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EQUILIBRIUM ANALYSIS OF FLAT ELEMENTS OF THE SAW WORKING ELEMENT PACKAGE

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Abstract: In the article, one of the most important results is that the magnitude of the bending rigidity of the saw working element package is a function of only the magnitude of the compression force of the package and the geometric dimensions of the contact areas of the flat elements, but does not depend on the mechanical properties of the materials of the rod elements. Here, a certain analogy can be traced with the linear rigidity of a flexible thread, determined only by the magnitude of its stretching force and also independent of the properties of the thread material.

Keywords: saw working element package, disk element, force factors, tensile force, equilibrium conditions, bending deformation, flexible package, moment of force factors, reaction moment, rigidity, the Component designs, power factor, carrying element, worker organ, mechanical parameters, parameters to acerbity.

Introduction. Note that due to the fact that the main load-bearing element of the saw working element package is a flexible thread that works only in tension, when it is bent, the deformation of the geometric axis of the thread and the displacement of all elements in the plane of action of the bending moment greatly exceeds the linear deformations and displacements of flat elements in the direction of the longitudinal axis of the rod and their angular deformations and displacements relative to the neutral axes of the elements [1]. Therefore, to simplify the issue, we will neglect the longitudinal deformations and angular displacements of flat elements. Then the flat elements will have only linear displacements in the plane of action of the bending moment.

The theoretical analysis of the saw working element package operation will be performed under the following additional assumptions.

1. The saw working element package consists of flat disk elements, which do not necessarily have the same thickness. The tightening element is an absolutely flexible thread, threaded through the holes in the disk elements, and its axis, in the absence of bending deformation of the saw working element package, coincides with the coordinate axis Z.

2. All force factors are applied to the saw working element package in its plane of symmetry of the flat disk elements.

3. The greatest thickness of the flat disk elements, the transverse dimensions of the holes in them and the flexible thread are small values compared to the length of the saw working element package and the transverse dimensions of the disk elements.

The equilibrium conditions, taking into account the above and the adopted basic assumptions of one of the flat elements, are shown in figure 1.

Discussion. A flat element with a small thickness dz experiences pressure from other elements located on both sides of it. Under the considered conditions, the pressure values and their distribution on the contact areas on both sides are equal and opposite in sign. Therefore, they completely balance each other and are not shown in the figure [2,3].

In the flexible thread, there is a tensile force N(θ), where: θ is the angle of inclination of the vector N to the OZ axis.

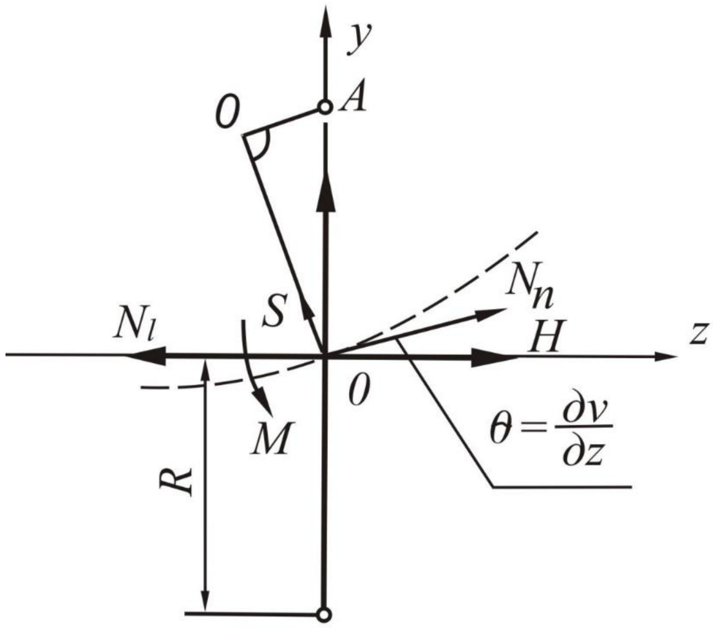

Figure 1. Equilibrium conditions for one plane element.

For simplicity, we assume that on the left the magnitude of the force is equal to the horizontal component of the tensile force of the thread, equal to the nominal tensile force of the thread in the absence of bending $N_l = -N$ and is directed along the OZ axis, and on the right -$-N_n = N/cos^{\theta}$, where $\theta = dv/dz$is the angle of rotation of the cross section of the flexible thread when its geometric axis is bent due to the bending deformation of the flexible stack rod. Since the length dz is small, we can assume that all of them are applied at the intersection point of the geometric axis of the thread and the plane of symmetry of the flat element O [4,5].

Then the equilibrium conditions of the linear force factors in projections on the OZ axis have the form:

$$\sum Z = 0 \quad \text{or} \quad H = Ncos\frac{\partial v}{\partial z} \tag{1}$$

From the figure it can be seen that the magnitude of the horizontal component of the tensile force of the thread can be determined as follows:

$$H = Ncos\frac{\partial v}{\partial z} \tag{2}$$

From (2) it follows that in the absence of external force factors and deformations $H = N$, i.e., the compression force of the package elements is equal to the tensile force of the flexible thread. When deformations appear, $N > H$ is always, i.e., the tensile force of the flexible thread is greater than the compression force of the flat elements of the package, the value of which is constant. Thus, when working on bending of the package of the saw working element, the tensile force of the flexible thread becomes greater than the constant compression force of the package [6,7,8].

For the projections of these same forces on the coordinate axis OY we have

$$\sum Y = 0$$

or

$$T = Htg\frac{\partial v}{\partial z} = nsin\frac{\partial v}{\partial z} \tag{3}$$

where T is the resultant frictional force on both contact surfaces that prevent the flat element from moving relative to adjacent elements under the action of the vertical component of the tensile force of the thread [9,10].

