Торайғыров университетінің хабаршысы
ҒЫЛЫМИ ЖУРНАЛЫ

НАУЧНЫЙ ЖУРНАЛ
Вестник Торайгыров университета

Торайғыров университетінің ХАБАРШЫСЫ

Энергетикалық сериясы

1997 жылдан бастап шығады

ВЕСТНИК Торайгыров университета

Энергетическая серия

Издается с 1997 года

ISSN 2710-3420

№ 4 (2025)

ПАВЛОДАР

НАУЧНЫЙ ЖУРНАЛ
Вестник Торайгыров университета

Энергетическая серия
выходит 4 раза в год

СВИДЕТЕЛЬСТВО
о постановке на переучет периодического печатного издания,
информационного агентства и сетевого издания
№ 14310-Ж
выдано
Министерство информации и общественного развития
Республики Казахстан

Тематическая направленность
публикация материалов в области электроэнергетики,
электротехнологии, автоматизации, автоматизированных и
информационных систем, электромеханики и теплоэнергетики

Подписной индекс – 76136

https://doi.org/10.48081/GMYR4462

SRSTI 45.41.33

https://doi.org/10.48081/IXOC3704

A. Y. Batyrbek[1], G. A. Sivyakova[2], V. K. Tytiuk[3]
[1,2]Karaganda Industrial University, Republic of Kazakhstan, Temirtau;
[3]Kryvyi Rih National University, Ukraine, Kryvyi Rih.
[1]ORCID: https://orcid.org/0000-0002-3369-6804
[2]ORCID: https://orcid.org/0000-0001-7689-8433
[3]ORCID: https://orcid.org/0000-0003-1077-3288
*e-mail: a.batyrbek@tttu.edu.kz

MODELING THE ELECTROMECHANICAL SYSTEM OF DRUM FLYING SHEARS USING LAGRANGE EQUATIONS

This article presents mathematical modeling of the electromechanical system of drum flying shears in the conditions of the rolling mill CWBRM-1700 of JSC «Qarmet» based on the Lagrange equations of the second kind. The study is aimed at developing a dynamic model that takes into account the interaction of electric drives, kinematic links and cutting elements under high speeds and loads. The aim of the work is to develop a mathematical model of the electromechanical system of drum flying shears of the rolling mill CWBRM-1700 based on the Lagrange equations of the second kind, taking into account elastic deformations, inertial characteristics and external loads. Generalized coordinates, forces and moments acting in the system are determined, taking into account the technological features of the transverse metal cutting process. Based on the Lagrange equations, differential equations of motion are obtained that describe the dynamics of the system. An analysis of the influence of the parameters of electromechanical components on the accuracy and stability of the shears is carried out. The results of the study allow us to quantitatively evaluate the influence of elastic connections in the mechanical structure of flying shears on the kinematic characteristics of the upper and lower cutting drums, as well as to determine the amplitude-frequency parameters of the resulting dynamic loads. The developed mathematical model creates the possibility of precise optimization of technological cutting modes, a significant increase in energy efficiency and a significant reduction *in dynamic loads. This study can be useful for specialists in the field of automation of rolling production and design of electromechanical systems.*

Key words: modeling, multimass system, electromechanical system, electric drive, hot rolling mill, elastic connections, drum flying shears.

Introduction

Continuous wide-band rolling mill (hereinafter referred to as CWBRM-1700) of JSC «Qarmet» are complex electromechanical systems in which an important role is played by equipment for transverse cutting of metal, in particular, drum flying shears. The efficiency of their operation directly affects the quality of products and the productivity of the entire mill [1; 2, P. 18]. However, the dynamic processes that occur during cutting of a moving strip are accompanied by significant mechanical loads, vibrations and elastic deformations, which requires an accurate mathematical description of the system to optimize control and increase reliability [3; 4; 5]. In this regard, the use of Lagrange equations of the second kind for constructing a mathematical model of drum flying shears seems promising, since this method provides a systematic account of kinetic and potential energy, as well as dissipative forces [6; 7].

The scientific novelty of the study lies in the application of the Lagrangian formalism for modeling a multi-mass system of flying scissors taking into account real operational factors, which allows for more accurate prediction of the dynamic operating modes of the equipment.

Materials and methods

Drum-type shears with an inclined upper knife are installed between the roughing and finishing groups of stands of the hot rolling mill «1700» of sheet rolling shop 1 of JSC «Qarmet» and are designed for cutting the front and rear ends of the rolled product (Figure 1).

