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Synthesis of adaptive electric drive control system of horizontal looper

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Abstract: The article presents studies on the electromechanical system of a metallurgical horizontal looper in the steelmaking industry. During the operation of this unit, parameters in the system changes due to variations of length and mass of the steel strip, these variations significantly change elastic properties and reduce moments of inertia. Various methods of combating elastic vibrations in electromechanical systems are analyzed in this article. The article presents a description of experiments with a horizontal looper. A mathematical model for two extreme positions of the unit was developed based on experimental results. Simulation experiments were made and their results are presented. A new control system structure is proposed to reduce vibrations in the electromechanical system of a horizontal looper. A power-up sensor, adjuster and velocity derivative feedback were added into the model structure. The proposed feedback link structure takes into account the change of steel strip length. From the experimental data it follows that the proposed system provides effective damping of mechanical vibrations in the steel strip if its length during operation is changed.

Key words: electromechanical system, horizontal looper, synthesis of damping system, variable inertia moment

1. Introduction

Electric drives interconnected through the processed flexible material find application in textile, papermaking, metallurgical production and production of synthetic materials [1].

In practice, electromechanical systems are very common where the values of resistance, mass, stiffness and inertia moments change during operation. Such systems include electric drives of rotor type industrial aggregates: screens, ball mills, mixers, granulators, rotary granular filters for gas purification, etc. The other types of systems with variable mechanical parameters, namely the moment of inertia, include winding and unwinding mechanisms for rolling steel sheet, wire, papermaking production [2].

Some multi-motor units connected through the processed material are widely used in industry. A lot of works has been devoted to controlling the units, taking into account mechanical properties of the object. Among them there are works aimed at controlling dynamic processes of the material being processed. In most cases, the material is considered absolutely rigid and weightless [3]. In other cases it is assumed that the material has a constant mass, but can change its rigidity due to temperature changes [4, 5]. If calculating such systems, there may be an assumption that the material has constant mass and rigidity [2]. These assumptions cannot be applied to horizontal loop devices (loopers) of metallurgical production, as they have a number of specific features. When the device is operating, the length of the metal strip changes significantly. In an unfilled horizontal looper, the strip length is insignificant, and it can be considered as rigid. When the looper is filled, the elastic properties of the strip appear. Also, when the strip length is changed, its mass and consequently the inertia coefficient changes. It should also be considered that the electro-mechanical system of the horizontal looper is a single-engine system.

The aim of the work is the synthesis of an electric drive system that provides an increase in the unit's productivity by reducing the vibration during the looper work.

2. Description of the study object

Horizontal looper No. 1 of the continuous aluminizing hot unit of JSC "ArcelorMittal Temirtau" is intended for accumulating of the strip during processing in order to ensure continuous operation of the machine. It refers to electromechanical systems where the moment of inertia and elastic properties of the strip change simultaneously.

The kinematic diagram of the looper is shown in Figure 1.

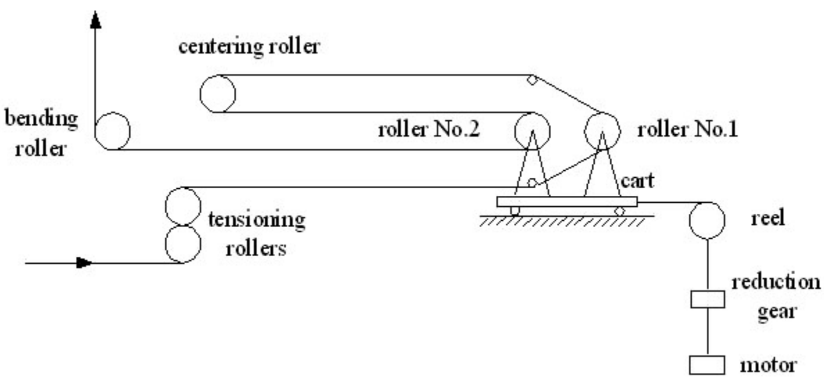


Fig. 1. Kinematic diagram of the loop device

The filling of the looper is carried out by moving the cart with rollers No. 1 and No. 2. There are two basic positions of the looper: empty and filled.

The dynamics of vertical loopers has been studied quite well. The elastic properties of the strip in such loopers seem lesser than in horizontal ones [4, 5].

The electric drive of a horizontal looper in a cold rolling mill is a multi-mass system that cannot be described as a standard double- or triple-mass system [3]. The feature of this device depends on the change of the strip length inside it. When analyzing the electric drive system of a horizontal looper, it is necessary to take into account the change of the strip elasticity and see the device as a distributed mechanical system with varies length of the strip in the looper. Also, the change of the steel strip mass should be taken into account and, therefore, the change in the inertia coefficient of the electric drive.

Currently, the horizontal looper is equipped with the direct current drive, a subordinate regulation control system, inner current loop, and external speed loop.

Vibrations occur in this mechanism, and their amplitude and frequency depend on the degree of infilling of the looper.

