NONLINEAR CONTROL OF THE INTERSTAND LOOPER IN HOT STRIP MILLS: A BACKSTEPPING APPROACH

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Abstract: A hot strip mill for the production of steel is a complex process where nonlinear dynamic phenomena occur and many control loops need to be tuned taking into account their interference. The control problem of the interstand tension and looper arm position is addressed in this paper. With respect to the conventional PID control approach, our investigation aims at exploiting nonlinear control techniques based on the backstepping methodology in order to improve the performance along with the requirements of the most modern rolling mills where short off-gauge lengths are desired even for thin products (0.7-1.2 mm) and for *setup on the fly* of the plant. The design of the controller is described in detail and several simulation trials are reported showing the effectiveness of the control scheme under the action of typical disturbance inputs.

Keywords: Steel Rolling Mills, Tension Control, Nonlinear Control.

1. INTRODUCTION

A hot strip mill is one of the most effective tools in flat rolling. The last part of a hot strip mill – the finishing mill, allows reducing the thickness of the flat steel to the planned value by means of consecutive rolling stands reaching even very thin values. This complex process presents several control aspects that are continuously subject to research activities in order to achieve excellence in quality of the produced steel and in achieving improved productivity of the plant.

In the finishing mill, to achieve the required reduction and the final qualities and tolerances, several passes of rolling are executed by tandem rolling with 5–7 successive stands.

Tension control turns out to be the key to stable mill. For instance, we experienced that during *Automatic Gauge Control* (AGC) tuning, strip tension can present wide fluctuations due to quick roll gap movements. On the other hand, when tension fluctuations are kept limited as much as possible, the performance of the AGC system can be improved as desired. Proper positioning of the looper is also important for stable operations, so the problem is to design a simultaneous control scheme for looper position and interstand tension.

In the literature, many control techniques have been proposed for the control of the interstand looper (see, for instance (An *et al.*, 2001; Asano *et al.*, 2000)). In particular, multivariable control techniques have been proposed to take into account the interaction between these two main control loops (see e.g. (Cuzzola and Boriani, 2003; Hearns *et al.*, 2004; Fukushima *et al.*, 1988; Hearns and Grimble, 2000)). Multivariable controllers have been experimented in real steel plants (for instance, the Sovel SA mill, Greece and the Algoma Mill, Canada, have been equipped of multivariable controllers).

The introduction of a nonlinear control technique (see (Hesketh *et al.*, 1988),(Furlan *et al.*, 2004) and the references cited therein for a recent review about the topic) leads to two further advantages. First of all, the controller is not based on a linearized model and, consequently, it is expected to be more efficient in a wide range of working situations. Secondly, the application of a *gauge setup "on the fly"* used to increase the productivity of a plant, produces transient modes of behavior for which the nonlinear effect can hardly be considered negligible and needs a more accurate compensation based on nonlinear control techniques, see (Kugi *et al.*, 2004).

In the paper, due to the particular structure of the dynamic model of the interstand looper, the nonlinear control scheme is designed according to a standard backstepping methodology (see (Krstić *et al.*, 1995; Hesketh *et al.*, 1988)) which turned out to be particularly effective.

The paper is organized as follows. In Section 2 the basic control problem under concern will be formulated while in Section 3, the modelling of the interstand dynamics will be addressed. Subsequently, in Section 4, the design of the backstepping controller will be extensively illustrated and, finally, simulation results showing the effectiveness of the control scheme will be reported in Section 5.

NOTATIONS

- h^i strip exit thickness for the i-th stand;
- σ strip tension;
- w is the strip width;
- v^i strip exit speed to the i-th stand;
- V^{i+1} strip entry speed to the $(i+1)$-th stand;
- r radius of the looper roll;
- R work roll radius;
- θ looper angular position;
- M_r looper roll mass;
- M_b looper arm mass;
- M_{cp} looper counterweight mass;
- l looper length ;
- l_{cp} counterweight length;
- T_u the motor torque on the looper;
- η the looper angular speed;
- ρ the steel density.

2. THE TENSION CONTROL PROBLEM IN THE FINISHING MILL

As is well known, the looper is located approximately midway between adjacent stands of the steel finishing mill. The looper is raised above the pass line so that it forms a loop of the stored strip.

