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DYNAMIC CHARACTERISTICS OF THE TECHNOLOGICAL COMPLEX BASED ON PARALLEL MECHANISMS OF KINEMATICS

Key words

Technological complex spatial systems, drives, frequency response, pulse characteristics, natural frequencies, damping parameters, mathematical models, transfer functions.

Abstract

The technique and results of experimental studies of dynamic characteristics of spatial Drive Systems, which includes six coordinate parallel kinematic machine and system manipulation as six coordinate table. The amplitude-frequency and impulse response relative movement of the table and platform machine based on the analysis of characteristics from the mathematical model of the spatial system drive as the transfer function and defines the numerical parameters of the model.

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Introduction

Technology equipment based on spatial Drive Systems is effective and has great development prospects. Application of mechatronic control systems for this equipment allows one to create computer-integrated methods of various shapes, with methods to increase the quality of technical equipment to improve it in the direction of improving the dynamic characteristics. Therefore, the study of dynamic characteristics of spatial Drive Systems is relevant.

The problem in general is to determine the dynamic characteristics of the spatial system drives.

The problem is related to important scientific and practical tasks of technological equipment, including equipment for manipulating engineering objects.

Recent studies and publications [1, 2] show the results of the studies of process equipment based on mechatronic drive systems with parallel kinematic constraints, including the characteristics of hardness [3], and features of programming equipment [4]. Selected publications devoted to the study of dynamic precision machinery [5, 6] are available in different approaches to the dynamic characteristics of the equipment. Typically, it analyses dynamic error by testing a model rule on body movement.

Results of studies of transient and frequency characteristics of spatial drive systems were found in the literature.

Unsolved aspects of the problem are the experimental determination of the transition and the spatial frequency characteristics of actuators and identification of their parameters to construct a mathematical model of the dynamics of the system drive.

The aim of research outlined in this paper is the experimental determination of transient and frequency characteristics of the spatial system drives and their identification to describe the dynamic processes in the system.

The tasks of the research is to develop methods of experimental measurements and their conduct, building a dynamic model of spatial Drive Systems and identification of its parameters, mathematical modelling spatial system drives, and to determine its dynamic characteristics.

1. Research Methods dynamic processes

Dynamic processes in a spatial system drive observed considerable complexity. The presence of spatial low hard rod system results in low stiffness and uncertainty of partial dynamic subsystems. Therefore, to determine the dynamic characteristics of the system used experimental methods.

The spatial drive system is part of the processing facility that combines a parallel kinematic machine and the manipulating of the system, including moving table space placed drives. Research was carried out on a specially designed technology sector. It is designed for a multi-machining within the workspace of 500x500x500 mm. The number of managed coordinates is 12. Six of them are provided with actuators to move the tool. The other six manage coordinates and provide the necessary spatial position of the table. The complex allows the necessary precision machining: <0.05-0.1> mm when using open-loop control circuits 0.005 and 0.001 mm using feedback measuring of the spatial position of the executive body. The machine includes an executive body in the form of platform 1, which moves in space using 2 rods of variable length (Fig. 1).

a) b)

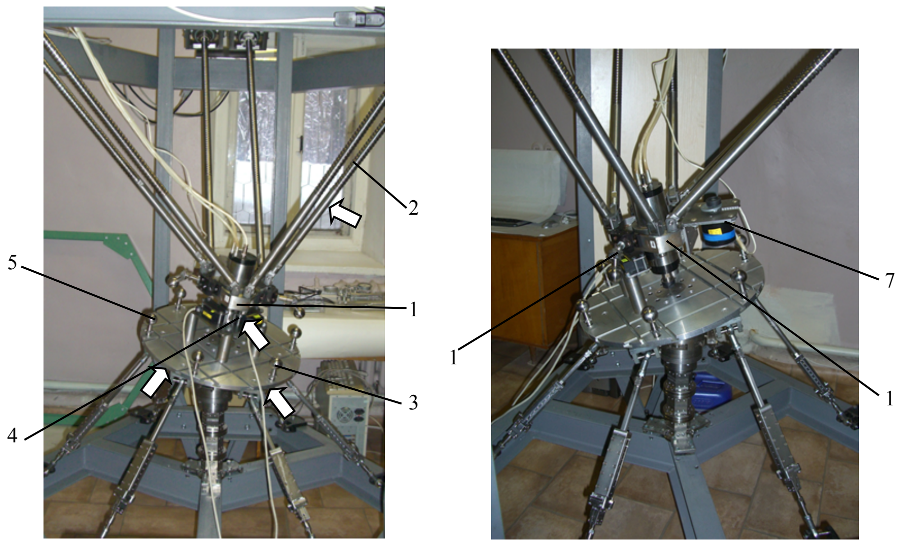

Fig. 1. Technological complex that includes a parallel kinematic machine and rolling table and – front view, b – side view

