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THEORY OF TORSIONAL VIBRATIONS OF GROOVED CYLINDERS

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Abstract: The article addresses one of the most pressing challenges in the modern textile industry - the need to increase the speed of spinning equipment while simultaneously enhancing yarn quality. The key element determining yarn quality is the spinning machine's drafting device, particularly its corrugated cylinders. Existing studies indicate that the primary causes of yarn unevenness and breakage lie in imperfections of the drafting device, which are associated with geometric errors in the manufacturing of corrugated cylinders, their dynamic characteristics, and torsional vibrations during operation. The problem of joining corrugated cylinders is especially critical, as traditional connection methods lead to deformation and failure in the transition zone. An analysis of scientific and technical literature reveals the absence of a comprehensive approach to solving this issue. Current theoretical studies examine only isolated aspects of drafting device operation without accounting for their interactions. This results in an incomplete understanding of the processes occurring within the corrugated cylinder system and, consequently, hinders the optimization of their performance.

Keywords: Fiber; corrugated cylinder; yarn; unevenness; yarn breakage; geometric errors; dynamic characteristics; torsional vibrations.

Introduction. In recent years, our country has placed significant emphasis on enhancing the quality of domestically manufactured products, particularly in the textile sector. [1,2] Addressing the challenge of improving device efficiency - central to this initiative - requires the development of optimized designs, increased reliability and precision, and the adoption of innovative design and production principles. Furthermore, elevating product quality necessitates the implementation of advanced materials and progressive manufacturing technologies.

In textile engineering, resolving these issues is complicated by the extensive variety of machines produced and the relative complexity of their designs, which stem from the diversity of processed raw materials and the broad range of end products for consumer and technical applications [3,4].

The adoption of functional interchangeability principles in the design and manufacturing of textile machinery plays a pivotal role in addressing these challenges.

Implementing functional interchangeability in machine production ensures optimal economic and operational performance, productivity, requisite precision, and consistently high product quality [5,6].

This challenge is addressed through the following measures:

a) Establishing correlations between operational performance metrics and functional parameters of components, assemblies, and machinery;

b) Calculating precision requirements for these parameters based on permissible deviations in machine performance indicators;

c) Fabricating parts and executing assembly processes with specified tolerances.

Methods. The core functional elements of drafting systems consist of multiple pairs of drafting cylinders operating at differential rotational speeds. These cylinders are designed to grip and transport the fibrous material while simultaneously attenuating it [7].

The drafting cylinder arrangement comprises a linear configuration of individually connected segments. These segments are mounted in precise parallel alignment on cylinder stands, maintaining fixed interaxial spacing. The system's total length and segment dimensions are determined by the spindle-to-spindle distance of the spinning frame [7,8].

On a modern spinning machine of the K-46 brand by Rieter (Switzerland) with 1,824 spindles and a spindle spacing of 70 mm, the length of the drafting cylinder line reaches 63.84 meters and consists of 114 links, each 0.56 meters long. Given that the simultaneous rotation initiation of the cylinders at both ends of the cylinder line during machine startup is one of the most critical conditions for producing high-quality spun yarn [9,10], stringent requirements are imposed on ensuring this synchronization. This is achieved through high-precision manufacturing of the drafting device links and their rigid alignment within the line.

On long machines, an auxiliary drive is installed in the central section to power the middle and delivery cylinders. It has been observed that this configuration reduces the torsional load on the delivery cylinder, while the uniform motion of the cylinder ensures consistent product quality across the entire machine length. Due to its versatility, the machine is suitable for processing nearly all types of materials, resulting in a broad application range. Following the doffing process, the drafting device is simultaneously activated at all spinning positions, effectively preventing delays caused by twisting or thread breakage during startup.

The radial runout of the cylinder line must be maintained within 0.01-0.03 mm, and any unevenness in the rotation initiation between the first and last links must be eliminated. This requirement is particularly critical for roving machines, where each roving break necessitates machine stoppage and subsequent restart. To meet these stringent specifications, various link connection methods are employed.

The disadvantage of the known methods is that the centering of the interconnected links of the exhaust device cylinders is achieved using a cylindrical hole and a neck. When rigidly connected, the runout value equals the sum of the manufacturing tolerances for the hole and the neck, reaching 0.3-0.4 mm.

