A REVIEW OF TORSIONAL VIBRATION CONTROL METHODS AND A PROPOSAL OF DISTURBANCE OBSERVER-BASED NEW TECHNIQUES

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Abstract: Vibration suppression and disturbance rejection in torsional system are important issue in the future motion control. In this paper, first, a brief review on various control strategies is given. Next, a new control technique based on the disturbance observer, the "resonance ratio control" is proposed. By realizing Manabe's model polynomial, the 2-inertia non-stiff system can be controlled effectively although the controller's order is only 2.

Keywords: vibration suppression, disturbance rejection, robust control, resonance control

1. INTRODUCTION

The problem on vibration suppression and disturbance rejection in flexible system originates in steel rolling mill system, where the load is coupled to the driving motor by a long shaft. The small elasticity of the shaft is magnified and has a vibrational effect on the load speed.

As the newly required speed response is very close to the first resonant frequency, only conventional P&I control are no longer effective. To overcome the problem, various control strategies have been proposed mainly for controlling the 2-inertia system, the simplest model (Hori, 1992; 1995a). Here, the history of control theory can been seen. In this paper, a brief review of them will be given.

Next, a new control technique based on the disturbance observer is proposed as a simple and practical strategy. It is the "resonance ratio control" based on the "fast disturbance observer". The resonance ratio is the ratio of the resonance and anti-resonance frequencies in 2-inertia system. By feeding back the torsional torque estimated by the disturbance observer, the virtual motor inertia moment can be changed to any arbitrarily value. Yuki (1993) suggested that vibration can be suppressed effectively by adjusting the resonance ratio to be about $\sqrt{5}$. Sugiura (1994) used the same ratio in speed control system. In this paper, it will be shown that $0.8\sqrt{5}$ is the optimal ratio when realizing "Manabe's model polynomial".

2. STEEL ROLLING MILL SYSTEM and its MATHEMATICAL MODEL

Our aim is to control the roll speed in the presence of (1) torsional vibration, (2) system parameter variation, (3) disturbance torque T_L, and (4) in the absence of a dedicated loadside speed sensor. P&I controller was designed to control one-inertia system, where the coupling shafts are assumed to have infinite stiffness. However, recent new requirements have forced us to develop newer techniques, i.e., (1) faster speed response, (2) rejection of disturbance on the loadside, and (3) robustness to parameter variations.

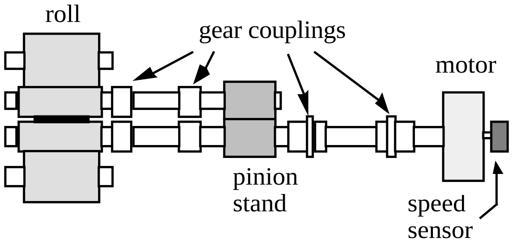


Fig.1 Typical configuration of steel rolling mill system.

Now the problem is not restricted anymore to the case of rolling mill (Harakawa, 1988). New areas such as flexible robotic joint, precise positioning system (Schäfer, et.al., 1991) and large scale space structure have the same problem. Parameter variation is a very much existent phenomenon in these areas.

Fig.1 illustrates the typical configuration of steel rolling mill system. As this system is the distributed parameter system, using the modal analysis it should be modeled as a multi-inertia system connected by springs. The 2-inertia model shown in Fig.2 is the simplest model considering up to the first mode. Fig.3 shows its block diagram. Numerous papers are found to deal with control of such system. In particular, 2-inertia system has been investigated as the benchmark problem in ACC'90 and '91 (Wie, et.al., 1990) In SICE, Japan, similar bench mark problems are presented and several techniques are evaluated (Hara, et.al., 1995).

The state equations take the form of

$$\begin{pmatrix} \dot{\omega}_M \\ \dot{\theta}_s \\ \dot{\omega}_L \end{pmatrix} = \begin{pmatrix} \frac{-B_M}{J_{M0}} & \frac{-K_s}{J_{M0}} & 0 \\ 1 & 0 & -1 \\ 0 & \frac{K_s}{J_L} & \frac{-B_L}{J_L} \end{pmatrix} \begin{pmatrix} \omega_M \\ \theta_s \\ \omega_L \end{pmatrix} + \begin{pmatrix} \frac{1}{J_{M0}} \\ 0 \\ 0 \end{pmatrix} T_M + \begin{pmatrix} 0 \\ 0 \\ \frac{1}{J_L} \end{pmatrix} T_L \quad (1)$$

State variables are ω_M, θ_s and ω_L. Control input is the motor torque T_M. Output variable which can be measured is the motor speed ω_M. Controlled variable is the load speed ω_L and disturbance T_L is injected into the load.

