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MODELING OF THE ELECTRIC DRIVE OF THE MAIN MOTION OF THE ROLLING CAGE AS A MULTI-MASS ELECTROMECHANICAL SYSTEM

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МОДЕЛЮВАННЯ ЕЛЕКТРОПРИВОДУ ГОЛОВНОГО РУХУ ПРОКАТНОЇ КЛІТІ ЯК БАГАТОМАСОВОЇ ЕЛЕКТРОМЕХАНІЧНОЇ СИСТЕМИ

Шевченко І.С., Руднєв Є.С., Романченко Ю.А.

The model of rolling stand of a thick-plate rolling mill 3000 has been designed in the work in order to determine the oscillations frequencies that occur during operation, the effect of their amplitude on the dynamic deviations of the speed of the working body from the specified one. Presented results of the research dynamic to rolling cage 3000 at presentation her as seven masses electromechanic system. Shown influence clearance in mechanical issue on dynamic of the mechanism. The research by the method of mathematical modeling in the design and operation of mechanical equipment is substantiated. The design diagram of mechanical part of the electromechanical system is presented. Using the simulation results it was confirmed that the influence of internal viscous friction in shafting on the oscillation damping is not significant in relation to the damping properties of electric drive. Therefore, in the first approximation, it can be ignored. The electric drive of the rolling stand was considered as a TP-D system with speed and current regulators at their standard settings to the modular optimum. To reduce the magnitude of the elastic moments in the kinematic chain of the stand the armature current intensity generator in the electric drive is used. Simulation of the processes was carried out in Simulink of the MATLAB package. The stand model is designed according to design scheme and reflects the branching into two channels with their combination through an elastic element – the material that is rolled. Based on real geometry and taking into account the properties of material the stiffness of shafts of mechanical transmissions were calculated. The model was set to a rolling program with a variable speed – in order to compensate the thickness difference, which corresponds to modern technological trends. Comparing the simulation results, it was found that the presence of a gap provokes the appearance of self-oscillations, the damping of which in a real mechanical system will occur due to damping properties of the shafts. The damping of oscillations takes place due to the damping properties of the electric drive. The appearance of a gap in the spindles leads to an increase in the system vibration frequency (70-80 Hz).

Keywords: *rolling cage, springy viscous element, mechanical fluctuations.*

Introduction. The design and operation of mechanical equipment requires finding the real maximum values of the forces (moments) arising in the kinematic chain, their dependence on various factors (acceleration-deceleration, elements stiffness, clearance values, etc.). Almost the only method for researching such processes is mathematical modeling.

Objective. The work is devoted to the design of a rolling stand model of 3000 thick-plate rolling mill in order to determine the oscillation frequencies that arising during operation, the effect of their amplitude on the dynamic deviations of speed of working body from the specified speed.

Research results. The equipment of finishing mill stand includes (Fig. 1): DC electric motor 1; protective gear clutch 2, gear stand 3; hinges 4 from the side of the gear stand; spindle shafts 5, spindle joints 6 from the stand side; couplings of 7 spindle blades; support 8 and work 9 rolls. Between the working rolls there is a strip 10, which is rolled.

The design diagram of the mechanical part of the electromechanical system is shown in Fig. 2. Mechanical inertias of single parts, which are expressed by the moments of inertia J_j (j = I, II, III, IV, V, VI) are shown by rectangles, the areas of which are proportional to these inertias. The kinematic connection between inertias are shown by elastic-dissipative elements in the form of a parallel connection of t stiffness (C_{ke}) and scattered (dissipation δ_{ke}) coefficients. The first one are determined by the ratio of elastic moments in such elements to their deformation, and the latter's are determined by the level of losses in them in the presence of a deformation rate.

