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SIMULATION MODEL OF THE FINISHING MILL HYDRAULIC GAP CONTROL SYSTEM

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Abstract

This paper deals with a hydraulic gap control system modelling of four-high finishing mill. Control system model has been worked out on the basis of simplified dynamical models of respective system elements. In order to estimate steady state control quality in case of low mill loading, the nonlinear mill deformation model has been introduced. The simulation tests for different rolling parameters have been presented.

Key words: hydraulic gap control, mill deformation, observer

1. INTRODUCTION

To assure good working parameters in modern mill equipment the Hydraulic Gauge Control System is used. The task of this system covers fast rolls adjustment as well as side effect compensation of mill deformation when plate is rolled.

One of the main plate quality parameters, a thickness, depends on operating accuracy of roll position control system. The control of thickness is especially important in last pass, where the loading forces are relatively low and nonlinear behaviour of mill deformation appears [Roberts 1988, Guo 1991, Ginzburg 2000]. Modelling of features of such system aids to evaluate and test innovated solutions for product quality improvement.

To evaluate the plate thickness quality performance, the finishing mill HGC system has been modelled and then implemented by means of Matlab/Simulink software. The dynamical behaviour of the system has secondary significance for steady state accuracy, therefore dynamical models of mill stand elements were simplified by linear equations. To assure a stability of the model the root loci method has been used. The main usage of the presented model mainly covers steady state control error estimation due to maladjustment of mill deformation model in control system and real mill deformation curve.

2. THICKNESS CONTROL BY MEANS OF HGC SYSTEM

The basic diagram of hydraulic system for roll gap control is presented in figure 1. Because during hot rolling process direct measurement of roll gap for the feedback purposes is impossible, therefore in HGC system plate thickness has to be calculated by gaugemeter equation:

$$h_2 = s_0 + \Delta s_k \tag{1}$$

where:
h_2 – exit thickness,
s_0 – no-load roll gap,
Δs_k – mill deformation.

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For thickness estimation in equation (1) the total mill deformation value in operating point is necessary. The mill deformation observer on the basis of force signals from transducers estimates this deformation. The parameters of observer are estimated in off-line mode. Because the control system must work quite fast, the linear form of gaugemeter equation is implemented:

$$h_2 = s_0^{'} + \frac{F}{K} \tag{2}$$

where:
F – rolling force,
K – mill modulus,
$s_0^{'}$ – no-load roll gap for linear gaugemeter equation.

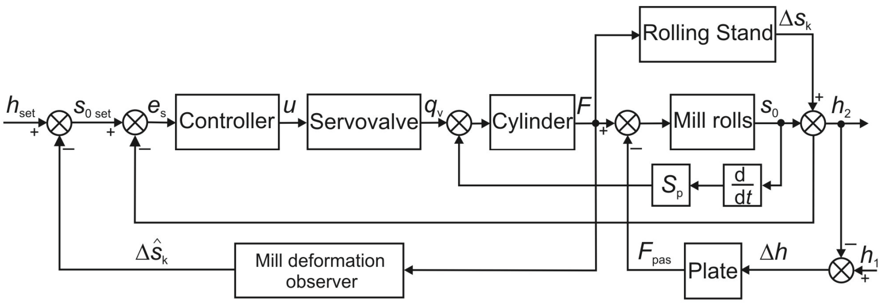

Figure 1. Block diagram of hydraulic gap control system with mill deformation observer

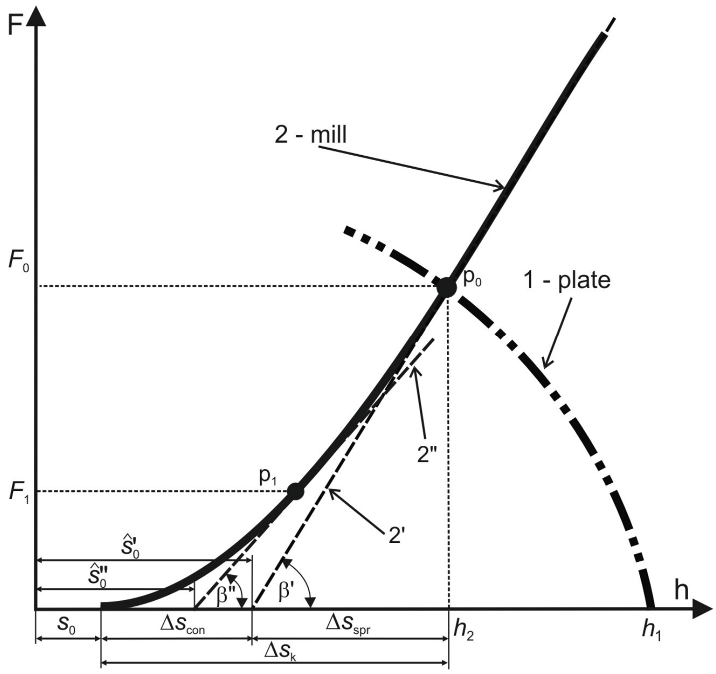

Figure 2. The gaugemeter equation method for plate thickness estimation in case of range of low forces

