Motion Control for Advanced Mechatronics

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(Invited Paper)

Abstract—**Motion control is now recognized as a key technology in mechatronics. The robustness of motion control will be represented as a function of stiffness and a basis for practical realization. Target of motion is parameterized by control stiffness which could be variable according to the task reference. However, the system robustness of motion always requires very high stiffness in the controller. The paper shows that control of acceleration realizes specified motion simultaneously with keeping the robustness very high. The acceleration is a bridge to connect such robustness and variable stiffness. For practical applications, a technique to estimate disturbance is introduced to make motion controller to be an acceleration controller. Motion control of flexible structure and identification of mechanical parameters are also described.**

I. Introduction

ONE of the most important elements in mechatronic technology is undoubtedly motion control. However, the word "mechatronics," registered as a trademark by Yaskawa Electric Co., in 1971 did not always include a concept of motion control [1].

In the 1970's, industries began to replace mechanical elements with electronic ones to achieve higher reliability and less maintenance. Also the mechatronic devices were designed to occupy smaller space in the final products. Totally function of reliability, availability, and serviceability has been very much improved in relatively more compact products.

In the 1980's, a remarkable progress in mini- and micro-computers and power electronics technology made it possible to improve the performance of motion. For example, vector controlled induction motor has higher cut-off frequency almost up to three times in the speed control loop compared to the same-sized dc motor. Following these results, the novel theories of control were tested in such mechatronic systems. In the late 1980's and the early 1990's, mechatronics seemed a showcase of various application of control theories.

The first IEEE Workshop on Advanced Motion Control (AMC'90) held in 1990 pointed out the importance of physical interpretation of motion control, though the proceedings included many examples of modern control techniques [2]. The phenomena observed in the early 1990's also came from the so-called "software-servo technology." Generally major part of software applied to motion control carries out the indispensable routines for diagnostics and sequential procedures. Only small area is assigned for programming control algorithms. The area was hardly sufficient for conventional PID controller. Recently the fast processor has gradually enabled more complicated algorithms within a shorter sampling time. Since the software-servo technology has generated more room for control algorithms, higher performance and flexibility have been realized without additional investment. Then the novel algorithms have gained high evaluation from the practical viewpoint because the quality of motion was improved. The motion control is now recognized as an important area in mechatronics [3]–[6].

The paper intends to show recent advances in motion control covering control and energy conversion [7] for a tutorial purpose. The physical meaning is emphasized rather than mathematical exactness. As is well known, control and estimation are twin aspects of system design. The fact holds in motion control. The robust control and the estimation of parameters have the same basis. The several examples shown later seem different approaches; however, the single interpretation is possible from the physical viewpoint. The paper, at first, defines the stiffness in relation to various motion control. This concept leads to both the meaning of robustness and the general structure of motion control. Then the paper points out the necessity of modification against flexible structure. Several examples will assure the concluded remarks at the end of the paper.

II. Target of Motion Control

A mechanical system governed by the Lagrange equation is represented both geometrically and dynamically. The kinematics is represented as a set of algebraic equations which gives constraints of motion. The dynamics is a set of differential equations based on dynamic equilibrium of force. A motion controller generates a set of inputs to the actuators according to motion reference. A motion reference is synthesized in the reference generator. The sensor signal, the database and the commands from other motion systems and/or human operators are input signals for the reference generator. There will be some intelligent process with composite structure in the reference generator. The general motion control totally consists of the motion controller and the reference generator. However, the paper lays stress on the motion controller. Fig. 1 shows a schematic relation of each component of mechatronics.

From the control point of view, the output of the motion will be position and/or force. A simple case is continuous path tracking, however, the need for force control is increasing because the industrial demand to the dexterous motion is growing up. A simplified index which covers various motion

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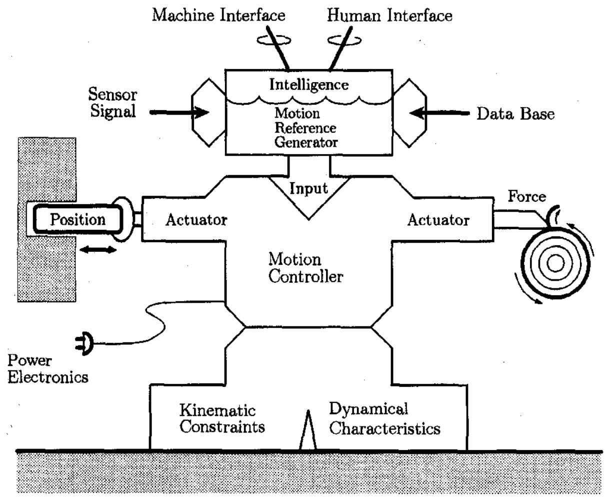


Fig. 1. A schematic relation of each component of mechatronics.

TABLE I
STIFFNESS AS A MOTION INDEX

target of motion	stiffness κ
position	∞
compliant	finite
force	0

is preferable, though there are various candidates of motion representation. One of such indices is a stiffness.

Suppose that x is a position of motion of a controlled object and f is a totally imposed force on that. From the kinematic and the dynamic equation, the following holds

$$f = g(\ddot{x}, \dot{x}, x). \tag{1}$$

The stiffness κ is defined in the partial differentiation

$$\kappa = \frac{\partial f}{\partial x}. \tag{2}$$

The ideal position control inhibits any deviation of position against any deviation of force. That means κ will be infinite in such a case. Naturally an integrator in the forward loop compensates the steady error and δx will be zero at infinite time. However, such function does not reflect in (2). On the other hand, the ideal force control inhibits any force deviation against any position deviation. Therefore, κ is zero in the ideal force control. In the compliance control, there must be a relation between position and force. For instance, a virtual compliance control will have a mechanical impedance computed in the controller according to the specified dynamics. Table I shows that κ is a good parameter as an index which represents a target of motion.

III. ROBUSTNESS OF MOTION CONTROL

A. Concept of Robustness

Various specifications for motion in industry require versatile ability in the controller. An efficient way to overcome this problem is to divide the function of motion control into two parts. Control flexibility is suitably realized in the motion reference generator since a kind of intelligence is indispensable in this part. It is necessary to track the motion reference accurately in the motion control part. The more intelligent a motion reference generator becomes, the more robust a motion controller should be. This is a kind of master-slave structure.

There is an interpretation on robustness of motion controller, which makes the conception visible in mind. Suppose the moving body whose position is controlled along the predetermined path. Such a rigid body should knock down or break any obstacles on the path and go forward to the end of path, if a motion controller is ideally robust. So-called obstacle avoidance issue is solved in the motion reference generator by synthesizing an appropriate reference of trajectory. The robustness of the motion controller assures to the utmost the "high-fidelity" to the input reference.

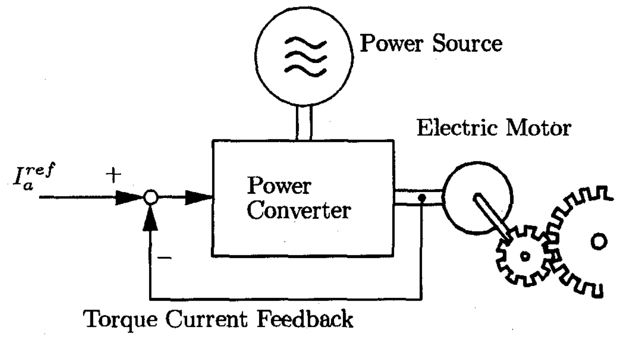


Fig. 2. A typical electric drive system.

