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Improved Drive Control for Multi-Stand Cold Rolling Mills

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Abstract: In multi-stand cold rolling mills an accurate synchronization of the mill drives is necessary for a good product quality and for a safe process operation. During the standard commissioning procedure the control performance of each drive is optimized individually. However, in normal operation mode the performance is then not always optimal due to the coupling of the drives by the strip being rolled. Based on a model of the drives and the strip tension force the control performance of the coupled drives is analyzed. A linear observer is proposed to support the standard PI speed controllers. Thus control performance can be improved with a minimum of additional effort. The observer needs only a few parameters and can cover all operating points of a tandem cold rolling mill.

1. INTRODUCTION

In the steel industry the demands to suppliers of drives and automation systems become more and more challenging due to the pressure from the market. Startup-times of new plants and shutdown-times during modernizations have to be kept at a minimum while the guarantees for product quality, plant availability and throughput are more and more ambitious. For cold rolling mills this means that not only the strip thickness and shape must be kept within certain tolerances and that process speed should be maximized, but also that off-gauge lengths have to be kept at a minimum. One key-issue for achieving these goals in multi-stand (tandem) cold rolling mills is the accurate synchronization of all mill drives.

Table 1. Basic and technological control loops

Basic Control	Actuator
rollgap position / roll force	hydraulic or spindle
roll speed	AC/DC drive
coiler / uncoiler tension force	AC/DC drive
Technological Control	**Actuator**
strip thickness	rollgap position
strip tension force	rollgap position or roll speed
shape control	roll bending and zone cooling

The procedure during commissioning of cold rolling mills is usually as such: after all mechanical and electrical installations have been done, first the basic control loops (see table 1) are set-up and optimized usually by step-response tests: mill drives, hydraulic rollgap position systems and coiler and uncoiler control. For those standard industrial applications P or PI controllers are widely used. Good control performance can be achieved since the processes are mostly linear and PI control allows to optimize the control loops in a straight-forward way. Once those systems are running properly, strip can be threaded into the mill and the technological control functions as strip thickness, strip tension force and shape control can be set-up step by step. Those control loops are superimposed to the basic control loops so that there is a cascaded control structure. Compared to the basic control loops they are more complex because of their coupling among each other and the dependence on the working point determined mainly by product and mill speed. Although several non-linear and MIMO-control concepts have been proposed in literature (Kugi et al. (2000), Geddes and Postlethwaite (1998), Hoshino et al. (1988))), the control concept for most applications is still decentralized PI(D) control with decoupling and feed forward loops and adaptive control parameters. Especially for tension force and thickness control decoupling networks have been proposed and applied successfully (Kroll and Vollmer (2004), Bryant (1973), Edwards (1975)). Nevertheless the coupling of basic and technological control loops has not strongly been considered yet in those control applications, assuming that the control performance of the basic control loops is not reduced with strip threaded in the mill. In the following sections it will be shown that this assumption is not always correct. Especially at low mill speed there is a strong cross-coupling of drive speed and the strip tension force. Consequently the dynamics of the mill drives are much slower compared to the unthreaded case. Furthermore the performance of the superimposed

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technical control loops is limited, since inner and outer control loops then operate in the same dynamic range.

