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# Analysis of the Stress-Strain State of the Spatial Parallel Manipulator Used in Military Robotic Systems of the Republic of Kazakhstan

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**Abstract:** In modern conditions, military robotics remains one of the priority branches of science and technology. Similar to the combat arms, modern combat robots are divided into three groups: ground, flying and floating. The most complex in design and combat use are ground robots. According to their functional purpose, ground military robots are divided into reconnaissance, combat, engineering and rear. By the degree of automation, robotic machines can be remotely controlled, autonomous (operate according to a program installed in the on-board computer) and also combined. Military robots differ in size, list of tasks, chassis design, hull configuration. The most intensive development of ground-based military robots is being conducted in the USA, Israel, Russia and some other countries.

## INTRODUCTION

In the armies of some foreign states, robotic complexes of various purposes are already in service now which are regarded as one of the most important attributes of the military technology of the future. The high level of equipping the Armed Forces of these states with robotic means provides them with the ability to conduct modern network-centric wars based on the massive application of robots.

According to the long-term plans of the US Department of Defense, the development of ground-based robotic complexes for various purposes whose share should make at least 30% of the total amount of military equipment by 2020 will lead to a significant increase in the combat capabilities of the Armed Forces, while reducing the number of servicemen and equipment and will significantly reduce the loss of personnel during the conduct of hostilities. In Russia, it is planned to combine several combat and reconnaissance robots into a combat robotic system. With a view to realizing this project, in each military district and fleets, special separate mouths of combat robots and the formation of their control bodies have been set up.

The creation of armed shock robotic complexes, united in robotic systems, with their integration with unmanned vehicles of various purposes, is a new stage in the development of modern automated high-precision weapons.

Considering the problems of different robotic systems, we chose the scheme of the spatial parallel manipulator used in robotic complexes and also conducted its analysis of the stress-strain state.

At present, in engineering design calculations, linear analysis continues to be the most common means of assessing their performance. The model of a perfectly elastic body, due to the properties of perfect elasticity, homogeneity, isotropy, physical and geometric linearity, introduces considerable simplifications into the calculation and when solving many important problems it allows one to obtain a result that quite reliably estimates the performance of engineering structures. It is often overlooked that it is at best only the first approximation which is valid in the nearest neighborhood of the initial state. However, some of the modern structural materials do not possess all of the above properties. Therefore, in order to solve a number of critical problems, it is necessary to have theories that allow us to adequately reflect the real properties of the material, even when such properties differ significantly from those of a perfectly elastic body. The latter circumstance is determined both by the nonlinearity of the characteristics of structural materials and by the change in the metric characteristics of the structure itself in the process of deformation. Accounting for these factors is the subject of a nonlinear analysis.

Essentially necessary to make certain refinements in the model of an ideally elastic body are first of all thin-walled spatial engineering constructions. Under difficult conditions of their loading, high stress levels, the property of relative stiffness is violated and leads to a significant complication of the equations of equilibrium and geometric relationships: they become nonlinear. The corresponding problem of the theory of elasticity is usually called a geometrically nonlinear problem. The desire to apply more thin-walled structural materials with increased strength as well as the rejection of the relative rigidity of the body, requires making appropriate changes both in the content of the model of the perfectly elastic body itself and in the basic dependencies of the linear theory of elasticity. Thus, it is precisely thin-walled structures that constitute the class of problems for which the development of nonlinear analysis methods, taking into account the mutual influence of large elastic displacements and the nonlinearity of structural materials, is of decisive importance.

The problems of the mechanics of deforming bodies are inherently nonlinear. There are several main reasons for the appearance of nonlinear terms in the main dependencies of the theory of elasticity:

- Taking into account the geometric nonlinearity, when the displacement of the structure causes significant changes in its geometry, so that the equilibrium equations are compiled for the deformed state
- Nonlinearity, determined by the nonlinearity of the connection between deformations and stresses, i.e. physical nonlinearity of materials
- The nonlinearity of the mechanical characteristics of the interaction of structural elements, i.e., structural nonlinearity
- Combinations of different categories of nonlinearity

Geometrically nonlinear problems arise in the study of the stress-strain state (VAT) of bodies that do not have the property of relative rigidity. In this case, in deriving the dependencies between deformations and displacements, one can not neglect the rotation angles of the elements in calculating their length and linear deformations in the expressions for the rotation angles. In the general case, the geometric and structural nonlinearities are simultaneous in nature. However, they can often be considered separately, independent from each other, since in the applied region seldom single-valued nonlinear effects appear simultaneously.

The results of the calculation within the framework of the linear theory are not always sufficiently accurate which can adversely affect the reliability of the constructive system or its economy. Therefore, in such cases, in order to make real conclusions about the VAT of constructive systems under static and dynamic influences, it is necessary to use a nonlinear theory that can also serve to assess the justification of the application and the accuracy that the linear theory gives. The main directions of the development of nonlinear analysis are currently developing in two directions improving the computational models of complex systems in order to ensure the accuracy and adequacy of nonlinear analysis and the development of effective and cost-effective analysis algorithms which makes it possible to conduct it at an acceptable cost and time.

The modern theory of analysis of continuum mechanics and engineering design on the basis of the finite element method (MCE) has been developed to such an extent that it can be effectively applied to solve very complex nonlinear problems. It is based on the developed theory of continuum mechanics, worked out methods of discretization of structures, effective numerical methods for the formation and solution of large systems of nonlinear equations and rapidly developing computer technology. However, the complexity of nonlinear analysis, based on multi-step or iterative algorithms, is incommensurably higher than the laboriousness of linear analysis which puts forward increased demands on memory resources and computer speed[1-5].

Analysis of nonlinear systems is conceived as a continuous process of modeling. Therefore, it is very important that each element of the design model either be in conditions similar to the conditions of the original system, or every inaccuracy in the simulation leads to unpredictable accumulation of errors. Further, having overcome the difficulties of mathematical formulation of the nonlinear model, the researcher faces a whole series of questions, the answers to which are difficult to obtain. Among them is the choice of the decision algorithm, since the approved linear analysis apparatus can no longer be used; lack of firm confidence that the solution is unique; difficulty or inability to verify the results by a physical experiment; existence of only a few rules of control, giving confidence in the correctness of the solution. Since, all solutions of non-linear analysis are built on the basis of certain incremental theories, we must accept the inevitable error of each step and the accumulation of error in the calculation process. Therefore, it is often necessary to use some heuristic techniques to develop a strategy for solving and developing specific procedures that save the computer's memory and reduce the amount of computing work[6].

The modern achievements of nonlinear analysis are based on the accumulated experience of rational modeling of complex structures. The concept of modeling, based on the closeness of the real construction and the computational model, is the factor that gives the researcher confidence in the result. Although a rigorous mathematical formulation of the existence and convergence of solutions is rarely possible, a physical and numerical experiment that largely replenishes this gap comes to the rescue. When calculating complex structures, an important element of the analysis is the comparison of results obtained using different models, methods and algorithms. It is the complexity of the analysis that becomes the determining factor that gives confidence in the reliability of the result obtained. The most complicated problems of the mechanics of a deformable solid are nonlinear dynamic problems. In them, it is necessary to take into account the variability of the parameters of VAT over time. There is no need to convince in complexity of the joint account of nonlinear and dynamic effects. In addition, for a number of problems, mathematical theories have not yet been constructed which could more or less accurately describe the experimentally determined features of nonlinear non-stationary deformation of materials.