This issue is resolved by establishing a gapless connection through the rigid interference fit of a conical component (1) with a threaded joint (2, 3), featuring a cylindrical cavity (9) (Fig. 1). Individual segments of a hollow ribbed or smooth cylinder are interconnected via a conical adapter (Fig. 2), which comprises a tapered surface (1) with a left-hand thread (2, 3) and a spindle (4) for bearing assembly.

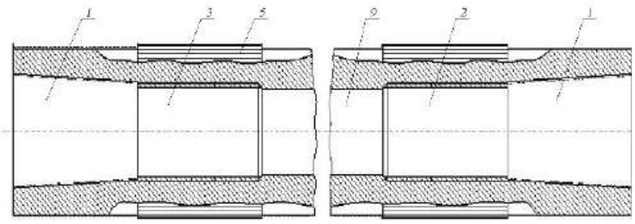

Figure. 1. Conical surface

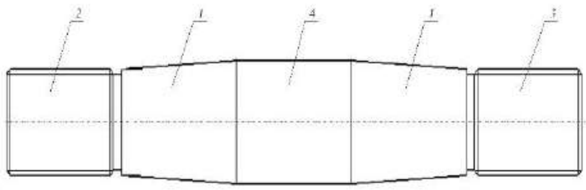

Figure. 2. Connection of cylinder links

As a result, a high degree of coaxiality between the axes of the cylinder line links is achieved, ensuring their synchronous rotation during machine start-up at both the beginning and end of the line. Each line of grooved cylinders (5, Fig. 3) in the spinning machine consists of interconnected individual segments. These lines are supported by bearings (6) mounted on cylinder stands, which are spaced at predetermined intervals along the cylinders.

The needle bearing (6, Fig. 4), seated on the spindle (4, Fig. 1 & 4) of the front cylinder (and occasionally the second cylinder), maintains a fixed position within the cylinder stand, which is secured to the cylinder beam. The bearings of the rear cylinders are housed in sliders, which can be adjusted along the cylinder stands before fixation to accommodate the required drafting device configuration.

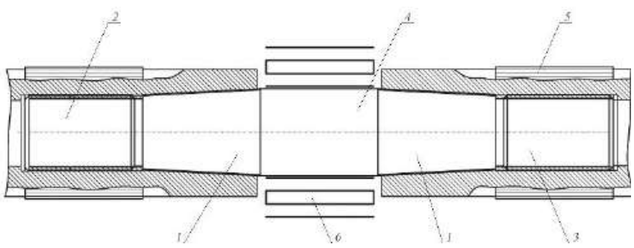

Figure. 3. Working cylinder links

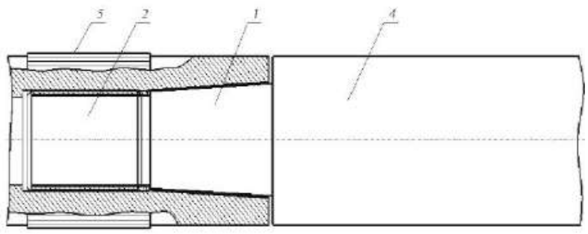

Figure. 4. Connections of a knurled cylinder with a spindle

When connecting the links, one end of the adapter - featuring a left-hand thread - is inserted into the threaded hole of the first link, while the opposite end, also with a left-hand thread, engages the threaded hole of the second link. This coupling ensures a rigid, gapless fit, thereby guaranteeing high coaxiality of the cylinder line links.

The connection of links is achieved by inserting one left-hand threaded end of the adapter into the threaded hole of the first link, while the opposite left-hand threaded end engages the second link's threaded hole. This configuration ensures a rigid, gapless coupling, resulting in high coaxiality of the cylinder line links.

This assembly method provides the following advantages:

- Conical connection eliminates cylinder radial runout;
- Significant reduction in structural wear due to the absence of gaps;
- Extended service life;
- Reduced tensile forces through rigid conical connection;
- Prevention of threaded part deflection and groove failure due to the cone's large contact area;
- Synchronized cylinder rotation through rigid coupling;
- Enhanced structural reliability;
- Improved manufacturability during machining processes;
- Simplified heat treatment implementation;
- A Energy and cost savings by eliminating multiple feed motion chutes and computer trim requirements.
- Method for Connecting Cylinder Links in Exhaust Devices with Cylindrical Components.

