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STUDY ON DETERMINING THE BENDING AND TORSIONAL STIFFNESS OF PACKAGED WORKING BODIES

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Abstract: The article reports on the study of bending and torsion of a package of alternating flat working and spacer disk elements, assembled by means of longitudinal compressive force. As a result, a phenomenological method for determining the bending and torsional stiffness of such structures is substantiated.

Keywords: torsion performance; bending performance; package of alternating elements; longitudinal force; compression; disk element; flat element; working element; spacer element; torsional stiffness; phenomenological method.

Introduction. The use of composite structures in technology.

In a number of technological machines across various industries, working bodies are used that represent a package of flat working and spacer disk elements, formed by longitudinal compressive force transmitted by the central shaft, which perform the main technological functions. As an example, one can cite assembled shafts of various designs, widely applied in cotton cleaning: saw cylinders of gins, linters, and lint cleaners, as well as in dyeing-finishing production of the textile industry.

Operational experience of such machines shows that, during operation, inevitable bending combined with torsional oscillations occurs, caused by random fluctuations in technological load. These not only negatively affect the technological process but may also cause loosening of the shaft's tightening nut, leading to a complete loss of compression force of the package and the creation of an emergency situation \[1]. At the same time, the central shaft is subjected to complex deformation. These circumstances indicate the relevance of studying the bending–torsional vibrations of packaged working bodies of cotton-cleaning machines.

It is clear that the bending stiffness of such working bodies is determined as the sum of the bending stiffness of the shaft and that of the disk package (or the packaged rod). Since the determination of shaft bending stiffness is well known, the subject of study can be the determination of the bending stiffness of the packaged rod. Similarly, the torsional stiffness will equal the sum of the torsional stiffness of the shaft and that of the package of saws and spacers. Since the determination of shaft torsional stiffness is also well known, the subject of research may be the determination of the torsional stiffness of the package of saws and spacers.

A solution is known for the problem of determining the bending stiffness of a component of packaged working bodies in the form of a package rod consisting of working and spacer disks clamped by longitudinal compressive force. This stiffness is defined as a function of the magnitude of compressive force and two influence functions

\[1]. The main drawback of this solution is the necessity of experimentally determining the two influence functions.

In recent years, when studying such complex objects, high efficiency has been achieved using methods of diacoptics, the finite element method, and phenomenology. Methods of diacoptics and finite elements, which are based on dividing the research object into parts, require a high degree of problem individualization or the use of very powerful computers.

The phenomenological method, which is based on studying the object as a whole, without dividing it into parts, and allows abstraction from the physics of the processes, can prove effective in this case. When performing a phenomenological analysis of the torsion of the package of saws and spacers, we neglect the physical and mechanical phenomena occurring in its volume and instead study the change in torsional stiffness of the package as a function of the compressive force \[2,3].

From our preliminary theoretical and experimental studies, confirmed by other authors, the following is known:

1. The torsional stiffness steadily increases with increasing compressive force. At the same time, the numerical value of stiffness asymptotically approaches the calculated value defined for a package operating as a monolithic body. In this sense, a monolithic package could, for example, be created by gluing the saws and spacers together.

2. The rate of increase of both bending and torsional stiffness decreases steadily from a maximum value at zero compression, asymptotically approaching zero.

3. Based on experimental results, or at least extrapolating them, one can always determine the value of compressive force N_0 at which the bending and torsional stiffness of the package differs from its calculated monolithic value by no more than a predetermined small finite quantity.

4. The practical ranges of stiffness increase and the decrease in the rate of growth are limited by technical conditions and the mechanical parameters of the packaged working body.

Analyzing the above, we can conclude that the rate of increase of the torsional stiffness of the saw–spacer package, at appropriate compressive forces, can be considered in the first approximation proportional to the compressive force.

