^{\max}}{W_\rho} \le [\tau]$$

Where: $W_p = \frac{M_b^{\max}}{[\tau]} = \frac{2460kg \cdot sm}{1500kg \cdot sm^2} = \frac{246}{150} = 1{,}64sm^3$

Polar moment of inertia $J_p = \frac{\pi d^4}{32} = 0{,}1 \cdot 10^4 = 1000sm^4$, $G = 8 \cdot 10^5 \ kg/sm^2$.

$$\varphi = \frac{M_b^{\max} \cdot l}{G \cdot J_n} = \frac{2460 \cdot 220}{8 \cdot 10^5 \cdot 1000} = 6{,}7 \cdot 10^{-7} rad$$

Another term for the resource-efficient shaft is a hollow shaft. It is manufactured by machining the central portion of the existing linter saw cylinder shaft to create a cavity with a diameter of 50 mm. This design modification was considered one of the main objectives of the dissertation.

The research conducted in this study showed that the torsional resistance of the hollow shaft does not differ significantly from that of the existing shaft. This is because the maximum shear stress occurs at the outer surface of the shaft cross-section and decreases toward the center, becoming zero at the center of the cross-section [46].

DOI: 10.65322/j.srd.05192026.58

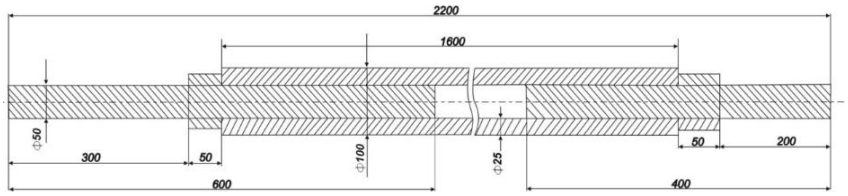

Figure 2. Dimensional design diagram of the resource-efficient shaft

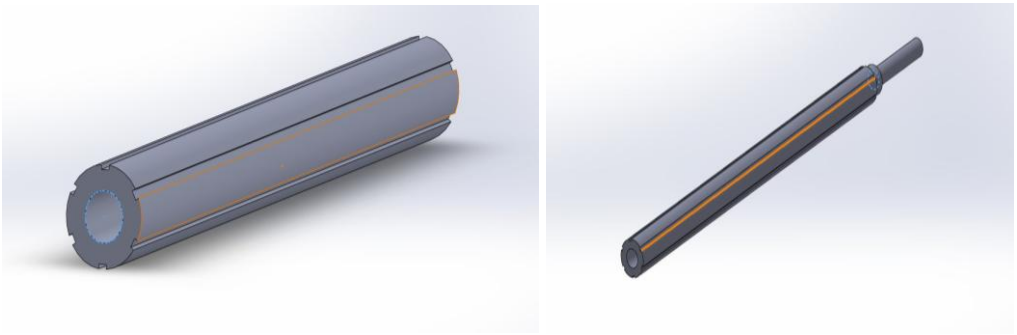

Figure 3. 3D Model of the Resource-Efficient Shaft

If the outer radius of the shaft cross-section (Figure 3) is denoted by R and the inner radius by r, the polar moment of inertia \(I_p\) can be expressed as follows:

$$I_p = 2\pi\int_r^R \rho^3 d\rho = \frac{\pi}{2}\left(R^4 - r^4\right) \tag{2.5}$$

$$I_p = \frac{\pi}{32}\left(D^4 - d^4\right) \approx 0{,}1\left(D^4 - d^4\right);$$

Here, the section modulus takes the following form:

$$W_p = \frac{I_p}{\rho_{\max}} = \frac{\pi\left(R^4 - r^4\right)}{2R} = \frac{\pi\left(D^4 - d^4\right)}{16D}$$

$$J_x = \left(\frac{\pi d^4}{64} + \frac{\pi d^2}{4}\cdot y_c\right) - \left(\frac{bt^3}{12} + bt\left(\frac{d-t}{2} + y_c\right)\right)\cdot n$$

The moment of inertia of this cross-section is (J_x). In addition, the shaft investigated in this study is analyzed as a statically determinate beam. The relevant parameters are specified as follows. First, for a solid shaft with a full cross-sectional area, the beam length is l = 3 m, and the uniformly distributed load is q = 1.08 kN/m. The analysis is performed using the example of a uniformly distributed load [46].