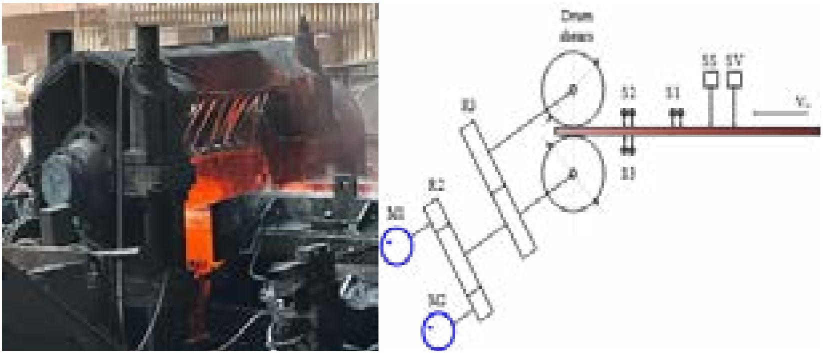

Figure 1 – General view of drum shears (a), process flow diagram and electromechanical equipment layout diagram (b) of drum shears of the hot rolling mill «1700»: S1, S2 – existing strip presence sensors; S3 – strip presence sensor (DELTA IRIS); SS – lane presence sensor (DELTA Rota – Sonde); SV – lane speed sensor (DELTA Velas DL 4068); VS – strip speed

At present, a reversible thyristor DC electric drive with independent excitation is used as an electric drive for flying shears at the CWBRM-1700 hot rolling mill. The shears are driven by two DC electric motors through a cylindrical gearbox. A double-anchor DC electric motor of the P19-75–7K type with independent excitation is used as a DC drive motor. Double-anchor motors are often used in flying shear electric drives due to their lower moment of inertia compared to single-anchor motors of the same power. The P19-75–7K type motor has the following technical characteristics: rated power – 1750 kW; armature voltage – 600 V; rotation speed – 190 rpm; rated current – 2980 A. The excitation windings are powered by built-in exciters of the SINAMICS 6RA8095-4GV62-0AA0 TC. The converters are selected with the required parameters of the built-in exciters. Figure 2 shows the electrical circuit diagram of the drum-type flying shear drive.

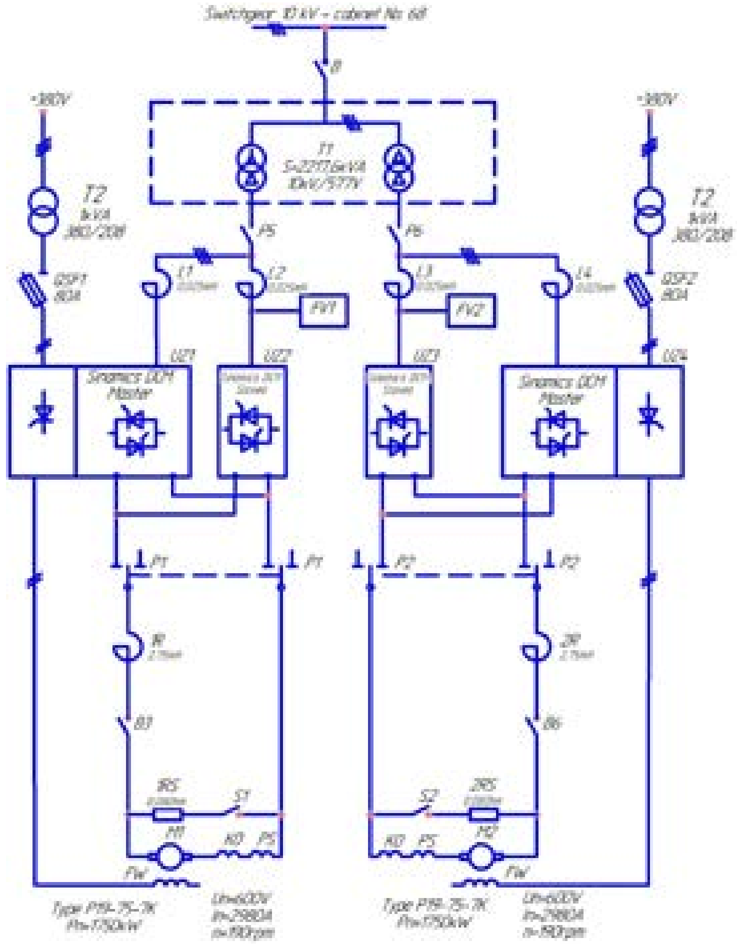

Figure 2 – Electrical schematic diagram of the drum-type flying shear drive

Mathematical modeling of a DC motor.

When modeling a DC motor with separate excitation, the following generally accepted assumptions are used:

1) the excitation current has a constant value;

2) we neglect the saturation value both along the main magnetic flux contour and along the scattering contour;

3) we do not take into account the influence of the eddy current circuit;

4) the machine is fully compensated, i.e. the influence of the anchor reaction is absent.

