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METHODS OF COMPUTER MODELING, APPROXIMATION AND MECHANICAL BEHAVIOR OF ROBOTIC SYSTEMS UNDER DYNAMIC INFLUENCES

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ABSTRACT

The methodology of modeling and analysis of the stress-strain state of a three-stage dynamic semi-natural modeling stand (stands) made of homogeneous material under dynamic loads is considered. Semi-simulation stands are used to simulate the flight characteristics of aircraft in ground conditions. The simulators are a complex structure consisting of moving elements connected by bearings, gearboxes, gears and engines. Accounting for the interaction of these elements is a complex task that requires modeling that is identical to the behavior of the stand elements in real conditions. One of the main parameters characterizing the effective workability of a test stand is its inertia characteristics, which allow to quickly and accurately implementing the given commands. One of the most effective and tested methods of structural studies is the finite element method. In the present study the methodology of modeling bearings, reducers, gear wheels in the stand by replacing them with systems identical in stiffness and deformability taking into account the applied research method, the method of finite elements. As a result of the study the methods of modeling a complex dynamic structure containing bearings, reducers and gearboxes. Methodology for modeling of three-layer structures consists of external bearing layers and a filler layer between them, preventing shear stresses and convergence of bearing layers. The stress-strain state of the stand under dynamic loads is investigated.

Keywords: *Method, Modeling, Approximation, Stands, Finite element method, Mechanical behavior, Dynamic loading.*

1. INTRODUCTION

The research work is aimed at developing a methodology for the dynamic calculation of robotic systems containing elements such as bearings, gear rims and gearboxes that are not adequately interpreted in numerical methods - finite element method.

The novelty of the research lies in the uniqueness of robotic systems: multi-stage semi-natural modeling stands, developed methods for identifying bearings, gear rims and gearboxes by a system of rod structures identical in rigidity and developed method for calculating structures for dynamic loads.

Robotic structures refer to complex dynamic systems containing complex surfaces, support bearings, ring gears, gearboxes and motors contributing to the movements of robotic systems channels [1, 2]. Semi-natural simulation stands (Figure 1) refer to such structures that simulate the flight characteristics of products under test in ground conditions [3]. Therefore, the simulation and approximation methods and calculations presented in this paper are applicable to robotic systems, weaving machines and, in general, to dynamic systems containing the elements described above.

Figure 1: Three-stage stand

The approximation of shell, lamellar, rod and volume structural elements is sufficiently developed and a large library of finite elements is available to adequately simulate the design of robotic systems [4, 5]. The approximation of such elements of robotic systems as support bearings, ring gears, gearboxes and motors is not well developed. The direct approximation of such structures in the robotic system cannot be implemented, and it leads to an unjustified increase in the number of finite elements and does not allow to adequately simulating the performance of such structures. This study presents a methodology for modeling support bearings, ring gears and gearboxes by replacing the represented elements with rod systems corresponding to the stiffnesses of the elements being replaced.

2. METHODOLOGY

The problem of calculating robotic systems related to dynamic systems containing bearings, gear rims and gearboxes that are inadequately interpreted in the system of numerical methods is a complex task that requires a large amount of theoretical and experimental methods in the development and manufacturing process. Therefore, the problems considered in this paper are important and relevant.

The interpretations proposed in the calculation systems of the units of robotic systems that ensure the movement of channels take into account, in the main, the rigidity in the plane of the location of the bearings and gear rims. To interpret gearboxes, it is necessary to develop a program for taking into account the stiffness of the gearbox by methods of discipline of machine parts and mechanisms, comparing the results obtained with experimental results.

We consider in more detail the methodology of approximation of support bearings, ring gears and gearboxes. For this purpose, we calculate the stiffness of such elements included in the stand.

Figure 2 shows the kinematic diagram of the gearbox.

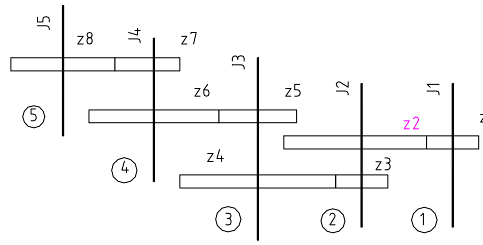

Figure 2. Pitch reducer kinematic diagram

2.1 Initial Data

Initial data moments of inertia of the shafts: J_5, J_4, J_3, J_2, J_1 (kg×m×s^2), angular velocity: υ_5 (1/s), angular acceleration: ω_5 (1/s^2), frequency: f (Hz).

Transmission ratios: $i_{54}=z_8/z_7$, $i_{43}=z_6/z_5$, $i_{32}=z_4/z_2$, $i_{21}=z_2/z_1$, $i_{11}=z_1/z_1$.