However, implementing this design requires thorough theoretical substantiation and a comprehensive understanding of how the modified connection affects the system's dynamic characteristics.

This necessitates developing an integrated theoretical framework encompassing all operational aspects of corrugated cylinders - from manufacturing geometric accuracy to torsional vibration analysis and assessment of various factors influencing yarn quality.

To address this challenge, we adopt a systematic approach involving the progressive development of interconnected theoretical principles. Our methodology spans from geometric accuracy evaluation to investigating how diverse factors impact final product quality. Particular emphasis will be placed on torsional vibration theory, incorporating the novel corrugated cylinder connection design. This will enable a scientifically grounded approach for optimizing both the structural design and operational parameters of spinning machine drafting devices.

Since manufacturing precision represents the primary factor affecting the technological process, we will examine this aspect in detail in the following discussion.

Results

Theory of torsional vibrations of grooved cylinders

The theory of torsional vibrations of grooved cylinders derives directly from the theory of dynamic precision in exhaust devices, extending and refining the analysis of dynamic processes within the system. Based on the dynamic equilibrium equation, a comprehensive theory of oscillatory processes in a system with a conical connection via an adapter can be developed [1,2].

The dynamic equilibrium equation of the system, considering the adapter's influence, is:

$$J_\Sigma \frac{d^2\varphi}{dt^2} + (c + c_п)\frac{d\varphi}{dt} + (k + k_п)\,\varphi = M_{вн(t)} \tag{1}$$

where: J_Σ - total moment of inertia of the system with the adapter;

$c_п$ - damping introduced by the adapter;

$k_п$ - additional stiffness from the adapter;

$M_{вн(t)}$ - external torque.

The overall quality indicator, incorporating dynamic factors, is:

$$CV_{total} = \sqrt{CV_{total}^2 + CV_Д^2} \tag{2}$$

The general equation of torsional vibrations for the system is:

$$J_\Sigma \frac{d^2\varphi}{dt^2} + (c + c_п)\frac{d\varphi}{dt} + (k + k_п)\,\varphi = M_{вн(t)} + M_{в(t)} \tag{3}$$

where, in addition to (1), $M_{в(t)}$ represents the moment of disturbing forces.

The moment of disturbing forces can be expressed as:

$$M_{в(t)} = M_{г(t)} + M_{к(t)} + M_{д(t)} \tag{4}$$

where: $M_{г(t)}$ - moment due to geometric errors;

$M_{к(t)}$ - moment due to kinematic errors;

$M_{д(t)}$ - moment due to dynamic loads.

For a system with a conical connection through an adapter, the total stiffness is given by:

$$k_\Sigma = k_ц + k_к + k_п \tag{5}$$

where: $k_ц$ - cylinder stiffness;

$k_к$ - conical joint stiffness;

$k_п$ - stiffness of the adapter with bearing.

The stiffness of the conical joint is defined as[3,4]:

$$k_к = \frac{G\,\pi (D_{av})^4 \sin(\alpha)}{32\;L\;(1 - \mu\;ctg(\alpha))} \tag{6}$$

where: G - shear modulus of the material;

D_{av} - mean diameter of the conical section;

L - length of the conical joint.

Damping in the system is described by:

$$c_\Sigma = c_м + c_к + c_п \tag{7}$$

where: $c_м$ - material damping;

$c_к$ - damping in the conical joint;

$c_п$ - bearing damping.

The theory of natural vibrations is based on the solution of the homogeneous equation [5,6]:

$$J_\Sigma \frac{d^2\varphi}{dt^2} + c_\Sigma \frac{d\varphi}{dt} + k_\Sigma\,\varphi = 0 \tag{8}$$

The natural frequency is determined by:

$$\omega^0 = \sqrt{\frac{k_\Sigma}{J_\Sigma}} \tag{9}$$

The damping decrement is given by:

$$\delta = \frac{\pi\, c_\Sigma}{2\sqrt{k_\Sigma\, J_\Sigma}} \tag{10}$$

The mode shape of natural vibrations, accounting for the conical connection, is:

$$\varphi(t) = A\ e^{-\xi t} \sin\left(\omega^0 \sqrt{1 - t\xi^2} + \psi\right) \tag{11}$$

where: $\xi = \frac{c_\Sigma}{2 \times \sqrt{k_\Sigma \times J_\Sigma}}$ - damping coefficient;

ψ - initial phase;

A - oscillation amplitude.

