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ELECTROHYDRAULIC SYSTEM FOR AUTOMATIC GAGE CONTROL (AGC) FOR TANDEM COLD MILL PLANT IN SARTID SMEDEREVO

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Abstract : Electro hydraulic servosystem for the AGC has better characteristics than electromechanic (five timesgreater speed of rolling, greater speed of positioning ,smaller dead–zone, smaller time of roll gap adjusting start, smaller time of maximum speed reaching,greater unloading speed).

Keywords: *Automatic control system, electromenechanical, electrohydraulic system,mechanical system, servovalve.*

I. INTRODUCTION

In Hot and Cold Mill Plants were mounted electrohydraulic servosystems for automatic gage control (instead electromechanic systems with screw thread) during modernisation. For the applications of the mostly control algorithms are necessary knowledge of mathematical models as components as complete object of automatic control. Detail analysis of these servosystems will help for better understanding and for the next optimisations by apllications of modern control methods.This work is based on results from literature [1].

II. ELECTROMECHANICAL SYSTEM FOR AUTOMATIC GAGE CONTROL

Automatic control system works on the following manner:if aberation of the strip thickness at the stand exit is happened then a signal from a thickness gage 9 by converter is led to summator 14 where this signal compares with set up signal.. Error signal is led to regulator and from regulator to electromotor 13. .Electromotor 13 moves screw and changes rolling force and thickness of streep too. The greater disadvantage of this regulation type is the great friction between screw and screw nut which disable very quick and very precision work.

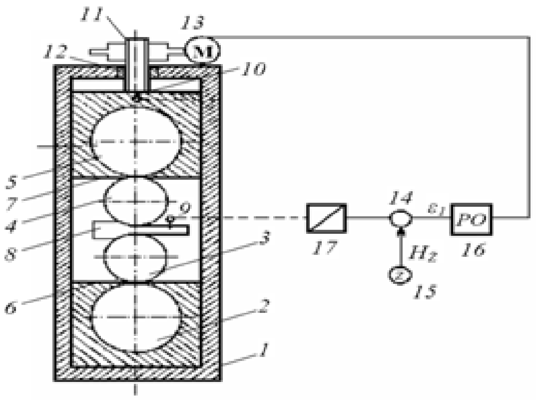

Figure1. Mechanical system scheme for AGC

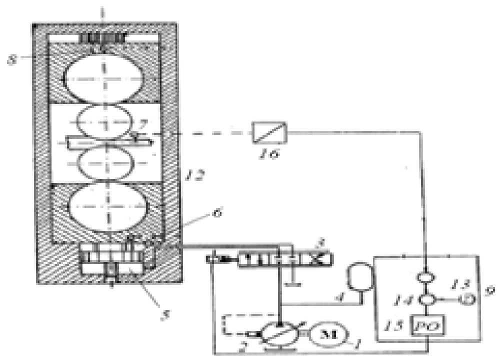

Figure 2. Electrohydraulic system sheme for AGC

III. ROLLING STANDS ELASTIC DEFORMATION AND PLASTIC DEFORMATION CURVE

Sum of elastic deformations of the all loaded rolling stand parts is rolling stand elastic deformation.Elastic deformation of rolling stand can be determined by theoretical or experimental method(theoretical method is rarely used because it is hard to determine the clearances between parts of rolling stand).

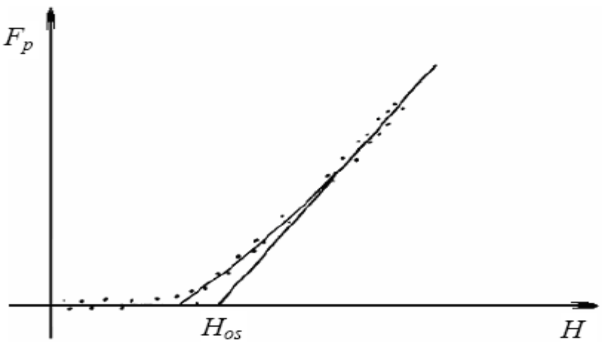

Figure 3. Curve of elastic deformation of the rolling stand

From the figure 3 is possible to write equation (1):

$$H_i = H_{os} + \frac{F_p}{E_s} \quad (1)$$

the following symbols are defined:

H_i - strip thickness at the exit from the rolling stand,

H_{os} - value of the initial gap between rolls,

F_p - modulus of rolling stand elasticity.

