Gauge and Tension Control during the Acceleration Phase of a Steckel Hot Rolling Mill

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The focus in this work is on linear model identification of a Steckel hot rolling mill process on a point during the acceleration phase of the process. It furthermore focuses on the controller design based on the linear models and on controller implementation on a non-linear simulator of this process. The linear models are identified for different cases simulated with and without gauge meter compensation and controlled tensions as part of the simulator. The system identification is accompanied by a heuristic justification of the data obtained. A diagonal PID/PI controller as well as MIMO H\textsubscript{\infty} controllers, of which the designs are based on the linear models, are implemented on the simulator. From the system identification data for the different linear models it could be seen that gauge meter compensation successfully counteracts the adverse effect of mill stretch and eliminates oscillations in exit gauge, which result from tension oscillations. Simulations of the different controllers in closed loop with the non-linear plant simulator show that good control can be achieved by controllers of which the designs are based on linear diagonal models. One of these controllers, a diagonal H\textsubscript{\infty} controller, tested on a non-linear simulator which incorporated gauge meter compensation and inner loop tension control, was found to be the most suited for a switching control system.

KEY WORDS: steel making; hot strip rolling; Steckel rolling mill; system identification; acceleration phase; tensions; exit gauge; diagonal control; H\textsubscript{\infty} control; gauge meter compensation.

1. Introduction

In the steel making industry rolling mills require large capital expenditure. Improving the throughput and quality produced by a Steckel mill can have significant financial implications.\textsuperscript{1} The drive for improved quality results from an over capacity of steel production and therefore from an increase in competition. Quality improvement is meant in the sense that the strip output thickness lies within narrower tolerances. How an increase in throughput can be achieved is described in the next section.

This work is a continuation of the work done by Scholtz.\textsuperscript{2} His focus was on the derivation of a non-linear model and the development of a simulator for the Steckel hot rolling mill. This model is used here to estimate linear models for controller design purposes and to serve as plant for controller implementation.

Numerous papers on the control of hot rolling mills can be found in literature. Applications of advanced control methods such as H\textsubscript{\infty} and H\textsubscript{2} (Linear Quadratic) are described.\textsuperscript{1,4} These controllers were designed for the threading speed phase of rolling and for a multi stand mill respectively. In contrast, this work describes controllers, which were designed for the speed up phase of a reversing hot rolling mill. Furthermore research into thickness control of multi stand rolling mills using Generalized Predictive Control (GPC) is reported.\textsuperscript{5}

With respect to control research on rolling mills the current trend is towards the application of multi-input-multi-output (MIMO) control.\textsuperscript{4} Although rolling mills are very complex multivariable systems, the most often implemented control systems are single-input-single-output (SISO) multi-loop controllers.\textsuperscript{6} The main reason for this is the ease of commissioning of the individual control loops.

In this work a diagonal controller (i.e. a SISO multi-loop) was designed for the non-linear simulator, with interactions considered as disturbances.

An H\textsubscript{\infty} controller was also designed using a diagonal linear model, which was identified from a non-linear simulator supplemented with gauge meter compensation and inner loop tension control.

A third controller, also an H\textsubscript{\infty} controller, was designed from a full linear model identified from a non-linear simulator with supplements as mentioned above. From the simulations of this simulator it was found however, that disturbances by input/output interactions on the tensions made the relationships between the model inputs and outputs non-linear. The full linear model could not satisfactorily accommodate these non-linear relationships. Therefore this linear model and its corresponding controller are not discussed in this work. The diagonal (SISO multi-loop) and the diagonal H\textsubscript{\infty} controller schemes were implemented on the non-linear simulator.

In the following section issues of the rolling process are
described. Modeling considerations and system identification (SID) accompanied by a heuristic justification of the SID data are given in Sec. 3. Section 4 contains a description of the controller design methods, while in Sec. 5 the controller design parameters are given. In Sec. 6 the controller implementation on the simulator is described. In the same section the results from the implementation are discussed. In Sec. 7 it then concludes with the main findings made in this work.

