Optimization of Motion Regimes for Machines and Mechanisms With a Mehatronics' Control

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Summary. Motion control of machines and mechanisms with a mechatronics control is now recognized as a key technology in mechatronics. The robustness of such motion control will be represented as a function of stiffness and a basic for practical realization, for example, in the precise agriculture. Target of such 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 of machine/mechanism. Optimization of motion control of flexible structure and identification of mechanical parameters are also described.

Key words: optimization, motion, machines, mechatronics, control.

INTRODUCTION

One of the most important elements in mechatronic technology is undoubtedly motion control (of machine/mechanism). 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 junction of reliability, availability, and serviceability has been very much improved in relatively more compact products (and, for example, in the agricultural machines and mechanisms).

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 regimes. 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 later 1980's and the early 1990's, mechatronic seemed slow case of various applications of control theories and of optimization theories of control as well.

The phenomena observed in the early 1990's [2] also came from the so-called „software-servo technology". Generally major 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. Than the novel algorithms have gained high evaluation from the practical viewpoint because the quality of motion was improved (in precise agricultural technologies as well). The motion control is now recognized as an important area in mechatronics [3]–[6].

This paper intends to show recent advances in motion regimes of machines/mechanisms 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 aspect of system design. The fact holds in motion control of machines and mechanisms. The robust control and the estimation of parameters have the same basic. The several example shown later seen 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 of machines and mechanisms. 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.

MATHERIALS AND METHODS

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 constrains 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 motions 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.

From the control point of view, the output of the motions 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 is preferable, though, there are various candidates of motion representation. One of such indices is stiffness.

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

$$f = g(\ddot{x}, x, x) \quad (1)$$

The stiffness k is defined in the partial differentiation:

$$k = \frac{\partial f}{\partial x} \quad (2)$$

The ideal position control inhibits any deviation of position against any deviation of force. That means k 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, k 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 mechanical impedance computed in the controller according to the specified dynamics. Table 1 shows that k is a good parameter as an index which represents a target of motion.

Table 1. Stiffness as a motion index

target of motion	stiffness k
position	∞
compliant	finite
force	0

RESULTS AND DISCUSSION

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 path and go forward to the end of part, 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 for the utmost the "high-fidelity" to the input reference.

The dynamical equation is excited by input force. Most of mechatronic systems adopt electrical actuator for the purpose. Fig. 1 shows a typical electric drive system.

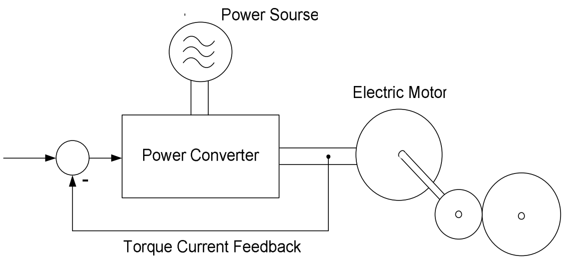


Fig. 1. A typical electric drive system

Most of power converters use switching device 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 electronic interference of current and magnetic flux.

There are three types of the interference as shown in Table 2. The stepping motor, which is not in Table 2, is widely used for simple positioning. It has similar characteristics of synchronous motor, however, it generates high torque ripple and is not appropriate for fine and smooth motion. Each motor in Table 2 generates torque be 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:

$$T_m = K_t \cdot I_a \, , \tag{3}$$

where: 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. 2.

