

ROBUST MECHANICAL SYSTEMS WITH MECHATRONIC DEVICES: PARAMETERS IDENTIFICATION AND VIBRATION CONTROL

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Abstract. The disturbance estimated with the help of advanced mechatronic devices is used for realization of robust mechanical system, such as agricultural machines. In the actual application, the estimated disturbance is effective not only for the disturbance compensation but also for the parameter identification in the mechanical system. The identified external force is applicable to sensorless force feed-back control in mechanical system and is utilized for a realization of mechanical vibration control. The progress of robust control technologies makes it possible to realize high performance motion control. In the industrial drive system such as a modern agricultural machine with advanced mechatronic device system for motion control, however, the developed technology is not enough to obtain the stable and high speed motion response since the mechanical vibration arises under the high accuracy positioning control. The mechanical vibration control is also taken in the field of the motion control. In particular, a vibration control based on the external force feedback brings the sophisticated advantages to the mechatronic system. The paper 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.

Introduction

The equivalent disturbance of the motion system described by (1):

$$I \cdot \frac{d\omega}{dt} = K_t \cdot I_a^{ref} - T_l, \quad (1)$$

where I – inertia; K_t – torque coefficient of electric motor; T_l – load torque, may be obtained as 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 [1]. The equivalent disturbance is

$$\tilde{d} = -\frac{T_l}{I} + \left(\frac{K_t}{I} - \frac{K_{to}}{I_o} \right) \cdot I_a^{ref}, \quad (2)$$

where I_a^{ref} – torque current reference, $I = I_o + \Delta I$, $K_t = K_{to} + \Delta K_t$. (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).

Suppose the first derivative of \tilde{d} is zero. An augmented state equation is

$$\frac{d}{dt} \begin{bmatrix} \theta \\ \omega \\ \hat{d} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & 0 & 0 \end{bmatrix} \cdot \begin{bmatrix} \theta \\ \omega \\ \hat{d} \end{bmatrix} + \begin{bmatrix} 0 \\ K_{to}/I_o \\ 0 \end{bmatrix} \cdot I_a^{ref}. \quad (3)$$

Here θ – angle (position of motor’s rotor), $\omega = \frac{d\theta}{dt}$, t – time.

By Gopinath’s method [2], the following estimation process is obtained:

$$\hat{\dot{d}} = k_1 \cdot \theta + z_1. \quad (4)$$

z_1 should satisfy (5), where k_1 and k_2 are free parameters:

$$\frac{d}{dt} \begin{bmatrix} z_1 \\ z_2 \end{bmatrix} = \begin{bmatrix} 0 & -k_1 \\ 1 & -k_2 \end{bmatrix} \cdot \begin{bmatrix} z_1 \\ z_2 \end{bmatrix} + \begin{bmatrix} -k_1 \cdot k_2 \cdot q \\ (k_1 - k_2^2) \cdot q + \frac{K_{to}}{I_0} \cdot I_a^{ref} \end{bmatrix}. \quad (5)$$

Equations (3) and (5) lead (6):

$$\hat{d}^0 = \frac{k_1}{(s^2 + k_2 \cdot s + k_1)} \cdot \left(s^2 \cdot \theta - \frac{K_{to}}{I_0} \cdot I_a^{ref} \right) = \frac{k_1}{(s^2 + k_2 \cdot s + k_1)} \cdot \tilde{d}^0 \quad (6)$$

Here \tilde{d} is obtained with the help of equation (2) and $s \equiv \frac{d}{dt}$.

Problem statement

The disturbance estimated by (6) 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 (2), the equivalent disturbance \tilde{d} , which is estimated by the disturbance observer, includes the load torque Tl and the parameter variation torque $\left[\left(\frac{K_t}{I} \right) - \left(\frac{K_{to}}{I_0} \right) \right] \cdot I_a^{ref}$.

Analysis of modern information sources on the subject of the article

The progress of robust control technologies makes it possible to realize high performance motion control. In the industrial drive system such as modern agricultural machine with advanced mechatronic devices and so on, however, the developed technology is not enough to obtain the stable and high speed motion response 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 [1-25]. In particular, a vibration control based on the external force feedback brings the sophisticated advantages to the mechatronic system. A vibration control strategy based on the external force feedback called "resonance ration control" in multiple resonance system is proposed below. In this case, the external force may be obtained by using the identification process shown in Fig. 2.

Statement of purpose and problems of research

In the industrial drive system such as a modern agricultural machine with advanced mechatronic device's system for motion control, however, the developed technology is not enough to obtain the stable and high speed motion response since the mechanical vibration arises under the high accuracy positioning control. The mechanical vibration control is also taken up in the field of the motion control. In particular, a vibration control based on the external force feedback brings the sophisticated advantages to the mechatronic system. The paper 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.

Parameter identification in mechanical system

The load torque Tl consists of friction and external force effects in the mechanical system as follows:

$$T_l = \underbrace{T_{coulomb} + T_{viscosity}}_{\text{coulomb and viscosity friction effect}} + \underbrace{T_{ext}}_{\text{external force effect}}. \quad (7)$$

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 function effect in the

mechanical system. For example, the friction effects are well identified as Stribeck friction model [3]. Fig. 1 shows an example of the identified friction effect with the help of this model.

