The computer modeling and analysis of complex dynamic structures of robotics from composite materials

Kamil Khayrnasov[1*], *Anton* Sokolskiy[1], and *Vladimir* Isaev[1]

[1]Moscow Aviation Institute (National Research University), 125993 Moscow, Russia

Abstract. The methods of designing and modeling dynamic stands from a composite material are considered. One of the complex elements of the stands is the elements that ensure the dynamic behavior of the channels of the stand: bearing supports, gear rims, gearboxes, motors, servo drives, etc. Approximation of these elements in numerical methods for calculating and studying dynamic structures (finite element method) is a difficult task. Direct approximation of these elements does not satisfy the adequacy. Various approaches are used to adequately approximate such elements. In this paper, a methodology is proposed that consists in determining the rigidity parameters of bearing supports, gear rims, gearboxes and motors. In the future, these elements are replaced by rod systems corresponding to the replaced elements in terms of stiffness parameters. The effectiveness of the proposed method is confirmed by a comparison with of available experimental methods. The shell structures and stands were modeled in ANSYS modeling module. The considered methods for designing stands can be applied to robotic systems, weaving machines and dynamic systems for various purposes, since they contain all the elements inherent in the considered stand. The finite element approximation of the stand was carried out by elements of various types.

1 Introduction

The purpose of this study is a method for calculating robotic systems from composite materials using the example of calculating a semi-natural simulation bench. The significance of the work is determined by the fact that, until now, stands have not been made from composite materials with higher specific strength characteristics compared to magnesium alloys traditionally used in the manufacture of stands.

For a numerical study of the mechanical behavior and stress-strain state of the stand, it is necessary to develop a methodology for designing a model in one of the automated simulation systems. In the present study, the simulation of the stand was carried out in ANSYS computer-aided design system [1], designed for modeling and analysis. As an object of study, we consider the bench shown in the Figure 1 [2] containing surfaces of double curvature, plates and shells, elements consisting of hollow structures designed to

* Corresponding author: kamilh@mail.ru

increase the rigidity and strength characteristics of the bench structure, which are one of the important characteristics of the bench performance [3-5]. The stand shown in the Figure 1 has three degrees of freedom and is designed to simulate the flight characteristics of the tested products in laboratory conditions. Support bearings, gear rims, gearboxes and motors are used as elements that facilitate the movement of the bench channels [6, 7]. The identification of such elements for numerical calculation is a complex task that requires a large amount of theoretical and experimental research [8]. In this study, a methodology is proposed for the identification of support bearings, gear rims and gearboxes with bar structures identical in rigidity to the simulated elements. Moreover, such an approximation allows taking into account the rigidity not only in the reference plane, but also from the reference plane, i.e. three-dimensional stiffness most closely corresponding to, for example, the stiffness of a support bearing.

The stand is made of a three-layer composite material containing a filler of a light composite material in the form of a stand model and carrier layers obtained by modeling on the surface of the stand model. These shell structures are assigned the characteristics of a five-layer composite material. To analyze the failure of a five-layer composite material (carrier layers), maximum stresses, maximum deformations, Tsai-Wu, Tsai-Hill, and Hoffman criteria for the destruction of a multilayer composite material were used.

2 Materials and methods

2.1 Equation

To solve the problem, we use the Lagrange equations. The Lagrange equations make it possible to obtain the stress-strain state of robotic systems under dynamic influences [3–7]. To do this, it is necessary to determine the kinetic, potential energy of the system under consideration and the work of external forces:

$$\frac{\partial}{\partial t}\frac{\partial T}{\partial \dot{q}_k}+\frac{\partial U}{\partial q_k}=Q,$$

where T is the kinetic energy of deformation, U is the potential energy of deformation, q is the vector of generalized displacements, the index k is the number of degrees of freedom, t is time, the dot above the letter means differentiation with respect to time.

2.2 Relationship between stresses-strains

In the present study, the stand material is a composite material [8,9] There are two approaches to identify the relationship between stresses and strains for a multilayer composite material, when the base of the layers is located at different angles: the method of reduced stiffnesses and taking into account the dependence for each layer separately. The reduced stiffness method used in this study calculates the generalized characteristics of a multilayer composite material characteristic of a homogeneous material. In the method under consideration, the relationship between stresses and strains for each layer of a multilayer composite material, the number of resolving equations depends on the number of layers of the multilayer composite, which leads to an increase in the resolving equations as for the method of reduced characteristics multiplied by the number of layers of the multilayer composite. This approach leads to an unreasonably large number of resolving equations, an increase in computational errors, and is practically not used when considering large structures, which also includes a multi-degree bench for semi-natural simulation.