Naturally, in the vast majority of cases, only numerical methods are applicable to the solution of nonlinear dynamical problems: for example, the FEM, the most important advantage of which is the existence of stable methods of numerical integration of systems of differential equations of motion, describing the motion of mechanisms and composed taking into account the elasticity of the links[7, 8].

Based on the number of works published in recent years and the number of researchers who have devoted themselves to the study of problems, it can be concluded that the scope of the FEM in nonlinear analysis is one of the most relevant research areas in continuum mechanics. Methods for solving nonlinear problems or, respectively, nonlinear FEM equations can be divided into three main groups: incremental, iterative and mixed (incremental-iterative). Within each, special methods or methods have been developed that are adapted to nonlinear problems.

However, along with this, further work is being done in this area, especially because common programs are not equally good for all non-linear problems. According to the developed algorithm, a package of applied programs was compiled, and the dynamic VAT of the Posel'e-Lipkin plane mechanism was calculated and analyzed (Fig. 1).

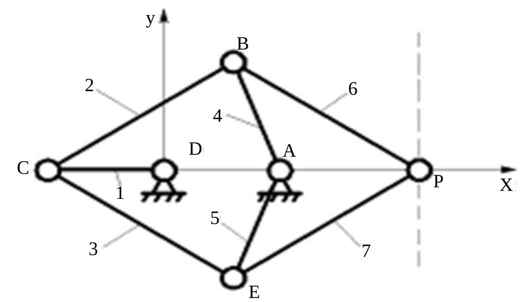


Fig. 1: Mechanism of Poselje-Lipkin

This mechanism with light parts and operating at high speeds, serves to build a straight path with the help of point P. It consists of 7 links made of steel rods of round cross section with a diameter of 0.006 m and a post. The links are connected in 6 rotational kinematic pairs. The points A, C and P must always lie on the straight line passing through point A. The condition AC•AP = const is always satisfied. When AD = CD, the point C must move along the arc of the circle and the point P-exactly along the straight line but in connection with the account of the elasticity of the links these conditions are not satisfied[9-11]. The mechanism has the following geometric dimensions:

$$l_2 = l_6 = l_3 = l_7 = 0.3048 \text{ m}, \;\; l_4 = l_5 = 0.215 \text{ m}$$

The mechanism consists of eight links, counting the rack, with ten rotational kinematic pairs. In Fig. 1, the links of the mechanism are indicated by the numbers 1-7 and the post is accepted as the zero link. It is assumed that the mechanism has one degree of freedom, and the law of motion of the leading link 1 is described by Eq. 2:

$$\phi_1 = \sin 750t \tag{2}$$

Where:

$\varphi_1$ = The crank angle relative to the inertial reference frame XY

t = The time

Known kinematics of the mechanism, obtained by the method of generalized coordinates by Masanov, etc. The dynamics of this mechanism, the elastic elements of which are made of isotropic material was studied linearly by Masanov. The order of the System of Linear Algebraic Equations (SLAE). The number of iterations over time 67 and by nonlinearity.

The calculation program was tested to solve a similar problem for small displacements. When only the forces of

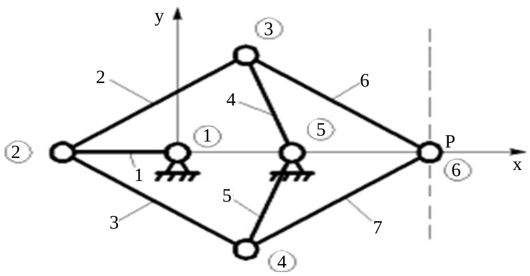


Fig. 2: The Poselje-Lipkin mechanism for finite element simulation

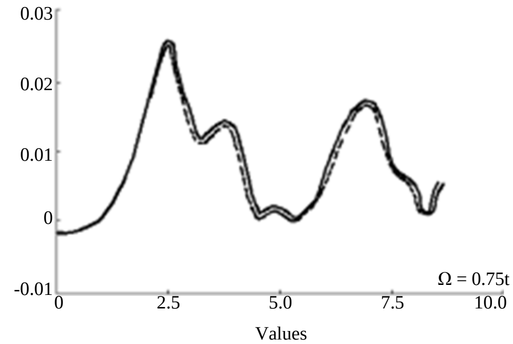


Fig. 3: Elastic movements of the V node 6

inertia act when all links are considered elastic, due to the elasticity of the links, the dimensionless deviation of the node 6, with the angle of rotation of the leading link $\varphi_1 = \sin\Omega$ according to the developed algorithm is shown in Fig. 2. Here $\Omega = 7.50$ t. Test problem, according to the developed algorithm.

The material is boron aluminum[12]. Elastic and geometric parameters of the mechanism, consisting of anisotropic links:

$$\begin{aligned} &E_1 = 2\cdot10^5\,\text{МПа}, E_2 = 0.9\cdot10^5\,\text{МПа}; G_1 = 80\cdot10^3\,\text{МПа},\\ &G_2 = 40\cdot10^3\,\text{МПа}; G_3 = 63\cdot10^3\,\text{Мпа}; \nu_1 = 0.25,\\ &\nu_2 = 0.13;\ \rho = 2640\ \text{кг/м}^3; d = 0.006\ \text{м} \end{aligned} \tag{3}$$

Where:

| | | |
|---|---|---|
| $E_1$, $E_2$ | = | The elastic modules |
| $G_1$, $G_2$, $G_3$ | = | The elastic modules |
| $v_1$, $v_2$ | = | Poisson’s coefficients |
| ρ | = | The density |
| d | = | The diameter of the links |

Table 1 shows a comparison of elastic displacements and in Table 2 longitudinal and shear forces, normal stresses obtained from linear and nonlinear calculations from the action of inertia forces in the isotropic links of the Poselje-Lipkin mechanism[13]. As can be seen in Table 1, the values of displacements in elements 2 and 3, obtained with nonlinear calculation, exceed by 20-25%, the values of displacements obtained with linear calculation. In the remaining elements which are not shown in the table, the movements increase by 14-18%. And internal forces and stresses with allowance for geometric nonlinearity, decrease by 21-23% in elements 1 and 2, in other elements by 15%. Taking into account the anisotropy, when $\varphi = 0$, $\psi = 0$ leads to an increase in displacements by 50%[14].

Figure 3-5 show graphs of maximum longitudinal and transverse displacements, longitudinal and transverse forces, normal and tangential stresses from the combined action of inertia forces and external load F = sinωt, in the isotropic links of the Poselje-Lipkin mechanism.

