Controller for Improving the Quality of the Tandem Rolling of Hot Metal Strip

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Abstract—The tandem hot metal strip rolling process presents a difficult control challenge because of its highly complex and nonlinear nature. This challenge is heightened by the hostile hot metal rolling environment which precludes the location of certain sensors to measure variables that are important for control. Based on our previous work using a state-dependent Riccati equation technique for development of a controller for the tandem cold metal rolling process, it is considered that a similar basis could be expanded upon to realize an improved method for control of this more complex application. In this paper we present a comprehensive model of this process plus the results of our first efforts in the development of a suitable controller, which for control of this application is different than previous methods. The results of simulations of the controller coupled to the model show a strong potential for improvement in the quality of the final product.

I. INTRODUCTION

A significant process in the areas of manufacturing and processing of metals is the tandem rolling of hot metal strip. Fig. 1 depicts a typical hot metal rolling process wherein large metal slabs which have been produced in a previous rolling or casting operation are placed in a reheating furnace and heated to temperatures suitable for intermediate processing and subsequent entry into the tandem hot strip finishing mill.

In the mill the strip is passed through a set of five to seven pairs of independently driven work rolls, with each work roll supported by a back-up roll of larger diameter. Between each pair of work rolls there is a looper (Fig. 2) which is a mechanism consisting of an arm and roll driven by hydraulics or an electric motor to keep the strip at a reference tension.

As the strip passes through the individual pairs of work rolls in the finishing mill the thickness is successively reduced. This is caused by hydraulic cylinders which produce very high compressive stress in a small region (the roll bite, or roll gap) between the work rolls. After exiting the last stand, the strip travels toward a coiling mechanism over a run-out table where it is cooled by water sprays to achieve desired metallurgical properties. An extremely hostile hot rolling environment precludes the reliable measurement of significant process variables such as strip speed and intermediate stand exit thickness.

It is a difficult task to develop a controller for this complex nonlinear process which has a wide range of uncertainties, major disturbances, and significant time delays. However, in almost all of the conventional and advanced control concepts implemented to date or simulated in academia (cf. [1] presents a survey of advanced concepts), only two adjacent mill stands with their associated looper have been considered. Although many improvements have been realized, most concepts have failed to fully address the complex dynamic interactions between all of the mill variables. A few attempts have been made treating the mill as a single system using linearized models, but with resulting complexities that were found to be unacceptable in a practical setting mostly due to difficulties in making adjustments to controller settings during commissioning. Thus there is a need for a completely different approach that can address the complex interactions between variables, reduces the computational efforts associated with linearization, that is user-friendly to design and commissioning personnel and that can significantly improve the quality and yield of the final product.

It is considered that the methods that were highly successful in our previous work with tandem cold rolling...
[2]-[4] can be expanded upon to develop a radically different technique for controlling the tandem hot rolling process. However, the control of this process is more difficult as it must address the additional issues of the lack of measurement of significant variables, a more complex model due to the higher strip temperatures, and control of the loopers which are not present in a tandem cold mill.

In this paper we present the results of our initial investigation into the development of a technique to control this complex process in its entirety as a single system and show by simulation that the method developed has a strong potential for improvement in performance. We also note that for this investigation, a method of active compensation for mill roll eccentricity such as that described in [4] is assumed to be operable so that any eccentricity components remaining after compensation are taken to be insignificant.

Except where noted otherwise, symbols used herein are as given in Table I.

| TABLE I |
| LISTING OF SYMBOLS |
| A(s) state-dependent matrix | o subscript, stand output value |
| a(s) state-dependent vector | P specific roll force |
| B control matrix | Q(s) state weighting matrix |
| C(s) state-dependent output matrix | R undeformed work roll radius |
| E Young's modulus | R(s) control weighting matrix |
| e subscript, estimated value | f forward slip |
| F total rolling force | g(s) state-dependent vector |
| h strip thickness | i subscript, stand i |
| in subscript, stand input | J performance index |
| Jlpr looper moment of inertia | K(s) solution to Riccati equation |
| L0 length between c/l of stands | k constrained yield stress |
| L strip length between stands | m subscript, measured value |
| M mill modulus | o or op subscript, operating pt value |
| Md1 loop torque, bending | op subscript, stand output value |
| Md1 loop torque, friction | Qp work roll peripheral speed |
| Md2 loop torque, total load | σ tension stress |
| Md3 loop torque, lpr mass | σv av tension stress = (σiy + σout)/2 |
| Mfcp work roll speed actuator ref | τp time constant, roll gap pos controller |
| Mlpr torque applied to looper | τy time constant, work roll spd controller |
| Mout loop torque, strip tension | ϕ angle at neutral plane |
| m subscript, measured value | ω looper angular velocity |