2. Process Description

Figures 1 and 2 show the basic components of a Steckel finishing mill. Scholtz modeled the stand of the Steckel mill as a system consisting of two roll packs. They are held by roll chocks and bearings kept in position by hydraulic actuators and load cells, which in turn are mounted in a frame. Hydraulic jacks are situated between the chocks of the top and bottom work rolls. The stand components have been described as discrete elements, i.e. lumped masses and springs and continuous mass elastic beams.

The purpose of the hydraulic actuators is to generate a force which forces the roll packs towards each other and to position the upper roll pack according to the strip exit thickness requirements. The resulting elastic bending of the work rolls is then counteracted by extension of the hydraulic jacks.

The rolling process itself starts in the Steckel finishing mill after the strip has been processed in the crop shear stage. When the strip enters the finishing mill stage it is passed underneath the entrance coiler furnace and threaded into the roll gap by the entry side pinch roll unit. On the mill delivery side the strip is fed to the delivery side coiling mandrel and as soon as the first wraps have been coiled and the strip tension has assumed its set up value, the coiler accelerates together with the mill. In the worst case the strip front tension assumes its set up value during the acceleration phase in a higher speed range of the process. This is a worse case compared to front tension increase before the acceleration phase because at higher coiler speeds less coiler motor torque is available for control purposes. A constant strip tension is maintained during the pass. When the tail end of the strip enters the roll gap, the mill is slowed down such that the tail end stops before the pinch roll unit on the current delivery side of the mill. The roll gap is then preset for the next pass and the process is reversed, such that the entry side pinch roll unit now feeds the entry side coiling mandrel. Reversal is done until the strip has the desired thickness. Typical of this reversing process is that the strip ends are exposed to greater cooling and require increased rolling forces for deformation.

Since the strip exit gauge is influenced by the elastic mill frame stretch, gauge meter compensation is employed to combat this effect. This is done by using positive force feedback to compensate for the mill stretch. Depending on the rolling force deviations, \( \delta F \), the compensator adds a trim to the hydraulic stroke, \( \delta x \), such that the exit thickness deviation, \( \delta h_2 \), due to mill stretch is zero according to the equation:

\[
\delta h_2 = -\delta x + \frac{\delta F}{M} \tag{1}
\]

where \( M \) is the mill modulus. What happens in the roll gap is that the law of mass flow continuity holds across the roll gap, i.e.

\[
v_1 \cdot h_1 = v_2 \cdot h_2 \tag{2}
\]

This law is based on the assumption that material spread in the width is negligible. The strip metal entering the finishing mill at temperatures in the region of 1000°C flows plastically in the roll gap. Because of plasticity the whole roll gap is filled with strip material. The entering strip exerts a separating force, the specific roll force, \( P \) [N/m], on the work rolls causing them to bend and the arc of contact to flatten elastically.

3. Models for Controller Design

3.1. Modeling Considerations

The available non-linear model for the Steckel mill is not in a format that can directly be used for controller design.
purposes. System identification (SID) is applied to the simulator of the process in order to obtain a linear time invariant (LTI) estimate of the non-linear model around a specific operating point. ARX (Auto Regression with eXternal Input) linear models were obtained using the Matlab System Identification Toolbox.\(^3\)

For the linear model the outputs of interest are the deviations in centerline exit thickness of the strip, \(\delta h_{l, g}\), and the strip tension changes on the entrance and exit sides of the roll gap, \(\delta T_{e}\) and \(\delta T_{f}\) respectively. Deviations in hydraulic stroke set points, \(\delta x_{sc}\), on both sides of the mill frame and deviations in back and front coiler speeds, \(\delta v_{bc}\) and \(\delta v_{fc}\), respectively, are the manipulated variables. Intervention into the process is possible through these variables. The speeds \(v_{e}\) and \(v_{f}\) are the entrance and exit speeds at the roll gap respectively. In Laplace transfer function format the LTI model is thus given by:

