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DEVELOPMENT AND APPLICATION OF COMPUTER MODEL TO STUDY THE MODES OF DYNAMIC LOADING IN MECHANICAL OSCILLATORY SYSTEMS

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Створено комп'ютерну імітаційну модель дослідження режимів динамічної навантаженості механічних коливальних систем. Розроблена модель орієнтована на дослідження динаміки одноступінчастої евольвентної зубчастої передачі, за умови дії зовнішнього зусилля змінного характеру. Комп'ютерна модель реалізована засобами моделюючого середовища MATLAB-Simulink, з використанням принципів електронного моделювання. На основі результатів моделювання отримано оцінку динамічних зусиль у вузлах зубчастої передачі, залежно від виду функції навантаженості

Ключові слова: Simulink-модель, електронне моделювання, динамічне навантаження, коливальна система, зубчаста передача

Создана компьютерная имитационная модель исследования режимов динамической нагруженности механических колебательных систем. Разработанная модель ориентирована на исследование динамики одноступенчатой эвольвентной зубчатой передачи, при условии воздействия внешнего усилия переменного характера. Компьютерная модель реализована средствами моделирующей среды MATLAB-Simulink с использованием принципов электронного моделирования. На основе результатов моделирования получена оценка динамических усилий в узлах зубчатой передачи, в зависимости от вида функции нагруженности

Ключевые слова: Simulink-модель, электронное моделирование, динамическая нагрузка, колебательная система, зубчатая передача

1. Introduction

One of the most complex mechanical systems that are widely used in machine building and instrument making is different types of reducers. The functionality of such mechanical devices may be different, depending on the type of application tasks. The most commonly used are the gearboxes with toothed gears that can be considered the typical nodes of mechanisms [1]. The main and the most responsible element of the tooth gear design is the kinematic couples with shaft sections that rotate on bearings. Weights on bearings may be considered as localized or distributed, they have different functionality, depending on the tooth gear type. The level of shaft oscillations and dynamic forces on their bearings considerably affect the basic parameters for tooth gear operation – accuracy, dynamic load, resource, performance, reliability, etc. [2].

An in-depth study of the dynamics of tooth gears implies consideration of influence of various factors on dynamic loading and the oscillation process in the framework of internal and external dynamics. Research into dynamic processes, with the aim of simplification, is mainly associated with exploring dynamics of single-stage gears. Knowledge of the reaction of an individual section to alternating loading is important for the dynamic analysis of the whole mechanical system both in the case of single-stage and multistage gears. Tooth gear as a complex mechanical oscillatory system, contains a large number of various structural elements. The reaction of each of them to the influence of dynamic loading differs significantly. Schemes of dynamics of mechanical oscillatory systems in the classical formulation are composed of typical sections, which, in terms of their reaction to the vibration and shock effects, may be represented in the form of inertial masses, springs and dampers [3]. Equivalent mechanical oscillatory system may be represented in the form of damped linear spring-mass system [4]. Reliable information about the actual dynamic processes in the oscillatory system, subjected to the influence of alternating loads, can be obtained based on mathematical modeling and computer experiment, connected with it [5, 6].

An important stage in the study of modes of dynamic loading of tooth gear as a mechanical oscillatory system is an analysis of dynamic phenomena and consideration of the processes associated with surge and dumping of loading. Typical modes of surge and dumping of loading, for example, for a metal cutting machine, are modes of tool cutting in and coming out, for a rolling mill – grip of a work piece with rolls [7], etc. Analysis of operating conditions of tooth gears that are used, for example, in rolling production, shows that despite the specific features of operation (depending on the particular equipment), they are characterized by high loadings with a significant dynamic component [7].

Thus, we can conclude that the study of modes of dynamic loading of mechanical oscillatory systems of the class of tooth gears makes it possible to perform an assessment of dynamic forces within their design elements and is an important scientific and technical problem. This makes relevant to design and implement appropriate simulating computer models, based on the use of modern simulating environments such as MATLAB-Simulink, into the broad research practice.

2. Literature review and problem statement

Development of computer models of studying the influence of load changes on the dynamics of oscillations in tooth gear requires consideration of contemporary trends in the development of means of computer modeling, their specific features and the classification of tooth gear elements by the frequency range of their oscillations. To discover the physical essence of flow of dynamical processes and to obtain data on the modes of dynamic loading of tooth gear, different approaches, based on developing and studying mathematical models of discrete and continuous systems, are used [8]. Modeling of dynamic processes in mechanical oscillatory systems and elastic environments are the focus of a number of publications [9–15].

