MODELING AND CONTROL OF PLATE THICKNESS IN HOT ROLLING MILLS

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Abstract: It is still common practice in industry to handle the thickness control problem of a rolling mill around a working point. The assumption of linearity is no longer valid if the mill is operated in a wider working range. In a reversing mill this is especially the case when the plate enters and leaves the mill stand from pass to pass. A non-linear control strategy is proposed to improve the regularity of the thickness at the ends of the plate in order to decrease the amount of the plate to be cut off by the crop shear. The performance of the controller is examined using an advanced simulation model of a finishing mill.

Keywords: hot rolling, reversing finishing mill, heavy plate thickness control, non-linear control, regularity of plate ends

1. INTRODUCTION

With a heavy plate production of 2 mill. tons in 2003, DILLINGER HUETTE GTS (DH) is the major European heavy plates producer. In the plants in Dillingen, Germany, and Dunkirk, France, plates of big dimensions are produced. Lengths of up to 50 meters, widths of up to 5.5 meters and thicknesses coming up to 400 millimeters are by no means uncommon. There is an ever increasing demand for higher quality of the rolled steel products on the market and for higher productivity of the plants. From a control theoretic point of view quality primarily is a byword for holding the closest possible plate thickness tolerances. Depending on the special type of rolling, whether hot or cold, whether thin strip or heavy plate, different effects limit the achievable performance of the overall closed-loop control system. Several MIMO (multi-input multi-output) controllers have successfully been designed for multi-stand mills, see, e.g., (Nakagawa *et al.*, 1990) or the H_∞-approach in (Grimble and Hearns, 1999), to control the thickness and the interstand tension, and for single-stand mills, see, e.g., (Pedersen and Wittenmark, 1998), to enhance the product quality over the plate width. Often controllers are based on linear process models, what in general is no big drawback, as in many cases the mill is predominantly operated close to a fixed working point. For scenarios where the mill is operated in a wider working range it is useful to take into account the non-linear behavior of the system, see., e.g., (Kugi *et al.*, 2000). When rolling plates in a reversing mill the assumption of linearity certainly is violated whenever the plate enters and leaves the mill stand. When entering with the leading plate end - in the following called plate head - the rolling force rapidly meets the desired force of, in our case, up to more than 90 MN. The leaving end is referred to as the plate tail in the following. The entering of the plate is surely the biggest but not the only disturbance to the process. As there may be several minutes between the moment when the

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plate leaves the reheat furnace and the end of its pass schedule its edges become significantly colder than its filet part. The huge resulting hardness variations are a second disturbance to the process which in context with conventional control concepts yield a considerable over thickness, in particular at the ends of a plate. While the mean temperature profile of the plate is more or less well known it is in contrast not possible to measure the plate thickness directly in the roll gap due to the hostile industrial environment. Often, as in our case, the only thickness measuring device is located many meters behind the mill stand and cannot be used for control purposes within one pass. Thus, there is a strong need for well adapted models and for a deep understanding of the process in order to develop new control concepts for the stand which help to increase the productivity of the plant. To achieve the highest plant performance and the best product quality even in future times, we choose to make a mechatronic approach where the constructional setup as well as the controllers, the actuators and sensors are redesigned and optimized from a synergetic point of view.

The paper is organized as follows: In a first step, a mathematical model of the mill stand and its components is given in Section 2. Special emphasis is laid on the derivation of a rather simple dynamic mill stand model considering all essential effects arising in practice. In Section 3 we will compare some simulation results of the plate thickness with measured data at the plate head to validate the derived model. Section 4 is devoted to the controller design. The controller consists of a servo compensating pressure controller in the innermost control loop and a non-linear outer control loop for the plate thickness. A few simulation results are presented in Section 5 and a short discussion on the control concept and the simulation results is given in Section 6.

2. MATHEMATICAL MODELING

The mill stand under investigation is a four-high stand equipped with an upper and a lower bending traverse. The principal mill configuration is depicted in fig. 1. The back-up rolls serve to avoid too strong bending of the work rolls and are itself supported by the traverses and the hydraulic back-up roll bending (BURB) systems. The bending of the work rolls can be influenced with hydraulic work roll bending (WORB) systems. Chocks in which the rolls are mounted allow the rolls to move vertically in the mill housing. The upper elements (traverse, back-up roll and work roll) can be positioned outside a pass with a screw system, while a hydraulic positioning system allows to change the position of the lower elements during a pass. A detailed mathematical formulation of the components of the mill is given in the following.

