INTERNATIONAL MULTI DISCIPLINARY JOURNAL FOR RESEARCH & DEVELOPMENT
eISSN 2394-6334 Impact factor: 8.854 Volume 13, issue 06 (2026)

STABILITY OF A COMPOSITE LOAD-BEARING CYLINDER UNDER TORSION CONSIDERING THE EFFECT OF TENSILE FORCES

Otabek A. Mirzaev
Associate Professor, Department of Labor Protection and Technical Safety,
Karshi State Technical University, Uzbekistan.
E-mail: kiamoa@mail.ru

Abstract.
This article presents a theoretical analysis of the stability of a composite load-bearing cylinder subjected to torsion while taking into account the effect of tensile forces acting in both the axial and circumferential directions. The analysis is based on the ring forces occurring in a cylindrical shell. In solving this problem, a coordinate system was employed in which the deflection function was assumed to represent the helical wave pattern that develops after the loss of stability. For a clearer understanding of the results, numerical data are presented and discussed.

Keywords: cylinder, stability, function, results, equation, shell, boundary conditions, strength, torsion, Poisson’s ratio, analogous systems, elasticity, approximations, table, drum, values, longitudinal direction.

INTRODUCTION

Research on the stability of thin-walled shells is largely associated with the use of a rather cumbersome computational apparatus. Therefore, when studying shell stability, various methods are widely applied, in particular variational methods, and simplifying hypotheses are introduced regarding the nature of the shell's shape change during buckling. One common simplifying approach is to divide shell stability into general and local stability, or as mechanicians often say, local buckling [1].

Accurate modeling of the complex properties of composite materials and the effects of tensile loads on them is possible only with the help of modern electronic resources and advanced software tools [2]. This problem cannot be fully solved by classical analytical methods; therefore, it is necessary to use high-quality electronic databases. The main need is related to the following: the use of advanced methods; access to current research; specialized software and models; comparison of experimental data with models. The complexity of this topic and the need to apply modern approaches make the use of electronic resources not merely important but absolutely essential, since without them it is practically impossible to achieve accurate and reliable results.

The results obtained from studying shear deformation allow us to move on to solving the problem of strength verification under torsion. In practice, torsion is encountered very often: axles of rolling stock, transmission feed cylinders, shells – all these are examples of members subjected to torsion [3].

The feed cylinder in the feed zone of spinning machines rotates about a fixed axis under the action of a system of forces applied to it [1,2]. When drawing the sliver from the canvas or can, no significant axial force arises and no deformation of the sliver is observed; therefore, no redistribution of fibers along the length of the sliver occurs.

In the space, aviation, and construction industries, lattice shells of various configurations and purposes are used [1]. Their application is due to the combination of high strength characteristics, low weight, and system connectivity [3–5].

In the current century, one of the key problems in mechanical engineering is the study and analysis of the strength of elastic shells across different industries. No less important is accounting for the influence of the contacting external environment, as well as the use of effective and proven mathematical calculation methods that ensure reliable results [6].

Torsional vibrations are defined as periodic angular oscillations of masses concentrated on a shaft, causing twisting of its individual sections [11].

RESEARCH METHODS AND METHODOLOGIES

The theoretical research is based on the theory of shell dynamics. The stability of a composite feed cylinder under torsion, taking into account the action of tensile forces, has been studied using the analytical method of statics and dynamics of shells and the theory of elasticity. The problems of stability of the feed cylinder shell are solved by applying appropriate boundary conditions and a deflection function. It should be particularly noted that numerical calculations have been performed to assess the influence of the main geometric and mechanical parameters on the stability characteristics of elastic shells.

RESEARCH RESULTS AND DISCUSSION

In the feed and discretization zone of a pneumomechanical spinning machine, if we conditionally disregard the edges and corner sections of the feed table walls (it should be noted that a twisting process also occurs there), then when the teeth of the discretizing drum grip the fiber sliver, a twisting process also takes place on the table surface. This phenomenon affects the stability of the composite feed cylinder. As the object of study, we take the composite feed zone and depict it for clarity in the figure (see Fig. 1).