Driven by the executive body of the machine (platform), it has six degrees of freedom and provides spatial movements. The recovered detail is mounted on a rolling table (3), which can change its position in space by means of manipulation. The system is based on a circuit-hexapod mechanism and has six options. The system is equipped with additional drives for micromanipulation of movement. The manipulation system provides installation details in the desired position. Regarding this provision is the executive body of the movement, which controls the high-speed spindle.

Experimental measurements of the movement of the executive body (1) relative to table (3) for various dynamic loads of the system were made. For measuring the movement of the executive body, we used highly sensitive noncontact laser triangulation measuring distances, Series RF603-10/2. The measurement has a working range of 2 mm with precision measuring of 0.2

microns. Two measuring devices were used (4, 5), installed in two mutually perpendicular directions at a distance of 11 mm from the reference on the cylindrical surface of the executive body.

During the experimental measurements, we determined the dynamic movement of the executive body (platform) with respect to the table with sinusoidal loads. Thus, a vibratory mechanism was applied (7), which provides a variable sinusoidal load in the frequency range up to 120 Hz [7]. The vibration mechanism was set for a harmonic load with a discrete variable frequency step of 10 Hz, namely 10, 20, 30, 40, 50, 60, 70, 80, 90, 100, 110, 120 Hz. At the same time, recorded amplitude and frequency of movement of the platform was recorded relative to the table in two ways. With increasing frequency, the fluctuation of the platform initially increases and then decreases (Fig. 2).

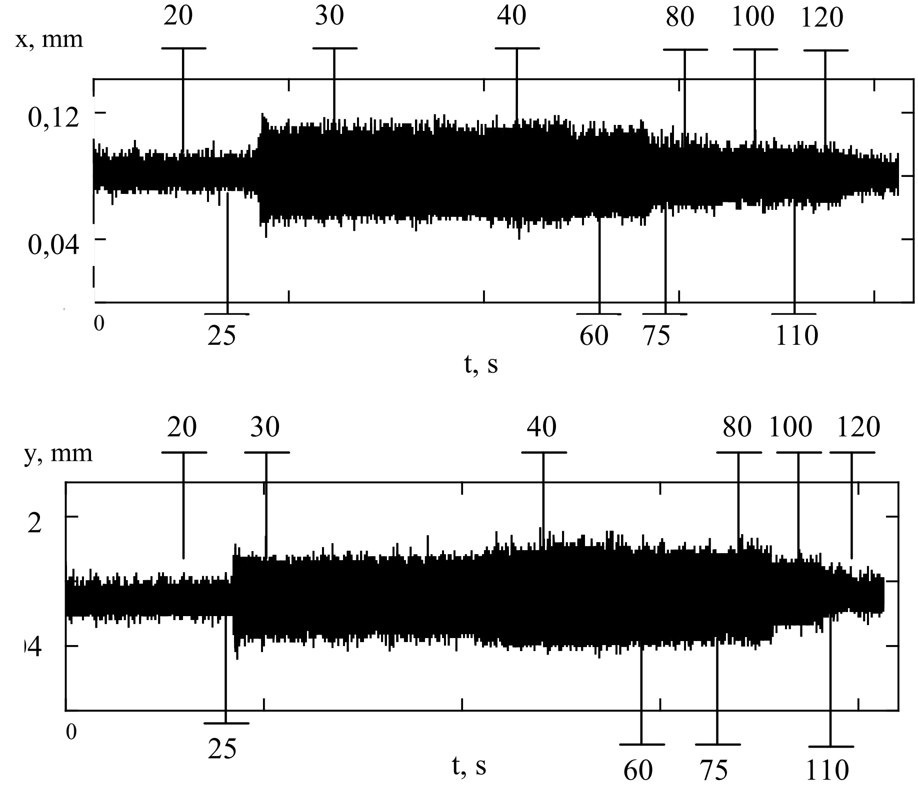

Fig. 2. Changes in the vibration platform scale depending on the frequency of the harmonic vibration force mechanism