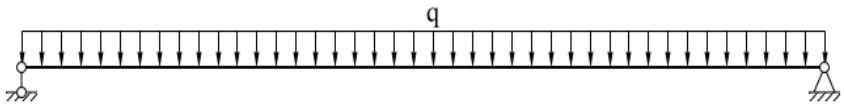

Figure 4. Uniformly Distributed Load Acting on the Shaft

We determine the reaction forces acting on the shaft and formulate the equations of static equilibrium.

$$\sum F_y = -q\cdot 3м + Y_A + Y_B = 0;$$
$$\sum M_B = q\cdot 3м\cdot 1.5м - Y_A\cdot 3м = 0.$$

Solving the static equilibrium equations (2.1) gives the following reaction force values:

DOI: 10.65322/j.srd.05192026.58

$$Y_a = 1.62\ kN;$$
$$Y_b = 1.62\ kN.$$

Based on the results obtained above, the shear force diagram is constructed.

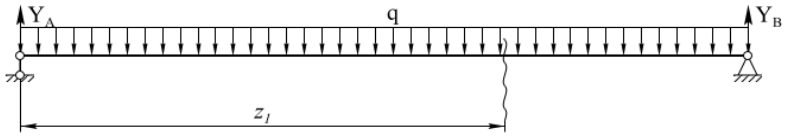

Figure 5. Directions of the Reaction Forces

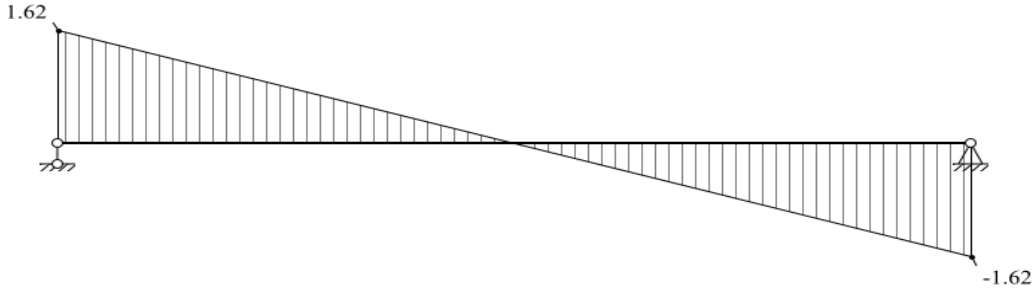

Figure 6. Shear Force Diagram

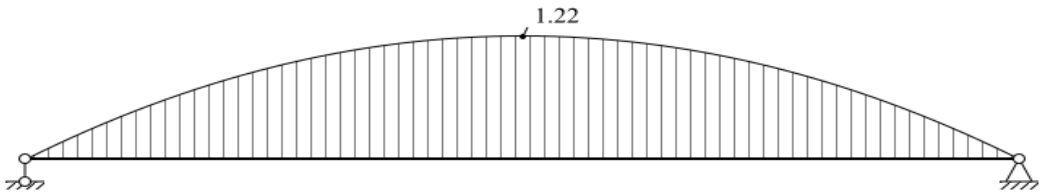

Figure 7. Bending Moment Diagram

In general, the following condition must be satisfied. ($0 \le z_1 \le 3м$), $Q_y = Y_A - q \cdot z_1$;

$$z_1 = 0;\ Q_y = 1.62kN,\ z_1 = 3m;\ Q_y = -1.62\ kN.$$
$$M_x = Y_A \cdot z_1 - q \cdot z_1^2/2;$$
$$Where:\ z_1 = 0;\ M_x = 0.$$
$$z_1 = 1.5m;\ M_x = 1.215\ kN \cdot m.$$
$$z_1 = 3m;\ M_x = 0.$$

For the proposed shaft with a hollow cross-section, the main cross-sectional area is considered as follows:

$$l = 3м,\ q = 0.88\ kN/m$$

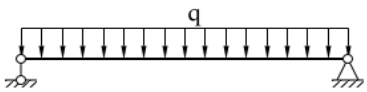

Figure 8. Uniformly Distributed Load Acting on the Proposed Hollow, Resource-Efficient Shaft