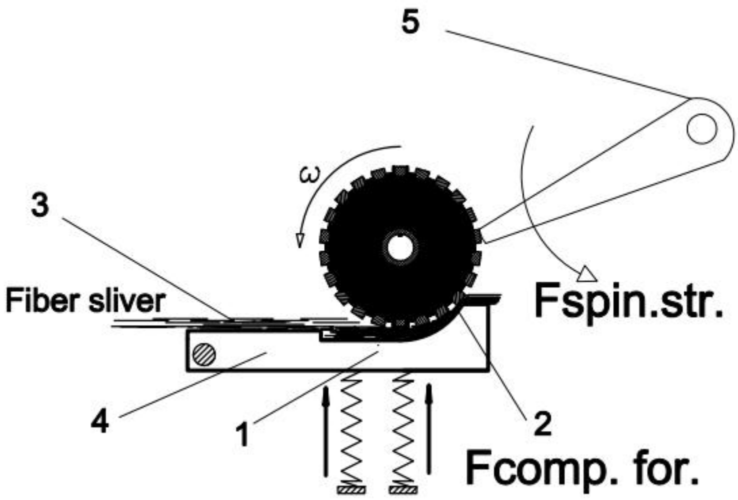

Figure 1. Feeding zone of the pneumomechanical spinning machine.
1 – feeding table, 2 – walls of the feeding table, 3 – fiber sliver, 4 – base of the feeding table, 5 – needle of the combing roller.

It should be noted that the moment of internal forces arising in any cross-section of rubber shells during torsion, which rotates that cross-section about the longitudinal axis, is called the torque. The magnitude and direction of the torque depend on the value of the external moments applied to the considered part of the elastic shells of the feed cylinder.

When solving this problem in a rectangular coordinate system for the deflection function, it is necessary to adopt an expression that reflects the wavy shape after buckling. The most suitable function will be,

$\omega = W(x)\cos\left(\frac{\lambda x}{R} - n\theta\right)$ (1)

Similarly, for the stress function we take...

$\varphi = F(x)\cos\left(\frac{\lambda x}{R} - n\theta\right)$(2)

Let us formulate the boundary conditions for the functions FF and WW. With the chosen expressions for the functions φφ and ωω, the boundary conditions for a rigid clamp are most simply written. In this case, at the ends of the feed cylinder shell, we must have...

$$\omega = W(x)\cos\left(\frac{\lambda x}{R} - n\theta\right) = 0 \qquad (3)$$

After taking the derivative, we obtain...

$$\frac{\partial\omega}{\partial x} = W^{I}(x)\cos\left(\frac{\lambda x}{R} - n\theta\right) - \frac{\lambda}{R} W\sin\left(\frac{\lambda x}{R} - n\theta\right) = 0 \qquad (4)$$

For the stress function φφ at the ends of the feed cylinder shell, we assume the conditions that the additional forces NxNx and NxyNxy after buckling are equal to zero.

$$N_x = \frac{\partial^2\varphi}{\partial y^2} = -\frac{n^2}{R^2}\cos\left(\frac{\lambda x}{R} - n\theta\right) = 0 \qquad (5)$$

$$N_{xy} = \frac{\partial^2\varphi}{\partial x \partial y} = -\frac{n}{R} F^{I}\sin\left(\frac{\lambda x}{R} - n\theta\right) - \frac{\lambda x}{R^2} F\cos\left(\frac{\lambda x}{R} - n\theta\right) = 0 \qquad (6)$$

After this, we write the torque that twists the shells of the feed cylinder.

$$M_{кр} = 2\pi R^2 N_{xy}^0$$

Excluding the function F_1 [---] from these equations and assuming that $W_1 \neq 0$, we obtain expressions for the critical torque $M_{кр}$ in the first approximation.