\[
\begin{bmatrix}
\delta h_{l, g} \\
\delta T_{e} \\
\delta T_{f}
\end{bmatrix} =
\begin{bmatrix}
g_{11}(s) & g_{12}(s) & g_{13}(s) \\
g_{21}(s) & g_{22}(s) & g_{23}(s) \\
g_{31}(s) & g_{32}(s) & g_{33}(s)
\end{bmatrix}
\begin{bmatrix}
\delta x_{sc} \\
\delta v_{bc} \\
\delta v_{fc}
\end{bmatrix}
\]

or in standard transfer function model notation,

\[
y(s) = G(s)u(s) \quad \text{..................................(4)}
\]

where \(y(s) = [\delta h_{l, g}(s) \; \delta T_{e}(s) \; \delta T_{f}(s)]^T\) and \(u(s) = [\delta x_{sc}(s) \; \delta v_{bc}(s) \; \delta v_{fc}(s)]^T\) are the controlled variable vector and the manipulated variable vector respectively. In this work all variables are given in SI units.

The potential economic benefit of operating the mill optimally on the speed up and slow down phases of rolling was investigated previously.\(^1\)\(^2\) The results of these findings motivated the decision to choose an operating point and point of linearization during the acceleration phase of the mill. A rolling pass with a strip thickness reduction from 13.3 to 9.97 mm and Stainless Steel Grade 304 as strip material was also chosen. At the point of linearization the main drive peripheral speed was forced constant at \(v_{\text{min}} = \Omega R = 3.5 \text{ m/s}\) (see Fig. 3) where \(\Omega\) is the angular speed and \(R\) the non-deformed radius of the work rolls. This was done in order to prevent speed deviations during acceleration having an effect on the controlled outputs. Figure 4 shows the main drive peripheral speed of a typical Steckel hot rolling mill during the pass considered.

A step of 1 mm was used for the hydraulic stroke. This value is based on a value for this variable found in literature.\(^3\) An upper limit for the speed was the maximum value for the strip back and front tensions with a stress value of 208 MPa. A speed step of \(\Delta v = \pm 0.2 \text{ m/s}\) for a duration of \(\Delta t_{\text{step}} = 60 \cdot 10^{-3} \text{ s}\) was used.\(^9\) During these step tests one of the inputs was stepped at a time while the other two and \(\Omega\) were kept constant.

### 3.2. System Identification

In this section input/output (I/O) data of some of the step tests performed on the simulator are given in the form of graphs accompanied by a heuristic discussion. Two different linear models were identified, called here the first and second linear model. Table 1 gives an overview of the differences between these two LTI models with respect to four criteria listed in the first column.

#### 3.2.1. The First Linear Model

The first linear model is estimated from a simulator with no gauge meter compensation and no inner loop tension control incorporated in it. The tension model was part of the simulator for all the SID data generated for the first linear model except for the data generated for transfer function \(g_{11}(s)\). One of the controllers designed in this work is diagonal. Therefore only the transfer functions on the diagonal of the first linear model were used for controller design.

The I/O data of the transfer function \(g_{11}(s)\) was obtained by subtracting the steady state evolution of the exit thick-
ness, i.e. the exit gauge simulation when no steps are applied, from the exit thickness data, which resulted from the application of a step in hydraulic actuator stroke. The steady state evolution of exit gauge is a result of mill stretch taking place when the simulation of the process starts. The thought behind this subtraction was to obtain the net dynamic effect due to an actuator stroke.\(^2\) \(g_{11}(s)\) was identified as a first order function with delay.

\[
g_{11}(s) = \frac{-4.1936 \cdot 10^{-1}}{5.0145s+1} e^{-2.5 \cdot 10^{-2}s} \quad ... (5)
\]

Figure 6 shows the response of the exit gauge and back and front tensions to initial conditions only, i.e. no steps applied. Oscillatory behavior is observed at a constant roll setting and speed. The reason for the tension increases following the increase in roll gap, i.e. exit gauge, is due to an increase in \(h_2v_2\), in which increases by a factor \(p/H_{1102}\) and \(q/H_{1100}\), since \(h_1\) was not changed. The factors \(p\) and \(q\) are some arbitrary reference values. The tensions increased because

\[
h_1v_{1b} > h_1v_{1a} = h_2v_{2b} \quad ... (6)
\]

where \(a\) and \(b\) refer to after and before increase of the exit gauge respectively. It is required that \(p > q\) for the tensions to increase.

