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Angular and Linear Movement of a System with a Composite Shaft of Technological Machines

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ABSTRACT: This article provides an analysis of the new design of the support of the composite shaft of technological machines with an elastic element consisting of a saw cylinder. The angular and linear displacement of the system is considered, as well as internal force factors using the APM software, and comparison of the results of the design scheme without taking into account the elastic element.

KEY WORDS: composite shaft, support, structural diagram, non-symmetrical arrangement of masses, saw cylinder, angular displacement, linear displacement, bending moment, transverse force.

I.INTRODUCTION

Issues of reliability, strength, durability and resource are the most important in modern technology. Due to ever-increasing requirements for speed, efficiency, reliability and to reduce the weight of machines, strength calculations are becoming more and more complex. They must take into account various operating modes, real properties of materials, loading conditions, technological, operational and other factors.

Structures of bearings for rotating shafts.
The disadvantage of the existing supports as part of any mechanisms and machines is the direct transmission of the vibrations of the rotating shafts in the bodies of machines and mechanisms to the bodies themselves, which leads to an increase in the vibration noise of the corresponding machines and mechanisms. In addition, the design does not allow parallel displacement of the shaft axis with vertical deformations of the supports with an asymmetrical arrangement of masses on the shaft, that is, the center of mass of the shaft is not in the middle along the length of the shaft. This leads to a violation of the movement of the machine due to violation of technological gaps.

In [1] it is noted that the bearings in which the bearing mates directly with the housing and the connecting surfaces of the rolling bearing to the housing are the outer diameter and width of the rings.

In another well-known design, the bearing support of the shaft contains a housing with a bearing mounted in it and an elastic element of variable cross section of an oval shape located between the outer surface and the housing. In this case, the major axis of the outer oval surface and is installed in the housing so that the axis of minimum rigidity coincides with the direction of the loading force [2].

The support structure for absorbing vibrations of rotating shafts comprises a housing with a bearing mounted in it and an elastic element made in the form of a sleeve placed between its outer surface and the housing. The bushing is made, for example, of rubber of circular cross section, while the axis of the bushing hole is displaced relative to its central axis in the direction opposite to the direction of action of the resultant force, loaded by no more than 15% of the inner radius of the bushing [3].

In the following design of the support for absorbing vibrations of rotating shafts, the elastic element is located in the housing and is made in the form of truncated conical bushings installed in such a way that the smaller base of the bushings

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is directed towards the outer surface of the housing, and the large diameter base is towards the inner surface of the housing. At the same time, the design allows the absorption of some axial vibrations of the shaft. But, this design also does not provide parallel displacement of the shaft axis due to the non-symmetrical arrangement of masses on the shaft (when the center of mass of the shaft is located outside the center of the shaft along its length) [4].

The disadvantage of the existing supports as part of any mechanisms and machines is the direct transmission of vibrations of the rotating shafts in the bodies of machines and mechanisms to the bodies themselves, which leads to an increase in the vibration noise of the corresponding machines and mechanisms. In addition, the design does not allow parallel displacement of the shaft axis with vertical deformations of the supports with an asymmetrical arrangement of masses on the shaft, that is, the center of mass of the shaft is not in the middle along the length of the shaft. This leads to a violation of the movement of the machine due to violation of technological gaps. In addition, the complexity and high costs in the manufacture of the structure, as well as the impossibility of ensuring parallel displacement of the shaft axis with an asymmetric arrangement of the masses of the shaft honors.

II. SIGNIFICANCE OF THE SYSTEM

Development of a new shaft support design.

The proposed design of the support provides a reduction in shaft vibration by parallel displacement of the axis of the shaft along the vertical with an asymmetric arrangement of masses on the shaft along its length is an urgent task. To dampen oscillations of rotating shafts, a new design of the support is proposed, containing a housing with a bearing mounted in it and an elastic element placed on its outer surface, while the thickness of the elastic elements is chosen in proportion to the distances from the bearing supports to the center of mass of the shaft along its length, while the ratio is chosen (Fig.1.):

$$\frac{l_1}{l_2} = \frac{D_1 - D}{D_2 - D} \tag{1}$$

where, l_1- distance from the first (right) bearing support to the center of mass of the shaft;
l_2- distance from the second (left) bearing support to the center of mass of the shaft;
D_1, D- diameters, respectively, of the outer and inner surfaces of the elastic support of the first bearing support;
D_2- diameter of the outer circumference of the second support.

