단일 스탠드 냉간압연공정의 정확한 두께 제어에 관한 연구
A Study on Accurate Thickness Control of a Single-Stand Cold Rolling Mill

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I. INTRODUCTION

Traditionally, the major portion of cold rolled metal strip was produced on large, tandem cold rolling mills having between 4 and 6 stands and producing up to 2 million tons per annum in coil form. The rapid growth in mini-mills and smaller sized flat products plants, especially in Asia, has encouraged the growth in single stand reversing mills. At the same time, there has been an increased emphasis on improved quality and a significant tightening in the thickness tolerances acceptable to downstream customers.

Because high speed response of roll position is available by a hydraulic screwdown system, the roll position AGC (automatic gauge control systems) is mainly used. However, a manipulation of roll position during rolling brings about not only change of exit thickness but change of front and back tensions which influence the exit thickness change, which inhibits fast closed loop response.

Rolling data exhibit an intriguing pause in the exit thickness in response to step change in roll position. The output appears to initially respond well but then pause, which is called "hold-up effect". This effect has been previously noticed. see Kondo et al (1988), Clark and Mu (1990) and Ueda et (1990). However, to the best of our knowledge, no satisfactory explanation has so far been advanced for its appearance. In order to pinpoint the cause of the hold-up effect, nonlinear and linearized model are built and two blocking zeros on the $j\omega$-axis from roll position to output thickness are found to be the cause of the hold-up effect which is fundamental and inescapable.

Finally, in order to avoid fundamental limitations in using the roll position. alternative input variables are brought into use. However each input has different fundamental limitations, a key issue is how to exploit the different characteristics of the multiple inputs. To accomplish this goal a Hard Load Sharing (HLS) algorithm was used to make full use of the available inputs and it is shown by simulation that the HLS scheme can be successfully applied to the rolling mill thickness control problem.

II. SIMULATION MODEL

2.1. Nonlinear Model

There are several variables that interact in a rolling mill stand. Those variables that attract particular attention are shown in Fig 1.

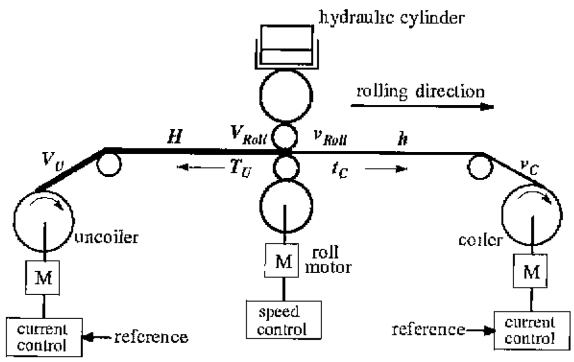

Fig. 1. Variables of Interest

To distinguish between the variables corresponding to the coiler and uncoiler, subscript C indicate variables corresponding to the coiler and subscript and U to the uncoiler.

For the purpose of deriving a model, the rolling mill process is divided into three parts. The first part is related to the dynamical behavior of the coiler and uncoiler reels. The second part corresponds to the rolls and the roll gap. The third part corresponds to the connection of the reels and the rolls, which is basically through tensions in the strip.

There also generally exits a current control loop. Usually, the electrical time constants are faster than those related with mass inertia such as the rotating reel. Furthermore. the current control loop is generally very fast. This suggests that a simplification can be made by assuming that the armature current is directly controllable. When the motor current reference is changed, the current control system corrects the motor terminal voltage or the motor revolution speed so that the actual motor current is consistent with the reference. Consequently, the correction of the motor revolution speed causes a change in the entry speed of strip as it goes into the roll bite and also a change in strip tension. On the other hand, the strip tension change caused by the roll position alteration is corrected by the current control system so that the strip tension returns to the original value.

For the motion of the reel, the variables involved are the electrical torque, the torque created by the strip tension, the radius of the reel and the angular velocity. The corresponding equations are

$$\frac{dr_U}{dt} = -\frac{H_o \omega_U}{2\pi} \qquad (1)$$

$$J_U \frac{d\omega_U}{dt} = T_U r_U - T_E$$

where H_o is the nominal input strip thickness, J_U is the uncoiler inertia, T_U is the uncoiler-side strip tension, T_E is the uncoiler motor torque, and r_U is the uncoiler radius. The equations for the coiler can be similarly determined.