2. TANDEM MILL NOTATIONS

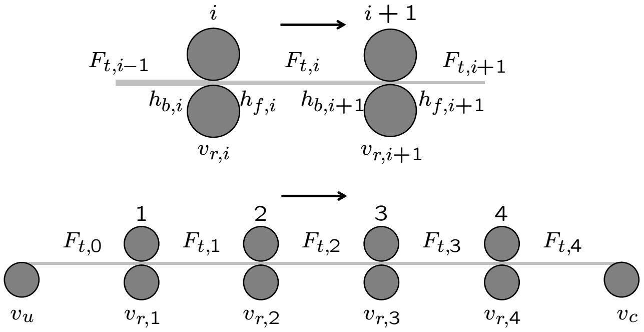

Fig. 1. Tandem mill notations

The discussion in this paper focuses on a 4-stand tandem cold rolling mill, but the results can be carried forward to any multi-stand mill. First the notations for tandem mill are sketched out: stand-specific variables or parameters (e.g. roll speed) are indexed with i. i denotes the number of the stand and reaches from 1 to n in a n-stand tandem cold rolling mill. In contrast strip-specific variables (e.g. strip tension force) are indexed from 0 to n since there are $n+1$ strip sections. In some case the index i is omitted if the consideration is restricted to a single stand. Back- and forward tension force are then referred to as $F_{t,b}$ and $F_{t,f}$.

Table 2. Symbols

Symbol	Unit	Description	Interval of i
$F_{t,i}$	N	strip tension force	0-n
$v_{f,i}$	m/s	exit speed	1-n
$v_{b,i}$	m/s	entry speed	1-n
$v_{r,i}$	m/s	roll speed	1-n
v_u,v_c	m/s	strip speed (unc-)coiler	-
f_i	-	forward slip	1-n
$T_{q,r,i}$	Nm	roll torque	1-n
$T_{q,m,i}$	Nm	motor torque	1-n
$T_{q,u}$,$T_{q,c}$	Nm	motor torque (unc-)coiler	-
$h_{f,i}$	m	exit thickness	1-n
$h_{b,i}$	m	entry thickness	1-n
R_i	m	work roll radius	1-n
R_u,R_c	m	radius (unc-)coiler	-
J_i	kgm^2	total inertia drive train	1-n
J_u,J_c	kgm^2	inertia (unc-)coiler	-
$c_{s,i}$	N/m	spring constant strip	0-n
k	N/m^2	strip hardness (yield stress)	-
μ_i	-	rollgap friction coefficient	1-n
$K_{b,i}$,$K_{f,i}$	m	substitute gain	1-n

3. MODELING

In this section the linear state-space model that will be used for the analysis of the interaction of strip tension force and roll speed is derived. Therefore the equations for the drive train, the rollgap and the strip segment between the stands are presented. Some assumptions must be taken into account to get a simple set of linear equations. This is possible since the neglected effects are not dominant. The assumptions are:

- The deformation area within the rollgap can be divided into the forward and backward slip area where the strip speed is faster respectively slower than the roll speed. Both areas are separated by the neutral point where strip and roll speed are identical. The relation between roll speed and exit speed can be expressed with the forward slip f:

$$v_f = v_r(1+f) \tag{1}$$

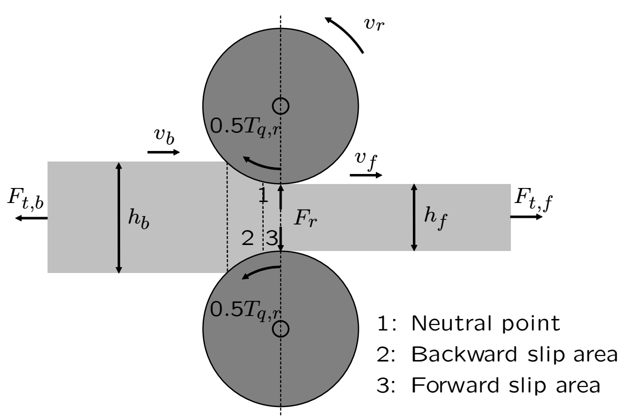

Fig. 2. Rollgap

- The calculation of roll force, roll torque and forward slip is very complex and non-linear, for further details the reader is referred to Bryant (1973) and Bland and Ford (1948). Since this paper does not focus on this topic it will be assumed that those variables are somehow dependent on entry and exit thickness, entry and exit tension force and the parameters material hardness, friction coefficient and work roll radius:

