Mathematical Modeling of Mechatronics System "Hydraulic Screwdown Mechanism - Electric Drive of Rolling Mill Stand"

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Abstract. The article provides a rationale for the mathematical model of a mechatronics system "hydraulic screwdown mechanism - electric drive of a rolling mill stand". The authors emphasized the relevance of the research of the developed method to limit dynamic loads imposed on electromechanical system rolls in compliance with which a roll gap is increased before metal bite and roll movement to a set position during rolling. The paper presents a block diagram for a complex mathematical model including local models of a hydraulic screwdown mechanism in the automatic gage control system (AGCS) and a rolling mill stand structure interconnected by metal. Further, the setup of a system for the adjustment of the screwdown mechanism position is considered. The authors provide analytical dependences to calculate the specific roll force and mill torque. The developed method was modeled, and it was proved that its implementation results in the decrease of the dynamic torque at metal bite. The article is concluded with the recommendations on the developed model use.

1. Introduction
Plate rolling mill operation is associated with the problems of the limitation of dynamic loads when metal is bitten with the horizontal rolls of a reversing rolling stand. Such problem is characteristic of the plate rolling mill 5000 of Magnitogorsk Iron&Steel Works. The main technological device of a rolling mill is a four-high mill stand providing for a roll force up to 12,000 tons. It is the most high-duty stand among all the similar mechanisms in Europe.

The solution of the aforementioned problem is complicated by the fact that the software of the stand automatic process control system is not accessible for the settings adjustment. It limits the possibilities to conduct active experiments at the rolling mill. That is why the most efficient way to conduct the research is mathematical modeling.

A mathematical model to be developed shall have a universal nature. It shall include the data from a mechatronics system of any sheet or wide-strip mill. But the model structure shall remain the same. The modeling ultimate objective is to develop recommendations to introduce the methods aimed at the reduction of dynamic loads on the rolling mill 5000.

2. Task statement
The method developed for the reduction of dynamic loads applied to the plate rolling mill provides for an automatic gap increase before metal bite. After some pause following the metal bite a hydraulic screwdown mechanism moves into a set position determined by the reduction value. To study this method, it is necessary to develop a mathematical model for a mechatronics system "hydraulic screwdown mechanism - electric drive of a rolling mill stand in the AGCS structure" taking into account the interconnection of these mechanisms by metal. One shall also take into consideration that modeling does not cover the research of "internal" coordinates of the electric rives (currents, voltages, magnetic flow, etc.). That is why the electric rive model shall be taken as a simplified one without taking into account the peculiarities of synchronous motor vector control [1].

3. Body
The stand of the plate rolling mill 5000 is equipped with two working rolls transmitting the rolling effort and electric drive power directly to the metal being rolled. The deformation zone in the stand is formed from the top with the mechanic screwdown mechanism and from the bottom - with the hydraulic screwdown mechanism [2]. The electric drive of each roll is individual. The mechanic joints do not have the mechanisms with the inertia (mass) comparable to the motor inertia and corrected inertia of the working and support rolls. That is why the electromechanical system is adopted as a dual-mass one with the elastic constraint and gap in mechanical joints. The model block diagram is presented in Figure 1. The block 2 determines the nature of transition processes in the mechanical part including natural oscillations damping, the block 3 models the gaps in mechanical transmissions while the blocks 1,4,5 are the standard blocks of a dual-mass model of electromechanical systems [3].

![Figure 1. Block diagram of the roll electric drive dual-mass system.](image)

Such structure is a component of a comprehensive model of the studied mechatronics system the diagram of which is given in Figure 2. The electric drive control system is a dual-circuit system with an external circuit of speed control. The internal closed loop for current control is represented by the aperiodic first-order factor. The structure also includes the hydraulic screwdown mechanism model as a part of the AGCS and the model of interconnection between hydraulic and electromechanical systems by metal which is a simplified deformation zone model in terms of its nature. They are built on the basis of known analytical dependencies considered below.

**Mathematical model of hydraulic screwdown mechanisms in the AGCS structure**
The conducted analysis of literature references showed that there are several variants of the hydraulic screwdown mechanism description as a control object [4-6]. Below are the main dependences to build a model reasoned in detail in [7,8]. The screwdown mechanism as a control object is described by three linearized differential equations.