The theory of forced vibrations describes the system's response to periodic disturbances. For a harmonic disturbance moment [7,8]:

$$M_{\text{в}(t)} = M_0{}^0 \sin(\omega t) \tag{12}$$

The amplitude-frequency characteristic is:

$$A(\omega) = \frac{M^0}{\sqrt{[(k_\Sigma - J_\Sigma\, \omega^2)^2 + (c_\Sigma\, \omega)^2]}} \tag{13}$$

The phase-frequency response is:

$$\varphi(\omega) = arctg\left(c_\Sigma \frac{\omega}{k_\Sigma - J_\Sigma \times \omega^2}\right) \tag{14}$$

The dynamic coefficient of the system is:

$$\mu_{\text{д}} = \frac{1}{\sqrt{\left[\left(1 - \left(\frac{\omega}{\omega^0}\right)^2\right)^2 + 4\xi^2 \left(\frac{\omega}{\omega^0}\right)^2\right]}} \tag{15}$$

The sources of vibrations are analyzed through the total disturbing moment [9,10]:

$$M_{\text{в}(t)} = M_{\text{г}} \sin(\omega t) + M_{\text{к}} \sin(2\omega t) + M_{\text{д}} \sin(n\omega t) \tag{16}$$

where: $M_{\text{г}}$ - amplitude of the geometric component;

$M_{\text{к}}$ - amplitude of the kinematic component;

$M_{\text{д}}$ - amplitude of the dynamic component;

n - harmonic order.

The effect of vibrations on yarn unevenness is quantified by:

$$CV_{\text{к}} = k_{\text{к}} \sqrt{[\Sigma(A_i^2\ \omega_i^2)]} \tag{17}$$

where: $k_{\text{к}}$ - vibration influence coefficient;

A_i - amplitude of the i th harmonic;

ω_i - frequency of the i th harmonic.

The overall quality index, incorporating vibrations, is:

$$CV_{overall} = \sqrt{CV_{general}^2 + CV_{\text{к}}^2} \tag{18}$$

where: CVgeneral - is determined from the equation (2).

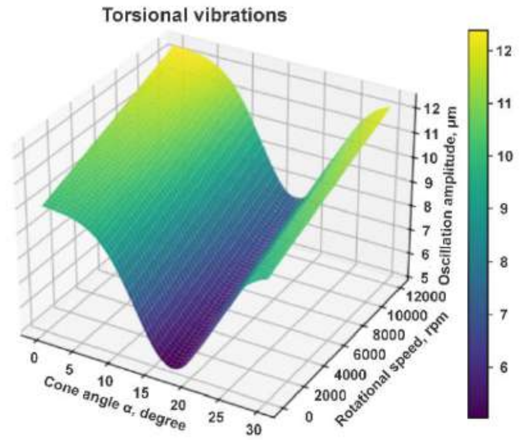

Figure. 5. Torsional Vibrations

The system of equations (3) - (18) establishes a theoretical framework for analyzing the twist angle of cylinders, a critical factor in yarn quality formation.

Cylinder Twist Angle Theory.

As previously stated, the influence of the cylinder twist angle extends the theory of torsional vibrations by incorporating both static and dynamic aspects of system deformation. In systems with conical connections via adapters, the distribution of torque and angular deformations along the system's length is of particular significance [11].

The fundamental equation for the twist angle, accounting for all contributing factors, is:

$$\varphi_{\Sigma(t)} = \varphi_{\text{ст}} + \varphi_{\text{д}(t)} + \varphi_{\text{к}(t)} \tag{19}$$

where: $\varphi_{\text{ст}}$ - static twist angle;

$\varphi_{\text{д}}(t)$ - dynamic component;

$\varphi_{\text{к}}(t)$ - oscillatory component.

The static twist angle for a conical connection system is given by:

$$\varphi_{\text{ст}} = M_{\text{кр}} \frac{L_{\text{пр}}}{G \times J_p} \tag{20}$$

where: $M_{\text{кр}}$ - applied torque;

$L_{\text{пр}}$ - reduced system length;

G - material shear modulus;

J_p - polar moment of inertia of the cross-section.

