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Trajectory Control of Direct Drive Robot using Two-Degrees-of-Freedom Compensator

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Abstract

In this paper, we propose a new design approach of a two-degrees-of-freedom compensator which assures the robust stability. First of all, we clarify the internal structure of the generalized two-degrees-of-freedom compensator. By adopting this structure, we can make a bridge between the generalized controller and the disturbance observer based controller. Secondly, based on the clarified structure we derive a robust stability condition, and propose a design algorithm of free parameter taking the condition into account. The proposed design algorithm is easy to implement and, as a result, we obtain lower order free parameter then that of the conventional design algorithm. Thirdly, we show by adopting an appropriate coprime factorization that the clarified structure can also be regarded as an extended version of the conventional PID compensator. Finally, we apply the proposed algorithm to a three-degrees-of freedom direct drive robot, and show some experimental results to verify the effectiveness of the proposed algorithm.

1 Introduction

To achieve high performance control of a robotic manipulator, we have to take into account disturbances added on each joint and nonlinear forces such as a frictional force, a gravity, and an interacting force. To meet these ends, lots of control strategies were proposed in the past such as the computed torque technique[1], the nonlinear decoupled feedback control[2], and the resolved-acceleration control[3]. These control methods need computations of the inverse dynamics and require much computational effort, therefore they are week uncertain deviations. On the other hand, new method based on the disturbance observer[4],[5] was proposed which is characterized by the facts that it does not need the inverse dynamics calculation and that it can be implemented with a simple microprocessor. However, this method depends on the intuitive approach in determining the disturbance estimation filter, and more systematic approach is desired. In recent years, the stabilizing compensators for a given plant has been derived based on the coprime factorization technique, which leads to the well-known Youla's parametrization of the controller[6]. Sugie et al.[7] extended the work to the two-degrees-of-freedom compensator and gave a very clear parametrization of the generalized two-degrees-of-freedom compensator. In spite of these theoretical results, the generalized compensators have not yet been used in industries. This is partly because the engineers are not familiar with the structure of the controllers which differs from those of the conventional and heuristic ones stated earlier. In this paper, we propose a new design approach of a two-degrees-of-freedom compensator which assures the robust stability. First of all, we clarify the internal structure of the generalized two-degrees-of-freedom compensator. By adopting this structure, we can make a bridge between the generalized compensator and the disturbance observer based controller. Secondly, based on the clarified structure, we derive a robust stability condition, and propose a design algorithm of a free parameter taking into account the condition. The proposed design algorithm is easy to implement and, as a result, we obtain lower order free parameter then that of the conventional design algorithm. Thirdly, we show by adopting an appropriate coprime factorization that the clarified structure can also be regarded as an extended version of the conventional PID compensator.

Finally, we apply the proposed algorithm to a three-degrees-of-freedom direct drive robot, and show some experimental results to verify the effectiveness of the proposed algorithm.

2 Internal structure of 2-DOF compensator

In this section, we make a brief review of the results of the parametrization of the two-degrees-of-freedom compensators. The block diagram of the two-degrees-of-freedom compensator is generally as shown in Fig.1. $P, r, u, d, andq$ represent the control plant and a command input, a con-

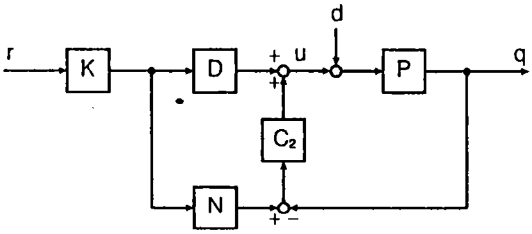

Figure 1: Internal structure of 2-DOF control system I.

