DOI: DOI: https://doi.org/10.15407/techned2025.06.044

CONTROL SYSTEM OF A VIBRATING PLATFORM DRIVEN BY A PERMANENT MAGNET LINEAR MOTOR

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Modern industry requires high-precision control of vibrating electromechanical systems. To optimize the operation of vibrating machines, it is important to develop control systems that balance energy efficiency and productivity across different dynamic modes. In this work, a control system for a vibrating platform driven by a permanent magnet linear motor has been developed. The control object is represented by a two-mass mechanical scheme that accounts for the elastic properties of the vibration suspension and the forces of Coulomb and viscous friction. A vibrating linear motor with a toothless stator structure is considered as the exciter of a periodic electromagnetic force. The motor's electrical model is represented by an equivalent circuit with lumped parameters, whose values are functions of the mover's displacement relative to the stator. Using the developed Simulink model, the resonant properties of the electromechanical system were studied, identifying the modes corresponding to the maximum values of mechanical power and efficiency. A control system for a vibrating platform driven by a permanent magnet linear motor has been developed, which tracks the phase of displacement (or acceleration) and maintains a specified current phase. Additionally, the motor current is regulated to achieve desired parameters of mechanical vibrations. The system was simulated in the Matlab/Simulink software package, and its transient processes were investigated under changes in the mass of the vibrating platform material. References 16, figures 7, tables 2.

Keywords: control system, electromechanical system, permanent magnet linear motor, resonant properties, two-mass mechanical system, vibration platform.

Introduction. Modern industry requires high-precision control of vibration electromechanical systems, which are widely used across various industries [1–5]. To achieve the optimal operating mode of vibration machines, it is essential to develop control systems that ensure a balance between energy efficiency and performance under different dynamic modes. In addition to maintaining the required technological parameters - such as vibration amplitude and acceleration – the drive must have low inertia to provide the necessary dynamic characteristics and high controllability. This leads to the increased requirements for vibration drives and motivates the search for innovative design solutions.

Vibrating machines typically operate in modes either above or below resonance [6], however, the highest efficiency can be achieved in modes close to resonance [7, 8]. Many types of vibrating machines (vibrating mills, separators, feeders, etc.) have variable load parameters. Therefore, to set the desired operating mode, it is necessary to use a control system to continuously maintain the resonant mode.

In certain processes like vibration processing, it is cruitial to ensure the vibration field's parameters are technologically optimal [9]. Moreover, while operating the adaptive vibration technological machine, the control system tracks two parameters – the frequency and amplitude of oscillations - and adjusts these parameters of the vibration drive if there is a change in the load mass.

The use of linear drives in vibration machines enables an energy-efficient near-resonant operation mode [8, 10–12]. To achieve this, the control system must maintain a specified phase angle between the electromagnetic force and the movement of the moving mass. The optimal angle value varies depending on the type of drive, mechanical scheme, and its parameters from 45° to 90° [8, 10, 13]. This angle can be determined beforehand through preliminary analysis of the vibration system and its electromechanical characteristics, and then used as a reference for the control system.

Another solution is to implement a resonant frequency search algorithm [14]. However, this method has two major drawbacks: first, the search process takes a significant amount of time, making such a system

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ineffective under variable load conditions. The second drawback is that the search result is the frequency of maximum oscillation amplitude, which is usually not the most efficient mode for work [7].

In this study, a permanent magnet linear motor (PMLM) is considered as the exciter of the vibration platform [7, 15]. Such motors have several characteristics that make them a promising solution for drives of vibration machines. Due to design features, there is no need to use mechanical gears, since electrical energy is converted directly into reciprocating motion. This reduces losses, improves dynamic characteristics, and the use of modern powerful permanent magnets ensures compactness, high productivity and energy efficiency.

The purpose of the study is to identify effective operating modes of a vibrating platform driven by a PMLM and their implementation under variable load conditions using an automatic control system.