TABLE II
TYPICAL ELECTRIC ACTUATORS

	dc motor	induction motor	synchronous motor
field	permanent magnet	field current by vector control	rotating permanent magnet with field orientation
torque	dc current	torque current by vector control	ac current with orientation

B. Actuation

The dynamical equation is excited by input force. Most of mechatronic systems adopt electrical actuator for the purpose. Fig. 2 shows a typical electric drive system.

Most of power converters use switching devices for power control. The regulation of torque highly depends on the switching frequency. Since the recent power converter uses IGBT's, FET's, and so on, for fast switching, the current feedback includes high gain inside the feedback loop and the torque current follows the current reference with delay of less than from 50 μs to 1 ms. The torque itself is produced by electromagnetic interference of current and magnetic flux.

There are three types of the interference as shown in Table II. The stepping motor, which is not in Table II, is widely used for simple positioning. It has a similar characteristics of synchronous motor, however, it generates high torque ripple

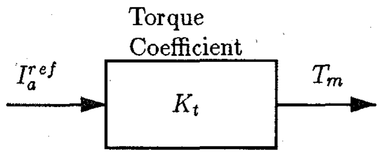


Fig. 3. Block diagram of actuator.

and is not appropriate for fine and smooth motion. Each motor in Table II generates torque by the product of torque current by field. The field and the torque current is controlled to be orthogonal to each other [8]. Then by integrating all the torque par small piece of surface of rotor, the total generated torque T_m is given simply as

$$T_m = K_t I_a. \tag{3}$$

K_t is a function of flux position and expanded in Fourier series and is called a torque coefficient. I_a is torque current. Fast switching devices make the power converter with feedback of torque current as a virtual current converter. In most cases, it is possible to regard I_a as I_a^{ref} (torque current reference). As a result, this chapter concludes that the actuation part is schematically represented in Fig. 3.

C. Disturbance

It is necessary to define the equivalent disturbance in order to consider the robust control of motion actuated by electric motor. The explanation and the interpretation of robustness and stiffness in motion control lead to definition of disturbance. The general definition for single-input and single-output (SISO) linear system is discussed. Such system has the following transfer function between input $U(s)$ and output $Y(s)$:

$$\frac{Y(s)}{U(s)} = K\,\frac{c_m s^m + c_{m-1}s^{m-1} + \cdots + c_2 s + c_1}{s_n + a_n s^{n-1} + a_{n-1}s^{n-2} + \cdots + a_2 s + a_1}. \tag{4}$$

Here $y(t)$ is output variable and $u(t)$ is input variable. If the disturbance $d(t)$ is additive in the input side, the system is represented in the following state equation:

$$\begin{aligned} \dot{\boldsymbol{x}} &= \boldsymbol{A}\boldsymbol{x} + \boldsymbol{b}u + \boldsymbol{e}d \\ y &= \boldsymbol{c}\boldsymbol{x}. \end{aligned} \tag{5}$$

Here

$$\boldsymbol{x} = \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{bmatrix}$$

$$\boldsymbol{A} = \begin{bmatrix} 0 & 1 & 0 & \cdots & 0 \\ 0 & 0 & 1 & \cdots & 0 \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & 0 & \cdots & 1 \\ -a_1 & -a_2 & -a_3 & \cdots & -a_n \end{bmatrix}$$

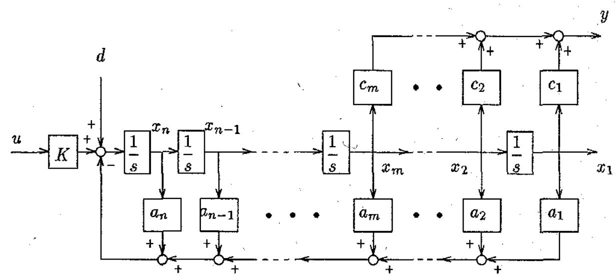


Fig. 4. Companion form of linear system.

$$\boldsymbol{b} = \begin{bmatrix} 0 \\ 0 \\ \vdots \\ 0 \\ K \end{bmatrix}$$

$$\boldsymbol{e} = \begin{bmatrix} 0 \\ 0 \\ \vdots \\ 0 \\ 1 \end{bmatrix}$$

$$\boldsymbol{c} = [c_1 \quad c_2 \quad \cdots \quad c_m \quad 0 \quad \cdots \quad 0].$$

$\boldsymbol{x}$ is a state vector, $\boldsymbol{A}$ is a system matrix, $\boldsymbol{b}$ is a distribution vector of input, $\boldsymbol{e}$ is a distribution vector of disturbance, and $\boldsymbol{c}$ is an observation column vector. Equation (5) is represented in Fig. 4.

The parameter variation and the disturbance should not give any significant effect to output in robust control. At first, the parameter variation is evaluated. Suppose that the variation of system matrix $\boldsymbol{A}$ and the distribution vector $\boldsymbol{b}$ is additive to the nominal state denoted by lower suffix $_0$:

$$\begin{aligned} \boldsymbol{A} &= \boldsymbol{A}_0 + \Delta\boldsymbol{A} \\ \boldsymbol{b} &= \boldsymbol{b}_0 + \Delta\boldsymbol{b}. \end{aligned} \tag{6}$$

$\Delta\boldsymbol{A}$ is a variation of $\boldsymbol{A}$ and $\Delta\boldsymbol{b}$ is a variation of $\boldsymbol{b}$. The variation of dynamic matrix $\boldsymbol{A}$ is the same to the variation of the coefficients of the characteristic equation of (4). An extended disturbance is defined by modification of (5):

$$\begin{aligned} \dot{\boldsymbol{x}} &= (\boldsymbol{A}_0 + \Delta\boldsymbol{A})\boldsymbol{x} + (\boldsymbol{b}_0 + \Delta\boldsymbol{b})u + \boldsymbol{e}d \\ &= \boldsymbol{A}_0\boldsymbol{x} + \boldsymbol{b}_0 u + (\Delta\boldsymbol{A}\boldsymbol{x} + \Delta\boldsymbol{b}u + \boldsymbol{e}d). \end{aligned} \tag{7}$$

The third term in the right side is an extended disturbance defined to have the dimension of torque or force

$$\tilde{d} = d + \boldsymbol{e}^t(\Delta\boldsymbol{A}\boldsymbol{x} + \Delta\boldsymbol{b}u). \tag{8}$$

By introduction of the extended disturbance, (5) is transformed to (9):

$$\begin{aligned} \dot{\boldsymbol{x}} &= \boldsymbol{A}_0\boldsymbol{x} + \boldsymbol{b}_0 u + \boldsymbol{e}\tilde{d} \\ y &= \boldsymbol{c}\boldsymbol{x}. \end{aligned} \tag{9}$$

D. Estimation of Disturbance and Acquisition of Robustness

There are various proposals to estimate the disturbance. The chapter introduces a disturbance observer. Since the extended disturbance is the function of time, it is approximated by polynomials of $p-1$ order [9]. Then (10) holds:

$$\frac{\mathrm{d}^{(p)}\tilde{d}}{\mathrm{d}t^{p}} = 0. \tag{10}$$

By putting (10) into (9), an augmented equation is obtained:

$$\begin{aligned} \dot{\tilde{\boldsymbol{x}}} &= \tilde{\boldsymbol{A}}_0\tilde{\boldsymbol{x}} + \tilde{\boldsymbol{b}}_0 u \\ y &= \tilde{\boldsymbol{c}}_0\tilde{\boldsymbol{x}}. \end{aligned} \tag{11}$$

Here, the order of the matrix is $n+p$ and $\tilde{\boldsymbol{x}}$ is as follows:

$$\tilde{\boldsymbol{x}} = \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \\ \tilde{d} \\ \dot{\tilde{d}} \\ \ddot{\tilde{d}} \\ \vdots \\ \tilde{d}^{(p-1)} \end{bmatrix}.$$