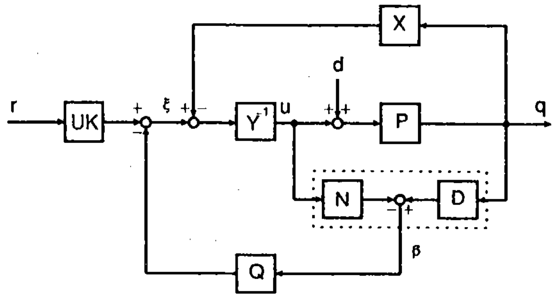

Figure 2: Internal structure of 2-DOF control system II.

trol input, an external disturbance, and a detected output(angle of joint), respectively. According to the results of Sugie et al., the parameter C_2 can be given as

$$P = ND^{-1} \quad (1)$$
$$C_2 = (Y - QN)^{-1}(X + QD). \quad (2)$$

Here, X, Y represent a particular solution that satisfies the following the Bezout Identity;

$$XN + YD = U, \quad (3)$$

where Q is a free parameter of the two-degrees-of-freedom compensator. U represents a unimodular transfer function, and $N, D, X, Y \in RH_\infty$.

The role of C_2 is to suppress the plant deviation or the external disturbance. Recalling that C_2 is parameterized also by the free parameter Q, in the one-degree-of-freedom system, it is interpreted that the parameter Q suppresses the effects of the plant deviation. In Fig.1, a designer of the controller can handle the parameter K and $Q(K, Q \in RH_\infty)$. K and Q determine the tracking performance and the feedback response, i.e. the robustness of the system, respectively. All stabilizing compensators for a given plant is derived based on the coprime factorization technique, and can be represented as equation (2) by using one free parameter Q. By using the expressions (1),(2),(3), the equivalent block diagram of Fig.1 is given by Fig.2[8]. Fig.2 reveals that the compensator is composed of 3 sub-controllers:

[1] Feedforward controller UK : This part of the controller assumes that the nominal plant is exact, and feeds forward the reference input to achieve the desired model.

[2] Feedback controller $Y^{-1}X$: This part is needed to stabilize the unstable plant, but can be omitted if the plant is stable. If the plant is exact, $y = NKr$ due to the feedforward path, and no signal is passed through this part.

[3] Disturbance observer : The disturbance is estimated and is fed back through $Y^{-1}Q$. Therefore, $Q(s)$ determines the robustness of the system. It should be noted that the stability is preserved even with time-varying stable $Q(s)$, which means that adaptation can be applied to $Q(s)$.

3 Identification of plant deviation and robust stability condition

In this section, we clarify how to identify the plant deviation based on the internal signals of 2-DOF compensator described above and propose a design method of the free parameter Q which assures the robust stability. Here, the plant uncertainty is expressed as the additive perturbations to the factors in a coprime factorization of the plant [10]. Here, we sate the several assumptions;

1. The nominal plant P is stabilized by a controller $C(s)$ parameterized by Q.

2. The perturbed plant $\tilde{P}$ is stabilized by the nominal controller $C_0(s)$.

$C_0(s)$ is obtained by setting $Q = 0$ in $C(s)$ and is expressed as

$$C_0(s) = Y^{-1}X. \quad (4)$$

Next, from the assumption 2, $\tilde{P}$ can be expressed as

$$\tilde{P} = (N + RY)(D - RX)^{-1} = \tilde{N}\tilde{D}^{-1}, \quad (5)$$

for some $R \in RH_\infty$ which represents the plant deviation [9]. In this case, R can be rewritten as

$$R = (P - \tilde{P})D\tilde{D}^{-1}U^{-1} \quad (6)$$

where N, D, X, Y satisfy the expression (1) and the Bezout Identity (3). Note that U is a unimodular transfer function. We see that R is actually an additive perturbation weighted by the frequency response of the plant.

By substituting $\tilde{P}$ for P in Fig. 2, we obtain a very simple block diagram of Fig.3, which clearly indicates how to identify the plant deviation R using the internal signal ξ and β of the two-degrees-of-freedom compensator (in case of SISO, $URU^{-1} = R$).

$$\beta = Dq - Nu \quad (7)$$
$$\xi = Yu + Xq \quad (8)$$

Also, the robust stability condition in Q can be obtained by applying small gain theorem to Fig.3 and is given by

$$\| RQ \|_\infty < 1, \quad (9)$$

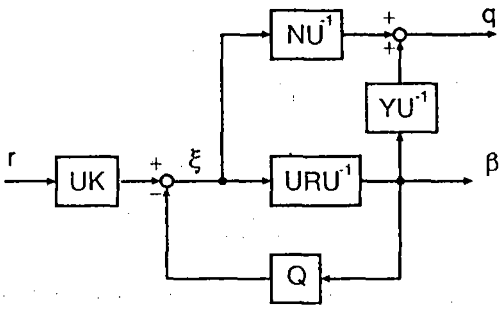

Figure 3: Internal structure of 2-DOF control system III.