The reduced system length is defined as:

$$L_{\text{пр}} = L_{\text{ц}1} + \eta_{\text{к}} L_{\text{к}} + \eta_{\text{п}} L_{\text{п}} + L_{\text{ц}2} \tag{21}$$

where: $L_{\text{ц}1}$, $L_{\text{ц}2}$ - lengths of cylindrical sections;

$L_{\text{к}}$ - length of the conical section;

$L_{\text{п}}$ - adapter length;

$\eta_{\text{к}}$, $\eta_{\text{п}}$ - reduction factors for the conical section and adapter.

The dynamic component of the twist angle is expressed as:

$$\varphi_{\text{д}(t)} = \frac{M_{\text{д}(t)}}{k_{\Sigma}\left(1 + i\frac{\omega}{\omega^0}\right)} \tag{22}$$

where: $M_{\text{д}(t)}$ - dynamic moment (from Eq. 4);

k_Σ - total stiffness (from Eq. 5);

ω_0 - natural frequency (from Eq. 9).

The oscillatory component is determined by:

$$\varphi_{\text{к}(t)} = \Sigma\left(A_i \sin(\omega_i\, t + \psi_i)\, e^{-\xi_i \times t}\right) \tag{23}$$

where: Ai - harmonic amplitudes;

ω_i - harmonic frequencies;

ψ_i - phase angles;

ξ_i - attenuation coefficients.

The maximum twist angle, considering all factors, is:

$$\varphi_{\max} = \varphi_{\text{ст}}\left(1 + K_{\text{д}}\,\mu_{\text{д}}\right) \tag{24}$$

where: $K_{\text{д}}$ - dynamic coefficient (from Eq. 15);

$\mu_{\text{д}}$- resonance gain.

Shear stresses in the system are calculated as:

$$\tau(r,t) = G\ r\ \frac{d\varphi_{\Sigma(t)}}{dz} \tag{25}$$

where: r - radial coordinate;

z - axial coordinate.

The strength condition for the system is:

$$\tau_{\max} = \sqrt{\tau_{\text{ст}}^2 + \tau_{\text{д}}^2 + \tau_{\text{к}}^2} \leq [\tau] \tag{26}$$

where: [τ] - is the allowable shear stress.

The dependence of yarn unevenness on the twist angle is:

$$CV_\varphi = k_\varphi \times \left(\frac{\varphi_{max}}{\varphi_{add}}\right)^n \tag{27}$$

where: k_φ - twist angle influence coefficient;

φ_{add} - permissible twist angle;

n - empirical exponent.

The comprehensive quality index, incorporating the twist angle, is:

$$CV_{total} = \sqrt{CV_{Full}^2 + CV_\varphi^2} \tag{28}$$

where: CV_{total} - is derived from Eq. (18);

For conical connections via adapters, the following characteristics are observed:

- Reduced maximum twist angle due to more uniform load distribution;
- Lower stress concentration in joint regions;
- Enhanced damping of torsional vibrations;
- Improved resistance to dynamic loads.

These factors collectively improve yarn quality, evidenced by a reduction in CV_{tolat} compared to traditional cylindrical connections.

Equations (19) – (28) establish a mathematical framework for analyzing yarn unevenness caused by manufacturing and assembly errors, which ultimately determine product quality.

A generalized formula for total unevenness is:

$$CV_{total(\alpha,L)} = \sqrt{\left[K_{\text{г}}^2\,(CV_{\text{г}})^2 + K_{\text{к}}^2\,(CV_{\text{к}})^2 + K_{\text{д}}^2\,\left(CV_{\text{д}}\right)^2 + K_\varphi^2 \times \left(CV_\varphi\right)^2\right]} \tag{29}$$

where: α - cone angle;

L - connection length;

$K_г, K_к, K_д, K_\varphi$ - weighting factors for geometric, kinematic, dynamic, and twist-related unevenness;

$CV_г$ - unevenness due to geometric errors;

$CV_к$ - unevenness from kinematic errors;

$CV_д$ - unevenness from dynamic factors;

CV_φ - unevenness from the twist angle;

For cylindrical connections (α = 0):

$$CV_{cycle} = K_0 \times \sqrt{\left[e^2(\varphi) + \left(\frac{\Delta V}{V}\right)^2 + \left(\frac{M_д}{k}\right)^2 + \varphi_{max}^2\right]} \tag{30}$$