For the purpose of control system design it is desirable to have a simplified formula for the roll force. A linear perturbation approximation is suitable for this purpose. The roll force change with constant output thickness is in linear form given by

$$\Delta F_D = W\left[\frac{\partial P}{\partial H}\Delta H + \frac{\partial P}{\partial T_U} + \frac{\partial P}{\partial t_C}\right] \qquad (2)$$
$$= A_1 \Delta H + A_3 T_U - A_4 t_C$$

where ΔH represents a disturbance in the strip entry thickness, ΔT_U and Δt_C represent entry and exit tension stress disturbance respectively. The partial derivatives in this equation can be evaluated at any particular steady state operating condition by using one of the complex roll gap theories or by making experimental measurements.

Any change in output thickness will introduce an additional force, so that the total force change may then be expressed as

$$\Delta F = \Delta F_D + W\frac{\partial P}{\partial h}\Delta h \qquad (3)$$
$$= A_1\Delta H - A_2\Delta h + A_3\Delta T_U - A_4\Delta t_C$$

In incremental form the following expression relates the change in thickness to changes in unloaded roll opening and roll force about the operating point

$$\Delta h = \Delta S + \frac{\Delta F}{M} \qquad (4)$$

where M is mill modulus. Δh is change in exit thickness, ΔS is change in unloaded roll opening and ΔF is change in total roll force.

Combining equations (3) and (4), an explicit expression for the exit thickness change can be obtained as

$$\Delta h = \frac{1}{M + A_2}\left(M\Delta S + A_1\Delta H + A_3\Delta T_U - A_4\Delta t_C\right) \qquad (5)$$

We assume that the exit strip velocity is a result of the tangential speed of the roll speed plus a slip effect. Hence we describe the exit velocity by

$$V_{Roll} = R_{Roll}\omega_{Roll}\left(K_{slip}(t_C - T_U) + K_R\right) \qquad (6)$$

where R_{Roll} and ω_{Roll} are respectively, the radius and the angular velocity of the roll. For most cold rolled products, the value of forward slip will be between zero and five percent. Forward slip increases with increasing delivery tension t_C and decreases with increasing entry tension T_U.

The interaction between the reels and the rolls is given by the connection of the subsystems through the strip. We make a simplifying assumption that the length of strip held between roll gap and coiler reel at any instant of time is considered to be equivalent to a massless spring whose ends are traveling with different velocities. Then a simplified model can be developed expressing the rate of change of the tension force in terms of the end velocities and an equivalent spring constant K_C. Assuming that the strip is stressed within its elastic limit and therefore obeys Hooke's Law, the tension in the coiler side strip is expressed as

$$t_C = K_C \int (v_C - v_{Roll})dt \qquad (7)$$

2.2. Simulation Result for Nonlinear Model

Simulation of the model has been performed in an operating condition when both reels have approximately the same radius. The closed loop corresponding to the

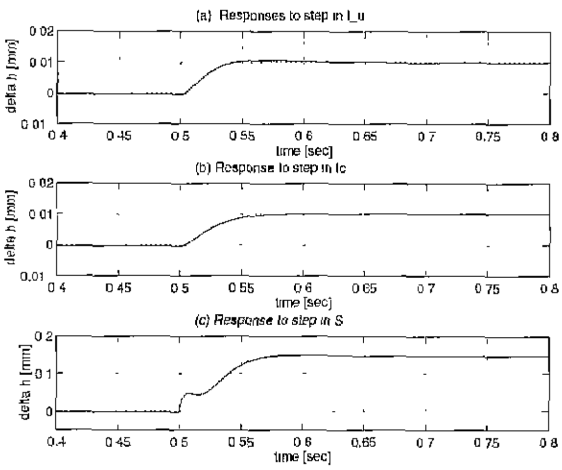

Fig. 2. Open loop step responses

speed control of the rolls is assumed to have a time constant of 1 [sec]. Also the reference for the armature current is set to a value such that, in the steady state, the input strip tension is 74 [kN] and the output strip tension is 61 [kN].

To examine the effect of the different input variables, step changes were made in the three inputs (I_U, I_C, S). Fig. 2 shows the manipulated variables affect the output. In the simulation, at t=0.5[sec], I_U changed from 30[A] to 20[A] and I_C from 20[A] to 10[A]. The rollgap was increased by 0.4[mm] at 0.5[sec].

