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Optimization of Dynamic Gear System Parameters of Metal Cutting Machine

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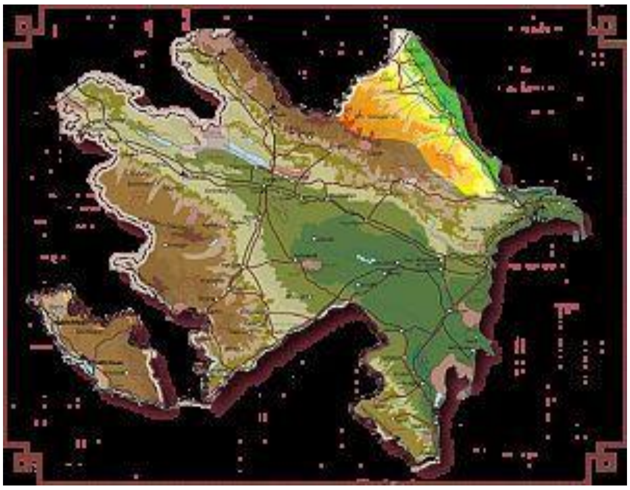
Abstract:- Work is devoted to questions of optimization of parameters oi dynamic system of tooth gearings of the machines, allowing on a design stage to choose such combinations of elements of a design at which realization are created machines with the best dynamic properties, operational and technological parameters. Methods have been developed for optimizing the oscillatory processes of machines in the parameter space of electromechanical systems, which allow: using simple analytical expressions to establish a relationship between the operational, design parameters and dynamic characteristics of the designed machine; predict the expected vibration level; synthesize a machine according to specified dynamic characteristics; create machines with the lowest dynamic coefficient and a machine with the best vibration damping. A technique has been developed for the optimal design of machine drives, allowing at the design stage to select such design and operational parameters, when assigning which the elastic system is significantly freed from the harmful effects of oscillatory processes and the machines become more productive, reliable, durable and minimally metal-intensive. The practical focus of the work opens up broad opportunities for the implementation of its results by design, engineering, and research organizations involved in the calculation, design, and research of machines and apparatus.

Keywords*: precision, durability, reliability, vibration resistance of machines and devices, create machines with the lowest dynamic coefficient.*

1. Introduction



Problem Statement:

Among the requirements set for modern mechanical engineering, performance, efficiency, accuracy, durability, reliability, vibration resistance of machines and devices are very important. The development of modern mechanical engineering, characterized by a continuous increase in the speed of movement of mechanisms, drive power, loads on parts, puts forward in the first place the optimization of the parameters of the dynamic system of mechanisms and machines.

Based on the above, we can conclude that the problem to which this work is devoted is an urgent problem and meets the challenges of further progress in modern mechanical engineering.

Analysis of the Literature:

Almost all industries use gears or gear mechanisms. Gear mechanisms are especially common in the machine tool and automotive industries. They can be used at both low and high speeds.

The design of high-quality, high-performance, accurate, durable mechanisms and machines should proceed primarily from the search for the optimal parameters of the dynamic system of the entire structure as a whole.

Mathematically, the search for optimal machine parameters consists in finding the parameter vector $\alpha = \{\alpha_1, \alpha_2, \dots, \alpha_j\}$ for which parametric, $\alpha_j^* \leq \alpha_j \leq \alpha_j^{**}$and functional $R_s = \{\alpha_1, \alpha_2, \dots, \alpha_j\} \geq 0$ constraints, at the same time the values of the optimized functions Φ_1(a), Φ_2(a),… reach their absolute extrema.

The solution to this problem is based on a global study of the parameter space with respect to local quality criteria. Modern problems of methodology for solving multicriteria problems are considered in works [1, 2, 3, 4, 5, 6, 7, 10]. Despite the existence of a large number of works in this area, at present there is no universal optimization method that can be used for a wide range of practical problems. The choice of this or that method for the successful solution of a multicriteria and multiparametric problem is associated with the need to solve the following questions in each specific case: identifying the type of problem; formation of algorithms for calculating target functions and functional limitations; formation of an admissible set of models; the formation of a decision rule and its use to streamline the options for feasible decisions, etc.

