Adaptive Automatic Gauge Control of a Cold Strip Rolling Process

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***Abstract*—The paper tackles with thickness control structure of the cold rolled strips. This structure is based on the rolls position control of a reversible quarto rolling mill. The main feature of the system proposed in the paper consists in the compensation of the errors introduced by the deficient dynamics of the hydraulic servo-system used for the rolls positioning, by means of a dynamic compensator that approximates the inverse system of the servo-system. Because the servo-system is considered variant over time, an on-line identification of the servo-system and parameter adapting of the compensator are achieved. The results obtained by numerical simulation are presented together with the data taken from real process. These results illustrate the efficiency of the proposed solutions.**

***Index Terms*—Automatic gauge control, adaptive control, inverse model, adaptive filtering, optimization**

I. INTRODUCTION

The economic efficiency of metal rolling processes is strongly correlated to the quality level of end-rolled products. The latest efforts to increase the quality of the end-rolled products in rolling processes have been mainly focused on the large-scale application and use of automation control advanced methods. This orientation is reflected in field literature by numerous approaches of rolling processes control. Thus, in [9] and [10], the predictive control is used for the hot rolled process, in [2] decentralized - coordinates control methods are used, in [1] and [4] – adaptive and robust control methods and in [5], Hinf control techniques are currently used. Artificial intelligence methods based on the inferential control [3], artificial neural networks [8] and neuro-fuzzy techniques [6] are also used. In the synthesis study [7] the main control methodologies recently explored, starting with the conventional controllers, up to predictive control, observer-based control, optimal control, internal model control (IMC), adaptive and robust control techniques, Hinf control and up to artificial intelligence techniques are briefly illustrated.

The main type of the rolling plant approached in the present paper is the one of the tandem mill type that consists in a group of 4-7 non-reversible stands with quarto structure, which is specific to high capacity lines. In this case, the process control is based, in a decisive measure, on the tension control in rolled strip, with the help of the loopers [7].

A special category of rolling processes is the one used for the production of very thin strips, where the uniformity of the thickness of the final product is very important. The rolling mills of this type contain only one reversible quarto structure stand and they are called “Quarto Mills”, generally of smaller capacity.

In this case, the process consists in a “rolling passes” sequence, with gradual decreasing of the gap between rolls for each pass and changing the rolling strip movement direction. Fig. 1 illustrates the simplified scheme of quarto reversible rolling mill, where: WR represents the working rolls, SR – support rolls, W – winding, U – unwinding, SH – hydraulic servo-system for rolls positioning and AGC – automatic gauge control system.

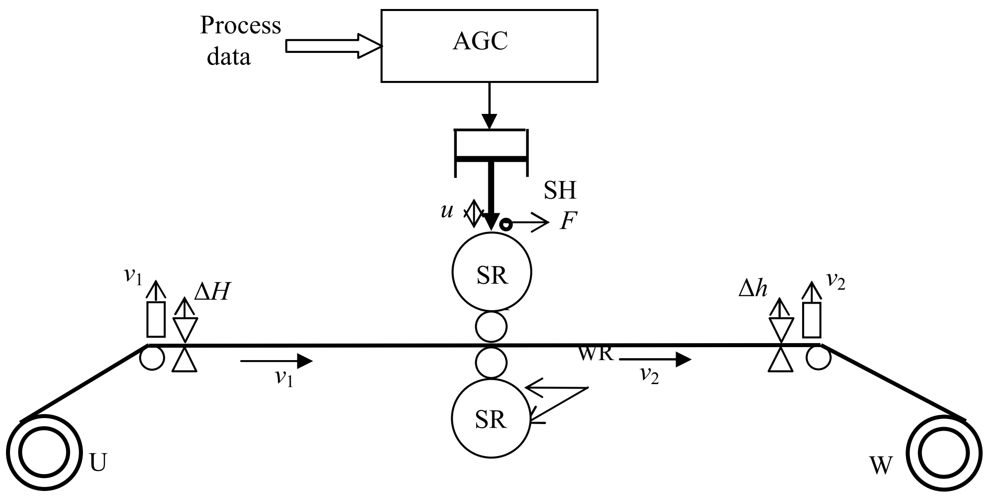

Figure 1. Quarto rolling mill structure.

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Each pass through this mill is similar to the rolling through a tandem rolling mill stand. The absence of the looper in the case of quarto rolling mill leads to significant particularities in the control strategy.

Let us consider H^*, the nominal value of the strip thickness at the rolling plant input. The real value of this thickness, (H), is different from the nominal one, due to the imperfection of the strip that is currently produced. Let us also consider $\Delta H = H^* - H$, the thickness error at the input of the rolling mill. The goal of the process control system is to obtain a thickness error of the strip (h) at the smallest output possible, with respect to the set point h^*.

The aim of AGC system is to minimize the thickness error variance, $\Delta h = h^* - h$. This objective can be attained by using two types of controls:

- by position control, that is by way of the modification of the distance u among the working rolls, using the hydraulic servo-system in closed loop (SH);
- by strip tension control, in the winding and/or unwinding section.

If the strip thickness is very small (under 0.3 mm), both controls are generated. For h >0.3 mm, AGC can act only by modifying the gap between rolls, the control u being applied through the hydraulic servo-system.

Through the measuring of the thickness error at the rolling plant input (ΔH), AGC system controls the gap among the rolls (u) in order to obtain the smallest output thickness error (Δh). Certainly, the control depends on the feeding speed of the strip (v_1) which determines the delay between the measurement moment of thickness deviation (ΔH) and the moment when the control u is applied. The efficiency of the controls applied to the rolling process is evaluated taking into account the measurement of the output thickness deviation (Δh), the objective being to minimize the spreading of this deviation. The main disturbance in this system is the random variation of rolled material plasticity module, especially after each pass, as a result of the material hardness. Due to this fact, the control loop gain must be adjusted, in order to compensate the unknown variations of the plasticity module. This adjustment capacity of the gain is an essential property of AGC and it must be accomplished by both control structures: the position control system and the tension control system [11]. Consequently, advanced control systems and modern electrical drive systems for the two control structures mentioned before are used. The linearizing control techniques applied to the quarto rolling mill can be noticed [12]. All these control structures take into consideration the hypothesis that the servo-system properties derived from the technological plant design do not considerable change during the various passes within the rolling process.

An original AGC structure, dealing with the position control of a quarto rolling mill is proposed in the present study. This structure offers the possibility to adapt the control algorithm to the current modifications of the rolling mill plasticity module and, in addition, to achieve the following two tasks:

- error compensation induced by the non-ideal dynamics of the roll positioning servo-system;
- control adapting when the dynamic proprieties of the servo-system are changing.

The paper has the following structure: the second section presents the principle scheme which illustrates the proposed control structure. The third section deals with the qualitative analysis regarding the influence of the compensation of the hydraulic servo-system dynamics on the system performances; the fourth section shows the results obtained by AGC numerical simulation, supposing that the rolls positioning servo-system is invariant over time and that it has a known dynamics. The fifth section presents the results obtained through numerical simulation of the AGC system, in the case when the servo-system has unknown dynamics and also time variable characteristics. Section sixth presents the on-line optimization algorithm, which offers the adapting capacity of the control structure to the modifications of the rolled material proprieties. The seventh section presents the results of the control structure proposed in the paper, obtained in simulation regime, using real data, undertaken from a thin strip rolling process (the thickness profile of the feeding strip). The conclusions can be found in the final section.