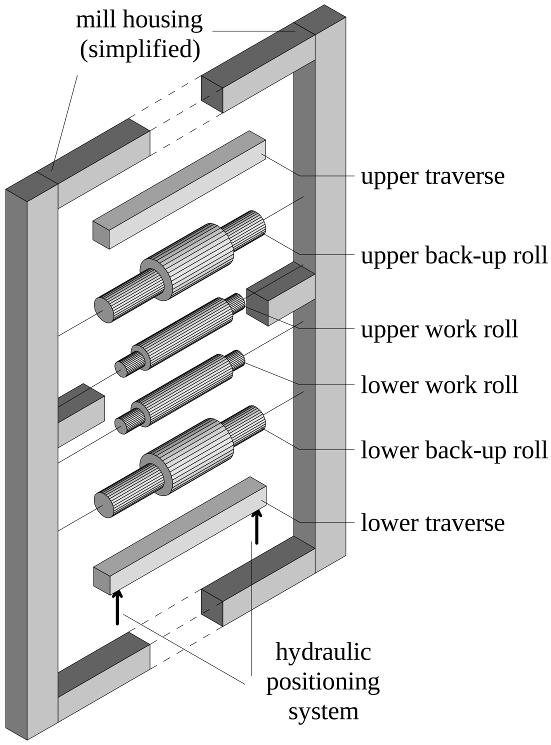

Fig. 1. Four-high stand of the finishing hot rolling mill in Dillingen.

2.1 *Mill Stand Model*

On the basis of the mill configuration as depicted in fig. 1 a simplified mill model is derived. Here we are only interested in the vertical behavior of the mill. The thickness profile of the plate in the horizontal direction is not considered in this contribution. The schematic diagram of the mill stand model is shown in fig. 2. The mass m_2 represents the mass of the lower mill housing, the mass m_4 is built up by the upper mill housing and the upper bending traverse. The remaining elements are summarized in the masses m_1 and m_3. The safety jackets are mounted between the upper bending traverse and the back-up roll chocks. They are not actuated during a pass. For the depicted model the equations of motion read as

$$\begin{aligned} \frac{\mathrm{d}}{\mathrm{d}t}x_i &= v_i, \qquad i = 1, ..., 4 \\ m_i \frac{\mathrm{d}}{\mathrm{d}t}v_i &= -m_i g + F_i, \qquad i = 1, ..., 4\,, \end{aligned} \tag{1}$$

with the gravitational constant g and the abbreviations

$$\begin{aligned} F_1 &= F_H - F_R - F_{f_A} + F_{1c} \\ F_2 &= -F_H + F_{f_A} + F_{LH} - d_2 v_2 + F_{2c} \\ F_3 &= -F_{SJ} + F_R - F_{f_B} + F_{3c} \\ F_4 &= F_{SJ} + F_{f_B} - F_{UH} - d_4 v_4 + F_{4c}\,. \end{aligned} \tag{2}$$

Here, $F_H = A_H p_H$ and $F_{SJ} = A_{SJ} p_{SJ}$ denote the hydraulic forces of the HGC (hydraulic gap control) cylinder and the safety jacket, F_R is the rolling force, d_2 and d_4 are viscous damping coefficients of the stand. The forces exerted by the upper and lower housing stretch elements are given by the inverse of