In paper [9], an analytical model based on the method of finite element together with the method of least squares was created to determine the dynamic forces in a rotor system. Article [10] explores the problem of dynamics for elastic media; in this case, dynamic coefficients of stress intensity are calculated as load frequency functions. Study [11] is dedicated to computer simulation of mechanics of destruction of structures under the influence of dynamic forces based on prediction of strength characteristics of its parts. Three-dimensional computer simulation of mono-axial dynamic force on the elastic material is examined in paper [12]. In this case, the cell automat is used as a computing method. In article [13], the approach to the implementation of models of elasticity of inhomogeneous materials within the method of discrete elements in combination with the cell automat was used. Study [14] is dedicated to the development of schemes of implementation of the vibro-impact system with one flat spring and locally concentrated masses. As a solution to the dynamic problem, the relationship between kinematic characteristics and parameters of the stressed state of oscillation system was obtained. In paper [15], mathematical modeling of the dynamic system "pendulum-cart" was performed. The obtained model allows conducting a wide nonlinear analysis of the systems, based on the response that includes dynamic transitions, bifurcation and chaos.

Analyzing the features of each study, we may conclude that a poorly developed direction in effective studying the dynamics of tooth gears, taking into account the influence of external factors on dynamic loading and oscillatory process, is the insufficient level of implementation of computer models, which would take into account linear and radial oscillations of tooth gear in three planes.

3. The aim and tasks of the study

The aim of present work is the creation of information technology to explore dynamics of mechanical oscillatory systems of the evolvent tooth gears under the mode of periodic loading change.

To achieve the set goal, the following tasks were to be solved:

– based on the mathematical model of tooth gear dynamics [16], which consists of 14 linear differential equations of second order, describing the longitudinally-transverse and radial oscillations of its elements in different planes, it is necessary to develop a computer model which would provide visual observation of reaction to external loading of alternating character of consequent oscillations in the studied mechanical system:

– φ, φ_1, φ_2, φ_3 – torsional (radial) vibrations of tooth wheels and shaft sections "engine–tooth wheel", and "tooth wheel-driving mechanism", respectively,

– ψ_1^y, ψ_1^z, ψ_2^y, ψ_2^z – radial oscillations of gears around axes y and z,

– x_1, x_2, y_1, y_2 – longitudinal oscillations of shafts on axes x and y;

– a computer model to be created should provide for the possibility of taking into account basic weight-inertial and rigidity tooth gear parameters and their changes within the limits, defined by the conditions of the experiment;

– results of the simulation should be represented in the form of combined oscillograms of functions of external loading and of reaction of the corresponding tooth gear element. Visualized simulation results should provide for the possibility to determine the dependence of oscillation amplitude of any type of tooth gear elements on the type of loading function and the ratio of loading operation periods and shaft rotation.

4. Development of computer model and procedure for conducting experimental research

In terms of tooth gear loading, a loading surge is a mode of overloading of particular intensity, which, under certain conditions, may lead to overstrain in sections of mechanical system [17]. Similar phenomena arise at load dumping (for example, at the tool output, metal output from rollers, etc.) that is caused by inertial properties of a mechanical system [17]. The loading moment, acting on the output shaft (working body), may be recorded [17] in the form $M_C(t)=\eta(t)M_c^0$, where M_c^0 is the value of the loading moment in the steady mode (after a surge or before dumping the load); η(t) is the functional multiplier. The most typical load cases of "rectangle" and "saw-tooth" pattern (Fig. 1, 2) with the defined values of period (T) and amplitude (A) were selected as a functional multiplier.

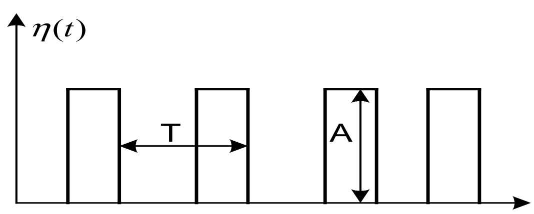

Fig. 1. Graph of dependence of η(t) of the "rectangle" type

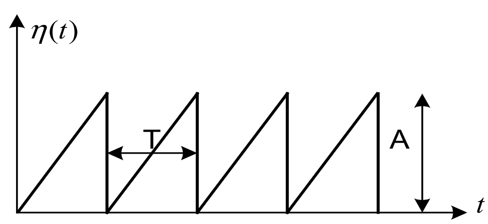

Fig. 2. Graph of dependence of η(t) of the "saw-tooth" type

In present study, features of the mechanism operation, which are manifested when considering elasticity of sections, are displayed. A surge is characterized by a loading increase over time, applied to the output section, and deformation of the driving mechanism, associated with it. In this case, the rotation velocity of the output section changes, which in turn influences the quality of the technological process. It should be noted that consideration of the mode of loading surge, based on the linear system of equations, is valid, if nonlinear properties of sections do not appear. If a mechanical system contains essentially non-linear sections, an analysis of modes of loading surge in linear statement allows the estimation of dynamic properties of a mechanical system only in the initial approximation [17]. For the loading dumping mode, non-linear properties of a mechanism manifest themselves to a larger extent, which must be considered in the design. In our case, in the study of dynamic processes, all sections of mechanical system are freed, so the dynamics of the process should be considered as linear.