II. The Control Structure Of The Rolling Mill Process

The thickness deviations at the output of the stand (Δh) derive from two sources:

1. due to the material properties, which can be produced:
   - by the thickness deviations of the feeding strip, ΔH;
   - by the deformation resistance, $\Delta\sigma$. These are determined mainly by the material hardness during various passes, but also by sheet chemical composition on strip length;
2. generated by the rolling plant. These deviations appear, mainly due to the stand yielding, depending on the stand elasticity module, M. The friction coefficient variation of the working rolls with the rolling strip ($\Delta\mu$), can also influence the thickness deviations (Δh).

The plastic deformation model of the rolling mills reflects the dependence of the output thickness error (Δh) by the variables mentioned before, but also by the distance between the working rolls, u. Let us consider:

$$h = h(H, u, \sigma, \mu, ...) \tag{1}$$

the generic model of the thickness strip at the stand output. The model can be written in finite variations as the following:

$$\Delta h = S_H \Delta H + S_u \Delta u + S_\sigma \Delta\sigma + S_\mu \Delta\mu \tag{2}$$

where

$$S_H = \frac{\partial h}{\partial H};\ S_u = \frac{\partial h}{\partial u};\ S_\sigma = \frac{\partial h}{\partial \sigma};\ S_\mu = \frac{\partial h}{\partial \mu} \tag{3}$$

are the sensitive functions with respect to the variables which influence the output thickness errors. The sensitive function S_H depends on the material plasticity module (m) and on the stand elasticity module, M:

$$S_H = \frac{m}{m + M} \tag{4}$$

The sensitive function, S_u, is given by the equation

$$S_m = \frac{M}{m+M} \tag{5}$$

In equation (2), ΔH and Δh represent the measurable disturbance and the measurable output, respectively. The variable Δu is the AGC control (the control of the gap between the working rolls). In the simplest formulation, the effect of the term $S_H \Delta H$ from the equation (2), which is generated by the thickness variation at the stand input, should be compensated by the term $S_u \Delta u$, equivalent to the AGC control, while the parameters S_σ and S_μ interfere in the disturbance term

$$\Delta w = S_\sigma \Delta\sigma + S_\mu \Delta\mu \tag{6}$$

that can not be evaluated.

Equation (7) represents the linearized equation of the process, during a pass:

$$\Delta h = A\Delta H + B\Delta u + \Delta w \tag{7}$$

where $A = S_H$ and $B = S_m$.

If we take into consideration the condition Δh=0 and not the immeasurable disturbance term, it results from equations (4), (5) and (7) the control that must be applied:

$$\Delta u = -\frac{m}{M}\Delta H \tag{8}$$

The stand plasticity module can be rather accurately measured during stand calibration, which is made before every working session, so that the available value ($\hat{M}$), could be used in (8), instead of variable M. On the other hand, the material plasticity module (m) is known only with a raw approximation. Using a preliminary estimation $\hat{m}$ of this variable, it is necessary to introduce the gain G in the control equation (8), so that $\hat{m}.G \cong m$. In these conditions, the control equation becomes

$$\Delta u = -\frac{\hat{m}}{\hat{M}}.G.\Delta H \tag{9}$$

The gain G is the adapting parameter of the process, with respect to the minimization of the error variance, Δh. In the case of the rolling process with several passes, each pass is numbered (the coefficient k representing the current number of the pass during the strip rolling). The output thickness error in the k^{th} pass is Δh_k and $\Delta H_k = \Delta h_{k-1}$. In these conditions, the equation (10) results from the equation (7).

$$\Delta h_k = A_k \Delta h_{k-1} + B_k \Delta u_k + \Delta w_k; \quad k = \overline{1,N} \tag{10}$$

In this equation

$$A_k = \frac{m_k}{m_k + M};\ B_k = \frac{M}{m_k + M};\ A_k + B_k = 1 \tag{11}$$

and N represents the total number of passes (maximum 5). In relation to the strip thickness (h_k), the adjusting process behaves as a dynamic variable parameter system. It is the result of plasticity module modification as a consequence of the rolled material cold harden. Fig. 2 presents the principle scheme of the controlled process.

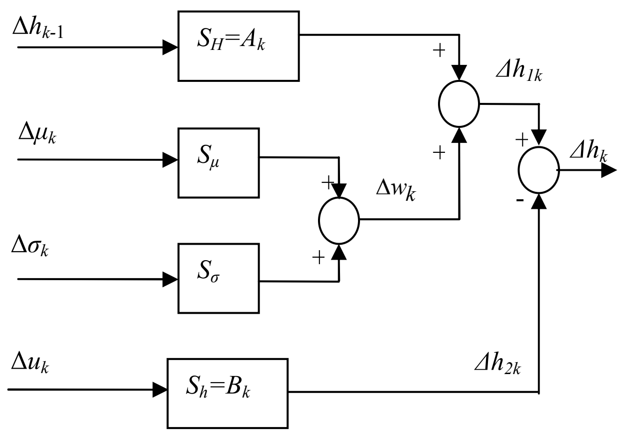

Figure 2. Mathematical model of the plastic deformation process.

In the fig. above, Δh_{1k} represents the thickness error that is independent from the control action and Δh_{2k} represents the thickness error obtained by way of the position control, Δu_k. According to the equation (8), the necessary position control is

$$\Delta u_k = -\frac{m_k}{M}\Delta H_k = -\frac{m_k}{M}\Delta h_{k-1} \tag{12}$$

The sign minus shows that the error Δh_{2k} should compensate the term Δh_{1k}, resulted from the rolling process. Further on, using a similar control to the one from the equation (12), the term Δh_{2k} will be defined by the following equation

$$\Delta h_{2k} = \frac{m_k}{M} B_k \Delta H_k \tag{13}$$

It can be seen in Fig. 2 that the variable Δh_{2k} is subtracted from the variable Δh_{1k}.

Fig. 3 presents the AGC system structure. The thickness error transducer at the stand input provides information shifted in advance compared to the one that interests in the control algorithm, specifically the thickness error of the strip between the working rolls. The delay is determined by two parameters: the distance between the sensor and the working rolls axis and by the lead speed of the strip, v_1. The signal given by the transducer is loaded in a shift register that exists into the controller (LI in Fig. 3). It is extracted from this register with a delay, calculated as a function of the measured speed, v_1. Knowing the error given by the strip speed transducer, the value of the thickness deviation available in the controller, $\Delta\hat{H}_k$ (see Fig. 3), is an estimation of the actual deviation (ΔH_k) of the strip thickness at the input of the working rolls. According to this signal that is available at every moment, the movement of the working rolls (Δu_k) must be assured, so that Δh_{2k} should compensate the thickness variation, Δh_{1k}.