Figure 6-8 show graphs of the maximum longitudinal and transverse displacements, longitudinal and transverse forces, normal and tangential stresses from the action of the external load F = e-αtsinωt, in the isotropic links of the Poselje-Lipkin mechanism. The external force is applied at node 6. The values obtained for linear and nonlinear calculations are compared. Moves also increase, and internal forces and stresses decrease. Large longitudinal displacements occur in elements 3 and 7, large lateral displacements of 2 and 3. The most loaded element is 1 and 5, the least 6. Taking an isotropy when $\varphi = 0$, $\psi = 0$ leads to an increase in movement by 50%[15].

Calculation of the dynamics of elastic RPM (Fig. 9) is carried out by computer simulation. To describe the finite element model (CEM) of the PPM, we divide it into rectilinear two-node rod elements connected at nodes through kinematic pairs. For APM, consisting mainly of individual rod links such a dismemberment is natural. The MRP nodes are numbered in the GCS which serves to identify them in the list of nodes. Elements have their numbers-initial and final, with the help of which their identification is made in turn.

When finite-element dynamic modeling of the PPM, we divide into 9 straight-line two-node rod elements with 9 nodes.

In the APM, the leading links are 1, 3, 5. The lengths of the links assume the following values: l1 = l3 = l5 = 1.3 m, l2 = l4 = l6 = l1 m, l7 = l8 = l9 = 1.5l1/cos30° m.

The inertial coordinate system XYZ is rigidly connected to a fixed base, the origin of which is at node 1. To describe the VAT of elastic PPMs in the second chapter, matrix equations of closure of the MRP contours are compiled and an algorithm and a program for analyzing displacements are developed. The manipulator kinematics was investigated by the Denavite-Hartenberg matrix method with six parameters[16]. The coordinates X, Y, Z of the nodes of the design model of the planning model are defined in the Global Coordinate System (GCS).

Table 1: Maximum longitudinal, transverse and angular displacements in the cross sections of elements over time, from the action of inertia forces in the isotropic links of the mechanism

| | Longitudinal movements $u\xi 10^{-4}$ (m) in the final section 3rd element | | Transversal movements $v\xi 10^{-4}$ (m) in the 8th section 2nd element | | Angular movements $\varphi\xi 10^{-4}$ (m) in the final section 2nd element | |
|---|---|---|---|---|---|---|
| Δt | Linear | Non-linear | Linear | Non-linear | Linear | Non-linear |
| 1 | 2 | 3 | 4 | 5 | 6 | 7 |
| 2 | 0 | 0 | 0 | 0 | 0 | 0 |
| 3 | -0.86 | -1.03 | 0.13 | 0.16 | -4.57 | -5.48 |
| 4 | 2.22 | 2.67 | 0.84 | 1.00 | -20.83 | -24.99 |
| 5 | 14.08 | 16.90 | 2.46 | 2.95 | -46.47 | -55.76 |
| 6 | 32.65 | 39.18 | 4.71 | 5.65 | -73.07 | -87.68 |
| 7 | 52.83 | 63.39 | 7.23 | 8.68 | -94.90 | -113.89 |
| 8 | 68.52 | 82.22 | 9.87 | 11.85 | -108.38 | -130.06 |
| 9 | 74.23 | 89.07 | 12.54 | 15.05 | -109.25 | -131.11 |
| 10 | 66.65 | 79.98 | 14.39 | 17.27 | -96.69 | -116.03 |
| 11 | 48.37 | 58.04 | 14.12 | 16.94 | -76.74 | -92.09 |
| 12 | 28.71 | 34.45 | 11.62 | 13.95 | -55.75 | -66.90 |
| 13 | 14.72 | 17.66 | 8.20 | 9.84 | -35.27 | -42.33 |
| 14 | 7.40 | 8.88 | 5.14 | 6.17 | -15.16 | -18.19 |
| 15 | 4.48 | 5.37 | 2.79 | 3.34 | 4.31 | 5.17 |
| 16 | 2.46 | 2.96 | 0.78 | 0.93 | 23.59 | 28.31 |
| 17 | -1.30 | -1.55 | -1.12 | -1.35 | 43.82 | 52.59 |
| 18 | -7.54 | -9.04 | -2.67 | -3.20 | 62.84 | 75.41 |
| 19 | -14.99 | -17.99 | -3.53 | -4.24 | 71.82 | 86.18 |
| 20 | -19.08 | -22.89 | -3.45 | -4.14 | 62.98 | 75.57 |
| 21 | -14.86 | -17.83 | -2.27 | -2.72 | 38.09 | 45.71 |
| 22 | -1.52 | -1.82 | -0.23 | -0.27 | 5.29 | 6.35 |
| 23 | 17.77 | 21.32 | 2.29 | 2.75 | -27.31 | -32.77 |
| 24 | 37.57 | 45.09 | 4.94 | 5.93 | -54.75 | -65.70 |
| 25 | 52.31 | 62.77 | 7.65 | 9.18 | -73.36 | -88.03 |
| 26 | 57.45 | 68.94 | 10.28 | 12.34 | -78.26 | -93.91 |
| 27 | 50.94 | 61.13 | 11.96 | 14.35 | -68.62 | -82.34 |
| 28 | 36.01 | 43.21 | 11.54 | 13.85 | -50.83 | -61.00 |
| 29 | 20.85 | 25.02 | 9.15 | 10.98 | -31.45 | -37.73 |
| 30 | 10.51 | 12.61 | 6.09 | 7.30 | -12.53 | -15.04 |
| 31 | 5.37 | 6.44 | 3.40 | 4.08 | 5.43 | 6.51 |
| 32 | 3.03 | 3.64 | 1.21 | 1.45 | 22.13 | 26.56 |
| 33 | 0.08 | 0.10 | -0.87 | -1.05 | 38.58 | 46.30 |
| 34 | -5.72 | -6.86 | -2.89 | -3.46 | 56.44 | 67.72 |
| 35 | -14.45 | -17.34 | -4.35 | -5.22 | 73.22 | 87.86 |
| 36 | -23.43 | -28.12 | -4.91 | -5.89 | 79.27 | 95.13 |
| 37 | -27.02 | -32.43 | -4.43 | -5.32 | 67.23 | 80.67 |
| 38 | -21.03 | -25.24 | -2.92 | -3.51 | 40.09 | 48.10 |
| 39 | -6.06 | -7.28 | -0.68 | -0.82 | 6.46 | 7.75 |
| 40 | 13.82 | 16.58 | 1.91 | 2.29 | -26.09 | -31.30 |
| 41 | 33.00 | 39.60 | 4.56 | 5.48 | -53.17 | -63.80 |
| 42 | 46.34 | 55.61 | 7.26 | 8.71 | -70.72 | -84.86 |
| 43 | 50.01 | 60.01 | 9.77 | 11.73 | -73.62 | -88.34 |
| 44 | 42.87 | 51.44 | 11.12 | 13.35 | -62.11 | -74.53 |
| 45 | 29.15 | 34.98 | 10.33 | 12.40 | -43.47 | -52.17 |
| 46 | 16.32 | 19.59 | 7.83 | 9.39 | -23.85 | -28.61 |
| 47 | 8.10 | 9.72 | 4.92 | 5.90 | -5.04 | -6.05 |
| 48 | 4.23 | 5.07 | 2.43 | 2.92 | 12.30 | 14.76 |
| 49 | 1.98 | 2.37 | 0.29 | 0.35 | 28.13 | 33.76 |
| 50 | -1.95 | -2.34 | -1.87 | -2.25 | 43.97 | 52.77 |
| 51 | -9.28 | -11.13 | -3.87 | -4.65 | 61.55 | 73.86 |
| 52 | -19.38 | -23.25 | -5.15 | -6.18 | 76.98 | 92.38 |
| 53 | -28.36 | -34.03 | -5.42 | -6.51 | 79.75 | 95.70 |
| 54 | -30.24 | -36.29 | -4.63 | -5.55 | 64.12 | 76.94 |
| 55 | -22.01 | -26.42 | -2.87 | -3.45 | 34.90 | 41.89 |
| 56 | -5.44 | -6.52 | -0.50 | -0.60 | 1.00 | 1.19 |
| 57 | 14.79 | 17.75 | 2.12 | 2.55 | -30.80 | -36.96 |
| 58 | 33.05 | 39.66 | 4.78 | 5.73 | -56.53 | -67.83 |
| 59 | 44.64 | 53.57 | 7.45 | 8.95 | -71.59 | -85.91 |
| 60 | 46.28 | 55.53 | 9.81 | 11.77 | -71.19 | -85.42 |
| 61 | 37.80 | 45.36 | 10.74 | 12.89 | -57.30 | -68.76 |