Tension control in finishing mills is very important owing to the following facts:

(1) The interstand tension must be kept as much as possible constant in order to avoid excessive fluctuations of the AGC and to guarantee the safety of the plant;
(2) closed–loop regulation of the interstand tension allows for independent thickness corrections to be made on individual stands while maintaining smooth mass flow through the mill.

The following aspects should also be taken into account in the design of the controller for the looper.

(3) Assuming negligible plastic deformations of the steel, the tension is governed by stretch and the elastic constant of the steel (Young Modulus). The stretch is determined by the difference between the stored strip length and the loop length formed in the in the two stands by the looper.
(4) the stored strip length between the stands is given as the time integration of the difference in the strip velocity between the upstream and downstream stands.
(5) the loop length can be adjusted by manipulating the looper angle, whereby stored length changes can be absorbed.

Since excess fluctuations of the angle may lead to unstable rolling, the looper angle should also be considered a controlled variable. Therefore, the typical control problem is to regulate the looper angle and tension simultaneously. In the next section, a state–space model of the interstand dynamics will be briefly described. This model will be subsequently used to design the nonlinear controller.

3. MATHEMATICAL MODELLING OF THE INTERSTAND DYNAMICS

In Fig. 1, a pair of consecutive rolling stands (with the looper as well) is depicted; the strip is depicted by the straight line. The geometrical quantities are defined in the figure and will be used to design the overall dynamic model. In the following, the dynamic models for each part of the system will be devised.

The looper dynamic behavior is described by the following law

$$J\ddot{\theta} = T_u - T_{load} - T_d + \omega_\eta$$

where:

- J is the looper inertia;
- T_{load} is the load torque;
- T_d represents the friction effect on the looper;
- ω_η represent the model uncertainties.

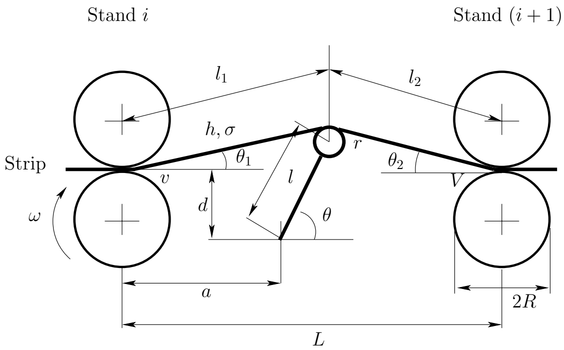

Fig. 1. Looper and interstand geometry.

The load torque is due to several phenomena

$$T_{load} = T_\sigma + T_s + T_{lw} + T_b$$

where:

- T_σ is the torque due to to the strip tension;

$$T_\sigma = \sigma w h F(\theta)$$

  and $F(\theta)$ is the arm of the strip tension with respect to the looper focus:

$$F(\theta) = [\,(l\sin\theta + r)(\cos\theta_2 - \cos\theta_1) + l\cos\theta(\sin\theta_1 + \sin\theta_2)\,]$$

- T_s is the torque due to the strip weight P_s

$$T_s = P_s l\cos\theta = \rho g w h[l_1(\theta) + l_2(\theta)]l\cos\theta$$

- T_{lw} is the torque due to the weight of the looper

$$T_{lw} = g\cos\theta\left[l(M_r + \frac{M_b}{2}) - l_{cp}M_{cp}\right]$$

- T_b is the strip bending torque.

During transient mode of behavior, the loop length L_s is subject to change if the exit strip speed of the upstream stand is different wrt the entry strip speed for of the downstream stand according to

$$L_s = L + \xi(t) = L + \int_0^t (v^i - V^{i+1})d\tau$$

where L is the initial loop length and consequently

$$\dot{\xi}(t) = v^i - V^{i+1} + \omega_\xi$$

where the term ω_ξ denotes the unmodeled dynamics.

Assuming an elastic behavior of the material, σ can be derived from the Young Modulus (E) of the steel according to

$$\sigma = \frac{E}{L + \xi(t)}\left[F_2(\theta) - \xi(t)\right]$$

with $F_2(\theta) = L_s - L$ and $L \gg \xi(t)$. It is worth pointing out that the Young Modulus cannot be considered independent of the steel grade or the strip temperature but in the following we will assume it uniform along the strip. From the last equation is it easy to derive the dynamic law valid for the strip tension

$$\dot{\sigma} = F_1\left[\frac{\partial F_2(\theta)}{\partial\theta}\dot{\theta} - \dot{\xi}(t)\right] = F_1\left[\frac{\partial F_2(\theta)}{\partial\theta}\eta - v^i - V^{i+1} + \omega_\xi\right]$$

where $F_1 = \dfrac{E}{L}$ is a constant.