Table 2. 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

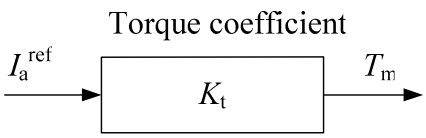


Fig. 2. Block diagram of actuator

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 $Y(s)$:

$$\frac{Y_{(s)}}{U_{(s)}} = K \cdot \frac{c_m \cdot s^m + c_{m-1} \cdot s^{m-1} + ... + c_2 s + c_1}{s_n + a_n \cdot s^{n-1} + a_{n-1} \cdot s^{n-2} + ... + a_2 s + a_1}. \quad (4)$$

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:

$$\begin{cases} \dot{x} = Ax + bu + e \cdot d \\ y = cx. \end{cases} \quad (5)$$

Here $x = \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{bmatrix}$, $A = \begin{bmatrix} 0 & 1 & 0 & \cdots & 0 \\ 0 & 0 & 1 & \cdots & 0 \\ \vdots & \vdots & \vdots & \cdots & \vdots \\ 0 & 0 & 0 & \cdots & 1 \\ -a_1 & -a_2 & -a_3 & \cdots & -a_n \end{bmatrix}$,

$b = \begin{bmatrix} 0 \\ 0 \\ \vdots \\ 0 \\ K \end{bmatrix}$, $e = \begin{bmatrix} 0 \\ 0 \\ \vdots \\ 0 \\ 1 \end{bmatrix}$, $c = [c_1 \quad c_2 \quad \dots \quad c_m \quad 0 \quad \dots \quad 0]$,

x is a state vector, A is a system matrix, b is a distribution vector of input, e is a distribution vector of disturbance, and c is an observation column vector. Equation (5) is represented in Fig. 3.

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

$$\begin{cases} A = A_0 + \Delta A, \\ b = b_0 + \Delta b. \end{cases} \quad (6)$$

where: ΔA is a variation of A and Δb is a variation of b.The variation of dynamic matrix 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):

$$\begin{aligned} \dot{x} &= (A_0 + \Delta A) \cdot x + (b_0 + \Delta b) \cdot u + e \cdot d = \\ &= A_0 \cdot x + b_0 \cdot u + (\Delta A \cdot x + \Delta b \cdot u + e \cdot d). \end{aligned} \quad (7)$$

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

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

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

$$\begin{cases} \dot{x} = A_0 \cdot x + b_0 \cdot u + e\tilde{d}, \\ y = cx. \end{cases} \quad (9)$$

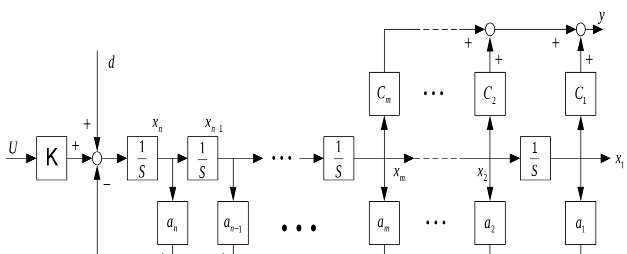


Fig. 3. Companion form of linear system

There are various proposals to estimate the disturbance. There 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:

$$\frac{d^{(p)}\tilde{d}}{dt^{p}} = 0. \qquad (10)$$

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

$$\begin{cases} \dot{\tilde{x}} = \tilde{A}_0 \tilde{x} + \tilde{b}_0 u \\ y = \tilde{c}_0 \tilde{x}. \end{cases} \qquad (11)$$

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

$$\tilde{x} = \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \\ \tilde{d} \\ \dot{\tilde{d}} \\ \ddot{\tilde{d}} \\ \vdots \\ \tilde{d}^{(p-1)} \end{bmatrix}, \tilde{A}_0, \tilde{b}_0, \tilde{c}_0 \text{ are as follows}$$

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

$$\tilde{b}_0 = \begin{bmatrix} 0 \\ 0 \\ \vdots \\ 0 \\ K_0 \\ 0 \\ \vdots \\ 0 \end{bmatrix} \Bigg\} (n+p), \quad \tilde{c}_0 = \overbrace{\begin{bmatrix} c_1 & c_2 & \dots & c_m & 0 & \dots & 0 \end{bmatrix}}^{(n+p)}.$$

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 observed 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 $\tilde{d}$, the input will be sum of two parts:

$$u = u^{ref} + u^{dis}. \qquad (12)$$

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, by the way, to optimize the system as well), the compensation input is made by using the estimated equivalent – disturbance:

$$u^{dis} = -(b_0^t b_0)^{-1} b_0^t e \hat{\tilde{d}} = -\frac{1}{K_0} \hat{\tilde{d}}. \qquad (13)$$

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 without any disturbance.