To create a computer model to studying the modes of dynamic loading of tooth gear, taking into account the specificity of research, preference was given to the principles of analog (digital) modeling [18]. It is necessary to consider the following special features as an advantage of this method of modeling, compared to the program method: high performance, simplicity of the task, entire own zero lag of the solving elements, actual lack of influence of the characteristics of deciding elements of the model on the research results, a possibility to simulate typical non-linear and piecewise-linear approximations of complex non-linear dependences [18]. Using the methods of implementation of the principles of analog modeling, presented in [19, 20], a simulation model to solve a system of differential equations was developed by means of the MATLAB-Simulink environment [16], on which the modes of tooth gear loading were actually studied. The computer simulation model was based on the previously developed schematic of simulation using the methods described in [21], it is depicted in Fig. 3.

As can be seen in Fig. 3, the simulation model consists of sub-blocks, the number of which corresponds to the number of differential equations of mathematical model of the tooth gear [16]. Each sub-block implements a solution for differential equation of the system, correspondent to it, and has an internal structure that is determined by the form of the differential equation, correspondent to it.

An example of internal structure of one of the sub-blocks of the simulation model, which is based on 12 assigned input parameters of mechanical system, solving differential equation for radial oscillations of the first tooth gear wheel around axis z, is presented in Fig. 4.

As it may be seen from Fig. 4, the block diagram of the sub-block contains the set of typical operating blocks – adders, integrators, multiplication blocks, etc, the structure of their connection with one another are determined by the appropriate equation of the mathematical model [16].

The interrelations between the sub-blocks of the Simulink model are defined, in their turn, by interrelations between equations, that is, by the structure of the system of differential equations of mathematical model, and are clearly displayed in the form of graph of relationships between equations and variables in [16].

To plot the graph of loading surge on the operating body, a generating functional device was created, the block diagram of which is depicted in Fig. 5.

As can be seen from Fig. 5, the functional device consists of generators of signals of various forms (G1-G14) from the set of tools in the MatLAB-Simulink modeling environment; amplifying blocks (U1-U14) with the possibility of setting an arbitrary transmission factor; switching keys (K1-K14), providing a specified composition of generators, connected to the adder; adder (Σ), which provides obtaining the weighted sum of generators signals in accordance with the assigned coefficients of amplifiers (U1-U14).