Neglecting friction terms, the transfer function from T_M to ω_M, which plays an important role in the closed loop design, is given by

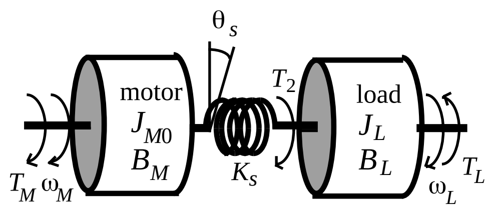


Fig.2 2-inertia system model.

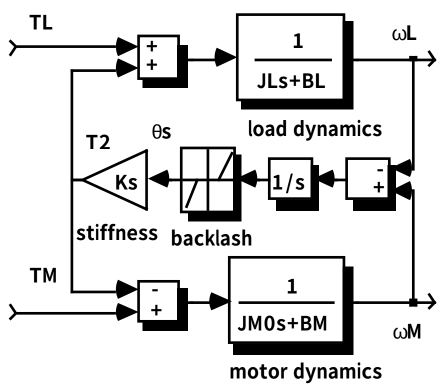


Fig.3 Block diagram of 2-inertia system.

$$\frac{\omega_M}{T_M} = \frac{1}{s} \frac{J_L s^2 + K_s}{J_{M0} J_L s^2 + K_s (J_{M0} + J_L)} = \frac{1}{J_{M0} s} \frac{s^2 + \omega_a^2}{s^2 + \omega_{r0}^2} \quad (2)$$

The Bode plot is drawn in Fig.4. The resonant and anti-resonant frequencies are given by

$$\omega_{r0} = \sqrt{\frac{K_s}{J_L}\left(1 + \frac{J_L}{J_{M0}}\right)} \quad (3)$$

and

$$\omega_a = \sqrt{\frac{K_s}{J_L}} \quad (4)$$

At these frequencies, the phase characteristics change drastically. The resonance ratio is defined by

$$H_0 = \frac{\omega_{r0}}{\omega_a} = \sqrt{1 + \frac{J_L}{J_{M0}}} = \sqrt{1 + R_0} \quad (5)$$

where R_0 is called the inertia ratio given by $R_0 = J_L / J_{M0}$.

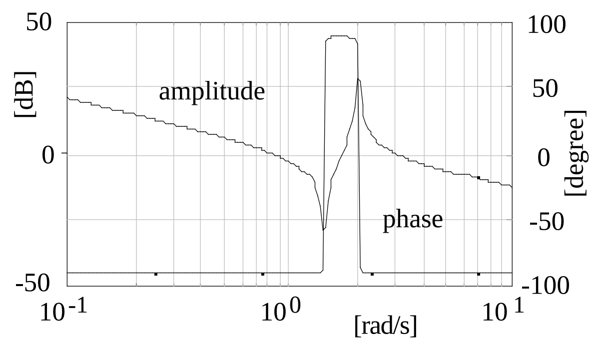


Fig.4 Bode plot of ω_M / T_M when R_0=1.

3. VARIOUS CONTROL STRATEGIES

3.1 Time Derivative Feedback

The starting point of vibration suppression is the speed derivative (i.e., acceleration) feedback as is shown in Fig.5 (Sugano, 1990). Vibration can be suppressed fairly well if the derivative gain K_d is adjusted appropriately.

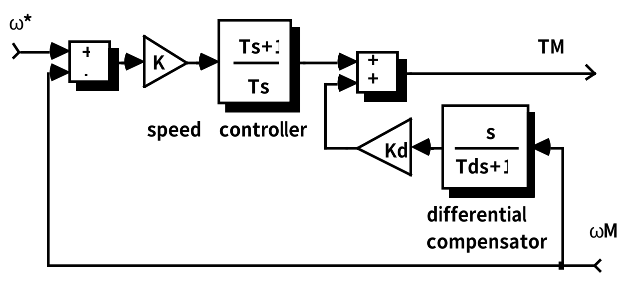


Fig.5 Speed derivative compensation.