Table 1: Continue

| Δt | Longitudinal movements uξ$10^{-4}$ (m) in the final section 3rd element Linear | Non-linear | Transversal movements vξ$10^{-4}$ (m) in the 8th section 2nd element Linear | Non-linear | Angular movements φξ$10^{-4}$ (m) in the final section 2nd element Linear | Non-linear |
|---|---|---|---|---|---|---|
| 62 | 24.55 | 29.46 | 9.52 | 11.42 | -37.87 | -45.45 |
| 63 | 13.26 | 15.91 | 6.86 | 8.23 | -18.19 | -21.83 |
| 64 | 6.53 | 7.84 | 4.07 | 4.88 | 0.36 | 0.43 |
| 65 | 3.46 | 4.15 | 1.72 | 2.06 | 17.16 | 20.59 |
| 66 | 1.01 | 1.21 | -0.42 | -0.51 | 32.50 | 39.00 |
| 67 | -3.91 | -4.70 | -2.63 | -3.16 | 48.36 | 58.03 |

Table 2: Maximum longitudinal, transverse forces and normal stresses in sections of elements, with the passage of time from the action of inertia forces in the mechanism of Poselje-Lipkin

| Δt | N·$10^{-2}$ H in the 45th section 1st element Linear | Non-linear | Q.$10^{-2}$ H in the 18th section 2nd element Linear | Non-linear | σ.$10^{-2}$Πa in the 45th section 1st element Linear | Non-linear |
|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 |
| 2 | 0 | 0 | 0 | 0 | 0 | 0 |
| 3 | 3.82 | 3.18 | 2.04 | 1.70 | 1.70 | 1.42 |
| 4 | 6.81 | 5.68 | 8.41 | 7.01 | 6.80 | 5.66 |
| 5 | 4.89 | 4.08 | 17.50 | 14.58 | 18.06 | 15.05 |
| 6 | -2.17 | -1.81 | 25.98 | 21.65 | 33.98 | 28.32 |
| 7 | -17.08 | -14.23 | 31.83 | 26.53 | 50.32 | 41.93 |
| 8 | -39.08 | -32.56 | 34.14 | 28.45 | 65.39 | 54.49 |
| 9 | -64.17 | -53.47 | 32.20 | 26.83 | 79.90 | 66.58 |
| 10 | -86.78 | -72.32 | 27.11 | 22.59 | 90.58 | 75.49 |
| 11 | -94.61 | -78.84 | 21.17 | 17.64 | 95.03 | 79.19 |
| 12 | -79.64 | -66.37 | 15.12 | 12.60 | 94.31 | 78.59 |
| 13 | -46.78 | -38.98 | 8.65 | 7.21 | 88.36 | 73.63 |
| 14 | -11.03 | -9.19 | 1.86 | 1.55 | 74.57 | 62.14 |
| 15 | 12.84 | 10.70 | -4.84 | -4.03 | 51.94 | 43.29 |
| 16 | 18.55 | 15.46 | -11.54 | -9.62 | 24.16 | 20.13 |
| 17 | 10.85 | 9.04 | -18.65 | -15.54 | -1.40 | -1.17 |
| 18 | 1.59 | 1.33 | -25.20 | -21.00 | -18.32 | -15.27 |
| 19 | -2.75 | -2.29 | -27.67 | -23.06 | -26.51 | -22.09 |
| 20 | -4.37 | -3.64 | -23.54 | -19.62 | -26.96 | -22.46 |
| 21 | -2.91 | -2.42 | -13.95 | -11.63 | -18.03 | -15.02 |
| 22 | -0.69 | -0.57 | -2.17 | -1.81 | -1.50 | -1.25 |
| 23 | -3.27 | -2.73 | 8.68 | 7.23 | 18.01 | 15.01 |
| 24 | -14.19 | -11.82 | 16.95 | 14.13 | 36.29 | 30.24 |
| 25 | -32.03 | -26.69 | 21.67 | 18.06 | 52.66 | 43.88 |
| 26 | -52.97 | -44.14 | 21.80 | 18.17 | 68.06 | 56.71 |
| 27 | -71.84 | -59.87 | 18.00 | 15.00 | 79.49 | 66.24 |
| 28 | -77.35 | -64.46 | 12.45 | 10.38 | 84.34 | 70.29 |
| 29 | -62.21 | -51.84 | 6.40 | 5.33 | 82.90 | 69.08 |
| 30 | -32.31 | -26.93 | 0.19 | 0.16 | 74.53 | 62.11 |
| 31 | -3.07 | -2.55 | -5.83 | -4.86 | 57.15 | 47.63 |
| 32 | 12.17 | 10.14 | -11.35 | -9.46 | 31.08 | 25.90 |
| 33 | 9.85 | 8.21 | -16.71 | -13.93 | 1.26 | 1.05 |
| 34 | -2.59 | -2.16 | -22.52 | -18.77 | -24.17 | -20.14 |
| 35 | -12.59 | -10.49 | -27.79 | -23.16 | -38.86 | -32.39 |
| 36 | -15.05 | -12.54 | -29.05 | -24.21 | -43.08 | -35.90 |
| 37 | -12.69 | -10.57 | -23.92 | -19.93 | -38.39 | -31.99 |
| 38 | -6.98 | -5.81 | -13.77 | -11.48 | -24.96 | -20.80 |
| 39 | -2.12 | -1.77 | -1.92 | -1.60 | -5.68 | -4.74 |
| 40 | -3.68 | -3.07 | 8.77 | 7.31 | 14.85 | 12.37 |
| 41 | -13.94 | -11.61 | 16.92 | 14.10 | 33.24 | 27.70 |
| 42 | -30.57 | -25.47 | 21.39 | 17.83 | 49.81 | 41.51 |
| 43 | -50.00 | -41.67 | 21.03 | 17.53 | 65.19 | 54.33 |
| 44 | -66.56 | -55.47 | 16.65 | 13.88 | 76.06 | 63.38 |
| 45 | -68.96 | -57.46 | 10.56 | 8.80 | 79.90 | 66.58 |
| 46 | -51.49 | -42.91 | 4.11 | 3.43 | 76.87 | 64.05 |
| 47 | -22.52 | -18.76 | -2.23 | -1.86 | 65.97 | 54.98 |
| 48 | 2.10 | 1.75 | -8.03 | -6.69 | 45.55 | 37.96 |
| 49 | 10.94 | 9.12 | -13.15 | -10.96 | 17.12 | 14.27 |