Fig.4 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.

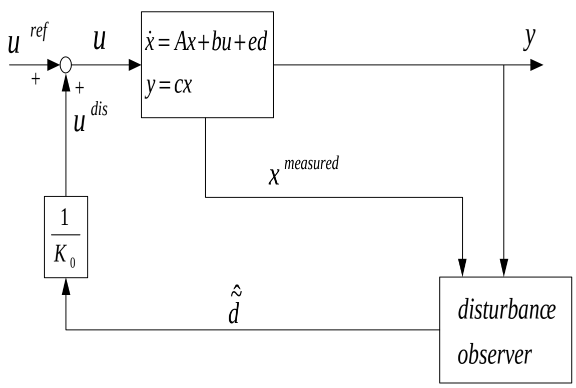


Fig. 4. Robust control based on disturbance observer

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. 5.

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

$$I\frac{dw}{dt} = K_t I_a^{ref} - T_l, \tag{14}$$

Here I – inertia; K_t – torque coefficient of electric motor; T_l – load torque.

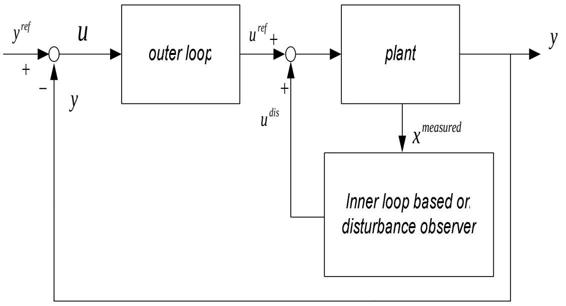


Fig. 5. Total system with robust control

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:

$$\tilde{d} = -\frac{T_l}{I} + \left(\frac{K_t}{I} - \frac{K_{t_0}}{I_0}\right) I_a^{ref}. \tag{15}$$

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

$$\frac{d}{dt}\begin{bmatrix}\theta \\ \omega \\ \tilde{d}\end{bmatrix} = \begin{bmatrix}0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & 0 & 0\end{bmatrix}\begin{bmatrix}\theta \\ \omega \\ \tilde{d}\end{bmatrix} + \begin{bmatrix}0 \\ K_{t_0}/I_0 \\ 0\end{bmatrix} I_a^{ref}. \tag{16}$$

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:

$$\frac{d}{dt}\begin{bmatrix}z_1 \\ z_2\end{bmatrix} = \begin{bmatrix}0 & -k_1 \\ 1 & -k_2\end{bmatrix}\begin{bmatrix}z_1 \\ z_2\end{bmatrix} + \begin{bmatrix}-k_1 k_2 \theta \\ (k_1 - k_2^2)\theta + \dfrac{K_{t_0}}{I_0} I_a^{ref}\end{bmatrix}. \tag{17}$$

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

$$\hat{\tilde{d}} = \frac{k_1}{(S^2 + k_2 S) + k_1}\left(S^2\theta - \frac{K_{t_0}}{I_0} I_a^{ref}\right) = \frac{k_1}{(S^2 + k_2 S) + k_1}\tilde{d}. \tag{18}$$

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

$$\alpha + \beta = -k_2, \quad \alpha\beta = k_1. \tag{19}$$

It is worthwhile reconsidering (19). 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:

$$I = I_0 + \Delta I, \tag{20}$$

$$K_t = K_{t_0} + \Delta K_t. \tag{21}$$

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

$$I_0\frac{dw}{dt} = K_{t_0} I_a^{ref} - \left(T_l + \Delta I\frac{dw}{dt} - \Delta K_t I_a^{ref}\right). \tag{22}$$

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

$$T_{dis} = T_l + \Delta I\frac{dw}{dt} - \Delta K_t I_a. \tag{23}$$

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

$$T_{dis} = I_0(-\tilde{d}). \tag{24}$$

T_{dis} contacts: 1) mechanical load($=T_l$),

2) varied self-inertia torque $\left[= \Delta I (\frac{dw}{dt}) \right]$,

3) 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. 6.