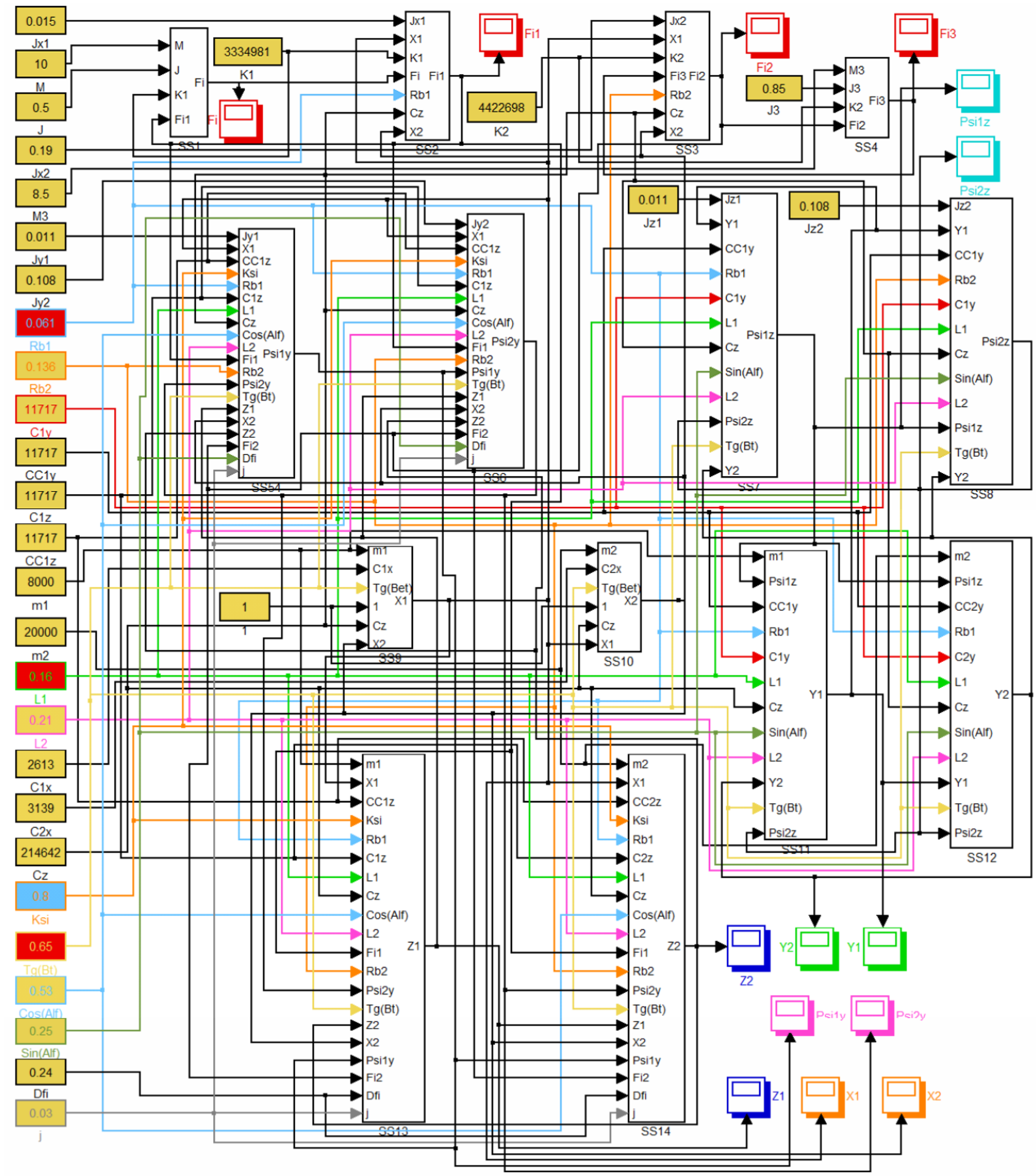

Fig. 3. Simulink-model to study the modes of dynamic loading in mechanical oscillatory systems of the class of tooth gear

Generating part of the device includes generators of signals of the following forms: G1 – sinusoid, G2 – rectangle, G3 – saw-toothed, G4 – triangle, G5 – linearly growing, G6 – saw-toothed, G7 – functional combined generator, G8 – random discrete signal, G9 – random analog signal, G10 – stepwise growing signal. The signals of the generators, connected to the adder of generators, are converted into total signal η(t), which operates at the output of adder, and then delivered to the input of modeling device as a variable magnitude M_3. The possibility of arbitrary commutation of signals and assigning arbitrary weighting coefficients of amplifiers, provides receiving total loading function η(t) of any assigned form. A view of the function generation block, created by means of MATLAB-Simulink based on methods [22, 23] and the examples of oscillograms of functions of external loading, generated with its help, are shown in Fig. 6, 7.