The graphical interpretation of expression (2) is presented in figure 2. In example, we have mill deformation curve linearly interpolated using K factor. The line 2' represents case of higher mill loading when $K = \mathrm{tg}(\beta')$ estimates correctly mill deformation in point p_0. It is easy to notice that for greater forces (above p_0) mill behavior is also well approximated by this line. A different situation takes place at p_1. In this case, line 2" ($K = \mathrm{tg}(\beta'')$) gives better interpolation efficiency than line 2'. Using line 2' for mill deformation estimation at p_1 we get steady state error. This problem occurs always if the mill deformation curve is nonlinear. In the last pass when operating points are located in range of low rolling forces linear interpolation may be insufficiently accurate to achieve high-end thickness quality. To avoid malfunction of control systems the minimal value of operating rolling force is restricted by system vendor (e.g. four-high finishing mill in "Huta Częstochowa" F_{min} = 10 MN). Taking into account force limit, system must recalculate pass schedule to assure correct settings for deformation observer in last pass. Unfortunately this strategy sometimes does not makes possible to predict correct working point and does not guarantee thickness quality of plates at level determined in standards [Jakubiec 2006].

3. MODEL OF HGC SYSTEM

Simplifying an analysis of dynamical behaviour of rolling stand as well as servovalve-hydraulic cylinder the linearized form of equations has been assumed [Pizoń 1995]. In order to describe the model the Laplace transformation has been used [Kaczorek 1977].

3.1. Controller

In control system a proportional controller is applied (fig. 1). Its transfer function can be written as [Huzyak 1984, Ginzburg 2000]:

$$G_R(s) = \frac{U(s)}{E_s(s)} = K_R \tag{3}$$

where:
U – valve control voltage,
E_s – control error,
K_R – controller gain.

3.2. Servovalve

General form of servovalve transfer function can be formulated as [Pizoń 1995]:

$$G_{sv}(s) = \frac{Q_v(s)}{U(s)} = \frac{K_v}{T_n{}^2 s^2 + 2\xi T_n s + 1} \tag{4}$$

$$T_n = \frac{1}{\omega_n} \tag{5}$$

K_v – gain coefficient of hydraulic system,
T_n – time-constant,
f_n=1/ω_n – natural frequency,
ξ – relative damping coefficient.

Such description of electro-hydraulic amplifier can be used above frequency 270 Hz. For lower frequencies (to 50 Hz) the first-order transfer function approximates model sufficiently [Thayer 1965, Huzyak 1984, Ginzburg 1984]:

$$G_{sv}(s) = \frac{Q_v(s)}{U(s)} = \frac{K_v}{T_v s + 1} \tag{6}$$

T_v – servovalve time-constant.

3.3. Hydraulic cylinder

On the basis of flow rate balance equation of servovalve system, considering the hydraulic force F as directly proportional to the pressure in hydraulic cylinder and the speed of piston v_c is its displacement derivative, the transfer function of hydraulic cylinder can be determined:

$$G_{cyl}(s) = \frac{F(s)}{Q_v(s) + S_p V_c(s)} = \frac{S_p E}{V}\frac{1}{s} \tag{7}$$

S_p – piston area,
V – cylinder volume,
E – oil compression modulus.

3.4. Strip

Considering forces which act on the strip during the rolling process (the force F caused by piston displacement counteracts the force F_{pas} generated by the strip caused by pass reduction Δh) the transfer function of the strip G_{pas} has been derived:

$$G_{pas}(s) = \frac{F_{pas}(s)}{\Delta H(s)} = M\left(T_{pas}s + 1\right) \tag{8}$$

M – strip modulus,
B_{pas} – strip damping,

$T_{pas} = \dfrac{B_{pas}}{M}$ – time constant.

3.5. Rolling mill

Roll displacement equation determines the transfer function of the rolls in form:

$$G_w(s) = \frac{S_0(s)}{F(s) - F_{pas}(s)} = -\frac{1}{m_w}\cdot\frac{1}{s^2} \tag{9}$$

m_w – mass of rolls.