We formulate the equations of static equilibrium.

$$\sum F_y = -q \cdot 3м + Y_A + Y_B = 0;$$
$$\sum M_B = q \cdot 3м \cdot 1.5м - Y_A \cdot 3м = 0.$$

Solving the equilibrium equations yields the following reaction force values:

$$Y_A = 1.32\ kN;$$
$$Y_B = 1.32\ kN.$$

DOI: 10.65322/j.srd.05192026.58

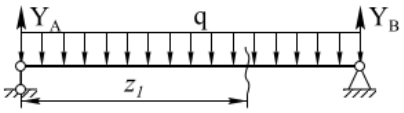

Figure 9. Directions of the Reaction Forces for the Proposed Hollow, Resource-Efficient Shaft

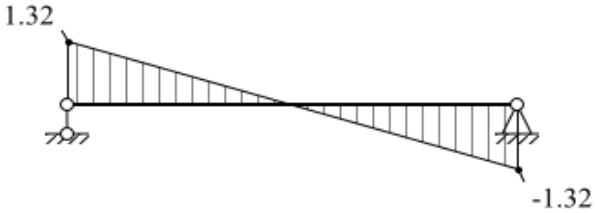

Figure 10 Shear Force Diagram for the Resource-Efficient Shaft

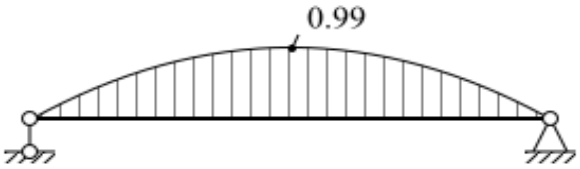

Figure 11. Bending Moment Diagram for the Resource-Efficient Shaft

In general, the following condition must be satisfied, namely: $(0 \le z_1 \le 3м), Q_y = Y_A - q \cdot z_1;$

$$z_1 = 0;\ Q_y = 1.32kN,\ z_1 = 3m;\ Q_y = -1.32kN,\ M_x = Y_A \cdot z_1 - q \cdot z_1^2/2;$$

Where $z_1 = 0;\ M_x = 0.z_1 = 1.5m;\ M_x = 0.99kN \cdot m.z_1 = 3m;\ M_x = 0.$

The results obtained by analyzing the shafts as statically determinate beams indicate that the proposed hollow shaft investigated in this study is capable of withstanding the applied forces and operating under the resulting loads during the technological process.

An oscilloscope was used to record measurement results during the measurement of the torsional moment. This made it possible to reliably monitor the dynamic loads arising during the experiment and the variation of the torsional moment over time (Figure 3.6). To facilitate the analysis of the recorded oscillograms and their interpretation in graphical form, the recording speed on the oscilloscope tape was set to 50 mm/s. The recording was performed in the second-speed mode, ensuring that the signals were displayed clearly, distinctly, and in a readily distinguishable form on the oscillogram. As a result, it became possible to analyze the variation characteristics of the torsional moment in detail and draw reliable conclusions.



Figure 12. Experimental Research Process

DOI: 10.65322/j.srd.05192026.58

Practical experiments conducted to test the resource-efficient shaft were carried out in the mechanical workshop of the To‘raqo‘rg‘on cotton-ginning enterprise and on the 5LP linter machine operating in the technological process (Figure 3.6). As noted in previous studies, the main objective of the experimental investigations was to determine the principal parameters of the shaft and develop appropriate modifications to its design.

Theoretical studies established that, at the linter machine’s saw-cylinder rotational speed of 730 rpm, the existing saw-cylinder shaft undergoes significant bending and vibration, exceeding the permissible limits. Furthermore, in the central region of the shaft, the technological clearances between the saw discs and the grid bars are disrupted, increasing cottonseed damage and reducing machine productivity.

The inner diameter of the saw-cylinder shaft was varied among three values: 45 mm, 50 mm, and 55 mm. The effects of changes in the inner diameter on the shaft mass, moment of inertia, strength, and dynamic characteristics were systematically investigated. For each selected inner diameter, the stress state, degree of deformation, and vibration characteristics were analyzed using both theoretical and experimental methods. Based on the results obtained, scientifically substantiated conclusions were developed to determine the optimal inner diameter of the saw-cylinder shaft.