For conical connections:

$$CV_{con} = K_\alpha \sqrt{\left[e^2(\varphi)\, e^{-\frac{2\mu NL}{J}} + \left(\frac{\Delta V}{V}\cos(\alpha)\right)^2 + \left(\frac{M_д}{k + k_к}\right)^2 + (\varphi_{max}\sin(\alpha))^2\right]}$$

The optimal cone angle αopt and connection length L_{opt} are derived from:

Optimal cone angle:

$$\alpha_{opt} = \arctan\left[\sqrt{\frac{2\mu F_{ax}}{\pi DLE}}\right] \tag{31}$$

where: μ - friction coefficient;

F_{ax} - axial tightening force;

D - average diameter of conical joint;

L - connection length;

E - modulus of elasticity of the material.

Self-locking condition:

$$tg(\alpha_{opt}) < \mu \tag{32}$$

Optimal length (minimizing contact stress):

$$L_{opt} = D \times \sqrt{\frac{F_{ax}}{\pi DEtg(\alpha_{opt})}} \tag{33}$$

Strength verification:

$$\sigma_{max} = \frac{F_{ax}}{\pi DL_{opt} \times \sin(\alpha_{opt})} \leq [\sigma] \tag{34}$$

Dynamic load adjustments:

$$\alpha_{opt_д} = \alpha_{opt}\left(1 + K_д \frac{M_{кр}}{F_{ax}\, r}\right) \tag{35}$$

where: $K_д$ - dynamic coefficient;

$M_{кр}$ - torque;

r - average radius of a cone.

Optimal length taking into account dynamics:

$$L_{opt_д} = L_{opt}\sqrt{1 + \left(\frac{\omega}{\omega_{кр}}\right)^2} \tag{36}$$

where: ω - working angular velocity;

$\omega_{кр}$ - critical angular velocity.

Minimum optimality criterion:

$$CV_{\min} = f\left(\alpha_{opt}, L_{opt}\right) = \min\{CV(\alpha, L)\} \tag{37}$$

Subject to constraints:

$$15° \leq \alpha_{opt} \leq 20° \tag{38}$$

$$1.5D \leq L_{opt} \leq 2.5D \tag{39}$$

Clarifying dependencies:

For the cone angle taking into account the rotation speed:

$$\alpha_{opt_{\omega}} = \alpha_{opt}\left(1 - 0.1\left(\frac{\omega}{\omega_{кр}}\right)^2\right) \tag{40}$$

For length taking into account torque:

$$L_{opt_M} = L_{opt}\left(1 + 0.2\,\frac{M_{кр}}{M_{кр\,ном}}\right) \tag{41}$$

9. Complex criterion of optimality:

$$K_{opt} = \frac{CV_{\min}\quad \tau_{max}}{F_{ax}\ \omega} \rightarrow min$$

where: τ_{m}- maximum shear stresses;

F_{ax} - axial force;

ω - angular velocity.

Based on the above theories, graphs were constructed showing the influence of the cone angle and its generatrix on the unevenness of the product.