To achieve this, the following were developed: a mathematical model of a vibration system driven by a PMLM; a control system with a sufficiently high adjustment speed to a given mode; and a simulation model of the system. The effectiveness of the control system was verified through simulation.

Model of the control object. Vibration platforms are used to generate controlled mechanical vibrations. Their operation is based on converting the energy of an exciter (electrical, mechanical, or hydraulic) into oscillatory movements of the platform or another object.

The design of the platform for forming concrete products driven by a vibrating linear motor is shown in Fig. 1, *a*. Here, the linear motor *1* is rigidly attached to the platform *2*, which is isolated from the foundation *3* by support springs *4*.

The mechanical scheme of the vibrating platform is most often considered as a two-mass, with two degrees of freedom (Fig. 1, *b*). The motor mover *5* is connected to the stator through elastic elements *6*. The system moves under the action of a periodic electromagnetic force F_e, whose frequency and amplitude are set by the control system.

The following assumptions are made: oscillating masses are perfectly rigid bodies; the stiffness of elastic elements is constant; the mass m_p also accounts for the attached load mass (such as the concrete mixture, etc.); the viscous friction coefficient of the vibrating platform b_p considers the dissipative properties of the elastic suspension of the platform and losses in the attached load mass; initially, the system is in a state of mechanical equilibrium, when there is a static equilibrium between the gravity and elasticity forces.

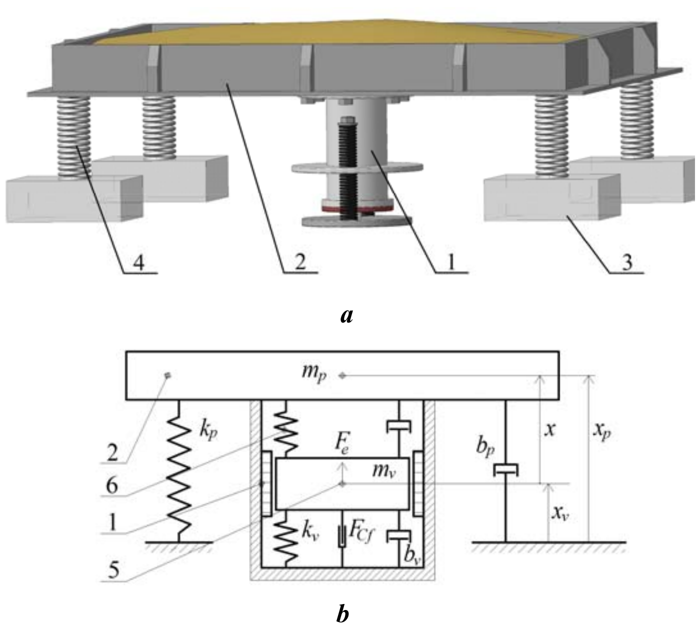

Fig. 1

If we take the position of mechanical equilibrium of the system (the position of the masses when there is no force F_e) as the reference point for the movement coordinate, the following equations of motion correspond to the given mechanical scheme:

$$\left.\begin{aligned} m_v \frac{d^2 x_v}{dt^2} + k_v x + b_v \frac{dx}{dt} + F_{Cf} \operatorname{sign} \frac{dx}{dt} = F_e; \\ m_p \frac{d^2 x_p}{dt^2} - k_v x - b_v \frac{dx}{dt} - F_{Cf} \operatorname{sign} \frac{dx}{dt} - F_e + k_p x + b_p \frac{dx_p}{dt} = 0, \end{aligned}\right\} \quad (1)$$

where m_v is the PMLM mover mass; x_v is the mover displacement relative to the fixed coordinate system; k_v is the stiffness coefficient of the vibrator springs; $x = x_v - x_p$ is the mover displacement relative to the stator; b_v is the viscous friction coefficient of the vibrator; F_{Cf} is the Coulomb friction force, F_e is the electromagnetic force; m_p is the mass of the vibrating platform with concrete mix and vibrator; x_p is the vibrating platform displacement; k_p is the stiffness coefficient of the vibrating platform springs; b_p is the viscous friction coefficient of the vibrating platform.