For linear model of rolling mill every single deformation in the mill can be considered additively for each value of the load force. So the sum of these deformations equals:

$$\Delta s_k(t) = \frac{F(t)}{K} = \sum_{i=1}^{n}\Delta s_i = F(t)\sum_{i=1}^{n}K_{ki} = F(t)K_k \tag{10}$$

K is a resultant mill modulus, K_k is an inverse of resultant of mill modulus.

On the basis of equation (10) the transfer function of the mill can be written as:

$$G_{kl}(s) = \frac{\Delta S_k(s)}{F(s)} = K_k \tag{11}$$

3.6. Deformation observer

A deformation observer applied in HGC system is considered in linear form. After correction of roll position set up value the new set up point is calculated from equation:

$$s_{0set} = h_{set} - \Delta\hat{s}_k \tag{12}$$

where observer signal is given by:

$$\Delta\hat{s}_k = F\hat{K}_k + \hat{s}_0 \tag{13}$$

4. MODEL STABILITY

In order to determine validity of the model presented in figure 1 its stability has been tested. In the first step, the stability of the controlled plant i.e. the rolling mill has been estimated. In the next step, the gain value has been adjusted for the modelled structure of the HGC system.

4.1. Mill stability

As the most important effect in HGC control system should be the exit thickness convergence to the set up value. This convergence must be maintained despite of entry thickness change during the rolling process. The exit thickness can be expressed in form [Jakubiec 2005]:

$$H_2(s) = -F(s)\frac{K_k M T_{pas} s + M K_k + 1}{m_w s^2 + M T_{pas} s + M} + H_2(s)\frac{M T_{pas} s + M}{m_w s^2 + M T_{pas} s + M} + F(s) K_k \quad (14)$$

Using the end-value theorem [Kaczorek 1977] and introducing steady state values in operating point p_0 the steady state value of the exit thickness can be calculated as:

$$\lim_{t\to\infty} h_2(t) = \lim_{s\to 0}\left(-\frac{1}{s} F_{(p0)} s \frac{K_k M T_{pas} s + M K_k + 1}{m_w s^2 + M T_{pas} s + M} + \frac{1}{s} H_{1(p0)} s \frac{M T_{pas} s + M}{m_w s^2 + M T_{pas} s + M} + \frac{1}{s} s F_{(p0)} K_k \right) \quad (15)$$

Equation (15) is convergent to the steady state thickness value denoted as $H_{2(p0)}$, which is determined by expression:

$$H_{2(p0)} = F_{(p0)}\left(-\frac{M K_k + 1}{M} \right) + H_{1(p0)} + F_{(p0)} K_k \quad (16)$$

after rearranging the (16) we have:

$$F_{(p0)} = M\left(H_{1(p0)} - H_{2(p0)} \right) \quad (17)$$

Considering relationships (1), (10) and (16) we also get:

$$F_{(p0)} = K\left(H_{2(p0)} - S_{0(p0)} \right) \quad (18)$$

Equations (17) and (18) determine the equilibrium point of forces as well as the level of mill deformation for these forces. They prove the validity of the presented model.

4.2. Stability of the HGC system

The stability test allows to adjust gain value for the HGC controller. This analysis has been made using root locus method [Pełczewski 1980]. It is also a very effective tool for stability determination for given parameters M and K_k as well as for adjusting the controller gain. Distribution of poles of closed loop system, presented in figure 3, gives information about transient behaviour of the HGC system.

5. SIMULATION MODEL OF HGC SYSTEM

Model implementation of hydraulic roll gap control system with linear deformation observer by means of Matlab/Simulink software has been presented in figure 4. The simulation parameters were assigned on the basis of technical data of the control system [Clecim 1993] as well as of process values occurred during the last pass: S_p=0,98 [m^2], V=0,118 [m^3], T_{pas}=0,01 [s], K_k=2,35·10^{-10} [m/N], m_w=117·10^3 [kg], M=1,01·10^{10} [N/m], E=2,0·10^9 [N/m^2]. These data were derived from four high finishing mill 3600 mm placed in Huta Częstochowa S.A. (working roll diameter: 1000 mm, support roll diameter: 1800 mm, maximal allowed force: 60 MN, operational thickness range: 5-40 mm).