We see from Fig 2 that the output (Δh) response to I_U and I_C are almost the same. Also, we see that the gain from rollgap S to the output is much bigger than from I_U and I_C. Close inspection of Fig 2 reveals that the response of to a step change in S is not monotonic but instead exhibits a temporary "hold-up" phenomenon after approximately 10[msec] following the step change in S. This response will be the subject of a detailed investigation in the following sections.

2.3. Linearized Model

The rolling model is inherently nonlinear. The nonlinearities come from the mass flow feedback, slip effect and magnetic flux change due to the change of radius of the reel. Mass flow feedback and slip effect are linearized and it is assumed for simplicity, that the reel radius is constant.

By linearizing the mass flow feedback and the slip effect, we can get a linear model which has 7 states and 4 inputs. The system is actually time-varying since reel radii and related mill parameters change accordingly. However, for the purpose of simulation, we will fix the radius at some nominal value.

The resulting linearized model can then be written in state space form as

$$x = Ax + Bu + P_x\Delta H$$
$$y = Cx + Du + P_y\Delta H \qquad (8)$$

where

$$y = \begin{bmatrix} T_U \\ t_C \\ \Delta h \end{bmatrix} = \begin{bmatrix} \text{input strip tension} \\ \text{output strip tension} \\ \text{output strip thickness change} \end{bmatrix}$$

$$u = \begin{bmatrix} I_U \\ I_C \\ S \end{bmatrix} = \begin{bmatrix} \text{armature current to uncoiler} \\ \text{armature current to coiler} \\ \text{roll gap} \end{bmatrix}$$

ΔH = *input strip thickness disturbance*

The various transfer function of the linear model were found to be as follows:

$$T_{hIu} = \frac{K_{Iu}\prod_{l=1}^{5}(s+Z_{Iu_l})}{\prod_{m=1}^{7}(s+P_{Iu_m})}, \quad T_{hIc} = \frac{K_{Ic}\prod_{l=1}^{5}(s+Z_{Ic_l})}{\prod_{m=1}^{7}(s+P_{Ic_m})} \qquad (9)$$

$$T_{hS} = \frac{K_S\prod_{l=1}^{6}(s+Z_{S_l})}{\prod_{m=1}^{7}(s+P_{S_m})}, \quad T_{hH} = \frac{K_H\prod_{l=1}^{7}(s+Z_{H_l})}{\prod_{m=1}^{7}(s+P_{H_m})}$$

where T_{hS} is the transfer function from roll position to output thickness and other transfer functions are similarly defined.

Fig. 3. compares the response of the 7 state linear model with the nonlinear model. The figure corresponds the response of a step of 0.2 [mm] in the input thickness applied at 1.5 [sec] and when the roll position is closed by 0.4 [mm] at 2 [sec]. It can be seen that the correspondence is remarkably good. This gives us confidence to use the linear model to gain quantitative insight into the performance of the system. We also see that the same hold-up effect is present in the linear model response. We therefore proceed in the next section to see if an explanation for the effect can be obtained from linear theory.

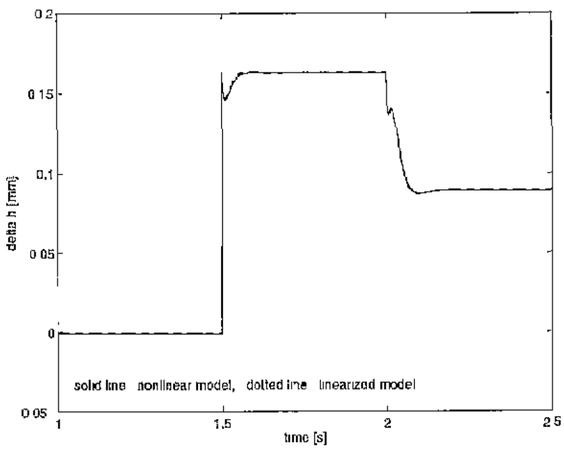

Fig. 3. Comparison between nonlinear and linearized model responses

III. HOLD-UP EFFECT

3.1 Pole-zero Analysis of the Linearized Model

A key observation is that T_{hS} has two purely imaginary zeros on the $j\omega$-axis at approximately 86 [rad sec^{-1}]. The impact of the $j\omega$-axis zeros identified is that, any attempt to move the roll position with frequency 86 [rad sec^{-1}], will not change the output thickness.