Summing up, the dynamics of the considered nonlinear system to be controlled can be described by means of the following nonlinear state equations:

$$\begin{cases} \dot{\theta}(t) = \eta(t) \\ \dot{\eta}(t) = \frac{1}{J}\left[T_u - T_{load} + \omega_\eta\right] \\ \dot{\sigma}(t) = F_1\left[F_3(\theta)\eta - \overline{v}(t) - \omega_\xi\right] \\ \dot{\overline{v}}(t) = -K\overline{v}(t) + Ku_v(t) \end{cases} \quad (1)$$

where $\bar{v} = v^i - V^{i+1}$ and the parameter K is used to describe the closed loop effect of the stand Automatic Speed Regulator (ASR).

The control problem addressed in the next section will consist in designing the control signals T_u and u_v in order to regulate both the strip tension and the looper angular position.

4. DESIGN OF THE NONLINEAR CONTROLLER

First, we note that the *Looper Current Control* yielding the looper torque has a very fast dynamics. Therefore, we can write $T_u = c_l u_1$ where c_l is a known constant gain. Hence we have

$$\dot{\eta} = \frac{1}{J}\left[c_l\, u_1(t) - F_3(\theta)\sigma - T_s - T_{lw} + \omega_\eta\right]$$

Letting

$$u_1(t) = \frac{1}{c_l}\left[F_3(\theta)\sigma_{ref} + T_s + T_{lw}\right] + \frac{J}{c_l}\hat{u}_1(t)$$

where $\hat{u}_1(t)$ is the new control signal, we obtain the state equation

$$\dot{\eta}(t) = \frac{1}{J}F_3(\theta)\left[\sigma_{ref} - \sigma(t)\right] + \hat{u}_1(t)$$

with σ_{ref} denoting the strip tension reference. The control objective is to design a nonlinear controller providing the control signals $\hat{u}_1(t)$ and $u_v(t)$ that keep the tension error as small as possible while guaranteeing a stable behavior for the looper dynamics.

By introducing the error variables[1] $\theta_e(t) \triangleq [\theta(t) - \theta_{ref}]$ and $\sigma_e(t) \triangleq [\sigma(t) - \sigma_{ref}]$, the state equations can be rewritten as

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[1] All the figures reported in this paper refer to the SOVEL SA mill, Greece. In the following we consider the set point $\theta_{ref} = 25$ [°], $\sigma_{ref} = 3.9$ [MPa].

$$\begin{cases} \dot{\theta}_e = \eta \\ \dot{\eta} = -\frac{1}{J} F_3(\theta)\sigma_e + \hat{u}_1 \\ \dot{\sigma}_e = F_1\big[F_3(\theta)\eta - \bar{v} - \omega_\xi\big] \\ \dot{\bar{v}} = -K\bar{v} + Ku_v \end{cases} \tag{2}$$

If we set the control variables to zero, then $[\bar{\theta}_e, 0, 0, 0]$ is an equilibrium point for any possible value of $\bar{\theta}_e$. This means that a stabilizing controller will achieve its objective irrespective of the actual value of the looper angle θ. In practice, this is not an issue because the goal is to control the tension with a stable dynamics of the looper and the actual angle is not of interest provided it belongs to some reasonable interval (e.g., $10° \div 70°$).

Now, we design the controller using a backstepping (BS) procedure (see, for instance, (Krstić *et al.*, 1995)). To this end, for the sake of notational convenience, we rewrite (the meaning of the symbols is obvious) the state equations (2) as

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = -\frac{1}{J} F_3(x_1)x_3 + \hat{u}_1 \\ \dot{x}_3 = F_1\big[F_3(x_1)x_2 - x_4\big] \\ \dot{x}_4 = -Kx_4 + Ku_v \end{cases} \tag{3}$$

where the uncertainty term ω_ξ will be neglected in the following and will be re-considered in the simulation analysis. Dynamic system (3) is in strict–feedback form. Therefore the integrator–backstepping procedure can be applied without conceptual difficulties.