Figure 7 shows the input/output (I/O) data used for the identification of \(g_{21}(s)\). The graphs of the I/O data, used for the identification of \(g_{31}(s)\), have similar trends as those used for \(g_{21}(s)\) and are therefore not shown. The transfer functions for \(g_{21}(s)\) and \(g_{31}(s)\) were identified as

\[
g_{21}(s) = \frac{7.712 \cdot 10^4s - 3.0847 \cdot 10^9}{s^2 + 4.5298s + 2.966} e^{-7.25 \cdot 10^{-2}s} \quad ... (7)
\]

and

\[
g_{31}(s) = \frac{6.4215 \cdot 10^4s - 2.5685 \cdot 10^9}{s^2 + 4.2697s + 2.8926} e^{-7.75 \cdot 10^{-2}s} \quad ... (8)
\]

As it can be seen from Fig. 7 the model identified for \(g_{21}(s)\) is inaccurate. The same applies to the I/O data used for the identification of \(g_{31}(s)\). This inaccuracy however does not matter since the first linear model was used to design a diagonal controller.

The transfer functions \(g_{22}(s)\) and \(g_{33}(s)\) were modeled as integrators.\(^2\) The I/O data shown in Figs. 8 and 9 also show integrator behavior. The transfer functions \(g_{22}(s)\) and \(g_{33}(s)\) therefore have in this work been identified and approximated as

\[
g_{22}(s) = \frac{-1.444 \cdot 10^8}{s + 0.0001} \quad ... (9)
\]

and

\[
g_{33}(s) = \frac{1.088 \cdot 10^8}{s + 0.0001} \quad ... (10)
\]

The influence of a step in back coiler speed, similar to the input given in Fig. 10, on exit gauge is approximately 60 times smaller than the influence of a step in hydraulic stroke, such as the input shown in Fig. 5. Similar arguments apply to the influence of front coiler speed on exit gauge (see Fig. 11). Therefore the corresponding transfer functions were chosen as
The output data in Figs. 10 and 11 were obtained by subtracting the exit gauge, i.e. when no step was applied, from the step test data. This subtraction was done for the same reason as in the case of \( g_{11}(s) \). The stronger increasing trend of the output data of Fig. 11 compared to Fig. 10 is because back tension is twice as effective as front tension in decreasing the separating force between the work rolls.\(^{15}\)

The off-diagonal transfer functions \( g_{23}(s) \) and \( g_{32}(s) \) were identified as

\[
g_{23}(s) = \frac{-1.2461 \times 10^4 s + 6.1196 \times 10^8}{s^3 + 4.0453 \times 10^3 s + 4.0318 \times 10^6} e^{-2.15 \times 10^5 s} \quad \cdots (12)
\]

and

\[
g_{32}(s) = \frac{2.3159 \times 10^4 s - 9.0907 \times 10^9}{s^3 + 4.3528 \times 10^6 s + 9.2486 \times 10^7} e^{-2.0 \times 10^7 s} \quad \cdots (13)
\]

### 3.2.2. The Second Linear Model

The second linear model consists of transfer functions on the diagonal, estimated from a simulator with inner loop tension control and gauge meter compensation incorporated in it. From the steady state gain of \( g_{11}(s) \) of this model it can be seen that the hydraulic stroke input has a larger effect on exit gauge when compared to the first linear model \((87\% \text{ vs.} \ 41\% \text{ respectively})\). This is the result of gauge meter compensation, which is applied to the simulator for the second linear model. The transfer function \( g_{11}(s) \) was obtained as

\[
g_{11}(s) = \frac{-0.8768 \times 10^5}{10.52 s + 1} e^{-2.5 \times 10^5 s} \quad \cdots (14)
\]

