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Full factorial experiment in research the parameters of a combined shaft of technological machines

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Abstract. This article provides the results of a full-factorial experiment for a combined shaft with an elastic support. The influence of the main parameters of the shaft on the bending vibrations of the shaft is considered, as the input parameters are the thickness and hardness of the elastic element, as well as the distance from the resultant force to the supports.

1 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 weight reduction 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 [1].

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 [2] it is noted that bearing mating directly with the housing and the connecting surfaces of the rolling mechanism are experiencing more pressure on 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 is installed in the housing so that the axis of minimum rigidity coincides with the direction of the loading force [3].

The article presents a constructive diagram of a new saw cylinder and the results of a study of the bending of a gin saw cylinder, determining how to find the deflection and

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technological gaps, especially between the saw and grates of the gin and linter, within the limits determined by the technological regulations for ginning and lintering. The process of bending oscillations of the saw cylinder, between which there is a package of working steel saws and aluminum spacers, compressed by the longitudinal compressive force imparted by the central shaft, is considered. The results of the calculation of the bending of the spline shaft, dusty gin cylinder are presented [4].

Results of studies on extending the time operation of gin and linter grates by hardening of the working area by high-frequency current are given in article. Mathematical models for calculation of key parameters of the mode of hardening of the working area of gin and linter grates are made. As a result of the executed researches the following conclusions are drawn: change of a gap between grates in a grate lattice leads to change of parameters of process of ginning and finally to deterioration of products. Mathematical models for calculations of parameters of process of hardening of the working area of grates are received. On the received mathematical model's key parameters of hardening process current of high frequency which have the following values are calculated: depth of the tempered layer - 2.0 mm, the optimum frequency of current-60000, specific power - 500 W/cm^2, hardening time - 10, studying hardness tempered and not tempered sites of grates by method of Rockwell showed increase in hardness of the tempered site to HRC 55 against HRC 45 at not tempered by the working area [5].

In technological machines, elastic elements, in particular rubber, are often used to absorb dynamic loads. The structural scheme and the principle of operation of the sawmill cantilever grate installed in the housing with the help of a rubber gasket, consisting of two rubber elements with different rigidity, were presented. The results of the study show that when sawing raw cotton, saws hit the working part of the chimneys, causing blockages, which leads to fiber damage, as well as a decrease in the overall quality of the fiber. In this case, the total stiffness was obtained with a non-linear characteristic. Mathematical model and its solution by an approximate analytical method. As a result of the numerical solution of the problem, graphical dependencies were constructed and, based on their analysis, the required values were recommended [6-8].

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. 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 [9].

This paper introduces a new solution method based on modal analysis to calculate periodic nonlinear response to harmonic forcing. The method can be used for piecewise linear (PL) elastic multi-degree-of-freedom (MDOF) systems with two linear states, provided that the unloaded equilibrium position of both states coincides, and the system is in both states once during a period of forcing. The method formulates a linear system of equations, which yields the initial conditions of the periodic response as a function of the time spent in each state and the forcing phase. Error functions are constructed to secure the switching between the states of the system at the given time instances and the nonlinear equations are solved for the candidates of periodic responses with the scanning of the parameter space. Finally, the vibrations with unwanted switches are filtered out to obtain the actual periodic responses. An example with support vibration shows that the proposed method is capable to find disconnected branches of the periodic responses to a harmonic excitation. As the dimension of the scanning space is independent of the discretization, the method scales almost linearly with the increase of the number of degrees-of-freedom [10].

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 [11, 12].

2 Objects and methods of research

2.1 Development of a new shaft support design

In [4], the authors considered the design scheme of the saw cylinder and the results of the study of the bending of the shaft of the gin saw cylinder, determining, how to find deflection and technological gaps, especially between saw gin saws and a grate installed by technological regulations. The process of bending vibrations of saw blades of a gin saw cylinder, consisting of a package of steel saw blades and aluminum spacers, compressed by a longitudinal compression force transmitted by a central shaft, is considered. It is proposed that if the external loads are known, then when calculating the internal force factors in the sections, the shaft should be considered as a beam pivotally fixed in rigid supports. Such a model of the shaft shape and fastening conditions is close to reality for shafts rotating in rolling bearings. A conditional support for shafts resting at the ends on plain bearings is located at a distance from the inner end, but not further from the inner edge of the bearing.