The methodology and software developed by us for the formulation and solution of problems of searching for admissible and Pareto models of machines was created on the basis of the method for investigating the space of parameters (LP-search), proposed by IM Sobol and RB Statnikov [9]. In this case, both well-known programs and algorithms are implemented, modified in relation to the problems of machine design, and new ones developed by the author and this article.The peculiarities of these algorithms for finding the optimal parameters are that they take into account the specifics of the design problems of gear mechanisms of metal-cutting machines and allow at the design stage to identify, from the standpoint of the proposed quality criteria, the real resource capabilities of the machines under study.

The purpose of the article is to develop and implement a methodology for optimizing the parameters of a dynamic system of gear transmissions of metal-cutting machines, taking into account their real characteristics, which makes it possible to predict the expected level of vibrations, to synthesize mechanisms with the best design, operational parameters, the appointment of which resulted in productive, durable, vibration-resistant machines with minimal dynamic loads and metal consumption.

2. Section 1. Dynamic Model and the Equation of its Motion

Let us consider the optimization of the parameters of the dynamic system of a numerical control lathe (SNC) model 16K20T1, the kinematic diagram of which is shown in fig. 1.

The drive includes: a source of motion - an asynchronous electric motor with a power of 10 kW with a rotation of n = 1460 rpm, transmission mechanisms - a V-belt transmission, gear speeds and working machines - a spindle assembly and a support.

The main drive of the 16K20T1 CNC machine consists of an electric motor, a five-speed gearbox and a spindle assembly. The main drive of the machine at pshp = 71 rpm, the dynamic model that is shown in Fig. 2 has 12 degrees of freedom [8], the corresponding differential equations of motion have the form:

$$J_1\varphi_1 + \beta_{01}\varphi_1 + \beta_{12}(\varphi_1 - \varphi_2) + c_{01}\varphi_1 + c_{12}(\varphi_1 - \varphi_2) = M_1$$