$$F_r = F_r(h_b, h_f, F_{t,b}, F_{t,f}, k, \mu, R) \tag{2}$$

$$T_{q,r} = T_{q,r}(h_b, h_f, F_{t,b}, F_{t,f}, k, \mu, R) \tag{3}$$

$$f = f(h_b, h_f, F_{t,b}, F_{t,f}, k, \mu, R) \tag{4}$$

  The roll force is not further considered in this paper but is mentioned here for the sake of completeness.
- The strip density and strip width are not changed during the deformation process. Therewith the mass flow equation can be reduced to:

$$h_b \cdot v_b = h_f \cdot v_f \tag{5}$$

- The mechanical deformation of the material is symmetrical to the pass-line so that the roll torque is distributed equally to 50% on the upper and lower rolls.
- The motor is stiffly coupled to the rolls, that means that the whole drive train (motor, rolls, gear and shaft) can be reduced to a one mass system with a total inertia J. All variables are referred to the load side.
- Upper and lower drive train are identical and controlled individually. Due to the symmetry it is then sufficient to model only one drive train.
- Friction effects in the drive train are not considered.
- The dynamics of the torque control loops are neglected:

$$T_{q,m} = T_{q,m,Ref} \tag{6}$$

- The strip segment between the stands i and $i+1$ is regarded as a mass-less spring with the substitute spring constant $c_{s,i}$. Corresponding to Hook's law the change of strip tension force is then given by the difference of the strip entry speed at stand $i+1$ and the strip exit speed at stand i:

$$\dot{F}_{t,i} = c_{s,i}(v_{b,i+1} - v_{f,i}) \tag{7}$$

- The deviation from the operating point is small, so that the mass-flow and rollgap equations can be linearized. In the following all equations are presented in the linearized form with Δ-values.

3.1 Roll speed

The angular speed of the rolls can be described by Newton's law for rotational motion:

$$J_i \Delta\dot{\omega}_i = \sum \Delta T_{q,i} = \Delta T_{q,m,i} - 0.5\Delta T_{q,r,i} \tag{8}$$

The roll torque $T_{q,r,i}$ that is needed to bring up the deformation work to the material depends on many variables and parameters (equation (3)). Since only the interaction of strip tension force and roll speed is discussed in this paper, the linearized roll torque is divided into three terms:

$$\begin{aligned}\Delta T_{q,r,i} &= \frac{\partial T_{q,r,i}}{\partial F_{t,i-1}}\Delta F_{t,i-1} + \frac{\partial T_{q,r,i}}{\partial F_{t,i}}\Delta F_{t,i} + \Delta T_{q,Dist,i}\\ &= K_{b,i}\Delta F_{t,i-1} - K_{f,i}\Delta F_{t,i} + \Delta T_{q,Dist,i} \qquad (9)\end{aligned}$$

The first two terms describe the dependency of the torque from the back and front tension force. The partial derivations are nearly independent from the mill speed and are replaced by the terms $K_{b,i}$ and $-K_{f,i}$ in the following. The last term $\Delta T_{q,Dist,i}$ includes the dependence of roll torque from all other parameters and variables as e.g. thickness changes or variation in hardness. They can be interpreted as an external disturbance for the derived tension force and roll speed model.

With $\omega_i = \frac{v_{r,i}}{R_i}$ the final equation for roll speed is:

$$\begin{aligned}\Delta\dot{v}_{r,i} = \frac{R_i}{J_i}[\,\Delta T_{q,m,i} &- 0.5K_{b,i}\Delta F_{t,i-1}\\ &+0.5K_{f,i}\Delta F_{t,i} - 0.5\Delta T_{q,Dist,i}] \qquad (10)\end{aligned}$$

3.2 Uncoiler and coiler speed

The uncoiler and coiler speed can be derived similar to the roll speed. The difference is that the load torque is determined by the product of strip tension force and radius and that there are no torque disturbances, which arise from the deformation in case of roll speed:

$$\Delta\dot{v}_u = \frac{R_u}{J_u}[\,\Delta T_{q,u} + \Delta F_{t,0}R_u] \tag{11}$$

$$\Delta\dot{v}_c = \frac{R_c}{J_c}[\,\Delta T_{q,c} - \Delta F_{t,n}R_c] \tag{12}$$