The system of equations of a DC motor in canonical form has the well-known form:

$$\begin{cases} L_\Sigma \dfrac{dI_A}{dt} = U_A - k\phi \cdot \omega - I_A R_\Sigma \\ J\dfrac{d\omega}{dt} = k\phi \cdot I_A - M_c \end{cases} \tag{1}$$

where L_Σ – is the total inductance of the armature circuit, H;

R_Σ – total active resistance of the anchor circuit, Ohm;

U_A – armature voltage, V;

I_A – armature current, A;

ω – and the angular velocity of the anchor 1/s;

J – moment of inertia of the anchor, kg· m2;

$k\phi$ – engine flux coefficient, V·s;

M_C – moment of resistance of the electric drive, N·m.

Also in this work we will neglect the discreteness of the operation of the adjustable thyristor converter, using modeling based on average values.

One of the most common methods of mathematical modeling of complex mechanical systems is the use of Lagrange equations. The most important characteristic of a mechanical system is the number of degrees of freedom (DOF, the degrees of freedom) [8]. In many scientific fields, the degrees of freedom of a system are understood as the number of system parameters that can change independently of each other.

For mechanical systems, the number of degrees of freedom H is equal to the number of independent coordinates that completely describe the state at an arbitrary point in the system.

These coordinates are called generalized coordinates and are denoted by the symbols q_1, q_2, … q H. Thus, the number of degrees of freedom is simultaneously the minimum necessary number of coordinates with which it is possible to cover all possible states of the system. [9, P. 164]; [10]. Accordingly, the first and second derivatives of generalized coordinates are called generalized velocity and generalized acceleration.

In practice, simultaneously with generalized coordinates it is advisable to use auxiliary coordinates, which are determined from generalized coordinates by constraint equations. These coordinates are redundant, but they increase clarity when studying the operation of mechanical systems.

The behavior of an arbitrary mechanical system can be exhaustively described by the well-known Lagrange equations of the second kind in generalized coordinates, which have the following form:

$$\frac{d}{dt}\left(\frac{\partial K}{\partial \dot{q}_i}\right) - \frac{\partial L}{\partial q_i} = Q_i, i = 1,2 \cdots H \qquad (2)$$

where $L = (K - P)$ – Lagrange function;

K, P – respectively, kinetic and potential energy of a mechanical system;

Q_i is a generalized force of non-potential nature.

Generalized forces Q_i can be found from the sum of elementary works during virtual displacements:

$$\delta W = Q_1 \cdot \delta q_1 + Q_2 \cdot \delta q_2 + \cdots + Q_H \cdot \delta q_H \qquad (3)$$

Equating the coefficients of variations of generalized coordinates.

Since we are considering the motion of an electromechanical system of drum shears, in which all elements perform a rotational motion, we can make the following simplifying considerations:

1) the potential energy of the system under consideration includes only the energy of elastic deformation of individual elements of the system that have elastic properties;

2) to simplify the formulas, it is advisable to use the twist angle of the elastic element as a generalized coordinate for elements with elastic properties.

As shown in [9, p. [164]; [10], the Lagrange equations of the second kind (2) can be represented in expanded form in the form of a system of second-order differential equations:

$$\begin{cases} a_{11} \cdot \ddot{q}_1 + a_{12} \cdot \ddot{q}_2 + \cdots + a_{1H} \cdot \ddot{q}_H + c_{11} \cdot q_1 + c_{22} \cdot q_2 + \cdots + c_{1H} \cdot q_H = Q_1 \\ a_{21} \cdot \ddot{q}_1 + a_{22} \cdot \ddot{q}_2 + \cdots + a_{2H} \cdot \ddot{q}_H + c_{21} \cdot q_1 + c_{22} \cdot q_2 + \cdots + c_{2H} \cdot q_H = Q_2 \\ \cdots \\ a_{H1} \cdot \ddot{q}_1 + a_{H2} \cdot \ddot{q}_2 + \cdots + a_{HH} \cdot \ddot{q}_H + c_{H1} \cdot q_1 + c_{H2} \cdot q_2 + \cdots + c_{HH} \cdot q_H = Q_H \end{cases} \qquad (4)$$

where rrhcbd are the generalized inertial coefficients; rrhcbd are the generalized quasi-elastic coefficients.

For the sake of compactness of presentation, we present equation (4) in vector-matrix form:

$$A \cdot \ddot{q} + C \cdot q = Q \qquad (5)$$

where $q, \ddot{q}$ are the column vectors of generalized coordinates and their second derivatives of dimension [1×H]; A, C are square matrices of generalized inertial coefficients and generalized quasi-elastic coefficients of dimension [H×H]; Q is the column vector of generalized forces of dimension [1×H].