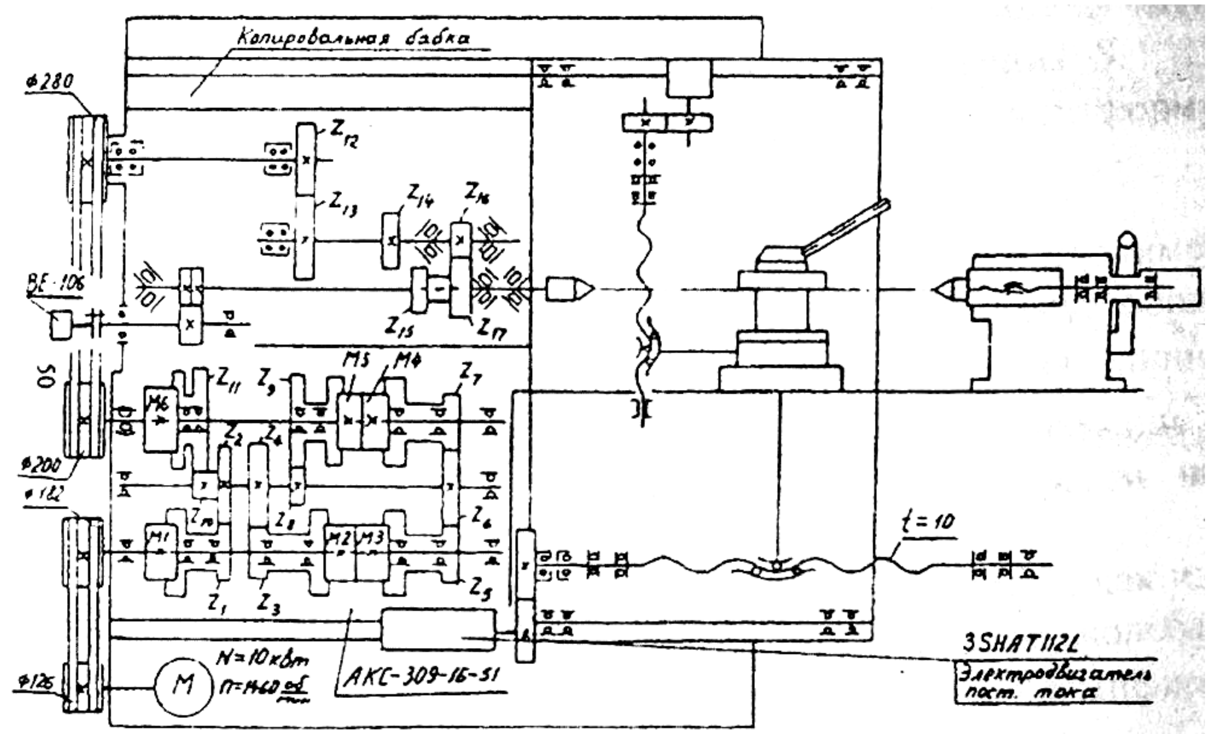


Fig.1. Kinematic diagram of the drive machine with CNC 16K20T1

$$\frac{1}{2w_3 M_k}\mathrm{M}_1 + \frac{S_k}{2M_k}\mathrm{M}_1 = S$$

$$J_2\ddot{\varphi}_2 - \beta_{12}(\dot{\varphi}_1 - \dot{\varphi}_2) + \frac{R_2}{R_3}\beta_{23}\left(\frac{R_2}{R_3}\dot{\varphi}_3 - \dot{\varphi}_4\right) - c_{12}(\varphi_1 - \varphi_2) + \frac{R_2 C_{23}}{R_3}\left(\frac{R_2}{R_3}\varphi_2 - \varphi_3\right) = 0$$

$$J_3\ddot{\varphi}_3 - \beta_{23}\left(\frac{R_2}{R_3}\dot{\varphi}_2 - \dot{\varphi}_3\right) + \beta_{12}(\dot{\varphi}_3 - \dot{\varphi}_4) - \beta_{23}\left(\frac{R_2}{R_3}\varphi_2 - \varphi_3\right) + c_{12}(\varphi_1 - \varphi_2) = 0$$

$$J_{11}\varphi_{11} - \beta_{1011}\left(\frac{R_{10}}{R_{11}}\varphi_{10} - \varphi_{11}\right) + \beta_{1112}(\varphi_{11} - \varphi_{12}) - c_{1011}$$

$$\left(\frac{R_{10}}{R_{11}}\varphi_{10} - \varphi_{11}\right) + c_{1112}(\varphi_{11} - \varphi_{12}) = 0$$

$$J_{11}\varphi_{11} - \beta_{1112}(\varphi_{11} - \varphi_{12}) - c_{1112}(\varphi_1 - \varphi_{12}) = M_p \quad (1)$$

where β_{ij}, c_{ij}, J_i - are parameters characterizing viscous resistance, torsional stiffness and moment of inertia of the masses of the system under consideration, respectively: $\varphi_1, \varphi_2, \dots, \varphi_{15}$ - are generalized coordinates-absolute angular displacements of the corresponding masses: $R_1, R_2, \dots, R_{12}$ - radiuses of pulleys and gears, to which concentrated masses are reduced; M_1 electromagnetic torque of the induction motor; M_p - moment of cutting forces; M_k, S_k is the critical moment and slip of the motor according to the static mechanical characteristic.

The second equation of system (1) is the equation of the "dynamic response" of an induction motor, which characterizes the dependence of the electromagnetic torque M_1 on slip *S*.

The values of the moment M_p - from the cutting forces P_z are calculated by the formula

$$RP_z = R(A + Df_{ш}^2 + E\ sin\ sin\ \omega_{ш} t\ ) \quad (2)$$

Formula (2) is obtained by interpolating the experimental dependence of the cutting force on the speed [8]. Here R is the radius of the workpiece $f_{ш} = \omega_{ш}/2\pi$, $\omega_{ш}$ - angular spindle speed; *t* - the current time value.