Therefore equation (1) contains two unknown values (rolling force and thickness of the strip at the exit from the stand) it is necessary to know dependence of rolling force from exit strip thickness for a concrete rolling conditions (entry strip thickness,friction coefficient,rolls diameters...).
Desired dependence is given by equation(2):

$$F_p = f(H_u, R, \lambda.....) \quad (2)$$

Curves given by equation (2) are strip plastic deformation curves(curves of plasticity and it is possible to obtain its by theoretical or experimental method. Figure 4 shows plastic deformation curve of strip.

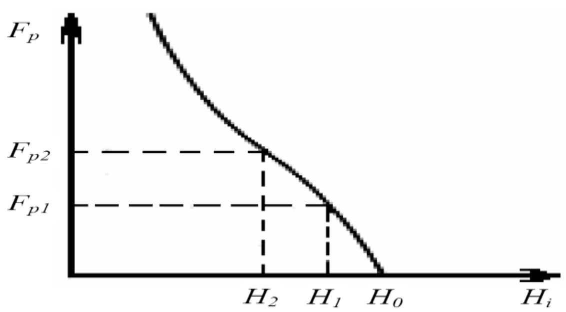

Figure 4. Plastic deformation curve of strip

From figure 4 it is obvious that greater rolling force enables smaller strip thickness at the rolling stand exit. Simultaneously equations (1) and (2) solvings give rolling force and exit strip thickness during strip deformation in real working stand. Graphic equations solution is shown on figure 5.

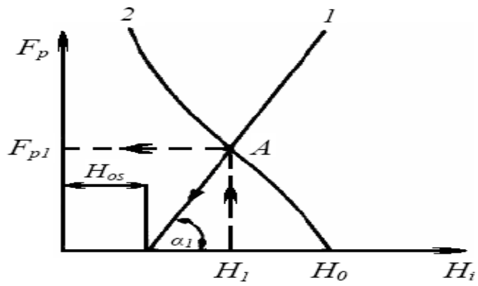

Figure 5. Graphic determination of the rolling force

Section n the straight line 1- the curve 2 determines point A. The coordinates of the point A (F_{p1} and H_1) determine rolling force F_{p1} which must act on the rolls to perform exit strip thickness H_1 . The gap between rolls increases from H_{os} to H_1 until thickness of the strip reduced from H_{os} at entry of the stand to H_1 at the exit of the stand.In the process computer are "memorized" straight line 1 and curve 2 and for demand H_1 it is possible to determinate F_{p1} and H_{os} (it is necesarry to draw a vertical line from the point H_1 and point of section with the curve 2 is point A; From point A it is necessary to draw a horizontal line and obtain F_{p1} : It is necessary to draw a line through a point A with angle and obtain H_{os} . However,if the strip has the greater hardness then the rolling force Fp1 will not be sufficient to perform H_1 and control system will increase the rolling force F_{p1} to get demand thickness of the strip at the exit of the stand.

IV. SERVOVALVE

Therefore a reprensentation of servovalve response throught the frequency range about 50 cps is sufficient (literature [1]), and a first-order expression is adequate. The time constant for the first-order transfer function is best established by 0.7 amplitude point (-3 db) (figure 6). Figure 6 shows a "Moog" servovalve dynamic response,together with the response of a first–order transfer function. The first-order aproximation is a quite good throught the lower frequency region. According to literature [2] and figure 6 we can write:

$$W_{sr} = \frac{1}{1+T_1S} \quad \text{while:} \quad T_1 = \frac{1}{2\pi\omega_1} \qquad (3)$$