3.3 Rollgap

The exit speed of a stand is given by the roll speed and the forward slip. The linearized form of equation (1) is:

$$\Delta v_{f,i} = \Delta v_{r,i}(1+f_i) + \Delta f_i v_{r,i} \tag{13}$$

Similar to the linearized torque equation the forward slip (equation (4)) is split up into three parts which describe the dependency from front and back tension force as well as all other influences summarized in $\Delta f_{Dist,i}$:

$$\Delta f_i = \frac{\partial f_i}{\partial F_{t,i-1}}\Delta F_{t,i-1} + \frac{\partial f_i}{\partial F_{t,i}}\Delta F_{t,i} + \Delta f_{Dist,i} \tag{14}$$

The entry speed is linked to the exit speed over the mass-flow equation (5):

$$\Delta v_{b,i} = \Delta v_{f,i}\frac{h_{f,i}}{h_{b,i}} + \underbrace{\Delta h_{f,i}\frac{v_{f,i}}{h_{b,i}} - \Delta h_{b,i}\frac{v_{f,i}h_{f,i}}{h_{b,i}^2}}_{=\Delta v_{b,Dist,i}} \tag{15}$$

3.4 Strip

For interstand tension forces, equation (7) can directly be expressed with Δ-values. For the mill entry and exit tension force the equation is also valid but one formally has to replace the speeds $v_{f,0}$ and $v_{b,n+1}$ by the uncoiler and coiler speeds v_u and v_c:

$$\Delta\dot{F}_{t,i} = c_{s,i}(\Delta v_{b,i+1} - \Delta v_{f,i}) \quad i = 1:n-1 \tag{16}$$

$$\Delta\dot{F}_{t,0} = c_{s,0}(\Delta v_{b,1} - \Delta v_u) \tag{17}$$

$$\Delta\dot{F}_{t,n} = c_{s,n}(\Delta v_c - \Delta v_{f,n}) \tag{18}$$

3.5 State-space equations

Equations (10) to (18) can be rewritten in state-space form by elimination of the variables $\Delta v_{f,i}$,$\Delta v_{b,i}$ and Δf_i. The states are the roll speeds, speed of uncoiler and coiler and strip tension forces:

$$\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{u} + \mathbf{E}\mathbf{d} \tag{19}$$

$$\mathbf{y} = \mathbf{x} \tag{20}$$

$$\mathbf{x} = \begin{pmatrix}\Delta v_u\\ \Delta F_{t,0}\\ \Delta v_{r,1}\\ \Delta F_{t,1}\\ \Delta v_{r,2}\\ \Delta F_{t,2}\\ \Delta v_{r,3}\\ \Delta F_{t,3}\\ \Delta v_{r,4}\\ \Delta F_{t,4}\\ \Delta v_c\end{pmatrix};\ \mathbf{u} = \begin{pmatrix}\Delta T_{q,u}\\ \Delta T_{q,1}\\ \Delta T_{q,2}\\ \Delta T_{q,3}\\ \Delta T_{q,4}\\ \Delta T_{q,c}\end{pmatrix};\ \mathbf{d} = \begin{pmatrix}\Delta\dot{F}_{t,Dist,0}\\ \Delta T_{q,Dist,1}\\ \Delta\dot{F}_{t,Dist,1}\\ \Delta T_{q,Dist,2}\\ \Delta\dot{F}_{t,Dist,2}\\ \Delta T_{q,Dist,3}\\ \Delta\dot{F}_{t,Dist,3}\\ \Delta T_{q,Dist,4}\\ \Delta\dot{F}_{t,Dist,4}\end{pmatrix};$$