A vibrating linear motor with a toothless stator structure is considered as the exciter of the periodic electromagnetic force F_e. The mathematical model of a linear vibration motor, as well as the main approaches to obtaining its characteristics and parameters, are presented in [7, 15, 16].

The equivalent electrical circuit of the motor is shown in Fig. 2, where R_s and L_s are the resistance and inductance of the stator winding, respectively; e is the EMF induced in the stator winding due to the mover's movement.

In the given equivalent scheme, the stator winding resistance R_s remains constant. The winding inductance L_s is a function of the mover's movement relative to the stator. The stator EMF is equal to $e = -{d\Psi_{pm}}/{dt}$, where Ψ_{pm} is the winding flux linkage caused by the field of permanent magnets, which is a function of the mover's position.

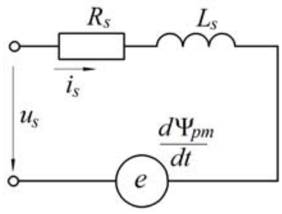

Fig. 2

Based on these considerations and the equivalent circuit, the voltage equation of the stator winding can be expressed as follows:

$$u_s = i_s R_s + \frac{d}{dt}\left(\Psi_{pm} + L_s i_s\right) = i_s R_s + \frac{d\Psi_{pm}}{dx}\cdot\frac{dx}{dt} + \frac{dL_s}{dx}\cdot\frac{dx}{dt} i_s + L_s\frac{di_s}{dt}, \qquad (2)$$

where u_s, i_s are the voltage and current of the motor winding, respectively; x is the mover displacement relative to the stator.

The motor electromagnetic force can be expressed as the derivative of the magnetic energy W_m with respect to the mover's displacement:

$$F_e = \left.\frac{\partial W_m}{\partial x}\right|_{i_s=\mathrm{const}} = \frac{d\Psi_{pm}}{dx} i_s + \frac{1}{2}\frac{dL_s}{dx} i_s^2 . \qquad (3)$$

For the toothless type of PMLM, the dependences of inductance and flux linkage on the displacement x are quite well approximated by sinusoidal functions:

$$\Psi_{pm} = \Psi_m \sin\left(\frac{\pi}{\tau}x\right); \qquad L_s = L_{av} + L_{sm}\cos\left(\frac{2\pi}{\tau}x\right), \qquad (4)$$

where Ψ_m is the amplitude of flux linkage; τ is the pole pitch; L_{av}, L_{sm} are the average and amplitude values of the stator winding inductance, respectively.

From the expressions above:

$$\frac{d\Psi_{pm}}{dx} = \frac{\Psi_m \pi}{\tau}\cos\left(\frac{\pi}{\tau}x\right); \qquad \frac{dL_s}{dx} = -\frac{2L_{pm}\pi}{\tau}\sin\left(\frac{2\pi}{\tau}\right). \qquad (5)$$

Simulink model of the system "linear motor-vibrating platform". To analyze the electromechanical properties of the system, its model was created in the Matlab/Simulink environment, as shown in Fig. 3. Here, the "Vibrator" block represents a virtual model of the "linear motor-vibrating platform" system, based on equations (1)–(5).

The model calculation was carried out with the parameters given in Table 1.

Besides the parameters listed in Table 1, it is assumed that the mass m_p ranges from 50 kg for an unloaded platform to 200 kg for a fully loaded one. It is also assumed that the viscous friction coefficient b_p depends linearly on the attached mass: 2500 kg/s for an unloaded platform to 7500 kg/s for a fully loaded platform.