The working rolls movement is performed by a hydraulic servo-system in closed loop whose transfer function is:

$$H_{SH}(s) = K\frac{\omega_n^2}{2 + 2\varsigma\omega_n s + \omega_n^2} \tag{14}$$

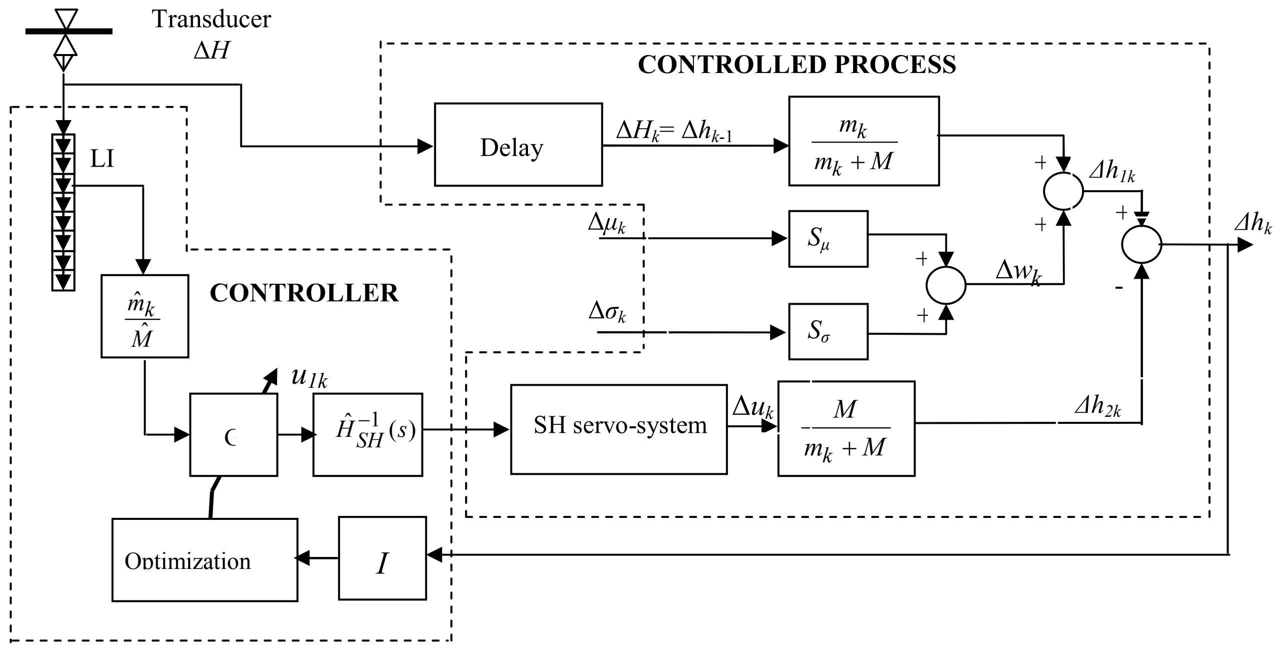

Figure 3. AGC system by rolls gap control based on measured thickness deviation ΔH .

In the controller's structure, there exists a subsystem that achieves an approximation of the inverse of the hydraulic servo-system transfer function $\hat{H}_{SH}^{-1}(s)$.

The introduction of the servo-system inverse subsystem aims to perform a transfer of the control u_{1k}, given by AGC system to the variation (Δu_k) of the gap among the working rolls as fast as possible.

Assuming the hypothetical ideal case in which $\hat{H}_{SH}^{-1}(s) = H_{SH}^{-1}(s)$, it results from the ACG structure (Fig. 3) that:

$$\Delta u_k = u_{1k} = G\frac{\hat{m}_k}{\hat{M}}\Delta\hat{H}_k \tag{15}$$

Since the main uncertainty of the model is given by the estimation of the plasticity module ($\hat{m}_k \neq m_k$), the gain G must be adopted, so that

$$G\frac{\hat{m}_k}{\hat{M}}\Delta\hat{H}_k = \frac{m_k}{M}\Delta H_k \tag{16}$$

In fact, the inverse of a dynamic system is a non-causal operation and the transfer function $\hat{H}_{SH}^{-1}(s)$, at causal limit, can only estimate the inverse system. In addition, the theoretical condition (16) is inoperable at the level of the current controls within the rolling pass, k.

A realistic approach consists in considering that the signals u_{1k} and Δu_k are distinct, representing the input and output respectively of the rolls positioning servo-system, which is provided with a predictive control for the dynamics improvement. This anticipative control is achieved by the causal limit inverse system, $\hat{H}_{SH}^{-1}(s)$.

The adjusting of the gain G is achieved through the performance-minimizing criterion, considering that the thickness deviation at the rolling mill stand output (Δh_k) is measured, meaning that its standard deviation can be determined.

$$I(G) = \sigma_{\Delta h_k} \tag{17}$$

Essentially, the real time calculus of the gain G for the variations Δh_{2k}, which are obtained based on the working roll position control, is required, aiming to compensate in an optimal manner the variations Δh_{1k}, with respect to the criterion (17).

III. Qualitative Analysis Of The Control Structure

The system performances are highly sensitive to the influence of the hydraulic system dynamics, since this dynamics makes the variations of the variables Δh_{1k} and Δh_{2k} to be dephased and, consequently, the compensation of the thickness variations is reduced. The purpose of the qualitative analysis, performed by numerical simulation, was to determine the system sensitivity with respect to the system properties to compensate the hydraulic servo-system dynamics using the causal limit inverse system.

The hypothesis that the transfer function of the servo-system is known in the form (14) and the causal limit inverse model has the transfer function

$$\hat{H}_{SH}^{-1}(s) = \frac{\omega_{n1}^2}{K\omega_n^2}\frac{s^2 + 2\varsigma\omega_n s + \omega_n^2}{s^2 + 2\varsigma_1\omega_{n1}s + \omega_{n1}^2} \tag{18}$$

has been adopted. In the equation (18), $\zeta_1 = 1/\sqrt{2}$ and ω_{n1} is greater then ω_n.

The width of the range frequency domain $[\omega_{n1}, \omega_n]$, in which the inverse system has an anticipative (derivative) character, determines the capacity to compensate the dynamics of the rolls positioning hydraulic servo-system. Fig. 4 and Fig. 5 present the Bode diagrams and the step signal responses of the uncompensated servo-system together with the case when a dynamic compensator is used.

During the system simulation, a slope variation for the gain G that includes the optimal value was required. Fig. 6 shows the evolution of the performance criterion (17), for two values of the frequency ω_{n1}, $\omega_{n1}^{(1)}$ and $\omega_{n1}^{(2)} > \omega_{n1}^{(1)}$. The curves 1 and 2 correspond to the natural frequencies $\omega_{n1}^{(1)}$ and $\omega_{n1}^{(2)}$. We may conclude that the system performance is influenced by the properties of the dynamic compensator (18).

Fig. 7 a and b present details that illustrate instantaneous values of the AGC controls (the variable u_{1k} from Fig. 3) and the values Δu_k of the servo-system output variable, in two cases: without and with dynamic compensator. The records are obtained for the parameter $\omega_{n1}^{(1)}$ (Fig. 7.a) and for $\omega_{n1}^{(2)} > \omega_{n1}^{(1)}$ (Fig. 7.b).

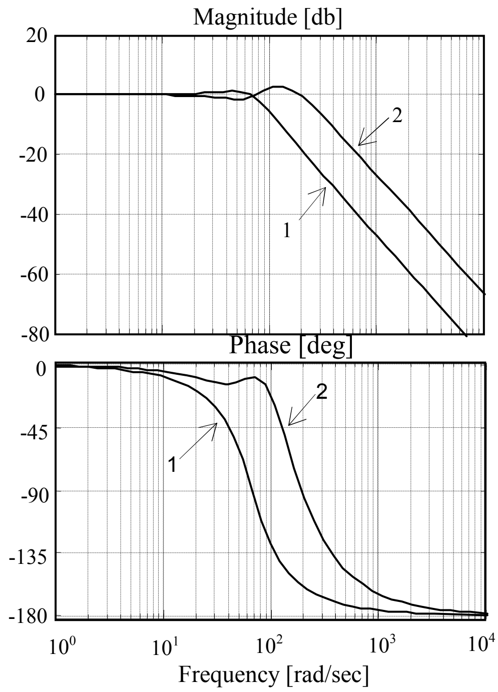

Figure 4. Bode diagrams of the uncompensated (1) and compensated (2) servo-system.