The external disturbances $\Delta v_{b,Dist}$ and Δf_{Dist} are summarized into one term $\Delta\dot{F}_{t,Dist,i}$:

$$\Delta\dot{F}_{t,Dist,i} =$$
$$c_{s,i}[\Delta v_{b,Dist,i+1} + \Delta f_{Dist,i+1}v_{r,i+1}\frac{h_{f,i+1}}{h_{b,i+1}} - \Delta f_{Dist,i}v_{r,i}]$$

The motivation for this notation is the following: once that a speed controller, which guarantees $\Delta v_{r,i} = 0$ for $t \to \infty$, has been designed, it can be shown by a static system analysis that those external disturbances cause static tension force deviations. It will later be the task of the superimposed tension force control loops to reject those disturbances. For further considerations in this paper concerning strip speeds this external disturbance is assumed to be 0. In consequence no static tension deviations occur if the all roll speed errors are controlled to 0.

The matrices have the following structures (only the signs of the matrix elements are sketched):

$$\mathbf{A} = \begin{pmatrix} 0 & + & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ - & - & + & + & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & - & 0 & + & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & + & - & - & + & + & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & - & 0 & + & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & + & - & - & + & + & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & - & 0 & + & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & + & - & - & + & + & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & - & 0 & + & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & + & - & - & + \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & - & 0 \end{pmatrix};$$

$$\mathbf{B} = \begin{pmatrix} + & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & + & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & + & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & + & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & + & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & + \end{pmatrix} \mathbf{E} = \begin{pmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ + & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & - & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & + & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & - & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & + & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & - & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & + & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & - & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & + \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{pmatrix}$$

The system is semi-stable, since the **A**-matrix has one eigenvalue at 0. This can be interpreted physically: if all mill drives, coiler and uncoiler would synchronously be accelerated (that means that no tension force disturbances occur) by applying an external torque, the total mill speed will increase continuously (integral behavior), since there is no torque that forces the system back into its old operation point.

4. ANALYSIS OF THE CONTROLLED SYSTEM

The task of the roll speed controllers is to make sure that the roll speeds $v_{r,i}$ are at the desired reference value. The tuning of those controllers usually does not take into account the coupling by the strip because the commissioning phase for the roll speeds is normally before any strip has been threaded (stand-alone). The plant for the speed controllers is then reduced to 4 independent SISO systems with integral characteristic. From linear system theory it is widely known that for speed control an integral controller is needed if torque disturbances occur. The PI controllers could be tuned by applying the rules of the symmetrical optimum, nevertheless in cold rolling applications the control performance of state-of-the-art drives is limited to rise times of about 50 to 150ms due to practical aspects as torque limits or mechanical resonances in the drive train.

It will now be analyzed how a speed control loop with a typical rise time of about 50ms behaves, if it is coupled with other drives with similar response times by a strip. Parameters from real plants are used for the parametrization of the models and controllers. The operation point of

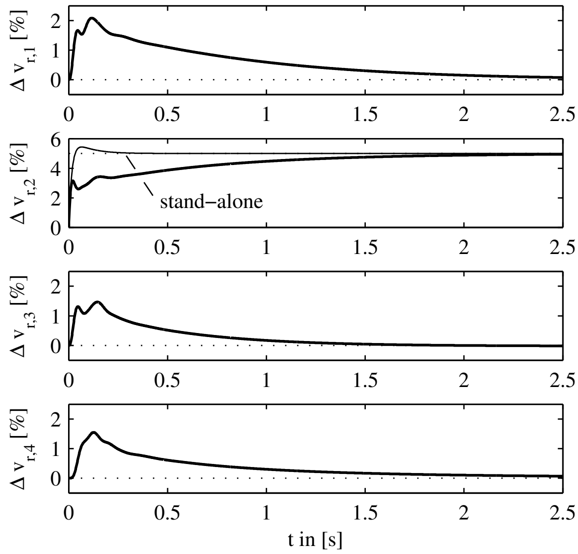

Fig. 3. 5% reference step roll speed stand 2

the threaded mill is with thick strip and low speed, superimposed tension force control loops are not considered. Fig. 3 shows the response of the controlled roll speeds when applying a 5% reference step at stand 2. One can observe, that all other mill drives are affected by the step and that it takes about 2 seconds to remove the static errors.