Transmission ratios referenced to the input shaft: $i_{51}=i_{54}\times i_{43}\times i_{32}\times i_{21}\times i_{11}$, $i_{41}=i_{51}/i_{54}$, $i_{31}=i_{41}/i_{43}$, $i_{21}=i_{31}/i_{32}$, $i_{11}=i_{21}/i_{11}$.

Total gear ratio: $i_{tot} = i_{54}\times i_{43}\times i_{32}\times i_{21}\times i_{11}$.

Gear ratio: $i_{gear}= i_{43}\times i_{32}\times i_{21}\times i_{11}$.

Moments of inertia reduced to the input shaft: $J_{51}=J_5/i_{51}^2$, $J_{41}=J_4/i_{41}^2$, $J_{31}=J_3/i_{31}^2$, $J_{21}=J_2/i_{21}^2$, $J_{11}=J_1/i_{11}^2$. Shaft accelerations: $\omega_4=\omega_5\times i_{54}$, $\omega_3=\omega 4\times i43$, $\omega_2=\omega_3\times i_{32}$, $\omega_1=\omega_2\times i_{21}$.

Moments on the shaft: M_5, $M_4=J_4\times\omega_4+M_5/i_{54}$, $M_3=J_3\times\omega_3+M_4/i_{43}$, $M_2=J_2\times\omega_2+M_3/i_{32}$, $M_1=J_1\times\omega_1+M_2/i_{21}$.

2.2 Shaft Load

$$P_{circ}=M_j/m\times z,\ P_{rad}=P_{circ}\times\tan 20^o, \quad (1)$$

where m is the module, z is the number of teeth, M_j is the shaft moment.

Support reactions R_1 and R_2 are determined by the methods of strength of materials taking the moment of inertia of the shaft section $J_{shaft}=3.14\times D_{eqv}^4/64$ based on D_{eqv} (equivalent constant diameter for bending).

The displacements and rotation angles at the force and support points, respectively, are determined using the BEAM program.

2.3 Deformation of the Bearing from the Fit on the Shaft and in the Housing

$$g_r = g_{r0} - \Delta d_1 - \Delta D_2, \quad (2)$$

where g_{r0} is the initial ball bearing clearance, Δd_1 is the increase in the outer diameter of the inner ring from the fit on the shaft, ΔD_2 is the decrease in the inner diameter of the outer ring from the housing fit is determined from the graphs [6], depending on the stiffness ratios of the mating parts:

- shaft (steel) $C_{shaft}=d_2/d$;
  bearing inner ring: $C_{inn}=d/d_1$;
  bearing outer ring: $C_{out}=D_2/D$;
  housing (steel): $C_h=D/D_1$,

where d_2 is the hollow shaft bore, d is the inner diameter of the bearing inner ring, d_1 is the outer diameter of the inner ring of the bearing, D, D_2 is the outer and inner diameter of the bearing outer ring, respectively, D_1 is the external diameter of the casing;

- strain and fit relations in the shaft-inner-ring connections: d_1/Hb;
  housings with outer ring D_2/Hk;
  $Hb=Hcp-H_1$, Hcp is the average value of the fit of the inner ring of the bearing, H_1 is the bump size, $H\kappa=Hcp-H_1$ (also for the bearing outer ring);
  D_1 is the increase in the outer diameter of the inner ring;
  D_2 is the reduction of the inner ring outer diameter.

2.4 Deformation of the Bearing under Load

$$\delta_r = \delta_{r1} + \delta_{r2}, \quad (3)$$

where δ_{r1} is the deformation in the contact of the most loaded rolling element with the raceway in the bearing, δ_{r2} is the radial compliance in the contact of bearing rings with the shaft and housing seating surfaces.

Deformation in the contact of the most loaded rolling element with the raceway in the bearing:

- preloaded:

$$\delta_{r1} = \beta \times \delta_{r0}. \quad (4)$$

- with radial clearance:

$$\delta_{r1} = \beta \times \delta_{r0} - g_r/2. \quad (5)$$

Coefficient β which takes into account the amount of interference or clearance in the bearing is determined from the graphs [6].

δ_{r0} for various types of bearings can be determined from the equations [6]:

- for the single-row ball:

$$\delta_{r0} = 2.0 \times 10^{-3} \times \sqrt[3]{\frac{Q_0^2}{D_T}}; \quad (6)$$

- for angular contact tapered bearing:

$$\delta_{r0} = 6.0 \times 10^{-4} \times \frac{Q_0^{0.9}}{\cos\alpha \times l^{0.8}}, \quad (7)$$

where $Q_0 = \frac{5 \times R}{i \times z \times \cos\alpha}$; R is the radial load on the support, i is the number of rolling element rows, z is the number of rolling elements in one row, $\alpha=\arccos(1-g_r/2\times A)$, α is the contact angle, D_T is the ball diameter, $d_{gr} = 0.5232 \times D_T$ groove diameter $A = D_T \times (2 \times d_{gr}/D_T - 1)$.