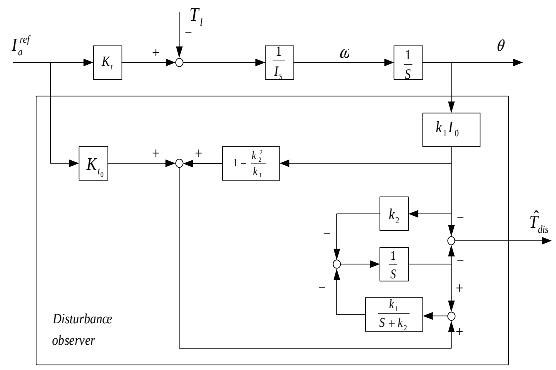


Fig. 6. Disturbance observed in motion system

According to the result of(12) and (13), compensation input is as follows:

$$I^{dis} = -\frac{I_0}{K_{t_0}} \hat{\hat{d}} = \frac{1}{K_{t_0}} \hat{T}_{dis} . \quad (25)$$

Robust motion controller has the schematic block diagram as shown in Fig.7.

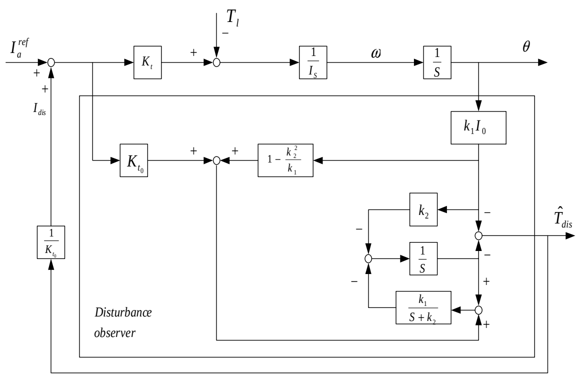


Fig. 7. A robust motion controller

There exists an integrator with high gain equivalently in the forward path as shown is Fig.8. Therefore, the robust motion controller eliminates steady state error.

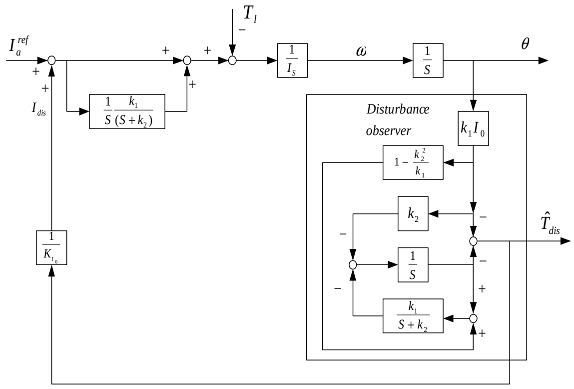


Fig. 8. Equivalent block diagram of Fig.7

Equation (18) shown that disturbance is estimated through low-pass filter. Generally there is such a low-pass filter in the observer structure. The poles of the observed determines the delay of the low-pass filter $G_T(s)$. $G_T(s)$ gives a certain effect to the control performance. Fig.6 is also transformed into Fig.9.

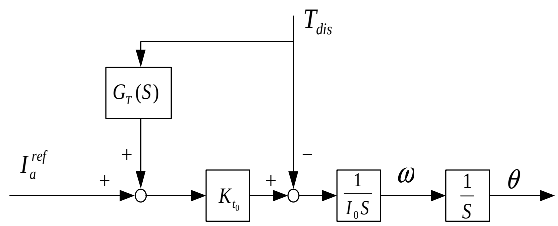


Fig. 9. A robust motion controller as an acceleration controller

Fig.9. transformed from Fig.6 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 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. 9 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 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, 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.