$\tilde{\boldsymbol{A}}_0$, $\tilde{\boldsymbol{b}}_0$, $\tilde{\boldsymbol{c}}_0$ are as follows

$$\tilde{\boldsymbol{A}}_0 = \overbrace{\begin{bmatrix} 0 & 1 & 0 & \cdots & 0 & 0 & 0 & \cdots & 0 \\ 0 & 0 & 1 & \cdots & 0 & 0 & 0 & \vdots & 0 \\ \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & 0 & \vdots & 0 \\ 0 & 0 & 0 & \cdots & 1 & 0 & 0 & \vdots & 0 \\ -a_{01} & -a_{02} & -a_{03} & \cdots & -a_{0n} & 1 & 0 & \vdots & 0 \\ 0 & 0 & 0 & \cdots & 0 & 0 & 0 & \vdots & 0 \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ 0 & 0 & 0 & \cdots & 0 & 0 & 0 & \cdots & 0 \end{bmatrix}}^{n+p}$$

$$\tilde{\boldsymbol{b}}_0 = \left.\begin{bmatrix} 0 \\ 0 \\ \vdots \\ 0 \\ K_0 \\ 0 \\ \vdots \\ 0 \end{bmatrix}\right\} n+p$$

$$\tilde{\boldsymbol{c}}_0 = \overbrace{[c_1 \quad c_2 \quad \cdots \quad c_m \quad 0 \quad \cdots \quad 0]}^{n+p}.$$

In (11), an equivalent disturbance defined by (8) seems a state variable. Clearly the system is uncontrollable, however, is observable. It is possible to construct an observer which estimates state variables. The minimum order of observer is, therefore, $n+p-m$. Gopinath's method is a systematic way to construct such an observer [10]. Once $\tilde{d}$ is estimated as $\hat{\tilde{d}}$, the input will be sum of two parts:

$$u = u^{ref} + u^{dis}. \tag{12}$$

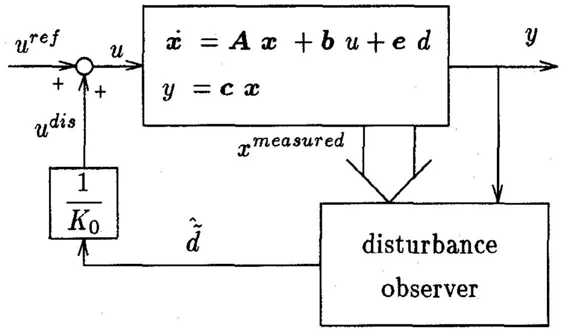


Fig. 5. Robust control based on disturbance observer.

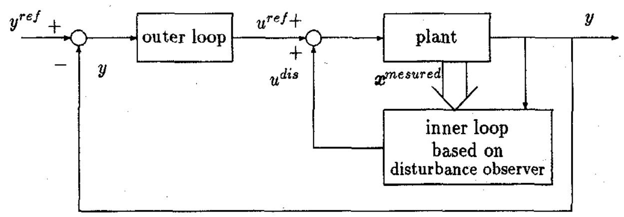


Fig. 6. Total system with robust control.

The first term in the right side is a driving input to excite the system. The second term is a compensation to suppress the equivalent disturbance and the system acquires robustness. To cancel the equivalent disturbance, the compensation input is made by using the estimated equivalent-disturbance:

$$\begin{aligned} u^{dis} &= -(\boldsymbol{b}_0^t\boldsymbol{b}_\mathbf{o})^{-1}\boldsymbol{b}_\mathbf{o}{}^t\boldsymbol{e}\hat{\tilde{d}} \\ &= -\frac{1}{K_0}\hat{\tilde{d}}. \end{aligned} \tag{13}$$

Since $\hat{\tilde{d}}$ will be delayed by the lag poles in the disturbance observer, the compensation of the equivalent disturbance will be also delayed by the same amount. It is possible to design such delay as small as possible not to make robust stability deteriorate. The compensation input u^{dis} will change the original system into the nominal system without any disturbance.

Fig. 5 visualizes a schematic diagram of the total system including the disturbance observer. It is noted that the design of u^{ref} comes from the motion reference generator.

Generally total controller will have cascade of the outer loop to bring the desired output and the inner loop by disturbance observer. The former will be a nest of the latter as shown in Fig. 6.

E. Robust Motion Controller

The previous design method is applied to the motion system described by (14):

$$J\frac{\mathrm{d}\omega}{\mathrm{d}t} = K_t I_a^{ref} - T_l. \tag{14}$$

Here

J inertia;
K_t torque coefficient of electric motor;
T_l load torque.

The disturbance is load torque. The parameter variations are the change of inertia and the change of torque constant of motor. The output is position detected by position detector. The equivalent disturbance defined by (8) is

$$\tilde{d} = -\frac{T_l}{J} + \left(\frac{K_t}{J} - \frac{K_{t0}}{J_0}\right) I_a^{ref}. \tag{15}$$

Suppose the first derivative of $\tilde{d}$ is zero. An augmented state equation corresponding to (11) is

$$\frac{\mathrm{d}}{\mathrm{d}t}\begin{bmatrix}\theta\\ \omega\\ \tilde{d}\end{bmatrix} = \begin{bmatrix}0 & 1 & 0\\ 0 & 0 & 1\\ 0 & 0 & 0\end{bmatrix}\begin{bmatrix}\theta\\ \omega\\ \tilde{d}\end{bmatrix} + \begin{bmatrix}0\\ \frac{K_{t0}}{J_0}\\ 0\end{bmatrix} I_a^{ref}. \tag{16}$$

By Gopinath's method, the following estimation process is obtained:

$$\hat{\tilde{d}} = k_1\theta + z_1.$$

z_1 should satisfy (17), where k_1 and k_2 are free parameters:

$$\frac{\mathrm{d}}{\mathrm{d}t}\begin{bmatrix}z_1\\ z_2\end{bmatrix} = \begin{bmatrix}0 & -k_1\\ 1 & -k_2\end{bmatrix}\begin{bmatrix}z_1\\ z_2\end{bmatrix} + \begin{bmatrix}-k_1k_2\theta\\ (k_1 - k_2^2)\theta + \frac{K_{t0}}{J_0} I_a^{ref}\end{bmatrix}. \tag{17}$$

Equations (16) and (17) lead (18):

$$\begin{aligned}\hat{\tilde{d}} &= \frac{k_1}{s^2 + k_2 s + k_1}\left(s^2\theta - \frac{K_{t0}}{J_0} I_a^{ref}\right)\\ &= \frac{k_1}{s^2 + k_2 s + k_1}\tilde{d}.\end{aligned} \tag{18}$$

Two poles of the observer are α and β, which are arbitrarily allocated in the complex plane. They satisfy (19):

$$\begin{aligned}\alpha + \beta &= -k_2\\ \alpha\beta &= k_1.\end{aligned} \tag{19}$$

It is worthwhile reconsidering (14). The parameters in (14) are the inertia and the torque coefficient. The inertia will change according to the mechanical configuration of motion system. The torque coefficient will vary according to the rotor position of electric motor due to irregular distribution of magnetic flux on the surface of rotor

$$J = J_0 + \Delta J \tag{20}$$

$$K_t = K_{t0} + \Delta K_t. \tag{21}$$

By substituting (20) and (21) into (14), (22) holds:

$$J_0\frac{\mathrm{d}\omega}{\mathrm{d}t} = K_{t0} I_a^{ref} - \left(T_l + \Delta J\frac{\mathrm{d}\omega}{\mathrm{d}t} - \Delta K_t I_a^{ref}\right). \tag{22}$$

