EXPERIMENTAL EVALUATION OF DISTURBANCE OBSERVER-BASED VIBRATION SUPPRESSION AND DISTURBANCE REJECTION CONTROL IN TORSIONAL SYSTEM

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Abstract. In the resonance ratio control, the estimation speed of the disturbance observer should have been much faster than the resonance frequency of the controlled system. However, too fast disturbance observer causes implementation problem. In this paper, we give the optimal speed of the disturbance estimation and propose a novel technique, slow resonance ratio control, which enables us to design the speed control and the vibration suppression control independently.

Keywords. vibration suppression, disturbance rejection, torsional system, robust control, disturbance observer.

Vibration suppression and disturbance rejection control in torsional system is an important problem in the future motion control. As the newly required speed response is very close to the primary resonant frequency of such systems, conventional techniques based on P&I controller is not effective enough. To overcome the problems, various control strategies have been proposed mainly for controlling 2-inertia system, the simplest model of the flexible system.[1][2][3][7][12][13][14]

In this paper, we will focus our discussion on the disturbance observer-based techniques. We proposed "resonance ratio control" several years ago and showed its excellent performance by simulation.[4][9] By feeding back the torsional torque estimated by the disturbance observer, the virtual motor inertia moment can be changed to an arbitrarily value. This means that we can change the resonance frequency and then the resonance ratio.

However, the estimation speed of the disturbance observer used in the resonance ratio control was assumed fast enough compared with the resonant frequency of the controlled object.[9] Too fast disturbance observer causes implementation problem. We then investigated the effect of estimation speed on various control performances.

In this paper, we propose a novel technique, "slow resonance ratio control", whose advantages are as follows:

(1) The optimal speed of the disturbance estimation is given by a explicit formula. It can be relatively slow.

(2) The speed controller can be independently designed from the vibration suppression control.

Finally, we explain the specially ordered experimental setup using two motors and adjustable flywheels connected by a flexible shaft. We can adjust not only the inertia moments and the stiffness of the shaft but also the backlash and friction. We confirm the effectiveness of the proposed method by experiments.

STEEL ROLLING MILL AND 2-INERTIA MODEL

Figure 1 illustrates the typical configuration of steel rolling mill system. This system is basically a distributed parameter system. By using the modal analysis, it can be modeled as a system having several inertia moments and springs.[12] For example, 12 inertia moments are needed. 2-inertia system given by Figure 2 is its simplest model. Figure 3 gives its block diagram representation.

In Figure 2, we assume

$$J_{M0} + J_L = 1\ ,\quad K_s = 1 \qquad (1), (2)$$

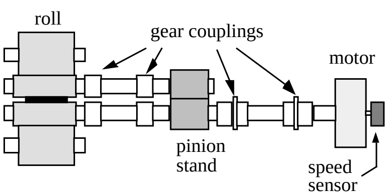


Figure 1: Typical configuration of steel rolling mill system.

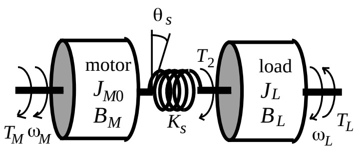


Figure 2: 2-inertia system model.

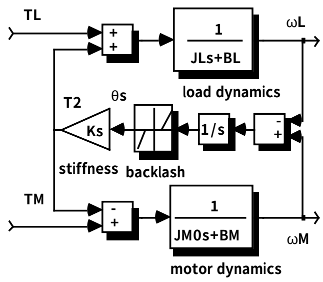


Figure 3: Block diagram of 2-inertia system.

for comparative analysis. These equations mean that the total inertia moment of the motor and the load, and the spring coefficient are fixed to 1, respectively. Various 2-inertia systems with different inertia ratios will be investigated under these relations.

The transfer function from T_M to ω_M, which is most important in the closed loop design, is given by

$$G_{11}(s) \approx \frac{1}{s}\,\frac{J_L s^2 + K_s}{J_{M0} J_L s^2 + K_s (J_{M0} + J_L)} = \frac{1}{J_{M0} s}\,\frac{s^2 + \omega_a^2}{s^2 + \omega_{r0}^2} \qquad (3)$$

This transfer function has two particular points: the resonant and anti-resonant frequencies given by

$$\omega_{r0} = \sqrt{\frac{K_s}{J_L}\left(1 + \frac{J_L}{J_{M0}}\right)} \qquad (4)$$

$$\omega_a = \sqrt{\frac{K_s}{J_L}} \qquad (5)$$

where R_0 is the inertia ratio given by $R_0 = J_L/J_{M0}$. At these frequencies, the phase characteristics change drastically.