In the existing drive system it is not possible to provide damping of the strip vibrations during their accumulation in the unit, since the system uses a gear drive with a reduction coefficient of 33.85. In this case, the inertia coefficient varies in the range from 1.059 to 1.163. This means that vibrations in the mechanical part do not affect the operating processes of the electric drive.

The amplitude of these vibrations in some cases exceeds the distance between the parallel branches of the strip, which leads to the collapse of the strip.

The aim of the work is the synthesis of the electric drive system capable of damping vibrations in the electromechanical system of a horizontal looper.

3. Setting the research objective

The identification of the most rational methods for achieving minimal vibration in electromechanical systems with elastic linkages gives possibility to improve the quality of transients at minimal costs and thereby extends the life of the equipment.

There are various methods of elastic vibrations damping in electromechanical systems [7]:

1) technological (due to process control);
2) constructive (based on the choice of optimal design parameters when designed drives);
3) introduction to the system of additional damping devices (hydraulic, pneumatic, mechanical, based on friction couples);
4) electrical (electromechanical, electrodynamical).

1) Technological methods

Currently, the only widely used in practice method of dealing with resonant vibration in a mill is the reduction of rolling speed, which reduces the amplitude of vibrations, but at the same time reduces the average operating speed of the mill [7].

The use of reduced levels of tension and the compression mode leads to weakening of bonds along the strip, whereby the transfer of vibrational energy between the stands takes place. However, the use of this technique causes technological limitations [8].

2) Constructive methods

Out of all parameters, the most interesting are those, whose combinations will provide the maximum damping of elastic vibrations [9].

Varying flexibility of the elastic mechanical transmission bonds, can affect the fulfillment of the transmission strength, their service life and conditions [10].

In the correct way, it is possible to change the inertial mass distribution coefficient γ by choosing the gearbox ratio and the rated speed of the electric drive, as well as the values of inertia moments of the drive elements. The designer choice of the drive discrete mass value is limited by certain (catalog) values of the inertia moments of the engine rotors, gearboxes, couplings, pulleys, etc. [8].

The possibilities of using constructive methods for existing mechanisms and assemblies are very limited, because it is not always possible to replace existing equipment. It must be kept in mind that the horizontal loop device is not a double-mass system, but a multi-mass one. During the operation of the device, almost all of its parameters change, and it is not possible to select the necessary values from the standard series.

3) Introduction to the system of additional damping devices

In the presence of elastic bands in the electric drive kinematics, mechanical damping of the actuator speed is traditionally used or deliberate reduction of the speed-work of the speed control loop two or more times, compared to the system without elastic bands [11].

The active influence on vibration by serial impact in the stand is quite promising from the point of view of eliminating resonant vibrations due to reverse-phase forced vibrations of one of the masses at the resonant frequency of the electromechanical system. The impact is applied to any of the vibration elements [7].

In various versions, mechanical designers apply mechanical damping devices in drives of machines; these devices as a rule are difficult to manufacture and set up and require very careful maintenance [8].

At the same time the use of damping devices is limited for controlled electric drives due to the fact that with a speed change the vibration parameters including the frequency also change.

4) Electrical methods

One of the rational methods for damping elastic vibrations is the synthesis of the structure and the choice of control system parameters, since it is relatively easy to implement and can be used for any electromechanical system, so does not require additional material costs [12, 13].

The following main methods are considered for use in the synthesis of control systems [12–15]:

- gain correction of the speed controller in the system of subordinate regulation;
- the system creation of subordinate regulation control with the introduction of feedback on the derivative actuator speed;
- the system creation of subordinate regulation control with the introduction of feedback on the second derivative of the actuator speed;
- the system creation of subordinate regulation control with the introduction of additional feedback on the difference between the engine and actuator speeds;

– creation of a control system with a modal controller (modulator);
– creation of a control system using a band-stop filter in the speed control loop;
– use of parallel correction with indirect measurement of the actuator speed and using flexible feedback connected to the input of the current regulator (adjuster);
– the system creation of subordinate regulation control using additional parallel corrective devices.

In this work, a control system that provides damping of elastic vibrations in a horizontal loop device was synthesized for a continuous aluminizing hot unit based on a horizontal loop unit.

4. Research methods

A continuous aluminizing hot unit refers to continuous units. It is not possible to carry out experiments while it is working, so the experiments were made indirectly. The complexity of the experiments and limited abilities of physical modeling predetermined further research, narrowing them to the use a mathematical model approach.

In addition, a model of looper for the electromechanical system was already developed. The simulation experiments were carried out in extreme operating modes: with an empty and full loop device. Accuracy of the simulation model was confirmed by comparison with experimental evidence [6]. The transient processes of current and speed corresponded to the actual values measured on the continuous aluminizing hot unit. Further, structural and parametric optimization of the electromechanical system was carried out and a simulation model of the modernized system was developed. The simulation experiments carried out on the developed model, were made in accordance with the proposed modernization.

To develop a mathematical model of such a complex unit, it is necessary to have information about the mechanical parameters of the strip, including elastic moments and vibration frequencies of the strip. But there are only 3 parameters of the horizontal loop device recorded by an oscillograph in the actually working unit: drive current, linear speed and material stock (loop margin) in the unit. The measurement of elastic moments in the strip branches and the distance between the branches is not possible due to the lack of necessary sensors in the operating unit.