2.3 Method for determining the structure of a composite material

The multilayer composite material changes its characteristics depending on the location of the warp of the composite yarns. To obtain the maximum rigidity of a multilayer composite material, it is necessary to position the base of the composite material in the direction of the maximum stresses acting in the structure under study under the action of operational loads. In this case, part of the layers in the multilayer composite material should be made at an angle to the maximum stresses to absorb shear stresses. The ratio of the arrangement of layers in a multilayer composite material is a complex task that requires a large number of theoretical and experimental studies. And it depends on the structure under study and the operating loads.

This study proposes an iterative method for arranging layers of a multilayer composite material along lines of maximum stresses. At the first stage, a model of a structure made of a homogeneous material under the action of operational loads is investigated. As a result of the study, we obtain the trajectories of maximum stresses. At the second stage, the structure model is made of a composite material with the arrangement of layers of a multilayer composite material along the lines of maximum stresses. The trajectories of maximum stresses obtained as a result of the studies make it possible to correct the location of the composite base and carry out the calculation of the structure. In this way, the location of the base of the composite material along the lines of maximum stresses is achieved and, thereby, a model of the structure of maximum strength and rigidity is obtained. Layers with an angle to the trajectories of maximum stresses for the perception of shear stresses are determined from the results of the analysis of the stress-strain state of the structure.

2.4 Modeling technique for gear rims, gearboxes and bearings

In this study, a method for approximating such elements of robotic systems as bearings, gear rims and bearing supports is proposed. The approximation technique is as follows. The rigidity of the specified elements is determined on the basis of the developed program using the formulas of the mechanics of machines and mechanisms. Next, a bar structure of the corresponding stiffness is modeled and the ring gear, bearing or ring gear is replaced in the model with a bar structure of the same stiffness. The performed calculations and studies have shown the validity of this technique. The results of comparison with the available experimental studies of showed good agreement.

2.5 Modeling and approximation of the stand

The stand is a three-layer shell structure, consisting of a stand model made of lightweight composite material (Table 1) with outer load-bearing layers of a five-layer composite material with a layer orientation of 0/45/0/-45/0 degrees. The stand was approximated by finite elements. Gear rims, bearing supports and gearboxes were approximated by bar structures of appropriate stiffness. The convergence of the results was determined by condensing the number of finite elements: the structure model was divided into n finite elements, the calculation was carried out, and the results obtained were compared with the results obtained by splitting into $2n$ finite elements. If the results differed by no more than 3%, splitting into n finite elements was considered sufficient to obtain acceptable results.

2.6 Algorithm for determining stiffness

We determine the rigidity of the gearbox using the methods of the theory of machines and mechanisms. One of the main parameters that reducers must satisfy is stiffness, gear ratio and natural frequency. The determination of the stiffness parameters of the gearboxes is carried out experimentally on the gearbox and theoretically. The experimentally determined gearbox parameters correspond most closely to the true stiffness of the gearbox. At the same time, this approach does not allow changing the gearbox parameters if the stiffness does not meet the operating requirements. Therefore, to determine the stiffness and dynamic characteristics of gearboxes, an algorithm was developed, and a program was compiled.

The stiffness parameters of the gearbox are determined using an application program written in Microsoft Excel. The program allows determining gear ratios, total gear ratio, moments of inertia reduced to the shaft, moments on the shaft, angular velocities and accelerations, moments of inertia, angular error, reduced stiffness, natural frequency, stiffness reduced to the input shaft of each shaft separately and gearbox as a whole. The developed program reduces the calculation time by more than 5...7 times in comparison with the analytical calculation. The program allows easily varying the components of the gearbox in order to obtain the optimal model. The reliability of the obtained results was compared with the experimental results. An error (~10%) is acceptable for this type of structure. To use the program, it is enough to set the input parameters of the constituent parts of the gearbox: bearings, tolerances and fits, acceleration, number of gear teeth, etc.

In accordance with the ideology of the concept of the complex for the design of specific elements of multi-stage dynamic stands, such as gearboxes, the calculation of which cannot be carried out in ANSYS analytical block, a program for calculating gearboxes was developed. The program allows conducting a comprehensive analysis of gearboxes, taking into account moments of inertia, angular velocities, angular accelerations, gear ratios, bearing stiffness and other characteristics, and calculating the stiffness and natural frequency of the gearbox.

For a better understanding of the foregoing, we will describe what a reducer is. The gearbox in multistage dynamic stands is designed to change the angular velocity, accelerations or engine effort. And it is a structure consisting of shafts and gears of various diameters. In turn, each shaft is a multi-stage body of rotation containing gears of various characteristics (number of teeth, diameters, gear ratios, etc.).