SLOW RESONANCE RATIO CONTROL AND ITS DESIGN METHOD

Figure 4 depicts our proposing new technique: the slow resonance ratio control. Using this configuration, I will explain our idea.

Ideal fast resonance ratio control [9][11]

When T_q=0 in Figure 4, it gives the ideal "resonance ratio control" based on the fast disturbance observer. In usual disturbance observer applications, 100% of the estimated disturbance is fed back to the motor torque, but in this case, 1-K of the estimated disturbance is fed back. We can change the virtual motor inertia moment to any value as

$$J_M = J_{M0}/K \qquad (6)$$

The inertia ratio can be changed to

$$R = \frac{J_L}{J_M} = \frac{J_L}{J_{M0}/K} = KR_0 \qquad (7)$$

The resonant frequency is then changed to

$$\omega_r = \sqrt{\frac{K_s}{J_L}\left(1+\frac{J_L}{J_M}\right)} \qquad (8)$$

The anti-resonant frequency does not change. In the resonance ratio control, by setting the new resonance ratio $H=\omega_r/\omega_a$ to be $2\sim\sqrt{5}$, effective vibration control is achieved.[4][9][11] However, the estimation speed of the disturbance observer is finite in actuality. From some simulations, it is known that the estimation should be done faster than the resonant frequency of the controlled object.

Slow resonance ratio control

When the estimation speed of the observer is finite, i.e., $T_q>0$, the ideal resonance ratio control is impossible. Generally by using Q as the low-pass filter part: $1/(T_q s+1)$ in Figure 4, the following two important transfer functions can be obtained.

$$\frac{\omega_M}{T_M'} = \frac{1+R}{s}\,\frac{s^2+\omega_a^2}{s^2+\{1+(1-Q)R_0+QR\}\omega_a^2} \qquad (9)$$

$$\frac{\omega_L}{T_M'} = \frac{1+R}{s}\,\frac{\omega_a^2}{s^2+\{1+(1-Q)R_0+QR\}\omega_a^2} \qquad (10)$$

We pay more attention to ω_L/T_M'. Its characteristics are as follows.

- When the observer is very fast, i.e., $T_q=0$ if $Q=1/(T_q s+1)$, by putting $Q\rightarrow1$, eq.(11) is obtained.

$$\frac{\omega_L}{T_M'} = \frac{1+R}{s}\,\frac{\omega_a^2}{s^2+(1+R)\omega_a^2} = \frac{1}{s}\,\frac{\omega_r^2}{s^2+\omega_r^2} \qquad (11)$$

- When the observer is very slow, i.e., $T_q=\infty$ if $Q=1/(T_q s+1)$, by putting $Q\rightarrow0$, eq.(12) is obtained.

$$\frac{\omega_L}{T_M'} = \frac{1+R}{s}\,\frac{\omega_a^2}{s^2+(1+R_0)\omega_a^2} = \frac{1}{s}\,\frac{\omega_r^2}{s^2+\omega_{r0}^2} \qquad (12)$$

These two curves have the intersection point at

$$\omega_0 = \sqrt{1+\frac{R+R_0}{2}}\;\omega_a \qquad (13)$$

and the amplitude there is given by

$$\frac{\omega_L}{T_M'} = \frac{1+R}{\omega_0}\,\frac{2}{R-R_0} \qquad (14)$$

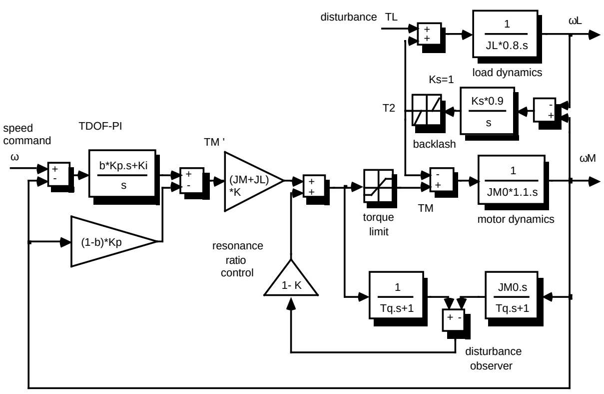


Figure 4: Configuration of the slow resonance ratio control.