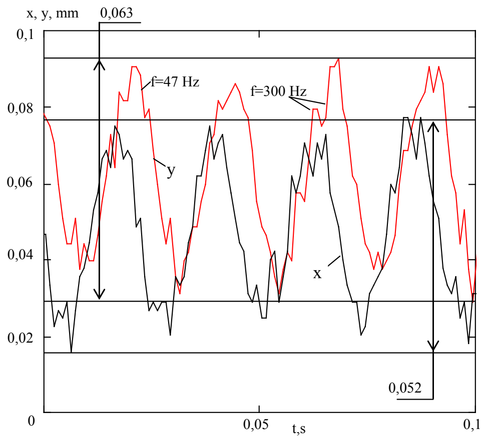

Fig. 3. Movement of the platform by the action of vibration loading at the resonant frequency (47 Hz)

The structure of the oscillation process of the platform is close to the harmonic oscillations of high frequency components. The maximum increase in amplitude occurs at a frequency of about 47 Hz. Thus, the fluctuation in the direction of the coordinate axis x and y reaches 0.052 and 0.063 mm (Fig. 3). The high frequency component has a frequency of about 300 Hz and a sweep of 0.002 to 0.01 mm. The vibrating mechanism provides a circulating load. Therefore, the trajectory of the platform is an ellipse. The value of the axes of the ellipse are 0.84 to 0.89 (Fig. 4).

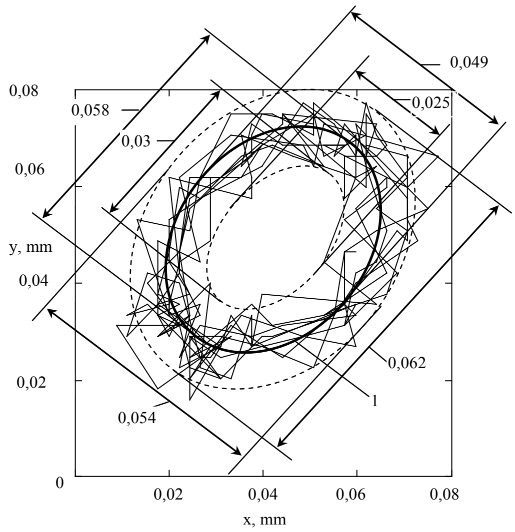

Fig. 4. The trajectory of moving pole platform with vibration load (curve 1 – the average trajectory, dashed curve - area location path)

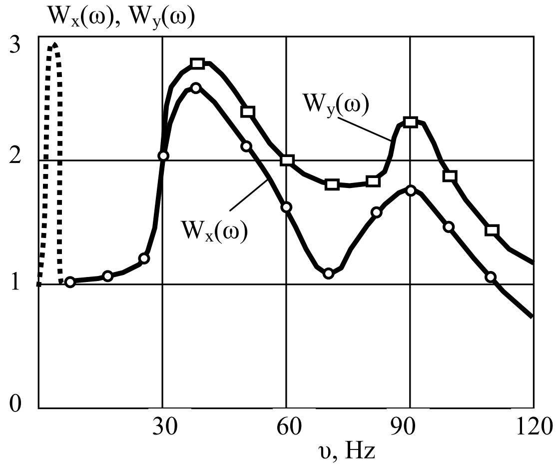

Fig. 5. Experimentally determined frequency response platform in its movements in directions x and y

Model parameters $K_1, K_2, T_1, T_2, \xi_1, \xi_2$ were determined according to the terms of the estimated amplitude-frequency characteristics of the experimental data.

The calculation of the amplitude-frequency response of the transfer function performed by a system meets the structural model and is defined as the following:

$$W_x(s) = \frac{K_1}{T_1^2 S^2 + 2\xi_1 T_1 S + 1} + \frac{K_2}{T_2^2 S^2 + 2\xi_2 T_2 S + 1} = W_1(s) + W_2(s) \qquad (1)$$

where S – Laplace operator.