3.2 Physical Explanation for $j\omega$-Axis Zeros

We recall that the strip acts as an (undamped) spring. However, damping is provided by a feedback mechanism through slip. Moreover, the rollgap change acts directly on

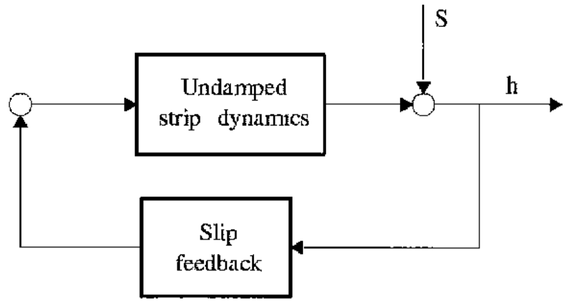

Fig. 4. Slip Feedback

the output (without dynamics). Hence, we can picture the physical interaction in Fig. 4.

If we let the transfer function of the undamped strip dynamics be $T(s)=N(s)/D(s)$ then the closed loop transfer function from S to h in Fig 4 is

$$T_d(s)=\frac{1}{1+k\frac{N(s)}{D(s)}}=\frac{D(s)}{D(s)+kN(s)}$$

where k is the gain of the slip effect. We thus see that the action of the feedback loop in Fig 4 has the effect of placing the open loop poles (including the undamped resonant pair associated with the strip) as zeros of the transfer function from S to Δh. This explains the presence of the zeros in the model. Actually in practice the resonant zeros appear at the average location of the open loop resonant poles corresponding to the coiler and uncoiler sides.

3.3. Single-loop Control Strategies

It is possible via the use of a soft sensor (Gaugemeter or Mass-Flow Estimator) to effectively eliminate the delay between the rollgap and the (estimated) exit thickness. Hence we will essentially ignore the time delay in the measurement of the exit thickness. Of course, the presence of any time-delay in practice would make matter worse and thus we are considering an ideal base-case for the control system performance.

The roll position is used as the actuator, which is in accordance with the usual practice in thickness control system. We study simple feedback controllers connecting the output strip thickness and the rollgap change. Simulation results for different PI controllers are given. It is assumed that the time constant of the roll positioning system is 7 [msec] and there is step change of 0.2 [mm] in input strip thickness.

Fig 5 shows the responses to the following feedback controllers between output thickness and roll position.

$$(a)\frac{s+50}{s},\ (b)\frac{s+100}{s},\ (c)\frac{s+500}{s},\ (d)\frac{0.1(s+1000)}{s}$$

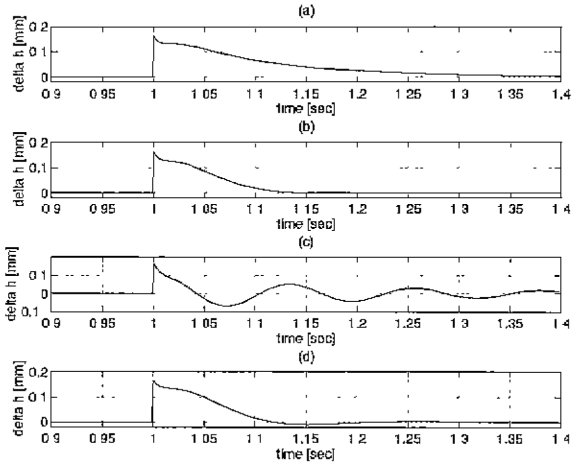

Fig. 5. Single loop feedback control with PI controller using rolling position alone

It can be seen from Fig 5 that the response time following the disturbance appears to be lower limited by about 50 [msec] This seems strange given that the actuator response time is 7 [msec].

We will use the response to a 0.2 [mm] input thickness disturbance as a quantitative measure of performance based on an integrated absolute error (IAE) criterion which is defined as follows

$$IAE=\int |h(t)-h^*|\,dt$$

where $h(t)$ is the output strip thickness and h^* is the nominal output thickness. IAE of four different PI controllers in Fig 5 are (a) 17.6×10^{-3}, (b) 8.9×10^{-3}, (a) 14.7×10^{-3}, (a) 10.0×10^{-3}. Various control strategies based on single loop controller lead to unsatisfactory results since all attempts to speed up the response result in a degradation in the transient response. An analysis supporting this claim is presented in the next section.