The motor is powered by a controlled sinusoidal current source with a fixed rms value of 10 A, and its frequency gradually increases from 1 Hz to 50 Hz over time. The amplitude-frequency characteristics of the system for four values of the mass m_p (50, 100, 150, 200 kg) are shown in Fig. 4. It reveals that, at these parameter settings, there are two resonant frequencies where the displacement amplitudes of the mover and the platform are maximum.

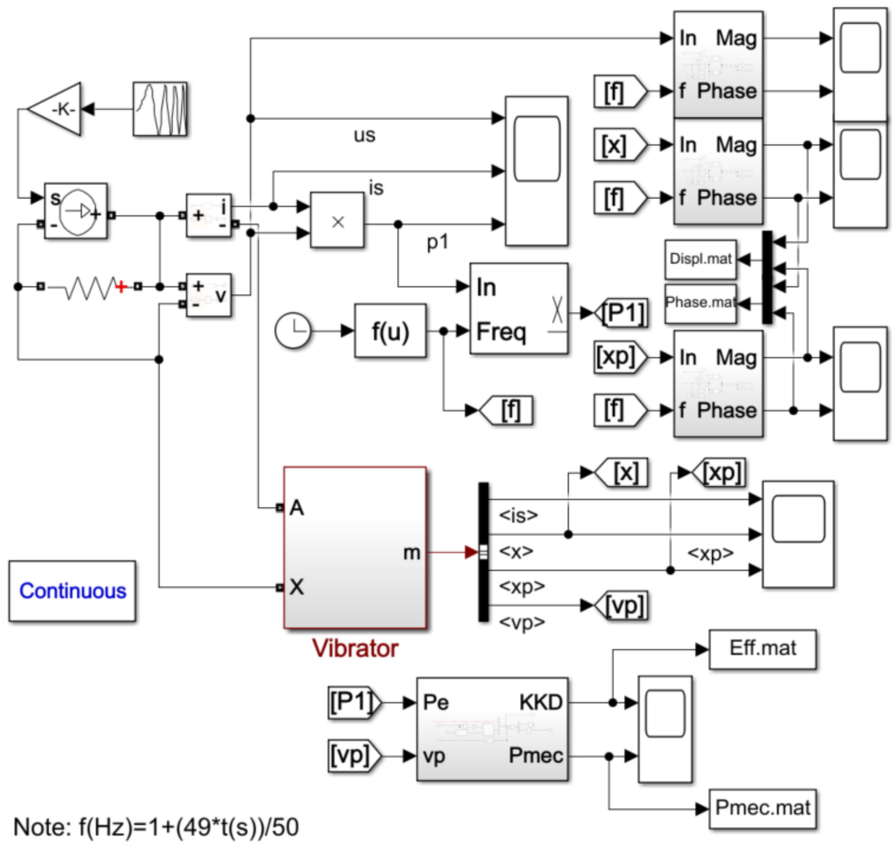

Fig. 3

Table 1

Motor winding resistance R_s, Ohm	2.67
Flux linkage amplitude Ψ_m, Vb	2.35
Pole pitch τ, m	0.071
Average value of the stator winding inductance L_{av}, H	0.07
Amplitude value of the stator winding inductance L_{sm}, H	0.0035
Mover mass m_v, kg	77
Viscous friction coefficient of the vibrator b_v, kg/s	1350
Stiffness coefficient of the vibrator springs k_v, N/m	0.687e+6
Coulomb friction force F_{Cf}, N	15
Stiffness coefficient of the vibrating platform springs k_p, N/m	1.97e+6

The dependencies of efficiency and mechanical power, shown in Fig. 4, are based on the following.

According to the adopted calculation model, the useful power supplied to the load is used to cover losses due to the equivalent viscous friction of the vibrating platform with the mixture and is equal to

$$p_l = \frac{b_p}{T} \int_{t-T}^{t} v_p^2 dt \,, \tag{6}$$

where v_p is the platform velocity.