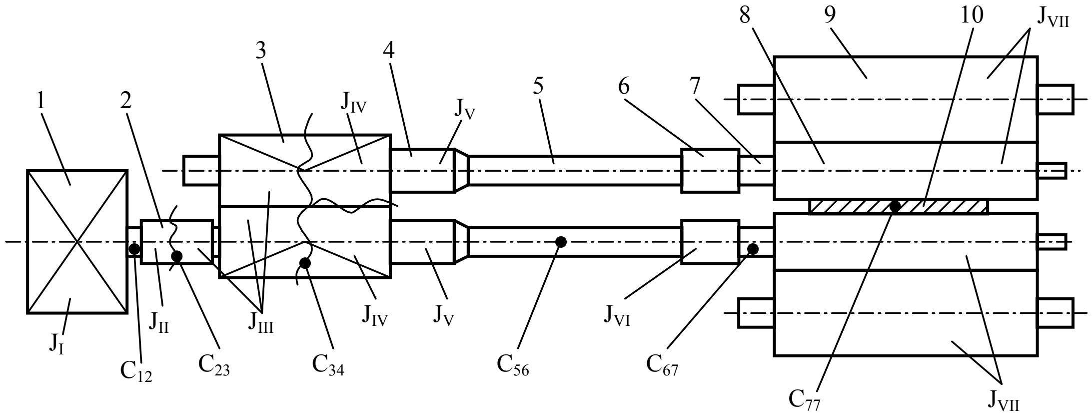


Fig. 1. Main line of finishing stand

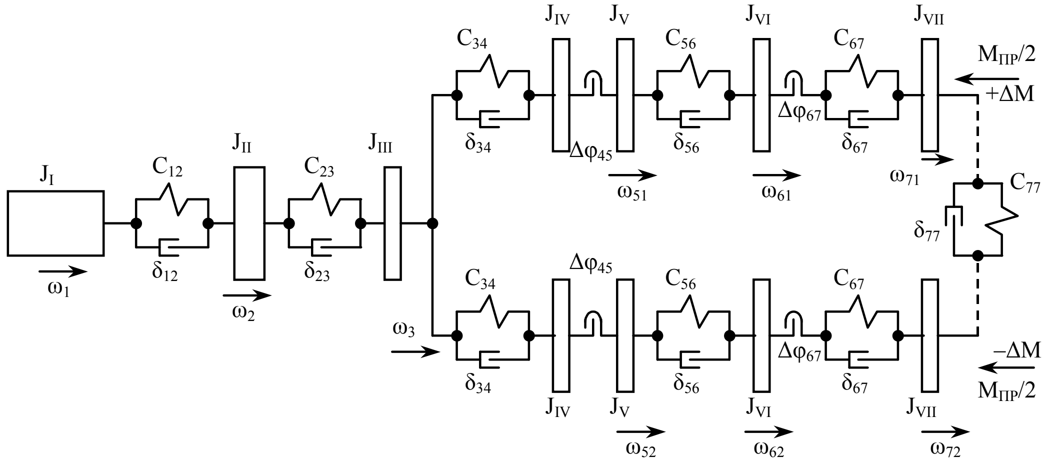


Fig. 2. Design diagram of mechanical part of the stand

The scattering coefficients are determined by the logarithmic damping decrement $\lambda_{\text{вт}}$

$$\delta_{\text{ке}} = 1 - e^{-2\lambda_{\text{вт}}} . \quad (1)$$

The size of the latter for metal (steel) elements is much smaller $\lambda_{\text{вт}} < 0{,}1$

$$\lambda_{\text{вт}} = \ln(A_1 / A_2) , \quad (2)$$

A_1, A_2 – are two consecutive values of oscillation amplitude in shafting, separated by one period,

$$T_{\text{ке}} = \Omega_{\text{ке}} / 2\pi , \quad (3)$$

$\Omega_{\text{ке}}$ – own circular frequency of oscillations, 1/s.