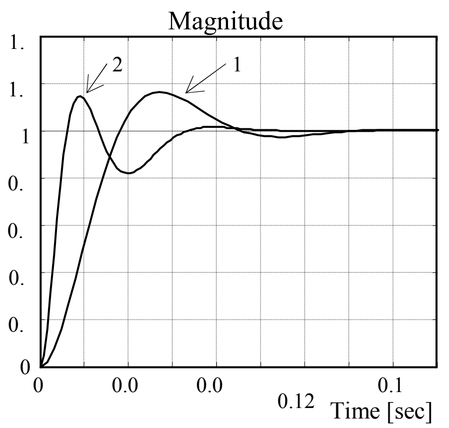

Figure 5. Step signal response of the uncompensated (1) and compensated (2) servo-system.

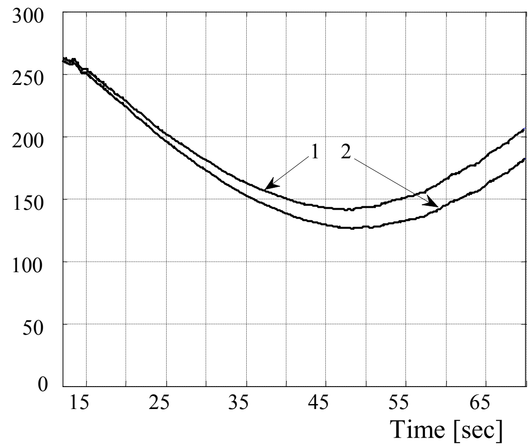

Figure 6. The performance criterion evolution for the frequencies $\omega_{n1}^{(1)}$ - (1) and $\omega_{n1}^{(2)}$ - (2); ($\omega_{n1}^{(2)} > \omega_{n1}^{(1)}$).

In the second case it can be noticed a decrease in the phase difference between the variables u_{1k} and Δu_k.

Figures 8.a and 8.b present the evolutions of the thickness deviations Δh_{1k}, Δh_{2k} and Δh_k, at the same value of the gain G for the two cases mentioned before. It can be noticed that the correction effect of the servo-system dynamics compensation on the final strip thickness deviation (Δh_k), is important, so that the control structure, which is analyzed in the paper, can be tuned with respect to two freedom degrees:

- the compensation of the servo-system (SH) dynamics, that means the phasing of the variable Δh_{2k}(t) in relation with Δh_{1k}(t) (in fact, the phase difference decrease between these two variables);
- the gain G, which determines the magnitude of the variable $\Delta h_{2k}(t)$.

IV. SYNTHESIS OF THE ANTICIPATIVE COMPENSATOR OF THE CAUSAL LIMIT INVERSE MODEL

It is assumed that the inverse model is given by the equation (18), in which ω_{n1} is adopted from practical considerations: the increase of ω_{n1} determines the enlargement of the frequency range in which the compensation of the hydraulic servo-system dynamics is performed, while the possibility of anticipative control effective implementing is reduced, due to the limitations of saturation type that exist in the physical systems and due to the increase of the noise level.

The determination of the causal limit inverse model of the hydraulic servo-system can be approached in two situations:

- the servo-system is invariant;
- the servo-system parameters vary in time, especially from one pass to another.

In the following, the case when the servo-system parameters are modified in time is considered. This option is justified by practical observations, which have shown that the dynamic properties of the servo-system modify in time, especially due to the rolling force. In this case, the identification is made on-line, using an adaptive filter, given by the following equation:

$$y(i) = p^T(i)x(i) \tag{19}$$

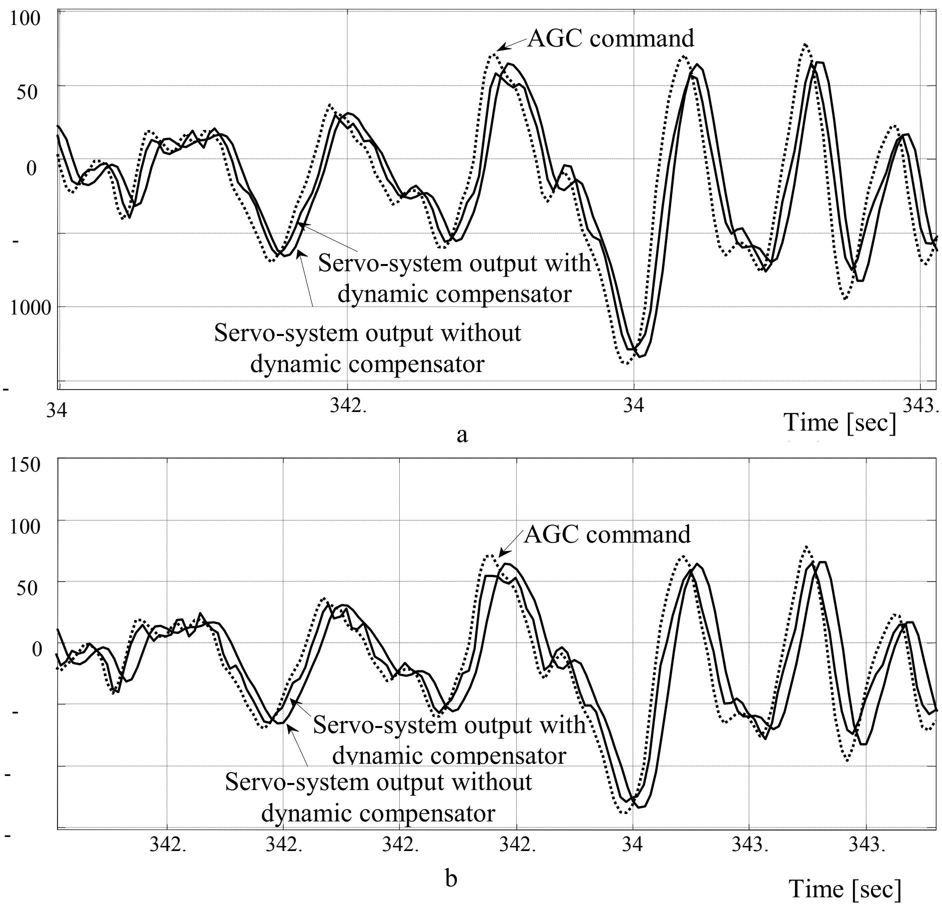

Figure 7. Instantaneous values of AGC control u_{1k} and the values Δu_k of the servo-system output, in the following two cases: the system runs without and with dynamic compensator.