Table 2: Continue

| | $N\cdot10^{-2}$ H in the 45th section 1st element | | Q.$10^{-2}$ H in the 18th section 2nd element | | σ.$10^{-2}$Πa in the 45th section 1st element | |
|---|---|---|---|---|---|---|
| Δt | Linear | Non-linear | Linear | Non-linear | Linear | Non-linear |
| 50 | 3.35 | 2.80 | -18.15 | -15.13 | -13.27 | -11.05 |
| 51 | -10.94 | -9.11 | -23.66 | -19.72 | -37.00 | -30.83 |
| 52 | -19.58 | -16.32 | -28.28 | -23.57 | -48.68 | -40.57 |
| 53 | -19.84 | -16.53 | -28.38 | -23.65 | -49.52 | -41.27 |
| 54 | -14.86 | -12.39 | -22.13 | -18.44 | -41.27 | -34.39 |
| 55 | -7.22 | -6.01 | -11.42 | -9.52 | -25.07 | -20.89 |
| 56 | -2.04 | -1.70 | 0.32 | 0.27 | -4.42 | -3.68 |
| 57 | -4.45 | -3.71 | 10.60 | 8.83 | 16.20 | 13.50 |
| 58 | -15.47 | -12.89 | 18.21 | 15.18 | 34.24 | 28.54 |
| 59 | -32.09 | -26.74 | 21.86 | 18.22 | 50.82 | 42.35 |
| 60 | -50.99 | -42.49 | 20.51 | 17.09 | 65.75 | 54.80 |
| 61 | -65.25 | -54.38 | 15.42 | 12.85 | 75.40 | 62.83 |
| 62 | -63.67 | -53.06 | 8.94 | 7.45 | 77.72 | 64.76 |
| 63 | -43.15 | -35.96 | 2.32 | 1.93 | 72.82 | 60.68 |
| 64 | -14.79 | -12.33 | -3.98 | -3.32 | 59.41 | 49.50 |
| 65 | 5.67 | 4.72 | -9.55 | -7.96 | 36.33 | 30.28 |
| 66 | 9.17 | 7.64 | -14.43 | -12.03 | 6.33 | 5.28 |
| 67 | -2.03 | -1.69 | -19.33 | -16.11 | -23.59 | -19.66 |

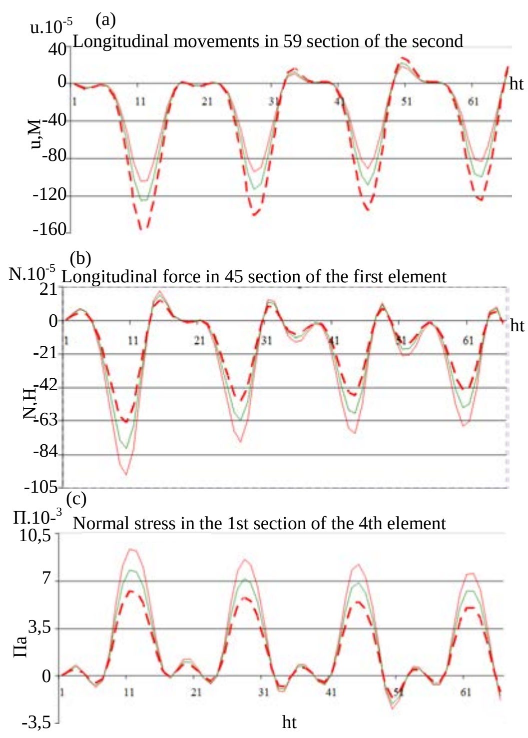


Fig. 4(a-c): Maximum longitudinal displacements, (a) Longitudinal forces, (b) Normal stresses and (c) The elements of the Poselje-Lipkin mechanism with the joint action of inertia forces and concentrated force: linear isotropic; nonlinear isotropic; linear anisotropic case

The PPM links are made of steel rods of round cross-section with a diameter of 0.006 m. The shape and dimensions of the cross section, the elastic properties of the materials are constant. The dimensions and design of the nodes are neglected.

The manipulator is under the action of the node forces of inertia of the FE, additional forces, nodal external forces, whose magnitude is 10 N and applied at the nodes 3, 4, 7. The order of SLAU: 54. Iteration over time: 807. Iteration over nonlinearity: 94.

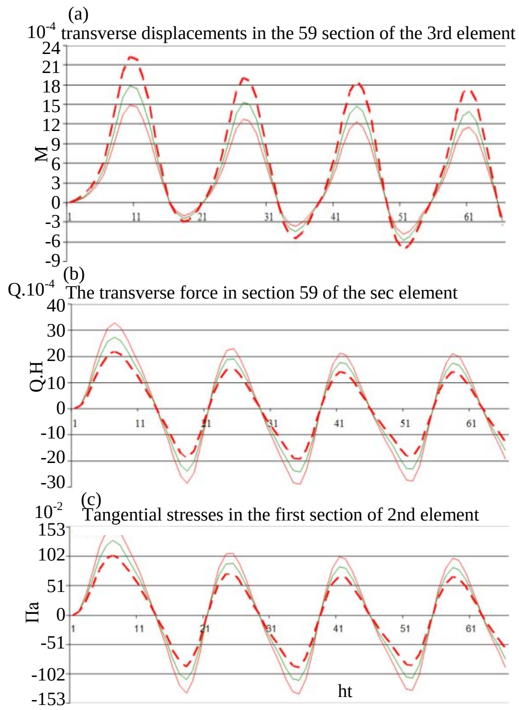


Fig. 5(a-c): Maximum lateral movements, (a) Transverse forces, (b) Tangential stresses and (c) The elements of the Poselje-Lipkin mechanism with the joint action of inertia forces and concentrated force: linear isotropic; nonlinear isotropic; linear anisotropic case