The second term of (22) is the disturbance torque T_{dis}

$$T_{dis} \equiv T_l + \Delta J\frac{\mathrm{d}\omega}{\mathrm{d}t} - \Delta K_t I_a. \tag{23}$$

Comparing (14), (15), and (23), the following equation holds:

$$T_{dis} = J_0(-\tilde{d}). \tag{24}$$

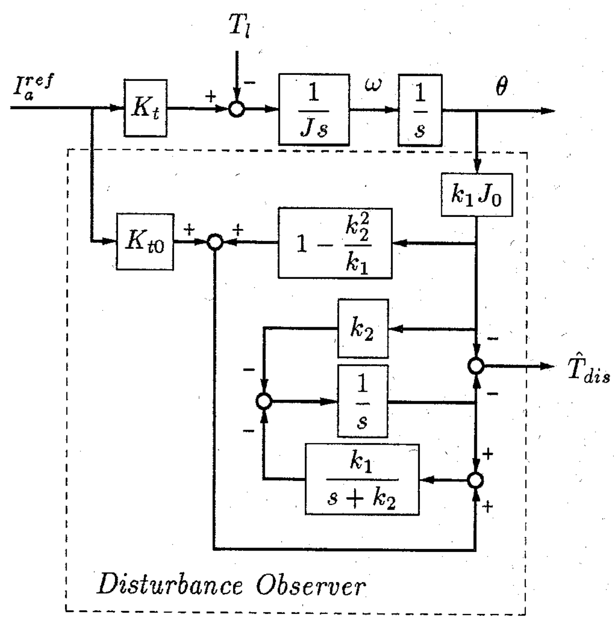


Fig. 7. Disturbance observer in motion system.

T_{dis} contains

- mechanical load ($=T_l$);
- varied self-inertia torque [$= \Delta J\,(\mathrm{d}\omega/\mathrm{d}t)$];
- torque ripple from motor ($=\Delta K_t I_a$).

The robust motion controller is designed to cancel the disturbance torque as quickly as possible.

The estimated disturbance torque is obtained from the position θ and the current reference as shown in Fig. 7. According to the result of (12) and (13), compensation input is as follows:

$$\begin{aligned}I^{dis} &= -\frac{J_0}{K_{t0}}\hat{\tilde{d}}\\ &= \frac{1}{K_{t0}}\hat{T}_{dis}.\end{aligned} \tag{25}$$

Robust motion controller has the schematic block diagram as shown in Fig. 8. There exists an integrator with high gain equivalently in the forward path as shown in Fig. 9. Therefore, the robust motion controller eliminates steady state error.

Equation (18) shows that the disturbance is estimated through low-pass filter. Generally, there is such a low-pass filter in the observer structure. The poles of the observer determines the delay of the low-pass filter $G_T(s)$. $G_T(s)$ gives a certain effect to the control performance. Fig. 7 is also transformed into Fig. 10.

Fig. 10 transformed from Fig. 7 clarifies the feedback effect of the disturbance. If there is no delay in the estimation process, the disturbance is completely canceled out. In fact, since there is definitely some time-delay in the estimation process, the controlled system is not robust in high frequency range determined by $1 - G_T(s) = G_S(s)$. $G_S(s)$ is called a sensitivity function which shows a performance limit of robust control in high frequency range. In most of low frequency area covered by $G_T(s)$, the motion system is robust.

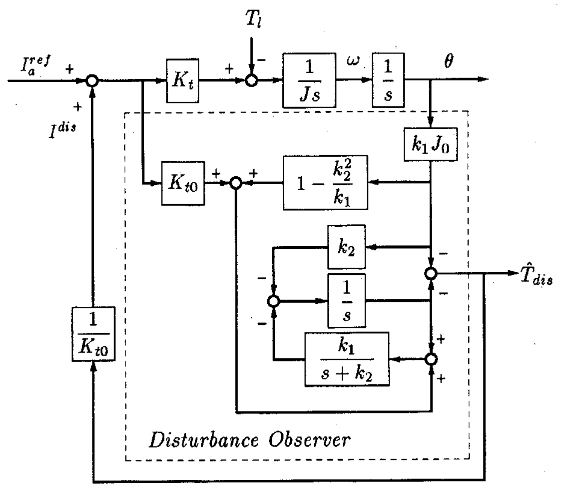


Fig. 8. A robust motion controller.

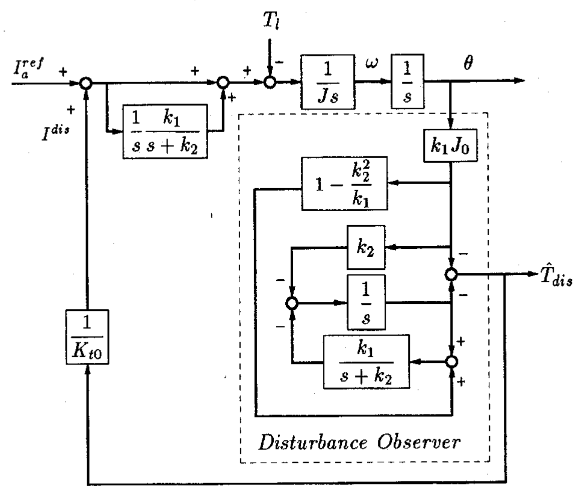


Fig. 9. Equivalent block diagram of Fig. 8.

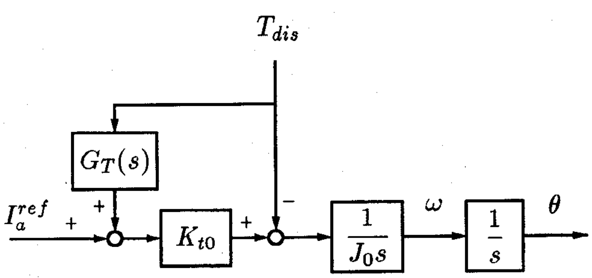


Fig. 10. A robust motion controller as an acceleration controller.

Fig. 10 shows another interpretation. It is possible to select nominal inertia and nominal torque coefficient as unity. This case shows that a current reference is also an acceleration reference.

The chapter reaches a result that robust motion controller makes a motion system to be an acceleration control system. The result implies a versatility of robust motion controller for both position and force control. If position signal is fed back,

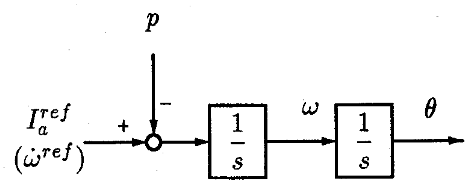


Fig. 11. Acceleration controller.

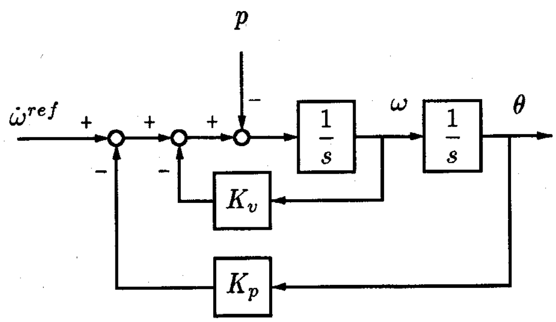


Fig. 12. State feedback in acceleration controller.

a high-gain feedback in the robust controller makes stiffness very high. On the contrary, only pure force error feedback makes total stiffness zero since there is no gain to the position.

IV. Position Control and Force Control

A. Position Control

Positioning is one of the important application in motion control. There are two kinds of industry requirements to the positioning:

- PTP path (Point-To-Point Path);
- CP path (Continuous Path).