Then, the efficiency of the "linear motor-vibrating platform" system is determined by the expression:

$$\sum \eta = p_l \Bigg/ \left(T^{-1} \int_{t-T}^{t} u_s i_s dt \right). \tag{7}$$

The electromechanical system parameters at the points of maximum mechanical power are listed in Table 2. The phase angles are calculated relative to the current, which is assumed to have an initial phase of zero. For each mass in the table, there are two mechanical power maxima, because a two-mass system with two resonant frequencies is considered.

As follows from the obtained data, the maximum mechanical power does not align with the highest amplitudes of the mover and platform oscillations. At the same time, the maximum efficiency values (see Fig. 4) correspond to the points of maximum mechanical power, which indicates the possibility of using these modes as reference for the control system.

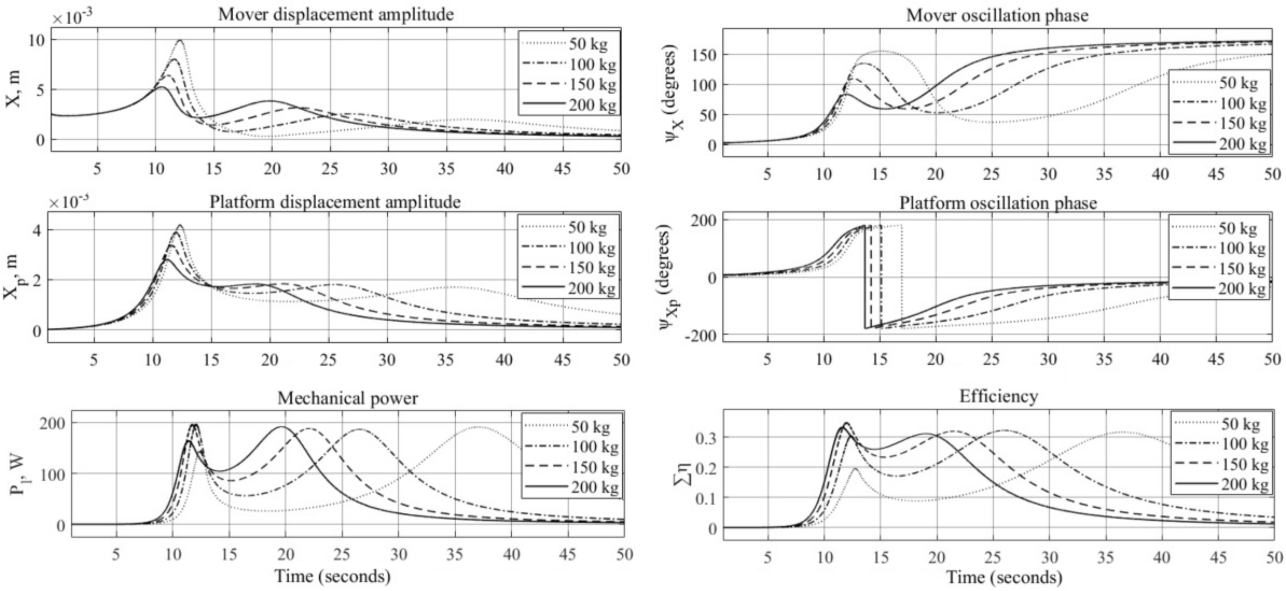

Fig. 4

Table 2

Mass of the vibrating platform m_p (kg)	50		100		150		200	
Frequency f (Hz)	13.12	37.31	12.83	26.99	12.48	22.63	12.16	20.23
Useful mechanical power P_l (W)	143.2	191.3	196	186.7	195.8	188.3	163.7	191.7
Platform displacement amplitude X_{pm} (m)	0.0042	0.0017	0.0039	0.0018	0.0034	0.0018	0.0028	0.0018
Platform oscillation phase ψ_{Xp} (degrees)	115.7	-95.5	122.7	-100.1	126.3	-104.7	131.4	-109.2
Mover displacement amplitude X_m (m)	0.0095	0.0020	0.0073	0.0026	0.0055	0.0032	0.0043	0.0038
Mover oscillation phase ψ_X (degrees)	106	91.2	102.2	92.8	90.7	92.9	76.4	91.7
System efficiency	0.178	0.316	0.288	0.321	0.337	0.317	0.328	0.309