IV. INVESTIGATION OF IMAGINARY ZEROS

4.1 Preliminary Step Response Study

We have seen in Section III that a key feature of the linearized model linking the rollgap change to the exit thickness is (a pair of) blocking zeros on the $j\omega$-axis. It is conjectured that these zeros are the origin of the hold-up effect in the step response relating S to Δh .

To check if these zeros are really the cause of the "hold-up effect", the following system was simulated

$$\frac{K(s+j\theta)(s-j\theta)}{(s+a)(s+b)(s+c)}$$

where K was chosen to give unity d.c. gain.

The parameters a,b,c were chosen to give real poles. Four cases were studied, namely

(1) poles : -1,-2,-3, zeros at $\pm$ 0.1j

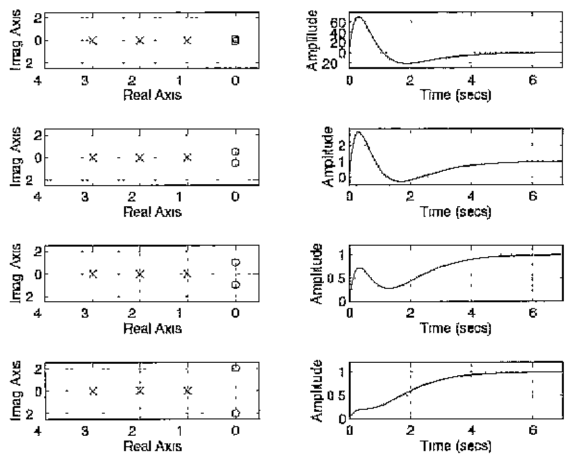

Fig 6. Effect of undamped zeros on transient response

(2) poles : -1,-2,-3, zeros at $\pm 0.5j$
(3) poles : -1,-2,-3, zeros at $\pm j$
(4) poles : -1,-2,-3, zeros at $\pm 2j$

Fig 6 shows the corresponding step responses.

We see from Fig 6 that the presence of $j\omega$-axis zeros leads to a hold-up effect. Moreover the faster the closed loop poles in relationship to the location of the zeros the more pronounced the effect becomes.

4.2. Fundamental Limitations

If one allows arbitrary pole positions, then the impact of $j\omega$-axis zeros reduces to the rather uninteresting observation that the frequency response must be zero at the zero location. The corresponding time domain constraints would therefore be uninteresting. However, for reasons of robustness it is usual to require that the poles lie in the strict left half plane (say to the left of $-\alpha$). Under these conditions, there is quantifiable fundamental limitation arising from $j\omega$-axis zeros. This is made clear in the following result.

THEOREM 1

Consider a system, G(s) having poles and zeros satisfying the following two constraints

(i) all poles have real part less than $-\alpha$ ($\alpha>0$)

(ii) there exists at least one pair of zeros on the $j\omega$-axis at $\pm j\omega_o$

Further, the steady state gain of the system is normalized to unity.

Under these conditions, the following integral constraints hold for the unit step response y(t) of the system:

$$\int_0^\infty \cos\omega_o t[1-y(t)]\,dt = 0$$
$$\int_0^\infty \sin\omega_o t[1-y(t)]\,dt = \frac{1}{\omega_o} \tag{10}$$

PROOF

The Laplace Transformation of the step response satisfies

$$Y(s) = G(s)\frac{1}{s}$$

in the region of convergence of the transform.

Now let $e(t)=1-y(t)$. Then

$$E(s) = \frac{[1-G(s)]}{s}$$

By assumption, $G(0)=1$, and hence $E(s)$ does not have a pole at $s=0$. Hence, in view of assumption (I), $s=\pm j\omega_o$ lies inside the region of convergence of the transformation E(s).