As a result the next experiment was conducted on the unit. The vibration period was determined visually, using a video recorder and a stopwatch timer. The vibration frequency was changing in the range of 0.3–0.5 Hz.

Vibration amplitude equal to the distance between the parallel branches of the strip was recorded.

The distance between the parallel branches of the strips in the looper is 0.85 meters. The visual observations indicated that the large portion of the strip (full loop device) has high amplitude vibration on a vertical strip up to the adjoining strip, i.e. the maximum value of the amplitude is limited by the distance between the parallel branches of the steel strip.

In the process of observation, it was found that the maximal vibrations occur when a rope is short and a strip is long, i.e. when the loop device is full. Therefore, the strip itself in the process of filling the device has the maximal influence on the dynamics of the electromechanical system of the loop device.

As part of the research the model was developed for two extreme positions of the horizontal loop device [6].

Figure 1 shows a structural chart of a horizontal looper in the electromechanical system with a standard scheme of subordinate regulation control drive.

The equations for the mathematical model:

$$
\begin{gathered}
M_H - M_{12} = J_H \frac{\mathrm{d}w_H}{\mathrm{d}t}, \\
M_{12} = c_5 \left(\phi_H - \phi_{p1}\right) + b_{12} \left(w_H - w_{p1}\right), \\
M_{12} - M_{23} - M_{56} = J_{p1} \frac{\mathrm{d}w_{p1}}{\mathrm{d}t}, \\
M_{56} = c_3 \left(\phi_{p1} - \phi_{dw}\right) \approx M_{67} = c_3 \left(\phi_{p2} - \phi_{dw}\right), \\
\begin{cases} M_{23} = c_1 \left(\phi_{p1} - \phi_{cr} \pm \dfrac{\phi_1}{2}\right) + b_{23} \left(w_{p1} - w_{cr}\right) & \text{if } \left|\phi_{p1} - \phi_{cr}\right| > \phi_1/2 \\ M_{23} = 0 & \text{if } \left|\phi_{p1} - \phi_{cr}\right| < \phi_1/2 \end{cases}, \\
M_{23} - M_{34} = J_{cr} \frac{\mathrm{d}w_{cr}}{\mathrm{d}t}, \\
\begin{cases} M_{34} = c_2 \left(\phi_{cr} - \phi_{p2} \pm \dfrac{\phi_2}{2}\right) + b_{24} \left(w_{cr} - w_{p2}\right) & \text{if } \left|\phi_{cr} - \phi_{p2}\right| > \phi_2/2 \\ M_{34} = 0 & \text{if } \left|\phi_{cr} - \phi_{p2}\right| < \phi_2/2 \end{cases}, \\
M_{34} - M_{45} - M_{67} = J_{p2} \frac{\mathrm{d}w_{p2}}{\mathrm{d}t}, \\
M_{45} = c_4 \left(\phi_{p2} - \phi_n\right) + b_{45} \left(w_{p2} - w_n\right), \\
M_{\sigma\varepsilon} - M_{56} - M_{67} = J_{dw} \frac{\mathrm{d}w_{dw}}{\mathrm{d}t},
\end{gathered} \tag{1}
$$

where: M_H, M_n, M_{dw} are the moments of the tensioning rollers, bending roller motors and the looper carriage; J_H, J_{p1}, J_{p2}, J_{cr}, J_n, J_{dw} are the moments of inertia of the tensioning rollers, roller No. 1, roller No. 2, centering roller, bending roller, the looper carriage; c_1, c_2, c_4, c_5 are the rigidities of the strip elastic couplings between masses in corresponding sections; c_3 is the rigidity of the rope elastic coupling between the carriage drum and the looper carriage; φ_H, φ_{p1}, φ_{p2}, φ_{cr}, φ_n, φ_{dw} are the rotational movements of the tensioning rollers, roller No. 1, roller No. 2, centering roller, bending roller, the looper carriage; φ_1, φ_2 are the gaps; w_H, w_{p1}, w_{p2}, w_{cr}, w_n, w_{dw}, are the speeds of rotation of the tensioning rollers, roller No. 1, roller No. 2, centering roller, bending roller, the looper carriage; M_{12}, M_{23}, M_{34}, M_{45}, M_{56}, M_{67} represent the moment of elastic forces; b_{12}, b_{23}, b_{24}, b_{45} are the damping factors.

These equations are valid for both empty and filled loop devices. In the equations, the coefficients c, b and J will vary for different positions depending on the length of the strip.

Based on Equations (1), a structural diagram of the loop device in the electromechanical system is developed and shown in Figure 2. The diagram is completed with a direct current electric drive, made according to the scheme with subordinate regulation.