Interestingly, all curves having any T_q pass this point. Hence, if T_q is selected so that this point is the local maximum, vibration suppression can be realized most effectively. Such T_q is given by

$$T_q = \sqrt{\frac{1+\frac{R+3R_0}{4}}{\left(1+\frac{3R+R_0}{4}\right)\left(1+\frac{R+R_0}{2}\right)}}\;\frac{1}{\omega_a} \qquad (15)$$

This is the optimal estimation speed (the optimal time constant) of the disturbance observer when we use the first order observer.

For reference, the optimal estimation speed given by Iwata in Umida's slow disturbance observer is given by [5][6]

$$T_q = \sqrt{1+\frac{R_0}{2}}\;\frac{1}{\omega_a} \qquad (16)$$

This value is close to the value to put R=0 in eq.(15). In other word, Umida's slow disturbance observer is the special case to put R=0 in the slow resonance ratio control.

Design of *K*

K is the ratio of R (the new inertia ratio that the resonance ratio control aims to realize) to R_0 (the original inertia ratio), i.e., $R=KR_0$. ω_a is also the function of R_0. When R_0 is given as a parameter of the original system, from eqs.(14) and (15), the optimal estimation speed T_q and the peak amplitude at ω_0 are the functions of only K.

Figure 5 draws the peak amplitude at ω_0 as the function of K. The peak at ω_0 decreases when K increases. On the other hand, from Figure 6, we can see that $\omega_q(=1/T_q)$ becomes bigger when K increases, which means that faster estimation is required. It must cause implementation problem. Hence we need a compromise.

From Figures 5 and 6, if we select $K=5\sim10$, the peak is relatively small while keeping ω_q not so big for a wide range of R_0. For smaller K, ω_q becomes much smaller, which reduces implementation problem, because the controller has no fast parts.

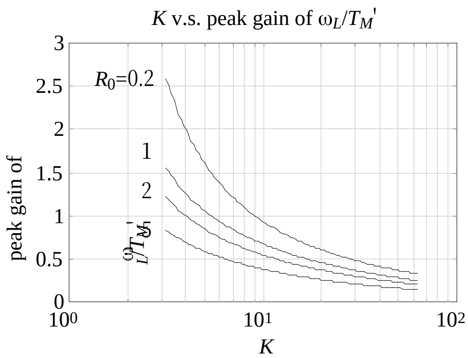


Figure 5: The peak amplitude at ω_0 v.s. K.
(When K increases, the peak at ω_0 decreases.)

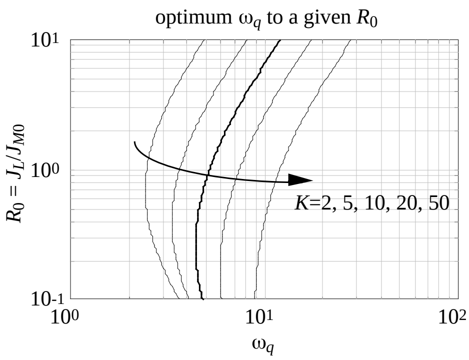


Figure 6: The optimal estimation speed of the disturbance observer.
(For bigger K, faster estimation is needed. $T_q = 1/\omega_q$.)

Design of the speed controller

In eqs.(9) and (10), ω_L/T_M' converges to $1/s$ when $s \rightarrow 0$, because we designed so as to keep their DC gains same regardless of R_0. It means that, in any R_0 cases, the 2-inertia system can be seen 1-inertia system having $J_{M0}+J_L=1$ as the total inertia moment.

It is very convenient if we can use the fixed P&I speed controller designed for the 1-inertia system. Here, we put

$$K_p = \frac{1}{T_\omega} \ , \quad K_i = \frac{K_p}{2.5 T_\omega} \qquad (17), (18)$$

T_ω is the specified response time of speed control. Here, we put $T_\omega=1/\omega_a$ hoping to realize the command response as fast as the anti-resonant frequency. Eq.(18) means that we selected the integral time constant to be 2.5 times of the speed control response. In simulation, Two-Degree-Of-Freedom P&I controller is used to reduce the overshoot in command response. It can be realized simply by putting b=0.5 in Figure 4.

Summary of the design procedure

The following is the summary of the design procedure of the "slow resonance ratio control" proposed in this paper.