while the transfer functions and were identified as

\[
g_{22}(s) = \frac{-6.9092 \times 10^5 s - 1.3594 \times 10^8}{s^3 + 9.0963 \times 10^5 s + 5.071 \times 10^6} \quad \cdots (15)
\]

and

\[
g_{33}(s) = \frac{-4.5934 \times 10^5 s + 9.0628 \times 10^6}{s^3 + 8.1846 \times 10^5 s + 5.5107 \times 10^6} \quad \cdots (16)
\]
The higher order response of the tensions to the speed steps of ±0.2 m/s is due to the control applied to the tension at a frequency 1000 times higher than the sampling frequency of the outer main loop of the simulator. The design of these controllers is discussed in Sec. 5. All of the off-diagonal transfer functions of the second linear model were chosen as zero because their outputs were at least 43 times smaller than to their corresponding inputs, which were of the same size and duration as those shown in all the previous I/O graphs.

4. Controller Design Methods

The design methods for the diagonal multi-loop as well as the diagonal $H_\infty$ controller for the LTI models of the Steckel hot rolling process are given in this section.

4.1. Diagonal Control

The application of SISO-PID/PI control to close the loop around the transfer functions on the diagonal, i.e. $g_{11}(s)$, $g_{22}(s)$ and $g_{33}(s)$ of the first linear model was considered as an option for a controller. This was done because the inputs applied to these transfer functions have a more direct influence on the outputs ($y_1, y_2, y_3$) than inputs applied to the off-diagonal transfer functions. This is apparent from the simulations in Sec. 3. Two controller design methods have been used for the individual controllers of the diagonal SISO multi-loop controller scheme.

4.1.1. Controller Design Method for $g_{11}(s)$

With $g_{11}(s)$ of the form

$$g_{11}(s) = G_{11}(s) = \frac{s e^{-\tau_c}}{v_a s + 1} \tag{17}$$

the computation of the controller $K_p(s)$ is based on an affine parameterization$^{[3]}$ and takes the form

$$K_p(s) = \frac{Q(s)}{1 - Q(s) G_{11}(s)}$$

$$= \frac{(\tau_v v_a)^2 + (\tau_v + 2 v_a)s + 2}{(2\tau_v \alpha_s)^2 + (2\tau_v \alpha_s + \tau_v) s} \tag{18}$$

where $G_{11}(s)$ is a first order Padé approximation of $G(s)$ and $Q(s) = F_1(s) G_{11}(s)$ with $F_1(s) = 1/(\alpha_s s^2 + \alpha_s s + 1)$ and $G_{11}(s)$ an approximated inverse of $G_{11}(s)$ and $\alpha_s$, $\alpha_l$ the tuning parameters.

The controller parameters of

$$K_{PID}(s) = K_p + \frac{K_i}{s} + \frac{K_ds}{\tau_ds + 1} \tag{19}$$

are obtained by comparison of the coefficients of $s$ in Eqs. (18) and (19).

4.1.2. Controller Design Method for $g_{22}(s)$ and $g_{33}(s)$

A SISO pole assignment technique, based on a polynomial approach$^{[3]}$ was used to find the PID controller parameters for the controllers for $g_{22}(s)$ and $g_{33}(s)$. If the controller and model transfer functions are respectively,

$$K(s) = \frac{P(s)}{L(s)} \quad \text{and} \quad G_{22}(s) = \frac{B_2(s)}{A_2(s)} \tag{20}$$

then $A_2(s)$ can be a chosen polynomial of degree $n_2 = 2n_1$ and the polynomials $P(s)$ and $L(s)$, with degrees $n_1 = n_2 = n$, such that,

$$A_2(s) L(s) + B_2(s) P(s) = A_2(s) \quad \tag{21}$$

For the PI synthesis the polynomials $P(s)$ and $L(s)$ were chosen as,

$$P(s) = p_s + p_0 \quad \tag{22}$$

$$L(s) = s L_0 \quad \tag{23}$$

$$L(s) = s L_0 \quad \tag{24}$$

By comparison of coefficients in Eq. (21) the coefficients $P(s)$ and $L(s)$ can be determined and thus also the parameters of the PI controllers.