Generalized inertial and quasi-elastic coefficients are found from the condition of equality of the coefficients of quadratic forms of the kinetic and potential energy of a mechanical system, respectively.

Let us compose the Lagrange equations of the second kind for a mechanical system of drum shears.

The calculation scheme of the mechanical system of drum shears can be significantly simplified by the following assumptions:

1) due to the insignificant length of the shafts connecting the drive motors with the reduction gear, it is permissible to neglect their elastic- viscous properties and exclude them from the calculation scheme. elastic-viscous elements [c1, b1] and [c_2, b_2];

2) due to the absence of elastic-viscous properties in the drive unit, it is advisable to present this unit as a single-engine unit, with an engine of double power and double the moment of inertia.

Taking these assumptions into account, we obtain the following three-mass calculation scheme of the mechanical system of drum shears, Figure 3.

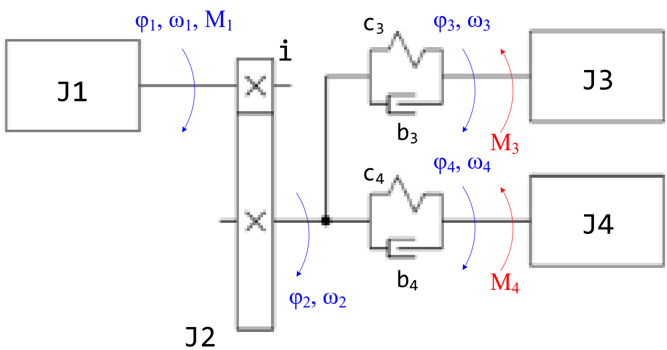

Figure 3 – Simplified three-mass calculation scheme of the mechanical part of drum-type shears

In the calculation scheme of Figure 3, the moment of inertia of the first mass J1 includes the moments of inertia of the two drive motors. Mass J2 includes both the moment of inertia of all elements of the reduction gear and the moment of inertia of the distribution gear. Masses J3, J4 represent the upper and lower drum of the drum shears with their own moments of inertia.

The elastic-viscous elements [c3, b3] and [c4, b4] model the shafts between the distribution gear and, respectively, the upper D1 and lower D2 shear drums.

As the first generalized coordinate we will take the absolute value of the rotation angle at the beginning of the kinematic chain:

$$q_1 = \varphi_1 \tag{6}$$

As the second generalized coordinate we will take the twist angle of the elastic shaft 2–3, which is equal to the difference in the angles of rotation of the ends of this shaft:

$$q_2 = \varphi_3 - \varphi_2 \tag{7}$$

Accordingly, we will take the angle of twist of the elastic shaft 2–4 as the third generalized coordinate, which is equal to the difference in the angles of rotation of the ends of this shaft:

$$q_3 = \varphi_4 - \varphi_2 \tag{8}$$

Let us express the positions and angular velocities of the shafts of the kinematic scheme through generalized coordinates:

$$\begin{aligned} \varphi_2 &= \frac{\varphi_1}{i} = \frac{q_1}{i} \\ \varphi_3 &= q_2 + \varphi_2 = q_2 + \frac{q_1}{i} \\ \varphi_4 &= q_3 + \varphi_2 = q_3 + \frac{q_1}{i} \end{aligned} \tag{9}$$

Definition of inertial coefficients. Let us formulate an equation for the kinetic energy of the mechanism and represent it as a function of generalized coordinates:

$$\begin{aligned} K &= \frac{1}{2}(J_1 \cdot \omega_1^2 + J_2 \cdot \omega_2^2 + J_3 \cdot \omega_3^2 + J_4 \cdot \omega_4^2) \\ K &= \frac{1}{2}(J_1 \cdot \dot{\varphi}_1{}^2 + J_2 \cdot \dot{\varphi}_2{}^2 + J_3 \cdot \dot{\varphi}_3{}^2 + J_4 \cdot \dot{\varphi}_4{}^2) \end{aligned} \tag{10}$$

Substituting equation (9) into (10) after reducing similar terms, we obtain the following equation:

$$K = \frac{1}{2}\left(q_1^2 \cdot \left(J_1 + \frac{J_2 + J_3 + J_4}{i}\right) + q_2^2 \cdot J_3 + q_3^2 \cdot J_4 + \cdots \right.$$
$$\left. + 2\frac{1}{i} \cdot q_1 \cdot q_2 + 2\frac{1}{i} \cdot q_1 \cdot q_3\right) \quad (11)$$