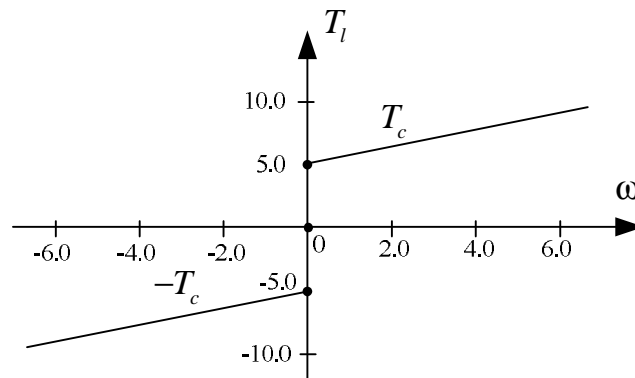


Fig. 1. An example of identified friction effect

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}/I_0 is adjusted in the observer design so that it is close to the actual value K_t/I . As a result, the disturbance observer estimates only the external force effect as follows:

$$\hat{d}^{\omega} = \frac{k_1}{(s^2 + k_2 \cdot s + k_1)} \cdot d^{\omega}_{K_{t0}/I_0 \rightarrow K_t/I} = \frac{k_1}{(s^2 + k_2 \cdot s + k_1)} \cdot \frac{T_{external}}{I} \quad (8)$$

The identification process of the external force is summarized in Fig. 2. The identified external force is applicable to sensor less force feedback control in mechanical system [4] and is utilized for a realization of mechanical vibration control as shown in the next section.

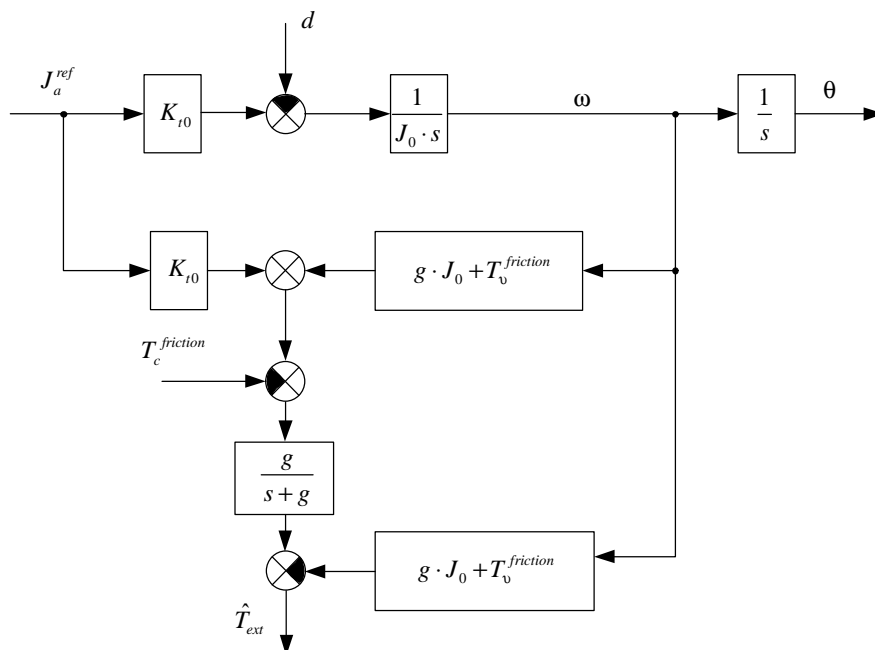


Fig. 2. Identification process of external force

Vibration control of mechanical resonance system

Dynamical Model of Mechanical Resonance System

In general, the dynamical behavior of the mechanical resonance system is described as multiple mass spring models. Fig. 3 (a) shows a schematic illustration of the multiple mass spring system and Fig. 3 (b) is a block diagram.