Table II lists the operating point strip thickness, the average strip temperature at the mill entry and at the exit of each stand, the peripheral speed of the work rolls, and the undeformed work roll radius of each stand. The basis for the prediction of the roll force in the roll bite area (Fig. 3) is Sims model [6] which we have enhanced by using the comprehensive empirical results of Shida [7] to better estimate the constrained yield stress of the material being rolled.

| TABLE II |
| MILL OPERATING POINT |
| Stand | hout (mm) | T (°C) | Vp (m/sec) | R (mm) |
| entry | 38.8 | 1058 | ---- | ---- |
| 1 | 21.6 | 988 | 1.188 | 360 |
| 2 | 14.4 | 973 | 1.823 | 336 |
| 3 | 8.6 | 957 | 2.957 | 353 |
| 4 | 6.1 | 938 | 4.294 | 343 |
| 5 | 4.7 | 922 | 5.665 | 388 |
| 6 | 3.9 | 904 | 6.946 | 348 |
| 7 | 3.5 | 894 | 7.880 | 369 |

II. MATHEMATICAL MODEL

The tandem hot strip rolling process is modeled as a set of mathematical expressions which relate the rolling parameters to each other. The salient features of the model, which has been developed and verified as part of previous work [5], are presented in what follows. The operating point of the mill is based on a fully threaded condition at operating speed, with a strip tension of 0.01 kN/mm² between adjacent stands, and with each looper at an angle of 15 degrees.

In Sims' model the specific roll force is represented as

\[ P = (k Q_P - \sigma) \sqrt{R_P \sigma} , \]  

(1)

where \( Q_P \) is a factor developed in Sims' paper which compensates for friction and any inhomogeneities of deformation, and \( R_P \) is approximated by the Hitchcock relation [8]. The exit thickness is estimated using the linearized relation for the output thickness as

\[ h_{out} = S + S_0 + \frac{F}{M} , \]  

(2)

The forward slip \( f \) is a measure of the strip speed exiting the roll bite and is defined as the ratio of the relative velocity of the exiting strip to the peripheral speed of the roll,

\[ f = \frac{V_{out} - V_0}{V_0} . \]  

(3)

A model that describes the forward slip and is more useful for control development is that presented in Ford, Ellis, and Bland [9] for cold metal rolling, except that for hot rolling the empirical relationship given in Roberts [10] for the
The coefficient of sticking friction is used in place of the coefficient for sliding friction which is used for cold rolling. A relationship for strip tension is derived from the relationship for Young's modulus,

$$\frac{d\sigma}{dt} = \frac{E}{L_0} \left[ \frac{dL(\theta(t))}{dt} + V_{in,j+1} - V_{out,j} \right], \quad \sigma(0) = \sigma_0. \quad (4)$$

The position of the hydraulic cylinder that sets the work roll position at the roll bite, and the peripheral speed of the work rolls, are modeled as single first order lags,

$$\frac{dS}{dt} = \frac{U_S}{\tau_S} \cdot S, \quad S(0) = S_0, \quad (5)$$

$$\frac{dV}{dt} = \frac{U_V}{\tau_V} \cdot V, \quad V(0) = V_0. \quad (6)$$

The interstand time delay, with its associated effects on thickness and temperature, is the time taken for an element of strip to move between adjacent stands and is approximated as

$$\tau_{d,j+1} = \frac{L}{V_{out,j}}. \quad (7)$$

The looper position angle is determined as

$$\frac{d\theta}{dt} = \omega, \quad \theta(0) = \theta_0, \quad (8)$$

where $\omega$ is the looper angular velocity which is derived from Newton's second law of motion, and is described as

$$\frac{d\omega}{dt} = \frac{1}{J_{lpr}} \left[ M_{lpr} + M_{fct} + M_{td} \right], \quad \omega(0) = 0. \quad (9)$$

where $M_{td} = M_{tan} + M_{sto} + M_{bas} + M_{bnd} + M_{fct}$. The torque $M_{lpr}$ is approximated as a first order lag which includes the looper hydraulic cylinder with its controller,

$$\frac{dM_{lpr}}{dt} = \frac{U_{lpr}}{\tau_M} \cdot \frac{M_{lpr}(0)}{\tau_M} = \frac{M_{lpr}(0)}{\tau_M}. \quad (10)$$

The friction torque of the looper mechanism is approximated for this investigation as

$$M_{fct} = k_{visc} \cdot \omega. \quad (11)$$

The Appendix gives calculations for the looper torques, moment of inertia, and $dL(\theta(t))/dt$.