In order to test properties of the HGC system, the nonlinear characteristics of mill deformation have been implemented (in block rolling stand model – fig. 4). These characteristics (further on, for given plate width, called the base characteristics) were modelled on the basis of calibration curve of the real plant [Dyja 1993] – figure 5. The change effect of the rolls and frame elastic deformation as well as

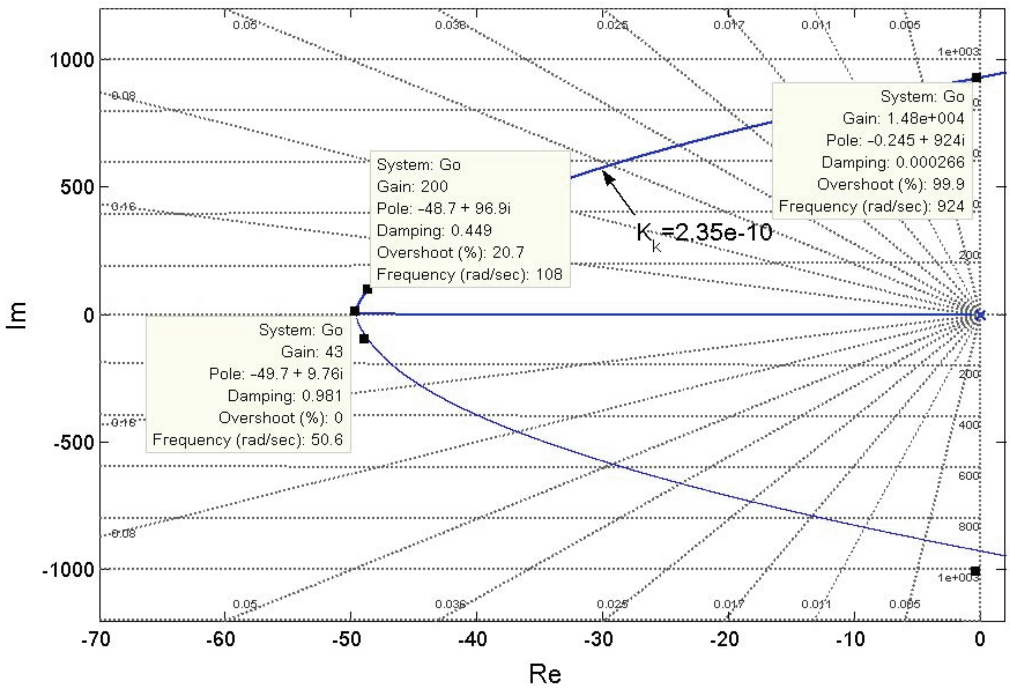

Figure 3. Distribution of poles for HGC system as a function of rolling controller gain

the change of the rolls diameter have been modelled by changing the slope and offset of the base characteristic. In the real system it corresponds to changing the width of the plate and temperature of rolls respectively.

On the basis of the force measurement the linear interpolation on the base characteristic has been made. In this way we get parameters for linear observer. Different conditions of rolling process causes adaptation of $\hat{K}_k$ and $\hat{s}_0$, in order to get minimal static control error.

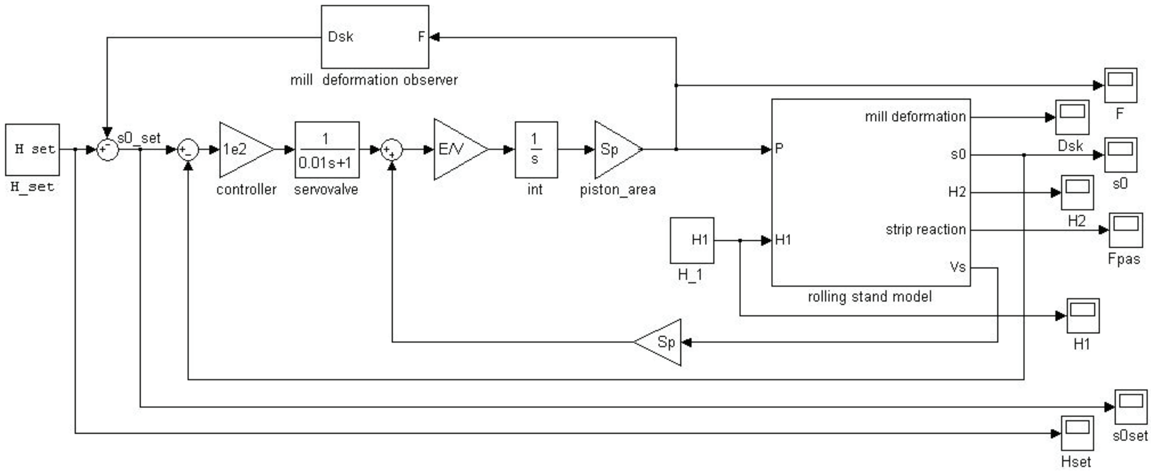

Figure 4. Simulation model of the HGC system with linear mill deformation observer implemented in Matlab/Simulink