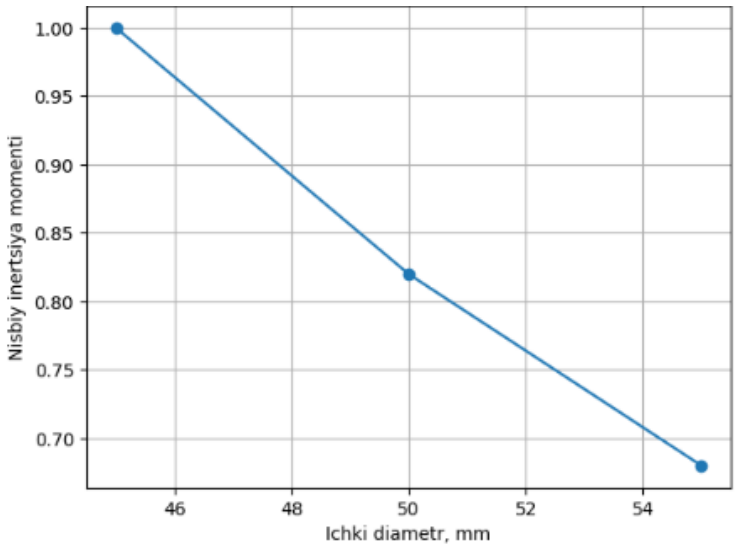

Figure 13. Variation in the Moment of Inertia as a Function of the Inner Diameter of the Saw-Cylinder Shaft

The graph in Figure 13 illustrates the variation in the moment of inertia as the inner diameter of the saw-cylinder shaft increases from 45 mm to 55 mm. The results show that the moment of inertia of the shaft decreases steadily as the inner diameter increases. This can be explained by the removal of material from the central region of the shaft cross-section, which changes both the total mass of the shaft and the distribution of mass relative to the axis of rotation.

The shaft with an inner diameter of 45 mm has the highest moment of inertia, which may result in relatively high dynamic loads at elevated rotational speeds. When the inner diameter increases to 50 mm, a significant reduction in the moment of inertia is observed, creating the potential to reduce vibration intensity during operation and lower energy consumption. When the inner diameter reaches 55 mm, the moment of inertia decreases further. However, at this point, limitations associated with the shaft's strength and stiffness may arise. Therefore, the increase in the inner diameter was evaluated comprehensively, taking into account not only dynamic performance but also the stress state and degree of deformation. Overall, the graphical analysis indicates that increasing the inner diameter of the saw-cylinder shaft within certain limits improves

DOI: 10.65322/j.srd.05192026.58

its dynamic characteristics, reduces its moment of inertia, and enhances vibration stability. The results are of considerable scientific and practical importance for selecting the optimal inner diameter.

In particular, during the maintenance and repair of the saw cylinder, it is not practically advisable to install the saw blades in the same groove every time. Continuous operation of the saw blades in a single groove causes that region to be subjected to higher stresses (Figure 14).



Figure 15. Stressed Region of the 5LP Shaft Where the Saw Blades Are Mounted

As a result, the geometric characteristics of the shaft cross-section change, leading to a non-uniform stress distribution. This may disrupt the dynamic balance of the shaft and cause an increase in vibration.

Therefore, the presence of six longitudinal grooves makes it possible to install the saw blades alternately in different grooves. This ensures a more uniform distribution of loads across the shaft cross-section, reduces stresses, and improves the long-term structural strength and operational reliability of the design. The selection of six grooves as the optimal design solution for reducing the shaft mass while maintaining the required strength, stiffness, and dynamic stability was substantiated through theoretical and experimental investigations.

Overall, optimizing the inner diameter of the saw-cylinder shaft and incorporating longitudinal grooves into its outer surface improved its structural and dynamic characteristics, reduced vibration, and enhanced operational convenience during maintenance and repair.