Frequency response is designed according to the following relationship:

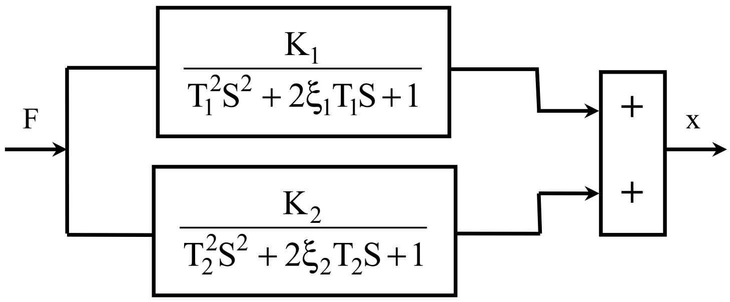

Fig. 6. Structural model to describe amplitude-frequency characteristic spatial Drive Systems

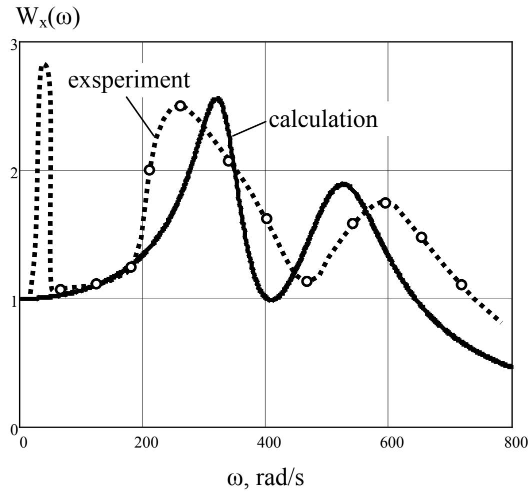

Fig. 7. Compliance with the estimated amplitude-frequency characteristics of the experimental data

Model parameters were determined according to the terms of the estimated amplitude-frequency characteristics of the experimental data.

The calculation of the amplitude-frequency response of the transfer function performed by a system meets the structural model and is defined as follows: where S – Laplace operator.

Frequency response is designed according to the following relationship:

$$W_x(\omega) = \mathrm{mod}[W_x(S), S \to j\omega],\ j = \sqrt{-1},$$

where ω - the frequency of which varies in the range 0 .. .

The calculation was carried out for the parameters:

$$K_1 = K_2 = 0{,}5,\ T_2 = 0{,}0030\ \text{s},\ \xi_2 = 0{,}08,$$
$$T_1 = 0{,}0019\,\text{s},\ \xi_1 = 0{,}14 \quad (2)$$

The calculation of the amplitude-frequency response with sufficient accuracy for practice meets the experimental data (Fig. 7). The resulting frequency response describes the dynamics of the system at frequencies above 10 Hz (62.3 rad/s). To determine the dynamic properties of the system at frequencies below 10 Hz pulse characteristics of the system used for platform vibration measurements were obtained relative to the table with impulse (shock) loads.

The experiments are qualitatively similar, but the resonance regions differ. The calculated and experimental resonance frequencies differ by 10–15%, which can be considered acceptable. The resonance in the low-frequency region is not considered in the mathematical model.

Shock to the executive body, table, and bar Drive Systems arrows are shown in Fig. 1. Measurements were performed of the mutual displacement table and platforms for shock loads on the platform. Movement was with a smooth damped oscillatory character. During the transition, oscillations of 10 to 15 take place (Fig. 8).

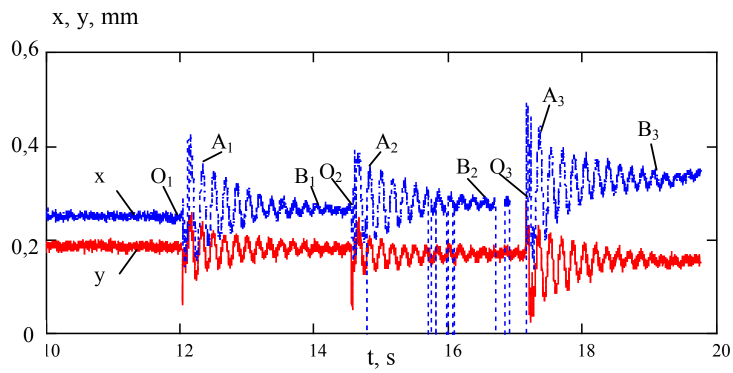

Fig. 8. Transients (pulse characteristics) in the system under impact loading of the platform