2.5 Radial Slack in the Contact Between the Bearing Rings and the Shaft and Housing Seating Surfaces

$$\delta_{r2} = \frac{4 \times R \times k}{\pi \times d \times B}\left(1 + \frac{d}{D}\right). \quad (8)$$

$k=0.015 mm^2/kg$, D, d, B, respectively, are the outer and inner diameters of the bearing and its width.

2.6 Calculation of Gear Rotation Caused by Elastic Deflection of the Shaft

$$\varphi_{ed} = \frac{2}{m \times Z} \times (w_{circ} + w_{rad} \times tg20^o), \quad (9)$$

where w_{circ}, w_{rad} are the values of shaft deflections in the middle plane of the transmission in circumferential and radial directions, respectively, taking into account the strain in the bearings, Z is the number of teeth, m is the gear module.

2.7 Calculation of the Error Caused by Torsion of the Shaft

$$\varphi_t = \Sigma(M \times l_i/(G \times J_p)), \text{ rad}, \quad (10)$$

where G is the shear modulus, $J_p=\pi\times(D^4-d^4)/32$ is the polar moment of inertia, l_i is the length of the twisting section, M is the moment on the shaft.

2.8 Calculation of the Error Caused by the Gear Suppleness

$$\varphi_{gear}=M/(k\times d^2\times b), \text{ rad.} \quad (11)$$

Here is denoted by: d is the pitch diameter of the ring gear, b is the working width of the ring gear, $k=368$ kg/mm^2 is the experimental coefficient.

2.9 Calculation of the Error Caused by the Keyed Connection

$$\varphi_k=M/k_k\times d_b^2\times l\times h\times z, \text{ rad,} \quad (12)$$

where d_b is the shaft diameter, l is the working length of the key, h is the key height, z is the number of keys, k_k=15kg/mm^3 for prismatic keys, k_s=25 kg/mm^3 for segmented keys.

2.10 Error Caused by Deformation of the Pin Connection

$$\varphi_{pin}=k_{pin}\times M, \text{ rad,} \quad (13)$$

where k_{pin} is the coefficient of proportionality [7].

2.11 Total Gearbox Stiffness

$$C=\left\{\sum_{j=1}^{n}\left[\frac{(\phi_{ed}+\phi_t+\phi_{gear}+\phi_k+\phi_{pin})_j}{M_j\times i_j^2}\right]\right\}^{-1},$$

$$kg\times\text{т}\times rad, \quad (14)$$

where M is the moment on the shaft, φ is the twisting angle, i is the gear ratio referenced to the input shaft.

2.12 Own Frequency of the Design

$$f=\frac{1}{\pi}\times\sqrt{\frac{C\times i_{tot}^2}{J_5}}, Hz, \quad (15)$$

where i_{tot} is the overall ratio, C is the stiffness, J_5 is the reduced moment of inertia.

Then, we replace the calculated stiffness with rod systems corresponding there and serviceability to the elements being replaced.

3. RESULTS AND DISCUSSION

As a result of the work, modeling, finite-element approximation [8-9], calculation on the dynamic load of a three-stage stand of semi-natural simulation was carried out. The general view of the stand and its finite-element approximation is shown in the Figure 3.

The simulation and study of the stand was carried out using ANSYS program with the finite element method.

Shell, planar three- and four-corner finite element, which has six degrees of freedom in each node, is the most adapted for modeling the stand structure [10, 11].

The finite element under consideration takes into account bending and membrane deformations, the material is isotropic with characteristics specified in the element plane and accepts concentrated, distributed, mass and temperature loads.

As a result of calculations, the natural frequencies and stresses in the median surface and at the boundary surfaces are determined.

The weakest link in discrediting the stand is the modeling of ball bearings. Accounting for their interaction with the structure is a challenging task. Geometrically identical modeling of ball bearings leads to an unreasonable increase in the number of equations, the error in solving which nullifies the effort of accurate modeling, so assumptions were made to account for ball bearings that allow the mechanism of ball-bearing interaction with the structure to be accounted for with sufficient reliability.

The assumptions are as follows:

- ball was replaced by a rod element having pinching at the ends, which is highly consistent with the behavior of the ball in the bearing;
- rigidity of the rod system simulating a ball bearing was taken to be equal to that of the ball bearing.

For ease of calculation, the gearboxes included in the stand were calculated separately, using the program developed, and the elastic compliance of the gearboxes was subsequently taken into account when calculating the design.