Thus, since

$$E(s) = \int_0^\infty e^{-st} e(t)\,dt = \frac{[1-G(s)]}{s}$$

and $G(\pm j\omega_o)$-0, then

$$\int_0^\infty e^{-j\omega_o t} e(t) = \frac{1}{j\omega_o}$$

Similarly

$$\int_0^\infty e^{j\omega_o t} e(t) = \frac{-1}{j\omega_o}$$

The result follows on using the fact that

$$\cos\omega_o t = \frac{1}{2}\left[e^{j\omega_o t} + e^{-j\omega_o t}\right]$$
$$\sin\omega_o t = \frac{1}{2j}\left[e^{j\omega_o t} - e^{-j\omega_o t}\right]$$

To get a feeling for the results in Theorem 1, let us consider the limiting case where ω_o is much smaller than the location of the poles. In this case, we can approximate $\cos\omega_o t$ and $\sin\omega_o t$ as follows over the settling time of the system

$$\cos\omega_o t \cong 1$$
$$\sin\omega_o t \cong \omega_o t$$

In this case, the integral constraints become approximately

$$\int_0^{t_s} (1-y(t))\,dt \cong 0$$
$$\int_0^{t_s} (1-y(t))t\,dt \cong \frac{1}{\omega_o^2} \tag{11}$$

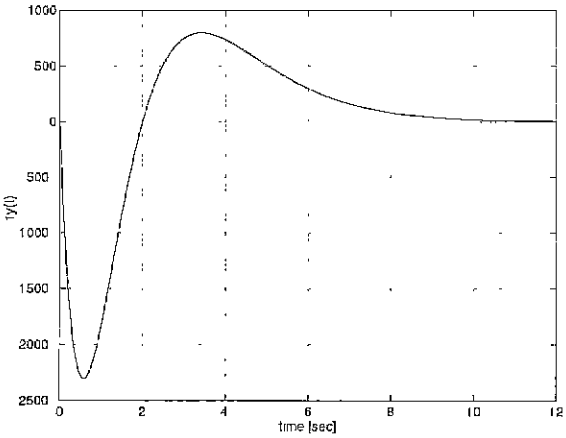

Fig. 7. Error response of $(10^4 s^2+1)/(s^3+3s^2+3s+1)$ to a step

where t_s is the settling time ($t_s << 1/\omega_o$).

We thus see that the mean value of the error response is approximately zero while the first moment about the origin is approximately $1/\omega_o^2$

To validate this analysis, a simulation was carried out of the following system

$$G(s)=\frac{10^4 s^2+1}{s^3+3s^2+3s+1}$$

Note that the poles are at (-1,-1,-1) whereas there are blocking zeros at $\pm j0.01$. The simulation response 1-y(t) to a unit step is shown in the Fig 7.

The response in Fig 7 has massive over and under shoots. However, they are easily explained by Theorem 1 and approximately by (11). One can easily evaluate

$$\int_0^{12}(1-y(t))dt=-2.1$$

$$\int_0^{12}(1-y(t))t\,dt=19932$$

Of course, exact correspondence would be obtained by using the precise constraints given in (10).

4.3. Implication for Rolling Mill Gauge Control

The fundamental constraint presented by Theorem 1 is applicable to the thickness control problem. However, it is unreasonable to expect that the closed loop poles would be placed in the far left half plane. The limiting results presented indicates that this would be an extremely undesirable thing to do. Instead, the actuator response of the rolling mill is almost 7 [msec] and it thus seems reasonable to aim for a settling time of 30 [msec].

The location of the blocking zero is at 86[rad sec-1], i.e. a period of approximately 73 [msec]. We refer to Fig. 8 for the corresponding graphs of $cos\omega_o t$ and $sin\omega_o t$. We see from these graphs that, in order that the correlation of (1-y(t)) with $sin\omega_o t$ be large but the correlation with $cos\omega_o t$ approximately zero, we must "hold-up" the error response so that there is enough weight in the part of the $cos\omega_o t$ waveform which is negative to give an overall result of zero for