The first BS step is very easy. By setting

$$\hat{u}_1 = -k_1 x_2 \tag{4}$$

where $k_1 > 1$ is a constant gain, we ensure that the dynamic behavior of the first state variable obeys the asymptotically stable law $\dot{x}_1 = -k_1 x_1$. Now we let $z_1 \triangleq x_2 - k_1 x_1$ and then

$$\dot{z}_1 = -\frac{1}{J} F_3(x_1)x_3 + \hat{u}_1 + k_1 x_2$$

Imposing

$$\dot{z}_1 = -k_2 z_2$$

where $k_2 > 1$ is another constant gain, we finish the first BS step. We obtain the new state equations

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = -\frac{1}{J} F_3(x_1)x_3 - k_1 x_1 \\ \dot{x}_3 = F_1\big[F_3(x_1)x_2 - x_4\big] \\ \dot{x}_4 = -Kx_4 + Ku_v \end{cases} \tag{5}$$

where in the second BS step the controller should be designed in such a way that the variable x_3 tracks the stabilizing term

$$\alpha_2(x_1, x_2) = \frac{k_2 J}{F_3(x_1)}(x_2 + k_1 x_1)$$

To carry out the second BS step, we consider the equations

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = -k_1 x_2 - \frac{1}{J} F_3(x_1)x_3 \\ \dot{x}_3 = u \end{cases} \tag{6}$$

We let $z_2 \triangleq x_3 - \alpha_2(x_1, x_2)$. Computing the time derivative and imposing a stable dynamics $\dot{z}_2 = -k_3 z_2$, with $k_3 > 1$, after some algebra we obtain

$$u = F_1\left[F_3(x_1)x_2 - \alpha_3(x_1, x_2, x_3)\right]$$

where the stabilizing term $\alpha_3(x_1, x_2, x_3)$ that state variable x_4 has to track is given by

$$\begin{aligned} \alpha(x_1, x_2, x_3) &= F_3(x_1)x_2 \\ &+ \frac{1}{F_1}\Bigg[k_2 J(x_2 + k_1 x_1)\left(\frac{\dot{F}_3(x_1)}{F_3(x_1)^2} - \frac{k_3}{F_3(x_1)}\right)x_2 \\ &\qquad + (k_2 + k_3)x_3\Bigg] \end{aligned}$$

To carry out the third BS step, we consider the equations

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = -k_1 x_2 - \frac{1}{J} F_3(x_1)x_3 \\ \dot{x}_3 = F_1[F_3(x_1)x_2 - x_4] \\ \dot{x}_4 = u. \end{cases} \tag{7}$$

We let $z_3 \triangleq x_4 - \alpha_3(x_1, x_2, x_3)$. Computing the time derivative and imposing a stable dynamics $\dot{z}_3 = -Kz_3$, again after some algebra, we obtain that the control signal u_v should take on the form

$$u_v = \alpha_3(x_1, x_2, x_3) + \frac{\partial \alpha_3}{\partial x_1}\dot{x}_1 + \frac{\partial \alpha_3}{\partial x_2}\dot{x}_2 + \frac{\partial \alpha_3}{\partial x_3}\dot{x}_3 \tag{8}$$

The computation of the partial derivatives involved in (8) finally gives

$$\begin{aligned} \frac{\partial \alpha_3}{\partial x_1}\dot{x}_1 &= \dot{F}_3(x_1)x_2^2 + \frac{k_2 J}{F_1}\Bigg[k_1 \frac{\dot{F}_3(x_1)}{F_3(x_1)^2}x_2^2 \\ &+ (x_2 + k_1 x_1)\frac{\ddot{F}_3(x_1)}{F_3(x_1)^2}x_2^2 - 2(x_2 + k_1 x_1)\frac{\dot{F}_3(x_1)^2}{F_3(x_1)^3}x_2^2 \\ &- \frac{k_1 k_3 x_2}{F_3(x_1)} + k_3(x_2 + k_1 x_1)\frac{\dot{F}_3(x_1)}{F_3(x_1)^2}x_2\Bigg] \end{aligned}$$

$$\begin{aligned} \frac{\partial \alpha_3}{\partial x_2}\dot{x}_2 = \Bigg\{F_3(x_1) + \frac{k_2 J}{F_1}\Bigg[(2x_2 + k_1 x_1)\frac{\dot{F}_3(x_1)}{F_3(x_1)^2} \\ - \frac{k_3}{F_3(x_1)}\Bigg]\Bigg\} \end{aligned}$$

$$\frac{\partial \alpha_3}{\partial x_3}\dot{x}_3 = (k_2 + k_3)[F_3(x_1)x_2 - \xi]$$

Remarks

(1) The nonlinear control law given by (4) and (8) guarantees the asymptotic stability of the equilibrium points. Clearly, the choice of the constant gains strongly influences the performances of the closed–loop system.