3.2 *Model Following Controls*

The control techniques in this category aim to realize similar response to that of the damped reference model by feeding back the output difference from the actual plant.

Simulator Following Control (SFC)

SFC (Hasegawa, 1986) shown in Fig.6 is an excellent and practical method. It is widely used in actual industrial drive systems like rolling mill and elevator system. The advantage of SFC is that it's function is "optional" and it can be easily adjusted on site.

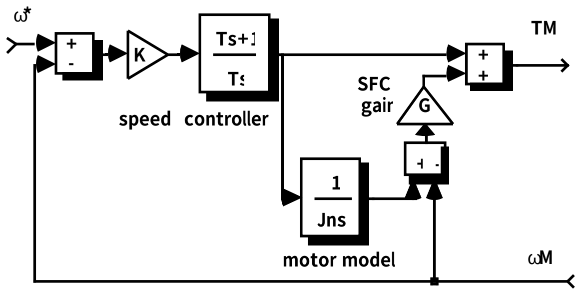


Fig.6 Simulator following control system.

Model Following TDOF Control

Fig.7 shows the model following Two-Degree-Of-Freedom control system proposed by Koyama, et.al. (1991). Speed controller 1 is the low gain controller. The main controller is controller 2 to control the model. The difference between the real plant output and the model output is fed back to the control input of the model.

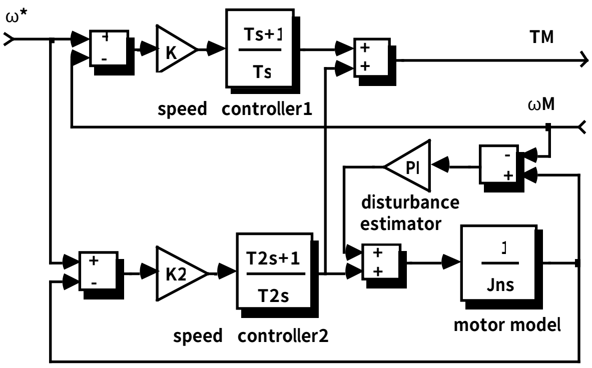


Fig.7 Model following TDOF control system.

3.3 *Application of Disturbance Observer*

Disturbance observer was originally proposed for disturbance rejection and robustification to parameter variation in single-inertia system (Ohishi, 1987). It is well known that, if this technique is applied to the 2-inertia system as it is, free vibration due to the load inertia and spring is induced. However, the disturbance observer has three parameters, i.e., the compensation gain, the cutoff frequency, and the nominal inertia moment used in the observer. By appropriately selecting these parameters, it is effective for vibration suppression.

Resonance Ratio Control

The resonance ratio control is based on the fast disturbance observer. Only the compensation gain is adjusted. This technique will be investigated in detail in the next chapter.

Slow Disturbance Observer

Fig.8 shows the slow disturbance observer proposed by Umida (1994). By putting k_1=1 and k_2=0 for simplicity, eq.(6) is obtained

$$\omega_M = \frac{1}{J_{M0}s}\left(\frac{s+\omega_q}{s+(J_L/J_{M0})\omega_q}T_M^* - \frac{s}{s+(J_L/J_{M0})\omega_q}T_2\right) \quad (6)$$

where $\omega_q = 1/T_q$. It is seen that the phase lead compensation from torque command and the derivative feedback from the torsional torque are applied. Iwata, et.al. (1994) showed that the optimal observer's cutoff is given by

$$\omega_q = \sqrt{\frac{2}{R_0^2+1}}\,\omega_a \quad (7)$$

which is a little slower than the anti-resonant frequency.

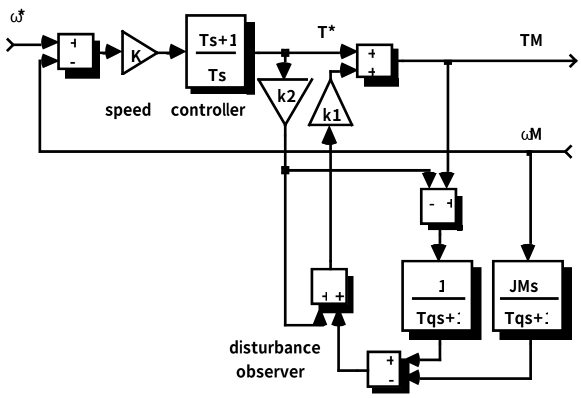


Fig.8 Slow disturbance observer application.