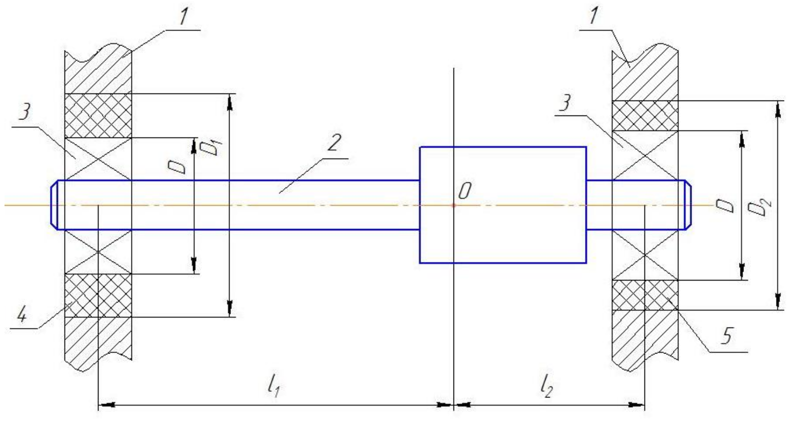

Fig 1. Construction of a support for absorbing vibrations

The recommended support for absorbing vibrations of the shafts with an asymmetric arrangement of the parts of the shaft and the working body along its length ensures parallel movement of the shaft axis during the operation of the machine. This provides the required technological clearances in the machine.

Support for absorbing vibrations of rotating shafts contains a housing in which fixedly elastic bushings with different thicknesses are installed. In this case, the inner diameters D of the elastic bushings are made the same, and the outer diameter of the elastic bushing is made larger than the outer diameter of the elastic bushing. The shaft is installed in the elastic bushings by means of bearings. The shaft is not symmetrical along its length and therefore the center of mass of

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the shaft is at point O, which is located at a distance from the support with the elastic sleeve l_1, и at a distance l_2, from the support with an elastic sleeve. In the process of working with an asymmetric distributed mass, forces act along the length of the shaft: weights, inertia, friction, etc. (not shown in the figure). Note that mainly due to the force of weight, the elastic bushings are deformed by different amounts. Their deformation corresponds to the thickness of the elastic bushings. Therefore, with an asymmetrical arrangement of masses on the shaft during operation, its axis moves parallel vertically, ensuring the absorption of peak load values on the shaft. This is ensured by the fulfillment of condition (1).

III. LITERATURE SURVEY

In order to study the effect of technological resistance on the deflection of the mass working shaft, we take for calculations the shaft of the saw cylinder of the fiber separator. The design of which is shown in Fig.2.

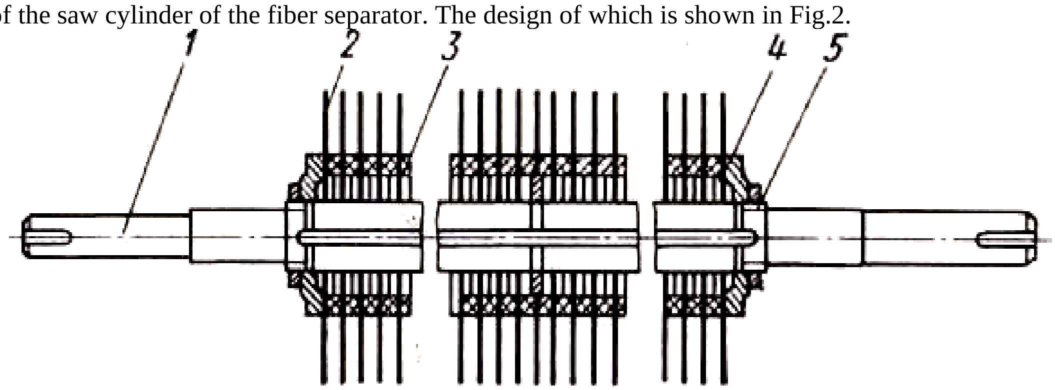

Fig. 2. General view of the genie saw cylinder
1 saw shaft, 2 saw blades, 3 saw spacers, 4 washers,
5-clamp nuts (right and left).