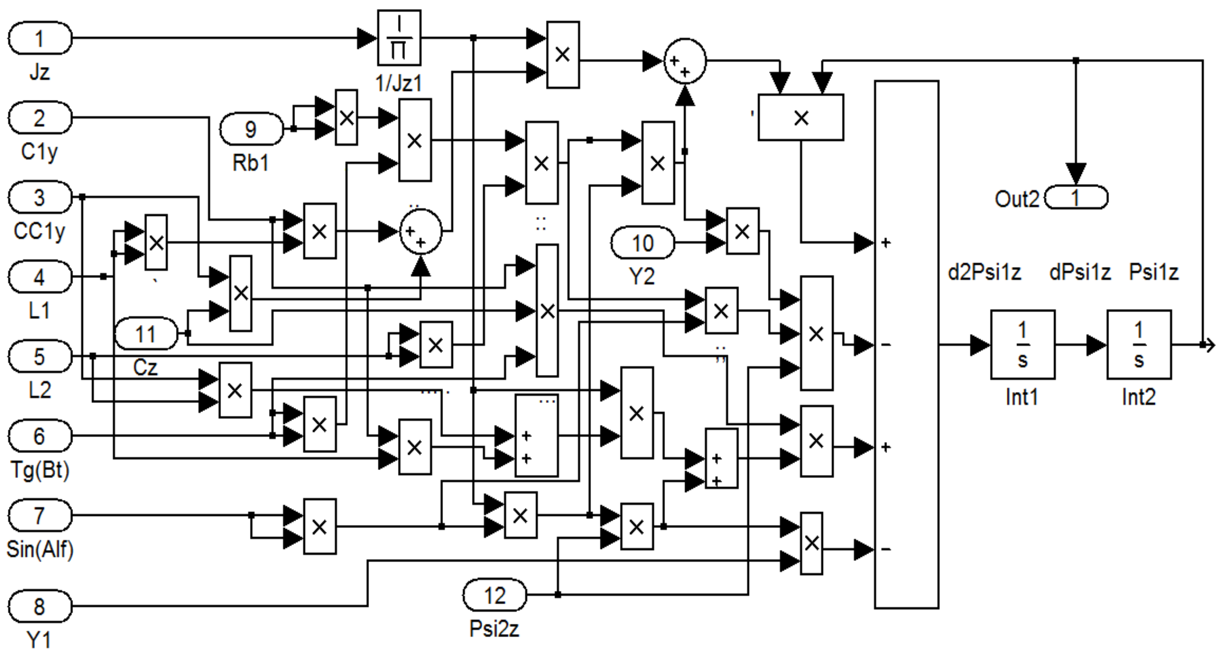

Fig. 4. Schematic of sub-block to reproduce a function of radial oscillations ψ_1^z

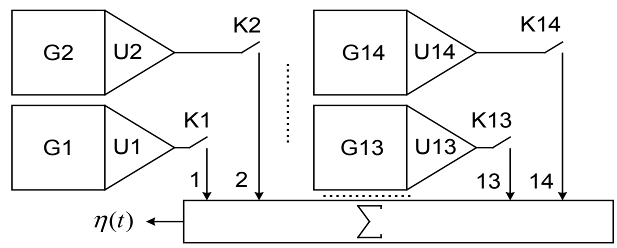

Fig. 5. Schematic of functional device for the generation of function of external load of arbitrary form