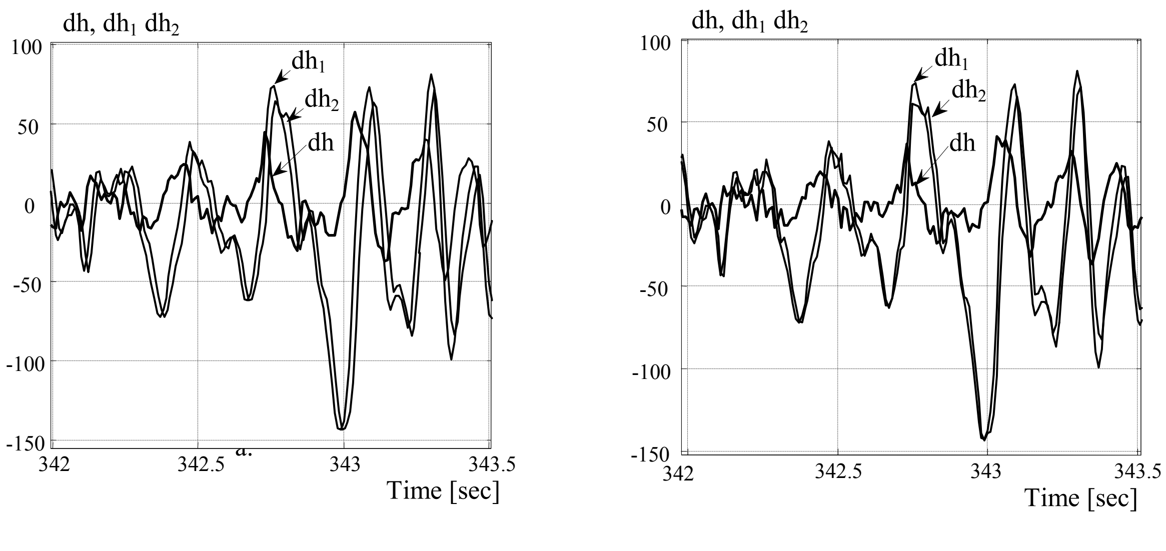

Figure 8. Instantaneous values of the deviations Δh_{1k}, Δh_{2k} and Δh_k, at the same value of the gain G, when the parameter $\omega_{n1}^{(1)}$ is used in the compensator (a) and to the use of the parameter $\omega_{n1}^{(2)} > \omega_{n1}^{(1)}$.

where i is the current sampling step,

$$p(i) = [b_1(i) \quad b_2(i) \quad a_1(i) \quad a_2(i)]^T \tag{20}$$

is the servo-system discrete time parameter vector and

$$x(i) = [u(i-1) \quad u(i-2) \quad -y(i-1) \quad -y(i-2)]^T \tag{21}$$

is the regression vector, in which u and y are the input and output variables of the servo-system. Parameter estimation is made by recursive least square (RLS) algorithm, which involves the recursive calculation of the following variables:

- adapting gain

$$g(k)=\frac{\lambda^{-1}C_{xx}^{-1}(k-1)x(k)}{1+\lambda^{-1}x^{T}(k)C_{xx}^{-1}(k-1)x(k)} \tag{22}$$

- previous error

$$e(k)=y(k)-h^{T}(k-1)x(k) \tag{23}$$

- parameters h

$$h(k)=h(k-1)+g(k).e(k) \tag{24}$$

- the inverse of the auto-correlation matrix

$$C_{xx}^{-1}(k)=C_{xx}^{-1}(k-1)-g(k).x^{T}(k)C_{xx}^{-1}(k-1) \tag{25}$$

where λ is the forgetting factor.

For the effective calculus of the inverse model, the diagram in Fig. 9 has been adopted, whose transfer function ($H_{SHI}(s)$) approximates the causal limit inverse model of the hydraulic servo-system:

$$H_{SHI}(s)=\frac{1}{1/K+H_{SH}(s)}\cong H_{SH}^{-1}(s) \tag{26}$$

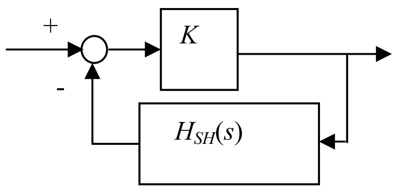

Figure 9. The scheme for obtaining the inverse system of the hydraulic servo-system causal limit model.

The parameter K determines the approximation error, meaning the frequency range in which the servo-system dynamics compensation is achieved.

The control scheme of the hydraulic servo-system, shown in Fig. 10, uses the subsystem $H_{SHI}(s)$ as anticipative controller, beside a PI classical controller. Fig. 11 shows the step signal response of this system, obtained by numerical simulation. A remarkable improvement of the servo-system dynamics was obtained when the inverse model compensator has been used, as compared to the case when the system works without compensator of causal limit model type. The constant K (see Fig. 10) imposes the frequency bandwidth of the servo-system dynamics compensator. The higher this constant is, the larger frequency range the compensation is made in. Consequently, the control magnitude applied to the servo-system input is greater. Therefore, the value of the constant K is limited by the implementation possibilities of the predictive control required by the limitations of saturation and noise level type.

Within the thickness control system, the anticipative and the PI controller are implemented by software and the transfer function $H_{SH}(s)$ is identified on-line, using an adaptive filter, based on the equations (19) – (25). The system presented in Fig. 3 was simulated using a sampling period T_s = 0.002 s. The performance criterion, given by the equation (17), was calculated using a recursive method, based on sample sets of 500 samples, therefore with a period of 1 second. During the system simulations, the gain G has a slope variation, so that it reaches the optimal value, within the domain of variation. Fig. 12 presents the performance criterion variation. The sensible decreasing of the variance in the optimum area is a consequence of the phase and magnitude adjusting of the variable Δh_{2k} with respect to Δh_{1k}. The dynamic compensator that is of causal limit inverse system type achieves the phase adjusting. As it can be seen in Fig. 13, the inverse system introduces a phase lead aiming to compensate the delay given by the hydraulic servo-system. Due to the fact that the dynamic compensator has a derivative character, its output contains a random component. The larger the frequency range in which the dynamics compensation is, the larger this component is (meaning the constant K is bigger). The evolution in the area of the criterion minimum of the variables dh_1, dh_2 and dh is shown in Fig. 14 (see also Fig. 12). It can be seen that the thickness variations dh are small, as a result of the two operations performed in the controller, meaning:

- phasing the variation dh_2 with respect to the variable dh_1, by means of dynamic compensator;
- the gain of the controller control so that the criterion (17) has a minimum value that means the variation dh_2 is as close as possible to the variable dh_1.

A supplementary correction aiming to generate in advance the control that determines the variable dh_2 for phasing the variables dh_2 and dh_1 can be used. This correction is performed by extracting the variable $\Delta\hat{H}_k$ in advance from the shift register LI (see Fig. 3).

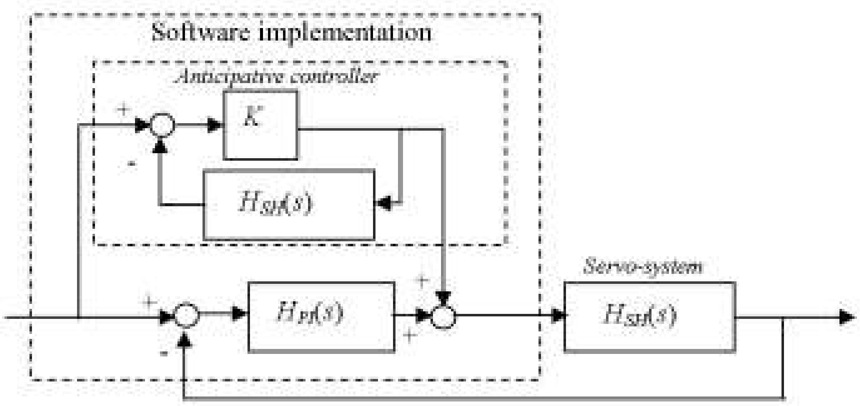

Figure 10. The servo-system control based on the anticipative controller.