The saw cylinder of the genie is designed to capture the fibers of the fly fibers with the teeth of the saw blades, tear them off from the seeds and carry them out through the slotted gaps in the grate to the air-removing device. In addition, simultaneously with the separation of the fiber, the saw cylinder, coming into contact with the raw roller on the arc of fiber capture into the working chamber, rotates it, which creates conditions for the constant supply of fresh air to the saw blades. The following technological requirements for the saw cylinder have been established: the saw cylinder must have a high gripping ability to ensure the specified performance and uninterrupted rotation of the raw roller; saw blades must be rigidly fixed on the shaft of the saw cylinder, do not change their position during operation. When the cylinder rotates, the saws pass strictly in the center of the slotted gap between the grates. One end of the saw shaft is closed with a safety sleeve, and the other end is connected to the motor shaft through a semi-rigid coupling. Along the entire working length of the shaft, a groove is milled, which includes the tongue of the saw blade, which protects the saw from turning. In the middle of the working length of the saw shaft, a fixing washer is mounted, from which saw blades are placed on both sides [5].

In [6], for a theoretical calculation of the effect of technological resistance (density and mass of raw roller, machine productivity) on the process of deformation of the saw cylinder shaft, a calculation was made, which consists of several stages:

-calculation for the bending of the saw cylinder shaft (the installation point of the saw blades will not be fixed).

-calculation of the bending of the saw cylinder, taking into account the saw blades and gaskets (in a static position).

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IV. METHODOLOGY

Calculation of the deflection of the composite mass shaft of technological machines.

We are interested in the question of the deflection of the composite mass shaft, as well as the vibration of the shaft by a parallel displacement of the axis of the shaft along the vertical with an asymmetric arrangement of the masses on the shaft along its length.

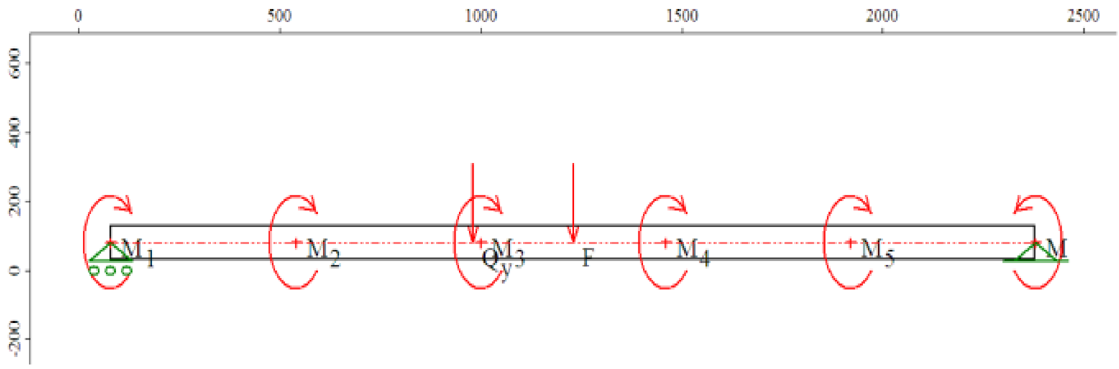
In order to make a mathematical description of the calculation object and, if possible, simply solve the problem, in the calculations, real structures are replaced by idealized models or calculation schemes. In this case, the calculation becomes approximate; using this method, the calculation of the saw cylinder for bending was called.

In software products of the APM line, the implementation of the FEM method is reduced to the analysis of the displacements of nodal points. The mathematical formulation of the method is based on the Lagrange variational principle, i.e. the principle of minimum potential energy of the system. The unknowns here are the displacements of the nodal points, and the stresses are determined by numerical differentiation of the displacements. The result of the SSS calculation are the components of the displacement of nodal points along three coordinate axes and the angles of rotation relative to these axes, the components of normal and shear stresses, and the principal stresses.

The loads acting on the mechanical system are given in the form of force factors (concentrated and distributed forces and moments), kinematic displacements and temperature fields. Thermal action, leading to the appearance of thermal stresses and thermal deformations, can be analyzed both separately and in combination with mechanical action.

When analyzing the static strength, the complex stress state of an arbitrary volumetric element is reduced to a uniaxial equivalent tension. With the help of APM software products, using the Mises, Mohr, Drucker-Prager reduction methods, it is possible to determine the largest shear stresses, the largest principal stresses, the largest deformations, etc.