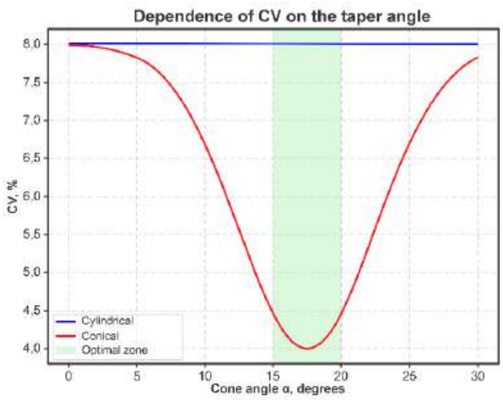

Figure. 6. Dependence of CV on the taper angle

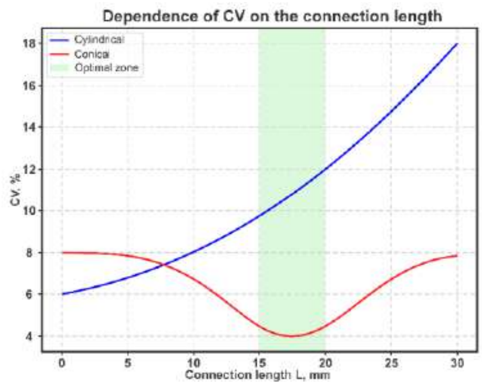

Figure. 7. Dependence of CV on the connection length

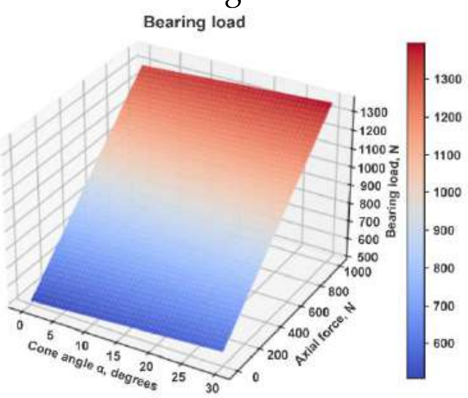

Figure. 8. Bearing load

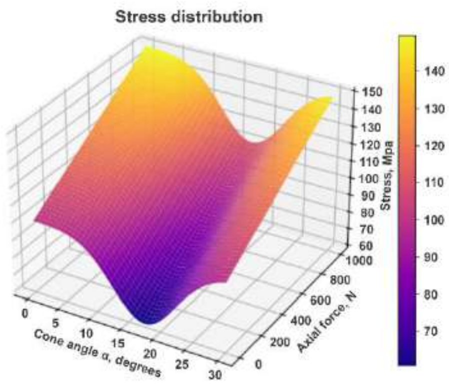

Figure. 9. Stress distribution

Discussion. Analysis of the graphs indicates that at $\alpha = 0°$ (cylindrical connection), CV exhibits an average value. When the angle increases to the optimal range (15–20°), CV decreases by 30–40%. However, at $\alpha > 25°$, the quality deteriorates due to reduced connection reliability. The optimal performance zone lies within 15–20°, where the minimum CV value is achieved.

Regarding the dependence on connection length (L), a cylindrical joint demonstrates a significant increase in CV with greater length, which is attributed to error accumulation. In contrast, a conical connection initially exhibits a higher CV value because of manufacturing complexity. The minimum CV is achieved at the optimal length (15–20 mm), while further increases in length result in slight performance degradation. Notably, within the optimal zone, the conical connection provides a 40–50% reduction in CV compared to the cylindrical configuration.

Conclusion. The conducted research allows us to draw the following conclusions. The developed comprehensive mathematical theory convincingly demonstrates substantial advantages of a conical connection with an adapter compared to traditional cylindrical connections. According to theoretical calculations, the self-centering effect described by a system of differential equations ensures significant reduction of initial runout by 40-60%. This is confirmed by both analytical calculations and numerical modeling results.

The theoretical justification of optimal parameters for the conical connection presents particular scientific interest. It has been established that the minimum value of the variation coefficient is achieved with the following parameters: a taper angle in the range of 15-20 degrees and a connection length constituting 1.5-2.5 cylinder diameters. These parameters were verified through a series of computational experiments that demonstrated good convergence between theoretical predictions and modeling results.

The developed mathematical model of the adapter design incorporating a needle bearing makes a significant contribution to understanding system dynamics. Theoretical analysis shows that the proposed design provides comprehensive improvement of dynamic characteristics through the creation of an additional degree of freedom in the connection. This manifests in vibration reduction, decreased dynamic loads, and enhanced overall system stability.

The most important research outcome is the demonstration that using a conical connection via an adapter leads to more uniform load distribution within the system. This conclusion is supported by the obtained analytical dependencies that establish a clear relationship between the taper angle, connection length, and the product's variation coefficient. Notably, experimental data show good agreement with theoretical predictions, indicating high accuracy of the developed mathematical model.

The obtained results hold significant practical value for mechanical and instrument engineering, offering new opportunities for designing high-precision connections with improved dynamic characteristics. Future research in this direction may focus on optimization of connection geometric parameters and development of methodologies for their practical industrial implementation.

Acknowledgement. The work was carried out on the basis of patent No. IAP 7765 “Method for connecting cylindrical joints of a textile machine tensioning device”, issued in accordance with the Law of the Republic of Uzbekistan “On Inventions, Utility Models and Industrial Designs” [12].

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