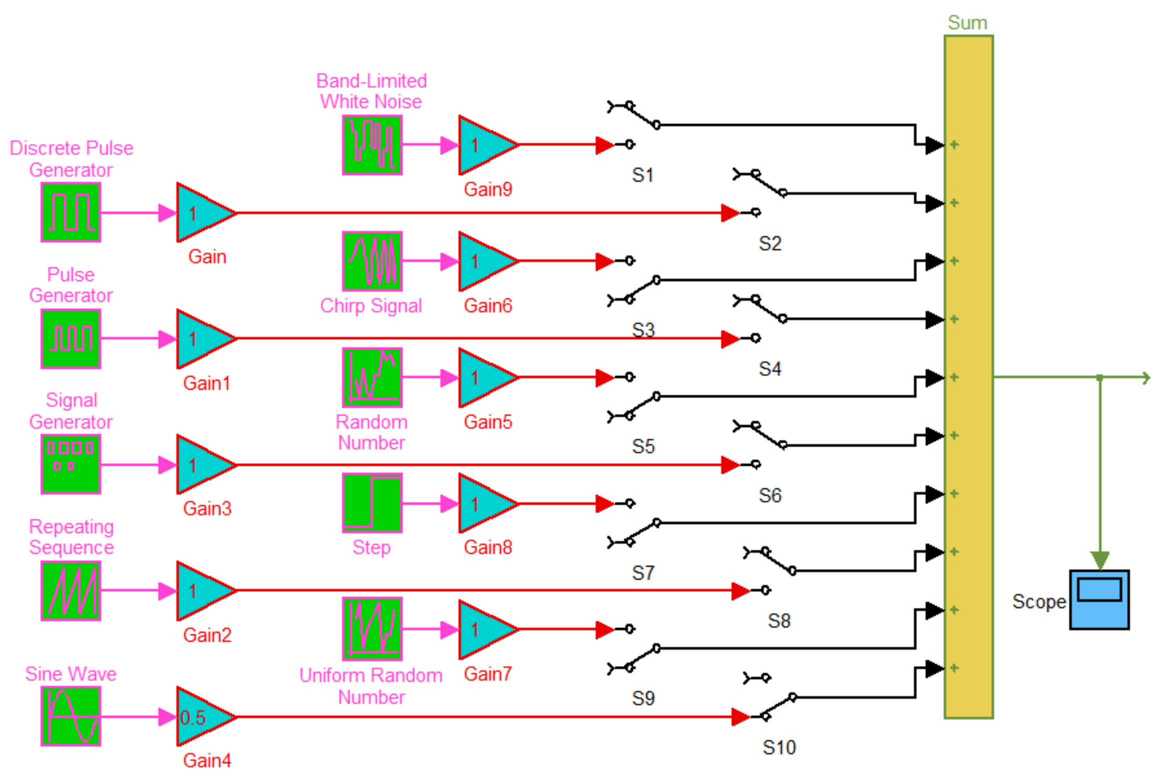

Fig. 6. Block for the generation of functions of external loading

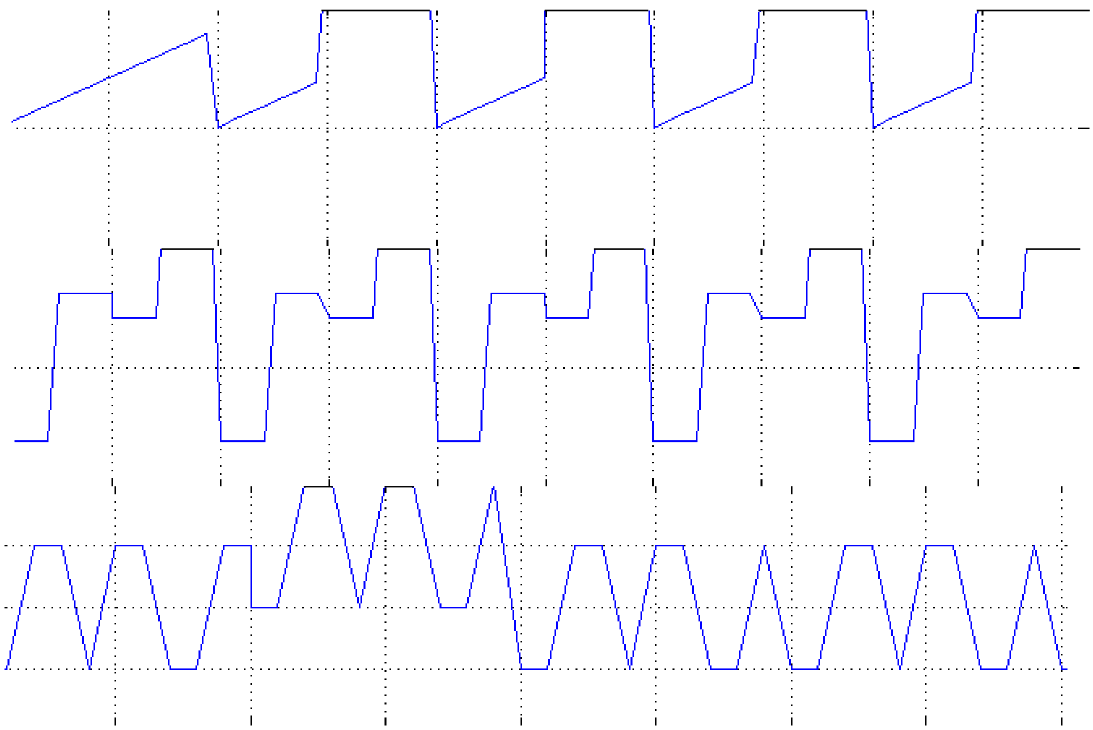

Fig. 7. Examples of oscillograms of external loading functions for the simulating modes: a — tool cutting in and coming out of metal cutting machine; b — grip of the work piece by rolls of rolling mill; c — motion of manipulator's hand

Fig. 8 presents a general simplified schematic of the model experiment on examining the modes of dynamic tooth gear loading.

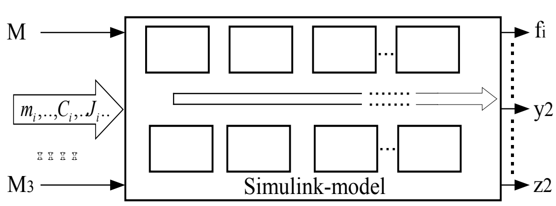

Fig. 8. General simplified schematic of the model experiment

The input magnitudes that affect the solution (Fig. 8) are: M – torque of the drive engine, M_3 – loading moment, m_i,..,C_i,..J_i.. – vector of generalized coordinates, which includes inertial–elastic, damping, and a number of other parameters of tooth gear dynamics. Loading functions, the view and parameters of which were described above (Fig. 1, 2), were used as M_3. Transverse oscillations of the output shaft in the horizontal plane – y_2 were selected as the original magnitude for observation.

In the simulating experiment, we studied single-staged cylinder helical evolvent tooth gear of the C1U-100 standard type, with transmission ratio z=2.22, nominal torque on the shaft 315 Nm, wheel width 10 mm, angle of engagement α=20°. Numerical values of basic inertial-elastic, damping, structural, geometric and dynamic parameters of the reducer are presented in Table 1.