On fig.3. A diagram of the saw cylinder shaft is proposed, taking into account saws and spacers in an equally distributed version when calculating which the transverse force is distributed along the length of the shaft upward, and the bending moment in a flat form, this occurs directly under the influence of an equally distributed force (g - saw blades and spacers (weight)).



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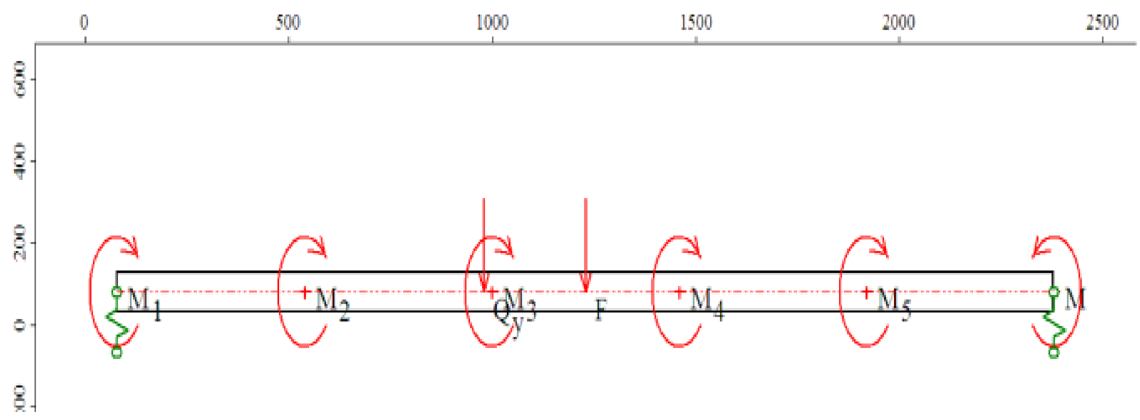

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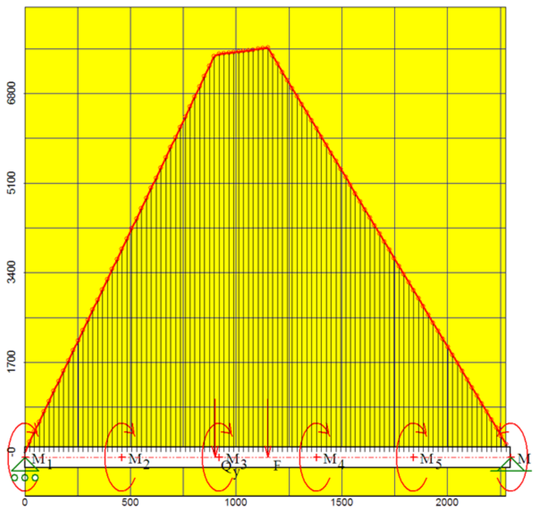
Fig. 3. Calculation scheme of the saw cylinder shaft:

a) design with serial bearings; b) construction with composite bearing support.

Since the static calculation is performed both for the considered object as a whole, and for each point of its arbitrary cross section. The calculation results are presented in the form of diagrams.

V. EXPERIMENTAL RESULTS

Bending moment in vertical plane



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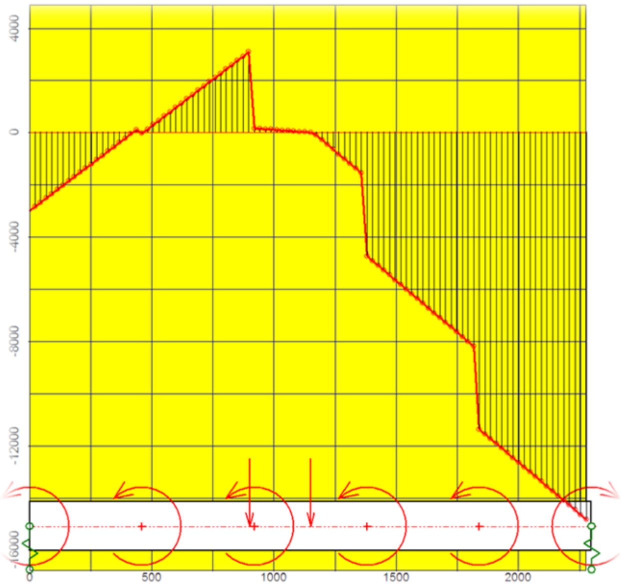