The equations of the model representing the nonlinear process dynamics initially are expressed in the form

$$\dot{x} = a(x) + Bu, \quad x(0) = x_0, \quad (12)$$

$$y = g(x). \quad (13)$$

The variables represented by the elements of the state, control, and output vectors are as shown in Table III.

For use in the controller simulation, (12) and (13) are modified to be expressed as

$$x = A(x)x + Bu, \quad x(0) = x_0, \quad (14)$$

$$y = C(x)x, \quad (15)$$

where $a(x)$ and $g(x)$ are factorized (nonuniquely) into the forms $A(x)x$ and $C(x)x$, where $A(x) \in \mathbb{R}^{n \times n}$ is a state-dependent matrix, $C(x) \in \mathbb{R}^{p \times n}$ is a state-dependent matrix, and with $x, y, u$, and $B$ as previously noted. The elements of the $A(x), C(x)$, and $B$ matrices are as determined in our previous work [5].

The model was successfully verified as part of previous work wherein our simulation results were compared against the verified simulation results of others and against data from actual installations.

III. CONTROLLER

A. The SDRE Technique

The controller is based on the use of the state-dependent Riccati equation (SDRE) technique, which is becoming recognized as a desirable and useful nonlinear method for the control of many nonlinear applications due to its simplicity and its capability for promoting physical intuition in the design process. A brief overview follows. More detailed treatments can be found in [11]-[13].

The SDRE technique is similar to the well-established LQR method except that the coefficient matrices in the state and output equations, and the control and state weighting matrices, are state dependent. This technique is developed by expressing the nonlinear plant dynamics in the form as noted previously in (14) and (15). The optimal control problem is then defined in terms of minimizing the performance index

$$J = \frac{1}{2} \int_0^\infty (x'Q(x)x + u'R(x)u)dt \quad (16)$$
with respect to the control vector $u$, subject to the constraint (14), where $Q(x) \geq 0$, $R(x) > 0$, $Q(x)$ and $R(x) \in \mathbb{C}^k$ for $k \geq 1$. Essentially (16) implies finding a control law which regulates the system to the origin. The state-dependent algebraic Riccati equation

$$A'(x)K(x) + K(x)A(x) - K(x)B R^{-1}(x) B' K(x) + Q(x) = 0$$

is solved pointwise for $K(x)$, which results in the control law

$$u = -R^{-1}(x) B' K(x) x.$$ (18)

The method requires that the pair $(A(x),B)$ be pointwise stabilizable (in a linear sense) for all $x$ in the control space in order to ensure a solution to (17) at each point. While local asymptotic stability is assured under some fairly mild conditions [12], in general asymptotic stability over the control space must be confirmed by simulation.

B. Application to Control of Tandem Hot Strip Rolling

To assure good quality of the final strip output and to assure the stability of rolling, it is essential to reduce excursions in the interstand tensions and looper positions, as well as reducing excursions in the strip thicknesses. This must be done with the consideration that measurements of several of the more significant variables such as strip speeds, output thicknesses at the first and intermediate stands, strip temperatures, and the material properties of the workpiece are unavailable. In response to this requirement, the controller is configured such that the effects of excursions in the unmeasured variables are estimated by using measured variables, the model, and data collected during normal processing functions to refine the model as needed. As in (2) for example, except at the last stand where a thickness measurement is available, the exit thickness of the strip is inferred by measurements of the roll force, the roll gap actuator position, and an estimation of the mill modulus which is refined based on data collected during actual process functions. Moreover, the nonlinear MIMO nature of the SDRE technique provides an effective means of handling the interactions between variables, both measured and unmeasured, throughout the entire mill. In addition, the SDRE technique is augmented by outer loop SISO trimming functions (trims) which reduce slight offsets in the looper position and provide zero steady-state error in the measured tensions and estimated strip thicknesses. The closed-loop control action of the trims also contributes heavily toward reducing the effects of uncertainties and of various unmodeled perturbations since many of both of these occur inside the control loops of the trims, so that the trims significantly enhance the robustness of the controller to these types of effects. Also, the trims are effective in reducing the effects of the interstand time delays, and provide a simple and user-friendly means of adjustment of certain controller settings which facilitates commissioning and operational functions. As noted previously, an SDRE-based system in general requires verification of stability by simulation as at present there is no supporting theory to assure it. This also carries over to the addition of the trims, i.e. stability is established by simulation.