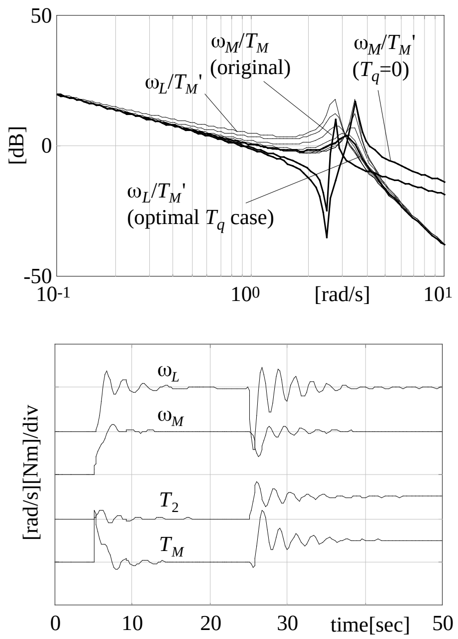
(1) Put K=5～10, i.e., $R=5R_0$～$10R_0$.
(2) Put the disturbance observer's estimation speed by eq.(15).
(3) Design the speed controller by eqs.(17) and (18).

SIMULATION RESULTS OF THE SLOW RESONANCE RATIO CONTROL

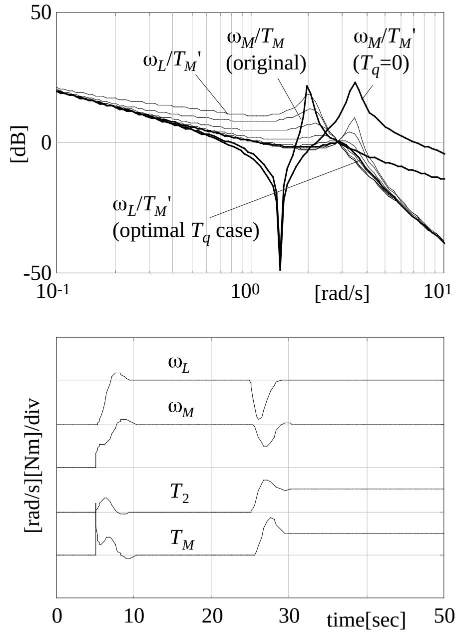
Here we will show the time response simulation of the "slow resonance ratio control" At t=5, the speed command ω^*=1 is given to observe its command response. At t=25, step disturbance of T_L=-0.5 is given to see the disturbance response.

In this simulation, the model constants include 10~20% errors, and backlash (+/-0.01) and torque limiter (+/-1.2) are introduced. We can know that the performances of the proposed method are same or even superior to other methods, e.g., the resonance ratio control, the optimal PID control, and even the state feedback control.[10][11][15]

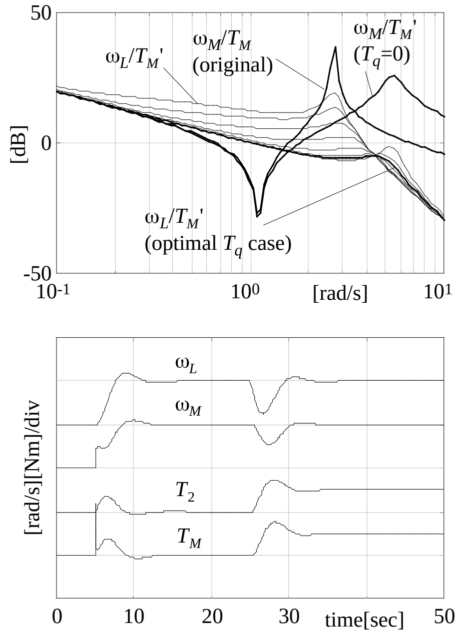
The speed controller here is designed for 1-inertia system without considering vibration suppression. Such an independent design has a great advantage in actual industrial application systems.



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



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



(c) $R_0=J_L/J_{M0}$=5

Figure 7: Simulation results when K=5.

EXPERIMENTAL RESULTS OF DISTURBANCE OBSERVER BASED CONTROLLERS

Experimental setup

Figure 8 illustrates the "Torsional Vibration System Experimental Setup" specially made by Mitsubishi Heavy Industry. It consists of two brushless DC (BLDC) motors, changeable backlash and friction mechanism, the load equipment and so on. The torques of BLDC motors are controlled fast and precisely enough by two high performance motor drivers.

Sensor information from shaft encoders and tacho-generators are read into the microcomputer via counter boards and A/D converters. After some control calculations, the torque commands are outputted to the drivers via D/A converters. Control algorithm is written by C language. We developed some more convenient programs, e.g., a frequency response measurement algorithm using M-series signal.