3.4 *State Feedback Control*

Various state feedback techniques have been proposed. The system poles can be located to any desired positions. As only the motor torque and speed are measurable, other state variables should be estimated by observer.

SFLAC (State Feedback and Load Acceleration Control) is proposed by Hori (1994a) as a typical application of state feedback. Fig.9 illustrates its configuration. It seems relatively complex, but its design concept is straightforward and clear. Dhaouadi (1992) proposed more sophisticated technique using two observers. It has superior control performance though the controller's order is much higher.

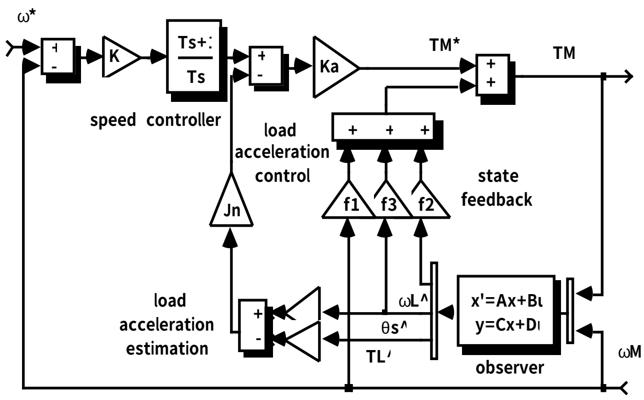


Fig.9 SFLAC.

4. RESONANCE RATIO CONTROL

4.1 System Model

In this chapter, "resonance ratio control" is proposed and investigated. First, to make discussion general,

$$J_{M0} + J_L = 1\ ,\quad K_s = 1 \qquad (8), (9)$$

is assumed in the two-inertia model. It means that the total inertia moment of the motor and load, and the spring coefficient are fixed. Various 2-inertia systems with different inertia ratios will be considered under this assumption.

4.2 Resonance Ratio Control

Fig.10 depicts the resonance ratio control using the fast disturbance observer. In conventional disturbance observer applications, 100% of the estimated disturbance is fed back. In this case, 1-K of the estimated disturbance is used (Hori, 1994b). Fig.11 shows the new system, when the disturbance observer's cutoff frequency is high enough. The virtual motor inertia moment is changed as

$$J_M = J_{M0}/K \qquad (10)$$

This means that the resonant frequency can be changed as

$$\omega_r = \sqrt{K_s\left(\frac{1}{J_M} + \frac{1}{J_L}\right)} \qquad (11)$$

and the resonance ratio as

$$H = \sqrt{1+R} = \sqrt{1+R_0K} \qquad (12)$$

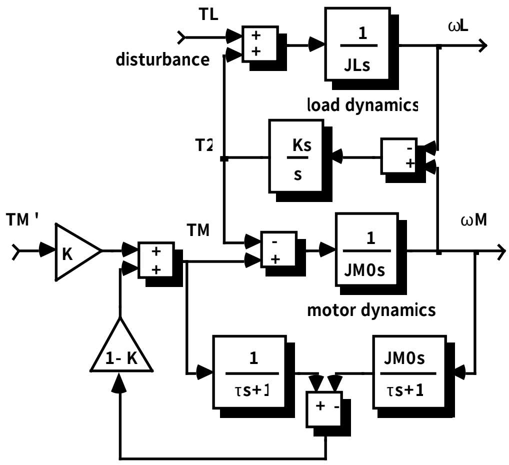


Fig.10 Resonance ratio control.

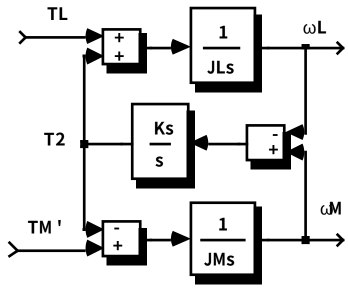


Fig.11 The effect of the resonance ratio control.

4.3 Normalization

Fig.12 is the block diagram from the new input torque T_M' to the motor speed ω_M. This is normalized as is shown in Fig.13 putting ω_a=1 and J_L=1. Here q is the parameter representing the resonance ratio given by

$$q = 1/H^2 = 1/(1+R) < 1 \qquad (13)$$

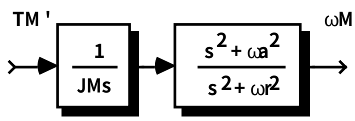


Fig.12 Block diagram from T_M' to ω_M.