To understand this effect a look to the eigenvalues of the coupled system will be taken for three cases: without any speed control, with P controllers in each stand and with PI controllers in each stand. From table 3 one can see

Table 3. Eigenvalues

λ	$\omega_0 = \lvert\lambda\rvert$	$D = \lvert\frac{Re\{\lambda\}}{\lvert\lambda\rvert}\rvert$
Without Control		
$-11.28 \pm j96.94$	97.60	0.1156
$-7.70 \pm j88.79$	89.12	0.0864
$-8.74 \pm j70.65$	71.18	0.1227
$-6.01 \pm j49.03$	49.40	0.1217
$-6.21 \pm j27.22$	27.92	0.2223
0	0	-
With P Control		
$-40.91 \pm j93.66$	102.20	0.4003
$-19.02 \pm j86.83$	88.90	0.2140
$-26.86 \pm j68.69$	73.76	0.3642
$-17.73 \pm j43.18$	46.68	0.3799
-43.02	43.02	1
$-13.01 \pm j29.76$	32.48	0.4004
With PI Control		
$-40.58 \pm j96.57$	104.74	0.3847
$-19.40 \pm j88.09$	90.20	0.2151
$-26.73 \pm j71.29$	76.13	0.3510
$-19.94 \pm j45.13$	49.34	0.4042
$-11.35 \pm j31.16$	33.17	0.3423
$-18.81 \pm j6.78$	20.00	0.9408
-2.41	2.41	1
-1.48	1.48	1
-0.56	0.56	1
With PI Control, Stand-Alone		
-61.94	61.94	1
-11.93	11.93	1

that P controllers (same gain as PI controllers) improve the dynamic characteristics of the system compared to the uncontrolled system: the eigenvalues are shifted to the left in the Laplace-domain, the damping is increased and the closed-loop system has no eigenvalue at 0. Nevertheless for practical application a pure P controller is not suitable because static speed control errors would not be removed. With PI controllers the number of eigenvalues increases but the dominating eigenvalues are slower by the factor 10 or more compared to P control. With this it is clear, that the poor control performance in threaded conditions arises from the integral action of the PI controller.

5. DECENTRALIZED OBSERVER DESIGN

One possibility to improve the system performance is to retune the parameters of the PI controllers, e.g. by using linear pole placement techniques. This method has the drawback, that it is based on a linear model as presented before. Since this model was derived by linearization around the working point, one would have to use an adaptive control law, which is not straight-forward for systems of this order. Also one has to take into account that some parameters such as roll gap coefficients and strip constants are affected by uncertainties so that a robust control design would require a high effort.

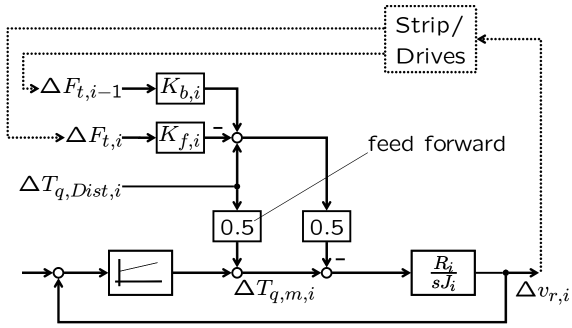

Fig. 4. Speed control loop with feed forward control

To find another method one has to take a look to equation (10). To remove any control error under static conditions the controller output must exactly match the torque disturbance (as already mentioned, all tension force disturbances are 0 for $t \to \infty$ if speed errors are canceled):

$$\Delta T_{q,m,i} \stackrel{!}{=} 0.5 \Delta T_{q,Dist,i} \tag{21}$$

In case of a PI controller this is realized by the I channel since the output of the P channel is 0 without control error. Another possibility to fulfill condition (21) is with a feed forward control loop if the torque disturbance is measurable (Fig. 4). If the static value of the disturbance is exactly known control errors can fully be removed while slow transient actions which arise from the integral action of the controller are avoided.