Interpolation coefficients A.B.D.E. characterizing the dependence of the cutting force P_z on speed, are entered into the computer as initial data.

The damping coefficient of a mechanical system, as is known from vibration theory, changes with a change in the vibration frequency. When solving differential equations, the damping coefficients $\beta_{ij\,,}$ are set for two arbitrary frequencies *f₁*, *f₂* lying in a certain frequency range of interest to us $f_{min} \leq f \leq f_{max}$.

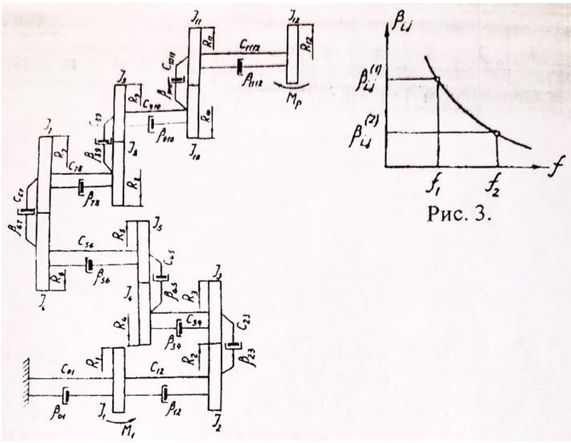
For other frequencies from the interval $f_{min} \leq f \leq f_{max}$ the damping coefficients are calculated automatically using exponential interpolation (Fig. 3).

$$\beta_{ij\,,}(f) = \beta_{ij}^{(1)} e^{\beta y(2)} \qquad (3)$$

3. Section 2. Parameters of the Dynamic Model

The materials of compiling dynamic models of drives of metal-cutting machines can be found in detail in the monograph of Professor Kh.S. Samidov [8], therefore, we will not dwell on their implementation here, but we will only recall the techniques and give some calculation results.

The dynamic system of drives of metal-cutting machines from the presence of circumferential and spacer forces of gears is a complex bending torsion system.



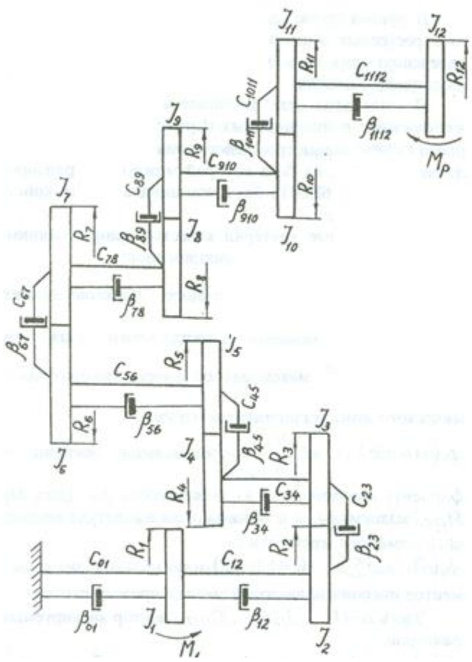


Fig. 2. Dynamic model of the main drive of a CNC lathe 16К20Т1 *nшп = 71 об/мин*

Therefore, the torsional compliance *($e_{кр}$)* of such a system should be summed up with the reduced equivalent torsional compliance during bending of shafts and subsidence of supports *($e_{equiv-ig}$)* and the intrinsic compliance of the gear train *($e_{зп}$)* i.e.

$$e = e_{кр} + e_{экв-изг} + e_{зп}$$

Calculations and experimental checks show that the torsional compliance (rigidity) of the shafts of the main drive of the 16K20T1 CNC machine is on average 52% of the total total compliance of the drive of this machine. In this case, the bending of shafts, deformation of bearings and gears reduced to torsional compliance is 26%, and the contact deformations of keyed and spline joints reduced to torsional compliance is 22% [8]. The masses (m_i) and moments of inertia of the parts (J_i) of the gears of metal-cutting machines have a mostly cylindrical shape, so their values were determined by the analytical method.