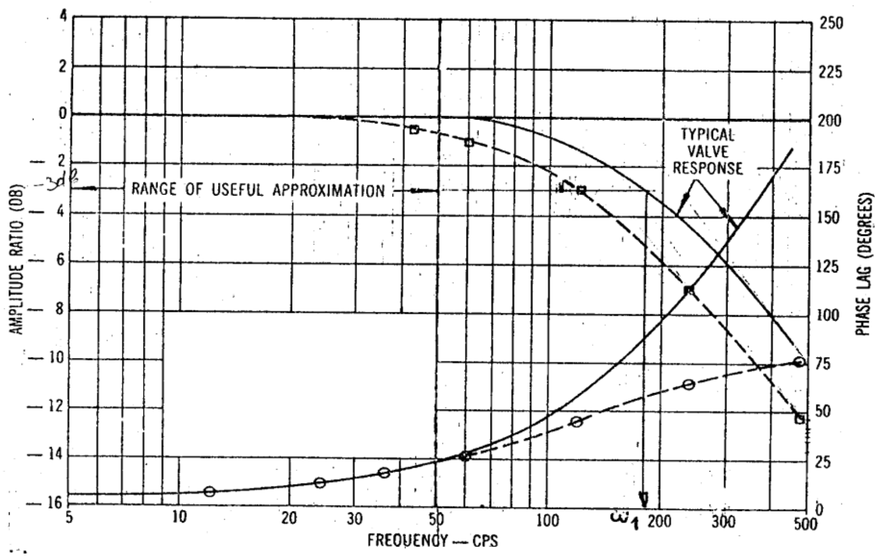

Figure 6. Bode diagram of the "Moog" servovalve

V. SERVOVALVE CONTROLLED PISTON

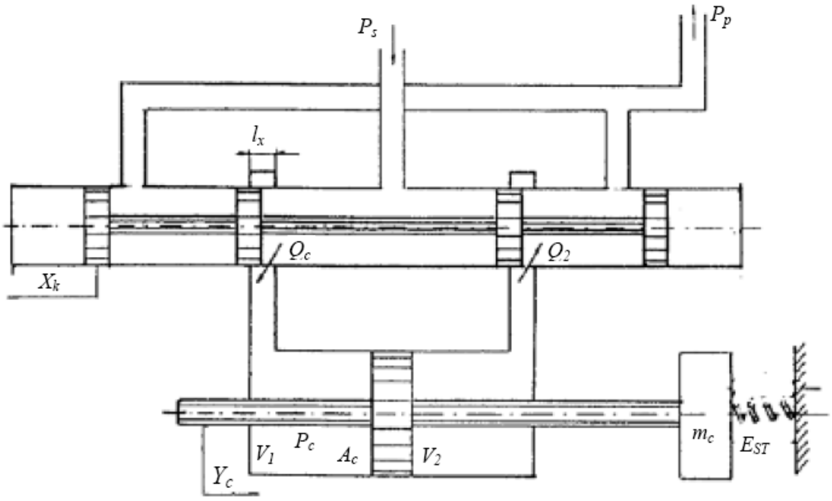

Figure7. Servovalve controlled cylinder piston

Figure 7. Servovalve controlled cylinder piston p_c; $F_{pc} = p_c A_c$; force induced by roll stand elasticity:

$$F_x = E_{ST} Y, \quad (4)$$

force of inertia: $F_{in} = m_c \dfrac{d^2 Y_c}{dt^2}$

while: $m_c = m_T + m_{kc}$;

m_T – load mass,

m_{kc} - mass of the cylinder piston,

m_c - “common” mass .

Force of friction in the gaskets is negligible because the gaskets are made from a special PTFE material. The Second Newton's law for cylinder piston is given by equation (5):

$$p_c A_c = m_c \frac{d^2 Y}{dt^2} + E_{ST} Y \cdot \quad (5)$$

Cylinder entry flow (Q_c) is consist of the flow for the piston moving Q_v and flow for the compressibility compensation Q_{LC} , while leakage is negligible. We can write flow equation (6):

$$Q_c = A_c \frac{dY}{dt} + \frac{V}{B} \frac{dp_c}{dt} \cdot \quad (6)$$

Flow which leaves servovalve is:

$$Q_{SR} = \mu_{SR} W_{SR} X_K \sqrt{\frac{2}{\rho}(p_S \quad p_C)} \cdot \quad (7)$$

We can combine equations (5) and (6) to yield equation (8):

$$\mu_{SR} W_{SR} X_K \sqrt{\frac{2}{\rho}(p_S \quad p_C)} = A_C \frac{dY}{dt} + \frac{V}{B} \frac{dp_C}{dt} \cdot \quad (8)$$

VI. GAUGES,TRANSDUCERS AND SIGNAL AMPLIFIERS

From literature [1] we can write equations for the position gage,transducer and signal amplifier:

$$W_{MP} = \frac{U_{MP}(s)}{Y_c(s)} = \frac{K_{PC}}{T_u S'+1}, \tag{9}$$

$$U_y = C_1 H_{iy}, \tag{10}$$

$$I = K_A (U_y - U_{MP}) \tag{11}$$

while: W_{MP} – transfer function for piston rod position gage together with the position-voltage transducer, U_{MP} - exit signal from the position gage, H_{iy} - demand piston rod position signal, C_1 - amlification of demand position signal, K_A - amplification of voltage-current transducer.