Computer simulation experiment of dynamic processes in the examined tooth gear transmission included the following: by numeric values of the model's parameters (Table 1), marked on the simulation model (Fig. 3), the variable load (surge-dumping) that corresponds to variable M_3, is supplied to the output shaft of the mechanism. Variable magnitude M, simulating the rotation frequency of the input shaft (f), acts simultaneously. Magnitude M was reproduced by harmonic sinusoid function, the period of which (T=1/f) corresponds to the rotation period of the input shaft, and the amplitude ($A=M_{max}$) corresponds to maximum torque on the shaft of the driving motor. Changing the rotation period of the input shaft (T_{rot}) in the range from $0.5T_{load}$ to $1.25T_{load}$, where T_{load} is the period of the function that represents the alternating external loading on the output shaft, the oscillogram of fluctuations in output shaft y_2 was registered with the help of the virtual oscilloscope (Y_2) of the simulation model (Fig. 3).

Table 1

Numeric values of basic parameters for tooth gear

Parameter of tooth gear	Numeric values
Tooth module	m=2,5 mm
Numbers of tooth wheels	Z_1=49, Z_2=109
Radii of dividing circles	r_{b1}=0.0612 m; r_{b2}=0.1362 m
Masses of tooth wheels	M_1=8 kg; M_2=20.305 kg
Moments of inertia of tooth wheels relatively to axis X	J_{x1}=0.1539 kg·m^2; J_{x2}=0,1919 kg·m^2
Moments of inertia of tooth wheels gears relatively to axis Y	J_{Y1}=0.1196 kg·m^2; J_{Y2}=0.1081 kg·m^2
Moments of inertia of tooth wheels gears relatively to axis Z	J_{Z1}=0.1196 kg·m^2; J_{Z2}=0.1081 kg·m^2
Moment of loading inertia	J=0.5 kg·m^2

Oscillograms of oscillations of the output shaft, aligned with the functions of the external loading at different ratios of the rotation period of the output shaft and the loading period, for loading cases of "rectangle" and "saw-toothed" patterns are shown in Fig. 9, where A_{y2} is the amplitude of oscillations of the output shaft, A_{load} is the amplitude of external load moment.

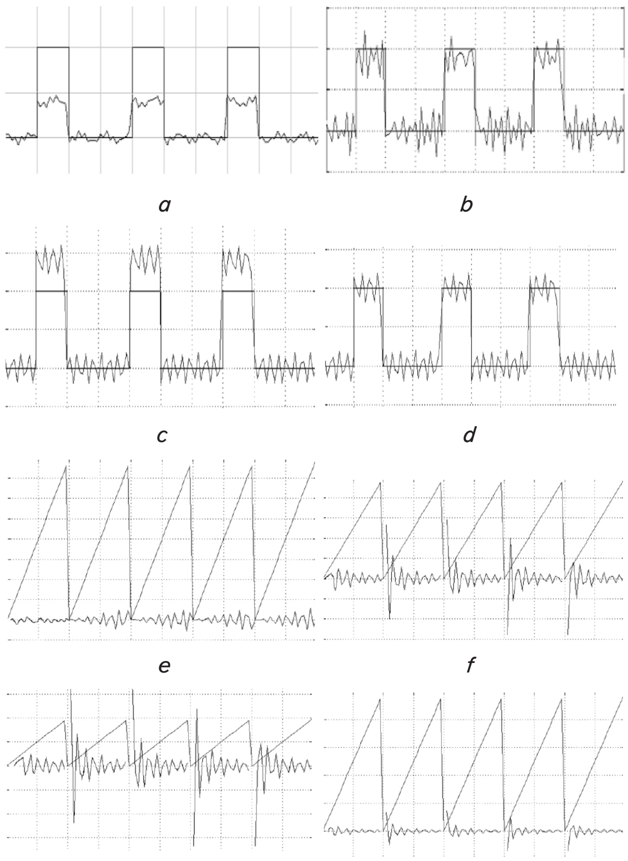

Fig. 9. Oscillograms of oscillations y_2 under condition of alternating load type: *a*–*d*–"rectangle"; *e*–*h*– "saw-tooth"

Numerical values of the ratio of amplitude of oscillations of the output shaft to the amplitude of external loading moment (A_{y2}/A_{load}), at different ratios of rotation periods of the shaft and loading action (T_{rot}/T_{load}), are represented in Table 2. Oscillograms Fig. 9, *a–d* correspond to the load type "rectangle", Fig. 9, *e–h* correspond to the "saw-tooth" type. Action period of external loading functions for both cases amounts to value T_{load}=0.3 s.