The motor in the diagram is surrounded by a dotted line. The mechanical parts of roller 1, roller 2, the bending roller, centering roller and tensioning rollers are also circled in the dotted

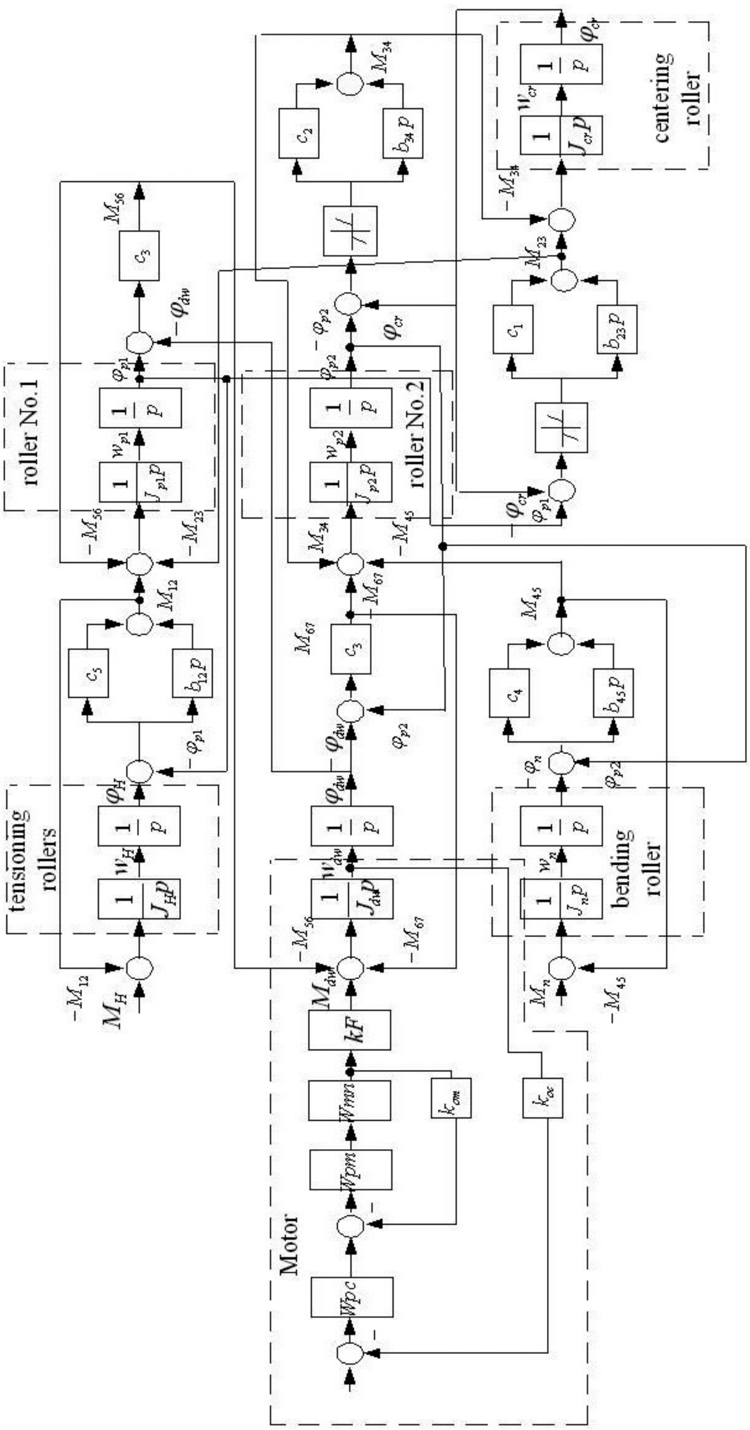


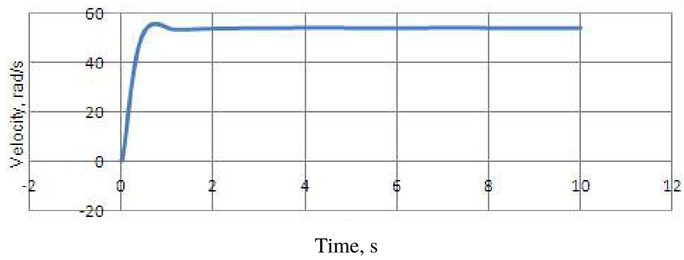
Fig. 2. Structural chart of the electromechanical system of a loop device

line. The rest of the blocks are elastic mechanical links. The input to the motor is the reference signal from the control system. The input actions M_H and M_n are the moments corresponding to the motors of the tensioning rollers and bending roller.

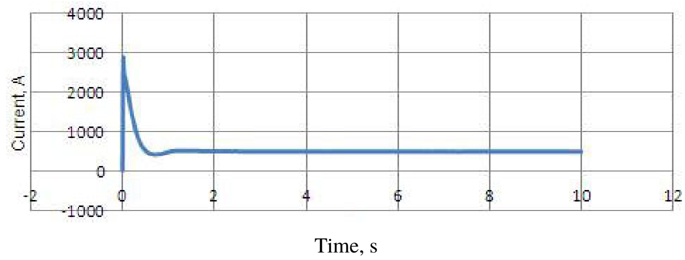
5. The simulation results

The simulation of the electromechanical system of a horizontal looper was performed in the MatLab application [16].

