ISSN 2181-8622

Manufacturing technology problems

Scientific and Technical Journal Namangan Institute of Engineering and Technology

Volume 10
Issue 1
2025

UDC 677.021

STATIC ANALYSIS OF THE SPINDLE SHAFT WITH A SPLIT CYLINDER

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Abstract: This article examines the static analysis of a split-cylinder shaft and the process of calculating its torsion angle along with its strength. Using static analysis, forces, moments, stresses, and deformations acting on the shaft are determined. The research focuses on calculating the strength of the split-cylinder shaft in initial cotton processing enterprises and ensuring that the torsion angle remains within the permissible limits. The static equilibrium process, the uniform distribution of forces, shear forces, and moment diagrams are analyzed in detail. Additionally, the article provides information about the material properties of the shaft and its performance under mechanical conditions. The study also highlights the importance of both static and dynamic equilibrium, as well as the necessary measures to improve safety and economic efficiency in technological machinery operations. This article offers practical recommendations for improving the construction of split-cylinder shafts and ensuring their effective performance in industrial sectors.

Keywords: Split-cylinder shaft, static analysis, torsion angle, elasticity modulus, moment of inertia, shear force, moment diagram, static equilibrium, dynamic equilibrium, technological process, force distribution, safety, engineering design, mechanical systems.

Introduction. The strength of a split-cylinder shaft must be high, and its rigidity is also crucial. In other words, the torsion angle along the length of the shaft must not exceed the permissible torsion angle for the material of the shaft. The split-cylinder shaft is one of the structural components with significant importance in mechanical systems. Its static analysis is applied in various fields, ranging from mechanics to the transportation industry. Static analysis allows evaluating the operating condition of the shaft, how it responds to loads, and its strength or safety under different conditions. We will consider the calculation of the strength of a split-cylinder shaft subjected to evenly distributed forces in initial cotton processing enterprises [1,2,3].

Materials and methods.

$$q = \frac{G_в + G_п + G_{мп}}{l_o};\ N/mm$$

In this $G_в = m_в \cdot g$; shaft mass, $G_п = m_п \cdot g$;saw mass, $G_{мп} = m_{мп} \cdot g$ sleeve mass

$$m_в = \rho_в \cdot V_в = \rho \cdot \left(\frac{\pi d^2}{4} \cdot l_0 \right)$$

$\Delta d = 10\,sm = 100\,mm$, $\Delta l = 3\,м = 300\,sm = 3000\,mm$, $Po`lat - 45$, $\rho = 7{,}8\,g/sm^3$,

$l_1 = 39.7sm = 397mm$, $l_2 = 237sm = 2370mm$, $l_3 = 23{,}3sm = 233\,mm$

$d_1 = 7sm = 70mm$, $d_2 = 10sm = 100mm$, $d_3 = 7sm = 70\,mm$;

$$m_{в1} = \rho \cdot \frac{\pi d_1}{4} \cdot l_1 = 7.8\left(\frac{г}{см^3}\right) \cdot \frac{3{,}14 \cdot 7^2}{4} \cdot 39.7 = 11911\,гр \approx 12kg$$

$$m_{в2} = \rho \cdot \frac{\pi d_2}{4} \cdot l_2 = 7.8\left(\frac{г}{см^3}\right) \cdot \frac{3{,}14 \cdot 10^2}{4} \cdot 237 = 145115{,}1gr \approx 145kg$$

$$m_{в3} = \rho \cdot \frac{\pi d_3}{4} \cdot l_3 = 7.8\left(\frac{г}{см^3}\right) \cdot \frac{3{,}14 \cdot 7^2}{4} \cdot 23{,}3 = 6990.6gr \approx 7kg$$

$$\sum m_в = m_{в1} + m_{в2} + m_{в3} = 12 + 145 + 7 = 164kg$$

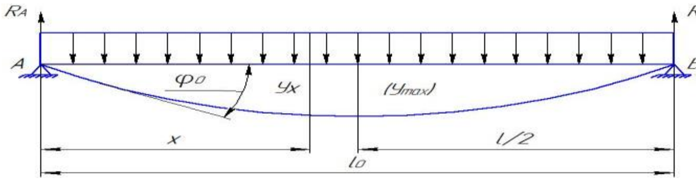
The mass of one saw $m_n = 0.57\ kg$, in total $m_a \cdot 130 = 0.57 \cdot 130 = 74.1 \approx 74\ kg$ sleeve mass