where $\| \ \|_\infty$ represents the H_∞ norm and defined by

$$\| G \|_\infty = \sup_\omega |G(j\omega)|. \tag{10}$$

The condition given by equation (9) is the sufficient condition for robust stability that must be satisfied by the free parameter Q(i.e, the disturbance estimation filter). The condition quantitatively clarifies the well-known fact that the cut-off frequency of the disturbance observer cannot be chosen at high frequency.

This stability condition has been often neglected in the design of the disturbance observer by assuming that the disturbance is totally created by an external source, which is not true in the case of the parameter uncertainties.

4 Design algorithm of free parameter Q

From the discussion of the previous section, the plant deviation can be identified using the internal signal ξ, β. In this section, we show the design strategy of $Q(s)$ for SISO system, which assures the robust stability condition of equation (9) and minimizes the sensitivity function. In general, the sensitivity function is given by $S = (Y - NQ)U^{-1}D$ using the coprime factorization. This conditional optimization problem can be formulated [8] as follows,

$$\min_Q \left\|(Y - NQ)U^{-1}D\right\|_2, \tag{11}$$

$$subject\ to \ \| RQ \|_\infty < 1, \tag{12}$$

where $\| \cdot \|_2$ represents the following expression,

$$\| G \|_2 = \left[\frac{1}{2\pi}\int_{-\infty}^{\infty} | \ G(j\omega) \ |^2 \ d\omega\right]^{\frac{1}{2}}. \tag{13}$$

The design approach [8] needs mush computational effort and results in a higher order free parameter. On the other hand, if the deviation R is quite small, we may well neglect the robust stability condition, and design a free parameter from a point of sensitivity minimization. In fact, as shown in the following section, even in case of the direct drive robot, plant deviation R is small.

Next, we consider a method of the sensitivity minimization. In general, Q should be calculated by solving the Nehari problem. However, in this paper we propose a new simple design algorithm of free parameter Q.

(1) Identify the plant deviation R.
(2) Design the free parameter Q.
  (2a) Calculate $Q_{minS} = YN^{-1}$.
  (2b) Check the robust stability condition.
    If it is satisfied, then $Q_{out} = Q_{minS}$ and go to (2d).
    If it is not satisfied, then go to (2c).
  (2c) Lower the gain of Q_{minS}, so that it satisfies
    the robust stability condition($\| RQ_{minS} \|_\infty < 1$)
  (2d) Determine a low-pass filter Q_{filter} to
    cut down the high frequency gain.
  (2e) Calculate $Q = Q_{out}Q_{filter}$.

In step (2d), Q_{out} is multiplied by a low pass filter, because precision of the identified R is degraded remarkably for high frequency band.

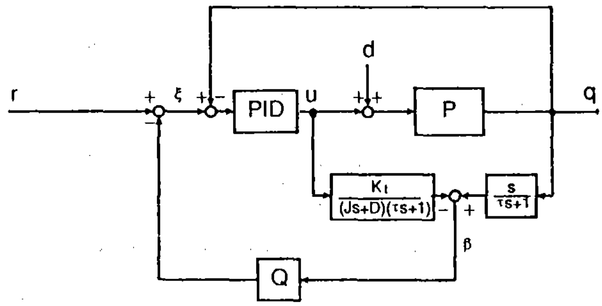

Figure 4: Control system of each Joint.

5 Control system and experimental results

5.1 Coprime factorization

In this section, we adopt a new coprime factorization which can be regarded as the extended version of the the conventional PID controller. In case of the independent joint control, by regarding the torque current as an input and the joint angle as an output, we get the following transfer function as a nominal plant.

$$P = \frac{K_t}{(Js + D)s} \tag{14}$$

where K_t is a torque constant, J is a inertia moment, and D is a damping coefficient.