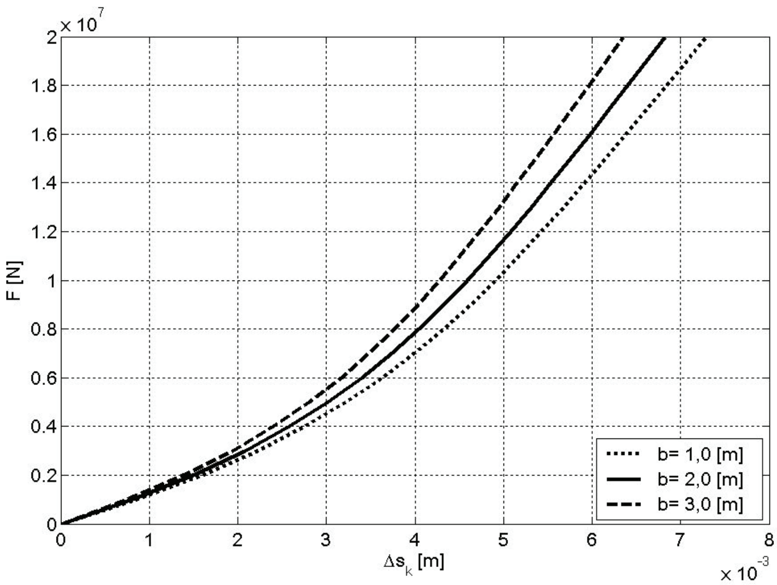

Figure 5. Base characteristics of mill deformation for different width values

6. SIMULATION TESTS

The correction efficiency of the linear deformation observer has been tested in two cases. The first one, the observer was adjusted to the operational point of the base characteristic. In the second case, for the same observer parameters, when the deformation curve has been modified due to change of rolling conditions. All tests have been made for sequential change of entry thickness and set up value of thickness. Time intervals of changing rolling parameters have been modified in simulation in order to get only steady state of the process, so the presented simulation time is shorter than period of real process.

The characteristics of the observer used in simulation tests have been presented in figure 6. In simulation No. 1 observer parameters were set to 1 mm thickness reduction at force 9.30 MN. Linearization at the working point for constant thickness reduction assures thickness control without error (simulation time periods: 0,5÷1 s, 1,5÷2 s, 2,5÷3 s, fig. 7ure -8). Out of this point, an error rises as much as the force differs from force 9.30 MN. In the next test the change of rolling conditions has been made by modification of the base characteristic. On the basis of testing results presented in [Pichler 2002] the mill stiffness has been modified by 10 %. Observer parameters were unchanged (fig. 6). As we can see in figures 10-11 for the same set up parameters control process is not efficient as in previous simulation. The time intervals 0÷0.5 s and 3÷3.5 without error are in fact a side effect of the crossing nonlinear and linear mill deformation characteristics (fig. 9). It is also presented situation when observer is adjusted correctly (simulations 1 and 3). We can notice, that slope change of the base characteristic must be corrected by adjustment of two linear observer parameters.

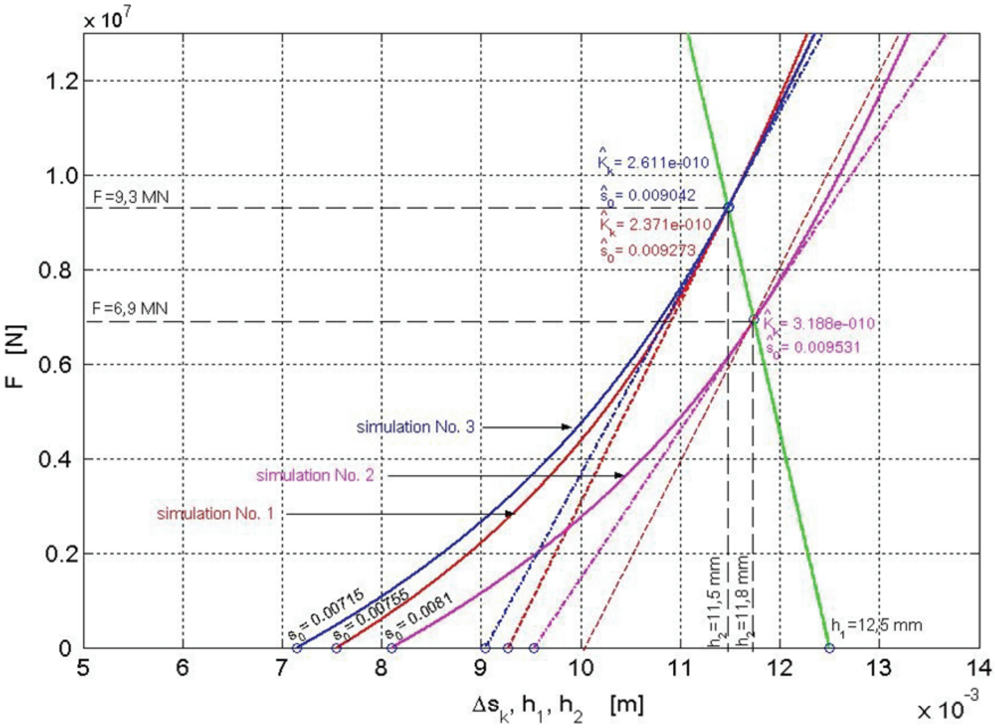

Figure 6. Linear observer and plate deformation characteristics used in simulations No.1÷3