When calculating the gearbox, the concept of equivalent diameter is used. A shaft of equivalent diameter is a shaft having a constant diameter along its length, the maximum deflection of which is equal to the deflection of a shaft comprising different diameters. To determine this parameter, a program was developed to determine the equivalent diameter of the gearbox shafts using Excel spreadsheets.

We describe the procedure for determining the equivalent shaft diameter. To determine the equivalent diameter of each shaft of the gearbox, firstly, using a standard program, we solve the problem of determining the reactions of the supports at the points of contact of the shaft gears with other gears, then we set the modulus of elasticity of the shaft material and for each different-sized section of the shaft, the length, outer and inner diameter (if the shaft in this area is hollow). The program automatically calculates the moment of inertia of the shaft sections, its cross section, center of gravity, deflection and, ultimately, the equivalent shaft diameter. Having determined the equivalent diameter of each shaft, we calculate the gearbox as a whole. The calculation of the gearbox is to determine the stiffness of the gearbox and its natural frequency.

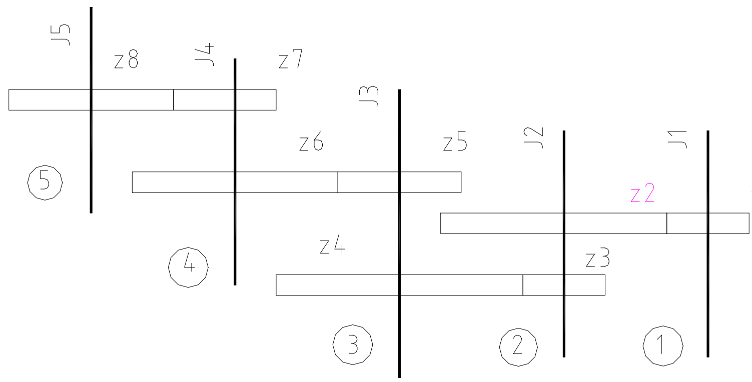


Fig. 3. Kinematic diagram of the pitch reducer

2.7 Initial data

Initial data moments of inertia of shafts $\mathbf{J_5,J_4,J_3,J_2,J_1}$ (kg×m×s^2), angular velocity υ_5 (1/s), angular acceleration ω_5 (1/s^2) and frequency **f** (Hz).

Gear ratios i54=z8/z7, i43=z6/z5, i32=z4/z2, i21=z2/z1, i11=z1/z1.

Gear ratios related to the input shaft i51=i54×i43×i32×i21×i11, i41=i51/i54, i31=i41/i43, i21=i31/i32, i11=i21/i11.

General gear ratio i_{tot} = i54×i43×i32×i21×i11.

Gear ratio i_{red} = i43×i32×i21×i11.

Moments of inertia referenced to input shaft J51=J5/i51^2, J41=J4/i41^2, J31=J3/i31^2, J21=J2/i21^2, J11=J1/i11^2.

Shaft accelerations $\omega_4=\omega_5\times i54$, $\omega_3=\omega_4\times i43$, $\omega_2=\omega_3\times i32$, $\omega_1=\omega_2\times i21$

Moments on the shaft M5, M4=J4×ω_4+M5/i54, M3=J3×ω_3+M4/i43, M2=J2×ω_2+M3/i32, M1=J1×ω_1+M2/i21.

2.8 Shaft load

Rocr=Mj/m×z, Prad=Rocr×tan20°,
where m is the module, z is the number of teeth and Mj is the moment on the shaft.

Reactions of supports R_1 and R_2 are determined by methods of resistance of materials, taking the moment of inertia of the shaft section Jshaft=3.14×Deq^4/64 based on Deqv - equivalent constant diameter for bending.

Displacements and angles of rotation at the points of application of force and supports, respectively, are determined by BEAM program.

2.9 Deformation of the bearing from fit on the shaft and into the housing

$$g_r=g_{r0}-\Delta d_1-\Delta D_2,$$

where g_{r0} is the initial clearance of the ball bearing, 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 fit into the housing are determined from the graphs [9], depending on the ratio of the stiffness of the mating parts:

- shaft (steel) $S_V=d_2/d$, bearing inner ring $S_{vn}=d/d_1$, bearing outer ring $S_H=D_2/D$, body (steel) $S_k=D/D_1$, where d_2 is the bore of the hollow shaft, d is the inner diameter of the bearing inner ring and d_1 is the outer diameter of the bearing inner ring, D, D_2, respectively, are the outer and inner diameter of the outer ring of the bearing, D_1 is the outer diameter of the housing
- ratios of deformations and fitting interferences in shaft-inner ring connections d_1/Hb, housings with outer ring D_2/Hk, $Hb=Hav - H_1$, Hav is the average value of the fit interference of the inner ring of the bearing, H_1 is the amount of unevenness and $Hk=Hav - H_1$, also for the outer ring of the bearing.