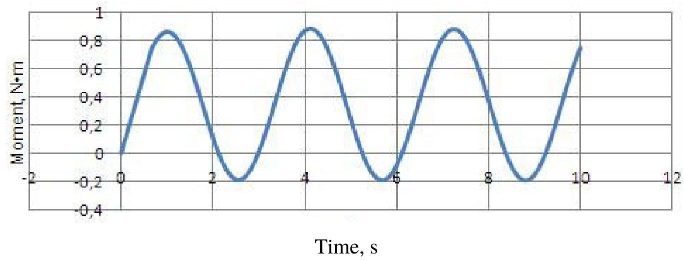
The simulation experiments were made to identify the parameters of vibration processes in the band. The simulation experiments were carried out on the models corresponding to the structural diagram of the loop device electromechanical system and are shown in Figure 3.



(a)



(b)



(c)

Fig. 3. The results of simulated experiments of transient processes in a strip with a fully filled loop device

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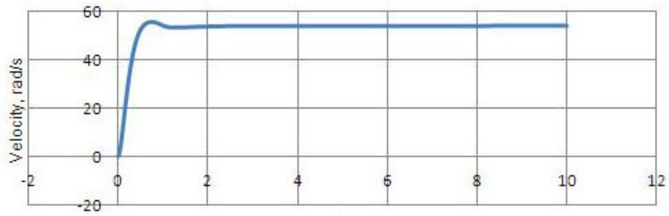
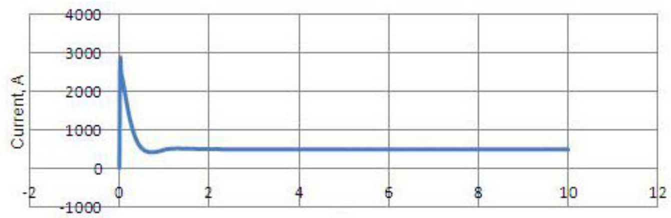
As an example, Figures 3 and 4 show the results of the simulation experiments for a fully filled (Figure 3) and for an empty loop device (Figure 4). These figures show the reaction of the control action on the engine. The graphs show the engine speed (a), motor current (b) and elastic moment (c) between roller No. 2 and the centering roller.



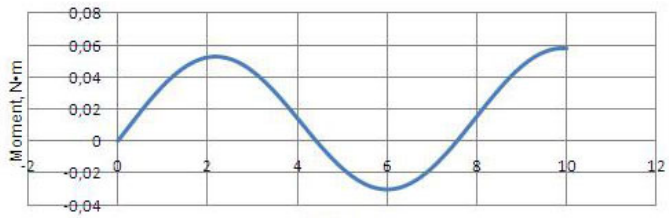
Time, s

(a)



Time, s

(b)



Time, s

(c)

Fig. 4. The results of simulated experiments of transient processes in a strip with an empty loop device

As can be seen from the graphs, the amplitude and frequency of the vibration moment for an empty loop device are much lower than for a completely filled. When the looper is empty, vibrations occur in the strip but the vibrations of the elastic moment amplitude are equal to 0.057 Nm, and when the looper is fully filled they are equal to 0.882 Nm. The difference between them is greater than 15 times. Moreover, the vibrations frequency is changed from 0.125 Hz to 0.33 Hz.

Taking into account the fact that the distances in an empty horizontal loop device, are not large, the elastic properties of the strip are barely noticeable.

The important fact is that the inertia factor in the device is changing from 1.059 up to 1.163, so the effect of the drive on the mechanical part is substantial. At the same time, the dynamic processes in the strip are weakly affected by transients in the drive. As can be seen in Figures 3 and 4, the vibrations in the elastic moment (c) practically do not affect the transient processes of speed (a) and current (b) in the electric drive.

This should be explained by comparing the presented experimental results.

6. Development of a modernized control system

The presence of elastic links leads to a significant complication of the object of an automatic control system in comparison with a rigid system. When the parameters of the object are such that the electromechanical connection between the actuator and the electric drive is strong, elasticity noticeably affects the speed loop, and under certain conditions – the current loop, in this case, the implementation of standard settings of the loops adopted in the rigid system usually becomes impossible. If the current and speed circuits are configured in the same way as in the rigid system, this will mean that the electric drive with an automatic control system does not have a noticeable damping effect on the vibrations of the mechanism, which damp only due to friction. This is usually unacceptable, so in this case it may be necessary to change control settings or system settings [17].

The looper electromechanical system is weakly damped. Such systems are sensitive to control influences. Intensity sensors serve to obtain the necessary time law of the reference-input signal of the control system. The intensity adjuster allows creating the processes of acceleration and braking of the electric drive at a given pace. For this reason, an intensity sensor is added into the system (block 1 in Figure 5). The circuit contains the same mechanical blocks as in Figure 2. In addition, it contains the blocks of the modernized control system, which are highlighted in yellow. These blocks will be described below. The input variables in the diagram in Figure 5 are the same as in Figure 2.

The integration into the direct chain of an aperiodic link with a time constant significantly greater than the links of the original automatic control system increases the stability margin of the automatic control system. The advantages include the reduction of high-frequency interference and transient variations. Therefore, this technique of increasing the stability margin is called damping with the suppression of high frequencies [18]. In Figure 5, the aperiodic link can be found as block 2.