$m_q = 0.214\,kg$, in total $m_{мп} \cdot 129 = 0.214 \cdot 129 = 27.6 \approx 28\ kg$

$$\frac{P}{l} = q = \frac{g(m_v + m_a + m_q)}{l_o} = \frac{10(164 + 74 + 28)}{3000} = 0{,}88\,\frac{N}{mm}$$

$$q = 0.88\,\frac{N}{mm};\ P = 2660\,N$$

$$y_{\max} = \frac{5}{384} \cdot \frac{0{,}88 \cdot (3 \cdot 10^3)^4}{2 \cdot 10^5 \cdot 490.625 \cdot 10^4} = \frac{5}{384} \cdot \frac{0{,}88 \cdot 81 \cdot 10^3}{2 \cdot 490.625} = 0.94mm$$



1-figure. Deflection diagram of the saw cylinder shaft

We will input the initial parameters.

$$\varphi_o \neq 0,\ y_o = 0,\ M_o = 0,\ Q_o = \frac{q \cdot l}{2} = R_A = R_B;\ q_o = -q.$$

$$EJY_{\max} = -\frac{5}{384} \cdot ql^4;\ |y|_{\max} = \frac{5}{384} \cdot \frac{ql^4}{EJ}$$

$E = 2 \cdot 10^5\ H/мм^2$ - elasticity of steel, q=1.08 N/mm^2, l=l_o =3m=3000 mm;

$$J = \frac{\pi d^4}{64} = \frac{3.14 \cdot 10^4}{64} = 490.625\,sm^4 = 490{,}625\ \cdot 10^4\,mm^4$$

$$|y|_{\max} = \frac{5}{384} \cdot \frac{0{,}88 \cdot (3000)^4}{2 \cdot 10^5 \cdot 490.625 \cdot 10^4} = 0.94mm$$

Maximum twisting.

$$|y|_{\max} = \frac{Pl^3}{48EJ_x}\ ;\ \theta_A = \theta_B = \frac{Pl^2}{16EJ_x}$$

$$y_{\max} = \frac{2660\cdot(3\cdot 10)^3}{48\cdot 2\cdot 10^5\cdot 490.625\cdot 10^4} = \frac{2660\cdot 27}{96\cdot 490.625} = 1{,}5mm;\ y_{\max} = 1{,}52mm\ .$$

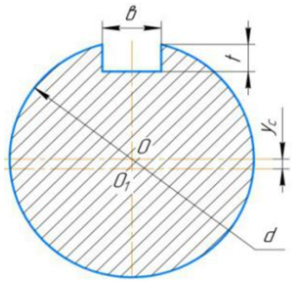
$J_o = \dfrac{\pi d^4}{64}$ - moment of inertia of the shaft.

J' - Moment of inertia at the section of the shaft where the saw is applied.

Moment of inertia calculated relative to the longitudinal central axis X_o

$$J_x = \sum\left(J_{x_o} + F_{y_c}{}^2\right)$$

Total cross-sectional area of the shaft: $F = \dfrac{\pi d^2}{4} - b\cdot t$



2-figure. Cross-sectional condition of the shaft

y_c – The distance from the central axis to the central axis of the circle;

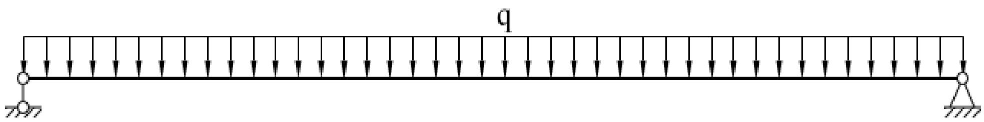
$y_c = \dfrac{S_x}{F}$, in this S_x – static moment, $S_x = -(bt)\cdot\dfrac{t}{2} = -\dfrac{bt^2}{2}$;

$$y_c = -\frac{\dfrac{bt^2}{2}}{\dfrac{\pi d^2}{4} - bt} = -\frac{bt^2}{2\left(\dfrac{\pi d^2}{4} - bt\right)}$$

We calculate the transition to parallel axes using the formula x_0 for the moment of inertia relative to the central axis x_0

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

This is the moment of inertia for the given section(J).Additionally, we will verify it within the static analysis of the beam for the shaft underinvestigation.We input the known parameters.For a shaft with a fully intact primary cross-sectional area, I=3m and q=1,08kN/m.Forces are analyzed as an example of distributed load.