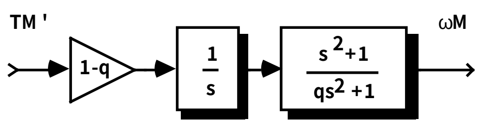


Fig.13 Normalized system by putting ω_a=1 and J_L=1.

4.4 Controller Design using Manabe Polynomials

As the speed controller $C(s)$, P, P&I and PID controllers are designed considering the closed loop characteristics.

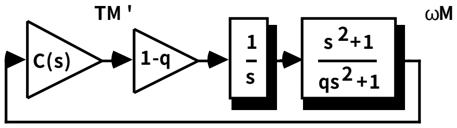


Fig.14 Design of the speed controller $C(s)$.

When $C(s)=K_p+K_I/s+K_Ds$ (PID controller) is applied, the characteristic equation of the closed loop system is given by

$$
\begin{aligned}
P(s) &= s^2(1+qs^2) + (K_D s^2 + K_p s + K_I)(1-q)(1+s^2) \\
&= \{q + K_D(1-q)\}s^4 + K_p(1-q)s^3 \\
&\quad + \{1 + K_D(1-q) + K_I(1-q)\}s^2 + K_p(1-q)s + K_I(1-q) \\
&= a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0
\end{aligned} \tag{14}
$$

Manabe's standard form (Manabe, 1991; also refer to Kessler, 1960) suggests that

$$\gamma_1 = \frac{a_1^2}{a_0 a_2} = \frac{\{K_p(1-q)\}^2}{\{1+K_D(1-q)+K_I(1-q)\}K_I(1-q)} = 2.5 \tag{15}$$

$$\gamma_2 = \frac{a_2^2}{a_1 a_3} = \frac{\{1+K_D(1-q)+K_I(1-q)\}^2}{\{K_p(1-q)\}^2} = 2 \tag{16}$$

$$\gamma_3 = \frac{a_3^2}{a_2 a_4} = \frac{\{K_p(1-q)\}^2}{\{q+K_D(1-q)\}\{1+K_D(1-q)+K_I(1-q)\}} = 2 \tag{17}$$

By solving these equations, the controller constants are given as follows by using q as the parameter. (The actual controller gains can be easily given after de-normalization by ω_a and J_L.)

$$K_p = \frac{10\sqrt{2}}{11},\ K_I = \frac{4}{11},\ K_D = \frac{5-16q}{11(1-q)} \qquad (18),(19),(20)$$

K_p and K_I are completely same to those of only P&I controller case. The differentiation gain K_D is the function of q. This means that, whatever resonance ratio $H=1/q^2$ is chosen, the criterion of Manabe Polynomial ($\gamma_1=2.5$, $\gamma_2=\gamma_3=...=2$) can be satisfied by corresponding choice of K_D. When $q>5/16$, i.e., $H<0.8\sqrt{5}$ and $R<2.2=11/5$, K_D should be negative. It is often reported that the positive feedback of acceleration is effective for vibration suppression. This corresponds to the case where the original inertia ratio R_0 is smaller than 2.2.

There are no need to use all components of PID controller. No usage of K_D has the advantage from the viewpoint of controller simplicity. This case gives the optimal resonance ratio as given by eq.(21). The order of the finally obtained controller is only 2.

$$q = 5/16,\ \ H = 0.8\sqrt{5},\ \ R = J_M/J_L = 5/11 \tag{21}$$

4.5 Simulation Results

Fig.15 shows the simulation results of two different cases, where the original inertia ratios are 1 and 0.2. TDOF P&I controller is used to suppress the overshoot in command response. Simulation is done under the condition with 10~20% model errors, backlash (+/-0.01[rad]) and torque limit (+/-1.2[Nm]).

Excellent performances can be seen both in vibration suppression and disturbance rejection. It is confirmed that the resonance ratio control is quite effective to wide range of inertia ratio. Also, the motor torque is negative just after the disturbance is added at t=25. This means that the disturbance rejection and vibration suppression are not consistent requirements for 2-inertia system with the original inertia ratio (R_0) smaller than 11/5=2.2.