Transients are the original plot O_1A_1, O_2A_2, O_3A_3 and the main plot is A_1B_1, A_2B_2, A_3B_3. The major areas of pulse characteristics are close to those of vibrational level. Therefore, to describe the low frequency components of the transition process, we applied oscillating link with the following transfer function:

$$W_3(S)=\frac{K_3}{T_3^2S^2+2\xi_3T_3S+1} \tag{3}$$

where t_0 – A proper beginning of the transition process, which set points O_1, O_2, O_3 (see Fig. 8). The periods of oscillations in the main areas of transient periods of measurements, which is ten complete oscillations of the transition process (intervals A_1B_1, A_2B_2, A_3B_3). As a result, measurements showed that the average period of oscillation for the measured transients is within $\tau_3 = 0{,}0154..0{,}0171\,\text{s}$. This corresponds to a cyclic frequency $\nu_3 = 5{,}85..6{,}49$ Hz and is in accordance with the circular frequency $\omega_3 = 36{,}7..40{,}8$ rad/s. This item corresponds to the natural frequency of oscillations of the oscillating link with a time constant $T_3 = \frac{1}{\omega} = 0{,}024..0{,}027\,\text{s}$. Adopted mean is $T_3 = 0{,}025\,\text{s}$, hence the cyclic frequency is $\nu_3 = 6{,}3\,\text{Hz}$. In this case, the maximum values of the scattering time constant determined by measurements are 5.5%, which is satisfactory given the complexity of dynamic system. To determine the damping parameters, ξ_3 is used bended transition to the main areas A_1B_1, A_2B_2, A_3B_3. This parameter for points A and B defined by relation (4), which trigonometric function is taken as 1. However, it was found that the damping parameter is within $\xi_3 = 0{,}026..0{,}030$ and the transfer factor $K_3 = 0{,}0017..0{,}0023$. The accepted damping parameter is $\xi_3 = 0{,}028$, with a transmission coefficient of 0.002. In the initial sections (see Fig. 8), there are complex variations, including the low-frequency component corresponding to model (3) and high-frequency components. The detailed analysis of transients in the initial sections is presented in Fig. 9.

Transients in the initial areas have significant differences in their implementations. For the analysis of high-frequency components, the fixed maxima and minima of processes are in the intervals of O_1A_1, O_2A_2, O_3A_3. For the measured process, the implementation periods specified an array of high-frequency components τ_{4i} ., $i = 1{,}2..N$ $(N = 44)$.

The established range of variation periods of the highs and lows of the process is $\tau_4 = 0{,}0215..0{,}0288$ s.

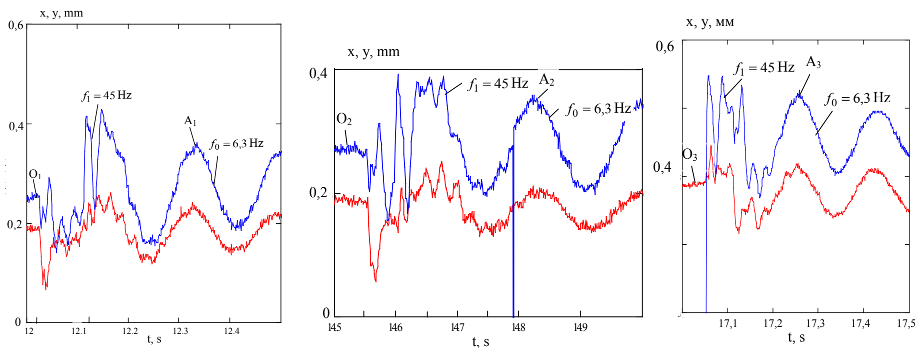

Fig. 9. The structure of the transition process in the initial area for three consecutive shocks of the system

This corresponds to a cyclic oscillation frequency $\nu_4 = 34{,}7..46{,}5$ Hz and in accordance with the circular frequency of oscillation $\omega_4 = 218..292$ rad/s. The average frequency is $\nu_4 = 45$ Hz (286 rad/s), which close to the previously defined resonant peak ($\nu = 47$ Hz) amplitude-frequency characteristics (see Fig. 7). Gain and attenuation settings for the high-frequency component are defined by the number of complete oscillations of the process, and the maximum amplitude of oscillation is defined by $K_4 = 0{,}003$, $\xi_4 = 0{,}06$.