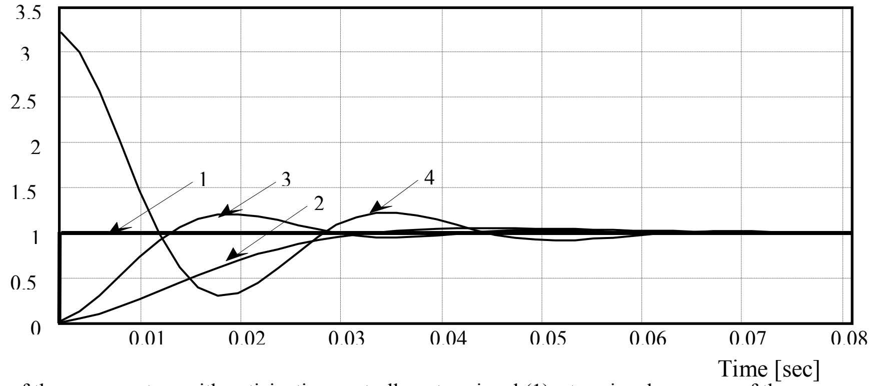

Figure 11. Step signal response of the servo-system with anticipative controller: step signal (1); step signal response of the servo-system without predictive controller (2) and with predictive controller (3); the control applied to the servo-system input (4).

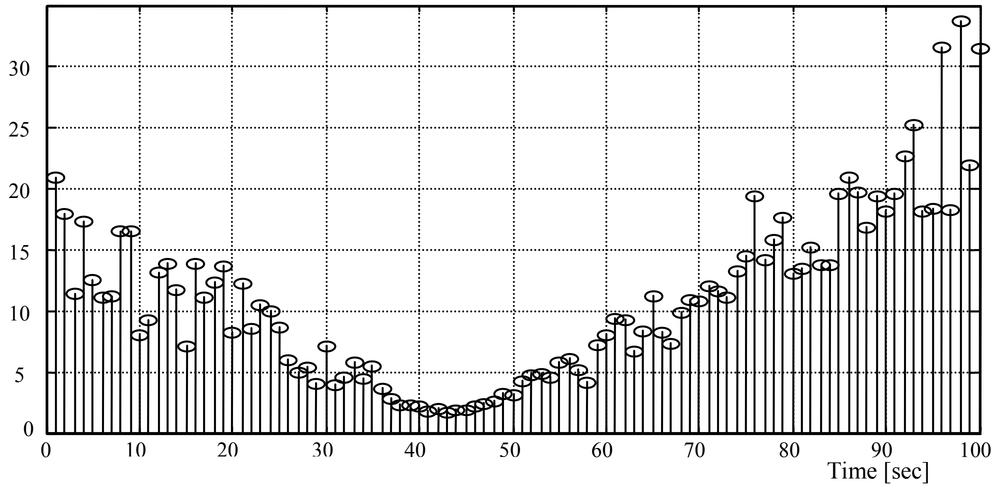

Figure 12. The evolution of the performance criterion considering a slope variation of the gain *G*.

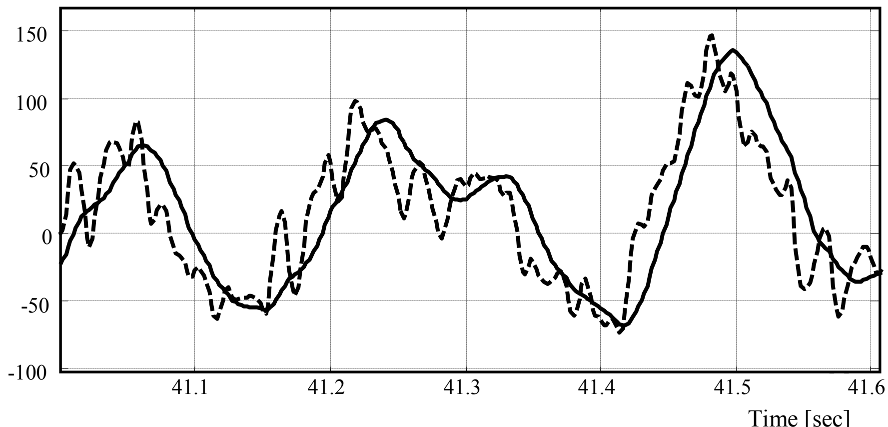

Figure 13. The evolution of the input variables (continuous line) and of the output variables (dotted line) from the dynamic compensator.

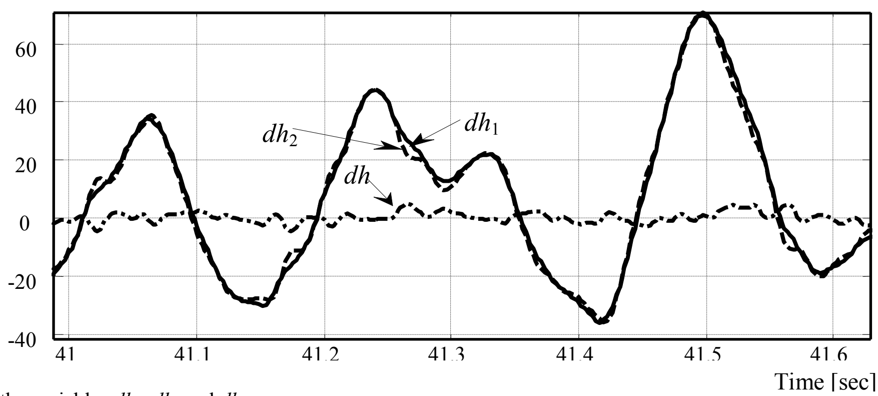

Figure 14. The evolution of the variables dh_1, dh_2 and dh.

V. CALCULUS OF THE OPTIMAL GAIN

The optimization procedure consists in the calculus of the gain G, in order to minimize the standard deviation of the variable Δ. The estimation of the standard deviation is made on a span of 500 numerical values of the variable $\Delta h(i)$, that is corresponding to a time span of 1 second. As it can be seen in Fig. 12, these estimations include a fluctuant component, due to the small number of numerical values that are taken into account.

We have tried to avoid the failures in determining the minimum value of the variable G, due to the very noisy rate of the variable Δh, case when the search procedure could stop in a local minimum.

Thus, an one-dimensional search method (relaxation) to determine the minimum of the criterion function was used. It runs with a constant step, as long as the criterion value is greater then an imposed threshold, after which it switches to a standard version of the method that uses the reducing to one half the searching step and reversing the search direction. This method was applied in the case of a strip pass through the rolling mill stand in a span of 100 seconds, the searching step being equal to 1 second.

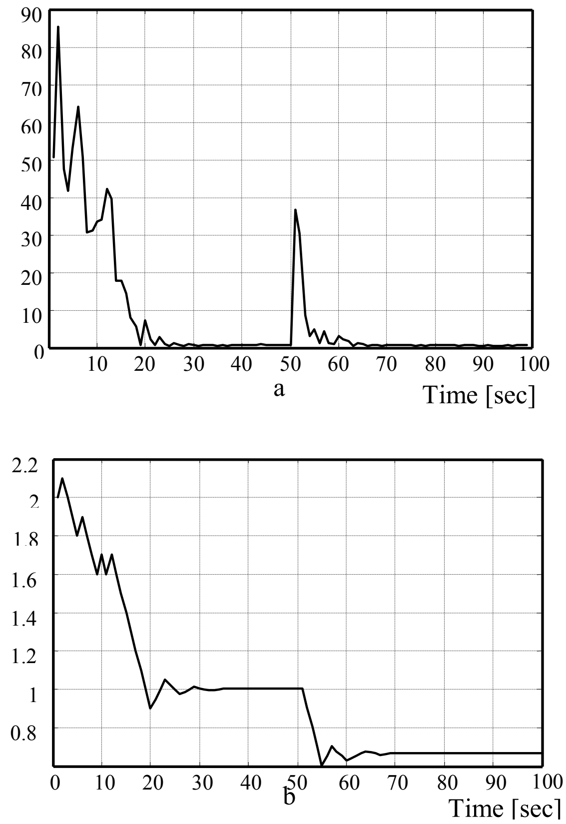

Figure 15. The evolution of the standard deviation Δh (a) and of the gain G (b) within the searching process.