The disturbance estimated by (18) is used for a realization of robust mechanical system. In the actual application, the estimated disturbance is effective fir not only the disturbance compensation but also the parameter identification in the mechanical system. As defined in (15), the equivalent disturbance $\tilde{d}$, which is estimated by the disturbance observer, includes the load torque T_l and the parameter variation torque $[(K_t/I)-(K_{t_0}/I_0)]I_a^{ref}$. Here the load torque T_l consist of friction and external force effects in the mechanical system as follows:

$$T_l = \underbrace{T_c^{friction} + T_v^{friction}\omega}_{\substack{\text{Coulomb and viscosity}\\ \text{friction effect}}} + \underbrace{Text}_{\substack{\text{External}\\ \text{force effect}}}. \qquad (26)$$

This equation means that the output of the disturbance observed 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. 10 shows an example of the identified friction effect. In Fig.10, the friction effects are well identified as Stribeck friction model [11].

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 angural accelerated motion, the system parameter K_{t0}/I_0 is adjusted in the observer design so that it is close to the actual value K_t/I. As result, the disturbance observer estimates only the external force effect as follows:

$$\hat{\hat{d}} = \frac{k_1}{\left(s^2 + k_2 \cdot s + k_1\right)} \cdot \tilde{d}\Big|_{k_{to}/I_o \to K_t/I_o} = \frac{k_1}{\left(s^2 + k_2 \cdot s + k_1\right)} \cdot \frac{Texternal}{I_0}. \qquad (27)$$

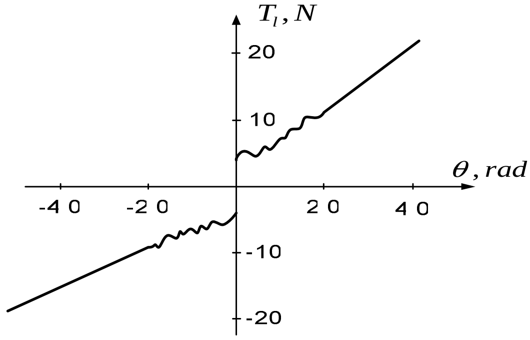


Fig. 10. An example of identified friction effect

The identification process of the external force is summarized in Fig. 11. The identified external force is applicable to senseless force feedback control in mechanical system [12] and is utilized for a realization of mechanical vibration control as shown in the next section.

As described before, the progress of robust control technologies makes it possible to realize high performance motion control of machines and mechanisms. In the industrial drive system such as a still rolling mill system and so on however , the development of technology is not enough to obtain the stable and high speed motion response since the mechanical vibration arises under the high accuracy positioning control. One may say such words about crane's systems as well. To address above issue, the mechanical vibration control is also taken up in the field of the motion control [13-15]. 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. 11.

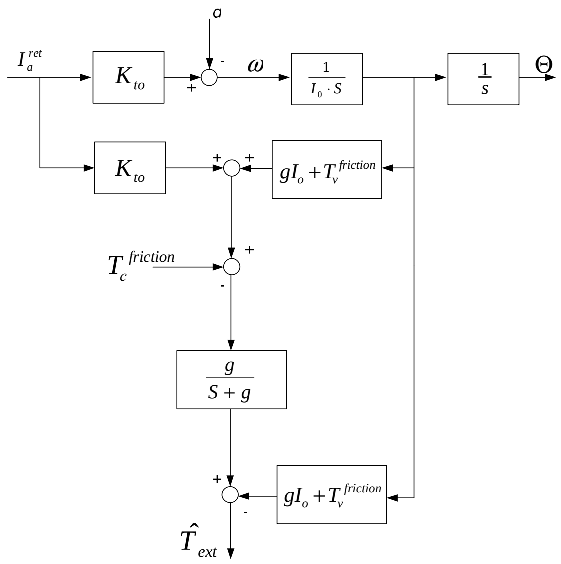


Fig. 11. Identification process of external force

In general, the dynamical behavior of the mechanical resonance system is described as multiple mass spring model. Fig. 12 (a) shows a schematic illustration of the multiple mass spring system and Fig. 12 (b) is a block diagram.