In this paper, based on the two-degrees-of-freedom compensator described in the early section, we adopt the following coprime factorization which clearly shows itself as an extended version of the conventional PID compensator.

$$\begin{aligned} N &= \frac{K_t}{(Js + D)(\tau s + 1)}, D = \frac{s}{\tau s + 1}, X = 1, \\ Y &= C_{PID}^{-1} \\ &= \frac{s(1 + \tau_d s)}{K_v s^2 + (1 + \tau_d s)K_p s + (1 + \tau_d s)K_i} \end{aligned} \tag{15}$$

Here, $N, D, X, Y \in RH_\infty$, and equation (15) satisfies equation (3). And K_v, K_p, K_i, τ_d are the gains of the PID compensator. With this coprime factorization, the overall system block diagram for each joint is given by Fig.4 in which select the free parameter $K = U^{-1}$. Note that the transfer function from r to q is given by NU^{-1}, therefore, the tracking performance can be specified by a adjusting PID parameters. Consequently, by using this coprime factorization, we can easily extend the conventional PID controller with some additional elements and get much better performances.

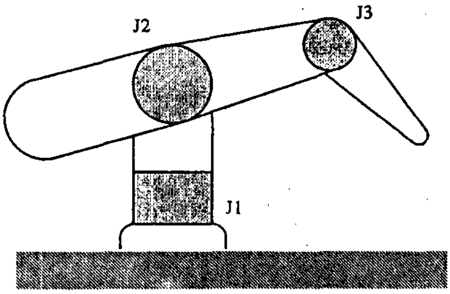

Figure 5: 3-degrees-of-freedom direct drive robot.

Table 1: Specifications of direct drive robot.

Joint number		1	2	3
Rated power	[W]	410		250
Maximum torque	[Nm]	100		30
Maximum speed	[rps]	1.2		2.4
Resolution of encoder	[p/rev]	1024000		655360
Link length	[cm]	40	22.5	24

5.2 Experimental setup

We applied the proposed compensator to the trajectory control of a three-degrees-of-freedom direct drive manipulator, shown in Fig. 5. Table 1. shows the specifications of the manipulator, and Fig.6 shows the experimental system setup. Each joint was driven independently by the control system of Fig.4. Only joint 2 and 3 were used. We used 80386 and DSP(Digital Signal Processor NEC–μ PD 77230) as the control units which generates the control signal and the identification signal and send it to each joint driver through D/A. A sampling period was 400[μsec]. The joint 2 and 3 more simultaneously. This results in the gravitational and the interactional disturbing forces. To identify R, we set $Q = 0$ in Fig.4 and measured 12500 pairs of (ξ , β) signal.

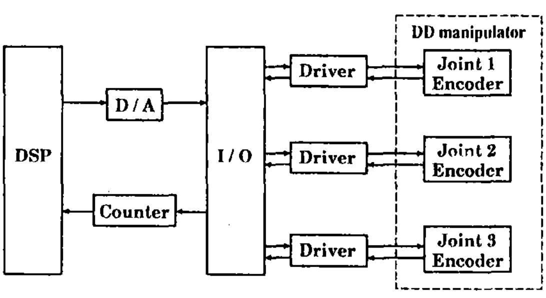

Figure 6: Schematic diagram of experimental system.

5.3 Experimental results

In the following, we present the experimental procedures and results in detail. At first, we attempted the following test to show the effectiveness of the proposed control system. Firstly, we tuned the PID gains so that if realize the desired tracking performance specification. Each parameter is given by $K_v = 0.144, K_p = 0.632, K_i = 0$. Secondly, to identify R, we let $Q = 0$ and input $\xi = 1$[rad](step signal). Fig.7 shows the identification result. Next, we calculated Q_{minS} as step (2a) and get simple $Q_{minS} = 2.500 \times 10^{-1}$(see Fig.7). As shown in Fig.7, $\| RQ_{minS} \|_\infty$ satisfies the robust stability condition so that we multiply Q_{minS} by Q_{filter} to get the final Q(step(2d)). Here, we selected 100[Hz] as the cut-off frequency of Q_{filter} and the obtained Q is given as follows;