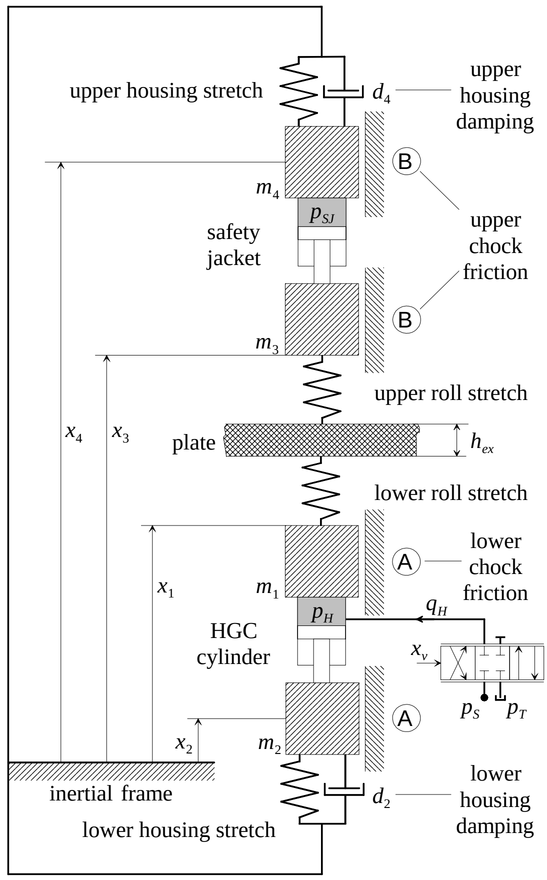

Fig. 2. Schematic diagram of the mill stand model.

the corresponding stretch functions f_{UH} and f_{LH} and the initial positions $x_{2,0}$ and $x_{4,0}$ of the masses m_2 and m_4 in the form

$$F_{UH} = f_{UH}^{-1}\left(x_4 - x_{4,0}\right) \tag{3}$$

$$F_{LH} = f_{LH}^{-1}\left(x_{2,0} - x_2\right) \ . \tag{4}$$

In $F_{ic}, i = 1, ..., 4$, all forces are summarized which are constant within one pass (such as balancing and bending forces). For the chock friction forces F_{f_A} and F_{f_B} a static friction model is used, the parameters of which are estimated from measurement data. Stick-slip effects are considered in the simulation model, while only viscous friction is assumed for the controller design. Given the stretch of the upper rolls

$$s_{UR} = f_{UR}\left(F_R, F_{3c}, F_{4c}\right) \tag{5}$$

and of the lower rolls

$$s_{LR} = f_{LR}\left(F_R, F_{1c}, F_{2c}\right) \tag{6}$$

the plate exit thickness h_{ex} is given by

$$h_{ex} = \left(x_3 + s_{UR}\right) - \left(x_1 - s_{LR}\right) \ . \tag{7}$$

With the stretch functions f_{UR} and f_{LR} different stretch effects are considered, e.g. the bending of the rolls due to BURB and WORB forces, the flattening between back-up and work rolls and the flattening of the work roll sides facing the plate. The stretch functions are physically motivated and their final representation is adapted by means of finite-element studies and quasi-static measurements. As the only measurable quantities are the displacement of the HGC cylinder $x_H = x_1 - x_2$, the displacement of the safety jacket $x_{SJ} = x_4 - x_3$, the pressure in the safety jacket p_{SJ} and the pressure in the HGC cylinder p_H, we rewrite (7) in the following form

$$\begin{aligned} h_{ex} = {} & URG - \left(x_H - x_{H,0}\right) - \left(x_{SJ} - x_{SJ,0}\right) \\ & + s_{UR} + s_{LR} + f_{UH}\left(F_{UH}\right) + f_{LH}\left(F_{LH}\right) \ . \end{aligned} \tag{8}$$

Here $URG = x_{3,0} - x_{1,0}$ denotes the size of the unloaded roll gap. This is the size of the roll gap before the plate enters the mill stand. In conventional mill automation systems (8) is treated quasi-statically by assuming all forces applied to the stretch functions to be equal to a measurable force, mostly F_H. Then, usually a linear thickness controller, mostly a PI-controller, is designed giving the reference signal to a controller for the displacement of the HGC cylinder x_H in the innermost control loop. This control concept is often referred to as gaugemeter control.