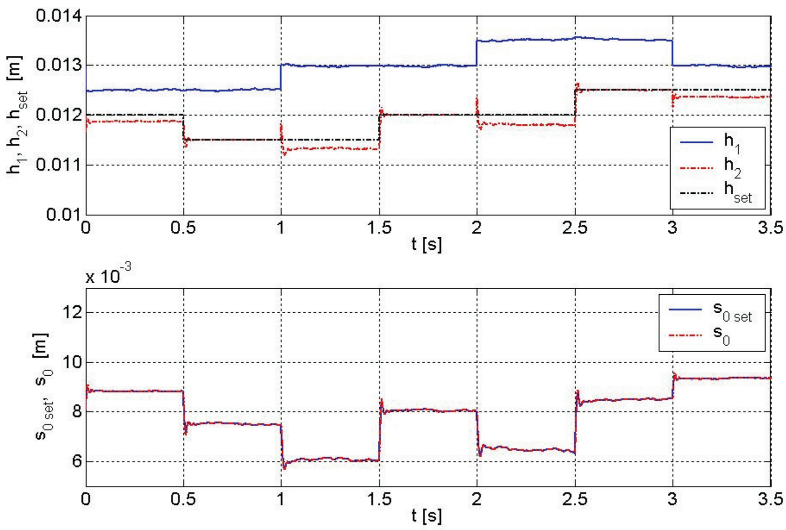

Figure 7. Exit plate thickness, estimation of piston set up value and piston displacement for sequential change of set up value of thickness and entry thickness – simulation No. 1

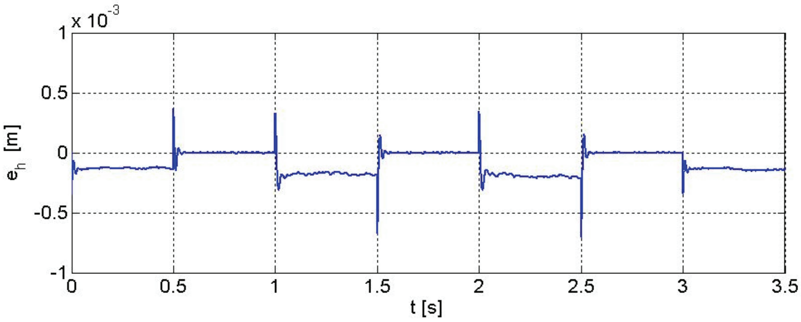

Figure 8. Error of thickness control – simulation No. 1

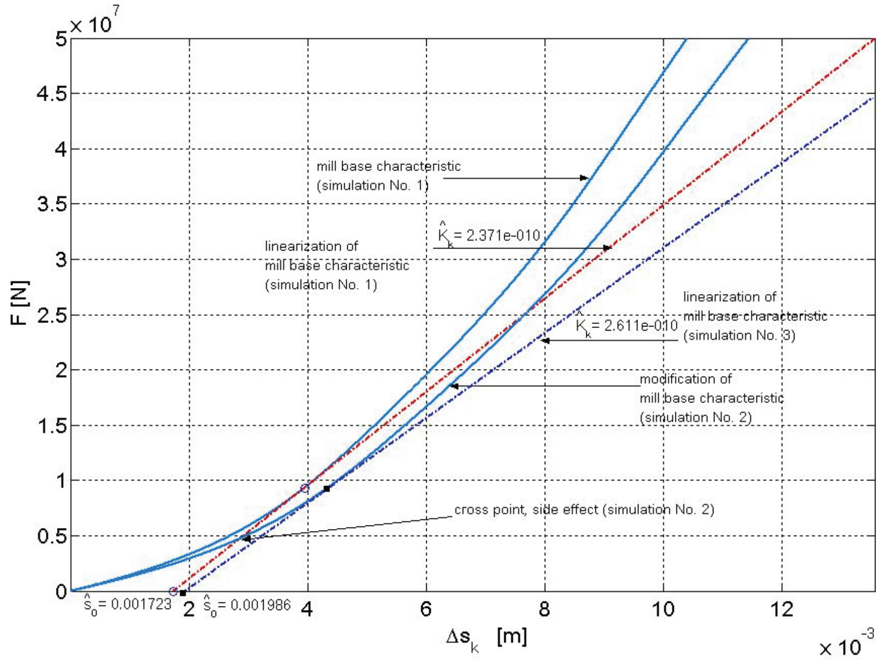

Figure 9. Base and modified mill deformation characteristic for simulations No. 1 and No. 2