Vibration platform operating frequency control system. The principle of the control system's operation is based on tracking the phase of displacement (or acceleration) and maintaining a specified phase of the current. To simplify the system structure, a moving coordinate system (x,y) is used, which moves synchronously with the mover. In this case, there are two control channels. The first, on the x-axis, corresponds to the phase of the current, and the second, on the y-axis, relates to its amplitude. Current regulation allows to maintain permissible electrical loads of the motor when changing the mechanical load parameters, as well as to implement the required mode of mechanical oscillations. The model of the control system in the Matlab/Simulink environment is shown in Fig. 5.

Here, the input signal from the acceleration sensor is fed to the input of the control system In(a). After converting into a moving coordinate system, the projection of acceleration along the x-axis is kept at zero using the frequency controller (the "Frequency controller" block), while the acceleration amplitude along the y-axis is regulated.

The current value from the acceleration controller output is converted by the "*xy* to *ab*" block into a fixed coordinate system and is used as a reference for the hysteresis current controller (the "Current Controller" block). The latter compares the reference value is* with the actual motor current, and if the difference is greater than the set hysteresis zone, it generates a signal to switch the power switches of the bridge inverter.

The "Step" and "Step1" blocks introduce a delay in switching on the frequency and acceleration controllers at the start of the vibrating platform. At the same time, the start occurs with the rated motor current and its fixed initial frequency.

Considering the nonlinearity of the control object, the parameters of the PID controllers were determined using the Response Optimizer/Simulink Design Optimization program. It should be taken into account that the response speed of the acceleration control system must exceed the response speed of the frequency control system; otherwise, the approach to resonance could cause significant acceleration fluctuations [8].

To investigate transient processes during the vibration drive operation, we will simulate the proposed control system under conditions of changing the processed material mass. As shown in Table 2, the change in the platform mass is accompanied by a change in the phase of the platform and mover displacements. At the same time, the phase angle of the mover's displacement at the second (high-frequency) resonance changes slightly (in the range from 91.2° to 92.9°). Therefore, we will use the mover's acceleration as the input signal, maintaining the average value of the phase angle between the current and the mover's displacement at 92.1°.