The kinetic energy of a mechanism can be represented as a complete quadratic form, which in the case of a mechanism with a number of degrees of freedom has the following form:

$$K = \frac{1}{2}\left(a_{11} \cdot q_1^2 + 2a_{12} \cdot q_1 \cdot q_2 + 2a_{13} \cdot q_1 \cdot q_3 + a_{22} \cdot q_2^2 + \cdots \right.$$
$$\left. + 2a_{23} \cdot q_2 \cdot q_3 + a_{33} \cdot q_3^2\right) \quad (12)$$

By equating equations (11) and (12), we obtain an equation for the matrix of inertial coefficients included in equation (5):

$$A = \begin{bmatrix} J_1 + \frac{J_2 + J_3 + J_4}{i} & \frac{1}{i} & \frac{1}{i} \\ \frac{1}{i} & J_3 & 0 \\ \frac{1}{i} & 0 & J_4 \end{bmatrix} \quad (13)$$

Definition of quasi-elastic coefficients. To determine the matrix of quasi-elastic coefficients, we will compile an equation of potential energy for the mechanism of drum shears. The potential energy of the mechanism is equal to the sum of the potential energies of all elastic-viscous links and, taking into account the adopted generalized coordinates, can be represented by the following equation:

$$\Pi = \frac{1}{2}\left(C_3 \cdot q_2^2 + C_4 \cdot q_3^2\right) \quad (14)$$

Representing the kinetic energy equation (14) in the form of a complete quadratic form similar to (12), we obtain the following expression for the matrix of quasi-elastic coefficients:

$$D = \begin{bmatrix} 0 & 0 & 0 \\ 0 & C_3 & 0 \\ 0 & 0 & C_4 \end{bmatrix} \quad (15)$$

Definition of generalized forces. In addition to the torque of the electric drive M1 and the moments of resistance of the drums M3 and M4, it is necessary to take into account the moments of resistance of the dissipative forces R_1 and R_2, arising during the deformation of shafts 2–3 and 2–4, respectively.

Taking into account the adopted generalized coordinates, the equations for dissipative forces can be expressed as follows:

$$\begin{aligned} R_3 &= b_3 \cdot (\dot{\varphi}_3 - \dot{\varphi}_2) = b_3 \cdot \dot{q}_2 \\ R_4 &= b_4 \cdot (\dot{\varphi}_4 - \dot{\varphi}_2) = b_4 \cdot \dot{q}_3 \end{aligned} \quad (16)$$

The equation of work for small virtual displacements of the drum shear mechanism has the form:

$$\delta W = M_1 \cdot \delta\varphi_1 - M_3 \cdot \delta\varphi_3 - M_4 \cdot \delta\varphi_4 - R_3 \cdot \delta q_2 - R_3 \cdot \delta q_3 \quad (17)$$

Substituting into (17) the equations of relations (9) after reducing similar ones, we obtain the following expression:

$$\delta W = \delta q_1 \cdot \left(M_1 - \frac{M_3 + M_4}{i}\right) - \delta q_2 \cdot (M_3 + b_3 \cdot \dot{q}_2) \quad (18)$$

$$-\delta q_3 \cdot (M_4 + b_4 \cdot \dot{q}_3)$$

Thus, the column vector of generalized forces for the drum shear mechanism can be represented in the form:

$$Q = \begin{bmatrix} M_1 - \dfrac{M_3 + M_4}{i} \\ -(M_3 + b_3 \cdot \dot{q}_2) \\ -(M_4 + b_4 \cdot \dot{q}_3) \end{bmatrix} \quad (19)$$

For the convenience of mathematical modeling, it is necessary to divide expression (19) into two components: external forces and internal viscous forces depending on the derivatives of the generalized coordinates.

Taking into account such a transformation, we write the vector-matrix equation of the mechanical system in the form:

$$A \cdot \ddot{q} = B \cdot \dot{q} + C \cdot q + M \cdot T \quad (20)$$

In this equation, the matrix A is defined above by equation (13). The matrix C = – D:

$$C = \begin{bmatrix} 0 & 0 & 0 \\ 0 & -C_3 & 0 \\ 0 & 0 & -C_4 \end{bmatrix} \tag{21}$$

According to (18), matrix B will have the form:

$$B = \begin{bmatrix} 0 & 0 & 0 \\ 0 & -b_3 & 0 \\ 0 & 0 & -b_4 \end{bmatrix} \tag{22}$$

Column vector of external forces T has the form:

$$T = \begin{bmatrix} M_1 \\ M_3 \\ M_4 \end{bmatrix} \tag{23}$$

Then the matrix M will have the form:

$$M = \begin{bmatrix} 1 & -\frac{1}{i} & -\frac{1}{i} \\ 0 & -1 & 0 \\ 0 & 0 & -1 \end{bmatrix} \tag{24}$$

Let us solve equation (20) with respect to the highest derivative. We multiply both parts of the equation on the left by the inverse matrix of generalized inertial coefficients A –1 . We obtain the equation:

$$\begin{gathered} \ddot{q} = A^{-1} \cdot B \cdot \dot{q} + A^{-1} \cdot C \cdot q + A^{-1} \cdot F \cdot T \\ \ddot{q} = K \cdot \dot{q} + L \cdot q + M \cdot T \end{gathered} \tag{25}$$

The matrix structural diagram that implements the solution of equation (25) is shown in Figure 4.