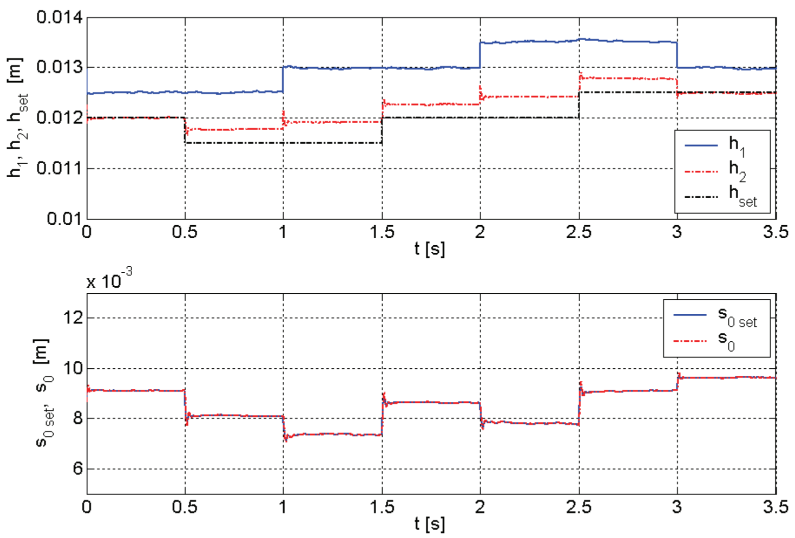

Figure 10. Exit plate thickness, estimation of piston set up value and piston displacement for sequential change of set up value of thickness and entry thickness – simulation No. 2

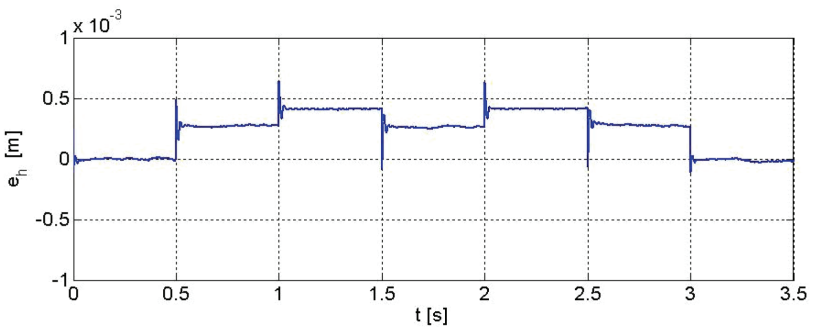

Figure 11. Error of thickness control – simulation No. 2

Figures 12-13 show results of the third simulation. In this case, observer parameters have been set correctly according to the base characteristic from simulation No. 2. For time intervals 0,5÷1 s, 1,5÷2 s, 2,5÷3 s control error equals zero like in simulation No. 1. It means good estimation of rolling mill deformation only in working point. For the rest time intervals linear observer estimates this deformation improperly.

7. CONCLUSIONS

To maintain the repeatable and good thickness quality, the HGC have to ensure accurate compensation of mill deformation. Hence the very important part of system lies in mill deformation observer. The thickness quality depends mainly on control system behaviour in last pass, in low range of forces. In this range nonlinearly of mill deformation appears and the linear approximation of this deformation in control system can not be estimated accurately.

The mill calibration curve taken from plant was used as the model of nonlinear behaviour of rolling mill. Test shows that for low forces, when linear observer is used, control quality strongly depends on operation point. The thickness errors can be even up to 300 μm if improper adjustment of parameters for linear observer is set. In this case a linear mill deformation model is strongly inaccurate and is only valid for linear part of deformation curve.

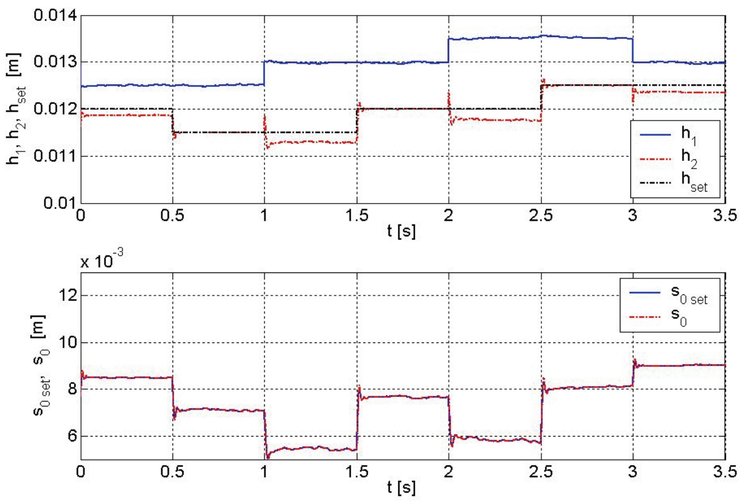

Figure 12. Exit plate thickness, estimation of piston set up value and piston displacement for sequential change of set up value of thickness and entry thickness – simulation No. 3