Filters in the current and velocity feedback circuits (blocks 3 and 4 in Figure 5) are added to suppress possible ripples to prevent the appearance of weakly damped vibrations.

With values $\gamma < 1.5 \div 2$, it is impossible to achieve effective damping of elastic speed variations of the actuator without a significant decrease of speed-work due to the choice of the structure and parameters of the speed controller. Methods of correction of such systems are associated with the use of certain corrective connections in terms of the mechanism speed [19].

The introduction of feedback on the derivative of the drive shaft speed leads to an effect equivalent increase of inertia ratio. Therefore, by introducing feedback on the derivative of the velocity, smooth transitions can be obtained, although in an unadjusted system the movement of the actuator shaft has a strong vibrational character [12].

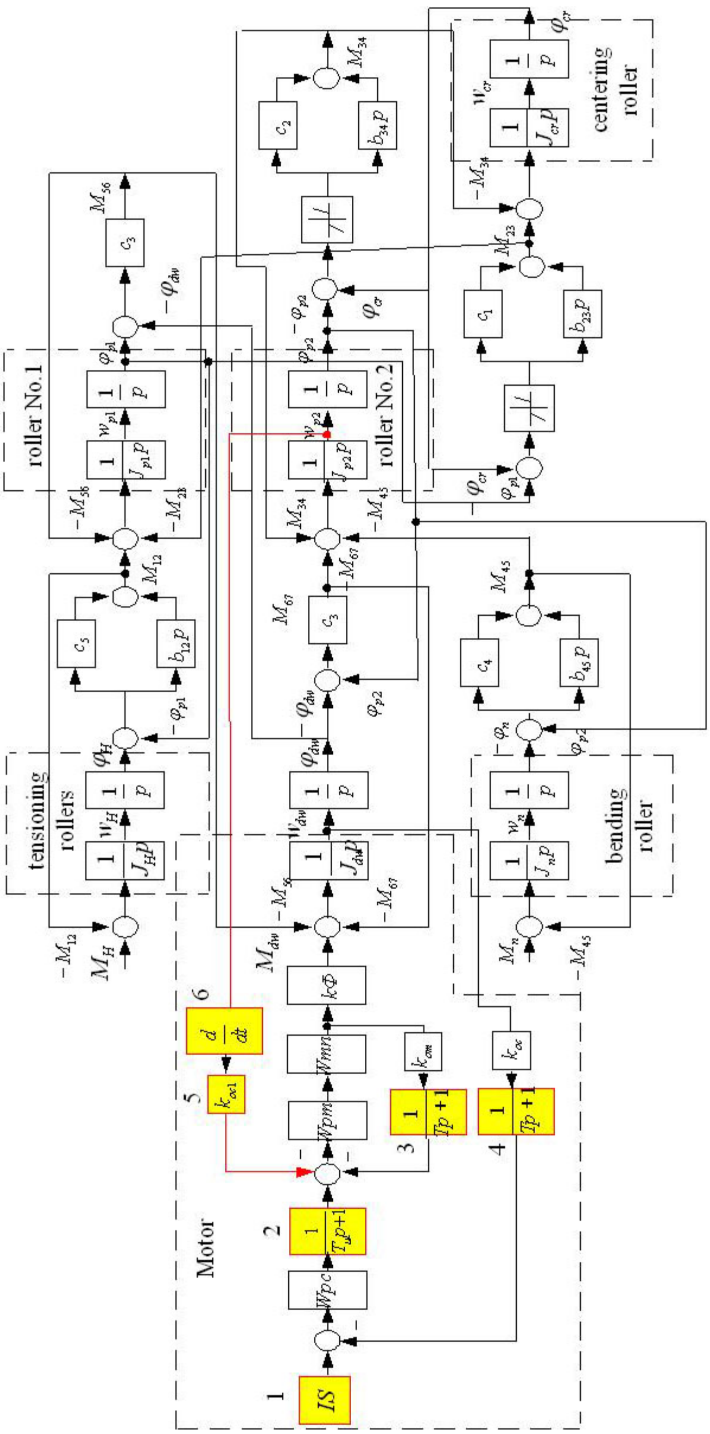


Fig. 5. Structural chart of the adjusted electromechanical system

A positive effect can also be obtained by connection to the input signal of a speed regulator of the second derivative of the actuator speed. However, two-fold differentiation is usually difficult to perform due to the presence of ripples at the output of the speed meter. Regarding the quality of the transition process for control, the same result can be obtained if a signal proportional to the difference between the drive speed and the brought to the drive actuator speed is input to the speed controller [19].