The displacement of the support from the center of the bearing towards the inner end is connected with a shift in this direction of the maximum contact pressure due to deformation of the shaft and bearing [13]. In the refined calculation, one should take into account the distribution of pressures along the contact length of the trunnion and the bearing, considering the elastic contact of the shaft and the bearing through a conditional contact layer. Loads from disks, pulleys, gears and other parts are also transferred to the shafts through the contact areas. The distribution of pressures (stresses) in the contact zones depends on a number of design and technological factors, and the calculation of these pressures in connections and transmissions is associated with significant mathematical difficulties.

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.):

$$\frac{l_1}{l_2} = \frac{D_1 - D}{D_2 - D} \qquad (1)$$

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.

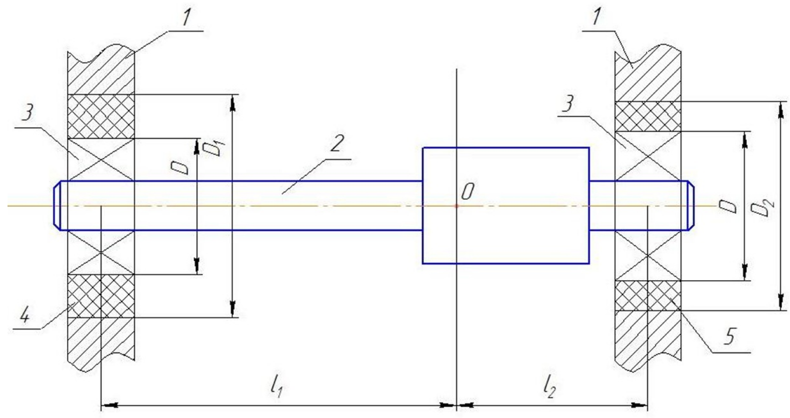

Fig. 1. Construction of a support for absorbing vibrations

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 [11].

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 the shaft is at point O, which is located at a distance from the support with the elastic sleeve l_1, 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).

2.2 Theory of the full factory experiment

For an experimental study and optimization of the parameters of the combined shaft, a full factorial experiment was carried out, in which the bending vibrations of the shaft were taken as the output parameter, taking into account the following input parameters:

- rigidity of the elastic element;
- operating forces of technological impact;
- distances of the generalized acting forces from the shaft supports.

In our opinion, three main factors should influence the normal and efficient flow of the process of rotating shafts. Therefore, when conducting research, a full factorial experiment 2^3 was chosen.

The implementation of a full factorial experiment includes:

- construction of an experiment planning matrix;

- conducting experiments by the planning matrix;
- processing the results of experiments, and obtaining regression models;
- analysis of regression models.

All the identified main factors vary at two levels (+1 and -1), and the number of experiments is 2^3=8. The planning matrix for the experiments is presented in Table. 1. After selecting the main factors and their levels of variation, it was determined by what main output parameters one can judge and evaluate the work, as well as optimize the technological and design parameters of saw gin [14].

Table 1. Experiment design and working matrix PFE $2^3 = 8$

№	The procedure for the implementation of the experience	Factor					
		x_1	x_2	x_3	x_1	x_2	x_3
1	1	-1	-1	-1	0,5	70	300
2	7	+1	-1	-1	1	70	300
3	8	-1	+1	-1	0,5	90	300
4	2	+1	+1	-1	1	90	300
5	5	-1	-1	+1	0,5	70	900
6	4	+1	-1	+1	1	70	900
7	3	-1	+1	+1	0,5	90	900
8	6	+1	+1	+1	1	70	900

In table 2 levels of variation of experimental factors are given.

Table 2. Levels of variation of experimental factors

№	Factor name	Units	Designation	The meaning of factors			Levels of variations
				-1	0	+1	
1	Elastic element thickness	mm	x_1	0.4	0.77	1	0.3
2	Elastic element hardness	Shore type A	x_2	686	784	882	98
3	Distances of the generalized acting forces from the shaft supports	mm	x_3	300	600	900	300

As the output parameters of the experiment "Y," we accept the bending vibrations of the shaft. The accuracy and reliability of the results of the experiment largely depend on the accuracy of control of all input and output parameters and their constancy. Therefore, each experiment was preceded by preparation with multiple control of the input and output parameters of the saw gin. The performance of the saw gin was regulated by changing the supply of raw cotton from the gin feeder and controlled by the timing method [16].