2.2 *Hydraulic Positioning System*

The mill under investigation is actuated by a single-acting, single-ended hydraulic piston actuator which allows to position the lower rolls at forces of up to $45\,\mathrm{MN}$ per mill side. All following considerations can easily be extended to the more general case, where especially for mills working at lower rolling forces the hydraulic positioning system consists of double-acting, double-ended hydraulic cylinders, see, e.g., (Kugi, 2001). The mass continuity equation yields

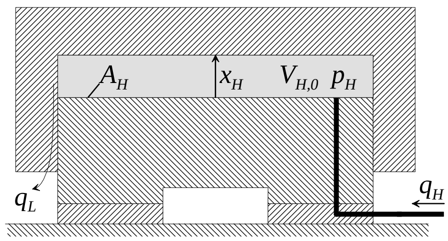

Fig. 3. Single-acting, single-ended HGC cylinder.

the differential equation for the pressure p_H in the cylinder chamber

$$\frac{\mathrm{d}}{\mathrm{d}t} p_H = E_H \frac{\left(q_H - q_L - A_H v_H\right)}{V_{H,0} + A_H x_H} \ , \tag{9}$$

where $v_H = \mathrm{d}x_H/\mathrm{d}t$ is the velocity, q_H is the volume flow to the cylinder chamber, $V_{H,0}$ is the compressible oil volume of the pipeline elements and A_H denotes the piston area. Furthermore, E_H is the isothermal bulk modulus of the oil and q_L is the leakage flow out of the chamber which will be neglected in what follows with no substantial restriction of generality. The volume flow to the cylinder coming from the servo valve reads as

$$q_H = f_{V_+}(x_v)\sqrt{p_S - p_H} - f_{V_-}(x_v)\sqrt{p_H - p_T} \tag{10}$$

with the supply and tank pressure p_S and p_T, the servo spool displacement x_v and the opening characteristics of the servo valve $f_{V_+}(x_v)$ and $f_{V_-}(x_v)$, with $f_{V_+}(x_v) = 0$ for $x_v < 0$ and $f_{V_-}(x_v) = 0$ for $x_v \geq 0$. In our case the valve is a three-stage critical center 4/3 directional valve with one closed connection. The valve is working at a supply pressure of $p_S = 450\,\mathrm{bar}$ with a rated flow of $700\,\mathrm{l/min}$. The pressures p_S and p_T are measurable. All simulations are performed using further models for the hydraulic supply system which are not presented in this contribution. The simulation model additionally contains a dynamic model for the servo valve. But for the controller design the servo valve is assumed to be ideally fast and thus x_v serves as the control input.

2.3 *Rolling Force Model*

Up to now, an exact analytical solution of the deformation process in the roll gap is not known. Depending on the special rolling scenario a variety of approximate solutions can be found in the literature. In this contribution we use the model of *Sims* which is one of various different models used for the hot rolling process of steel. The interested reader may find further details in the wide literature on rolling processes, e.g. in the original publication (Sims, 1954) and the references cited therein. We only want to briefly summarize the essential parts of the model here. Fig. 4 depicts a schematic diagram of the roll gap. Sims' model assumes the plate deformation to be

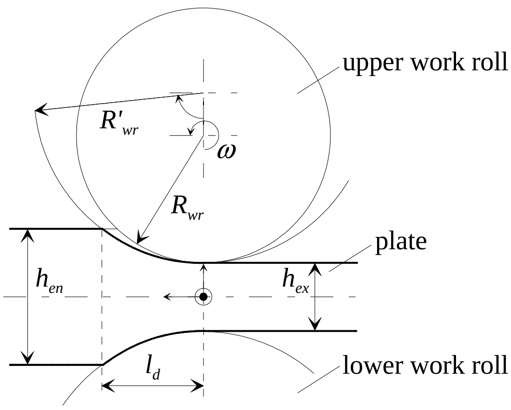

Fig. 4. Schematic diagram of the roll gap.