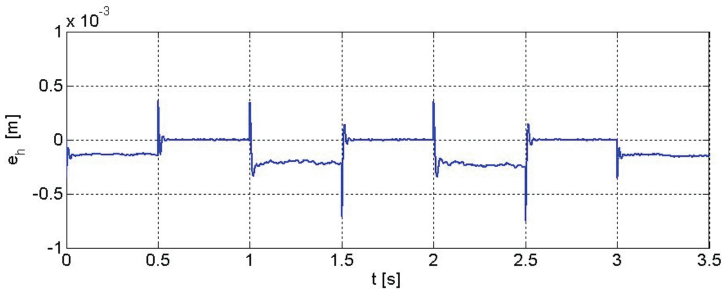

Figure 13. Error of thickness control – simulation No. 3

In real plant deformation characteristics strongly depend on rolling parameters. For this reason the set up thickness correction provided by deformation observer should be realized by fast nonlinear system, which would be able estimate compensation ratio on the basis of these parameters. Presented model of HGC system can be useful in developing and testing the new optimal structure nonlinear observer, to ensure good quality indexes in wider range of rolling forces.

REFERENCES

Clecim, 1993, *Technical documentation*.

Dyja, H., 1993, *Badanie naprężeń stojaka, sztywności klatki i ocena możliwości zwiększenia obciążenia klatki wykańczającej wydziału walcowni blach grubych*. Politechnika Częstochowska, Katedra Przeróbki Plastycznej Metali, (in Polish).

Ginzburg, V.B., 1984, Dynamic characteristics of automatic gage control system with hydraulic actuators, *Iron and Steel Engineer*, 57-65.

Ginzburg, V.B., Ballas, R., 2000, *Flat rolling fundamentals*, Marcel Dekker, New York.

Guo, R.M., 1991, Evaluation of dynamic characteristics of HAGC system, *Iron and Steel Engineer*, 52-61.

Guo, R.M., 1994, Material damping effect in cold rolling process, *Iron and Steel Engineer*, 28-35.

Huzyak, P., Gerber, T.L., 1984, Design and application of hydraulic gap control systems, *Iron and Steel Engineer*, 13-20.

Jakubiec, B., 2005, *Poprawa dokładności grubości walcowanych blach z zastosowaniem neuro-rozmytego układu nastawy szczeliny walców*, PhD thesis, Politechnika Częstochowska (in Polish).

Jakubiec, B., 2006, Statistical analysis of the thickness quality of hot-rolled plates, *14th Int. Conf. Production and Management in the Steel Industry*, Szczyrk, 166-171 (in Polish).

Kaczorek, T., 1977, *Teoria układów regulacji automatycznej*, WNT, Warszawa (in Polish).

Pełczewski, W., 1980, *Teoria sterowania*, WNT, Warszawa (in Polish).

Pichler, R., Beaumont, J., 2002, Advanced automation systems for plate mills, *Millenium Steel, Forming Processes*, 179-183.

Pizoń, A., 1995, *Elektrohydrauliczne analogowe i cyfrowe układy automatyki*, WNT, Warszawa (in Polish).

Roberts, W.L., 1988, *Flat Processing of Steel*, Marcel Dekker, New York.

Thayer, W.J., 1965, Transfer Functions for Moog Servovalves, *MOOG Inc. Technical Bulletin*, 103.

MODEL SYMULACYJNY UKŁADU HYDRAULICZNEJ NASTAWY WALCÓW WALCARKI WYKAŃCZAJĄCEJ

Streszczenie

Artykuł opisuje model symulacyjny hydraulicznego układu sterowania nastawy szczeliny między walcami dla walcarki wykańczającej typu kwarto. Model systemu sterowania został opracowany na podstawie uproszczonych modeli dynamicznych elementów systemu walcowania blach grubych. W celu oceny jakości sterowania w stanie ustalonym dla małych wartości obciążeń wprowadzono nieliniowy model odkształcenia klatki walcowniczej. Badania symulacyjne przeprowadzono dla różnych warunków pracy klatki.

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