$$Q(s) = \frac{2.500 \times 10^{-1}}{1.000 \times 10^{-2}s + 1.000}$$

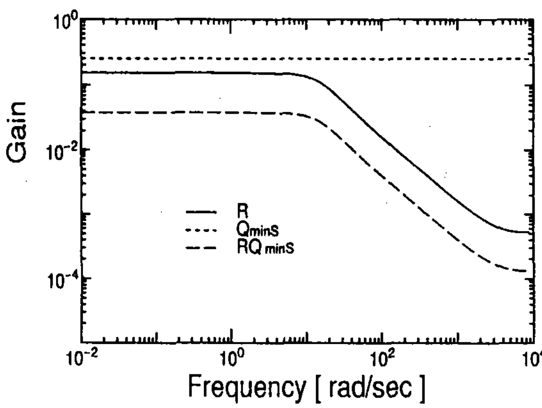

Fig. 7. Gain plot of R,Q_{minS},RQ_{minS}
(In case of $\| RQ_{minS} \|_\infty < 1$)

We compared the performance of proposed compensator using the above Q with that of the conventional PID controller. In particular, we tested their step responses and responses to a stepwise disturbance. Fig.8 and Fig.9 show the results. The PID gains were for the conventional controller $K_v = 0.18, K_p = 0.76, K_i = 0.1$, and the values were chosen so that almost same command tracking performances can be obtained in both the PID and the proposed compensator.

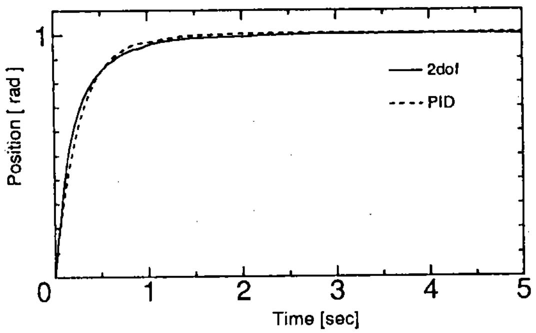

Figure 8: Response for stepnose command of joint 3 I.

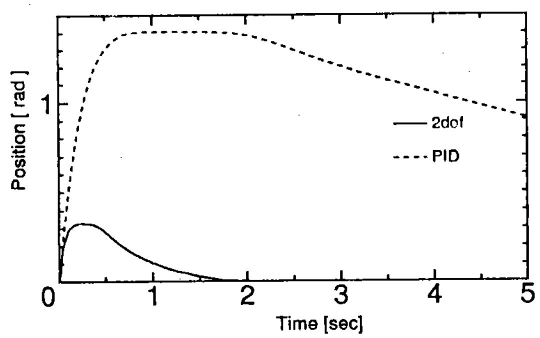

Figure 9: Response for stepnose disturbance command of joint 3.

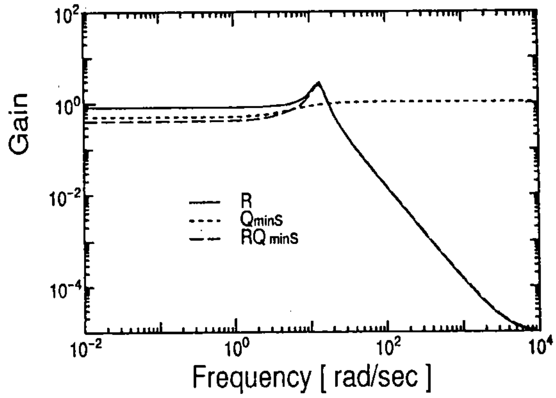

Figure 10. Gain plot of R,Q_{minS},RQ_{minS} (In case of $\| RQ_{minS} \|_\infty > 1$)

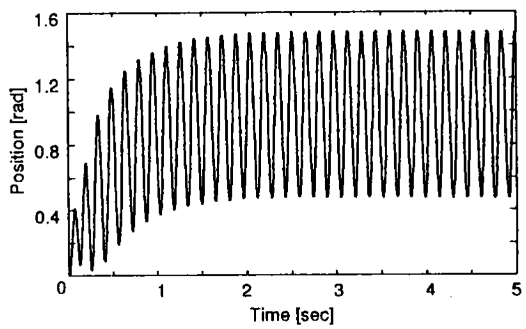

Figure 11: Response for stepwise disturbance command of Fig.10 II.