4.2 $H_\infty$ Controller Design Method

Since a MIMO (multi-input-multi-output) control method is discussed in this section, lower and upper case letters used in the figures here denote vectors and matrices respectively.
4.2.1. The $H_\infty$ Control Problem

When computing an admissible controller, $K(s)$, the $H_\infty$-norm of the closed loop transfer function $\tilde{H}_{g\tilde{u}}$ (see Fig. 14) has to be minimized such that, given $\gamma > 0$,

$$\|\tilde{H}_{g\tilde{u}}\|_{\infty} \leq \gamma$$

with

$$\min_{K(s)} \|\tilde{H}_{g\tilde{u}}\|_{\infty} = \gamma_{optimal}, \quad \gamma_{optimal} < \gamma$$

with the $H_\infty$-norm defined as,

$$\|G\|_{\infty} = \sup_{\omega \in \mathbb{R}} \delta(G(j\omega))$$

i.e. the least upper bound of $G$, where $\delta$ denotes the maximum singular value.\(^{14}\)

4.2.2. Weights for Plant Augmentation

As shown in Fig. 15 plant $G(s)$ was in this problem augmented by weighting functions, $W_i(s) = c_i(s) - I_{w_i}(i = 1, 2, 3)$, to form plant $P(s)$. Two bounds were taken into account when selecting weights: The performance bound given by

$$\delta(S(j\omega)) \leq \delta(S^{-1}(j\omega)), \quad \omega < \omega_c$$

and the robustness bound given by

$$\delta(T(j\omega)) \leq \delta(W^{-1}(j\omega)), \quad \omega > \omega_c$$

where $\omega_c$ denotes the crossover frequency and $S$ and $T$ the sensitivity and complementary sensitivity functions of the standard feedback configuration without weights, respectively.

In Fig. 15 $\tilde{u}_1 = \tilde{u}$ is the exogenous input vector and $\tilde{y}_1 = \tilde{y}$ is the output vector. Let plant $P(s)$ be given by its state space realization as

$$P(s) := \begin{bmatrix} A & B_1 & B_2 \\ C_1 & D_{11} & D_{12} \\ C_2 & D_{21} & D_{22} \end{bmatrix}$$

To be able to apply the $H_\infty$ synthesis successfully, $P(s)$ must have the following properties:

- the state space description of $P(s)$, Eq. (29), is detectable and stabilizable,
- $D_{11}$ and $D_{22}$ must be zero and
- $W_i(s)$ and/or $W(s)$ and $G(s)$ have to be proper.

4.2.3. $H_\infty$ Synthesis

The augmented plant, $P(s)$, is scaled by the iteration parameter, $\gamma$. The $H_\infty$ synthesis applied in this work is a standard one and is also available in Matlab.\(^{19}\) It involves an iteration consisting of the solution of two modified Riccati equations. The iteration is started with an estimate for $\gamma$. The next step is to determine unique, real, symmetric solutions of two algebraic Riccati equations (ARE).

5. Controller Design

5.1. Controller Specifications

Controller specifications are given in terms of the following requirements\(^{17}\):

i) Exit gauge is to be kept within 1% of its setup value in the presence of control error changes.

ii) Back and front tensions are to be kept within 40% of their setup values, i.e.

$$|T_i| \leq 1.32 \cdot 10^4 \pm 5.3 \cdot 10^4 \mathrm{N}$$

5.2. Diagonal Controller

The diagonal controller $K(s)$ has the transfer function matrix

$$K(s) = \text{diag}[k_{11}(s), k_{22}(s), k_{33}(s)]$$

A PID controller was designed for $k_{11}(s)$ as discussed in section 4.1.1 with $\alpha_1 = 0.55$ and $\alpha_2 = 0.03$. Overshoot and bandwidth of the SISO system, consisting of $k_{11}(s)$ and $\tilde{g}_{11}(s)$ as controller and plant respectively, can be influenced by adjusting only $\alpha_1$ and $\alpha_2$.