VII. MATHEMATICAL MODEL OF A ROLLING STAND

Linearisation of equation (1) gives equation (12):

$H_i = H_{osi} + \dfrac{F_p}{E_s}$ is given by:

$$H_i = H_{in} + (H_{osi} - H_{osin}) \left.\frac{\partial H_i}{\partial H_{osi}}\right| + (F_{Pi} - F_{Pin}) \left.\frac{\partial H_i}{\partial F_{Pi}}\right| . \tag{12}$$

From equations (1) and (12) we can write equation (13) (in relative variations):

$$h_i = \frac{H_{OSIN}}{H_{IN}} h_{osi} + \frac{F_{pin}}{H_{in} E_{si}} f_{pi} = h_i = l_{1i} h_{osi} + l_{2i} f_{pi}. \tag{13}$$

We consider cylinder piston rod, bottom work roll and bottom back up roll as “common” unit and therefore piston rod position change is the same as roll gap change. Therefore we can write equation (14):

$$h_i = l_{1i} y_c + l_{2i} f_{pi}. \tag{14}$$

In the literatures [1] and [3] we can find equation (14) for the any of Sartid Cold Mill stand:

$$F_{pi} = \lambda_1 F_{K_{1i}} + (1-\lambda_1) F_{K_{2i}} - \frac{2}{3} F_{ZZi} - \frac{1}{3} F_{ZPi} \cdot \sqrt{R(Hu_i - H_i)}.\ 0{,}6 + 0{,}4 \sqrt{\frac{H_i}{H_{ui}}} e^{\frac{\mu \sqrt{R_i (H_{ui}\ \ H_i)}}{0{,}12 H_i + 0{,}28 H_{ui}}} + \frac{2}{3} a_4 \sqrt{R_i H_i} \left(F_{K_{2i}} - F_{zpi}\right)^{3/2}$$

(15)

where:

$$F_{K_1} = a_1 \left(a_2 + \frac{H_{ui}}{H_0} \right)^{a_3},$$

$$F_{K_2} = a_1 \left(a_2 + \frac{H_{ui}}{H_5} \right)^{a_3}.$$

For the soft steel are:

$$a_1 = 55{,}3; a_2 = 1{,}002; a_3 = 0{,}27; \lambda_1 = 0{,}2; a_4 = \sqrt{\frac{1 \quad \gamma^2}{E}},$$

F_{zzi} - force of back tension,

F_{zpi} – force of front tension,

R – work roll diameter,

H_0 - thickness of the strip at the first stand entry,

H_5- thickness of the strip at the last stand exit.

Therefore we can write equation (16):

$$F_{pi} = (H_{ui}, H_i, F_{zzi}, F_{zpi}). \tag{16}$$

We can obtain equation (16) (in relative variations) by linearisation of equation (15):

$$f_{pi} = \frac{H_{in}}{F_{pin}} \frac{\partial F_{pi}}{\partial H_i}\bigg| h_i + \frac{H_{uin}}{F_{pin}} \frac{\partial F_{pi}}{\partial H_{ui}}\bigg| h_{ui} + + \frac{F_{zzin}}{F_{pin}} \frac{\partial F_{pi}}{\partial F_{zzi}}\bigg| f_{zzi} + \frac{F_{zpin}}{F_{pin}} \frac{\partial F_{pi}}{\partial F_{zpi}}\bigg| f_{zpi} + q_{1i} h_i + q_{2i} h_{ui} + q_{3i} f_{zzi} + q_{4i} f_{zpi}. \tag{17}$$

We can combine equations (12) and (17) and write equation (18):

$$h_i = \frac{l_{1i}}{1 \quad l_{2i} q_{1i}} y_{ci} + \frac{l_{2i} q_{2i}}{1 \quad l_{2i} q_{1i}} h_{ui} + \frac{l_{2i} q_{3i}}{1 \quad l_{2i} g_{1i}} f_{zzi} + \frac{l_{2i} q_{4i}}{1 \quad l_{2i} q_{1i}} f_{zpi} + a_{1i} y_{ci} + a_{2i} h_{ui} + a_{3i} f_{zzi} + a_{4i} f_{zpi} \tag{18}$$