From an analysis of the structure of the horizontal loop device, it follows that a positive effect can be achieved by adding a feedback on a roller speed (velocity) derivative due to the damping of the elastic wave, which spreads/expands along the metal strip. The electromechanical system of the horizontal loop device is distributed; vibrations occur when it is filling up. The signal on the derivative of the roller speed damps the expansion of the elastic wave due to the effect on the electric drive. The introduction of an additional feedback circuit with respect to the derivative of the second roller speed will allow suppressing the elastic wave, which expands along the strip when moving the loop device cart. The rollers in the horizontal loop device are non-driven; it is not possible to act on them. It only can be act on the electric drive of the trolley, but information about the speed of the rollers can be received if the appropriate sensors will be installed. Since the system is distributed, it does not make sense to use the cart speed for the feedback signal, because vibrations in the mechanical part do not have any effect on the electric drive. To organize the feedback, it is proposed to measure the speed of the second roller.

Derivative velocity feedback is equivalent to the dynamic moment M_{dyn} feedback (Equation (2)):

$$M_{dyn} = J\frac{\mathrm{d}w}{\mathrm{d}t}, \tag{2}$$

where: M_{dyn} is the dynamic torque, J is the inertia torque, w is the angular velocity.

Dynamic force is distributed along the strip. Since the elastic wave moves from the roller to the carriage, then, to input the signal proportional to derivative velocity of the second roller, we provide a feed-forward control, which compensates effect of the elastic wave, which spreads/expands along the metal strip.

A corrective action, in the form of a derivative velocity signal, is formed on the basis of the signal of the second roller velocity. As block 6, is shown in Figure 5.

The structural chart of the adjusted electromechanical system is shown in Figure 5.

In the process of the simulation experiments, it was found that it is not enough to introduce only feedback on the derivative of the velocity, because during operation of the loop device, the length of the strip in the looper changes.

The moment of inertia of the second roller depends on the parameters of the roller and strip. While the mass of the roller, width and thickness of the strip remain unchanged for the same rolled strip assortment, the strip length in the loop device varies significantly. The major influence on the nature of the processes has the inertia moment. With an increase in the length of a steel strip, its inertia moment increases, its vibration frequency decreases, and the amplitude increases, this leads to additional dynamic shocks. Since the amplitude of the vibrations directly depends on the inertia moment, so adaptation is made according to the dynamic moment.

The gain of the speed controller depends on the inertia moment:

$$W_{pc}(p) = k_{pc} = \frac{J_{\Sigma} \cdot k_T}{c \cdot k_c \cdot a_{\mu c} \cdot T_{\mu c}}, \tag{3}$$

where: J_Σ is the total moment of inertia of the electric drive, kgm^2; k_T is the current feedback coefficient, V/A; $a_{\mu c}$ is the setting factor; k_c is the speed feedback coefficient, V/rad/s; c represents the electromotive intensity coefficient and electromagnetic torque of the motor at a nominal excitation flux; $T_{\mu c}$ is the total (summarized) short time constant, s; p is the Laplace operator.

The total moment of inertia depends on the engine inertia moment and the strip inertia moment. During the loop device operation, the length of the strip changes significantly. Therefore, the moment of inertia of the strip also changes.

As a result of the analysis, for different positions of the drive, a linear dependence of the feedback coefficient on the derivative of the velocity and on the length of the steel strip, which is shown in Figure 6, is revealed.

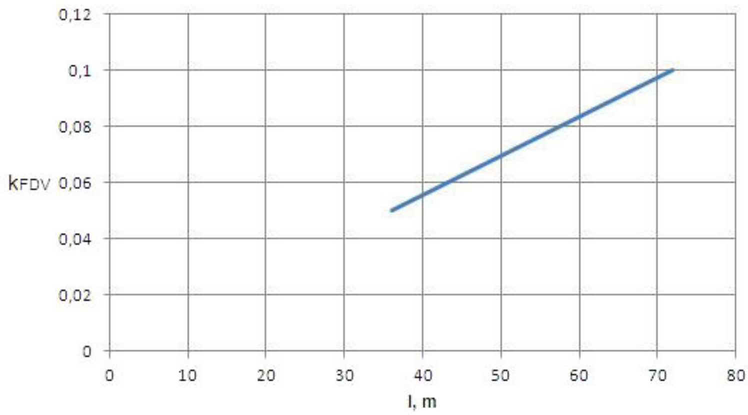


Fig. 6. Dependence of the feedback coefficient on the derivative of the velocity and on the length of the strip in the loop device

This dependence can be expressed analytically (Equation (4)):

$$k_{FDV} = k_{MCF} \cdot l, \tag{4}$$

where: k_{FDV} is the feedback coefficient of the derivative of the velocity; k_{MCF} is the feedback coefficient on the derivative of the velocity between k_{FDV} and the strip length l; l is the strip length, m.

In the study of a horizontal looper in an electromechanical system on simulation models the coefficient of the relationship $k_{MCF} = 0.0013889$ was determined. Taking it into account, the scheme in Figure 5 was added to block 5.

The new blocks and links in Figure 5 are marked in color.

Therefore, in this system, adaptation was provided by changing the inertia moment.

As a result, a link structure is proposed to take into account changes in the length of the steel strip when determining the feedback coefficient with respect to the derivative of the velocity (Figure 7).

About 20 experiments were carried out. The different operating time of the system was set: from 5 to 90 seconds. The improved performance was achieved over the entire period of time.