D_1 is the increase in the outer diameter of the inner ring,

D_2 is the reduction of the inner ring outer diameter.

2.10 Bearing deformation under load

$$\delta r= \delta r1+ \delta r2,$$

Here δr1 is the deformation in contact of the most loaded rolling element with the raceway in the bearing, δr2 is the radial compliance in contact of the bearing rings with the shaft and housing mounting surfaces.

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

- with preload:

$\delta r1= \beta \times \delta r0;$

- with radial clearance:

$$\delta r1= \beta \times \delta r0-gr/2.$$

The coefficient β, which takes into account the amount of interference or clearance in the bearing, is determined from the graphs [9].

The value of δr0 for bearings of various types can be determined from the equations [9]:

- for single-row ball:

$\delta_{r0} = 2.0\times10^{-3} \times \sqrt[3]{\frac{Q_0^2}{D_T}};$

- for angular contact tapered bearing:

$$\delta_{r0} = 6.0\times10^{-4} \times \frac{Q_0^{0.9}}{\cos\alpha\times l^{0.8}},$$

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

2.11 Radial compliance in contact of bearing rings with shaft and housing seating surfaces

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

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

2.12 Calculation of the rotation of the gear wheel caused by the elastic deflection of the shaft

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

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

2.13 Calculation of error due to shaft twist

$$\varphi_{tw}=\sum(M\times li/(G\times Jp)), \text{ radian},$$

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

2.14 Calculation of the error caused by the compliance of the ring gear

$$\varphi_{tooth}=M/(k\times d^2\times b), \text{ radian}.$$

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

2.15 Calculation of error due to keyed connection

$$\varphi_{shp}=M/k_{shp}\times d_b^2\times l\times h\times z, \text{ radian},$$

where d_b is the shaft diameter, l is the working length of the key, h is the height of the key, z is the number of keys, $k_{shp}=15kg/mm^3$ for parallel keys and $k_{shp}=25kg/mm^3$ for segmented keys.

2.16 Error due to pinning deformation

$$\varphi_{sht}=k_{sht}\times M, \text{ radian},$$

where k_{sht} is the coefficient of proportionality, M is the moment on the shaft [10].

2.17 Total gear stiffness

$$C=\left\{\sum_{j=1}^{n}\left[\frac{\left(\phi_{defl}+\phi_{tw}+\phi_{tooth}+\phi_{shp}+\phi_{sht}\right)_j}{M_j\times i_j^2}\right]\right\}^{-1}, kg\times m\times rad$$

Here i is the gear ratio related to the input shaft.

2.18 Natural frequency of the design

$$f=\frac{1}{\pi}\times\sqrt{\frac{C\times i_{total}^2}{J_5}}, Hz,$$

where i_{total} is the total gear ratio, C is the rigidity and J_5 is the reduced moment of inertia.

The resulting rigidity of thrust bearings, gearboxes and gear rims make it possible to replace them with a rod system identical in stiffness to the calculated parameters: gear rim at the connection of the fork and base, thrust bearings at the junction of the pitch ring with the course fork and gear wheel at the junction of the roll ring in the pitch channel. The reducers connect the motors with the movement elements of the bench channels.

Tables 1, 2 and 3 show the characteristics of the materials used: SAN Foam, carbon fiber and magnesium alloy [18- 20].

Table 1. Physical and mechanical characteristics of SAN Foam

Modulus of elasticity, MPa	Poisson's ratio	Shear Modulus, MPa	Tensile strength, MPa	Shear sz, MPa	Shear θz, MPa	Density,. kg/m³
60	0.3	23	1.1	0.8	0.8	81

Table 2. Physical and mechanical characteristics of carbon fiber

Modulus of elasticity, MPa	Poisson's ratio	Shear modulus, MPa	Tensile strength, MPa	Ultimate compressive strength, MPa	Shear strength, MPa	Density, kg/m³
$E_s=1.233\times10^5$	$\nu_{s\theta}=0.27$	$G_{s\theta}=0.5\times10^4$	$\sigma_s=1632$	$\sigma_s=704$	$\tau_{s\theta}=80$	1518
$E_\theta=0.778\times10^5$	$\nu_{\theta z}=0.42$	$G_{\theta z}=3.1\times10^3$	$\sigma_\theta=34$	$\sigma_\theta=68$	$\tau_{\theta z}=55$	
$E_z=0.778\times10^5$	$\nu_{sz}=0.27$	$G_{sz}=0.5\times10^4$	$\sigma_z=34$	$\sigma_z=68$	$\tau_{sz}=80$	

Table 3. Physical and mechanical characteristics of magnesium alloy

Modulus of elasticity (shear), MPa	Poisson's ratio	Density, kg/m³	Tensile strength, compression, MPa	Tensile strength, compression, MPa
$4.5\times10^4(1.6\times10^4)$	0.35	1740	250	190

3 Results and Discussion

The considered dynamic bench of semi-natural simulation is three-stage slewing mechanism, consisting of base, course fork connected to it, pitch and roll ring included in the pitch ring. With the tested product installed in the roll ring, the stand makes movements in three degrees of freedom, simulating the operation of the product. The dynamic stand is designed to simulate the flight characteristics of the product in ground conditions. The use of stands of this type can significantly reduce the cost of testing products that move in airspace or experience overloads during operation. The general view of the stand is shown in the Figure 4.