In the literature [1] are given all values for the coefficients g, l and a .Complete calculations, experimental results and producers catalogs give conclusion that in the rolling process with the»steady rolling speeds "(without considerations of taking in (accelerating)» and»taking out(slowing down)" the strip in (out) the rolling stand), the tension forces change smaller than 5% (lit [1]). Variations of the strip thickness which enters in the Cold Rolling Mill are very small because Sartid Hot Mill has electrohydraulic system for automatic gage control, too. Coefficients a_{2i} , a_{3i} and a_{4i} are much smaller than coefficient a1 (reference [1]) and according to these conclusions we can write equation (19) :

$$h_i = a_{1i} y_{ci}. \tag{19}$$

VIII. LINEAR MATHEMATICAL MODEL,BLOCK DIAGRAM AND STATE-SPACE REPRESENTATION OF THE SYSTEM

Linerisation of equations (3), (4), (5), (7), (10), (11) and equation (19) give linear mathematical model for a rolling stand:

Regulator: $q_{sr} = K_q x_k + K_c p_c,\ q_c = A_c \dot{y}_c + \frac{V}{B} \dot{p}_c,\ p_c A_c = m_c \ddot{y}_c + E_{ST} y_c,$

$$i = x_k + T_1 \dot{x}_k, \tag{20}$$

where: $i = K_A (u_z - u_{MP}),\ T_u \dot{u}_{MP} + u_{mp} = K_{pc} y_c,\ u_z = c_1 h_{iz}$, object; $h_i = a_1 y_c$.

We can draw block diagram of the system for the automatic gage control of one rolling stand (figure 8) using system of equations (20):

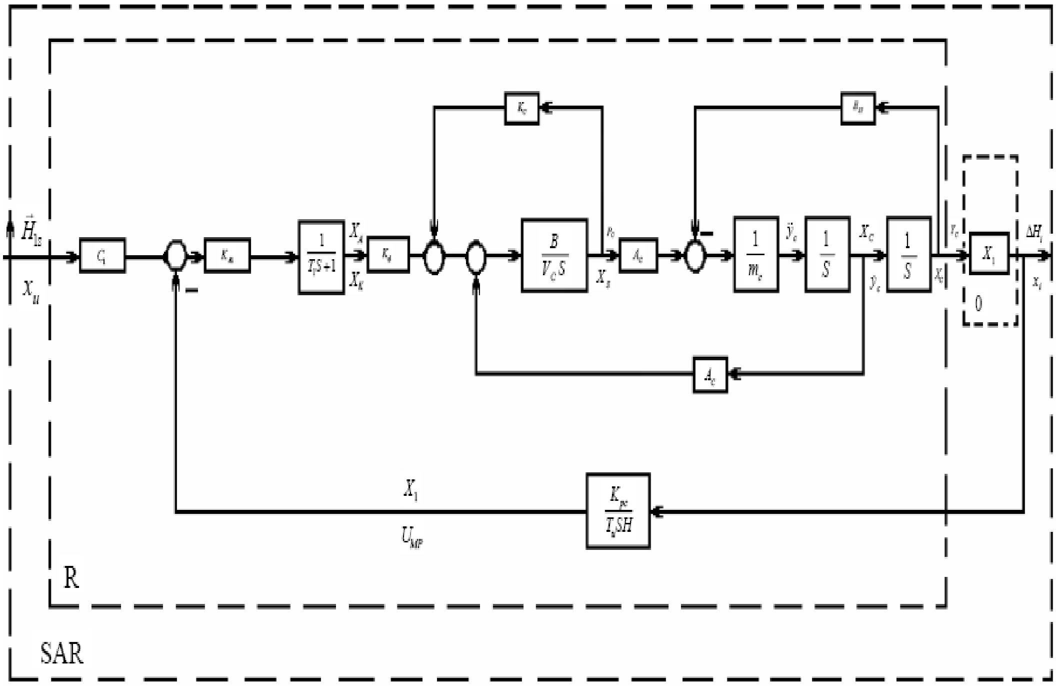

Figure 8. Automatic controlscheme- blockdiagramsconnection

Define state variables:

$$x_1 = u_{MP};\ x_2 = y_c;\ x_3 = y_c;\ x_4 = x_K;\ x_5 = p_c. \quad (21)$$

Input and output are:

$$x_u = h_{iz},\ x_i = h_i. \quad (22)$$

Notice that all state variables are easily measurable quantities (voltage from the position gage, piston rod velocity, piston rod position,servovalve spool position and the pressure in the cylinder). Output value is the change of the strip thickness and input value is demand change of the strip thickness. Then,from the definition of the state variables (21), equations (20) and figure 9, we obtain equations (23) and (24):

$$\dot{x}_1 = \frac{1}{T_u}x_1 + \frac{K_{pc}}{T_u}a_1x_3,\ \dot{x}_2 = -\frac{E_{ST}}{m_c}x_3 + \frac{A_c}{m_c}x_5,\ \dot{x}_3 = x_2, \quad (23)$$

$$\dot{x}_4 = \frac{K_A}{T_1}x_1 - \frac{1}{T_1}x_4 - \frac{C_1K_A}{T_1}x_u,\ \dot{x}_5 = -\frac{BA_c}{V_c}x_2 - \frac{B}{V_c}K_qx_4 + \frac{K_cB}{V_c}x_5, \quad (23\text{-}1)$$

$$x_i = ax_3. \quad (24)$$

IX. SIMULATION OF THE SYSTEM WORK BY COMPUTER AND EXPERIMENTAL MEASUREMENTS IN REAL SYSTEM

According to system of equations (22) we obtain step response by computer (for the calculation of the all necessary coefficients we use dates from lit ([1,3]). For the same demand strip thickness change of $0{,}5\text{-}10^{-3}$ mm was done series of experimental testing and result of that is shown at the figure 9, too. Comparison of this curves gives conclusion that nature of this curves are the same. Small difference between these curves is induced by impossibility of exactly determination and exactly testing the system characteristics.

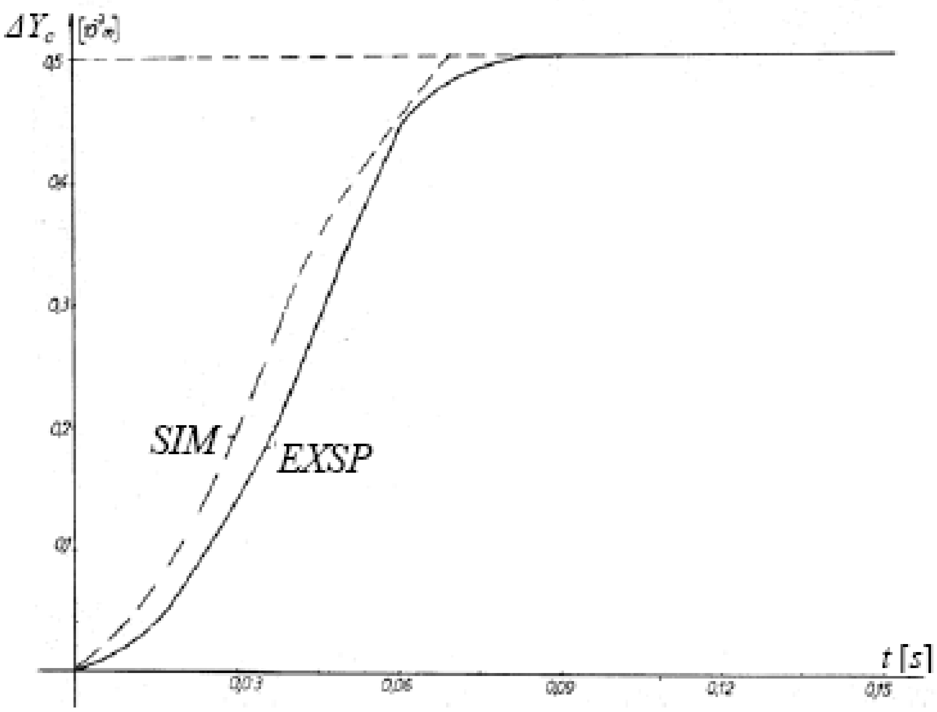

Figure 9. Step responses obtained by simulation and experimental measurements in the plant

X. CONCLUSION

Comparison of theoretical and experimental curves gives conclusion that all introduced assumptions are good. Analysis of the all system characteristics gives conclusion that system has good working. Electrohydraulic servosystem for the AGC has better characteristics than electromechanic system (five times greater speed of rolling, greater speed of positioning ,smaller dead–zone, smaller time of roll gap adjusting start, smaller time of maximum speed reaching,greater unloading speed...).

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