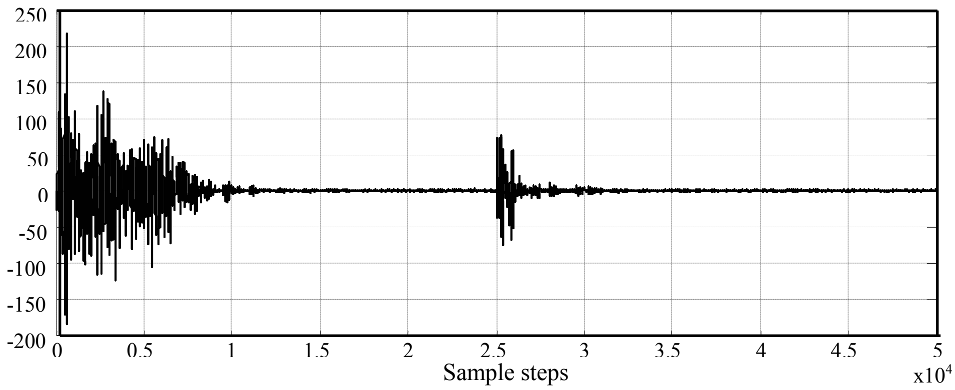

Figure 16. The evolution of the instant values of the deviation $\Delta h(i)$.

Figures 15 and 16 present the simulation results. Figures 15a and 15b show the standard deviation Δh and the gain G during the searching process. At the moment $t = 50$s, that is, after 2500 sampling steps, the rolled material plasticity module has been modified (this module has been increased through a step variation of 300%). It can be noticed the quick adapting of the parameter G and its stabilization to a new value, by minimizing the performance criterion. Fig. 16 shows the evolution of the instant values of the deviation $\Delta h(i)$, during the 50000 sampling steps.

VI. RESULTS OBTAINED BASED ON REAL INPUT DATA

In the previous sections, the input signal of the system, i.e., the thickness variations ($\Delta H_k(i)$), was synthetically generated, by filtering the white noise, so that the time series obtained to the filter output has statistical properties, similarly to a time series ($\Delta H_k(i)$) from a real process. We have to mention that in the model of the controlled process (see Fig. 3) the uncertainties regarding the parameters m and M are compensated by the optimal gain G and the transfer function of the servo system is identified in real time by means of an adaptive filter. This transfer function, given by the equation (14), is classic for the servo-system type that is used, and the parameters used in the simulations were chosen in such a manner that the simulated servo-system has dynamic properties close to those of a real servo-system. This rolling mill, used as reference within this paper, has the following basic properties: the thickness range: 0.07 – 0.8 mm; the strip speed: 0 – 240 m/min.; the strip tension: 50 – 750 daN to unwinding and 10 – 750 daN to winding; the working pressure to the hydraulic servo-system: 1 – 6.3 Mpa; the nominal power of the electrical engines: P_n = 37 kW etc.

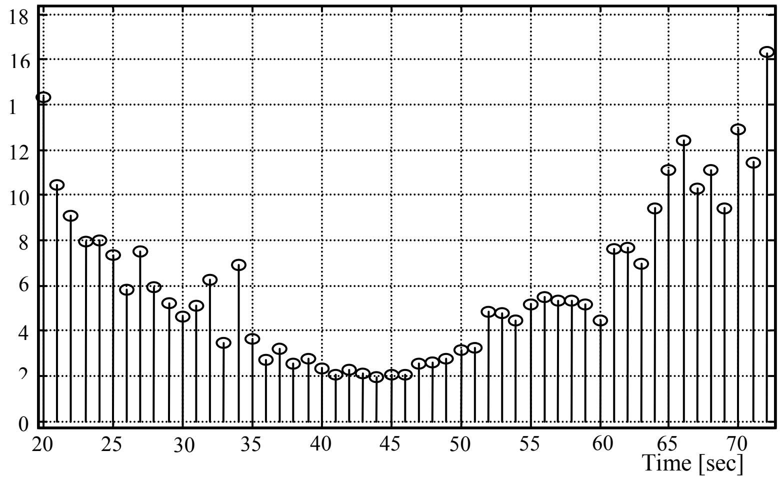

Figure 17. Performance criterion evolution when testing the system with real data input.

The analysis made through numerical simulation of the system from Fig. 3 has been done on the basis of real data, $\Delta H_k(i)$, supplied by this rolling mill. Similar to the results presented in section 4, the gain G has a slope variation, so that to reach the optimal value within sthe variation range. The evolution of the performance criterion, illustrated in Fig. 17, does not present significant differences from the case of input synthetic data use (see Fig. 12). Fig. 18 presents the evolution of the input and output variables from the dynamic compensator. Fig. 19 shows the evolution of the variables dh_1, dh_2 and dh.

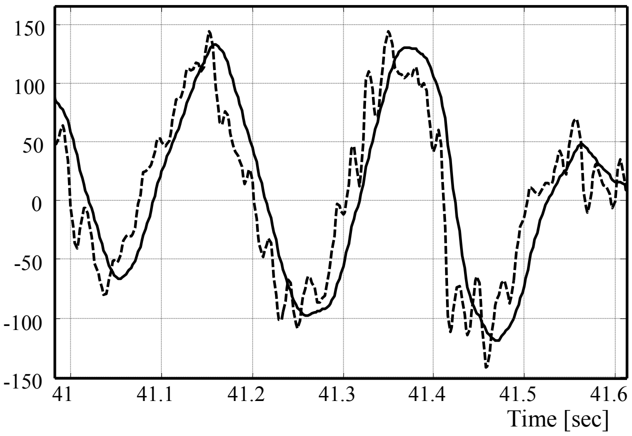

Figure 18. The evolutions of the input variables (dotted line) and of the output variables (continuous line) from dynamic compensator.

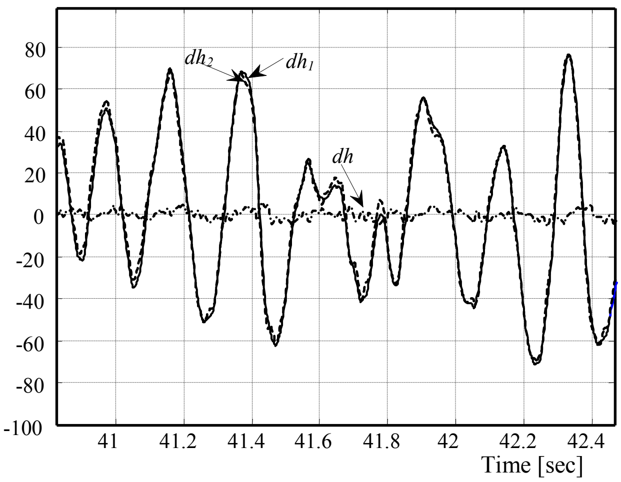

Figure 19. The evolution of the variables dh_1, dh_2 and dh.