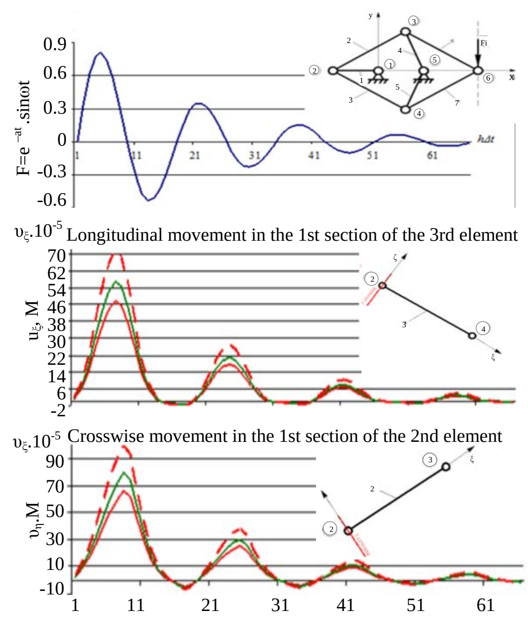


Fig. 6: Maximum longitudinal and transverse displacements in the elements of the mechanism under the action of only a concentrated force F = e-αtsinωt: linear isotropic; nonlinear isotropic; linear anisotropic case

The Newmark method is used to solve the dynamics equations. The integration step with respect to the time $\Delta t$ is chosen based on the angular velocity of the leading links and on the parameters that determine the required accuracy of the reproduced processes and the stability of the integration method. Within each step, the

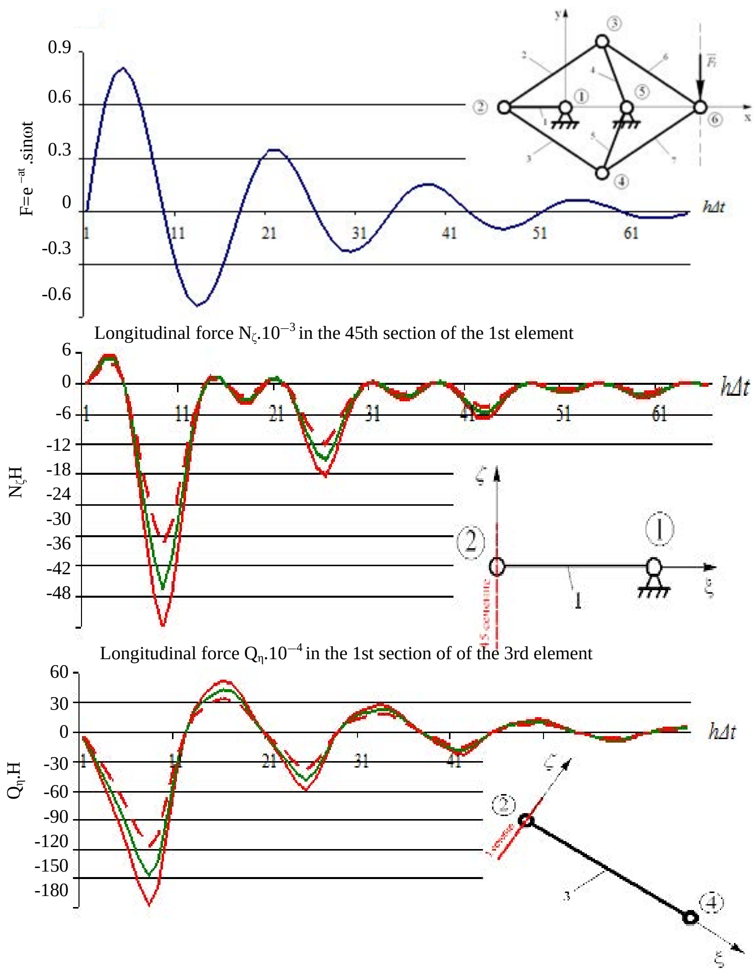


Fig. 7: Maximum longitudinal and transverse forces in the elements of the mechanism under the action of only a concentrated force F = e-αtsinαt: linear isotropic; nonlinear isotropic; linear anisotropic case

time taken into account is the change in both kinematic and elastic displacements, velocities and accelerations.

In the study of the dynamics of the RPM, elastic damping in the materials of the links is taken into account. The transformation of nodal displacements and elastic reactions of an element from the general coordinate system to local ones is carried out according to the developed algorithm, and internal stresses are located.

Figures 10-13 show the changes in longitudinal and transverse displacements, longitudinal forces from the joint action of inertia forces and concentrated force, in the links of the APM during the four turn of the leading link. The values obtained for linear and nonlinear calculations are compared.

In the APM, the largest longitudinal movements are observed in the elements 4 and 7, transverse displacements in the elements 4 and 6. Nonlinear movements in the elements 2, 4 and 6 exceed linear by 30% and in the remaining elements by 15%. The most loaded in the longitudinal direction are the nodal cross sections of the elements 1-6.

As a result of the study of the dynamic stress-strain state of elastic flat and spatial mechanisms with geometrically nonlinear anisotropic links.

First discrete finite element calculation dynamic model of elastic deformation of plane and spatial mechanisms with geometrically nonlinear anisotropic and isotropic links was proposed.

A finite-computational scheme for solving the basic system of non-linear equations of motion with variable linear and non-linear stiffness matrices as well as a mass matrix and elastic damping of the system with a representation of the coefficients of the latter through lower vibration frequencies of mechanisms with a choice for each mechanism of the time step, ensuring the stability of the calculation. A unified program complex, designed as a standard program, is compiled for the calculation of

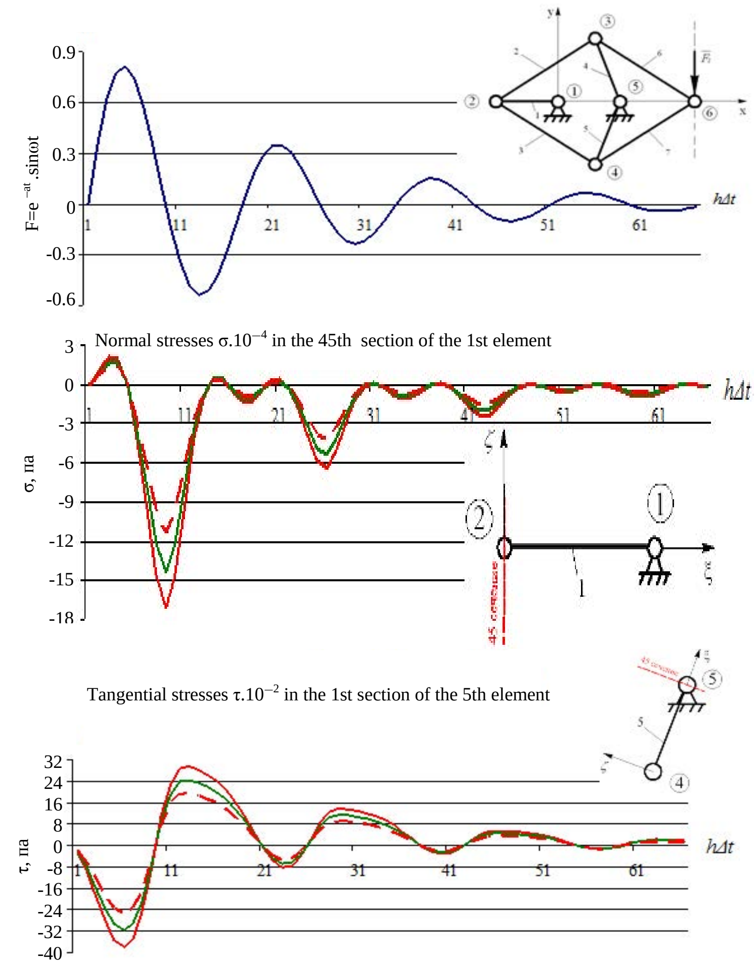


Fig. 8: Maximum normal stresses, tangential stresses in the elements of the mechanism under the action of only a concentrated force F = e-αtsinωt: linear isotropic; nonlinear isotropic; linear anisotropic case