The parameters of the dynamic model of the main drive of a 1620T1 CNC lathe at a spindle speed of $n_{шп}$ = *71 rpm* are presented in Table 1.

Table 1.

Machine parameters 16K20T1	1	2	3	4	5	6	7	8	9	10	11	12
J_i,НМс^2	0,120	0,032	0,013	10^{-4}·4	0,056	0,0317	0,491	0,008	0,008	0,022	0,056	0,112
C_{ij},10^{-4} НМ/рад	1,538	1,667	1,645	28,57	12,19	1,767	0,179	1,020	17,24	7,042	13,70	125,0
R_i,М	0,091	0,046	0,061	0,022	0,073	0,100	0,140	0,07	0,07	0,052	0,105	0,105
$\beta_{ij}^{(1)}$ НМс/рад	1,03	11,16	11,01	26,96	115,0	65,76	6,68	96,68	825,1	337,0	665,5	11796
$\beta_{ij}^{(2)}$ НМс/рад	7,99	8,60	8,50	14,80	63,40	4,50	0,46	6,50	56,03	22,80	44,52	325,0

4. Section 3. Criteria for Optimization and Limitations

The optimization problem is reduced to the study of the resource capabilities of the machine under consideration and the enthusiasm of these capabilities through the modernization of the existing model.

This goal was achieved by varying the rigidity and inertial parameters of the machine in the vicinity (± 30%) of the parameters of the current model and finding the best model on the basis of which it is possible to modernize the drive of the 16K20T1 machine without fundamental structural changes.

The minimality of the following target functionals is accepted as local quality criteria:

$\phi_1(a) = \frac{1}{f_{max}-f_{min}} \int_{min}^{max} M_{1112} df$ – the average value of the amplitude of the dynamic moment of the spindle unit of the machine;

$\phi_2(a) = max M_{1112}$ - the maximum value of the amplitude of the dynamic $f_{min} \leq f \leq f_{max}$ of the moment of the spindle assembly;

$\phi_3(a) = min \quad _{12} = min \frac{M_{12max}}{M_{12min}}$ – minimum value of dynamic factor $\quad _{12}$ in toughness c_{12}. Here M_{12max}, M_{12min} maximum and minimum dynamic moment amplitudes in stiffness c_{12} $\phi_4(a) = min \sum \quad J_i(i = 1,2, \ldots, 12)$ the minimum value of the moments of inertia of the moving links of the machine drive. Here $a = (J_i, \ldots, J_{12}, c_{01}, \ldots, c_{1112})$ - vector of variable parameters.

$\phi_5(a) = min\phi_1$- minimum value of natural frequency.

Constraints on variable parameters are given as

$0{,}7J_i \leq J_i \leq 1{,}3J_i$;

$0{,}7J_{ij} \leq c_{ij} \leq 1{,}3J_{ij}$

Functional limitations were as follows:

$M_{01max} \leq M_k; M_{12max} \leq M_k; max\sigma_{ni} \leq [\sigma_{ni}], i = 1,2, \dots$ where M_k – critical moment of the engine; $M_{0,1max}, M_{12max}$ – maximum dynamic loads in stiffness c_{01} и c_{12}; $\sigma_{ni}, [\sigma_{ni}]$ – reduced stresses in gear transmission elements.

5. Section 4. Problem Solving and Analysis of Results

The problem is solved on a digital computer using an automated algorithm, the block diagram of which is shown in Fig. 4 [8].

The number of parameters involved in the formation of torsional vibrations of the system is more than 60. Of these, 24 parameters (12 moments of inertia and 12 stiffnesses) varied within up to ± 30% of the initial values of the real structure. The process of discrete sensing of the space of the studied parameters of the implemented parameters was carried out using pseudo-random Sobol points [9]. In this case, the working body (mass J_{12}) was subjected to external influence M_p determined by the formula (2).

The limits of changing the parameters of the main drive of the 16K20T1 machine are presented in Table 2.

Table 2.