(2) The controller requires full–state measurements. This is a very strong assumption. In this respect, while measuring angular position and velocity of the looper is not a big issue and it is just a matter of availability of sensors, the availability of the other state variables remains a strong requirement. Therefore, a suitable estimation technique should be considered. In the next section, an Extended Kalman filter is used. However, other choices may be considered like, for instance, high gain nonlinear observers, etc.

5. SIMULATION RESULTS

As previously mentioned, an Extended Kalman Filter (EKF) has been designed to estimate the non–measurable variables σ and v^i assuming the angular position and velocity of the looper to be perfectly measurable. [2]

In the following we report some of the main simulation parameters and characteristics of the simulated disturbances. More specifically,we set controller gains as $k_1 = 100, k_2 = 1000, k_3 = 1000$. The covariance matrices of the EKF have been chosen as $Q = \text{diag}\,(10^{-4}, 10^{-4}, 10^{-3}, 10^2)$ and $R = \text{diag}\,(10^{-5}, 10^{-3})$. The initial values of the state variables have been set to the desired equilibrium state, that is: $\theta(0) = 25°$, $\eta(0) = 0\ [m/s]$, $\sigma(0) = 3.9\ [MPa]$, $\bar{v}(0) = 0\ [m/s]$.

In the following we propose some simulation experiments. In particular we consider the following three kinds of disturbances:

(1) $\Delta\theta_{ref}$ denoting a measurement disturbance on the angular looper position. We consider a step disturbance occurring for $t \geq 0.5\ [s]$ with an amplitude $\Delta\theta_{ref} = 0.2 \cdot \theta_{ref}$.
(2) $\Delta\sigma_{ref}$ denoting a measurement disturbance on the strip tension. We consider a step disturbance occurring for $t \geq 0.5\ [s]$ with an amplitude $\sigma_{ref} = 0.5 \cdot \sigma_{ref}$.

5.1 *Step disturbance on the looper position reference*

In Figures 2, 3 and 4, we report the time histories of the looper position error and strip tension error due to a step variation of the looper position reference. In Figure 4 we report the performance provided by the EKF for the estimation of the forward slip.

[2] We notice in passing that the knowledge of the exit strip speed v^i (and consequently of the strip tension σ) is not a trivial task (see (Roberts, 1988)). Fortunately, the use of the EKF offers also the possibility to achieve by–product an online estimation of the *forward slip* and consequently of v^i. This estimation represents an improvement compared to the offline estimation provided by the process computer.

As can be seen the performance of the proposed controlled seems robust towards the sudden variation of the looper working point-

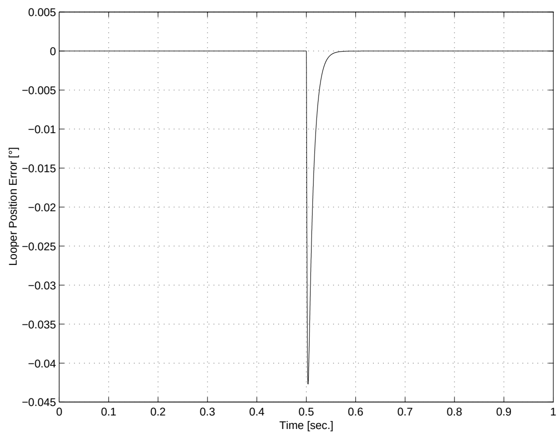

Fig. 2. Looper position error $\Delta\theta$ due to a step disturbance on the looper position reference

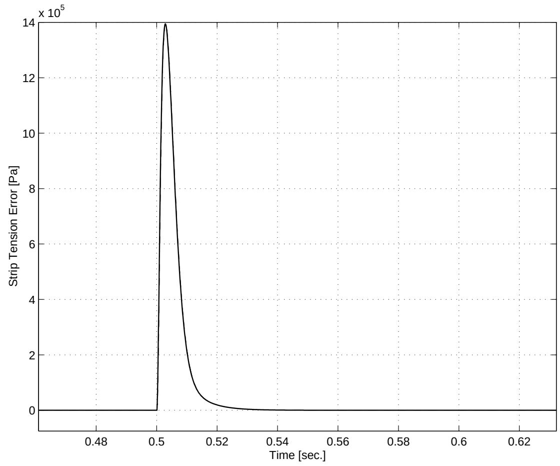

Fig. 3. Strip tension error $\Delta\sigma$ due to a step disturbance on the looper position reference