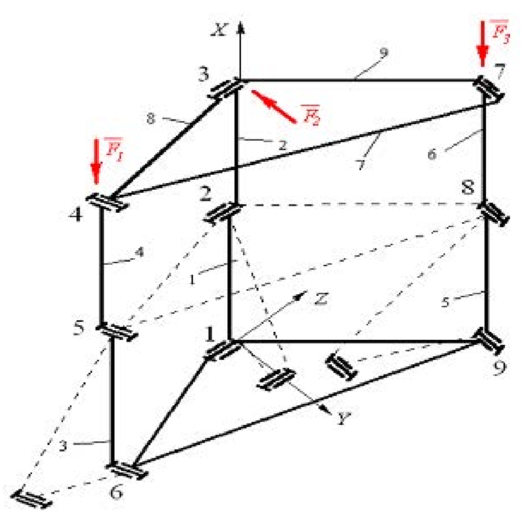

Fig. 9: Finite element planning model

dynamic and kinematic parameters, internal forces and stresses in the elements of plane and spatial mechanisms from the action of external variable forces and inertia forces. A standard program can be used to calculate the dynamic elastic state of plane and spatial mechanisms with known kinematic analysis.

Multivariate calculations of the values of dynamic displacements and internal forces in the elements of mechanisms with different initial parameters and speeds of the leading link; the results of calculations are analyzed and presented in the form of graphs and diagrams.

The basic FEM relations for planar and spatial mechanisms with rod elements for large displacements are obtained and nonlinear stiffness matrices of the first and second order CE with the use of the Lagrange equation are calculated. Algorithms have been developed and a package of applied programs has been compiled in a high-level language for calculating, analyzing and estimating the dynamic VAT of mechanisms with

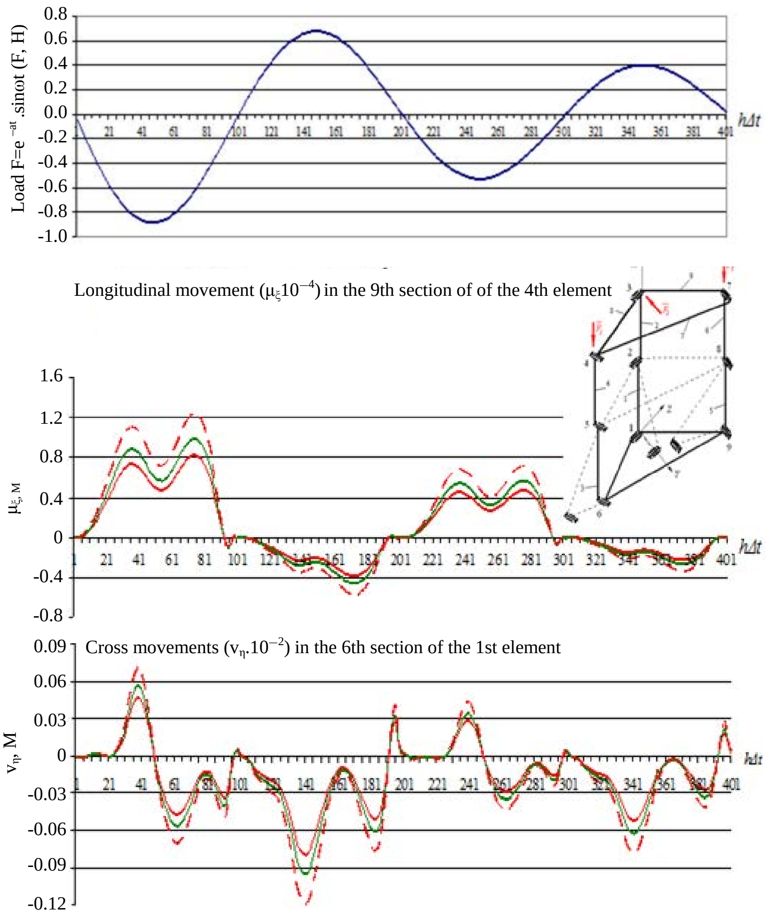


Fig. 10: Change in longitudinal and transverse displacements in the links of the RPM with the joint action of inertia forces and the concentrated force F = e-αtsinωt: linear isotropic; nonlinear isotropic; linear

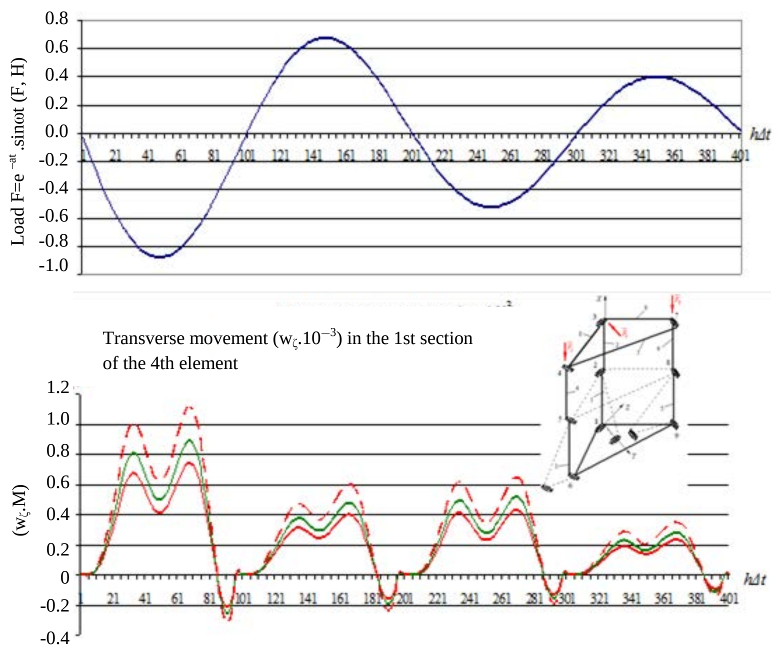


Fig. 11: Continue

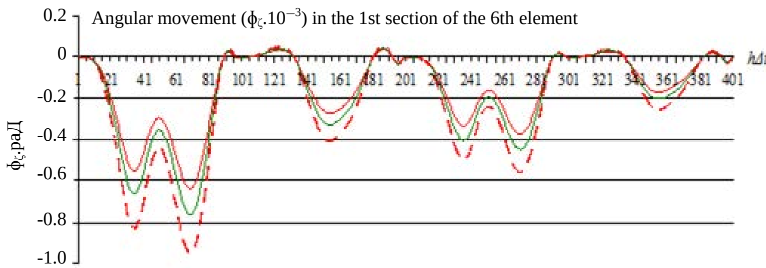


Fig. 11: Change of angular displacements in the links of the RPM with the joint action of inertia forces and the concentrated force F = e-αtsinωt: linear isotropic; nonlinear isotropic; linear anisotropic case