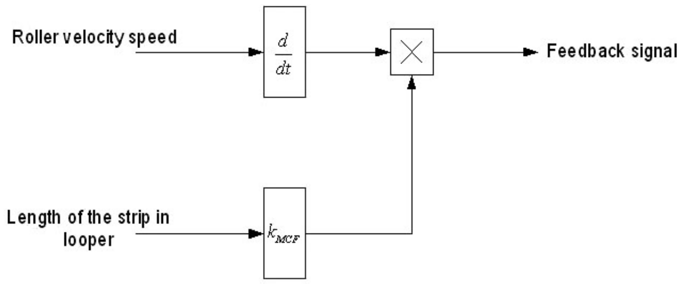
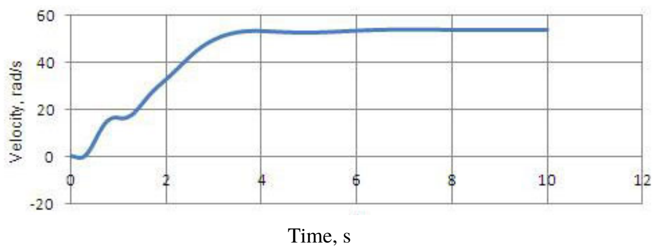
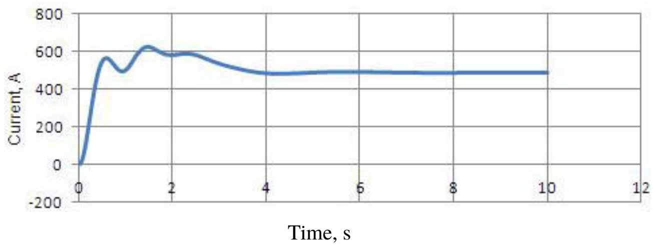


Fig. 7. Structural chart of a block that takes into account the change in the feedback coefficient with respect to the derivative of the velocity when strip length is altered

One of the experiments realizations for the adjusted automatic control system for a loop device filled to 100% is shown in Figure 8.



(a)



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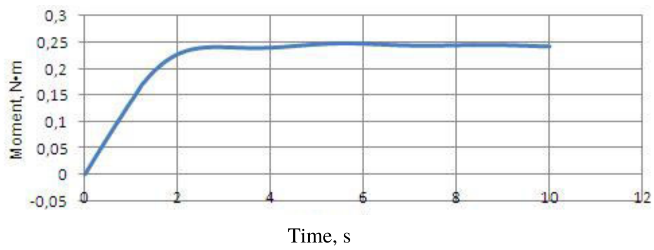

(c)

Fig. 8. Simulation results for the adjusted automatic control system

Figure 8 shows the results of the simulation experimental transients for a filled loop device in the modernized system. The graphs show the motor speed (a), motor current (b) and elastic moment (c) between roller No. 2 and the centering roller.

The amplitude of the elastic moment decreased to 0.25 Nm, i.e. 3.5 times less in response to a control action of the engine.

When comparing the transient processes of the speed and current, which are shown in Figures 3, 4 and 8, it can be seen that in Figure 8 damping appears due to the occurrence of additional fluctuations in the current and minor changes in the initial portion of the speed during acceleration. Damping occurs due to the fact that the bandwidth of the drive is expanded and it begins to damp vibrations.

Modernization is planned on the base of the existing control system.

Additionally, it is required to install a speed sensor on roller No. 2. The signal of the strip length in the loop device, which is necessary for constructing an adaptive system, can be received from the existing system of the automatically calculated strip stock in the horizontal loop device.

7. The discussion of results

Analyzes of various methods of suppressing elastic vibrations in electromechanical systems were made conducted in this article.

As a result of the experiments, significant vibrations of the metal strip were established in a horizontal loop device. This is related to a change in the strip length by 13 times during operation (from 24 580 mm to 329 480 mm), appropriate changes in the inertia coefficient could be from 1.059 to 1.163. The change of the inertia ratio significantly affects the dynamics of the mechanical part.

With an empty horizontal loop device the strip is almost rigid. Increased length of the steel strip improves its elastic properties. When the horizontal loop device is fully filled, the elastic properties of the strip have a significant effect on the entire electromechanical system. In this case some large-amplitude vibrations occur up to the adjoining strip.

An electromechanical system in a horizontal loop device has been developed as mathematical and simulation models. Synthesis of the drive control system of a horizontal loop device was made as well. The structure included a power-up sensor, adjusted devices and feedback for the derivative of the second roller velocity was proposed.

The introduction of an additional signal on the derivative of the velocity of the second roller suppresses the elastic wave, which is distributed along the metal strip in the looper.

Changing the ratio of the controller gear provides compensation of variable length of the stripes and inertia coefficient. The power-up sensor provides reduction of vibration in the strip due to the soft start. A periodic link increases the margin of control system stability.

The presented results of the simulation experiments showed that the proposed actions ensure the adaptation of the control system parameters to a variable strip length and inertia coefficient of the electric drive. These methods eliminate elastic vibrations in the electric drive elements of a horizontal looper.

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