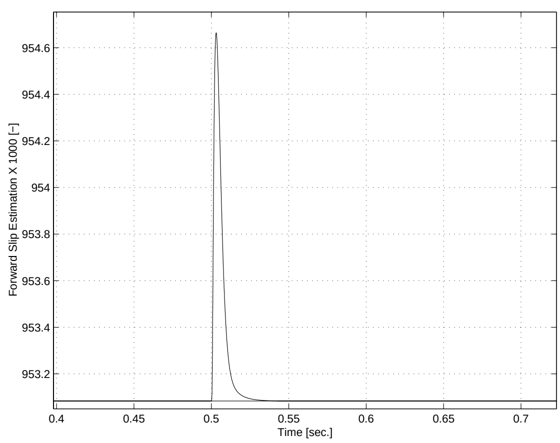

Fig. 4. Forward Slip Estimation by EKF with a disturbance on the looper position reference

5.2 *Step disturbance on the strip tension*

In this subsection we point out (Figures 5, 6 and 7) the performance obtained with a step variation in the strip tension. This type of disturbance can turn out the most critical in many situations and can be due to a sudden problem of the upstream stand or a variation in the strip thickness.

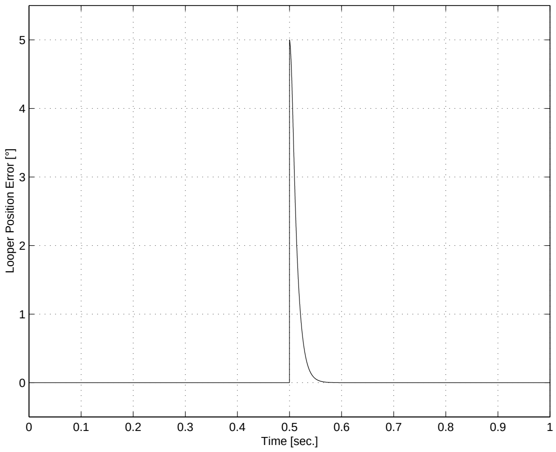

Fig. 5. Looper position error $\Delta\theta$ due to a step disturbance on the strip tension

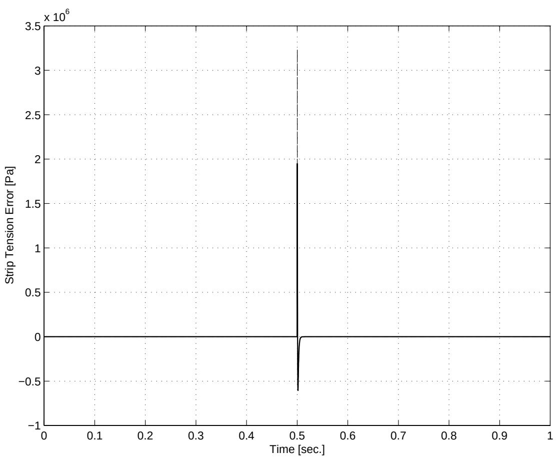

Fig. 6. Strip tension error $\Delta\sigma$ due to a step disturbance on the strip tension

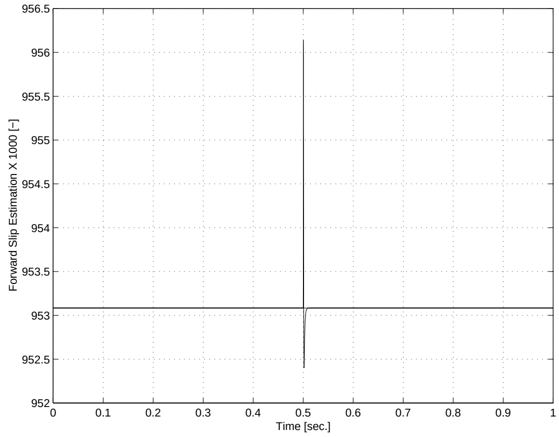

Fig. 7. Forward Slip Estimation by EKF with a step disturbance on the strip tension

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