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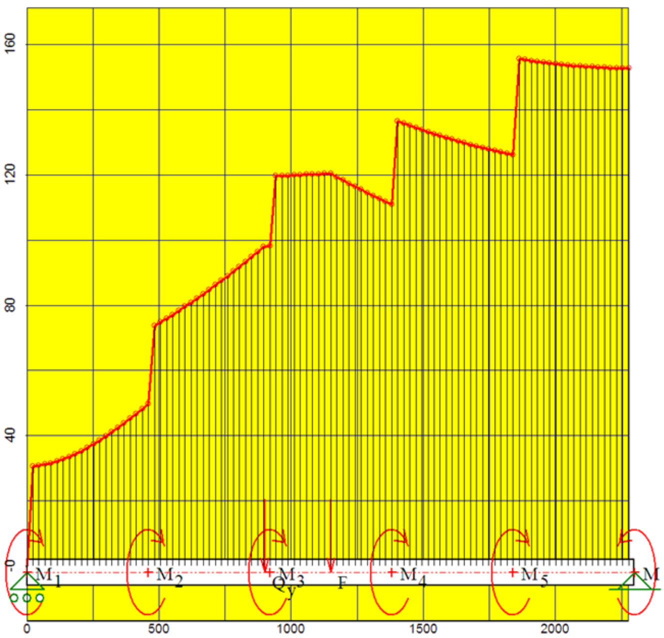
Fig.4. Diagram of the bending moment of the saw cylinder shaft:
a) design with serial bearings; b) construction with composite bearing support.

Equivalent strain



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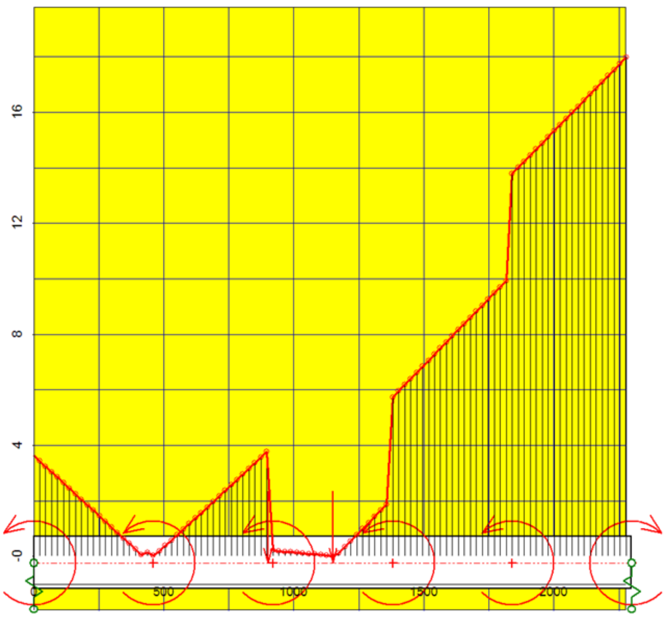

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Fig.5. Saw cylinder shaft voltage diagram:

a) design with serial bearings; b) construction with composite bearing support.

The results of processing the obtained solutions with variations in technological resistance show that with a change in technological resistance (mass of raw roller), the reaction forces on the bearings increase linearly, the growth of which directly affects the bending moment of the cross section and the angular displacement of the shaft.

VI. CONCLUSION AND FUTURE WORK

The use of the recommended shaft support will allow, due to the absorption of vibrations of rotating shafts, to reduce the load on the bodies of machines and mechanisms, therefore, the deformable characteristics of technological machines and mechanisms are significantly reduced.

The proposed support can be recommended for use as a vibration-absorbing support for a bearing in the main machines of the cotton processing industry (cotton and fiber cleaners, separators, linters, roller and saw gins, etc.).

Using the APM software, a numerical method for calculating the bending moment of a system consisting of a saw cylinder is proposed, and a calculation scheme is considered taking into account a composite bearing with an elastic element. The deflection of the genie saw cylinder shaft was analyzed taking into account the mass of the raw roller. Plots of loading of the saw cylinder shaft are constructed.

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