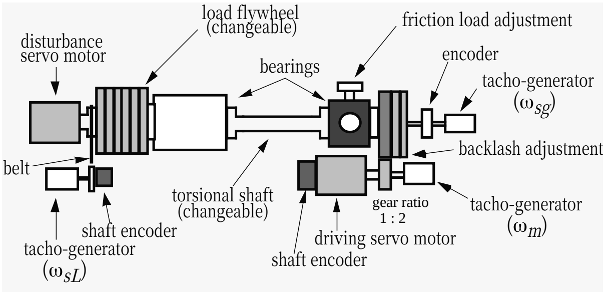


Figure 8: Experimental setup of torsional vibration system.

Table 1 gives the experimented control methods and their parameters. The inertia moments are converted to the load-side (i.e. the torsional shaft side) quantities. We implemented

(1) original disturbance observer designed for 1-inertia system,

(2) fast resonance ratio control using fast disturbance observer, and

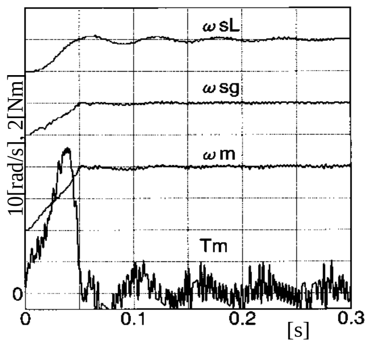
(3) slow resonance ratio control proposed in this paper.

The speed controllers of (1) and (3) are designed for 1-inertia system, and in (2) we used Manabe Polynomial method.[8] In all experiments, IP speed controllers are used by putting b=0 in Figure 4.

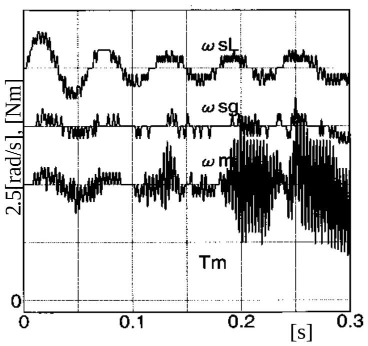
TABLE 1- Tested control methods and the parameters

parameters \ controller	disturbance observer for 1 axis	fast resonance ratio control	slow resonance ratio control
system parameters			
inertia moment of motor	J_{M0}=4.016 × 10^{-3}[kgm^2]		
inertia moment of load	J_L=2.921 × 10^{-3}[kgm^2]		
stiffness constant	K_s=39.21 [Nm/rad]		
resonant frequency anti-resonant frequency	ω_{r0}=152.3[rad/s], ω_a=115.9[rad/s]		
inertia ratio resonance ratio	R_0=J_L/J_{M0}=0.7273, H_0=1.314		
control period	T_s =1[ms]		
parameters in speed control			
K_p (proportional gain)	0.804	0.435	0.535
K_i (integral gain)	26.6	14.26	17.71
parameters in vibration control			
K (= R/R_0)	—	3.025	2.368
ω_q (= 1/T_q) *	2.0ω_a	3.0ω_a	1.7ω_a

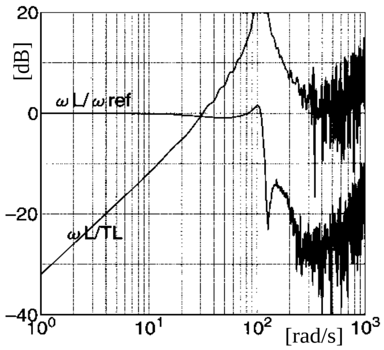
* T_q: time constant of the disturbance observer