3-figure. The distributed load condition of the forces acting on the shaft

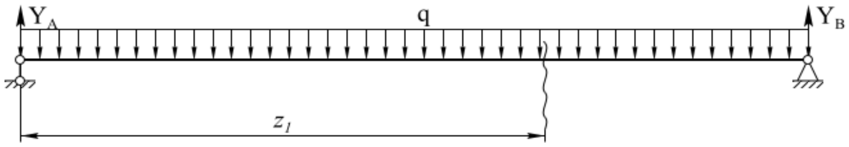
We determine the reaction forces acting on it and formulate the static equilibrium equations.

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

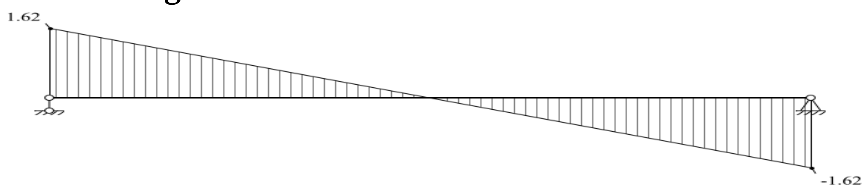
Solving the static equations (2.1) provides the following reaction values

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

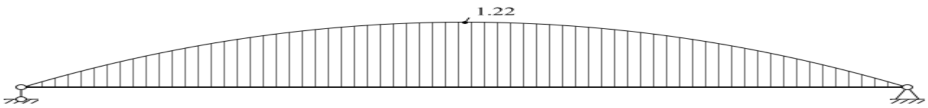
Based on the results obtained above,we contruct the shear force diagram



4-figure.The direction of the reaction forces



4-figure. Shear force diagram



5-figure. Bending moment diagram

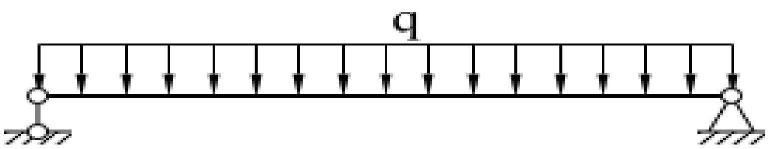
In the general case, the following condition must be satisfed ($0 \le z_1 \le 3м$), $Q_y = Y_A - q \cdot z_1$;

$$z_1 = 0;\ Q_y = 1.62kN,\ z_1 = 3m;\ Q_y = -1.62\ kN.$$

$M_x = Y_A \cdot z_1 - q \cdot z_1^2/2$; In this $z_1 = 0$; $M_x = 0$. $z_1 = 1.5m$; $M_x = 1.215$ kN·m. $z_1 = 3m$; $M_x = 0$.

For the shaft with a cross-sectional area having an internal cavity that, we propose.

$$l = 3м,\ q = 0.88\ kN/m$$



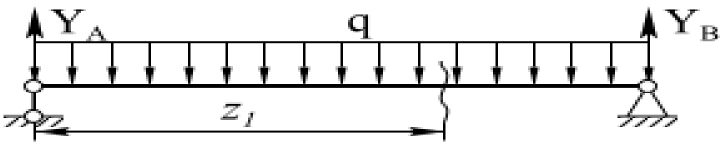
6-figure. The distributed load condition of the forces acting on the proposed grooved shaft.

We formulate the static equilibrium equations.

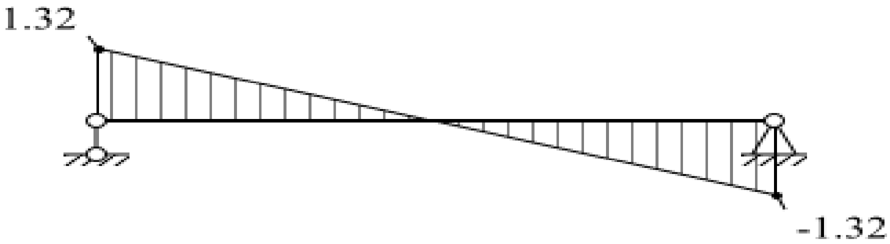
$$\sum F_y = -q \cdot 3м + Y_A + Y_B = 0;\ \ \sum M_B = q \cdot 3м \cdot 1.5м - Y_A \cdot 3м = 0.$$