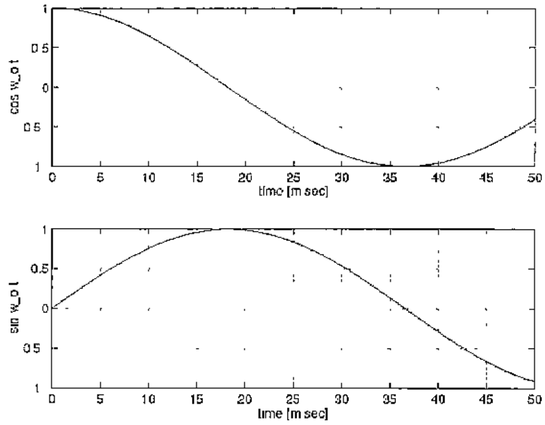

Fig. 8. $\cos\omega_o t$, $\sin\omega_o t$, ω_o=86,

$$\int_0^{t_s}(1-y(t))\cos\omega_o t\,dt$$

Thus $y(t)$ cannot settle to the steady state value 1 until significantly after 20 [msec] has elapsed. We thus have obtained a major insight into the control problem. We see that hold-up phenomenon is fundamental and is a direct consequence of the location of the $j\omega$-axis zeros, together with the location of the closed-loop poles of the system.

We have seen that the conventional SISO control system linking exit thickness to roll position is deleteriously affected by the hold-up phenomenon. Moreover, this effect is fundamental and unavoidable within this control architecture. We therefore proceed to investigate ways of circumventing the hold-up problem keeping in mind that this must be achieved by a fundamental change to the basic architecture of the controller.

V. HARD LOAD SHARING ALGORITHM APPLIED TO THICKNESS CONTROL

We have seen that the problem is directly related to the presence of $j\omega$-axis zeros in the rollgap to exit thickness transfer function. However, inspection of the linear model reveals that the $j\omega$-axis zeros are not zeros of the multivariable system. A preliminary investigation using the coiler and uncoiler motor currents to control exit thickness proved to be perfectly satisfactory save for the fact that these inputs are very limited in their authority due to hard amplitude constraints. Thus use of these currents, in isolation, is an unsatisfactory solution. However, it does seem feasible that the currents could be used in a coordinated way with the rollgap to effectively control exit thickness

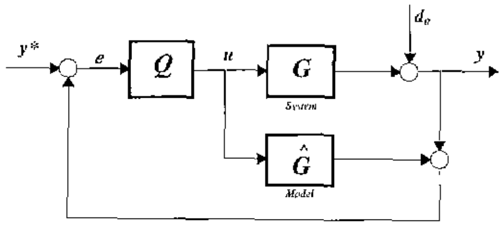

Fig. 9. Block diagram representation of all stabilizing controllers when the plant G is stable

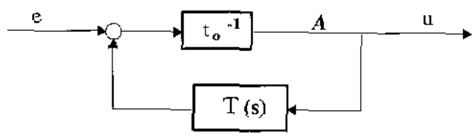

Fig. 10. Biproper controller implementation via feedback

The method we propose is developed in Shim et (1996) and can be briefly described as follows:

The class of all stabilizing controllers for a plant having all its open loop poles in a desirable region (e.g. at least stable which is the case of rolling mill) is shown in Fig 3 where Q is stable-proper. Assuming no modeling error then the output disturbance sensitivity, input disturbance sensitivity and complementary sensitivity are given respectively by

$$S=(I-GQ),\quad S_i=(I-GQ)G,\ T=GQ$$

The above expression suggest that Q should be a stable-proper approximation to the inverse of G. Furthermore, T shows that diagonal decoupling is achieved if Q is chosen so that GQ is diagonal. In the presence of saturating actuators, one could proceed by simply implementing Q in feedback form – see Goodwin et al (1993). We can write Q as

$$Q^{-1}=t_o+T(s)$$

Here we assume that Q is biproper but by prefiltering the reference signal and the feedback signal this assumption can be removed. so without loss of generality we can assume that Q is biproper. A feedback realization of Q is depicted in Fig 10.

When Q is implemented by feedback as in Fig 10 T(s) contains the dynamics of G(s). Therefore there are three copies of G(s). By saturating the input to each of these copies simultaneously, a match between the plant, the model and the model inverse can be achieved – Goodwin et al (1993). This methodology was applied to the rolling mill model and have been found to give satisfactory results. We thus conclude that the hold-up effect due to tension interactions can be satisfactorily dealt with via multivariable design. However, the decoupling controller uses three outputs (output strip thickness, input and output strip tension) as inputs to the controller. In practice it is difficult to measure or estimate tensions accurately With exit thickness as its only output, the rolling mill becomes a nonsquare system. Since nonsquare systems have their own unique properties, design methodologies for nonsquare systems must be approached differently from those for square systems.