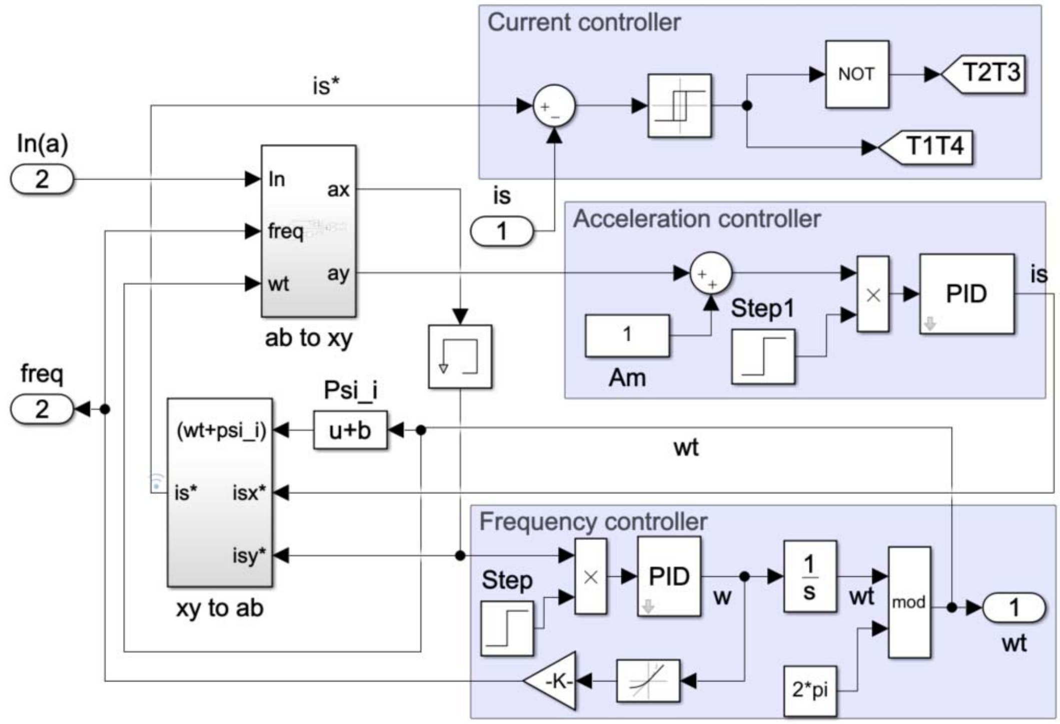

Fig. 5

The simulation results are shown in Fig. 6. Here, the vibrator operating frequency is changed by the controller when the mass of the material and platform changes from 160 kg at startup to 110 kg at the end of the 30 s calculation period. Simultaneously, the phase angle between the motor current and the mover displacement ψ_X is maintained by the controller at 92.1°. The oscillation frequency is fixed at 20 Hz at the startup After the transition period, the controller is activated, causing the frequency to increase to 21.9 Hz. As the material mass changes, the operating frequency adjusts accordingly, reaching 25.7 Hz at the end of the calculation period.

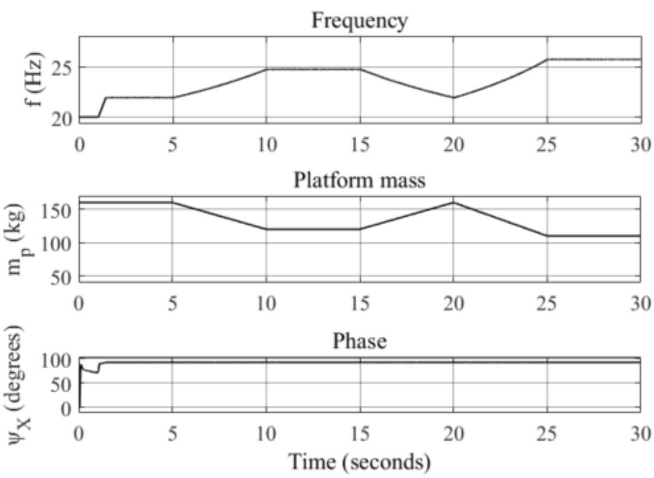

Fig. 6

The calculation results of instantaneous electromechanical processes are shown in Fig. 7.

The mover acceleration control system maintains it constant at 80 m/s^2. However, the current RMS value, which is fixed at the moment of startup (12.6 A), changes with the load variation and is 11.5 A at the end of the calculation period.

Conclusions. The work develops a control system for a vibrating platform driven by a permanent magnet linear motor. The system is simulated in the Matlab/Simulink software package, and its transient processes are investigated.

The frequency conditions corresponding to the points of maximum mechanical power were determined, and the control system operation during changes in the mass of the vibrating platform material was also examined.