When structural schemes are used to analyze the dynamics of a system, then the viscous friction coefficient in shafting $\beta_{\text{ке}}$ is introduced in them:

$$\beta_{\text{ке}} = \frac{2\lambda_{\text{вт}} C_{\text{ке}}}{\Omega_{\text{ке}} \sqrt{4\pi^2 + \lambda_{\text{вт}}^2}} . \quad (4)$$

Since the elements of the kinematic chain (shafts, spindles) are distributed masses, then the approaches proposed in [1] were used to bring the latter (J_j) to concentrated ones.

The influence of internal viscous friction in shafting on the oscillation damping is not significant in relation to the damping properties of the electric drive itself. Therefore, in the first approximation, it can be ignored, which was also confirmed by the simulation results.

On the design diagram $\Delta\varphi_{з}$ makes it possible to take into account possible clearances in the coupling of spindle blades, as well as in the hinges on the side of gear stand. The stiffness coefficient C_{77} of the rolled strip takes into account the interaction influence of rolls with each other in case of an uneven distribution of the rolling moment $M_{пр}$ between them [2].

The electric drive of the rolling stand was considered as a TP-D system with speed and current regulators at their standard settings to the modular optimum. To reduce the magnitude of the elastic moments in the kinematic chain of the stand the armature current intensity generator in the electric drive is used.

Simulation of the processes was carried out in Simulink of the MATLAB package. The stand model that taking into account the features of electric drive and elastic properties of the kinematic links is shown in Fig. 3. The model is designed according to design scheme and reflects the branching into two channels with their combination through an elastic element – the material that is rolled. The actual drive system is folded into the Subsystem EP subsystem. Creating of a gap in the kinematic connection of the hinge-spindle ($\Delta\varphi_{67}$) is implemented by blocks of insensitive Dead Zone1, Dead Zone2. In modeling, the internal dissipation of elastic elements was not taken into account – the coefficients β_{ij} were taken to be zero.

Following values of iinertia J1=67500 kgm^2; J2=8700 kgm^2; J3=950 kgm^2; J45=700 kgm^2; J6=450 kgm^2; J7=4075 kgm^2.

Based on real geometry and taking into account the properties of material the stiffness of shafts of mechanical transmissions were calculated.

The oscillation frequencies of elastic elements with such parameters are:

$$f_{ij} = \frac{1}{2\pi}\sqrt{\frac{C_{ij}(J_i + J_j)}{J_i J_j}}$$

$$f_{12} = 54{,}17\ \text{Гц};\ f_{23} = 132{,}54\ \text{Гц};\ f_{3(45)} = 392{,}4\ \text{Гц};$$
$$f_{(45)6} = 117{,}77\ \text{Гц};\ f_{67} = 250\ \text{Гц}.$$

The simulation results are shown in Fig. 4. It shows the graphs of coordinates in transient processes: set speed (ω_z); mismatch between ω_z and speed of the drive end of the roll $\Delta\omega = \omega_z - \omega_7$; the electromagnetic torque of the engine (M_e) and the total torque (M_c) of the load during rolling; moment in one of the spindles (M_{56}).

The model was set to a rolling program with a variable speed – in order to compensate the thickness difference, which corresponds to modern technological trends.

Taking into account the gap, its value was set to $\Delta\varphi_{67} = 0{,}004$ radians. Comparing the results in Fig. 4,a and Fig. 4,b, we see that, without taking into account the gaps, the damping of oscillations occurs due to damping properties of the engine. The presence of the gap provokes the appearance of self-oscillations, the damping of which in a real mechanical system will occur due to the damping properties of the shafts.