It should be noted that the compressive force $\left[0,\, N_0\right]$ increases linearly over the entire range of its variation. Taking this into account, we may, in rough approximation, accept that the value of the compressive force in this range equals the half-sum of its values at the boundary points of the range, i.e.:

$$N = \frac{O + N_0}{2} = \frac{N_0}{2}$$

II. Discussion.

Based on these assumptions, we can conduct a mathematical study and develop mathematical models of the processes of changes in bending and torsional stiffness of the saw–spacer package depending on variations in the compressive force of the package. According to the accepted positions and assumptions, we can assume that the increment of bending and torsional stiffness with a change in compressive force is proportional to the difference $D_{pn}-D_n$, $H_{pn}-H_n$ and inversely proportional to its magnitude $\frac{N_0}{2}$ i.e..

$$dD_n=\frac{2(D_{pn}-D_n)}{N_0}dN,\ dH_n=\frac{2(H_{pn}-H_n)}{N_0}dN \quad (1)$$

Here: dD_n – increment of the bending stiffness of the package rod;

D_{pn} – bending stiffness calculated for the monolithic package rod;

D_n – current value of the bending stiffness of the package rod.

dH_n – increment of the torsional stiffness of the saw–spacer package;

H_{pn} – torsional stiffness calculated for the monolithic saw–spacer package;

H_n – current value of the torsional stiffness of the saw–spacer package.

The influence of all other unaccounted factors, also in the first rough approximation, is determined by a phenomenological function of unaccounted factors A_k, which for now is taken as constant.

$$dD_u=\frac{2A_u(D_{pn}-D_n)}{N_0}dN,\ dH_k=\frac{2A_k(H_{pn}-H_n)}{N_0}dN \quad (2)$$

The solution to the obtained equation has the form:

$$D_u=D_{pn}\left(1-e^{-\frac{2A_uN}{N_0}}\right)=D_{pn}\eta_u,\quad H_k=H_{pn}\left(1-e^{-\frac{2A_kN}{N_0}}\right)=H_{pn}\eta_k \quad (3)$$

Here: η_u - the phenomenological function of bending stiffness of the package rod.

η_k - the phenomenological function of torsional stiffness of the saw–spacer package.

The expression for bending stiffness and torsional stiffness of a package consisting of alternating working and spacer disks, and functioning as a monolithic body, takes the following form:

$$D_{pn}=\frac{(l_p+l_n)E_pJ_pE_nJ_n}{l_nE_pJ_p+l_pE_nJ_n},\quad H_{pn}=\frac{(l_p+l_n)G_pJ_{pp}G_nJ_{pn}}{l_nG_pJ_p+l_pG_nJ_n}.$$

Here: l_p, l_n – thicknesses of working and spacer disks;

E_p, E_n, G_p and G_n – elastic moduli of materials of the working and spacer disks;

J_p, J_n, J_{pp} and J_{pn} – axial and polar moments of inertia of the cross-sections of working and spacer disks.

Now, based on the last formula and solution, we can write expressions for determining the bending stiffness of the package rod and torsional stiffness of the saw–spacer package as follows:

$$D_u = \frac{(l_p + l_n)E_p J_p E_n J_n}{l_n E_p J_p + l_p E_n J_n}\left(1 - e^{-\frac{2A_u N}{N_0}}\right) = \frac{(l_p + l_n)E_p J_p E_n J_n}{l_n E_p J_p + l_p E_n J_n}\eta_u$$

$$H_k = \frac{(l_p + l_n)G_p J_{pp} G_n J_{pn}}{l_n G_p J_{pp} + l_p G_n J_{pn}}\left(1 - e^{-\frac{2A_k N}{N_0}}\right) = \frac{(l_p + l_n)G_p J_{pp} G_n J_{pn}}{l_n G_p J_{pp} + l_p G_n J_{pn}}\eta_k$$

III. Conclusions:

As a result of the research, we obtained a phenomenological method for determining the bending and torsional stiffness of the component of a packaged working body in the form of a package of flat disk elements, in which the number of experimentally determined influence functions has been reduced to one.

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