purely plastic. In consideration of the yield condition derived by *Orowan* and making some assumptions on the friction between the work rolls and the plate, Sims calculates the rolling force F_R by integrating *von Karman's differential equation* to, see, e.g., (Hensel and Spittel, 1978)

$$F_R = b_m\, l_d\, k_{fm}\, Q_f\ . \tag{11}$$

Here, b_m denotes the plate width, l_d is the length of the arc of contact, k_{fm} is the average yield strength in the roll gap and Q_f is a factor depending on the geometry of the roll gap. The length of the arc of contact can approximately be calculated by

$$l_d \approx \sqrt{R'_{wr}\triangle h} = \sqrt{R'_{wr}(h_{en} - h_{ex})}\ , \tag{12}$$

where the radius R'_{wr} of the deformed work roll is given as a function of the radius of the undeformed work roll R_{wr} by *Hitchcock's formula*

$$R'_{wr} = R_{wr}\left(1 + \frac{16\left(1-\nu^2\right)}{\pi E}\frac{F_R}{b_m(h_{en} - h_{ex})}\right)\ , \tag{13}$$

with the entry and exit thickness of the plate h_{en} and h_{ex}. Furthermore, E denotes *Young's modulus* and ν is the *Poisson ratio* of the rolls. The average yield strength k_{fm} is a function of the temperature T of the plate in the rolling gap, the logarithmic pass reduction $\varepsilon = \log\left(\frac{h_{en}}{h_{ex}}\right)$ and the circumference velocity of the work rolls ω. Following the propositions of *Hill* and *Siebel*, see, e.g., (Schwenzfeier, 1979), the geometry factor Q_f has the form

$$\begin{aligned} Q_f &= x_{Q_f} + B_1 \exp\left(B_2\frac{l_d}{h_m} + B_3\right) \\ x_{Q_f} &= B_4 + B_5\frac{l_d}{h_m}\ , \end{aligned} \tag{14}$$

with the mean plate thickness $h_m = (h_{en} + h_{ex})/2$ and the characteristic ratio of the length of the arc of contact and the mean plate thickness l_d/h_m. The parameters $B_1, ..., B_5$ are found by means of regression analysis. With (11) - (14) we can formulate the rolling force model in an implicit form

$$f_{F_R}(F_R, h_{en}, h_{ex}, T, \omega) = 0 \tag{15}$$

which will be used both for simulation purposes and for the controller design. For all simulation studies the length of the arc of contact l_d from (12) is assumed to increase and decrease in a smooth manner at the beginning and the end of each pass.

3. MODEL VALIDATION

To validate the dynamical mathematical model of the mill stand derived in the previous section, measurement campaigns have been performed. The plate thickness was measured with a radiometric thickness measuring device located several meters behind the mill stand. The plates were rolled using a conventional gaugemeter controller. The simulated and measured plate thickness of the plate head for one characteristic pass are compared in fig. 5. The abrupt beginning of the measured profile is due to a delay in the thickness measuring device. Simulated and measured thickness profiles match with quite good accuracy which also motivates to use the models derived so far for the controller design in the next section. Note that of course the plates delivered to the customers are always cropped and thus meet the highest quality requirements.

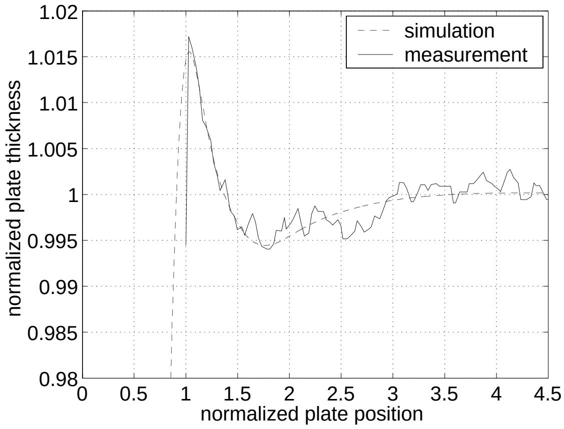

Fig. 5. Comparison of simulation results with radiometric thickness measurement data for the uncropped plate head of a characteristic pass.