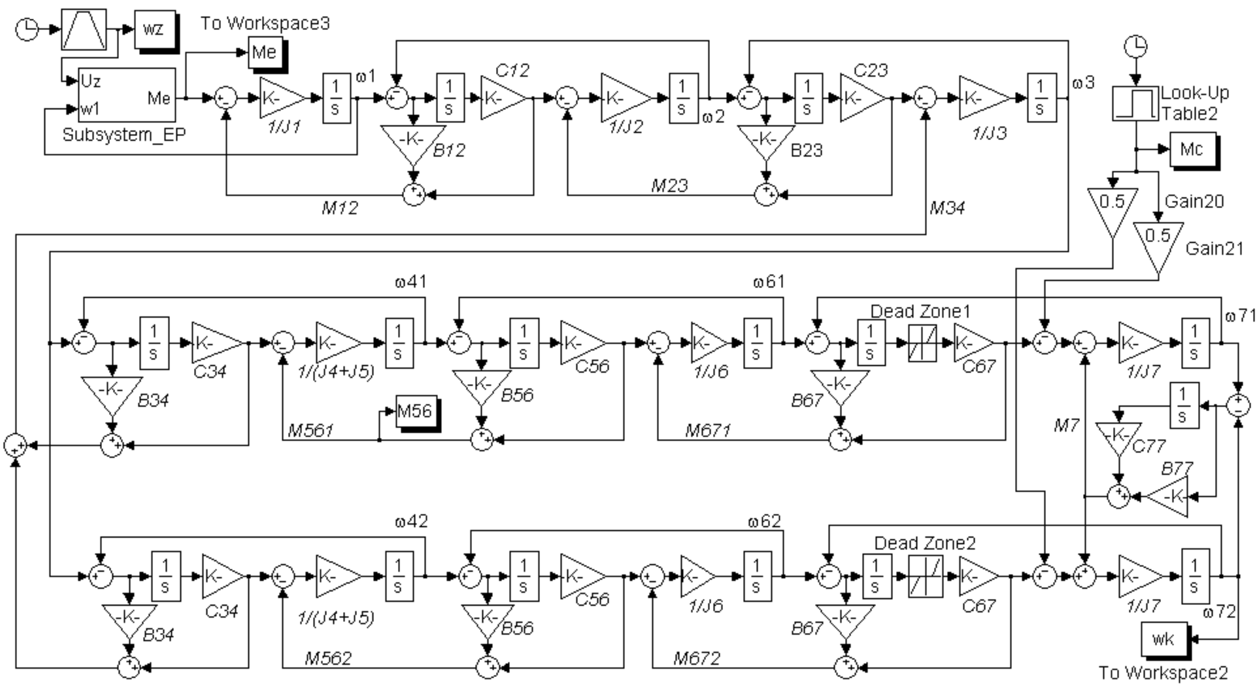
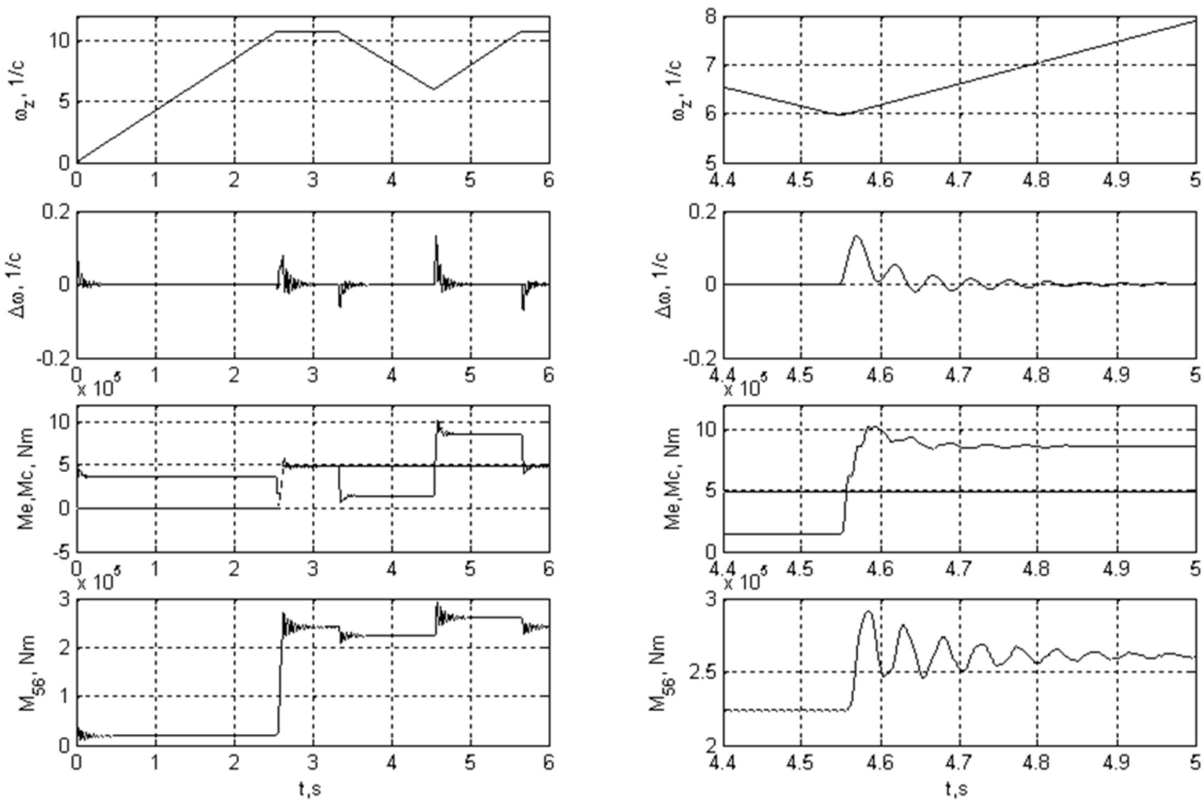