By the theory of a full-factorial experiment, when conducting experimental studies, each of the factors varies only at two levels - the minimum (coded value -1) and the maximum (coded value +1).

In models, the value of the regression coefficients characterizes the contribution of the corresponding factor to the value of the output parameter when the factor moves from the main level to the upper or lower. The contribution of the factor in the transition from the lower to the upper level in the value of the output parameter is called the effect of the factor.

The larger the regression coefficient, the higher the effect of this factor, i.e. the stronger the effect of the factor on the output parameter. Thus, according to the magnitude of the regression coefficients in the models, the factors are ranked according to the strength of their influence on y, the sign in front of the regression coefficient determines the nature of the influence of the factor on y. Factors whose coefficients have a plus sign (+) increase the value of the output parameter, and those with a minus sign (-) reduce it [15].

3 Results and their discussion

3.1 Receiving the regression analysis formula

According to the results, the regression equation looked as follows.

$$y=0.832+0.003x_2+0.017x_2+0.024x_3-0.0299x_1x_3$$

The analysis of the presented regression equation shows that on the parameter of bending vibrations of the shaft (y), the main influence comes from the hardness of the elastic element (x_2), the distance of the generalized acting forces from the shaft supports (x_3) and the interaction of factors (x_1 x_3). All incoming factors (x_1, x_2, x_3) have a positive effect, and the joint interaction of factors (x_1 x_3) negatively affects the bending vibrations of the shaft (y). To study these dependencies, the curves were numerically calculated using the regression equation for various values of the main factors. The results of calculations after processing using the EXCEL program are presented in the form of graphs (Fig. 1.).

Fig. 2, shows the dependence of the bending vibration of the shaft on the thickness of the elastic element x1, where four curves y=y(x) are shown. Variants are considered where the curves correspond to the minimum, maximum and intermediate values of the factors x_2 and x_3. It can be seen from the graph that the bending vibrations of the shaft on the first curve, i.e. at the minimum values of the main factors $x_2=686$ and $x_3=300$ *mm*, increase from *8.0 rad/s* to *23.7 rad/s*, and at the maximum values i.e. $x_2=882$ and $x_3=900$ *mm*, increases from *23.7%* to *27.0%.* The third and fourth curves also show an increase in bending vibration from *11 rad/s* to *14.3 rad/s* and from *16.2 rad/s* to *19.5 rad/s*, respectively. The analysis of graphical dependencies shows that all variations of changes in input factors have a positive effect on the bending vibrations of the shaft.

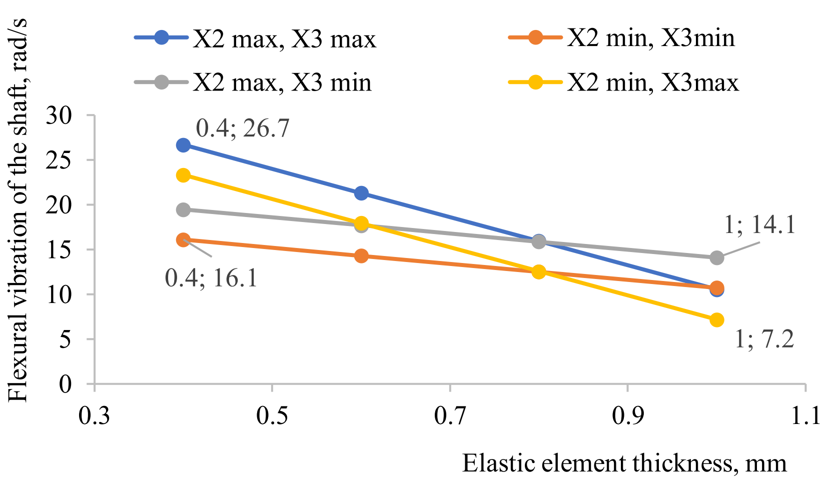

Fig. 2. Elastic element thickness versus shaft flexural vibrations

Consider the influence of the hardness of the elastic element on the bending vibrations of the shaft in the following variations of the input parameters (Fig. 3). Variants are considered where the curves correspond to the minimum, maximum and intermediate values of the factors x_1 and x_3. It can be seen from the graph that the bending vibrations of the shaft at the values of the main factors $x_1=0.4$ *mm* and $x_3=900$ *mm* increase from *8.0 rad/s* to *11.3 rad/s*, and maximum values i.e. $x_1=1.0$ *mm* and $x_3=900$ *mm*, increases from *23.7%* to *27.0%*. In the intermediate curves, there is also an increase in the bending vibration, respectively, from *11 rad/s* to *14.3 rad/s* and from *16.2 rad/s* to *19.5 rad/s*.