4. CONTROLLER DESIGN

The controller is based on a cascaded concept with a non-linear pressure controller in the innermost control loop and an implicit non-linear controller for the plate thickness in the outer control loop. Applying the theory of exact linearization, see, e.g., (Isidori, 1996), to (9) and (10), we construct a simple servo compensating pressure controller yielding a closed loop

$$\begin{aligned} \frac{\mathrm{d}}{\mathrm{d}t} p_H &= \bar{q}_H = -k_p \left(p_H - F_{H,d}/A_H \right), \quad k_p > 0 \\ q_H &= A_H v_H + \bar{q}_H \frac{V_{H,0} + A_H x_H}{E_H} \\ x_v &= \begin{cases} f_{V_+}^{-1} \left(q_H / \sqrt{p_S - p_H} \right) & , \quad x_v \geq 0 \\ -f_{V_-}^{-1} \left(q_H / \sqrt{p_H - p_T} \right) & , \quad x_v < 0 \end{cases} , \end{aligned} \tag{16}$$

where $F_{H,d}$ denotes the desired hydraulic force of the HGC cylinder and serves as the control input for the outer plate thickness controller. The controller parameter k_p has to be chosen appropriately. In case the piston position signal x_H is strongly corrupted by noise and cannot be differentiated w.r.t. time one can find elegant methods to overcome this problem, see, e.g., (Schlacher *et al.*, 2001) and (Kugi, 2001). For the design of the outer controller the profiles of the plate temperature T and the plate entry thickness h_{en}, the circumference velocity of the work rolls ω and the forces $F_{ic}, i = 1, ..., 4$, are assumed to be known. The forces F_{UH} and F_{LH} exerted by the stretched housing elements cannot be measured. They can either be substituted quasi-statically by a measurable force, e.g. F_H or F_{SJ}, or can be observed with a trivial observer. For the sake of convenience we assume that the chock friction forces $\tilde{F}_{f_A} = d_1 v_1$ and $\tilde{F}_{f_B} = d_3 v_3$ are purely viscous and that the roll stretch elements are flipped to the sides of the masses m_1 and m_3 where the hydraulic cylinders are located. On these conditions the mathematical model of the plate thickness (cp. (1) - (2)) for the controller design can be written as

$$\begin{aligned} \frac{\mathrm{d}}{\mathrm{d}t} h_{ex} &= v_{ex} = v_3 - v_1, \quad h_{ex} = x_3 - x_1 \\ \frac{\mathrm{d}}{\mathrm{d}t} v_{ex} &= \frac{-m_3 g - F_{SJ} + F_R - \tilde{F}_{f_B} + F_{3c}}{m_3} \\ &\quad - \frac{-m_1 g + F_H - F_R - \tilde{F}_{f_A} + F_{1c}}{m_1} . \end{aligned} \tag{17}$$

Based on the idea of flatness, see, e.g., (Fliess *et al.*, 1995) and (Rudolph, 2003), for the simplified model (17) an implicit non-linear control law can be designed in the following way: By applying a desired hydraulic force

$$\begin{aligned} F_{H,d} &= F_R - F_{1c} + m_1 g - v_{ex,d} d_1 - m_1 \chi \\ &\quad + \frac{m_1}{m_3} \left(F_R - F_{SJ} + F_{3c} - m_3 g \right) \end{aligned} \tag{18}$$

to the system with F_R from (15), with h_{ex} from (8) and assuming $m_1 = m_3$ and $d_1 = d_3$, we get

$$\frac{\mathrm{d}}{\mathrm{d}t} v_{ex} = \frac{d_1}{m_1} \left(v_{ex,d} - v_{ex} \right) + \chi \, . \tag{19}$$

Choosing χ in the form

$$\chi = \frac{\mathrm{d}}{\mathrm{d}t} v_{ex,d} - \left(\alpha_1 e + \alpha_2 \int e \mathrm{d}t \right) - \alpha_3 \frac{\mathrm{d}}{\mathrm{d}t} e \tag{20}$$

with $e = h_{ex} - h_{ex,d}$ the error dynamics of the outer closed loop system is given by

$$\frac{\mathrm{d}^2}{\mathrm{d}t^2} e + \left(\frac{d_1}{m_1} + \alpha_3 \right) \frac{\mathrm{d}}{\mathrm{d}t} e + \alpha_1 e + \alpha_2 \int e \mathrm{d}t = 0 \, , \tag{21}$$

provided that the inner pressure controller is assumed to be ideal. The error dynamics can be adjusted by an appropriate choice of the controller parameters α_1, α_2 and α_3. Anti-wind up strategies have to be arranged for the integral part in (21) as it may happen that the servo spool position saturates (see also fig. 8).