5. CONCLUSION

A review was given on some techniques to suppress vibration and to reject disturbance in torsional system. Next, the "resonance ratio control" based on the "fast disturbance observer" was proposed and its excellent performance was demonstrated by simulation.

A slight performance degradation is seen in Fig.15(b) where the original inertia ratio is extremely small. Fig.16 shows the performance degradation of resonance ratio control when the disturbance estimation becomes slower. How to decide the observer's cutoff frequency is the next problem.

Recently, H∞ control, μ-synthesis and even LMI are applied to this problem. However, following practical requirements should be taken into account, e.g., (1) Design concept is clear, (2) Controller is easily adjustable on site, (3) Controller can be easily implemented, and (4) Controlled system is robust to backlash and torque limit (Hori, 1995). In these aspects, a relatively classical method as proposed in this paper is also attractive.

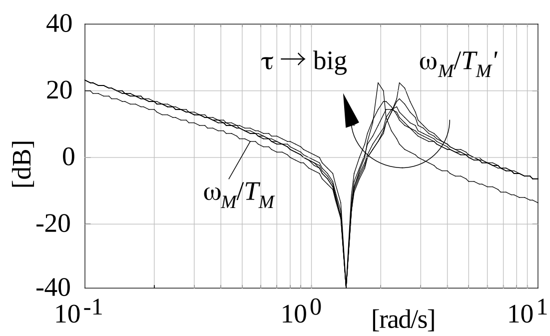
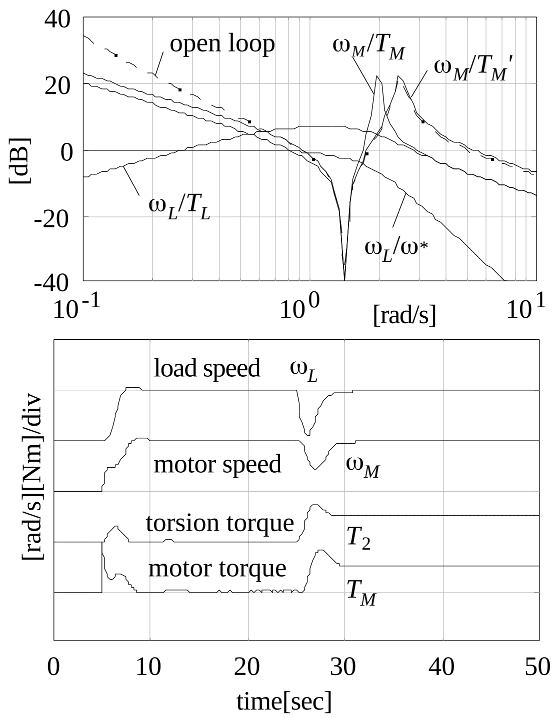
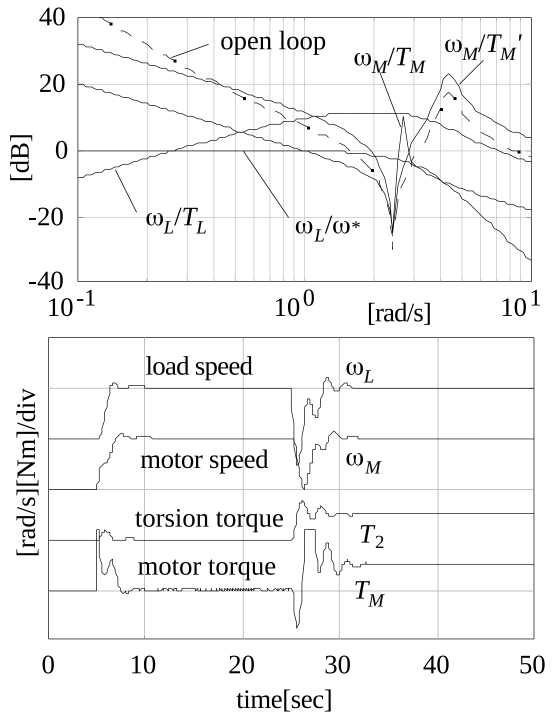


Fig.16 Performance degradation of resonance ratio control due to the slower disturbance estimation when R_0=1.



(a) $R_0 = J_L/J_{M0} = 1$



(b) $R_0 = J_L/J_{M0} = 0.2$

Fig.15 Simulation results.

(In the time response simulation, at t =5, ω^*=1(step) is commanded and at t =25, T_L=-0.5 (step) is added.)

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