Use of automated electric drives for limiting dynamic loads in shaft lines of roll mill stands

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Abstract: This study introduces the results of scientific research and pilot testing of modernised automated electric drives of the plate mill stands. The experimental tests of electromechanical systems of the reversing stand of the mill 5000 showed the intolerable level of dynamic loads when biting. The problem of reducing dynamic loads and elastic vibrations in shaft lines was stated; it can be solved through enhancing the automated electric drive. The method for limiting loads through biting metal with accelerated rolls and compensating the dynamic component of speed was suggested. The functional diagram of the electric drive to implement the suggested method was designed. In order to study the suggested technical solution, the dynamic mathematic model of the mill stand electric drive with elastic coupling was developed. The results of modelling dynamic modes and experimental studies carried out on the mill 5000 were introduced. The operability and efficiency of the suggested solutions were substantiated.

1 Introduction

Stated a long while ago, the problem of reducing dynamic loads in electromechanical systems of plate and broad-strip hot-rolling mill stands remains acute. Shock loads emerging when biting lead to faults of the mechanical and electric equipment, increasing maintenance costs and defect ratio. The main types of defects caused by dynamic loads in lines of primary electric drives are faults of mills, lathe spindle nose destruction, and faults of shaft bearings [1, 2].

Numerous authors devote their attention to issues related to enhancing the electric equipment and control systems of mills. Among these works, there are some introduced by major companies, such as Siemens, SMS-Demag, Voest Alpine, General Electric, and several Japanese companies [3–7]. Scientists of South-Ural and Magnitogorsk State Universities actively carry out research in this field [8–10]. The body of publications, including [11–13], are dedicated to issues related to reducing dynamic loads of the plate mill 5000 equipment.

The key device of the plate mill 5000 is the four-high stand ensuring roll force of up to 12,000 tons. This stand is one of the most powerful among the existing analogues, designed for several-pass rolling. The rolling goes lengthwise-transverse with staggered width [14]. The stand structure is described with the kinematic diagram (see Fig. 1). The stand is equipped with two working rolls transmitting the roll force and drive power to the treated metal. The electromechanical screw-down mechanism regulates the deformation zone from above, and the hydraulic screw-down mechanism from below. The drive of each working roll is provided by the synchronous motor through universal spindles.

A typical dynamic mode of reversing stand electromechanical systems is shock load impact taking place at the time of biting. The mode is accompanied by significant increase of the motor torque and elastic torque in shaft lines of the upper and lower rolls – this leads to early wear, lower process stability, and other negative consequences.

2 Problem statement

Fig. 2 shows typical oscillograph records of electric drive speeds and torques when biting. Before biting, motors rotate at threading speed, while their rolling torque is almost equal to zero and can alternate in signs. This leads to free motion in motor-to-rolls mechanical transmission. As a result, free motion when biting results in torque overcontrol (Window 2) and dynamic speed reduction (Window 1).

The features of the transient processes shown:
1. After biting, the electric drive is accelerated to the working speed for higher performance.

2. Dynamic speed reduction of the lower roll electric drive comes to 18% (from 49 down to 40 rpm). In the upper roll this parameter comes to 17%.

3. Torque overcontrol comes to 41% (max 240%, average equal to 170% of the rating).

4. In idle mode and before biting, the alternating-sign torque emerges, which is indicative of free motions in the mechanical transmission.

The following reasons of dynamic loads were found during analysing the introduced and other oscillograph records:

1. Impact torques related to free motions spindle joints cannot be maintained but can be removed by means of the electric drive.

2. Inconsistency between the sheet feeding speed and linear shaft speeds.

3. Biting conditions are inadequate as the roll nips are initially set after the pass according to the specified stock material thickness with no regard for biting conditions.

Therefore, the problem of development and industrial implementation of main electric drive control methods that are to ensure limitation of dynamic torques in shaft lines of the stand was stated. The following methods were studied during solving the problem:

- electric drive control method that ensures the take-up of clearance in spindle joints by means of biting material during roll acceleration;
- electric drive control method compensating the dynamic speed reduction.

In the first case, the preliminary take-up of clearances is conducted, as material biting takes place in the electric drive acceleration zone. The second method is featured for compensation of the electric drive dynamic speed reduction through preventive speed increase. The developed method for limiting the dynamic torque that combines advantages of two mentioned ones is described below. This integrate method is the derivative of the look-ahead electric drive speed compensation method substantiated in [15].