Thus, we found that high transient components are close to the resonance frequency 47 Hz, which is available on the amplitude-frequency characteristic (see Fig. 7). Low-frequency oscillations (approximately 6.3 Hz), which are present in the transition process to the amplitude-frequency characteristic is not observed due to lack of measurement characteristics at frequencies up to 10 Hz.

The theoretical generalization of the measured impulse response of the proposed additive mathematical model is a sum of two oscillating units (Fig. 10).

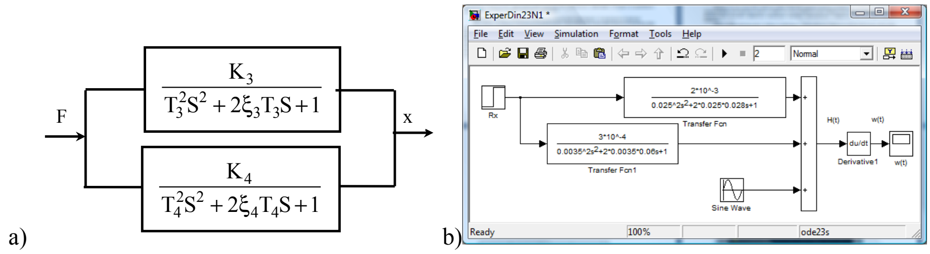

Fig. 10. Structural mathematical model for describing the impulse response (a) and the procedure for calculating the impulse response of the system (b)

The parameters of the model adopted by the result of experimental measurements of mean values are as follows:

$$K_3 = 2\cdot 10^{-3},\ T_3 = 0{,}025\ \text{s},\ \xi_3 = 0{,}028\ ,\ K_4 = 3\cdot 10^{-4},\ T_4 = 0{,}0035\ \text{s},\ \xi_4 = 0{,}06 \qquad (5)$$

The calculation of the impulse response has some difficulty because of the need to develop the impulse load on the system. Therefore, to calculate the impulse response of the procedure involves determining the system response to a unit speed function, i.e. determining the transition function $H(t)$ [7]. The differentiating transient impulse response function is presented in Fig. 10b. To take into account the noise component, which takes place during the experiments, the mathematical model introduced block SineWave was used, which generates a sinusoidal out the error of experimental measurements. The simulation results for these parameters determine the low-oscillation process in which low frequency and high frequency trace elements (Fig. 11).

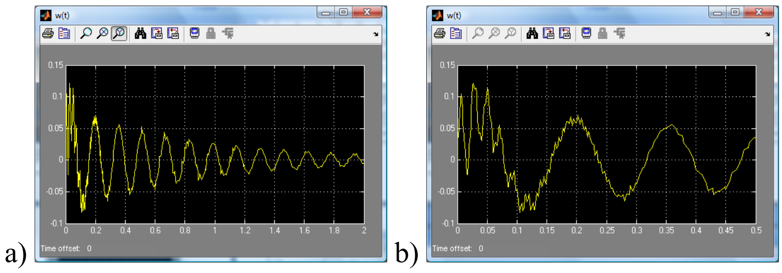

Fig. 11. Results of the calculation of impulse response on the developed model: a - general view of the characteristics defined low-frequency component, b - the structure of the process at the initial site

Comparison of calculations (Fig. 11a) with the experimental measurements (see Fig. 8) confirms compliance in calculating the low-frequency component of the experimental data.

The calculation of high-frequency component (see Fig. 11 b) meets the requirements for experimentally measured transients (see Fig. 9). The frequency of natural oscillations of about 45 Hz they correspond to the first resonance peak amplitude-frequency characteristics (see Fig. 5 and Fig. 7). Accordingly, we can conclude that the lower power models are shown in Fig. 6 and Fig. 10a, which are identical (within the error of determination of coefficients). Therefore, the combined model to describe the dynamics of the mutual displacement of the table and the platform will include three parallel-connected vibrational levels (Fig. 12).