(7)

$$\frac{M_{кр}^{I}}{2\pi E\delta R^2} = \frac{[2+(\frac{1}{2R})^2\lambda^2]^2}{[8+4(\frac{1}{2R})^2(n^2+3\lambda^2)+(\frac{1}{2R})^4(n^2+\lambda^2)^2]\lambda n} + \frac{4R^2\delta^2[8+4\left(\frac{1}{2R}\right)^2(n^2+3\lambda^2)+(\frac{1}{2R})^4(n^2+\lambda^2)^2}{3(1-\mu^2)n\lambda l^4} + \frac{4N_x^0R^2[2+(\frac{1}{2R})^2\lambda^2}{E\delta n\lambda l^2} + \frac{N_y^0 n}{E\delta\lambda} \qquad (8)$$

Having performed similar calculations for the second approximation, noting that $h=\frac{1}{3}1$, we obtain expressions for the critical torque $M_{кр}$ in the second approximation:

$$\frac{M_{кр}^{II}}{\pi E\delta R^2} = \frac{[1+(\frac{1}{3R})^2\lambda^2]^2}{[3+2(\frac{1}{3R})^2(n^2+3\lambda^2)+(\frac{1}{3R})^4(n^2+\lambda^2)^2]\lambda n} + \frac{27R^2\delta^2[3+2\left(\frac{1}{3R}\right)^2(n^2+3\lambda^2)+(\frac{1}{3R})^4(n^2+\lambda^2)^2}{4(1-\mu^2)n\lambda l^4} + \frac{9N_x^0R^2[1+(\frac{1}{3R})^2\lambda^2}{E\delta n\lambda l^2} + \frac{N_y^0 n}{E\delta\lambda} \qquad (9)$$

In the case of long shells of the feed cylinder, we can assume that...

$$(\frac{1}{2R})^2\lambda^2 \gg 1,\ n^2 \gg \lambda^2$$

After expression (8), it can be approximately written in the form...

$$\frac{M_{кр}^{I}}{\pi E\delta R^2} = \frac{\lambda^2}{n^2} + A\frac{n^2}{\lambda} + B\frac{\lambda}{n} + C\frac{n}{\lambda} \qquad (10)$$

$$A = \frac{\delta^2}{12(1-\mu^2)R^2}, B = \frac{N_x^0}{E\delta}, C = \frac{N_y^0}{E\delta}.$$

Consider the case when $N_x^0 = N_y^0 =$Then...

$$\frac{M_{кр}^{I}}{\pi E\delta R^2} = \frac{\lambda^3}{n^5} + A\frac{n^3}{\lambda} \qquad (11)$$

After this, we find the minimum of this expression with respect to λλ.

$$\frac{\partial}{\partial\lambda}\frac{M^{I}_{кр}}{\pi E\delta R^2}=\frac{3\lambda^3}{n^5}-A\frac{n^3}{\lambda}=0 \qquad (12)$$

Whence

$$\lambda=\frac{n^2\sqrt{\frac{\delta}{R}}}{\sqrt{6}\sqrt{1-\mu^2}} \qquad (13)$$

Then for n=2 the following smallest values of $M^{I}_{кр}$ will be obtained:

$$M^{I}_{кр}=1.7E\delta^2R\frac{\sqrt{\frac{\delta}{R}}}{\sqrt[4]{(1-\mu^2)^2}} \qquad (14)$$

Reference and result data on the stability of the composite feed cylinder under torsion, taking into account the effect of tensile forces on it, are presented in Table 1.

Table 1.

№	Notation	Parameter name	Numerical values or their limit boundaries
1	δ	Thickness of the cross-section of the feed cylinder shell wall	4 мм
2	**E**	Modulus of longitudinal elasticity of the cylindrical shell of the composite feed cylinder	$40\frac{H}{мм^2}$
3	**μ**	Poisson's ratio for the shell material of the composite feed cylinder	0.47
4	**h**	Thickness of the rubber shell of the composite feed cylinde	4 мм
5	**n**	Natural numbe	2
7	R	Radius of the composite feed cylinder (elastic part)	7 мм

The resulting dependence of the critical torque on the elastic modulus of the feed cylinder shells in the feeding zone of pneumomechanical spinning machines is shown in Fig. 2.