Table 2

Dependence of output shaft oscillations amplitude on the loading action period

Figure	Fig. 9, *a*	Fig. 9, *b*	Fig. 9, *c*	Fig. 9, *d*	Fig. 9, *e*	Fig. 9, *f*	Fig. 9, *g*	Fig. 9, *h*
T_{rot}/T_{load}	0,5	0,75	1	1,25	0,5	0,75	1	1,25
A_{y2}/A_{load}	0,42	0,94	1,48	1,26	0,082	0,94	2,5	0,12

Resonance characteristics of transverse oscillations of the shaft (y_2) for rectangular and saw-tooth loading patterns are depicted in Fig. 10.

As can be seen in Fig. 10, resonance characteristics are obtained by changing the value of the rotation period of the output shaft of tooth gear (T_{rot}), relative to the frequency of external loading action (T_{load}). Each graph shows the dependence of the ratio of amplitude of oscillations of the output shaft (A_{Y2}) to the amplitude of external loading moment (A_{load}) on the ratio (T_{rot}/T_{load}) in the range from 0.5 to 1.5 for rectangular and saw-tooth functions, respectively.

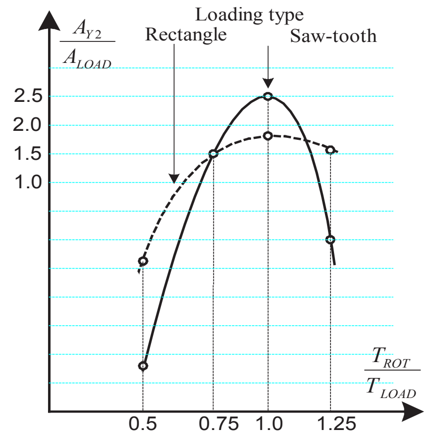

Fig. 10. Resonance characteristics of transverse oscillations of shaft (y_2) for rectangular and saw-tooth load patterns

5. Discussion of results of developing the simulation Simulink model

Created simulation Simulink model is universal and may be applied to studying the modes of dynamic loading for a single-stage tooth gear of any type. The advantage of the obtained simulation model compared with the program method of modeling is the possibility of receiving, visual observing, and saving oscillograms for any of 14 oscillations, described by the equations in mathematical model. The oscillograms, obtained during the experiment, are of great diagnostic value, because they objectively reproduce dynamic operating mode of mechanical oscillatory systems and dependence of the dynamic forces of its nodes on the form of the external load and the ratio of periods of shaft rotation and external loading. The disadvantage of the developed computer model is a relatively cumbersome complex structure and the impossibility of independent application without modeling environment. A need to change the structure when setting up for other type of problems can also be considered a shortcoming of present model.

The proposed computer model is a further development of previously created software complex DINAMIKA, intended to analyze oscillatory processes of mechanical systems, without considering external dynamics. Further improvement of the developed model is supposed to be carried out in the direction of applying the blocks to solve differential equations in partial derivatives, for dynamic analysis of distributed systems, and the use of neural network modeling.

6. Conclusions

1. A simulation computer model to study loading dynamics in mechanical oscillatory systems of the class of tooth gear was developed, which allows obtaining results of modeling dynamic operating modes in mechanical oscillatory system, according to the assigned basic weight and inertia, rigidity and damping parameters.

2. Results of simulation are displayed in the form of combined oscillograms of the action of external loading function and reactions of tooth gear elements, the oscillations of which are described by the equations of mathematical model [16]. When analyzing the obtained oscillograms, one can draw the following conclusions:

– for the "rectangle" loading pattern, the amplitude of lateral oscillations of the output shaft depends on the ratio of rotation periods and loading (T_{rot}/T_{load}), and reaches its maximum value in the resonant mode at their ratio T_{rot}/T_{load}=1 (Fig. 10). As can be seen in Fig. 9, *c*, in the resonant mode for "rectangle", the amplitude of bending moment of shaft exceeds the amplitude of moment acting in loading approximately by 1.48 times;

– at loading of the "saw-tooth" pattern, we observe the same regularity in the dependence of amplitude of transverse oscillations of the shaft on the ratio of corresponding periods, but, as it is seen from the oscillogram (Fig. 9, *e*), the value of the maximum bending moment acting on the shaft section in the resonant mode exceeds loading approximately by 2.5 times;

– the character of oscillograms for the "saw-tooth" loading pattern indicates that, in the linear section of increase in the loading moment, an increase in the amplitude of shaft oscillations is not observed. A sharp jump in the amplitude of oscillations occurs at the moment of loading dumping (Fig. 9);

– the resonance character for the "saw-tooth" loading pattern has a more rapid view compared with the "rectangle" pattern, in particular, the amplitude of dynamic loading on the shaft for "saw-tooth" loading exceeds this value for "rectangular" loading approximately by 1.4 times (Fig. 10).

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