1. The fluid flow rate equation for coordination small increments:

\[
Q = S_G \frac{dS_{sd}}{dt} + \frac{V_{sd}}{E} \frac{dP_G}{dt} + rP_G
\]  

(1)

where \(Q\) – rate of the fluid flowing into a hydraulic cylinder piston cavity; \(S_{sd}\), \(S_G\) – position and area of the hydraulic cylinder piston; \(V_{sd}\) – volume of the piston cavity in the piston initial position; \(E\) – volumetric modulus of elasticity of the hydraulic fluid; \(P_G\) – pressure inside the piston cavity; \(r\) – leakage coefficient.
2. The equation describing the effort ratio inside the stand:

\[ P \cdot S_G = P + m \frac{d^2 S_{ad}}{dt^2} + b \frac{dS_{ad}}{dt} \]  \hspace{1cm} (2)

where \( P \) – full roll force; \( m \) – mass of the stand mechanical parts moving together with a hydraulic cylinder as well as the mass of the fluid in the pipeline running from the battery to the hydraulic cylinder reduced to the piston area; \( b \) – viscous friction coefficient determining the value of the friction force proportionate to the hydraulic screwdown mechanism travel speed.

3. The linearized equation of the servo-motored valve for coordinate small increments:

\[ T_x \frac{dQ}{dt} + Q = k_y u_y - k_{yu} P \]  \hspace{1cm} (3)
where $T_y$ – servo-motored valve time constant; $u_y$ – control voltage; $k_y = \frac{dQ}{du_y}$ – constant flow rate coefficient of the servo-motored valve; $k_p = \left(\frac{dQ}{dm}/0\right)$ – stiffness coefficient of difference flow characteristics at the pressure $(P_0)$ inside the piston cavity taken as the initial one.

The given equations (1)-(3) do not take into account:
- wave processes in the pipeline connecting the battery, servo-motored valve and hydraulic cylinder;
- pressure loss in the pipeline;
- dry friction.

The transmitting function of the screwdown mechanism position adjuster at the technical optimum setting [9]:

$$ W_{opt}(p) = \frac{1}{W_{op}(p) k_y k_{opt} 2T_y p} $$

where $k_{opt} = \frac{U_{sp max}}{S_{sd max}}$ – feedback coefficient on the hydraulic cylinder piston position; $W_{opt}(p)$ – transmitting function of the hydraulic screwdown mechanism as a control object.

If we ignore the high-order values in the transmitting control function, we can apply a proportional screwdown mechanism position adjuster with the amplification coefficient $k_{prg}$. Figure 3a, presents logarithmic amplitude frequency characteristics (LAFC) of the open and closed circuits of the position adjuster at technical optimum setting. They are built for the proportional adjuster with the coefficient $k_{prg} = 95$. The curves of the transition process for the screwdown mechanism position at the change of the task with $k_{prg} = 95$ are given in Figure 3b.

At such setting, the screwdown mechanism position re-adjustment is equal to 30%. At the same time, in compliance with [10], the re-adjustment value shall not exceed 5%. To fulfill this condition, the amplification coefficient is taken as $k_{prg} = 65$. The corresponding LAFC of the open and closed circuits of the position adjustment and the transition process graph are also given in Figure 3a and Figure 3b. In this case, the setting provides for the fulfillment of the set requirement.

Hydraulic screwdown mechanisms control is effected with the help of an indirect system regulating the sheet (plate) gage built on the basis of the Golovin-Sims equation [11]:

$$ H = d + \frac{P}{M_K} $$

where $H$ – thickness of the plate at the stand outlet; $d$ – roll gap size; $P$ – roll force; $M_K$ – stand stiffness modulus.

Based upon this expression, the AGCS block diagram is built in the model in Figure 2.

**Mathematical model of rolling in the deformation zone**

At the mathematical model development the authors assumed the following:
- there is no flare at rolling;
- initial rolling gage at the stand inlet does not change within the analyzed time period;
- distribution of the metal delivery rate is equal along the whole section depth.
Figure 3. LAFC of the open and closed circuits of the sceredown mechanism position adjustment (a) and the graph of the transition process at the sceredown mechanism position change (b) at $k_{prg} = 95$ and at $k_{prg} = 65$. 
The rolled sheet as a control object is described by the known equations. To determine the forward flow at rolling, the authors use the Dresden formula [12]. To calculate the roll force, the authors use the expression proposed by A.I. Tselikov [13]:

\[ P = \frac{B \cdot X \cdot H}{2\mu} \left[ \frac{1}{\phi} \left( \frac{X_{i+1}}{X_i} \right) e^\phi - \left( \frac{X_{i+1}}{X_i} - 1 \right) - \frac{1}{\phi} \right] \]  