The stand was modeled using the finite element method [11-13], shell and flat three-four nodal finite elements with six degrees of freedom at each node.

The considered finite elements take into account bending and membrane deformations, the material is isotropic with characteristics specified in the plane of the element and perceives concentrated, distributed, mass and temperature loads.

When replacing bearing supports, gearboxes and gear rims with rod elements, the following assumptions were made:

- bearing elements of a ball bearing: balls were approximated by rods with rigid fastenings at the ends;

- resulting rod system matched the stiffness of the ball bearing.

The stand was modeled in ANSYS computer-aided design system, and bench was approximated by 420510 finite elements [14, 15].

Let us consider the procedure for modeling a stand of a three-layer structure made of a composite material

- Modeling of bench in CATIA, assigning material characteristics for the structure, which is a lightweight porous material such as foam. This material serves as a light intermediate layer in a three-layer structure between the carrier layers, preventing the bearing layers from approaching and absorbing mainly shear stresses.
- Next, we create surface elements for the entire structure, to which we further assign the characteristics of a multilayer composite package of five unidirectional layers with a given orientation: 0/45/0/-45/0 degrees [16-26].

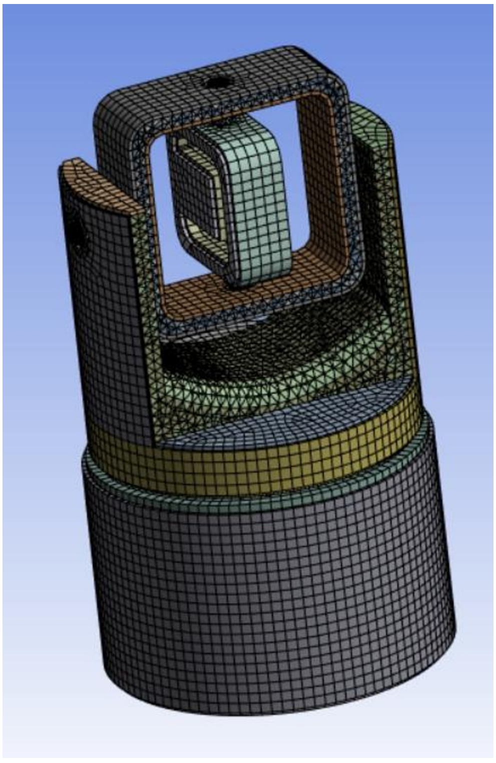


Fig. 4. Finite element approximation of the stand model and the direction of the stand movement from the angular operating speed of 500 deg/s applied to the course fork

Figure 5 shows the stresses in the stand at an angular velocity of 500 deg/s calculated