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. 13 shows the total block diagram of the acceleration controller based on the external force feedback. Fig. 13 is transformed into Fig. 14 without any approximation. In the latter discussion, Fig. 14 is used for the analysis and the design of the vibration control for various mechanical systems, machines and mechanisms.

In Fig. 14, 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.

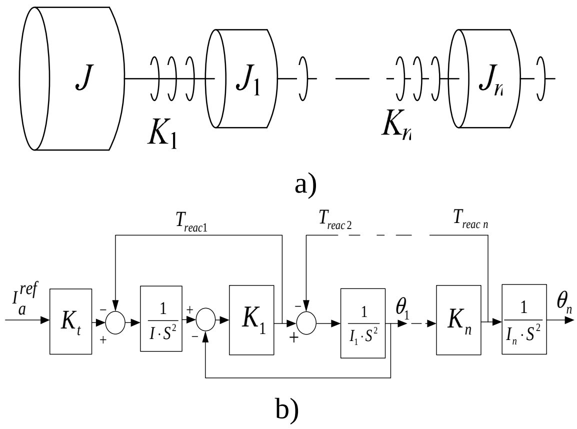


Fig. 12. A mode of mechanical resonance system K_f: Equivalent Total Stiffness of Load side;

$$I_a = I_1 + I_2 + \ldots + I_n$$

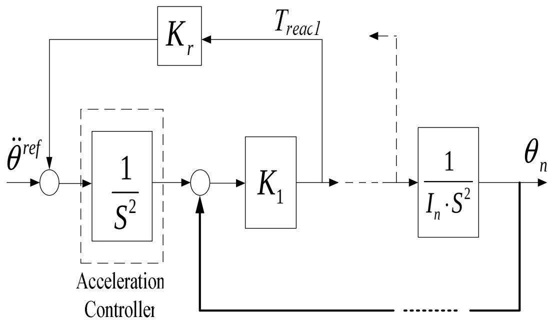


Fig. 13. Acceleration controller based on external force feedback

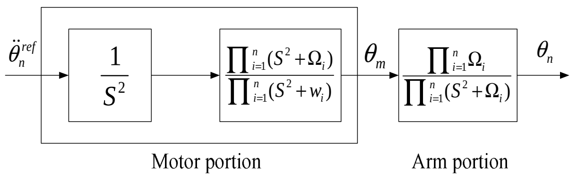


Fig. 14. Equivalent transformation of Fig. 13

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. 14, the total block diagram of the mechanical system is rewritten as slows in Fig. 15. Fig. 16 shows the root loci of Fig. 15.

From Fig.16, starting angle of each oscillation pole θ_i is obtained as follows:

$$\theta = 270° - 2\alpha_1;\ \theta_2 = 270° - 2\alpha_2, .. \theta_{n-1} = 270° - 2\alpha_{n-1}, \quad (28)$$

$$\theta \le \alpha_j \le 90°; \quad 90° \le \theta_j \le 270°. \quad (29)$$

The above equations mean that the controller based on the external force feedback mares 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 (for example, for the crane's system). Then the controller gains K_p, K_v, 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" for machines, mechanisms and other mechanical systems.