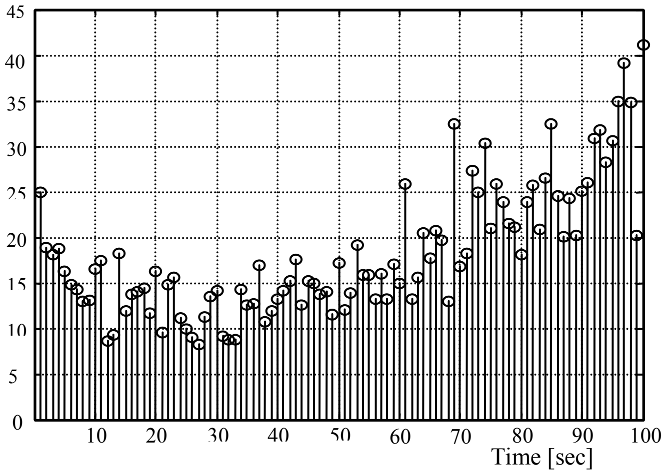

Figure 20. The evolution of the performance criterion when the system runs without dynamic compensator.

These results are similar to the ones presented in Fig. 13 and Fig. 14 respectively. For emphasizing the importance of the dynamic compensator in the system working, a functioning regime where a system with the transfer function $\hat{H}_{SH}^{-1}(s)$ was eliminated from the scheme given in Fig. 3, has been achieved. In this case, the evolution of the performance criterion, presented in Fig. 20, reflects a considerable deterioration of the process quality. The explanation for the system performance decrease results from Fig. 21, where the evolution of the variables dh_1, dh_2 and dh is presented. It can be noticed that the dephasing of the variable $dh_2(i)$, with respect to the variable $dh_1(i)$, leads to the considerable increase of the difference $dh(i)=dh_2(i)-dh_1(i)$.

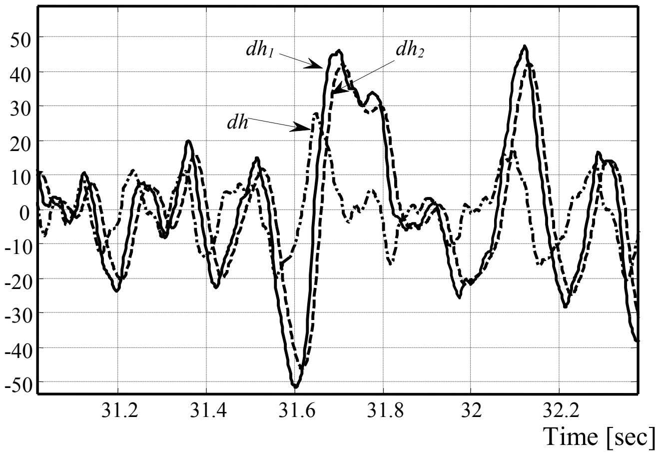

Figure 21. The evolution of the variables dh_1, dh_2 and dh when the system runs without dynamic compensator.

VII. CONCLUSIONS

The AGC structure proposed in the paper, aimed to control the position of a quarto rolling mill, offers the possibility to adapt the control algorithm to the current modifications of the strip plasticity module and, moreover, it performs two important properties:

- the compensation of the errors induced by the defective dynamics of the rolling mills positioning servo-system. This compensation is achieved through a causal limit inverse model. The width of the frequency domain, in which the compensation of the hydraulic servo-system dynamics is done, is controllable, in order to create the possibility of effective implementation of the predictive control, taking into consideration the limitations of saturation type existing in the physical system and the noise level increase;
- the control adjustment with respect to the changes of the dynamic properties of the servo-system. This function is performed by identifying the hydraulic servo-system in real time and the corresponding adjustment of the dynamic compensator of causal limit inverse model type.

Feasibility of the proposed solutions is determined by the adaptation of the spectral properties of the time series $\Delta H_k(i)$ to the dynamic properties of the hydraulic servo-system used in simulation. Dynamic properties of this servo-system are similar to those of the real servo-system from the rolling mill used in the present paper as a reference. With the excessive increase of rolling speed, the spectrum of the time series $\Delta H_k(i)$ is moving toward the higher frequency range and, as a consequence, the tracking capacity of the servo-system is reduced, as well as the possibilities of dynamic error compensation through the inverse model at causal limit. For this reason, we considered crucial the testing of the proposed system with input signals $\Delta H_k(i)$ taken from the real plant. The system main non-linearities are of saturation type. The proposed control structure uses an anticipative controller, which has the possibility to modify the dynamic properties, so that the non-linearities of saturation type be avoided.

The results obtained through numerical simulation, including the ones obtained based on real data supplied by real process, illustrate the feasibility of the solutions proposed in this paper.

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REFERENCES

[1] Asada H., Kitamura A., Nishino S., and Konishi M.., Adaptive and robust control method with estimation of rolling characteristics for looper angle control at hot strip mill, ISIJ International, Vol. 43 (No. 3), pp. 358–365, 2003 (The Iron and Steel Institute of Japan).

[2] Asano K., Yamamoto K., Kawase T. and Nomura N., Hot strip mill tension – looper control based on decentralization and coordination, Control Engineering Practice, Vol. 8, No. 3, pp. 337–344, 2000.

[3] Hearns G. and Grimble M. J., Inferential control for rolling mills, IEEE Proc. Control Theory and its Applications, 147 (No. 6), pp. 673–679, 2000.

[4] Hearns G. and Grimble M. J., Robust multivariable control for hot strip mills, ISIJ International, Vol. 40 (No. 10), pp. 995-1002, 2000.

[5] Imanari H., Morimatsu Y., Sekiguchi K., Ezure H., Matuoka R., Tokuda A. and Otobe H., Looper H_∞ control for hot strip mill, IEEE Transactions on industry applications, 33 (3), pp. 790–796, 1997.

[6] Janabi-Sharifi F., A neuro–fuzzy system for looper tension control in rolling mills, Control engineering practice, Vol. 13, No.1, pp. 1-13, 2005.

[7] Choi J. I. S., Rossiter J. A. and Fleming P. J., Looper and tension control in hot rolling mills: A survey, Journal of Process Control, Vol. 17, Issue 6, July 2007, pp. 509-521.

[8] Konishi M., Imajo S., Imai J., Nishi T., Modeling of gain tuning operation for hot strip looper controller by recurrent neural network, Proceedings of the 2004 IEEE International Conference on Control Applications, 2004, pp. 890–895.

[9] Bulut B., Ordys A. W., Grimble M. J., Application of efficient nonlinear predictive control to a hot strip finishing mill, Proceedings of the 2002 IEEE International conference on Control Applications 2002, pp. 373–378.

[10] Bulut B., Katebi M. R., Grimble M. J., Predictive Control of Hot Rolling Processes, Proceedings of the American Control Conference, Chicago, Illinois, June 2000, pp. 2058-2062.

[11] Rath G., Model Based Thickness Control of the Cold Strip Rolling Process, PhD Thesis, Leoben, April 2000.

[12] Rabbah N., Bensassi B., Control of a Multivariable Web Winding System, The Annals of “Dunarea de Jos” University of Galati, Fascicle III, 2007, ISSN 1221-454x, Electrotechnics, Electronics, Automatic Control and Informatics, pp. 107-113.