Unfortunately the torque disturbance cannot be measured but it can be observed by using an observer for disturbance variables (Isermann (1989), Franklin et al. (2002)). Therefore the process model must be extended by a linear disturbance model. The simplest model that can cover non-zero disturbances for $t \to \infty$ is an integrator model, which has the advantage that no additional parameters are needed.

Fig. 5 shows the block diagram of the submodel covered by equation (10) and the disturbance model together with the observer. A submodel can be used instead of the total process model because the strip tension forces over which the drive is coupled to other drives are measurable. So it is possible to realize a decentralized observer for each mill drive which only needs a few parameters and which is independent of the operating point of the mill. The observer design for this second order system (**L**-matrix) is straight-forward.

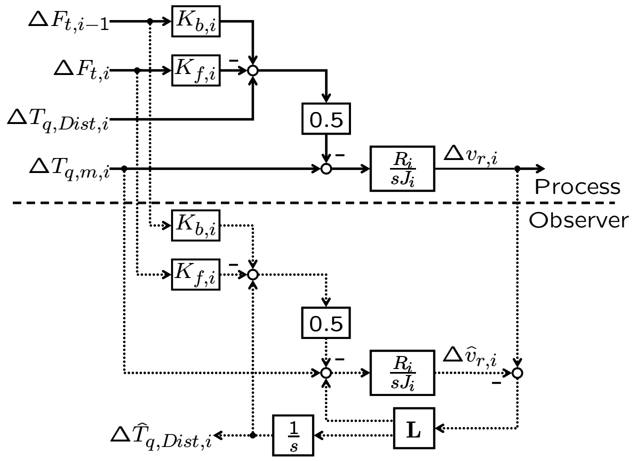

Fig. 5. Submodel with torque observer

The overall control structure is shown in Fig. 6: decentralized speed controllers are controlling the mill drives which are coupled by the strip. This classical control structure is extended by decentralized observers which calculate the torque disturbance by the means of motor torque, roll speed and front and back tension force. The observed torque disturbance is fed forward to the controller output so that the I channel of the controller is theoretically not necessary. Another advantage of the observer is, that it can simply be added to classical PI controllers without having to change the whole structure and that it can easily be switched on and off since it is a feed-forward loop.

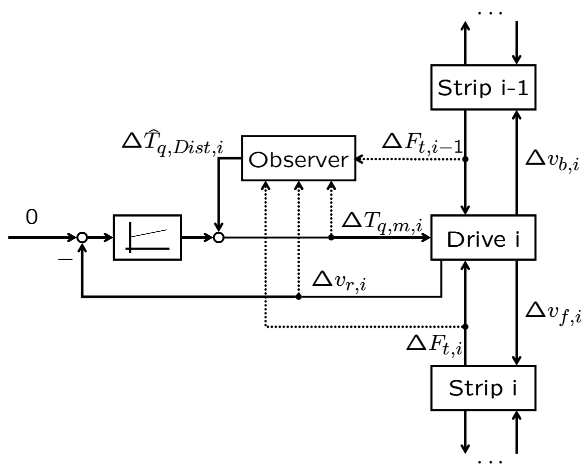

Fig. 6. Overall control structure

6. SIMULATION RESULTS

The control performance of the mill drives and superimposed tension force control loops have a strong influence on off-gauge lengths, although many other factors as control concept, setup quality and instrumentation have to be taken into account. A typical disturbance for the speed controllers at threading is a ramp in strip thickness. Usually the strip thickness is larger during threading and is then changed to its reference value over a ramp after strip tension has been established. This causes a disturbance in roll torque which has to be matched by the roll speed controller. Such a disturbance is used to show the improvements that can be achieved by extending a classical control concept with the presented observer.