The trapezoidal profile of speed reference is used for PTP path tracking. During acceleration and deceleration, the period of constant acceleration is controlled to attain maximum speed.

As to CP tracking, a trajectory of motion is predetermined and it is possible to know the several steps ahead at any time. If the motion reference generator knows two steps ahead, the velocity and the acceleration reference are calculated as well as position reference. The robust motion controller makes the motion system an acceleration controller as shown in Fig. 11 with unity gain for nominal inertia and torque coefficient [11]. p is an equivalent acceleration to the disturbance

$$p = J_0^{-1} G_s(s) T_{dis}. \tag{26}$$

Fig. 11 has two unstable poles at origin. One simple way to stabilize the system is state feedback, i.e., position and velocity feedback as shown in Fig. 12. Two poles will be allocated arbitrarily in the left complex plane. The transfer function $P_m(s)$ from acceleration reference to position is

$$P_m(s) = \frac{1}{s^2 + K_v s + K_p}. \tag{27}$$

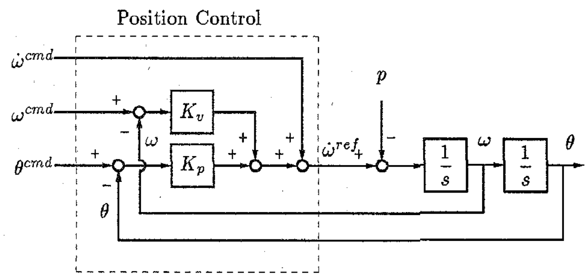


Fig. 13. Position control block diagram.

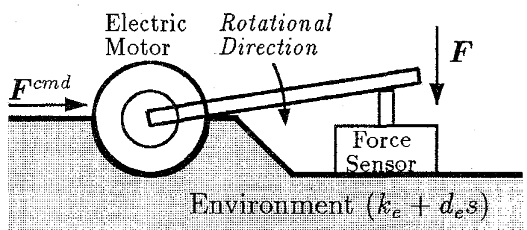


Fig. 14. Mechanical loop for contact task.

To recover the delay due to $P_m(s)$, inverse system is placed in front of $P_m(s)$

$$P_m^{-1}(s) = s^2 + K_v s + K_p. \tag{28}$$

Since the input of $P_m^{-1}(s)$ is position reference θ^{ref}, which will have first and second derivatives as the any place of predetermined trajectory, CP tracking control is constructed as shown in Fig. 13 by combination of (27), (28), and Fig. 12. The transfer function from θ^{cmd} to θ is as follows:

$$\theta = \theta^{cmd} - \frac{p}{s^2 + K_v s + K_p}. \tag{29}$$

The second term of right side is an error due to disturbance. Most of them is suppressed in robust control part and the little remained error is attenuated by the velocity and position feedback. It is noted that the forward gain from position command to position is unity.

B. Force Control

In the industry, the current control feedback in Fig. 2 has been widely used as a torque control. This loop has only a function to make a power converter to be a controlled current source as previously mentioned. The target of force control is a control of force at the end-effector accurately. The robustness of the force control system is also required. Therefore, Fig. 11 is also a basis for force control and there should not be high forward gain to position in order to keep stiffness κ as low as possible. If the force sensor is ideal, there would be no forward gain for the position and zero stiffness is attained. However, very small deviation proportional to the imposed force could exist in the force sensor and the robustness will be suffered.

There are two categories for force control:

- noncontact motion;
- contact motion.

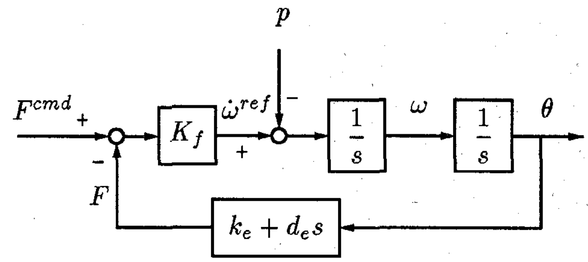


Fig. 15. Block diagram of force control (I).

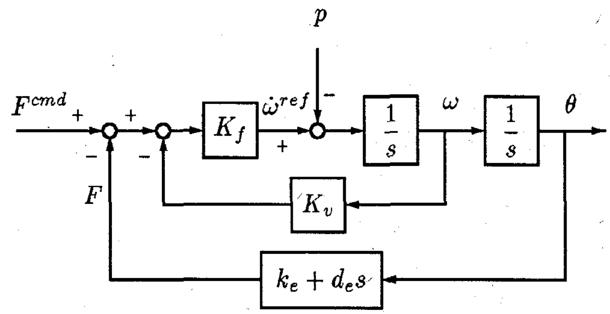


Fig. 16. Block diagram of force control (II).

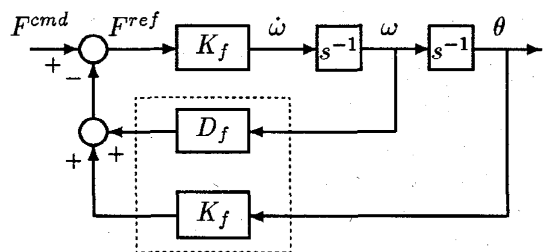


Fig. 17. Impedance control.

In noncontact motion, a force control is substantially an acceleration control. An end-effector moves along force reference until it collides with a fixed environment.

In contact motion as shown in Fig. 14, there will be a force sensor between the end-effector and the control object. The sensor will detect very small deviation proportional to the imposed force. Then at that moment, a mechanical loop including environment is set up as shown in Fig. 14, where k_e is stiffness of sensor and environment, d_e is viscosity of environment, and K_f is a forward gain. Fig. 15 shows that the system is oscillatory with natural angular frequency of $\sqrt{K_f k_e}$, when the damping of environment d_e is very small. Once the end-effector touches the environment, a closed loop in Fig. 15 is completed. Then the system is oscillatory and the end-effector is repulsed from the environment. When the end-effector separates from the environment, the closed loop in Fig. 15 is eliminated. Again the end-effector is stable and approaches the environment and touches it again. This process repeats over and over.

This hunting phenomena is overcome by adding damping loop as shown in Fig. 16. Generally it is difficult to know the stiffness and the damping of the environment *a priori.*

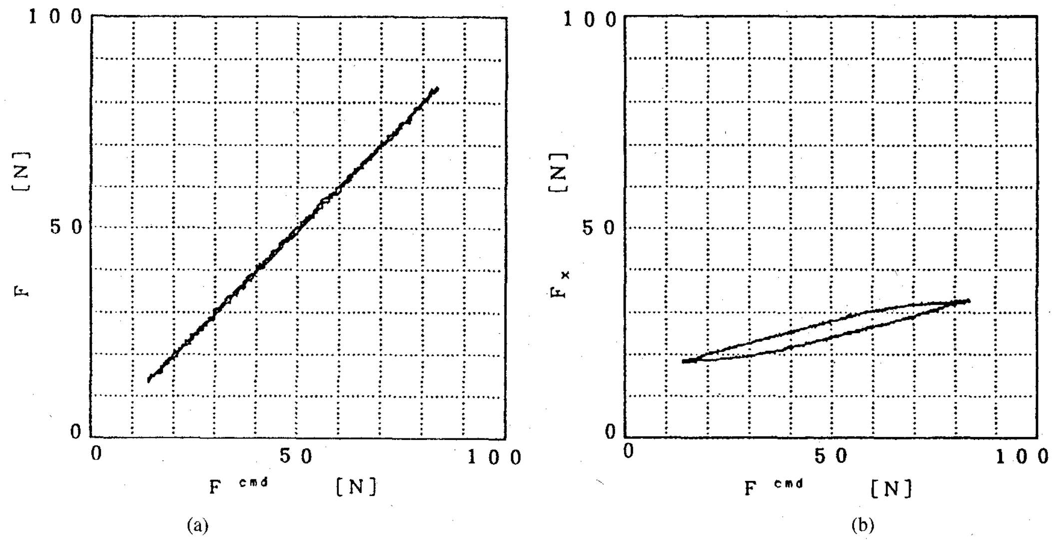


Fig. 18. Static characteristics of force response in force control.