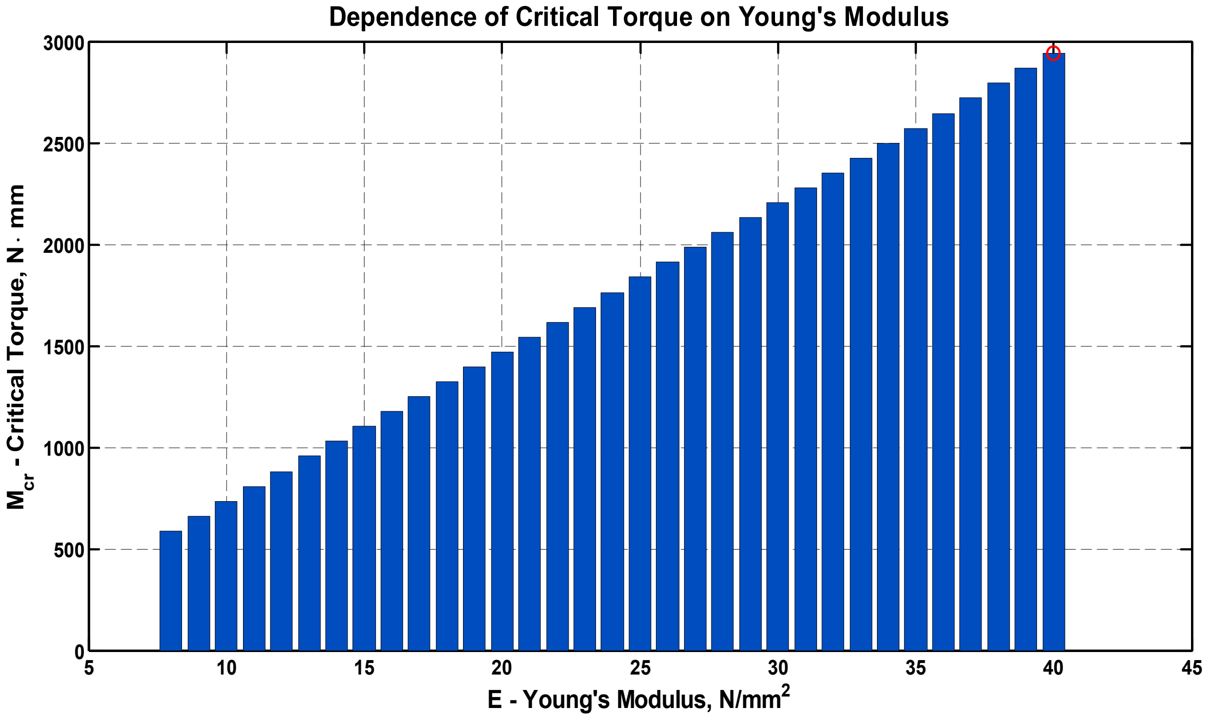

Fig. 2. Dependence of the critical torque on the elastic modulus of the feed cylinder shells in the feeding zone of pneumomechanical spinning machines

Applying as a generalized working model of the load-bearing structure its representation as a device for transmitting force dynamics having radial solutions when selecting force transmission methods.

Studying the stability of a composite cylinder under torsion using design methods makes it possible to model real structures, optimize parameters, and visualize results [10].

Conclusion: Theoretical studies of the feed cylinder with an elastic element in the feeding zone of pneumomechanical spinning machines make it possible to determine its torsion and ensure stable operation of the structure. Due to the presence of the elastic shell, the dynamic loads that occur during start-up and stopping of the feeding zone are reduced.

Straightening the material by means of the elastic shell helps to increase the efficiency of the feed cylinder [12].

It should be noted that dynamic analysis of the torsion of the feed cylinder section equipped with an elastic shell allows determining its performance under various operating conditions.

Analysis of the deformations and stresses arising during torsion of the feed cylinder with an elastic element in the feeding zone of pneumomechanical spinning machines enables more accurate calculation of the acting loads, ensures structural safety, and extends the service life of the rubber elastic shell.

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