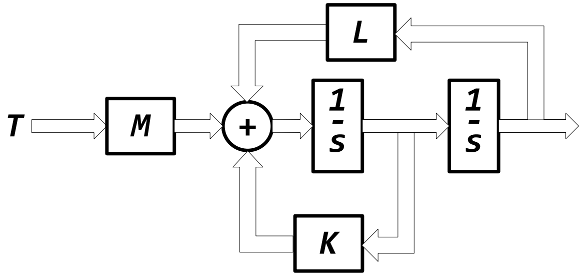

Figure 4 – Matrix structural diagram implementing the solution of the equations of a mechanical system in the form of Lagrange equations of the second kind

Coordinate transformation for observing the state of the model. The mathematical model of a mechanical system based on the Lagrange equations of the 2nd kind usually does not contain variables corresponding to the real angles of rotation of the shafts and the angular velocities of individual elements, which causes inconvenience when analyzing the model.

The actual rotation angles of the shafts and the angular velocities of the individual elements can be calculated based on the values of the generalized coordinates. In the case of the mechanical system of drum shears, Figure 5, these expressions are obtained above and are presented by formula (9).

Proper angular velocities are calculated using the same expressions, written relative to the first derivative of the rotation angles and generalized coordinates.

In matrix form, these expressions can be written as follows:

$$\varphi = F \cdot q; \ \omega = F \cdot \dot{q} \tag{26}$$

where *φ, ω* are respectively the column vectors of the proper rotation angles and angular velocities of the elements of the mechanical system; F is the coordinate transformation matrix.

In the case of a mechanical system of drum shears, we assume that the column vectors of the proper rotation angles and angular velocities of the elements of the mechanical system have the dimension [1×4], taking into account the mass J_2.

Then, in accordance with equation (9), the coordinate transformation matrix will have the form:

$$F = \begin{bmatrix} 1 & 0 & 0 \\ \frac{1}{i} & 0 & 0 \\ \frac{1}{i} & 1 & 0 \\ \frac{1}{i} & 0 & 1 \end{bmatrix} \tag{27}$$

Solving equations in MATLAB. To solve the differential equations of a multimass mechanical system in the form of Lagrange equations, presented in vector-matrix form by equations (25, 26), we use the MATLAB program, optimized for performing vector - matrix calculations. Based on the matrix structural diagram implementing the solution of the equations of a mechanical system in the form of Lagrange equations of the second kind, Figure 4, a mathematical model of the mechanical part of drum shears was developed in the MATLAB/Simulink environment, presented in Figure 5.

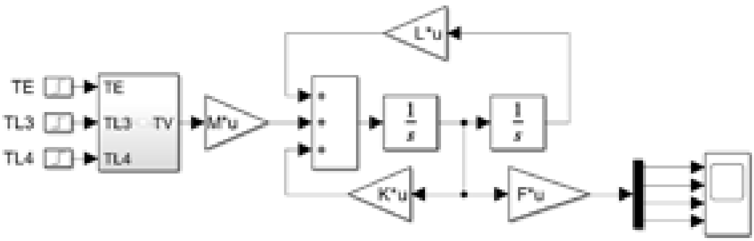

Figure 5 – Mathematical model of the mechanical part of drum shears in the MATLAB/Simulink environment

This model is universal, suitable for calculating an arbitrary mechanical system whose equations are presented in the form of Lagrange equations. The specificity of the object under study in this model is determined by the structure of the matrices *K, L, M* and the coordinate transformation matrix *F.* It is also necessary to correctly form the input vector of moments TV. The calculation of the numerical values of the parameters of the mathematical model of drum shears, Figure 5, is performed in a separate, pre-executed MATLAB script.

The results of the mathematical model of the mechanical part of drum shears are presented in Figure 6.