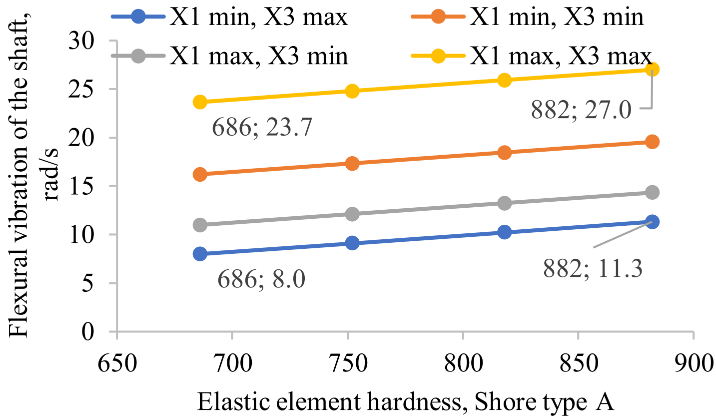

Fig. 3. Elastic element hardness versus shaft flexural vibrations

An analysis of the graphical dependences of the flexural vibration of the shaft on the distance between the generalized force and the shaft support x_3 is shown in fig. 4, where four curves $y=y(x)$ is shown, where the curves correspond to the minimum, maximum and intermediate values of the factors x_1 and x_2. It can be seen from the graph that at maximum values of the thickness of the elastic element $x_1=1.0$ *mm*, the bending vibrations of the shaft decrease from *10.7 rad/s* to *7.2 rad/s*. At the maximum values of the hardness of the elastic element on the Shore, scale type A $x_2=818$ *mm*, the bending vibrations of the shaft increase from *19.5 rad/s* to *26.7 rad/s*.

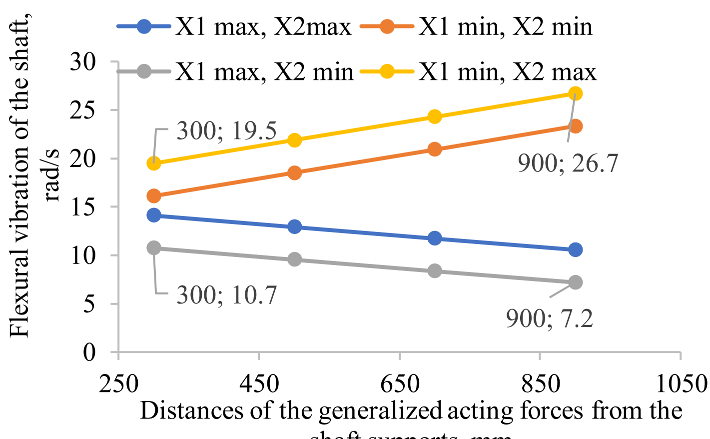

Fig. 4. Graph of the dependence of the bending vibration on the thickness of the elastic element and the distance of the generalized acting forces from the shaft supports

4 Conclusions

A full factorial experiment was carried out to study the influence of the bending vibration of a composite shaft on the hardness of the elastic element, the distance of the generalized acting forces from the shaft supports and the thickness of the elastic element. Such an approach makes it possible to take into account most of the factors affecting the objective function, by varying which it is possible to select the values of the factors that extremize the response function. Based on the results obtained, it can be concluded that the bending vibration of a composite shaft depends on the combined influences of the input factors. The recommended values of the parameters of the full-factorial experiment for the combined shaft are:

- thickness of the elastic element 0.5 mm and 0.75 mm;
- stiffness of the elastic element - 90 Shore A (polyurethane);
- distances of the generalized acting forces from the shaft supports - 600mm.

It is also assumed that during the experimental studies of the combined shaft under production conditions, it is advisable to take into account the results of the recommended values of the input factors.

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