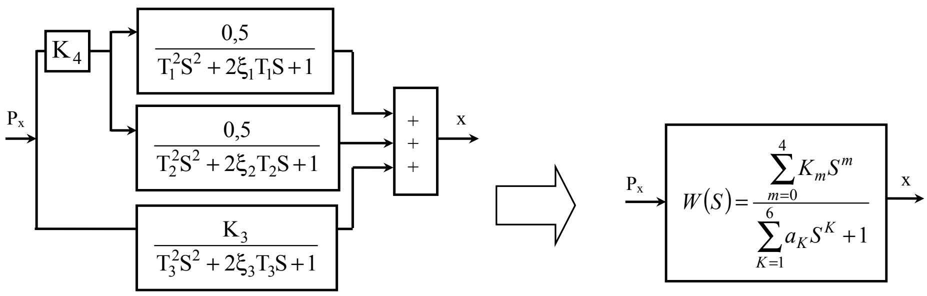

Fig. 12. Unified mathematical model for relative moving table platform is based on a comprehensive study of frequency and pulse characteristics of the system (a) and block diagram of the model (b)

This model generalizes both high-and low-frequency processes in a dynamic system when the force is in the plane of the table or platform. Parameters of the model are given above. They have been identified as a result of data processing.

Using the transformation rules, a block diagram structure model (Figure 12a) is given to one unit of the transfer function (Fig. 12b):

$$W(S)=\frac{\sum_{m=0}^{4}K_mS^m}{\sum_{K=1}^{6}a_KS^K+1} \tag{6}$$

where the coefficients of the transfer function defined by the parameters of the model (5) are shown in Fig. 12. For example:

$$a_6=T_1^2T_2^2T_3^2=2{,}03\cdot10^{-14}\ \text{s}.$$

These models generalize the experimentally determined dynamic movement of the table relative to the platform in the direction of x-axis under the influence of dynamic loading in the direction of x-axis. Given the presence of six rods that hold the table or platform, and a first approximation can be considered a small change in radial stiffness on the corner of the table. In this model, (6) describes the movement of the platform relative to the table in any radial direction.

Conclusions

1. We ound that the dynamic movement of the platform relative to the table, due to the dynamic load on the platform, with the amplitude-frequency response of the three resonant peaks at frequencies around 6.4 Hz 6 .. 45 .. 47 .. 85 Hz and 95 Hz. The resonance peak performance in the low-frequency region (6.3 Hz) has been shown in experimental measurements of amplitude-frequency response.
2. Experimentally determined dynamic circuit switching characteristics of the moving platform is with respect to the corresponding component of the short-period frequency 42 to 48 Hz and the long-period component with a frequency of 5.85 to 6.4 Hz.
3. We substantiated that the experimentally determined dynamic characteristics of the system meet the dynamic model of three parallel-connected oscillating parts with constant time $T_1 = 0{,}0019$, $T_2 = 0{,}003$, $T_3 = 0{,}025$ s and damping parameters $\xi_1 = 0{,}14$, $\xi_2 = 0{,}07$, $\xi_3 = 0{,}028$. The transfer function of this model can be presented as a ratio of a polynomial of 4th degree and a polynomial to the 6th degree.
4. As a direction for further research, it is recommended to analyse the dynamic system processing facility with the release of specific dynamic partial subsystems.

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Charakterystyki dynamiczne mechanizmów technologicznych opartych na kinematyce równoległej

Słowa kluczowe

Złożone systemy przestrzenne, napędy, pasmo przenoszenia, charakterystyki impulsowe, częstotliwości naturalne, parametry tłumienia, modele matematyczne.

Streszczenie

W artykule przedstawiono metodykę oraz wyniki badań eksperymentalnych charakterystyki dynamicznej przestrzennych systemów napędowych maszyny, na które składa się sześć współrzędnych równoległej kinematyki i manipulacji systemu. Częstotliwości i amplitudy odpowiedzi impulsowej względnego ruchu maszyny, obejmującej stolik oraz platformę ustalono na podstawie analizy cech modelu matematycznego systemowego dysku przestrzennego, który traktowano jako funkcję przenoszenia umożliwiającą zdefiniowanie parametrów numerycznych modelu.