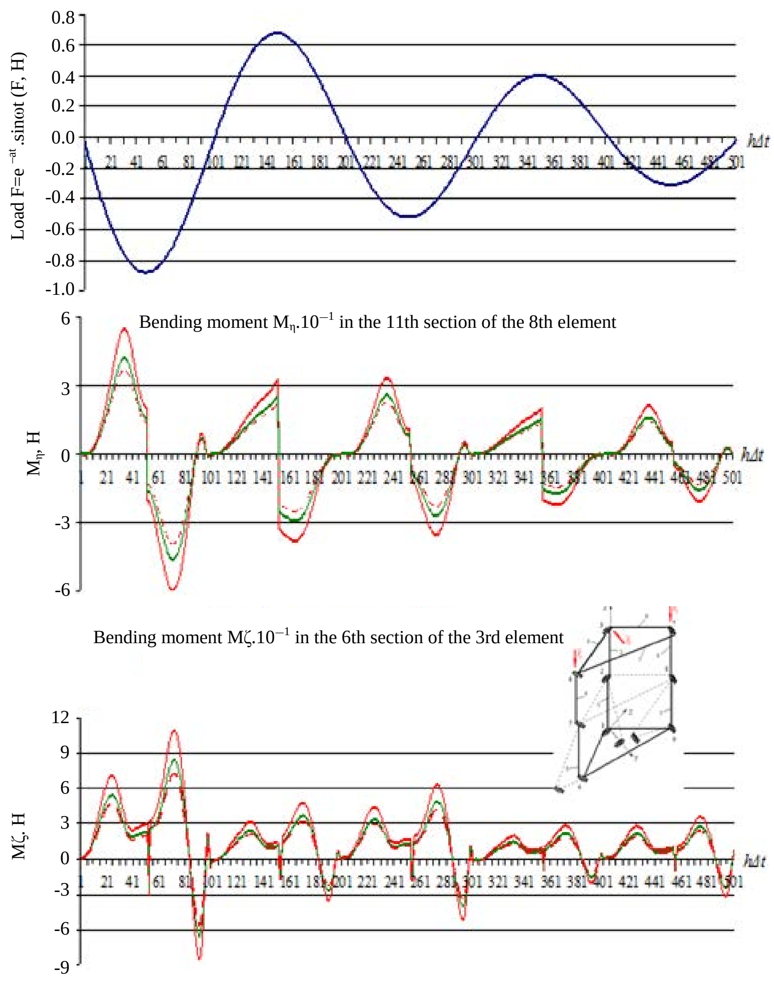


Fig. 12: Change in longitudinal and transverse forces in time in the links of the RPM with the joint action of inertia forces and the concentrated force F = e-αtsinωt: linear isotropic; nonlinear isotropic; linear anisotropic case

anisotropic and isotropic elastic links for geometric nonlinearity depending on arbitrary external variable forces and initial physical and geometric characteristics. The obtained results fully correspond to the tasks and completely cover their solutions. The results and conclusions of studies of the dynamic VAT of nonlinearly elastic plane and spatial mechanisms with rectilinear anisotropic links, make it possible to select the most

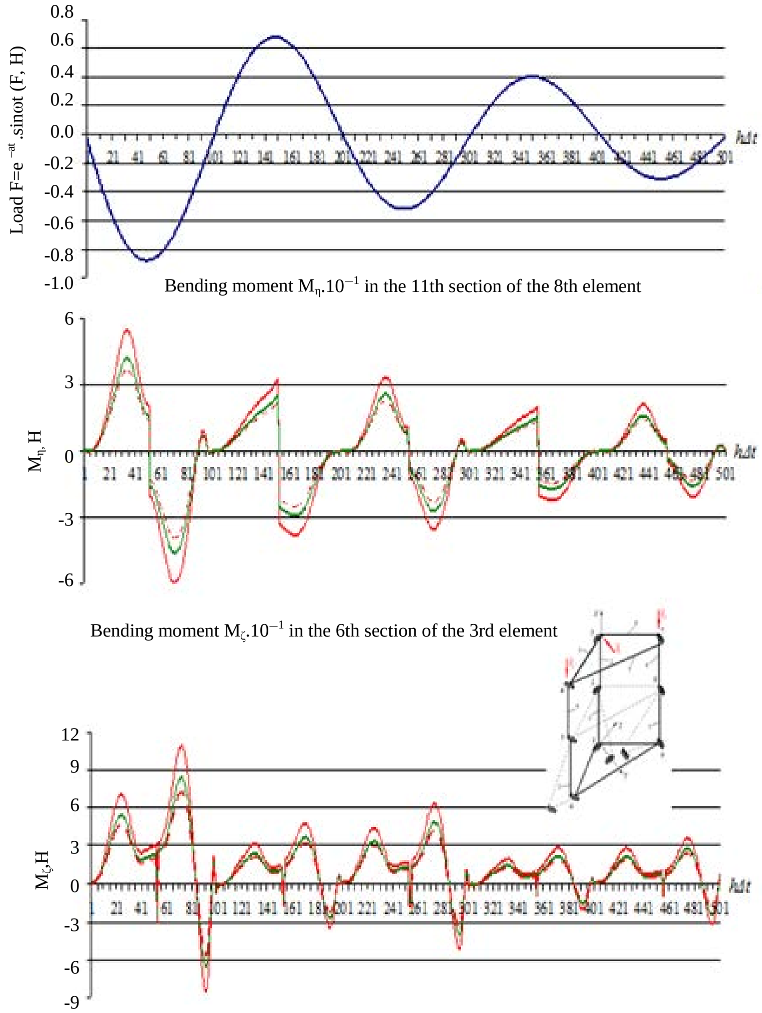


Fig. 13: Change of bending moments in time in the links of the RPM with the joint action of inertia forces and the concentrated force F = e-αtsinωt: linear isotropic; nonlinear isotropic; linear anisotropic case

optimal parameters of the mechanisms, establish maximum stresses, forces, elastic displacements, velocities and accelerations at any points of the computational elements of mechanisms under various loading conditions. Part of the results of scientific research and developed standard programs is used in our research institute.

The proposed mechanical-mathematical discrete design model for the dynamics of nonlinear elastic spatial mechanisms with different kinematic pairs and a single program complex for calculating dynamic VAT compiled on its basis for large elastic displacements of the links of mechanisms, in contrast to modern achievements, allows to take into account the elastic anisotropic properties of each link, the influence of static and various dynamic forces.

## CONCLUSION

Thus, the created mechanical-mathematical discrete design model can be used to develop and evaluate the efficiency of manipulators of various robotic systems in the interests of the Armed Forces of the Republic of Kazakhstan as combat, engineering and rear robots.

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