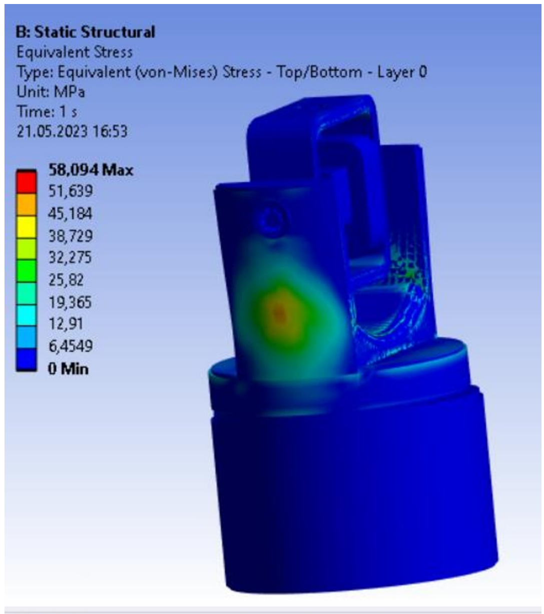
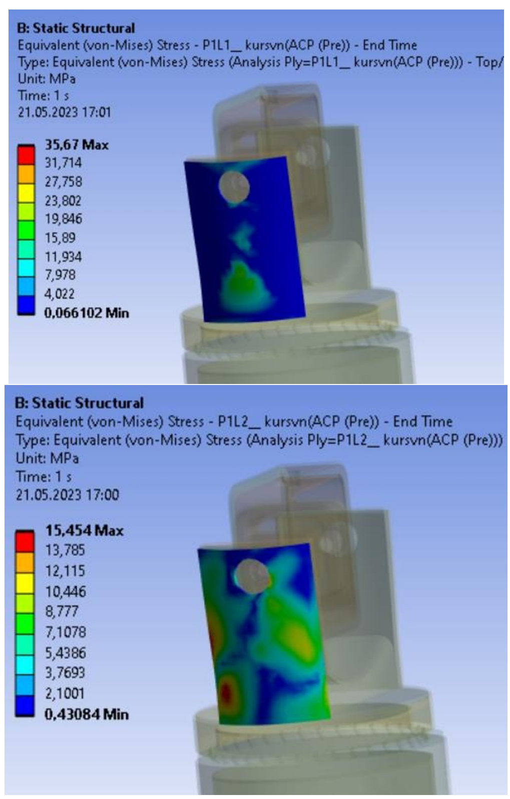
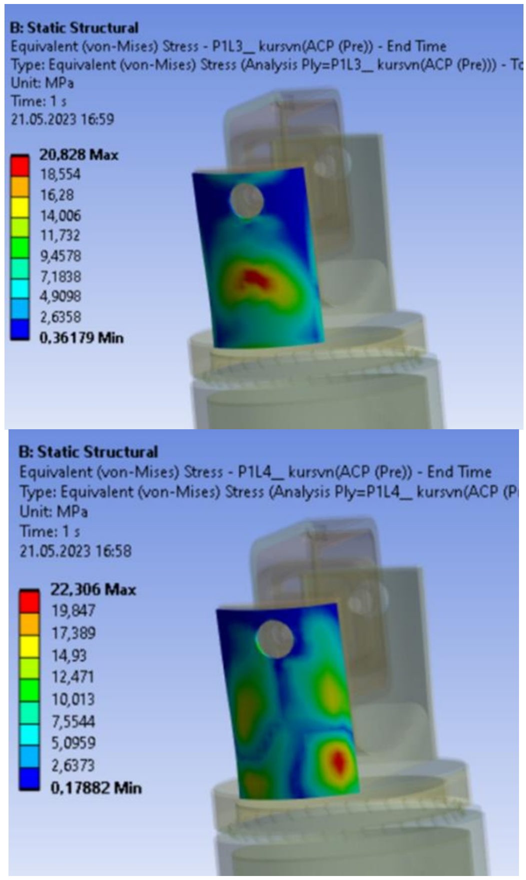


Fig. 5. Stresses in the stand at an angular velocity of 500 deg/s

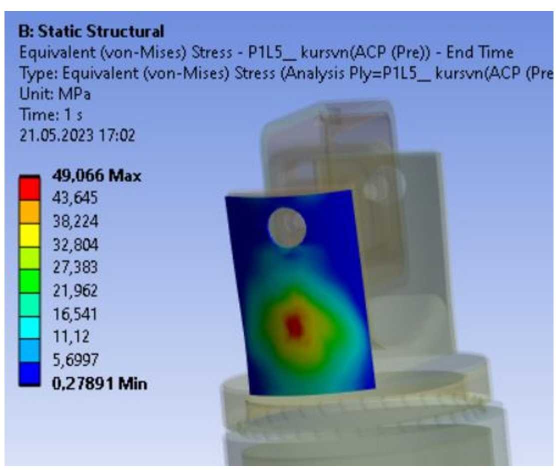
Figure 6 shows the stresses of a composite package of five unidirectional layers with a given orientation: 0/45/0/-45/0 degrees layer by layer



Stresses in the 1st and 2nd layers of the composite material



Stresses in the 3st and 4nd layers of the composite material



Stresses in the 5st layer of the composite material

Fig. 6. The stresses of a composite package of five unidirectional layers with a given orientation: 0/45/0/-45/0 degrees layer by layer

On figure 7 shows the maximum stresses in a five-layer package. Here it is indicated: e - deformation, 2 - direction, e - tension, the number in brackets means the number of the layer, tw and ho - means the destruction of the layered structure according to the criterion of destruction of Tsai-Wu, Tsai-Hill and Hoffman.