An inserted forward gain K_f should be chosen so that the total system is stabilizable. The transfer function from the equivalent disturbance acceleration $(-p)$ to the position is as follows:

$$\frac{\theta}{(-p)} = \frac{1}{s^2 + K_f(K_v + d_e)s + K_f k_e}. \tag{30}$$

A virtual stiffness κ_f is determined as follows:

$$\begin{aligned}\kappa_f &= \frac{\partial(-p)}{\partial\theta} \\ &= K_f k_e.\end{aligned} \tag{31}$$

This gives a performance limit of force control due to the very small displacement to measure the force. When the force control approaches to the ideal one by reducing stiffness, the response will be slow.

C. Impedance Control

The stiffness of the system is modified to have a specified mechanical impedance. In this case, position and force signal are used to generate acceleration reference based on the specified impedance. Fig. 17 shows an example of such impedance control system. In Fig. 17, the stiffness corresponding to the virtual spring coefficient, the artificial damping and the equivalent mass realize a mechanical impedance.

It is noted that if the gain of the position is zero, the impedance control becomes the force control. The zero gain of the force is the same as the position control. It is possible to turn continuously the motion control to both the position control and the force control by adjusting the control gains in impedance control. In other words, the impedance control is the general form of motion control [12].

D. Experimental Examples of Force and Position Control

The experimental examples of force and position control are shown in Figs. 18 and 19. These results show that the disturbance is well compensated by the disturbance observer and the desired motion responses are obtained in force and position control.

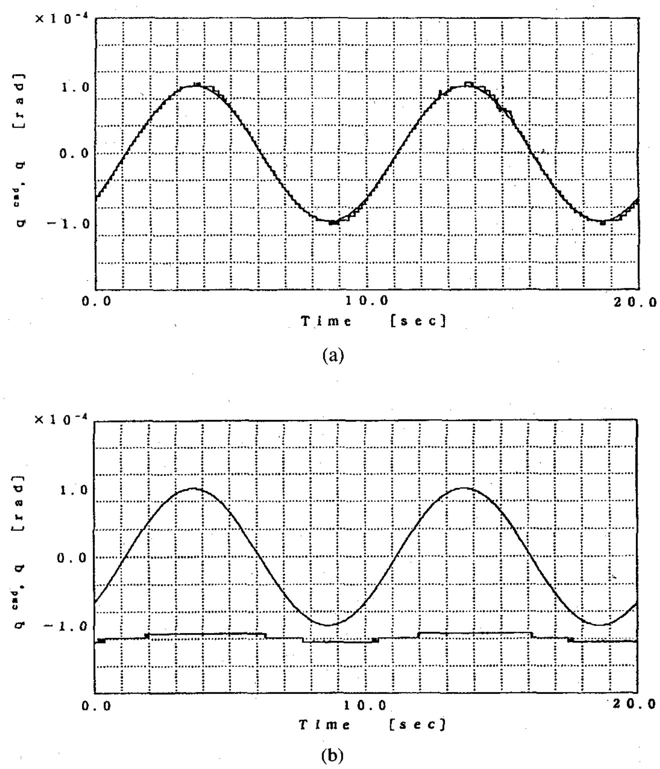


Fig. 19. Precise positioning control.

V. Parameter Identification in Mechanical System

In the above discussion, the disturbance estimated by (18) is used for a realization of robust mechanical system. In the actual application, the estimated disturbance is effective for not only the disturbance compensation but also the parameter identification in the mechanical system. As defined in (15), the

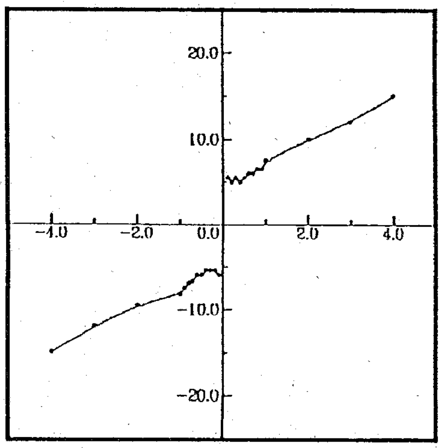


Fig. 20. An example of identified friction effect.

equivalent disturbance $\tilde{d}$, which is estimated by the disturbance observer, includes the load torque T_l and the parameter variation torque $[(K_t/J)-(K_{t0}/J_0)]\,I_a^{ref}$. Here the load torque T_l consists of friction and external force effects in the mechanical system as follows:

$$T_l = \underbrace{T_c^{friction} + T_v^{friction}\omega}_{\text{coulomb and viscosity friction effect}} + \underbrace{T_{ext}}_{\text{external force effect}}. \quad (32)$$

This equation means that the output of the disturbance observer is only the friction effect under the constant angular velocity motion. This feature makes it possible to identify the friction effect in the mechanical system. Fig. 20 shows an example of the identified friction effect. In Fig. 20, the friction effects are well identified as Stribeck friction model [13].

The external force effect is also identified by using the estimated disturbance. Here it is assumed that the friction effects are known beforehand by the above identification process. By implementing the angular accelerated motion, the system parameter K_{t0}/J_0 is adjusted in the observer design so that it is close to the actual value K_t/J. As a result, the disturbance observer estimates only the external force effect as follows:

$$\begin{aligned}\hat{\tilde{d}} &= \frac{k_1}{s^2+k_2s+k_1}\,\tilde{d}\,|_{K_{t0}/J_0 \to K_t/J_0} \\ &= \frac{k_1}{s^2+k_2s+k_1}\,\frac{T_{external}}{J}.\end{aligned} \quad (33)$$

The identification process of the external force is summarized in Fig. 21. The identified external force is applicable to sensorless force feedback control in mechanical system [14] and is utilized for a realization of mechanical vibration control as shown in the next section.

VI. Vibration Control of Mechanical Resonance System

As described before, the progress of robust control technologies makes it possible to realize high performance motion control. In the industrial drive system such as a steel rolling mill system and so on, however, the developed technology is not enough to obtain the stable and high speed motion response

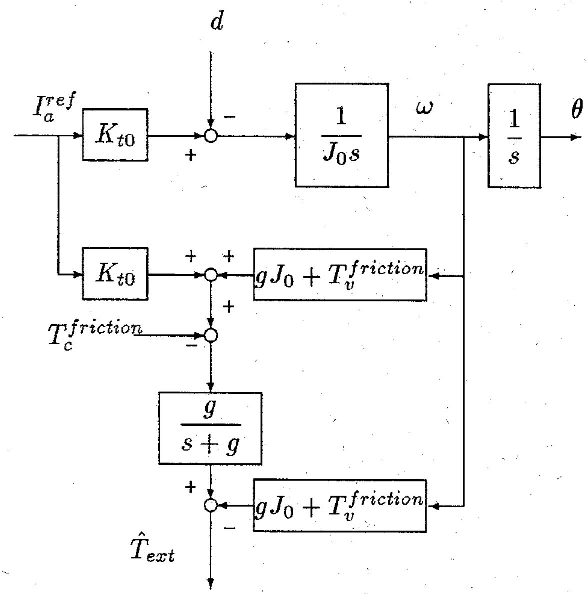
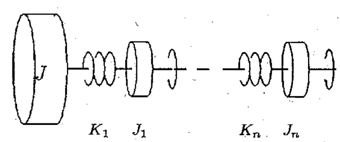
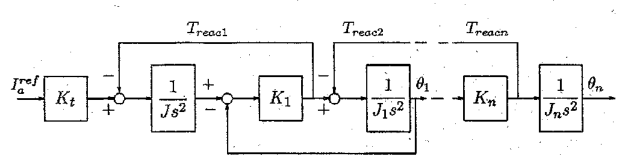


Fig. 21. Identification process of external force.