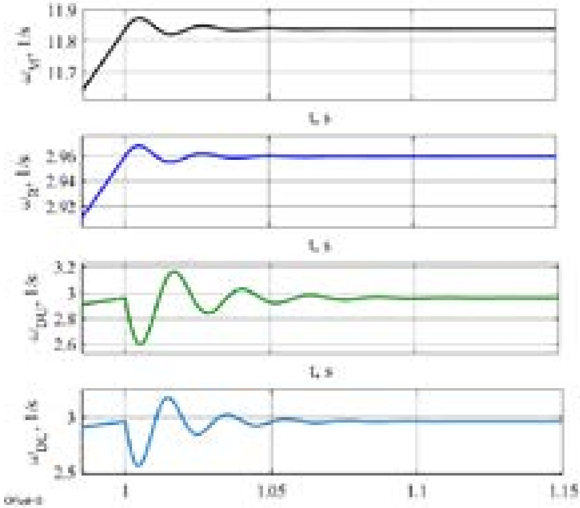

Figure 6 – Results of the mathematical model of the mechanical part of drum shears

The mathematical model considered the transient process of starting the drum shears under the action of a constant driving moment TE. At the time t = 1 s, the balancing moments of resistance TL3 and TL4 are applied to the shear drums. When calculating the moments of inertia of individual elements of the mechanical part, the mathematical model takes into account the difference in the diameters of the upper and lower drums of the flying shears. Figure 6 shows a fragment of the transient process containing the process of applying loads to individual drums and transition to a steady state.

Results and discussion

The paper presents a fairly general approach to the synthesis of a mathematical model of mechanical systems in the form of vector-matrix equations with respect to generalized coordinates using the Lagrange equations of the second kind. Using the proposed approach, a mathematical model of the mechanical part of drum shears has been developed, presented in the form of a three-mass mechanical system taking into account the finite elasticity of mechanical connections. When calculating the parameters of the mathematical model of drum shears, real mass and size characteristics of drum shears installed on the continuous wide-strip mill CWBRM-1700 of JSC «Qarmet», Temirtau, were used.

MATLAB/Simulink was used to solve the equations of the mathematical model of the three-mass mechanical part of the drum shears.

The obtained results allow us to estimate the influence of elastic connections in the mechanical part of flying shears on the movement of the upper and lower drums, to estimate the magnitude of dynamic loads in the mechanical part. The difference in the movement of the upper and lower drums is due to the difference in the diameters of these drums.

The disadvantages of this study include the absence of direct current drive motors in the proposed model, which would allow us to clarify the nature of the processes occurring when a load is applied.

As a direction for further research, it is proposed to consider the possibility of complicating the equivalent mechanical circuit of drum shears.

Conclusion

A universal approach to the synthesis of equations of multi-mass mechanical systems with elastic connections and internal viscous friction based on the Lagrange equations of the second kind is proposed.

A mathematical model of the mechanical part of the drum shears of the continuous wide-strip mill CWBRM-1700 of JSC «Qarmet», Temirtau, was synthesized, presented in the form of an equivalent three-mass system.

The solution of the equations of this model is performed using MATLAB/Simulink.

The obtained results allow us to evaluate the influence of elastic connections in the mechanical part of flying shears on the movement of the upper and lower drums, and to evaluate the magnitude of dynamic loads in the mechanical part.

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Received 22.05.25.
Received in revised form 04.11.25.
Accepted for publication 05.12.25.

**Ә. Е. Батырбек[1], Г. А. Сивякова[2], В. К. Тытюк[3]*
[1,2]Қарағанды индустриялық университеті,
Қазақстан Республикасы, Теміртау қ.;
[3]Кривой Рог ұлттық университеті, Украина, Кривой Рог қ.
22.05.25 ж. баспаға түсті.
04.11.25 ж. түзетулерімен түсті.
05.12.25 ж. басып шығаруға қабылданды.