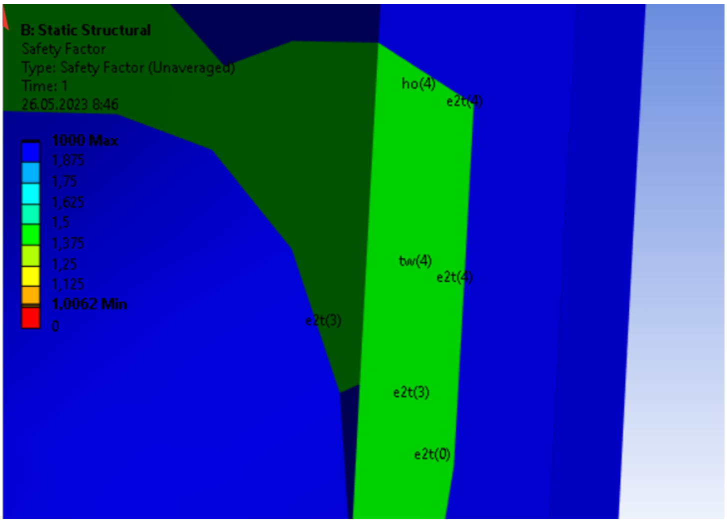


Fig. 7. The maximum stresses in a five-layer package. Here it is indicated: e - deformation, 2 - direction, t - tension, the number in brackets means the number of the layer, tw - means the destruction of the layered structure according to the failure criterion of tw - Tsai-Wu, ho - Hoffman

4 Conclusions

As a result of the work carried out, a simulation of a dynamic stand was performed, which a complex three-layer structure consisting of a stand layout is made of a lightweight composite material (table 1). By means of the simulation program, surfaces were created on the surface of the stand layout, which, later, were assigned the properties of a five-layer composite material (table 2). Table 3 shows the physical and mechanical characteristics of the magnesium alloy traditionally used in the manufacture simulation stands. The considered dynamic system contains bearings, gearboxes and gears that cannot be modeled directly. Therefore, a technique was developed for approximating such elements by systems that are identical in their characteristics: rod systems. To determine the stiffness parameters of bearing supports, gearboxes and gear rims, algorithms and programs for determining the stiffness characteristics have been developed. The reliability of the obtained results is confirmed by the correspondence of the electronic design data to the received drawing documents and test cases. In particular, a modified program for determining the stiffness-frequency characteristics of gearboxes was tested in comparison with experimental data, which showed their good agreement, within 10-15%. The stand was approximated by finite elements, volumetric and surface finite elements [11-15]. The performed calculations made it possible to determine the stress-strain state of the stand by layers (Figure 4) under dynamic loads: operational angular velocity around the vertical axis of the stand. To determine the destruction of multilayer composite materials, the following main criteria for the destruction of polymer composite materials were used: by maximum stresses, by maximum deformations, Tsai-Wu, Tsai-Hill and Hoffman [27-30]. As can be seen from Figure 5, considered in the most loaded place of the stand, the destruction of the five-layer composite material occurs in the fourth layer according to the theories of Tsai-Wu and Hoffman from the acting tensile stresses in the second direction, that is, in the direction perpendicular to the base of the composite material, which runs along the trajectories of maximum stresses obtained and applied for the orientation of the base of the composite material from the calculations. The calculations of a stress-strain bench made of a composite material under dynamic loads showed that the use of a composite material and three-layer structures in the design and manufacture of benches has an advantage in terms of bearing capacity compared to benches made of magnesium alloys, traditionally used in the preparation of benches for semi-natural simulation. The considered technique can be applied to robotic systems that include bearings, gear rims, gearboxes and motors. Such systems include robots, looms, etc.

5 Acknowledgements

This work was carried out with the financial support of the Russian Science Foundation, under the scientific project No. 22-29-20299 (recipient Khayrnasov K.Z., https://rscf.ru/project/22-29-20299/).

References

1 Y. Tian, C. Chen, W. Duan, Intelligent robotic systems for structural health monitoring: Applications and future trends. Automation in Construction, **139** (2022) https://doi.org/10.1016/j.autcon.2022.104273

2 B. Tao, Y. Feng, Y. Wu, Accuracy of dental implant surgery using dynamic navigation and robotic systems: An in vitro study, Journal of Dentistry, **123** (2022) https://doi.org/10.1016/j.jdent.2022.104170 robotic

3 Y. Tian, C. Chen, W. Duan, Intelligent robotic systems for structural health monitoring: Applications and future trends, Automation in Construction, **139** (2022) https://doi.org/10.1016/j.autcon.2022.104273

4 X. Xu, Y. Chen, Y. Gong, Assignment of parcels to loading stations in robotic sorting systems, Transportation Research Part E: Logistics and Transportation Review, **164**, (2022) https://doi.org/10.1016/j.tre.2022.102808

5 G. Boschetti, M. Faccio, R. Minto, 3D collision avoidance strategy and performance evaluation for human–robot collaborative systems, Computers & Industrial Engineering, **179** (2023) doi.org/10.1016/j.cie.2023.109225