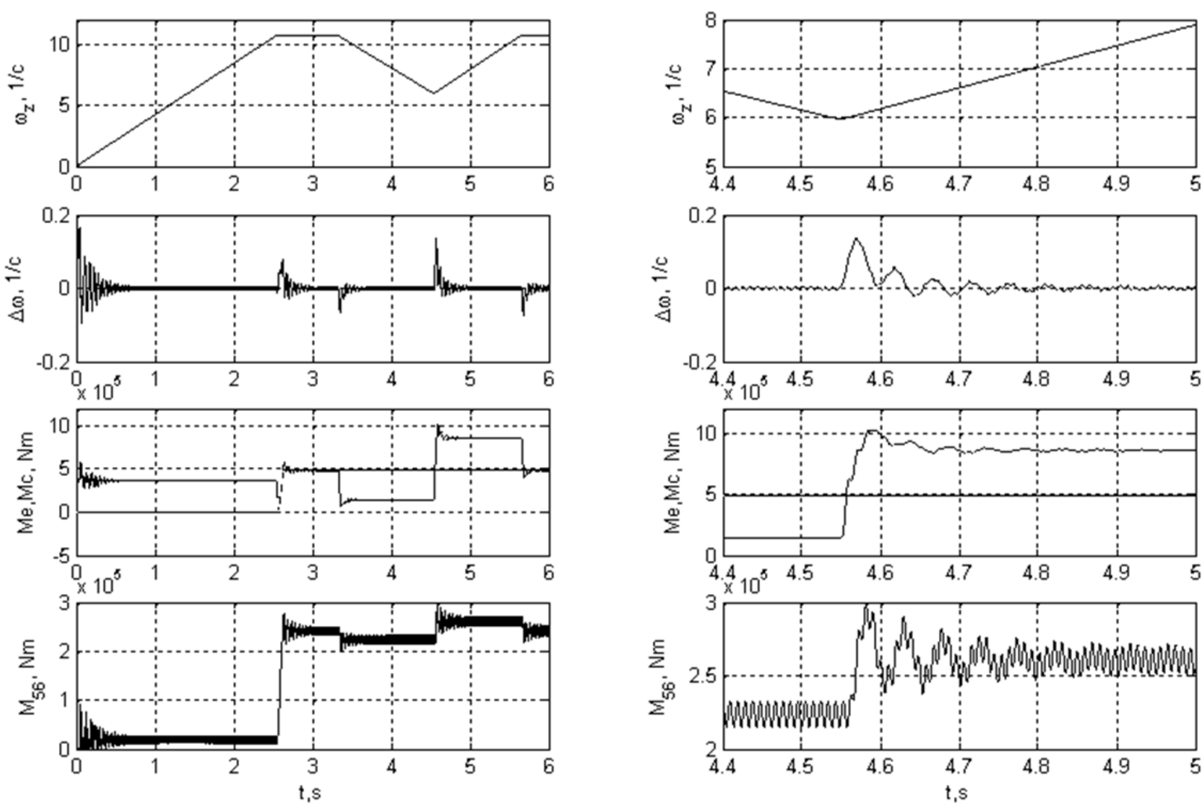