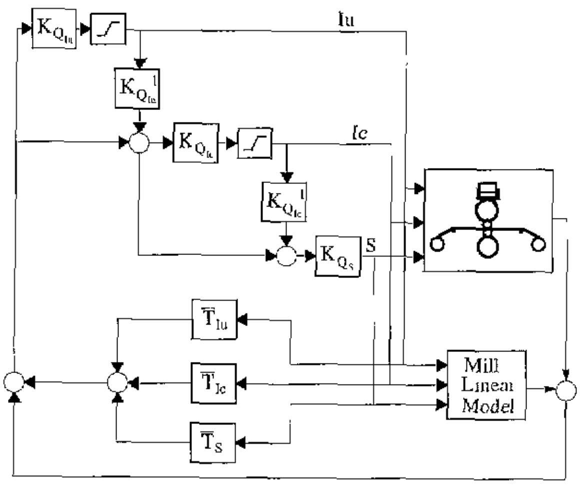

Fig. 11. Hard Load Sharing algrithm applied to feedback control

A key issue is how to exploit the different characteristics of the multiple inputs. Each input has different fundamental limitations. For example, the rollgap suffers from the hold-up effect whereas I_U and I_C have limited range. Thus the controller for the rolling mill should exploit the strength of each input while avoiding the weakness. This may only be achieved by using inputs in some coordinated fashion. To accomplish this goal Hard Load Sharing (HLS) algorithm developed by Shim et al (1996) is adopted. HLS algorithm makes full use of the available inputs for the nonsquare system having 3 inputs (roll position, coiler and uncoiler current) but only 1 output (exit thickness).

Our objective is to first use the coiler and uncoiler currents and then to only turn to the rollgap when the currents reach saturations.

We use the feedback version of HLS scheme, which is shown in Fig. 11 where

$$Q_{Iu}^{-1}=\frac{(s+Z_{Q_{Iu1}})(s+Z_{Q_{Iu2}})\prod_{l=1}^{5}(s+Z_{Iu\,l})}{K_{Q_{Iu}}\prod_{m=1}^{7}(s+P_{Iu\,m})}=\frac{1}{K_{Q_{Iu}}}+\overline{T}_{Iu}$$

$$Q_{Ic}^{-1}=\frac{1}{K_{Q_{Ic}}}+\overline{T}_{Ic}$$

$$Q_{S}^{-1}=\frac{1}{K_{Q_{S}}}+\overline{T}_{S}$$

Fig. 12 shows the simulation result when the HLS scheme is used in the feedback controller to cooperate

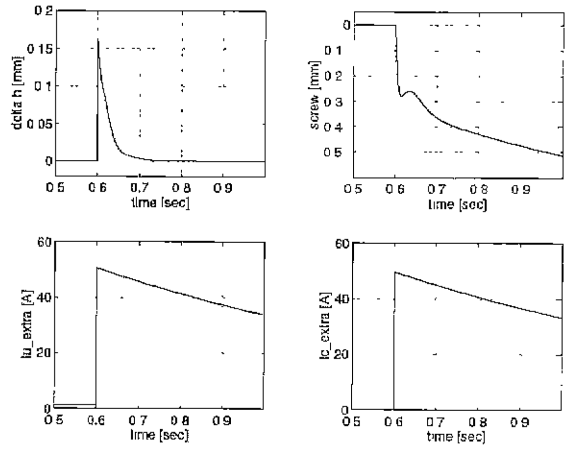

Fig. 12.

three inputs as well as returning I_U and I_C back to their nominal values following a disturbance.

The simulation result shows that a significantly better result is achieved with the proposed scheme than with the use of the rollgap change alone. When used 0.2 [mm] input thickness disturbance as a measure of performance IAE improved from 8.9×10^{-3} (with rollgap change only) to 4.0×10^{-3}. Thus we have achieved a 2:1 increase in performance by coordinated use of three available inputs.

VI. CONCLUSIONS

In this paper, accurate thickness control for a single-stand reversing cold rolling mill was studied. Hold-up effect which inhibits the achievement of rapid closed-loop response was identified and a general result on the impact of $j\omega$-axis zeros on fundamental limits in feedback control systems was developed. Having identified the cause of hold-up effect, a new multivariable design method, HLS algorithm, which makes full use of redundant inputs of non-square system, was applied. Simulation results demonstrate the usefulness of this approach in practice.

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