The uncertainties considered in modeling are the mill entry thickness ($\pm 5\%$), the Table II workpiece temperatures ($\pm 10^\circ \text{C}$, based on comparisons with mill data [5]), the carbon content ($\pm 30\%$) of the workpiece, plus the mill modulus ($\pm 10\%$). The uncertainties in measurement are strip tension stress ($\pm 1\%$, based on estimation using measurements of looper parameters), in measurement of roll gap actuator position, work roll peripheral speed, roll force, looper torque, angle, and angular velocity are on the order of 0.1% and are shown by simulation to be less significant. For purposes of this initial investigation the uncertainties in temperatures and carbon content are taken as invariant for the entire length of the strip, and Young’s modulus, strip width, and density are taken to be constant.

![Typical disturbances at mill entry](image_url)

The major disturbances addressed are variations in incoming thickness and temperature due to skid chill in the reheating furnace as shown in Fig. 4. The temperature disturbances in the incoming material are tracked from stand to stand as the strip moves through the mill, and are combined with the temperature uncertainties and the temperatures given in Table II. The disturbances and uncertainties are combined concurrently to simulate the more severe scenarios (Section IV).

The data presented are based on manufacturer’s information, typical operational characteristics, and experience. The percentages and values represent deviations from nominal operating point values.

C. Structure of the Controller

Fig. 5 depicts the controller structure wherein the effects of disturbances and uncertainties are modeled separately and depicted as shown for simplicity of presentation. The vectors $x$, $u$, and $y$ at the operating point are represented as $x_{\text{op}}$, $u_{\text{op}}$, and $y_{\text{op}}$. A coordinate change is performed by the introduction of the vector $z=x-x_{\text{op}}$ which shifts the operating point to the origin. The performance index (16) then is modified to be
\[ J = \frac{1}{2} \int_{0}^{\infty} \left( z'Qz + (u - u_{op})'R(u - u_{op}) \right) dt, \quad (19) \]

where for simplicity \( Q \) and \( R \) are taken for this initial effort as diagonal matrices with adjustable constant elements.

Each element of the state vector \( x \) is measurable, \( y_m \) represents the measurable elements of the output vector \( y \), \( y_c \) is a vector whose elements are the measured (or estimated) elements of \( y \), and \( \varphi_y \) is an algorithm which generates \( y_c \).

This algorithm uses inputs from measured variables to generate estimates of the tension stress and the stand output thickness. The uncertainties in the measured variables, in the estimated tensions and thicknesses, plus uncertainties and estimates of other variables, and the bases for these estimates as used in the simulation are as noted herein. The \( K_P \) and \( K_I \) blocks are diagonal matrices whose elements are the gains for the thickness and tension trims.

The output thickness at each mill stand is estimated using (2), with an estimate of the mill modulus \( M \) that is updated as \( M \) changes with operational effects such as roll temperature and mechanical wear. At the output of the last stand the strip thickness is determined using (2) and a refined estimation [4] of the mill modulus based on the measured thickness.

The pointwise solving of (17) sufficiently fast to properly control the process is assured by use of the matrix sign function technique [14]. The speed of solving (17) using this technique has been initially verified to be about 7 ms by previous simulation [5].

### IV. Simulation

Simulations (closed-loop) were performed using MATLAB/Simulink with the controller (Fig. 5) coupled to the model and using the looper properties noted in the Appendix. The results presented are for the operating point of Table II. A separate means of compensation for mill modulus uncertainty to handle variations in \( M \) in the linear approximation (2) is taken to be operative so that the \( M \) used for simulation is assumed to be within 2% of its nominal value (3980 kN/mm). The settings of the constant elements of the diagonal \( Q \) and \( R \) matrices and the gain settings of the trims were made intuitively and confirmed by simulation.

Simulations were performed for a fully threaded mill operating at constant speed. Nine cases were examined for uncertainties in the mill modulus ranging between +/-2%, in the interstand tension stress between +/-1%, in the looper positions between +/-0.1%, with uncertainty in the mill entry thickness of 5%, in the carbon content and temperature of the workpiece of 30% and +/-10 degrees C, and all applied concurrently with the disturbances (Fig. 4) and in a manner to simulate worst case scenarios. Some typical results are presented in Table IV.