3 Control method development

The developed method is actually the result of combining the mentioned methods for clearance take-up in shaft lines through extra acceleration and preventive pre-biting acceleration for the dynamic deviation of regulation. Fig. 3 shows the functional diagram of the control system that implements this method. It forms the setting chart for the electric drive speed (see Fig. 3b).

![Fig. 3](image)

**Fig. 3** Functional diagram of the Developed Control System – (a) and electric drive speed chart – (b)

The automated speed setting is formed as follows. The rolling model sends the table of trajectory points to the primary-level controller. As the table is interpolated, the signal of the specified linear speed passes through the power-up sensor and is transformed into the angular speed of each electric drive with consideration of roll diameters. The power-up sensor limits the acceleration. The acceleration–deceleration pace is formed by the interpolator when executing the trajectory that is formed by the second-level model of the automated process control systems (APCS) according to mill performance criteria and acquiring the required temperature.

According to the developed method, the clearance take-up is conducted through biting the metal in the place of electric drive acceleration. The acceleration start moment is determined according to the data acquired from the system monitoring the metal while staying on roller track. The additional speed is formed by the extra power-up sensor. Its output signal is summarised with the interpolator’s output signal. The extra acceleration pace was experimentally found equal to 0.22 m/s². The diagram was executed by software in the stand controller.

The system is distinguished by the calculation and speed reduction compensation blocks. The value of compensation is determined in the second-level model according to the expected electric drive torque and speed regulation system characteristics. In order to determine the compensation signal, the experimental dependence of dynamic speed reduction on the load torque was obtained. The slope of this parameter allows for fining the dynamic rigidity coefficient of the electromechanical system.

According to the speed chart given in Fig. 3b, by the biting moment, acceleration increases the electric drive speed by the value required to compensate the dynamic deviation. For that, the power-up sensor shall have different coefficients for acceleration and deceleration modes. During acceleration, the speed increase pace is set, and after biting, the extra acceleration setting falls down to zero. The advantage of such a way is that a setting signal, when formed, can ensure that the speed increment at the time of biting could be equal to the value required to compensate the dynamic speed reduction even at maximum clearances in mechanical transmissions. It removes the necessity in forming the additional acceleration signal for clearance take-up.

In order to study the suggested control method, the mathematical model of the electromechanical system with the elastic coupling was developed; it describes dynamic processes in shaft lines of the mill 5000 stand.

4 Dual-mass electromechanical system model

As described in Fig. 1, mechanical transmissions of the stand include no devices having mass comparable with the motor inertia and added inertia of working and backup rolls. This is why, during the model development, the electromechanical roll system...
was set as the dual-mass system with elastic coupling and clearance in mechanical joints.

The structural diagram of the model can be found in Fig. 4. Blocks 3, 5–7 are typical blocks of a dual-mass system model [16]. Block 4 determines the character of transient processes in the mechanical part, including the natural oscillation attenuation; Block 5 reflects clearances in mechanical transmissions. Block 8 designs the speed feedback at \( k_{fb} \) coefficient. The following designations are used: 

\[
J_1 = J_{\text{MOTOR}} \quad \text{— moment of first mass (motor)}; \quad J_2 = \text{adduced moment of second mass inertia (working and backup rolls, connecting shafts etc.)}; \quad c_{12}, \quad M_{12} = \text{elastic coupling stiffness and torque}; \quad \beta = \text{attenuation coefficient}; \quad M_1, \quad \omega_2 = \text{dual-mass system input torque and output angular speed}; \quad p = \text{Laplace operator}.
\]

The electric drive control system is a dual circuit with the external speed control circuit. The model assumes that the internal torque control circuit is technically optimal. This is why, in order to simplify the analysis of mechanical processes, it is presented as a first-order lag block (Block 2). Speed control circuit is symmetrically optimal equipped with the proportional-integral regulator (Block 1). For the purpose of torque limitation, the limitation block with adjustable limits is connected to the output of the speed control circuit.

Parameters of all model blocks are taken according to electric equipment readings and data obtained from mill oscillograph records. Comparing the model results with experimental data, the appropriateness of the model for the object studied was proven.