J_i, Nms ^{2}	Limits of change	$c_{ij} \cdot 10^{-4}$ Nm / rad	Limits of change
J_1	0,0840÷0,1560	C_{01}	1,0766÷1,9994
J_2	0,0224÷0,0416	C_{12}	1,1669÷2,1671
J_3	0,0091÷0,0169	C_{23}	1,1515÷2,1385
J_4	$10^{-4}2{,}8 \div 10^{-4}5{,}2$	C_{34}	1,9999÷3,7141
J_5	0,0392÷0,0728	C_{45}	8,5330÷15,847
J_6	0,2219÷0,4121	C_{56}	1,2369÷2,2971
J_7	0,3437÷0,6338	C_{67}	0,1253÷0,2327
J_8	0,0056÷0,0104	C_{78}	1,4140÷2,6260
J_9	0,0056÷0,0104	C_{89}	12,068÷22,412
J_{10}	0,0154÷0,0286	C_{910}	4,9294÷9,1546
J_{11}	0,0392÷0,0728	C_{1011}	9,5900÷17,810
J_{12}	0,0784÷0,1456	C_{1112}	87,500÷162,50

The total number of experimental *N=30*; the number of models that satisfy parametric and functional constraints was 24, of which 6 modules satisfy all 5 criterion constraints $\alpha^1, \alpha^7, \alpha^{12}, \alpha^{19}, \alpha^{22}, \alpha^{24}$.

A fragment of the results of calculations to optimize the parameters of the main drive of the 16K20T1 CNC lathe is presented in the test table 3. Criterion constraints are marked in the table with a solid line $\phi_\nu = \phi_\nu(\alpha^1)$, ν=1,2,…,5. Models $\alpha^7, \alpha^{12}, \alpha^{19}, \alpha^{22}$и α^{24} surpass the model in all the assigned quality criteria α^1 the current design of the machine 16K20T1.

Table 3. Fragment of the results of optimization of the parameters of the main drive of a CNC machine tool 16K20T1

№ models	ϕ_1 average	№ models	ϕ_2- max value	№ models	ϕ_3-min value	№ models	ϕ_4-min value	№ models	ϕ_5-min value
1	2	3	4	5	6	7	8	9	10
2	95,8	26	119,6	9	1,02	14	0,865	7	19,58
24	96,1	11	122,7	7	1,06	7	0,876	24	19,62
20	96,8	17	124,7	24	1,07	19	0,882	26	20,00
18	96,8	7	125,9	14	1,11	26	0,895	14	20,01
19	96,9	24	126,2	26	1,13	8	0,902	17	20,08
22	97,0	19	126,3	22	1,15	28	0,918	12	20,43
6	97,8	12	128,9	12	1,18	17	0,934	19	20,48
12	98,2	22	235,1	2	1,21	23	0,941	8	20,54
7	98,3	13	137,3	11	1,23	21	0,960	5	21,54
9	98,4	2	138,5	8	1,25	22	0,980	28	21,56
1	98,7	14	139,6	19	1,27	12	1,058	22	22,07
25	98,8	23	139,9	20	1,29	24	1,120	21	22,38
16	98,9	8	140,6	17	1,30	5	1,170	10	22,60
11	99,0	20	144,2	1	1,33	3	1,215	23	23,26
4	99,1	25	146,7	5	1,37	1	1,236	2	23,67
26	99,3	1	148,5	21	1,41	10	1,244	1	23,72
14	99,4	21	149,5	25	1,43	30	1,261	20	23,96
21	100,0	5	150,0	28	1,45	2	1,278	30	23,98
13	100,1	9	150,7	29	1,47	9	1,291	3	24,17
29	100,2	3	150,3	10	1,49	29	1,305	9	24,39
17	100,3	23	159,9	4	1,52	15	1,315	4	24,55
10	100,4	29	161,1	6	1,56	4	1,328	16	24,85
8	100,9	6	163,8	16	1,59	13	1,353	29	25,01
5	101,4	4	166,2	18	1,63	16	1,372	15	25,12
28	101,7	10	168,8	30	1,68	11	1,402	6	25,36
3	102,0	18	173,3	3	1,71	27	1,423	11	25,45
27	102,7	16	276,9	13	1,75	25	1,450	18	25,81
30	102,8	30	178,6	23	1,79	20	1,482	25	25,94
15	102,9	27	183,6	15	1,82	6	1,508	13	25,98
3	103,0	15	200,8	27	1,85	18	1,606	27	26,06

The most interesting and optimal in this formulation of the problem is the model α^7.