(a)



K_f: Equivalent Total Stiffness of Load side

$J_a = J_1 + J_2 + \cdots + J_n$

(b)

Fig. 22. A model of mechanical resonance system.

since the mechanical vibration arises under the high accuracy positioning control. To address above issue, the mechanical vibration control is also taken up in the field of the motion control [15]–[17]. In particular, a vibration control based on the external force feedback brings the sophisticated advantages to the mechatronic system. This section introduces a vibration control strategy based on the external force feedback called "resonance ratio control" in multiple resonance system. In this case, the external force may be obtained by using the identification process shown in Fig. 21.

A. A Dynamical Model of Mechanical Resonance System

In general, the dynamical behavior of the mechanical resonance system is described as multiple mass spring model. Fig. 22(a) shows a schematic illustration of the multiple mass spring system and Fig. 22(b) is a block diagram.

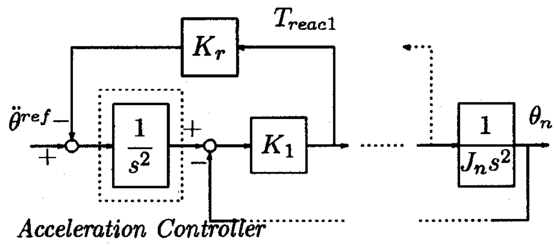


Fig. 23. Acceleration controller based on external force feedback.

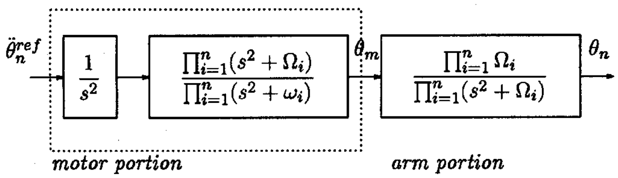


Fig. 24. Equivalent transformation of Fig. 23.

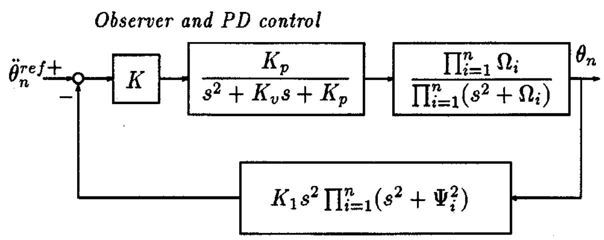


Fig. 25. Total block diagram of vibration suppression controller.

In the vibration control, the disturbance effect imposed on the motor portion is suppressed by applying the robust control technique, which is based on the disturbance observer in this section. Then, the motion system seems an acceleration controller. Furthermore, the identified external force is fed back through the feedback gain K_r. Fig. 23 shows the total block diagram of the acceleration controller based on the external force feedback. Fig. 23 is transformed into Fig. 24 without any approximation. In the latter discussion, Fig. 24 is used for the analysis and the design of the vibration control.

B. Stability Discussion

In Fig. 24, the following issues are considered to obtain the vibration suppression controller.

- The controller of the motor portion is designed so that the poles of the system do not cancel the zeros by the motor state feedback.
- The feedforward compensator is designed so that the location of the zeros is not changed.

In the vibration controller based on the external force feedback, PD control is applied to the motor position controller and the external force feedback gain is determined so that the above conditions are satisfied. To ensure the effectiveness of the external force feedback, the system stability is analyzed. In case PD control is applied to the motor portion of Fig. 24, the total block diagram of the system is rewritten as shown in Fig. 25. Fig. 26 shows the root loci of Fig. 25.

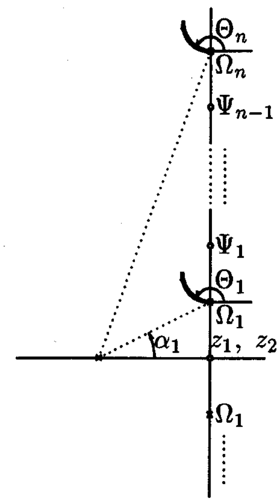


Fig. 26. Root locus of Fig. 25.

From Fig. 26, starting angle of each oscillation pole θ_i is obtained as follows:

$$\begin{aligned}\Theta_1 &= 270 - 2\alpha_1\\ \Theta_2 &= 270 - 2\alpha_2\\ &\vdots\\ \Theta_{n-1} &= 270 - 2\alpha_{n-1}\end{aligned} \tag{34}$$

$$\begin{aligned}0 &\le \alpha_j \le 90\\ 90 &\le \Theta_j \le 270.\end{aligned} \tag{35}$$

The above equations mean that the controller based on the external force feedback makes the oscillation poles stable. This is a basic concept of the proposed approach to obtain the stable motion response in the mechanical resonance system. In the actual design of the controller, only the first oscillation pole is considered to construct the vibration suppression controller. Then the controller gains K_p, K_v, and K_r are determined according to the resonance ratio which shows the ratio of the natural frequency of the motor side and the load side. The vibration control strategy based on the resonance ratio is called "resonance ratio control."

C. Resonance Ratio Control

As described before, all pole-loci of the mechanical resonance system move to the stable direction by the external force feedback. In the next step, the controller gains are determined according to the resonance ratio. Here it is assumed that the dominant oscillation pole of the mechanical system is the first oscillation pole. Then the transfer function of the system is described as follows:

$$\begin{aligned}\theta_m &= \frac{s^2+\omega_a^2}{\omega_a^2} G_1(s)G_2(s)\theta^{cmd}\\ \theta_a &= G_1(s)G_2(s)\theta^{cmd}\\ \omega_a &= \sqrt{\frac{K_f}{J_a}}\end{aligned}$$

$$
\begin{aligned}
\omega_m &= \sqrt{\frac{K_f}{J_a}\,(1 + K_r J_a)} \\
&= K\omega_a \\
K &= \sqrt{1 + K_r J_a}.
\end{aligned} \tag{36}
$$

Here ω_a and J_a are the equivalent frequency and inertia of the load side in Fig. 23. ω_m and K_f is the natural frequency of the motor side and the equivalent stiffness of the load side, respectively. K is the resonance ratio. The denominator $D(s)$ of the transfer function of $G_1(s)G_2(s)$ is given as follows:

$$D(s) = s^4 + K_v s^3 + (K_p + \omega_m^2)s^2 + K_v\omega_a^2 s + K_p\omega_a^2. \tag{37}$$

To simplify the controller design, $G_1(s)$ and $G_2(s)$ are defined as second order system and ζ_1, ω_1, ζ_2, and ω_2 are introduced to describe the motion performance in each system. Then $D(s)$ is also given as follows:

$$D(s) = (s^2 + 2\zeta_1\omega_1 s + \omega_1^2)(s^2 + 2\zeta_2\omega_2 s + \omega_2^2). \tag{38}$$

From (37) and (38), the following relations are obtained:

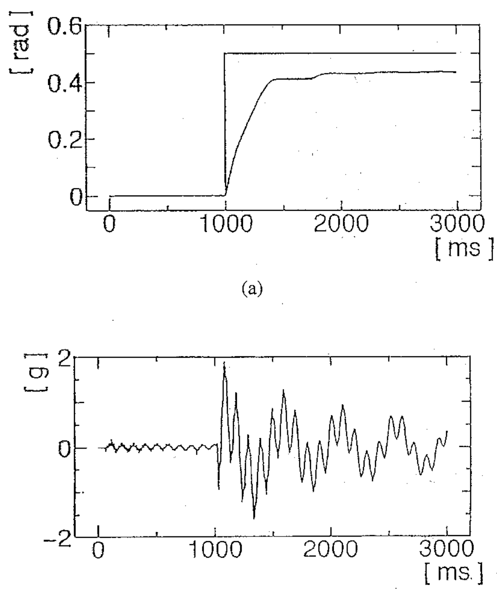
$$
\begin{aligned}
K_v &= 2(\zeta_1\omega_1 + \zeta_2\omega_2) \\
K_p &= \frac{\omega_1^2\omega_2^2}{\omega_a^2} \\
\omega_m &= \sqrt{-\frac{\omega_1^2\omega_2^2}{\omega_a^2} + \omega_1^2 + \omega_2^2 + 4\zeta_1\zeta_2\omega_1\omega_2}.
\end{aligned} \tag{39}
$$

The important goal in the vibration control is to suppress the vibration, so that $\zeta_1 = \zeta_2 = 1.0$ in (39). Also $\omega_1 = \omega_2 = \omega_a$ to obtain the high speed motion response in the load side. Finally, the following control gains are obtained with resonance ratio of $\sqrt{5}$

$$
\begin{cases}
K_r = \dfrac{4}{J_a} \\
K_p = \omega_a^2 \\
K_v = 4\omega_a
\end{cases}. \tag{40}
$$

D. Experimental Examples of Vibration Control

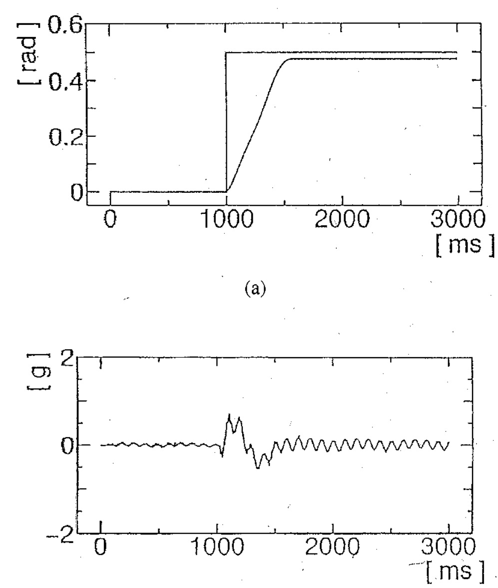
By using a set of the gains shown in (40), the vibration of the mechanical resonance system is well suppressed. Figs. 27 and 28 are the experimental results of PD control and resonance ratio control, respectively. These results clearly show that the resonance ratio control is effective for the vibration suppression in the mechanical resonance system.



(a)

(b)

Fig. 27. PD control in mechanical resonance system.



(a)

(b)

Fig. 28. Resonance ratio control in mechanical resonance system.

VII. Conclusions

The paper intends to give a tutorial of motion control technology in mechatronics. The robustness of the motion control makes the system more flexible. The stiffness of the motion, which correspond with the forward gain of the position, is defined to be a good index of robustness. The motion controller acquires robustness by estimating disturbance. Position control and force control are also discussed in relation to the robustness and the stiffness. The robustness and the identification is both sides of a motion control each other. The recent modern technique including two-degrees-of freedom control, H^∞ control has proved the same structure from physical point of view [18].

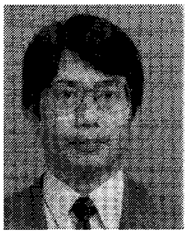
The estimated disturbance includes reaction force from the environment. The information is used for estimation of mechanical parameters. By direct use of reaction force, an antivibration control called a resonance ratio control for flexible structure is realized.

The paper little describes the reference generator. An intelligence in the reference generator is another key for intelligent mechatronics, however, it presuppose the robustness of the motion controller. From such a point of view, the roll of robust motion controller will be more important in mechatronics.

The further development, particularly in the connection of reference generator and controller of motion, will be expected.

REFERENCES

[1] Yaskawa Electric Co., Registered Trade Mark no. 46-32713, 1971.
[2] K. Ohnishi, Ed., in *Proc. Int. Workshop on Advanced Motion Contr.*, Yokohama, Apr. 1990.
[3] A. De Carli, Ed., in *Preprints, IFAC Workshop on Motion Contr. Intell. Automat.*, Perugia, Oct. 1992.
[4] S. Okuma, Ed., in *Proc. 2nd Int. Workshop on Advanced Motion Contr.*, Nagoya, Mar. 1992.
[5] H. Kazerooni and Y. Hori, Eds., in *Proc. 3rd Int. Workshop on Advanced Motion Contr.*, Berkeley, Mar. 1994.
[6] F. Pfeiffer and G. Brandenburg, Ed., in *Proc 3rd IFAC Workshop on Motion Contr.*, Munich, Oct. 1995.
[7] K. Ohishi *et al.*, Eds., *J. Robot. Soc. Jpn.*, Special Issue on Advanced Motion Control, vol. 11, no. 4, 1993.
[8] W. Leonhard, *Control of Electrical Drives.* Berlin: Springer-Verlag, 1985.
[9] J. Meditch and G. H. Hostetter, "Observers for systems with unknown and inaccessible inputs," *Int. J. Contr.*, vol. 16, no. 3, pp. 473–480, 1974.
[10] B. Gopinath, "On the control of linear multiple input-output systems," *Bell System Tech. J.*, vol. 50, no. 3, pp. 1063–1081, 1971.
[11] J. Luh, M. Walker, and R. Paul, "Resolved-acceleration control of mechanical manipulators," *IEEE Trans. Automat. Contr.*, vol. 28, no. 3, pp. 468–474, 1980.
[12] O. Khatib, "A unified approach for motion and force control of robotic manipulators," *IEEE J. Robot., Automat.*, vol. RA-3, no. 1, pp. 43–53, 1987.
[13] S. Southward, C. Radcliffe, and C. MacCluer, "Robust nonlinear stick-slip friction compensation," *ASME J. Dynamics Syst., Measurement, Contr.*, vol. 113, pp. 639–645, 1991.
[14] T. Murakami and K. Ohnishi, "Torque sensorless control in multi-degree-of-freedom manipulator" *IEEE Trans. Ind. Electron.*, vol. 40, no. 2, 1993.
[15] T. Murakami *et al.*, Eds., *J. Robot. Soc. Jpn.*, Special Issue on Advanced Motion Control in Vibration System, vol. 13, no. 8, 1995.
[16] K. Seto, K. Yoshida, and K. Nonami, Eds., in *Proc. JSME First Int. Conf. Motion, Vibration Contr.*, Yokohama, 1992.
[17] K. Yoshida and K. Nonami, Eds., in *Proc. 2nd Int. Conf. Motion, Vibration Contr.*, Yokohama, 1994.
[18] G. Buja, H. Fujita, and K. Ohnishi, Eds., *Recent Advances in Motion Control.* The Nikkan Kogyo Shimbun Ltd., 1990.



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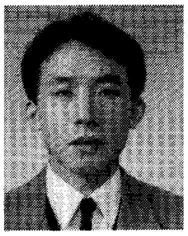

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