5. SIMULATION RESULTS

For the simulations to be presented in the following the input thickness h_{en} of the tail was chosen to be equal to the exit thickness h_{ex} of the head in the previous pass (see fig. 5). The temperature profile was assumed to decrease to 70% of the filet temperature at the plate tail in a smooth manner. As depicted in fig. 6, simulation studies show that the new non-linear controller significantly enlarges the plate length within the required thickness tolerances compared to a conventional gaugemeter controller. The corresponding rolling force is depicted in fig. 7. The servo spool position x_v for both controllers is given in fig. 8.

6. CONCLUSION AND OUTLOOK

It is interesting to note that in the case when the dynamics in (17) is neglected the controller presented here merges into the conventional gaugemeter controller. Compared to conventional mill automation

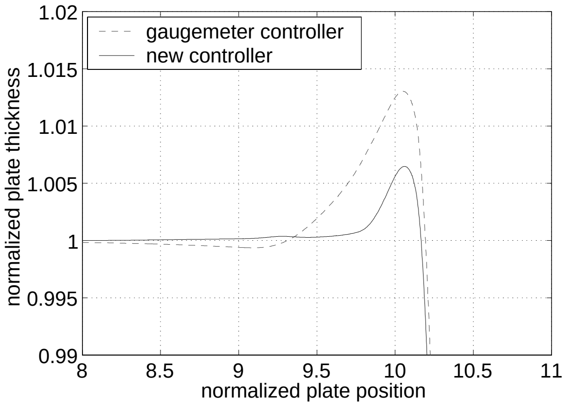

Fig. 6. Thickness of an uncropped plate tail with conventional gaugemeter controller and with new non-linear controller.

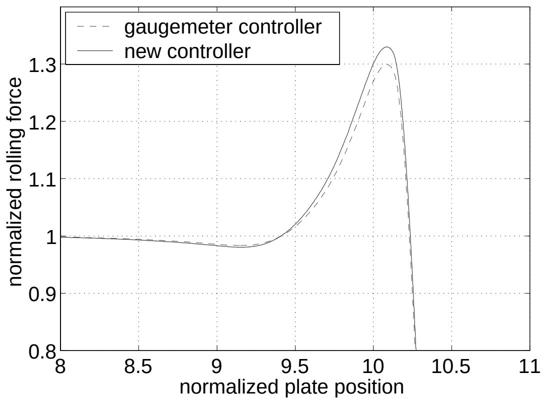

Fig. 7. Rolling force with conventional gaugemeter controller and with new non-linear controller.

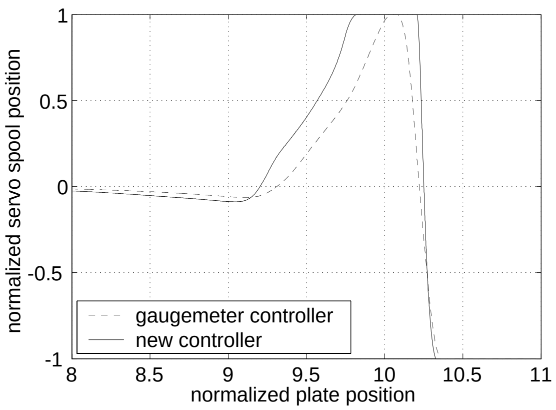

Fig. 8. Servo spool position with conventional gaugemeter controller and with new non-linear controller.

systems we use the actual non-linear mill stretch functions, which are used on-line for the pass schedule calculation, and a rolling force model instead of the usual deformation resistance and mill stretch coefficient. For the presented controller to be feasible in practice a forward slip model is required to track the position of the plate. Furthermore, the entry thickness and temperature profiles are needed. The control concept in combination with the advanced simulator also allows us to further investigate the influence of the performance of certain system components on the product quality and on the plant productivity, see, e.g., the saturating servo valve in fig. 8. Thus, apart from the control synthesis point of view this approach is also of great value when dealing with questions of optimal sensor and actuator positioning and the re-design of mill components.

7. ACKNOWLEDGEMENTS

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