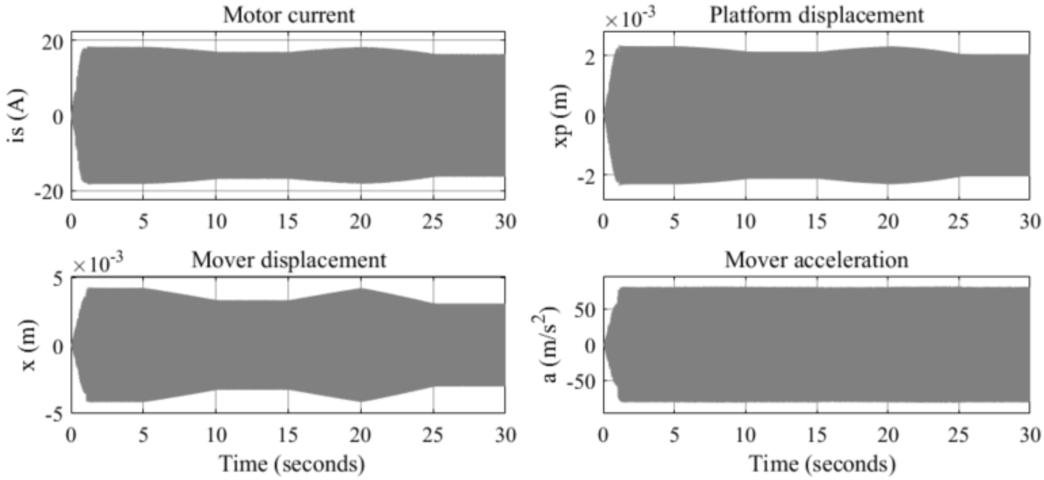

Fig. 7

The obtained results demonstrate that the proposed system for controlling the operating frequency of the linear electric drive of the vibration platform allows automatic adjustment of the system's operation mode as load parameters change. This ensures more efficient operation of the drive under conditions when the mechanical power of the electromechanical system reaches its maximum.

The drawback of the developed system is the relatively complex control algorithm, which requires the use of real-time processors. In addition, because the phase response of displacement (or acceleration) is nonlinear, the system is sensitive to the initial frequency at startup. Consequently, further research will focus on optimizing the control system and its practical implementation, including construction of a prototype.

The work was carried out within the framework of the budget program on the topic "Computer modeling and research of highly efficient electromechanical systems", state registration number 0124U000146.

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УДК 621.313.323

СИСТЕМА КЕРУВАННЯ ВІБРАЦІЙНОЇ ПЛОЩАДКИ З ПРИВОДОМ ВІД ЛІНІЙНОГО ДВИГУНА З ПОСТІЙНИМИ МАГНІТАМИ

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Сучасна промисловість вимагає високоточного керування вібраційними електромеханічними системами. Задля досягнення оптимального режиму роботи вібраційних машин важливо розробляти системи керування, що дають змогу забезпечити баланс між енергоефективністю й продуктивністю в різних динамічних режимах. У даній роботі розроблено систему керування вібраційної площадки з приводом від лінійного двигуна з постійними магнітами. Об'єкт керування представлений двомасовою механічною схемою, що враховує пружні властивості вібраційної підвіски та сили сухого і в'язкого тертя. У ролі збудника періодичної електромагнітної сили розглядається вібраційний лінійний двигун із беззубцевою структурою статора. Електричну модель двигуна подано схемою заміщення із зосередженими параметрами, значення яких є функціями переміщення бігуна відносно статора. За допомогою розробленої Simulink-моделі проведено дослідження резонансних властивостей електромеханічної системи та визначено режими, що відповідають максимальним значенням механічної потужності й ККД. Розроблено систему керування вібраційної площадки з приводом від лінійного двигуна з постійними магнітами, що відслідковує фазу переміщення (або прискорення) та підтримує задану фазу струму. Водночас струм двигуна регулюється задля досягнення заданих параметрів механічних коливань. Проведено моделювання такої системи в програмному пакеті Matlab/Simulink та досліджено її перехідні процеси під час зміни маси матеріалу віброплощадки. Бібл. 16, рис. 7, табл. 2.

Ключові слова: вібраційна площадка, двомасова механічна система, електромеханічна система, лінійний двигун із постійними магнітами, резонансні властивості, система керування.

Received 11.06.2025
Accepted 24.07.2025