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 asumed 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:

$$\begin{gathered} \theta_m = \frac{\left(s^2 + w_a^2\right)}{w_a^2} \cdot G_1(s) \cdot G_2(s) \cdot \theta^{cmd}, \\ \theta_a = G_1(s) \cdot G_2(s) \cdot \theta^{cmd};\ w_a = \sqrt{\frac{K_f}{I_a}}, \\ w_m = \sqrt{\frac{K_f}{I_a} \cdot (1 + K_r \cdot I_a)} = K \cdot w_a;\ \ K = \sqrt{1 + K_r \cdot I_a}. \end{gathered} \quad (30)$$

Here w_a and I_a are the equivalent frequency and inertia of the load side in Fig. 13. w_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 $G_1(s) \cdot G_1(s)$ is given as follows:

$$\begin{gathered} D(s) = s^4 + K_v \cdot s^3 + \left(K_p + w_m^2\right) \cdot s^2 + \\ + K_v \cdot w_a^2 \cdot s + K_p \cdot w_a^2. \end{gathered} \quad (31)$$

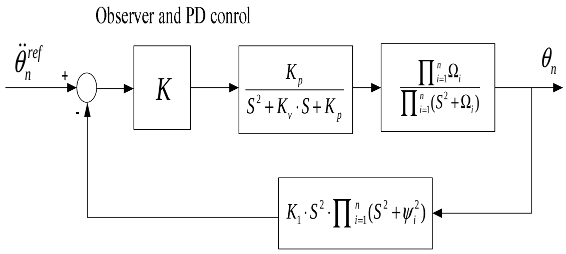


Fig. 15. Total block diagram of vibration suppression controller

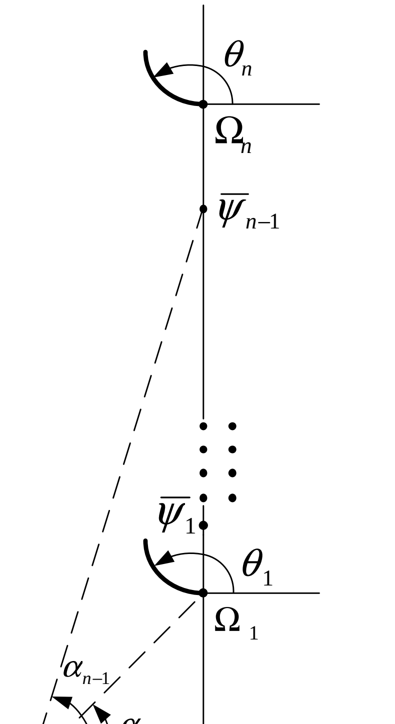


Fig. 16. Root locus of Fig. 15

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

$$\begin{gathered} D(s) = (s^2 + 2\zeta_1 \cdot \omega_1 \cdot s + \omega_1^2) \times \\ \times (s^2 + 2\zeta_2 \cdot \omega_2 \cdot s + \omega_2^2). \end{gathered} \quad (32)$$

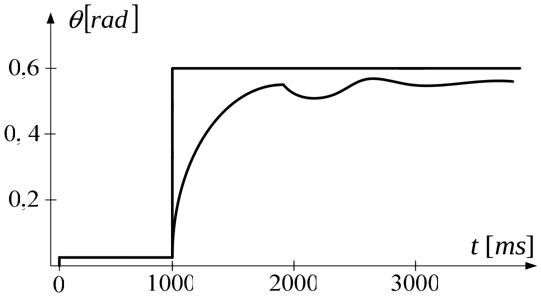
From (31) and (32), the following relations are obtained:

$$K_v = 2(\zeta_1 \cdot \omega_1 + \zeta_2 \cdot \omega_2), \quad K_p = \frac{\omega_1^2 \cdot \omega_2^2}{\omega_a^2},$$
$$\omega_m = \sqrt{-\frac{\omega_1^2 \cdot \omega_2^2}{\omega_a^2} + \omega_1^2 + \omega_2^2 + 4\zeta_1 \cdot \zeta_2 \cdot \omega_1 \cdot \omega_2}. \qquad (33)$$