(a) command response

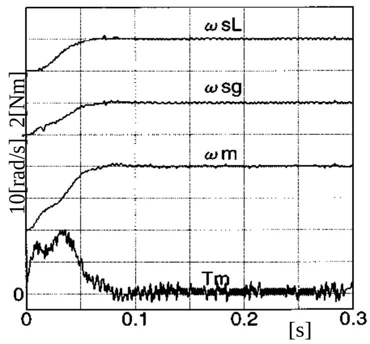


(b) disturbance response

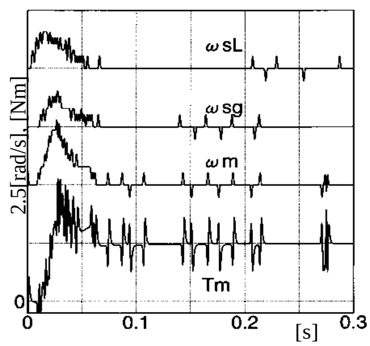


(c) frequency response from ω_{ref} and T_L to ω_L

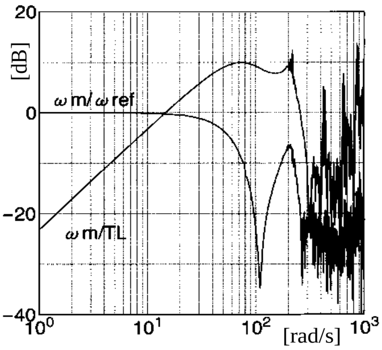
Figure 9: Original disturbance observer.



(a) command response

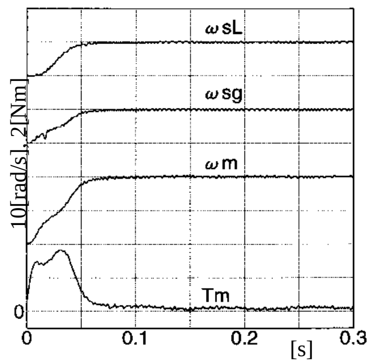


(b) disturbance response

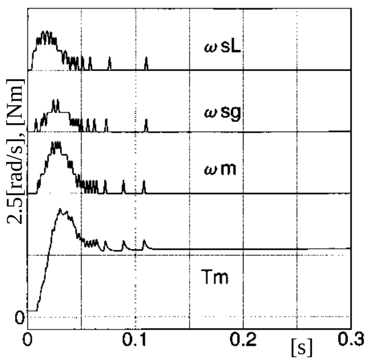


(c) frequency response from ω_{ref} and T_L to ω_L

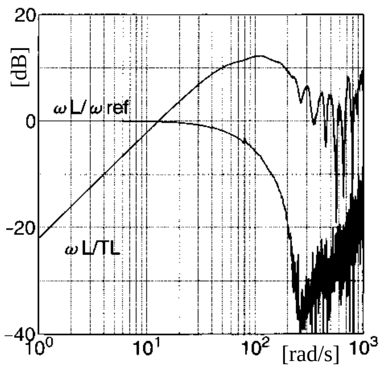
Figure 10: Fast resonance ratio control.



(a) command response



(b) disturbance response



(c) frequency response from ω_{ref} and T_L to ω_L

Figure 11: Slow resonance ratio control.

In the experiments of the command response shown in Figure 9(a)～Figure 11(a), the speed reference ω_{ref}=10[rad/s] is given at t=0. In experiments of the disturbance response shown in Figures (b), at t=0 the disturbance torque of 2[Nm] is added from the loadside motor.

In the experiments here, ω_{sL} is the load speed, ω_{sg} is the gear speed (We used the gear of 1/2 gear ratio.), ω_m is the motor speed and T_m is the motor torque. We tried to set the gear backlash to be 0, but there still remains little backlash. Also, note that the motor torque is limited by +/-3.84[Nm]. The torque commands are drawn in the figures.

Experimental results and discussion

The original disturbance observer in Figure 9 designed for 1-inertia system just suppresses the disturbance injected into the motor axis, which is the torsional torque in this case. As the result, big vibration was induced in the load speed and it considerably affected to the motor speed, too.

In the fast resonance ratio control shown in Figure 10, we implemented the disturbance observer as fast as we can. However it is not fast enough to realize the ideal resonance ratio control. Low frequency vibration can be suppressed effectively. However, the transfer function from T_L to ω_L has a harmful frequency peak around 200[rad/s]. Due to this peak, relatively big high frequency vibration remains in the motor torque.

The slow resonance ratio control shown in Figure 11 gives sufficiently stable vibration-less responses in frequency characteristics and motor torque waveforms. This is because there are no fast parts in the proposed controller.

CONCLUSION

In this paper, we proposed the "slow resonance ratio control" as an effective torsional system control method. We gave the explicit formula of the optimal estimation speed of the disturbance observer. We confirmed its superior performances by simulation and real experiment. In the original "fast resonance ratio control", we needed to design the speed controller considering the vibration suppression, where we dealt with the total system's transfer function or the state space representation. On the contrary, in the proposed slow resonance ratio control, we can use the speed controller independently designed for 1-inertia system. This is a great advantage in most industrial applications.

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