$k_{12}(s)$ and $k_{33}(s)$ were designed as PI controllers as discussed in Sec. 4.1.2.\(^{16}\) For $k_{22}(s)$ and $k_{33}(s)$, $A_{ij}(s)$ was chosen to be

$$A_{ij}(s) = (s + 95)^2(s + 12)$$

and

$$A_{ij}(s) = (s + 95)^2(s + 15)$$

respectively. The closed loop poles are chosen such that overshoot is reduced. This can be done, by making the slow poles less negative. The purpose of the fast poles in $A_{ij}(s)$ is to give a fast response, i.e. a large bandwidth. The tuning parameters that were obtained for $k_{12}(s)$ and $k_{33}(s)$ are given in Table 2 and have the form

$$k_{ij}(s) = k_{ij} \left(\frac{\tau_i \sigma + 1}{\tau_i \sigma}\right), \quad i \in \{2, 3\}$$

5.3. $H_\infty$ Controller for the Second Linear Model

Augmentation of the linear plant by weights was done as in Fig. 15 with the following weighting functions
The following steps were performed to tune the diagonal \( H_0 \) controller:

i) Adjust parameters in positions (2, 2) and (3, 3) of \( W_i(s) \) in order to affect the speed response. Large differences between values of the elements of \( W_i(s) \) were avoided, in order not to violate the performance bound Eq. (27).

ii) Adjust parameter in position (1, 1) of \( W_i(s) \) in order to meet the performance bound specification if it has been violated in step i).

iii) If necessary the gain of \( W_i(s) \) needs to be adjusted such that the controller’s performance for output \( y_i \) is the same as that of the diagonal controller. Steps 1 and 2 were repeated if the general performance of the controller deteriorated.

### 6. Results

This section contains information about the controller implementation and also the results of the implementation of the controllers designed in Sec. 5 on the nonlinear simulator. The controllers were tested for a step from 10 to 1.22 \( \times 10^4 \) N in front tension at \( t=0.125 \) s. This corresponds to the real situation when front tension undergoes a severe increase up to its setup value before or during the acceleration phase of the mill.

#### 6.1. Controller Implementation

The two controller schemes, designed in the previous section, were implemented in discretized form on the simulator. The discretization of the diagonal PID/PI controller was done by using the velocity form to compute the controller outputs. A minimal realization of the continuous state space matrices of the diagonal \( H_0 \) controller was discretized. In the form of discrete state space matrices, in which the scaling matrices were absorbed, the latter controller was implemented on the non-linear simulator.

Because the \( H_0 \) controller, \( K(s) \), was designed for a scaled plant, \( G_0(s) \):

\[
G_0(s) = Q_s G(s) Q_s^{-1}
\]

scaling matrices and had to be absorbed in as

\[
K(s) = Q_s^{-1} K(s) Q_s^{-1}
\]

in order to obtain the controller, \( K(s) \), which could be implemented, where

\[
Q_s = \text{diag}[1/\mu_{\text{max}}] = \text{diag}[1000, 2.5, 2.5]
\]

and

\[
Q_s = \text{diag}[1/\gamma_{\text{max}}] = \text{diag}[3333, 5 \times 10^{-7}, 2 \times 10^{-5}]
\]

The velocity form was also used for the discretization of the controllers of the inner loop tension control. The PI controllers designed in the previous section were used for the inner loop tension control loop. When the \( H_0 \) controller was implemented to a simulator with inner loop tension control the control actions, \( u_2 \) and \( u_3 \), of these PI controllers were applied to the inner loop, i.e. at a higher frequency than the action, \( u_3 \), of the \( H_0 \) controller in the simulator’s main loop. Before application of \( u_2 \) and \( u_3 \), of the \( H_0 \) controller these two control actions were summed with those calculated by the inner loop tension controller for the previous time step.

#### 6.2. Discussion of Results

According to the controller specification the maximum allowable exit gauge deviation is 99 \( \mu \)m while the tension deviations from the setup values are allowed to be within \( \pm 5.3 \times 10^3 \) N and \( \pm 4.9 \times 10^4 \) N for back and front tensions respectively.