(4)

where \( B \) – sheet gage; \( \phi = \frac{h}{h + H} \); \( m = \frac{2\mu \cdot l}{h + H} \); \( X_i = 1.15\sigma_i - \sigma_i \); \( X_{i+1} = 1.15\sigma_i - \sigma_i \) – coefficients; \( \sigma_i \) – metal yield strength before and after reduction in the stand (at loose rolling inside a reversing rolling stand these values are taken as equal ones, that is why they are not taken into account in Figure 2 [14,15]); \( \mu \) - friction coefficient between the working surface of the rolls and metal; \( l = \sqrt{R(h - H)} \) – length of the arc of roll contact with the account of roll flattening.

The model block diagram in Figure 2 is made with the use of the presented mathematical apparatus. In the model structure the dependences (4), (5) are implemented in the functional blocks \( P = f(h, H) \) and \( M_s = f(h, H, V_{pr}, P) \).

4. Modeling results

The developed model was used for the research of the method proposed for dynamic loads limiting. In compliance with the research programme, the gap before metal bite was made larger than it is required to obtain a target gage and was further decreased to the set value as the head part enters the stand. It was assumed that the load torque at metal bite changes abruptly.

The Figure 4 gives time dependences for the electric drive speed and torque change. Metal is bitten at the torque of 2.25 c with the increase of the roll gap and at different initial reductions: curves 1 – at 70%, curves 2 – at 80% and curves 3 – at 90% of nominal reduction. In 0.25 c after that, within 2.5–3 sec. the linear gap decrease occurs due to the hydraulic screwdown mechanisms movement. Accordingly, under the linear law the static load torque increases (window 2). Meanwhile, the mill
torque reaches the set value while the gap reaches its set size corresponding to the sheet gage after one pass.

The analysis of the presented graphs demonstrated that in case of metal bite at the preliminary roll convergence at 90% of the required roll gap, the torque re-adjustment (curve 3) is equal to 50%. At the graphs 1 and 2 the torque re-adjustment is equal to 13% and 27%, correspondingly. Consequently, the effect of torque fluctuation limiting depends on the initial roll setting.

Figure 4. Speed and dynamic torque transition processes for various gaps at metal bite.
5. Discussion
The modeling results confirm that the proposed method implementation provides for the dynamic torque decrease. The limitation is the roll gap size set before metal bite. The effective torque decrease can be reached if the roll set within the 70% of the set reduction (curve 2 in Figure 4). However, under such conditions it is necessary to significantly move the rolls under the load; such movement can cause section deformation in the rolling head part. Besides, at heavy reduction the load onto a hydraulic screwdown mechanism will significantly increase.

It is clear that this method application at the rolling mill 5000 will not always provide for the expected results. It is explained by a number of reasons, in particular, by the fact that a deformation zone filling at metal bite does not occur immediately as it was assumed at modeling but takes several seconds. That is why roll force changes in accordance with a complex law, therefore, the transition processes will be different from the presented ones. In this connection, it is necessary to conduct additional research using both mathematical modeling and direct studies on the object.

6. Conclusion
As a result of the conducted studies the authors obtained a relatively simple mathematical model of a mechatronics system "electric drive of a rolling mill stand-hydraulic screwdown mechanism in the AGCS structure" containing the models of the following devices and systems:
   - dual-mass electromechanical system of the main electric drive line;
   - hydraulic screwdown mechanism;
   - systems adjusting the screwdown mechanism position with a proportional position adjuster;
   - systems of gage indirect control based on the Golovin-Sims equations;
   - interconnection of electric and hydraulic drives by the treated metal.

The model is implemented as a programme in the graphical environment of imitation modeling of dynamic systems Simulink. In addition, the authors developed a special programme to calculate roll force and mill torque dependences. It allows for the calculation of mathematical model coefficients for various rolling modes applied at the mill in practice.

The authors modeled the developed method for dynamic torque limitation due to the increase of the roll gap before metal bite with its subsequent return into the set position during the rolling process. It was demonstrated that the proposed method provides a solution for the set tasks under the ideal bite conditions. However, its practical application under the conditions of the rolling mill operation needs to be further investigated.

The as-is model is suitable for the research of any modes associated with roll gap change during rolling, in particular to study the rolling process with contouring of wide rolling margins by horizontal rolls [16]. After proper follow-up revision such model may be applied to study the modes of mechatronics systems in hot continuous rolling mills [17]. For this purpose it shall be completed with a dynamic plate model in the interstad space [18,19].

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