### 5 Modelling the designed electric drive

During modelling, it was found that the pitfall of the suggested method was increase of load impact on the electric drive during gradual reduction of the speed setting at the time of biting. In order to remedy this disadvantage, the enhanced method for limiting the dynamic torque was developed. It was suggested to reduce the speed setting at the time of biting not abruptly (see Fig. 3b) but at the regulated pace. The control algorithm is described with curves (see Fig. 5) obtained through model calculations.

Biting takes place at \( t = 2.25 \text{ s} \). In order to compensate the dynamic speed reduction, the electric drive is further accelerated by the value of dynamic deviation (between 1 and 2.25 s — see Window 1). After biting, the deceleration goes at the set pace (with negative acceleration) changing in linear fashion. Negative acceleration values are taken as follows: \(-2.5, \ -3.5\) and \(-4.5 \text{ s}^{-2}\).

Curves 1, 2 and 3 correspond to each of them, respectively. The preliminary acceleration goes at the same acceleration values, so slopes of each curve before and after biting are the same.

Transient processes in speeds and torques at the time of executing the specified signals are demonstrated in Window 2 and 3, respectively. The technical effect is that when the electric drive decelerates, the negative dynamic torque occurs that partially compensates the dynamic torque that emerged at the time of biting. As a result, torque overcontrol (first extreme point) decreases (see Window 3). The value of the torque limitation depends on the electric drive deceleration paces. The modelling showed that negative acceleration varying from \(-2.5\) to \(-3.5 \text{ s}^{-2}\) would be optimal. In such case, speed reduction would be the least, while first and second extreme points would be equal for each chart, slightly exceeding the established value. The curves shown in Window 3 were calculated for the optimal acceleration belonging to the mentioned range.

In order to verify credibility of the obtained results, experimental studies were conducted and allowed for taking real-rolling conditions and equipment parameters into consideration.

### 6 Experimental studies

The control algorithms developed passed pilot tests on the mill 5000. Several experimental studies were carried out when rolling sheets of various materials. The mode corresponding to the considered dynamic limitation method was studied at various electric drive deceleration modes after biting. Typical oscillograph records for electric drives of the upper and lower rolls are shown in Fig. 6. There, the take-up of clearances in mechanical joints at the acceleration zone are clearly illustrated.

The take-up is accompanied by slight increases of torques and decreases of speeds (before and after \( t = 165.75 \text{ s} \)). The biting takes place at \( t = 166 \text{ s} \); after that, the pre-increased speed is being decreased at the pace of \(-2.5 \text{ s}^{-2}\).

As it appears from the oscillograph records introduced, biting takes place almost without overcontrol of electric drive torque in
both rolls. The possibility of dynamic speed reductions is eliminated.

Analyzing Fig. 6, we can prove the conclusions made based on modelling results:

1. Reduction of the speed setting after biting creates the additional negative torque that compensates the dynamic torque caused by shock load.
2. The value of the additional torque can be controlled through changing the speed reduction pace. This ensures higher efficiency of the dynamic torque limitation.
3. The credibility of conclusions was proven by results of extended experimental studies carried out on the mill.

7 Conclusion

After the studies conducted it was found that the developed method for limiting the dynamic torque by virtue of preliminary acceleration and compensation of the dynamic deviation of speed control can be reasonable for practical use. Its advantage is the compensation of the torque overcontrol at the time of biting by means of the negative dynamic torque occurring when the electric drive decelerates. The modelling proved the range of optimal negative acceleration: from $-2.5$ to $-3.5$ s$^{-2}$.

The algorithm ensuring implementation of the developed method was implemented into the mill 5000 control system software. Extended experimental studies of transient processes taking place during rolling were carried out and proved the reduction of the torque overcontrol down to 20% and speed reduction down to 5% of the established values.

The technical effect of implementing the developed method was the enhancement of the wear resistance of the mechanical and power equipment by virtue of limiting the shock loads. The economical effect is reduction of maintenance and replacement costs. Besides, the defect ratio is decreased, while general rolling quality grows. Also, condition-based maintenance becomes possible with implementation of the developed method.

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Fig. 6 Oscillograph records of biting when implementing the developed method for dynamic torque limitation through acceleration and dynamic deviation compensation

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