6 B. Lindqvist, S. Karlsson, G. Nikolakopoulos, Multimodality : Integrated combined legged-aerial mobility for subterranean search-and-rescue, Robotics and Autonomous Systems, **154** (2022) doi.org/10.1016/j.robot.2022.104134

7 Yi Guo, R. G. Parker, Stiffness matrix calculation rolling bearings using a finite element/contact mechanics model, Mechanism and Machine Theory (2012) doi:10.1016/j.mechmachtheory.2011.12.006

8 I. Hoopert, J. Tribology, 136 (2014)

9 P. R. N. Childs, Rolling Element Bearings , Mechanical Design (2021)

10 P. Dewangan, A. Parey, M. Haddar, Dynamic characteristics of a wind turbine gearbox with amplitude modulation and gravity effect: Theoretical and experimental investigation, Mechanism and Machine Theory, **167** (2021)

11 L. Liu, L. Zhu, X. Gou, Modeling and analysis of load distribution ratio and meshing stiffness for orthogonal spur-face gear drive under point contact, Mechanism and Machine Theory, **184** (2023) doi.org/10.1016/j.mechmachtheory.2023.105312

12 H.-G. Kim, R. Wiebe, Numerical investigation of stress states in buckled laminated composite plates under dynamic loading, Composite Structures, **235** (2020) doi.org/10.1016/j.compstruct.2019.111743

13 A. Vaibhav, A. Phadnis, V. Silberschmidt, Composites Under Dynamic Loads at High Velocities Comprehensive Composite Materials, Composite Materials, **8** (2018)

14 A. Chao Correas, H. Ghasemnejad, Analytical development on impact behaviour of composite sandwich laminates by differentiated loading regimes, Aerospace Science and Technology, **126** (2022) doi.org/10.1016/j.ast.2022.107658 1270-9638/

15 E. J. Barbero. Finite Element Analysis of Composite Materials Using ANSYS (2013)

16 O. G. Latyshev, A. B. Veremeychik, E. A. Zhukov Application of composite materials in stands for dynamic loading, Moscow, Publishing house of MSTU N.E. Bauman, (2011)

17 L. M. Gavva, V. V.Firsanov, Mechanics of Solids (Springer), 3 (2020)

18 O. V. Mitrofanov, Natural and Technical Sciences, **2(153)** (2021)

19 V. V. Vasiliev, E. V. Morozov, Advanced Mechanics of Composite Materials and Structures, Elsevier (2018)

20 A. Manes, A. Gilioli, C. Sbarufatti, M. Giglio, Experimental and numerical investigations of low velocity impact on sandwich panels, Compos. Struct., 99 (2013) doi.org/10.1016/j.compstruct.2012.11.031.

21 O. C. Zienkiewicz, R. L. Taylor, and J.Z. Zhu, Finite element method: its basis and fundamental, Butterworth-Heinemann, Oxford (2013)

22 S. Moaveni, Finite Element Analysis Theory and Application with ANSYS, Pearson Education, London (2015)

23 I. Koutromanos, Applied Fundamentals of Finite Element Analysis Linear Finite Element Analysis, John Wiley & Sons, New York (2018)

24 K. J. Bathe, Finite element procedures, Pearson education Inc., New York (2006)

25 A. Noman, M. Shohel, S. Gupta, *Investigate the mechanical strength of laminated composite carbon fiber with different fiber orientations by numerically using finite element analysis*, Material: Proceedings (2023) doi.org/10.1016/j.matpr.2023.02.132

26 J.-P. Lin, X. Liu, G. Wang, Static and dynamic analysis of three-layered partial-interaction composite structures, Engineering Structures, 252 (2021) doi.org/10.1016/j.engstruct.2021.113581

27 B. Zhang, J. Ge, J. Liang, Failure prediction for fiber reinforced polymer composites based on virtual experimental tests, Journal of Materials Research and Technology, 24 (2023)

28 I. Daniel, Yield and failure criteria for composite materials under static and dynamic loading. Prog. Aero Sci., **81**, 18e25 (2016) https://doi.org/10.1016/j.paerosci.2015.11.003.

29 J. Gu, P. Chen, L. Su, K. Li, A theoretical and experimental assessment of 3D macroscopic failure criteria for predicting pure inter-fiber fracture of transversely isotropic UD composites. Compos. Struct., 259 (2021) doi.org/10.1016/ j.compstruct.2020.113466.

30 Q. Sun, G. Zhou, Z. Meng, H. Guo, Z. Chen, H. Liu, H. Kang, S. Keten and X. Su, Failure criteria of unidirectional carbon fiber reinforced polymer composites informed by a computational micromechanics model. Compos. Sci. Technol., 172 (2019) doi.org/10.1016/ j.compscitech.2019.01.012.