Fig. 3. The structure of model in Simulink



a



b

Fig. 4. Simulation results:
a – without taking into account the gap; b – taking into account the gap

Conclusions. Based on the simulation results, it can be stated:

– the maximum value of the dynamic misalignment between the set and real speed of the rolls does not exceed 2% and it disappears at a strip length less than 1m;

– oscillation processes in the spindle lead to an increase of moment in them $M_{56} = (1,4 \div 1,6)M_H$. This is due to the imposition of the reflected elastic deformation wave on the main one, it is short-term and does not require an increase of the engine power, but only affects the durability of mechanical equipment of the stand. To reduce the influence of these phenomena, it is necessary to reduce the gaps in kinematic chain of drive, for example, by replacing Hooke's hinges with roller ones;

– the frequency of oscillations in the mechanical chain reaches 10 Hz, which corresponds to the data of source [3]. The damping of oscillations takes place due to the damping properties of the electric drive;

– the appearance of a gap in the spindles leads to an increase in the system vibration frequency (70-80 Hz).

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Шевченко І.С., Руднєв Є.С., Романченко Ю.А. Моделювання електроприводу головного руху прокатної кліті як багатомасової електромеханічної системи

У статті розроблено модель прокатної кліті товстолистового прокатного стана 3000 з метою визначення частоти коливань, що виникають при роботі, вплив їх амплітуди на динамічні відхилення швидкості робочого органу від заданої. Обґрунтовано дослідження методом математичного моделювання при проектуванні та експлуатації механічного обладнання. Наведена розрахункова схема механічної частини електромеханічної системи. Підтверджено за допомогою результатів моделювання, що вплив внутрішнього в'язкого тертя у валопроводах на загасання коливань не суттєвий по відношенню до демпфуючих властивостей самого електроприводу. Тому у першому наближенні його можна не враховувати. Електропривод прокатної кліті розглядався як система ТП–Д з регуляторами швидкості та струму при стандартних налагодженнях їх на модульний оптимум. Для зменшення величини пружних моментів у кінематичному ланцюзі стана в електроприводі використано задавач інтенсивності струму якоря. Моделювання процесів проводилось у середовищі Simulink пакету MATLAB. Модель кліті побудована за розрахунковою схемою та відображає розгалуження на два канали з поєднанням їх через пружний елемент – матеріал, що прокатується. Жорсткості валів механічних передач розраховувалися виходячи з реальної геометрії з врахуванням властивостей матеріалу. В модель задавалася програма прокатки зі змінною швидкістю – з метою компенсації різнотовщинності, що відповідає сучасним технологічним тенденціям. Порівнюючи результати моделювання встановлено, що наявність зазору провокує появу автоколивань, загасання яких в реальній механічній системі буде відбуватися за рахунок демпфуючих властивостей валів. Поява зазору в шпинделях призводить до збільшення частоти коливань системи (70-80 Гц).

Ключові слова: *прокатна кліть, пружньов'язкий елемент, механічні коливання.*

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