We see in Fig.8 that the command response of controller using PID only(dashed-line) and using two-degrees-of-freedom (solid-line) does not make a noticeable difference. In Fig.9, however, we show the responses to stepwise disturbance(that is, we set the tracking command to 0[rad] and added a step signal as disturbance) and could see that two-degrees-of-freedom compensator(solid-line) showes a far better disturbance rejection response (see Fig.9). Consequently, we see that two-degrees-of-freedom compensator guarantees not only the command tracking performance but also desirable disturbance rejection properly.

Next, to verify the validity of the robust stability condition (9), we performed the following test. Firstly we redesign the PID compensator, so that the deviation R become larger($K_v = 0.0338$, $K_p = 0.3016$, $K_i = 0$). Fig.11 shows the identified deviation R. And then, as a second step, we get Q_{minS} from YN^{-1} and obtain the Q by multiplying Q_{minS} by Q_{filter}

In this case, the step response was unstable, because the robust stability condition $\| RQ_{minS} \|_\infty < 1$ was violated. So a natural remedy to provide the stable response was as given in step (2c) in section 4, to lower the gain of Q_{minS} so that robust stability condition $\| RQ_{minS} \|_\infty < 1$ could be satisfied. After redesigning Q_{minS} and multiplying it by Q_{filter}, we obtained the following

$$Q_3(s) = \frac{3.575 \times 10^{-2}s + 1.508 \times 10^{-1}}{1.693 \times 10^{-2}s^2 + 3.531 \times 10^{-1}s + 3.016 \times 10^{-1}}.$$

In this case, Q_{minS} satisfies the condition $\| RQ \|_\infty <$ 1(Fig.13) and provides a stable, step response as shown in Fig.13

6 Conclusions

In this paper, firstly we have clarified the internal structure of the generalized two-degrees-of-freedom compensator and proposed a new coprime factorization which can explicitly be regarded as the extended version of the conventional PID compensator. Secondly, we show the simple design algorithm for Q which mainly focused on the sensitivity minimization. By using the proposed method, we could make up a two-degrees-of-freedom compensator which satisfies the robust stability, command tracking and disturbance rejection characteristics simple by extending the conventional PID controller,

Finally, we verified the effectiveness of the proposed

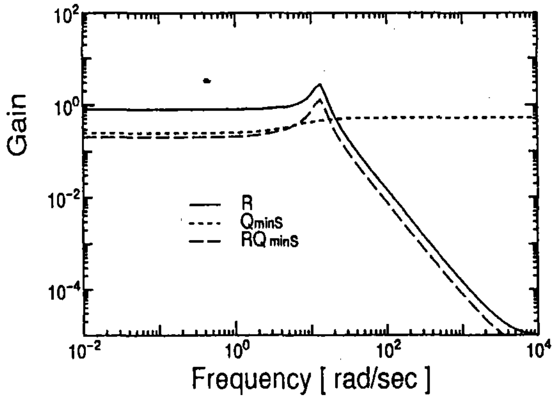

Figure 12. Gain plot of R,Q_{minS},RQ_{minS}
(In case of $\| RQ_{minS} \|_\infty < 1$)

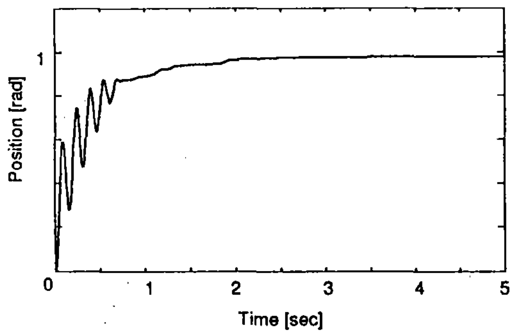

Figure 13: Response for stepwise disturbance command of Fig.12. III

method by the experiments using the direct drive manipulator.

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