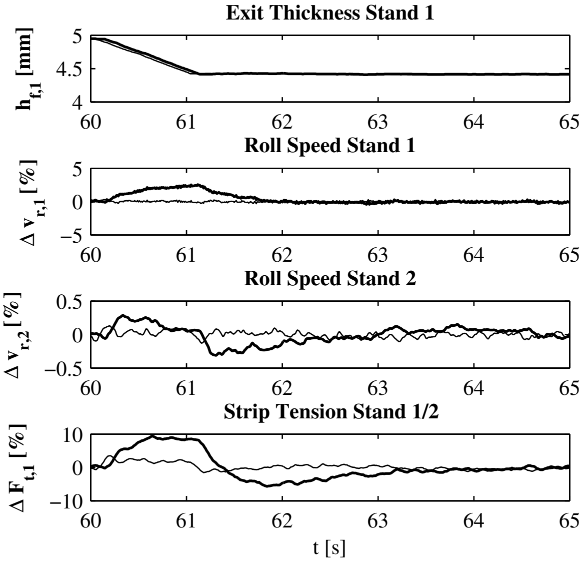

Fig. 7. Thickness ramp at threading, state-of-the-art drives (thick line: classical control; thin line: with observer)

A threading sequence of a thick strip into a 4-stand tandem mill has been simulated using a complex nonlinear rolling mill simulator (Brickwedde et al. (2007)). Parameters from a real plant have been used, the drives have been configured to reach an assumed rise time of 50ms. To gain realistic results, disturbances such as roll eccentricities, coil bump, setup errors, measurement noise, thickness and hardness variations and Coulomb friction effects are taken into account for simulations. The simulated threading sequence is also similar to the one used on the plant.

Fig. 7 shows that the deviation in roll speed at stand 1, where the torque disturbance occurs, is drastically reduced. Since the tension force deviations between stand 1 and 2 are therewith diminished the interaction with superimposed tension force control loops is reduced. Similar improvements are achieved in the stands 3 and 4. This has positive effects on off-gauge length, since the tension force is usually controlled by changing the strip thickness. From Fig. 8 one can see that the improvement is even much larger in mills with conventional drives with rise times of about 150ms. During full speed operation only minor improvements can be achieved with the observer because the coupling between strip tension force and roll speed is less with increasing roll speed.

7. CONCLUSION

The presented feed forward control based on a disturbance observer improves the speed control performance of multi-

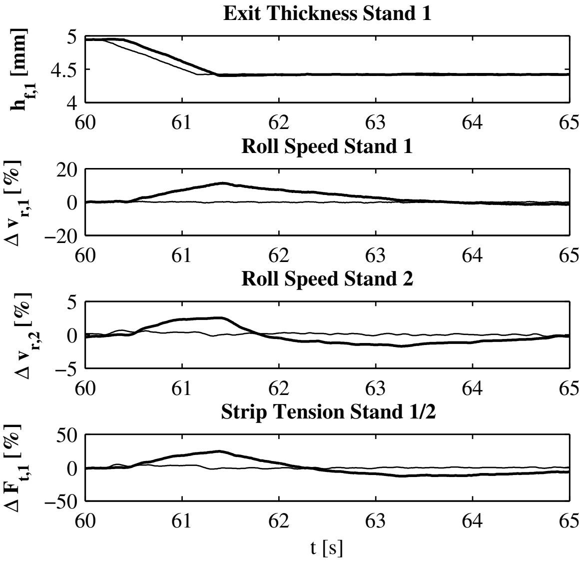

Fig. 8. Thickness ramp at threading, conventional drives (thick line: classical control; thin line: with observer)

stand cold rolling mills especially at low speed. This is not only of concern at threading or tail out but also during flying gauge change or weak-point passing. The structure is very simple and only few parameters, which are independent from the mill speed, are necessary. The decentralized observer can easily be integrated into classical control structures. The next step will be to apply the observer on a real plant, since simulation results are very promising.

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