Figures 16 and 20 show that both controller schemes meet specification i) since the exit gauge changes in these figures are less than \( 99 \mu \)m. Specification ii) is also met by both controllers because Figs 17, 18, 21 and 22 show that the tensions are regulated within the above mentioned tension limits. It can be seen in Figs. 19 and 23 that the control actions have points at \( t=0.125 \) s in which derivatives are nonexistent. At these points the control actions have nonzero initial values which is the result of the velocity form implementation of PI controllers. In practice the controller would command such a value but it would result in a response of the plant as if the control action is smoother because of the slew-rate limit imposed by the coiler’s moment of inertia. From the point \( t=0.125 \) s onwards the average slew-rates are 100 m/s and 115 m/s as can be seen from the rates at which \( v_n \) changes in Figs. 19 and 23 respectively. Therefore the diagonal PID/PI and the diagonal \( H_0 \) controller meet the requirements specified in section 5.1 iii). In Figs. 16, 17, 20 and 21 it can be seen that the control actions \( u_3 \) and \( u_2 \) of both controllers lie within the limits specified in 5.1 iii).

The weaker performance of the diagonal PID/PI controller can be attributed to model mismatch. In the case of the diagonal PID/PI controller the PID controller, \( k_t(s) \), was designed for a transfer function, \( G_t(s) \), for which the tension model was excluded during SID, i.e. SID was performed on an incomplete model.

#### 6.6. Context and Evaluation of Controller Designs

The controller designs in this work are meant to control the process for a time fraction of the speed up ramp at 3.5 \( \pm 0.2 \) m/s main drive speed. This means that the controller has 0.5 s to regulate the output variables within specified limits.

A set of controllers each operating at a different main drive speed range would be needed for control during the speed up phase, requiring a switching control system as discussed in Chang et al. These authors give a criterion for the switching time at which each controller in the set comes into action. According to Chang et al. the switching time is determined by a bounding function and a norm of a vector of which the components are the switching controller states as well as filtered errors of the controlled outputs. Developing a switching control strategy and its implement-
7. Conclusion

The investigations addressed in this work were aimed at the implementation of gauge and tension control on a speed up ramp of a simulated Steckel hot rolling mill process. For the design of two different controllers, two different linear models were identified from a non-linear plant simulator. For the second linear model gauge meter compensation and inner loop tension control were made part of the simulator to prevent tensions from oscillating and becoming negative. From system identification and implementation of a diagonal SISO multi-loop and diagonal $H_\infty$ controller on the non-linear simulator the following can be concluded.
(i) The first and second LTI models are suitable for diagonal SISO multi-loop and diagonal $H_\infty$ controller design respectively because model mismatch in their case does not affect their design to such an extent that the controllers cannot be used as switching controllers.

(ii) Gauge meter compensation and inner loop tension control make it possible that an LTI model with transfer functions only on the diagonal can be identified and used for design of a diagonal $H_\infty$ controller, i.e. the process gets decoupled to an extent.

(iii) The diagonal SISO multi-loop controller as well as the diagonal $H_\infty$ controller based on the second linear model, meet the requirements specified in 5.1, given the conditions under which they were tested. They are also suitable to be used in a switching control system.

This work shows that control on the speed-up ramp is possible and has the possible advantage of reducing offspecification strip produced during this phase of rolling. Such a benefit can become real when the suited controllers as designed in this work are used within a switching control system valid for the whole speed up ramp.

**Nomenclature**

- $\Omega$: Work roll rotational speed [rad/s]
- $R$: Work roll radius [m]
- $R'$: Deformed work roll radius [m]
- $\kappa_t$: Steady state gain
- $\tau_c$: Time constant
- $\tau_D$: Delay time
- $F_\delta(t)$: Design filter
- $\kappa_{pI}$, $\kappa_{D}$, $\tau_c$: PID controller parameters
- $Q_t$, $Q_e$: Scaling matrices

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