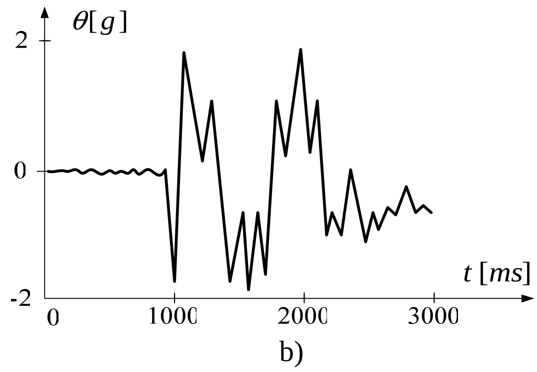
The important goal in the vibration control is to suppress the vibration, so that $\zeta_1 = \zeta_2 = 1{,}0$ in (33). Also $\omega_1 = \omega_2 = \omega_a$ to obtain the high speed motion response in the load side. Finally, the following control gains are obtained with resonance ratio of $\sqrt{5}$:

$$\begin{cases} K_r = 4 / I_a \\ K_p = w_a^2, \\ K_v = 4 w_a. \end{cases} \qquad (34)$$

By using a set of the gains shown in (34), the vibration of the mechanical resonance system is well suppressed. Fig. 17 and 18 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, various machine and mechanisms as well.



a)



b)

Fig. 17. PD control in mechanical resonance system

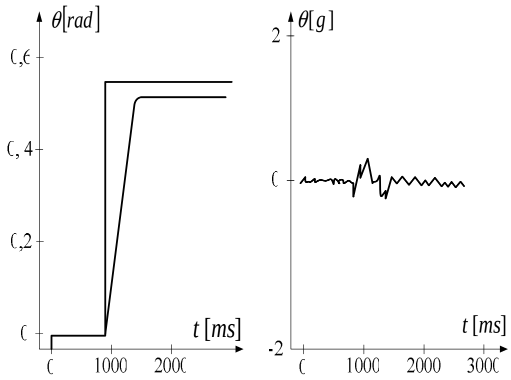


Fig. 18. Resonance ratio control in mechanical resonance system

CONCLUSIONS

1. The paper intends to give a tutorial of motion control technology in mechatronics and in various machines and mechanisms as well. The robustness of the motion control makes the mechanical 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. The robustness and the identification is both sides of an optimal control each other. The resent – modern technique including two-degrees-of freedom (optimal) control, H^∞ - control has proved the same structure from physical point of view [16]. 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 (optimal) control called a resonance ratio control for flexible structure is realized.

2. The paper little describes the reference generator. An intelligence in the reference generator is another key for intelligent mechatronics for mechanical systems, machines and mechanisms, however, it presuppose the robustness of the motion controller (for example, for crane's systems). From such point of view the role of robust motion controller will be more important in modern mechatronics.

3. The further development, particularly in the connection of reference generator and controller of motion for crane's systems, for agricultural machine and mechanisms, for various mechanical (resonance) systems, will be expected.

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ОПТИМИЗАЦИЯ РЕЖИМОВ ДВИЖЕНИЯ МАШИН И МЕХАНИЗМОВ С МЕХАТРОННЫМ УПРАВЛЕНИЕМ

Аннотация. Управление движением машин и механизмов в настоящее время признается в качестве ключевой технологии в мехатронике. Робастность управления движением представлена в виде функции от жесткости и есть основной для практической реализации, например, в точном сельском хозяйстве. Цель движения параметризирована жесткостью управления, которое, в соответствии с задачей, может быть переменной величиной. Однако, для достижения робастности движущейся системы необходима очень высокая жесткость управления. В работе показано, что управление ускорением системы реализуется одновременно с сохранением высокой робастности. Ускорение является переходом от робастности к переменной жесткости. Для практических применений введена методика оценки помех, что позволяет управляющему устройству движения управлять ускорением машины/механизма. Также описаны оптимизация управления движением гибкой структуры и идентификация механических параметров системы.

Ключевые слова: оптимизация, движение, машины, мехатроника, управление.