ЛАГРАНЖ ТЕҢДЕУЛЕРІНЕ НЕГІЗДЕЛГЕН БАРАБАНДЫ ҮШҚЫР ҚАЙШЫЛАРДЫҢ ЭЛЕКТРОМЕХАНИКАЛЫҚ ЖҮЙЕСІН МОДЕЛЬДЕУ

Бұл мақалада екінші типтегі Лагранж теңдеулері негізінде «Qarmet» АҚ ҮКЖО-1700 илемдеу орнағы жағдайында барабанды қайшылардың электромеханикалық жүйесін математикалық модельдеу ұсынылған. Зерттеу жоғары жылдамдықтар мен жүктемелер жағдайында электр жетектерінің, кинематикалық байланыстардың және кесу элементтерінің өзара әрекеттесуін ескеретін динамикалық модель жасауға бағытталған. Жұмыстың мақсаты серпімді деформацияларды, инерциялық сипаттамаларды және сыртқы жүктемелерді ескере отырып, II типті Лагранж *теңдеулері негізінде ҮКЖО-1700 илемдеу орнағының барабанды қайшыларының электромеханикалық жүйесінің математикалық моделін жасау болып табылады. Металды көлденең кесу процесінің технологиялық ерекшеліктерін ескере отырып, жүйеде әрекет ететін жалпыланған координаттар, күштер мен моменттер анықталды. Лагранж теңдеулерінен жүйенің динамикасын сипаттайтын қозғалыстың дифференциалдық теңдеулері алынады. Электромеханикалық компоненттердің параметрлерінің қайшының дәлдігі мен тұрақтылығына әсерін талдау жүргізілді. Зерттеу нәтижелері қайшының механикалық құрылымындағы серпімді байланыстардың жоғарғы және төменгі кесу барабандарының кинематикалық сипаттамаларына әсерін сандық бағалауға, сондай-ақ пайда болатын динамикалық жүктемелердің амплитудалық-жиілік параметрлерін анықтауға мүмкіндік береді. Әзірленген математикалық модель кесудің технологиялық режимдерін дәл оңтайландыруға, энергия тиімділігін едәуір арттыруға және динамикалық жүктемелерді айтарлықтай төмендетуге мүмкіндік береді. Бұл зерттеу илемдеу өндірісін автоматтандыру және электромеханикалық жүйелерді жобалау саласындағы мамандар үшін пайдалы болуы мүмкін.*

Кілтті сөздер: модельдеу, көп массалы жүйе, электромеханикалық жүйе, электр жетегі, ыстық илектеу орнағы, серпімді байланыстар, барабанды қайшылар.

**Ә. Е. Батырбек[1], Г. А. Сивякова[2], В. К. Тытюк[3]*
[1,2]Карагандинский индустриальный университет,
Республика Казахстан, г. Темиртау;
[3]Криворожский национальный университет, Украина, Кривой Рог.
Поступило в редакцию 22.05.25.
Поступило с исправлениями 04.11.25.
Принято в печать 05.12.25.

МОДЕЛИРОВАНИЕ ЭЛЕКТРОМЕХАНИЧЕСКОЙ СИСТЕМЫ БАРАБАННЫХ ЛЕТУЧИХ НОЖНИЦ НА ОСНОВЕ УРАВНЕНИЙ ЛАГРАНЖА

В данной статье представлено математическое моделирование электромеханической системы барабанных летучих ножниц в условиях прокатного стана НШПС-1700 АО «Qarmet» на основе уравнений Лагранжа второго рода. Исследование направлено на *разработку динамической модели, учитывающей взаимодействие электроприводов, кинематических связей и режущих элементов в условиях высоких скоростей и нагрузок. Целью работы является разработка математической модели электромеханической системы барабанных летучих ножниц прокатного стана НШПС-1700 на основе уравнений Лагранжа II рода с учетом упругих деформаций, инерционных характеристик и внешних нагрузок. Определены обобщённые координаты, силы и моменты, действующие в системе, с учётом технологических особенностей процесса поперечной резки металла. На основе уравнений Лагранжа получены дифференциальные уравнения движения, описывающие динамику системы. Проведён анализ влияния параметров электромеханических компонентов на точность и стабильность работы ножниц. Результаты исследования позволяют количественно оценить влияние упругих связей в механической структуре летучих ножниц на кинематические характеристики верхнего и нижнего режущих барабанов, а также определить амплитудно-частотные параметры возникающих динамических нагрузок. Разработанная математическая модель создает возможность точной оптимизации технологических режимов резания, существенного повышения энергоэффективности и значительного снижения динамических нагрузок. Данное исследование может быть полезно для специалистов в области автоматизации прокатного производства и проектирования электромеханических систем.*

Ключевые слова: моделирование, многомассовая система, электромеханическая система, электропривод, стан горячей прокатки, упругие связи, барабанные летучие ножницы.

Теруге 17.12.2025 ж. жіберілді. Басуға 30.12.2025 ж. қол қойылды.
Электронды баспа
29.9 Mb RAM
Шартты баспа табағы 22,2. Таралымы 300 дана. Бағасы келісім бойынша.
Компьютерде беттеген: А. К. Мыржикова
Корректор: А. Р. Омарова, Д. А.Кожас
Тапсырыс № 4494

Сдано в набор 17.12.2025 г. Подписано в печать 30.12.2025 г.
Электронное издание
29.9 Mb RAM
Усл. печ. л. 22,2. Тираж 300 экз. Цена договорная.
Компьютерная верстка: А. К. Мыржикова
Корректор: А. Р. Омарова, Д. А. Кожас
Заказ № 4494

«Toraighyrov University» баспасынан басылып шығарылған
Торайғыров университеті
140008, Павлодар қ., Ломов к., 64, 137 каб.

«Toraighyrov University» баспасы
Торайғыров университеті
140008, Павлодар қ., Ломов к., 64, 137 каб.
67-36-69
E-mail: kereku@tou.edu.kz
www.vestnik-energy.tou.edu.kz