Solving the static equations gives the following reaction values

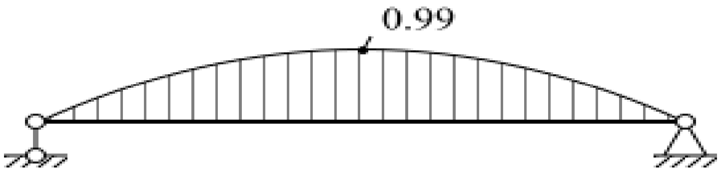
$$Y_A = 1.32\ kN;\ \ Y_B = 1.32\ kN.$$



7-figure. The direction of the reaction forces for the proposed grooved shaft



8-figure. Shear force diagram for the grooved shaft



9-figure. Bending moment diagram for the grooved shaft.

In the general case, the following condition must be satisfied, that is
$(0 \leq z_1 \leq 3м)$,$Q_y = Y_A - q \cdot z_1$; $z_1 = 0$; $Q_y = 1.32kN$, $z_1 = 3m$; $Q_y = -1.32kN$, $M_x = Y_A \cdot z_1 - q \cdot z_1^2/2$;
In this: $z_1 = 0$; $M_x = 0.z_1 = 1.5m$; $M_x = 0.99kN \cdot m.z_1 = 3m$; $M_x = 0$.

"The results obtained from the shaft's analysis as a beam show that the proposed grooved shaft is resistant to the forces acting on it and has the potential to operate under the forces in the technological process.

Results and discussion. The materials and dimensions of the shaft, spacer rings, and half-coupling, as well as the dimensions of the shaft's thread, were studied. This allowed the determination of the modulus of elasticity, the moment of inertia of the section, the inertia of rotation, mass, and the pitch of the nut's rotation during one revolution.

The balancing process of shafts in technological machines, especially the grooved shaft, ensures that they operate as required in technological processes. A lack of balance can lead to negative consequences, such as in the case of the jin machine [4]. Therefore, in our study, we examine both static and dynamic balancing. Typically, rotating parts, even if balanced in their design, may experience imbalances in practice due to uncertainties in manufacturing processes (such as casting, turning) or the non-homogeneity of materials (such as voids in castings), which can cause the center of mass to shift from the rotational axis. This results in inertia moments not being zero, and the rotational axis may not coincide with the main axis of inertia, indicating imbalance. Such imbalances must be eliminated by artificial means, a process known as balancing. There are two types of balancing: static and dynamic[5].

The purpose of static balancing is to bring the center of mass to the axis of rotation, meaning the rotational axis is brought back to the central axis of inertia. In this case, no centrifugal force arises during the rotation of the shaft, but a pair of inertia forces dependent on the magnitude of the centrifugal inertia moments may remain. If the shaft is not excessively long, the value of this pair of inertia forces will be small, and thus, static balancing alone can be sufficient. For shafts and rotors with high rotational speeds, dynamic balancing is necessary. Its goal is to return the rotational axis to the central inertia axis, ensuring that no centrifugal forces arise during the rotation and that no inertia forces dependent on the centrifugal moments of the masses occur. When the center of mass shifts from the axis of rotation, variable resistance to rotation can be observed, affecting the linkage to the shaft and potentially causing several malfunctions. Thus, static balancing of shafts becomes necessary[5].

Conclusion. The static analysis of the grooved shaft plays a crucial role in engineering, as it helps ensure the strength, safety, and efficiency of structures. Through static analysis, forces, moments, stresses, and deformations acting on the shaft are determined, while considering the material properties and loading conditions. All of this requires a systematic approach and the use of analytical, mathematical, and numerical methods. By correctly conducting static analysis, the proper design and service life of the shaft can be determined, making the structures reliable and efficient. Furthermore, it improves technical safety and is essential for ensuring the safety of large structures and transport systems. Additionally, proper analysis of material interactions and geometric shapes is necessary, even in the case of multilayer materials. The static analysis of the shaft also enhances economic efficiency, as errors in design or processing can lead to high costs in the long term. Therefore, correctly performing static analysis in the design and manufacturing process of the shaft is a key factor for successful and sustainable operation in all industrial sectors. Thus, the static analysis of the grooved shaft is of great importance not only theoretically but also practically, helping to improve performance efficiency in various industrial sectors[7,8,9].

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