The dynamism coefficient and weight (mass) of this model are, respectively, 20 and 30% less than that of the model α^{-1}- the original design.

6. Results and Discussion

1. As a result of solving the optimization problem, it was possible to improve the quality of the main drive of the machine under consideration according to a number of quality criteria: dynamic loads, dynamic coefficients of the elastic system and the weight (mass) of parts are reduced by 20%, 24% and 30%, respectively.

2. On the basis of the research carried out, the relevant recommendations have been developed, which have been introduced at the Azerbaijan Pipe-Rolling and Tbilisi Machine-Tool Plants for use in calculations, analysis of dynamics and optimal design of machines and apparatus.

3. Comparison of the magnitude and nature of the change in the elastic moments of the initial and optimized models show that the dynamic characteristics of any machine can be significantly improved by selecting its design parameters according to the developed method.

4. The practical orientation of the work opens up certain opportunities for the implementation of its results by design, engineering and research organizations involved in the study of dynamics and optimal design of machines for various purposes

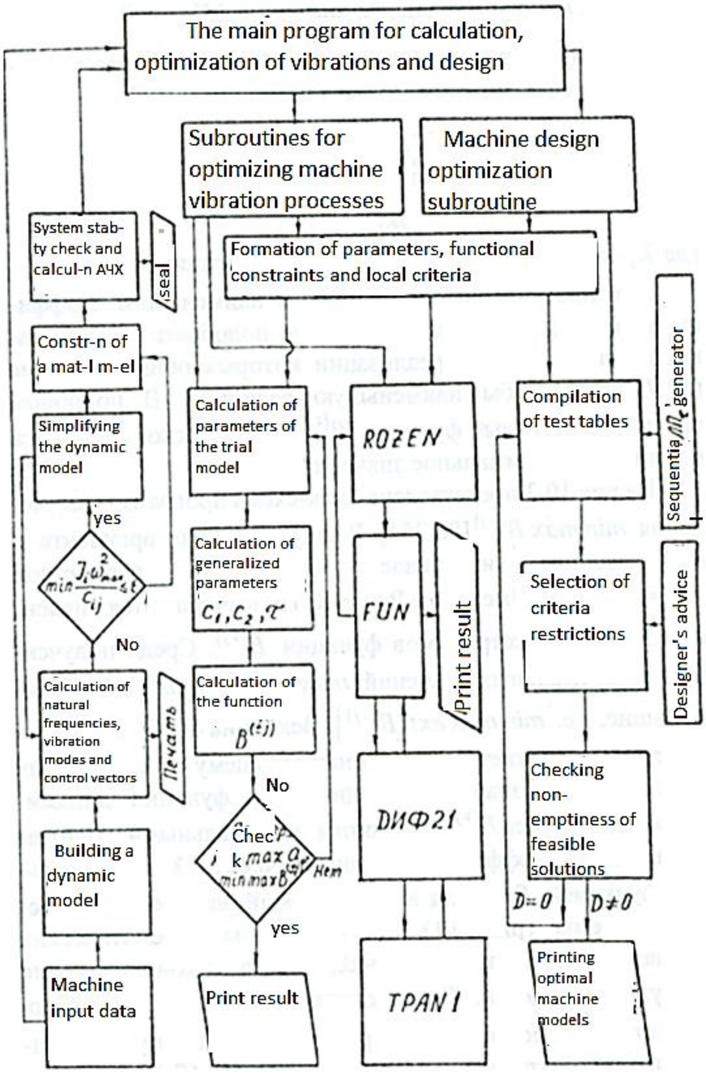


Fig. 4. Flowchart for calculating, optimizing vibrations and designing machines

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