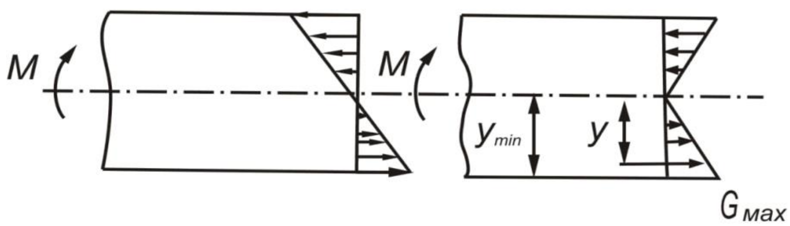

Figure 2. Picture of the work of the saw working element package on bending.

Let us turn to the conditions of balancing the moment force factors (Figure 2). As follows from Figures (2) and (3), the external bending moment M tends to rotate the flat element counterclockwise around the horizontal axis passing through point O on the geometric axis of the flexible thread and parallel to the coordinate axis OX and bend the saw working element package with its convexity downwards. The action of the external bending moment is balanced by the moment relative to point A – the extreme upper point of contact with the flat element located to the right of the component of the longitudinal force on the flexible thread, directed along the tangent to its geometric axis and determined by

$$S = N\frac{\partial v}{\partial z}$$

where S is the normal component of the tensile force of the thread.

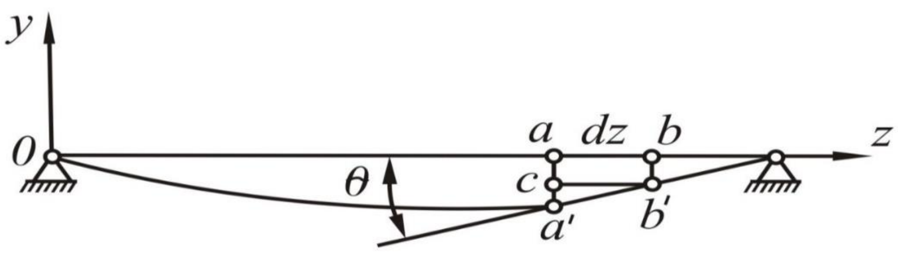

Figure 3. Conditions for balancing moment force factors.

From the same figure, we can determine the vector arm relative to point A as follows:

$$AD = R\frac{\partial v}{\partial z} \tag{4}$$

Knowing that the unknown bending stiffness of a flexible composite rod can be determined as follows:

$$C = \frac{MR}{\varepsilon_{max}}$$

Now, substituting (4) into the above formula, we obtain an expression for determining the bending rigidity of the saw working element package in the following form:

$$C = \frac{NR^2\left(\frac{\partial v}{\partial z}\right)^2}{\varepsilon_{max}} \tag{5}$$

Since we assumed the smallness of the proper longitudinal deformations of flat elements, it can be written in the form:

$$\varepsilon_{max} = \varepsilon = \frac{\partial w}{\partial z}$$

Then (5) can be rewritten as:

$$C = \frac{NR^2\left(\frac{\partial v}{\partial z}\right)^2}{\frac{\partial w}{\partial z}} \tag{6}$$

To simplify the expression obtained, we define the relationship between the angle of rotation of the cross sections $\frac{\partial v}{\partial z}$ and the longitudinal deformations $\frac{\partial w}{\partial z}$. Let us refer to figure 3.

Let us assume that the element of the deformable fiber of the saw working element package had positions *ab* and length *dz* before deformation.

After deformation with the displacement of each of its points strictly along the normal to the OX axis, this same element occupied the position $a\ b\ c$ with length $\frac{\partial z}{cos\theta}$, and the angle between the two positions of the element dz can be determined through the displacement in the plane of action of the bending moment, say coinciding with the vertical plane:

$$\theta = \frac{dv}{dz}$$

Then the relative longitudinal deformation of the element will be equal to:

$$\frac{dw}{dz} = \frac{\frac{dz}{coz\theta} - dz}{dz} = \frac{1 - cos\theta}{cos\theta}$$

Taking into account the smallness of the angle θ:

$$w \cong 1 - cos\theta$$

Now, moving on to the sine function of half the argument, we have:

$$\frac{dw}{dz} \approx 1 - cos\theta = 2sin^2\frac{\theta}{2}$$

Since for small values of the angle $sin\theta \cong \theta$, we can write:

$$\theta = \frac{dv}{dz}$$

Or, keeping in mind $\frac{dw}{dz} \approx \frac{\theta^2}{2}$, we have:

$$\frac{dw}{dz} \approx \frac{1}{2}\left(\frac{dv}{dz}\right)^2 \qquad (7)$$

Substituting (7) into (6) we obtain:

$$C = 2NR^2$$

Conclusions. Thus, the bending rigidity of the saw working element package can be defined as a first approximation as the doubled product of the nominal tensile force of the flexible thread or the compressive force of the flat elements by the square of the distance from the extreme point on the contact surface of the flat elements on the concave side of the bending saw working element package to the geometric axis of the flexible thread.

One of the most important results is that the bending rigidity of the saw working element package is a function only of the compressive force of the package and the geometric dimensions of the contact areas of the flat elements, but does not depend on the mechanical properties of the materials of the rod elements.

Here, a certain analogy can be traced with the linear rigidity of the flexible thread, which is determined only by the magnitude of its tensile force and also does not depend on the properties of the thread material.

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C O N T E N T S