The rotary support mechanism of the stand was modeled by a set of the finite elements described above. The total number of elements was 4 250, including rod elements taking into account the bearing supports [12-14].

The accuracy of the results obtained was checked by the convergence of the results depending on the number of partitioning elements. The results of the stress-strain state at partitioning into 5 006 finite elements and the accepted scheme differed by an acceptable error for the accepted type of calculations of 2%...3%.

The program accuracy was checked by solving test problems and comparing the obtained results with known analytical solutions. The discrepancy between the results was observed in the third decimal place.

A study of the stand model has been carried out under dynamic loading, which is a sinusoidal load applied to the top of the course fork in the direction of X axis (Figure 3). When reaching the frequency of the load corresponding to the natural frequency of oscillations, there is a significant increase in the amplitude of the stand oscillations. Thus, it is possible to determine the natural frequencies of the stand oscillations. Determination of natural frequencies is important for the design of stands, because the resonance phenomena at the coincidence of natural and forced external loads can lead to a significant increase in the amplitude and to the withdrawal of the stand from the serviceable state and its destruction. Therefore, knowing the natural frequencies of oscillations of the stand we need to deduce the forced vibrations from the range of natural frequencies by either increasing the rigidity of the stand in the right direction or changing the physical and mechanical characteristics of its materials. The proposed methodology for determining the natural frequencies of vibrations of robotic structures by applying a sinusoidal dynamic load with different frequency of oscillations can be applied to arbitrary dynamic systems. The Figure 3 shows the resonance response of the stand when the external frequency of oscillations coincides with the natural frequency of it.

Comparison with the available studies in the literature is difficult in the complex of problems solved in this article: these are dynamics, moving elements and parts that provide them: bearings, gear rims and gearboxes. In computer-aided design systems, there is an interpretation of such elements, mainly in the form of nodal points, at the same time, the scale of the element and the rigidity from the bearing plane are not taken into account. There is no algorithm for calculating structures for dynamic loads.

When studying the dynamic loading of a structure, it is necessary to set the initial displacements, which can be taken from the displacement of the nodal points of the finite elements, for example, from the solution of the deformed state of the structure under static load, multiplying the displacement of the nodal points by 10-8. Carry out the approximation of bearings, gear rims and gearboxes by a system of rod structures identical in rigidity.

4. CONCLUSIONS

As a result of the work, modeling of a structure of complex geometry with surfaces of double curvature, moving elements and parts that ensure the movement of channels is carried out. A program for calculating the rigidity of gearboxes, bearing supports and gear rims has been developed. Approximation of the design model was carried out [15, 16]: multi-stage bench of semi-natural modeling with the identification of gearboxes, bearing supports and gear rims by a system of bar structures of identical rigidity. A technique for calculating the structure for dynamic loads has been developed. The calculation of the stress-strain state of the semi-natural simulation stand for dynamic effects has been carried out.

The developed program for calculating the stiffness and frequency characteristics of support bearings, ring gears and gearboxes was compared with the available experimental data. The discrepancy did not exceed 10-15%. The carried out replacement of support bearings, ring gears and gearboxes by a system of rod systems of appropriate rigidity corresponds to the stress-strain state and natural frequencies of the stands obtained from the experimental studies. Thus, confirming the legitimacy of replacing support bearings, ring gears and gearboxes by a system of rod structures of similar rigidity.

The developed methods are applicable to robotic systems that have movable channels and elements that provide them: bearings, gear rims and gearboxes. Also to analyze the stress-strain state of robotic systems made of homogeneous materials on the action of dynamic operational loads to ensure their normalized characteristics and performance.

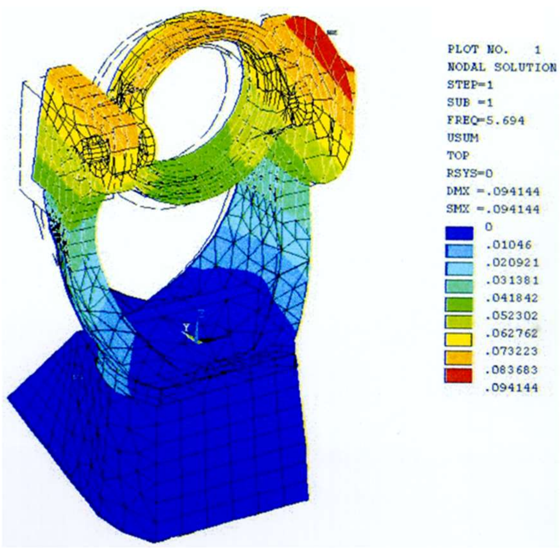

Figure 3. Finite element approximation of the stand and stand response to dynamic load

5. ACKNOWLEDGEMENTS

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