<table>
<thead>
<tr>
<th>Variable</th>
<th>0%</th>
<th>1%</th>
<th>-1%</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \sigma )</td>
<td>0.1%</td>
<td>1.1%</td>
<td>1.1%</td>
</tr>
<tr>
<td>( \theta )</td>
<td>0.1</td>
<td>0.1</td>
<td>0.1</td>
</tr>
<tr>
<td>( h_{out} ) to ( h_{out}^{ref} )</td>
<td>0.7</td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td>( h_{out}^{ref} )</td>
<td>neg</td>
<td>neg</td>
<td>neg</td>
</tr>
</tbody>
</table>

As noted in Table IV, the controller exhibits excellent performance in reducing excursions in the interstand tensions and looper angles which contributes strongly to the improvement in the quality of the final product. The excursions in the interstand thicknesses also are well-controlled which enhances the stability of rolling. The results of Table IV compare well with results that would be generally expected using conventional methods. Specific and detailed comparisons to conventional and advanced
techniques will be made as part of ongoing and proposed work depending on available resources.

V. CONCLUSION

The results of this first effort show that the controller offers a strong potential for improvement in performance which ultimately contributes to the upgrading of the quality of the final product, and therefore is suitable for continuing investigation as a method which offers an improvement in the control of the tandem hot rolling process.

APPENDIX

Looper Characteristics

Dimensions: \( L_0 = 5.48 \text{ m} \), \( y = 0.19 \text{ m} \), \( a = 1.94 \text{ m} \), \( r \) (radius) = 0.15 m, \( l = 0.76 \text{ m} \). Max angle: 40 deg. Pass line angle: 2.9 deg. Mass of looper arm: 300 kg. Mass of looper roll: 500 kg. Viscous friction constant: \( k_{vis} = -2.0 \text{ kNm/rad/sec} \). Moment of inertia: \( J_{spr} = 0.35 \) (in kgm\(^2\)/1000).

At the steady-state operating point: \( \theta = 15 \text{ deg} \), \( \omega = 0 \) rad/sec, \( l_1 = 2.68 \text{ m}, l_2 = 2.81 \text{ m} \), \( M_{tot} = -4.29 \text{ kNm} \), \( M_{net} = -1.52 \text{ kNm} \), \( M_{max} = -4.69 \text{ kNm} \), \( M_{bnd} = -0.13 \text{ kNm} \), \( M_{fct} = 0 \) kNm, \( M_{id} = -10.63 \text{ kNm} \), and \( M_{spr} = 10.63 \text{ kNm} \).

With \( A = l \sin \theta; B = l \cos \theta; C = A + r; D = B + a \), and \( F_{ten} = \sigma_{06} h_{out} W \), where \( F_{ten} \) is the tension force at the looper between stands 6,7, \( W \) is strip width (1290 mm), and where

\[
\begin{align*}
L_1 &= \left( (r-y)^2 + (a+B)^2 \right)^{1/2}, \quad (20) \\
L_2 &= \left( (r-y+2A)^2 + (L_0-a-B)^2 \right)^{1/2}, \quad (21)
\end{align*}
\]

\[
M_{ten} = F_{67} C \left[ \frac{D}{11} (L_0 - D) \right] B (C - y) \left( \frac{1}{11} + \frac{1}{12} \right), \quad (22)
\]

\[
M_{out} = (11+12) W h_{out6} B (74.556 \times 10^{-6} \text{ kNm}), \quad (23)
\]

\[
M_{max} = B (0.00981) \left( \frac{300}{2} + 500 \right), \quad (24)
\]

\[
M_{bnd} = \left( \frac{5 \times 10^{-5}}{4} \right) B W h_{out6} \left( \frac{1}{11} + \frac{1}{12} \right), \quad (25)
\]

and

\[
J_{spr} = \frac{300 l^2}{3} + 500 \left( l^2 + r^2 \right) \div 2. \quad (26)
\]

\[
\frac{dL(\theta(t))}{dt} = \frac{dL(\theta(t))}{d\theta} \cdot \frac{d\theta}{dt}, \quad (27)
\]

where

\[
L = L_1 + L_2, \quad (28)
\]

\[
\frac{dL}{d\theta} = \frac{d(L_1)}{d\theta} + \frac{d(L_2)}{d\theta}, \quad (29)
\]

\[
\frac{d(L_1)}{d\theta} = \frac{l((r-y) \cos \theta - a \sin \theta)}{\sqrt{L_1}}, \quad (30)
\]

\[
\frac{d(L_2)}{d\theta} = \frac{l((r-y) \cos \theta - (L_0-a) \sin \theta)}{\sqrt{L_2}}, \quad (31)
\]

and

\[
\frac{d\theta}{dt} = \omega, \quad (32)
\]

with

\[
L(\theta(0)) = L(\theta(0)) + L(\theta(0)) \cdot \theta(\theta) = \theta_0. \quad (33)
\]

REFERENCES