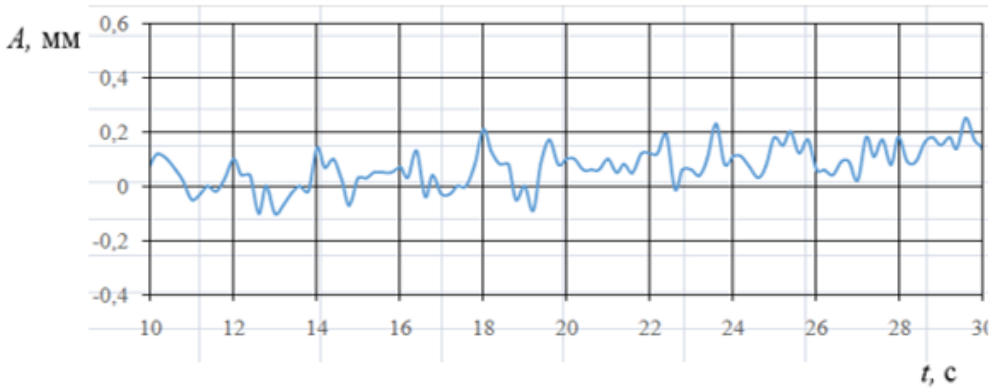

Figure 16. Maximum Vibration Amplitude Measured During the Operation of the Proposed Shaft

The vibration amplitude of the proposed shaft was measured, and the results showed that it remained relatively low, with a maximum vibration amplitude of 0.3 mm. This indicates that the shaft has sufficient dynamic stability and structural resistance under operating conditions.

DOI: 10.65322/j.srd.05192026.58

The steel shaft selected as the research object was loaded with a mass located at its center and supported by bearings at both ends. To determine the relationship between vibration amplitude and frequency, the system was considered as a linear vibration model with one degree of freedom. In this model, the mass of the shaft together with the centrally located mass was represented by the lumped mass \(m\), the shaft's bending stiffness by the stiffness coefficient \(k\), and the energy losses in the shaft and bearing mechanisms by the damping coefficient \(c\).

According to the technical specifications of the 5LP linter, the saw-cylinder shaft is connected via a half-coupling to an electric motor rated at 18.5 kW and operating at 730 rpm to drive the saw cylinder in rotational motion. During the research, it was established that the load on the electric motor increased when the saw cylinder operated both without a load and under loaded conditions. The load on the electric motor driving the saw cylinder was determined using an ammeter installed on the motor.

Conclusion. Cottonseed was fed from the 5LP linter feeder into the working chamber. The rotation of the agitator and saw cylinder caused the cottonseed to circulate, forming a seed roll. While the seed roll rotated at the required density without clogging, the load imposed on the electric motor by the saw cylinder during the process of removing lint from the seed surface was measured. The measured current was 34 A. The power required to drive the saw cylinder of the linter was 18.5 kW, resulting in considerable electrical energy consumption. At a rotational speed of 730 rpm, the weight of the cylinder contributed to shaft bending. This increased the load on the machine's electric motor and consequently raised electricity consumption.

In developing the proposed resource-efficient shaft design, the production conditions of the enterprise and other relevant resource requirements were taken into account. Structural modifications were introduced to reduce the shaft mass and optimize material consumption. Specifically, a hollow cavity was created inside the shaft, while grooves were machined into its outer surface to redistribute the material. As a result of these design modifications, the total shaft mass was reduced to 87 kg. Consequently, the proposed shaft design not only significantly reduced material consumption but also made it possible to decrease the moment of inertia of the rotating components, improve energy efficiency, and reduce dynamic loads during operation.

References

1. Miroshnichenko G.I. Fundamentals of designing machines for cotton processing. “Mashinostroeniye”, M. 1972, pp. 235-237.

2. Abdurasulov A., Obidov A., Mirzaumidov A., Otaqo‘ziyev D. Deformation of the shaft in torsion and the effect of torsion along with bending. // ISSN 2181 8622 Scientific and Technical Journal Namangan Institute of Engineering and Technology, Volume 9 Issue 2, 2024 208-216 p.

3. A.Abdurasulov, O‘.Muhammadsodiqova, A.Mirzaumidov. Creating a simplified structure of the linter saw cylinder shaft. Scientific journal of mechanics and technology, №4, 2024, 42-48 p.