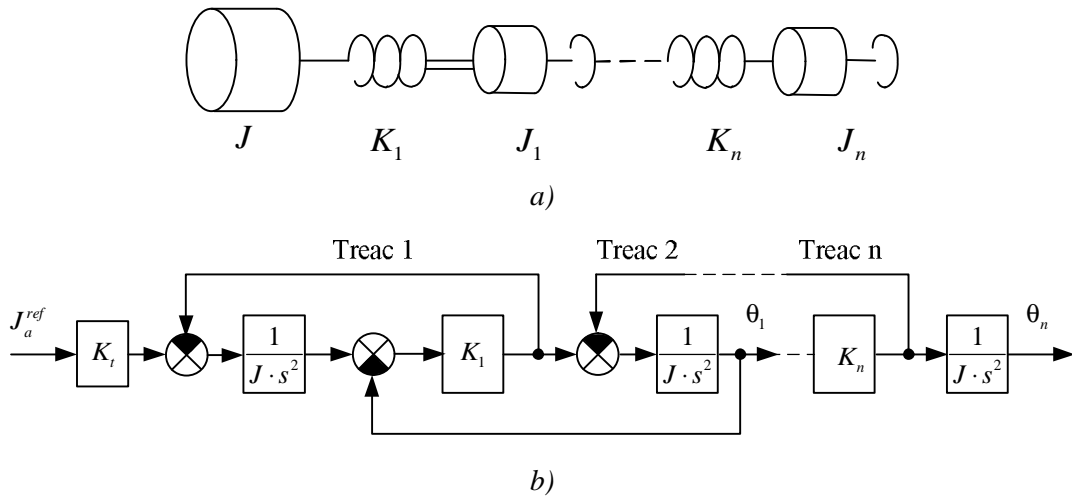


Fig. 3. A model of mechanical resonance system

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. 4 shows the total block diagram of the acceleration controller based on the external force feedback. Fig. 4 is transformed into Fig. 5 without any approximation. In the latter discussion, Fig. 5 is used for the analysis and the design of the vibration control.

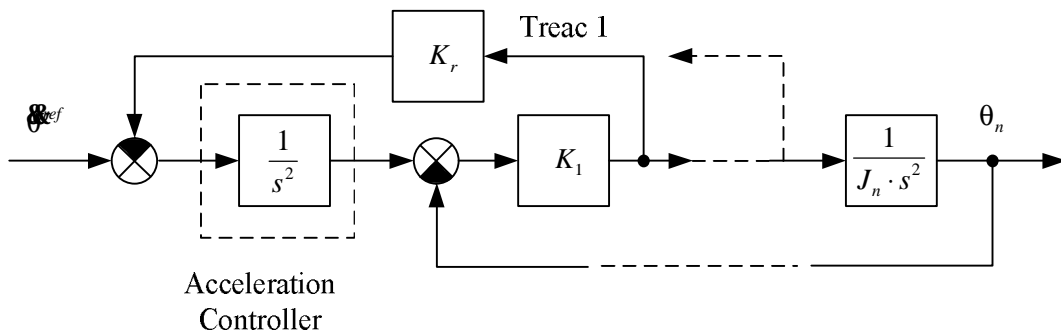


Fig. 4. Acceleration controller based on external force feedback

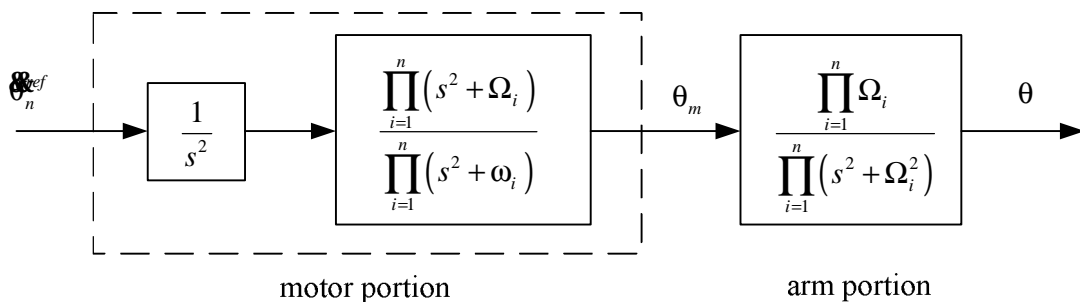


Fig. 5. Equivalent transformation of Fig. 4

Stability Discussion

In Fig. 5, 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 change.

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. 5, the total block diagram of the system is rewritten as shown in Fig. 6 Fig. 7 shows the root loci of Fig. 6.

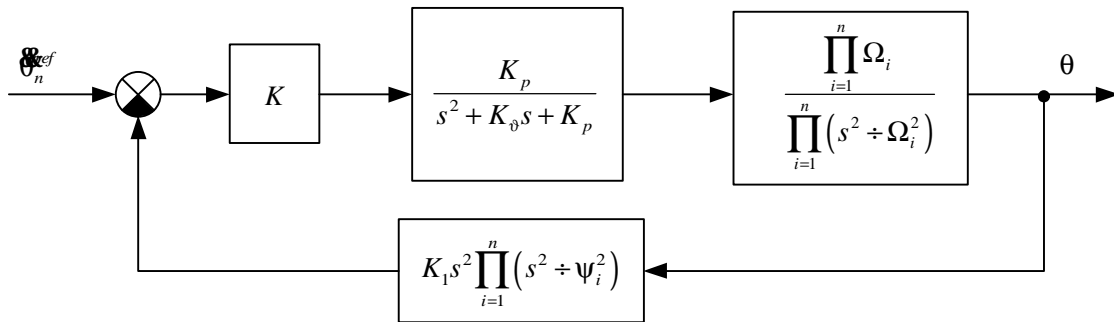


Fig. 6. Total block diagram of vibration suppression controller

From Fig. 7, starting angle of each oscillation pole θ_i is obtained as follows:

$$\theta_1 = 270^\circ - s\alpha_1; \theta_2 = 270^\circ - s\alpha_2; \mathbf{K} \theta_{n-1} = 270^\circ - s\alpha_{n-1}\sqrt{2}. \tag{9}$$

$$0 \leq \alpha_j \leq 90^\circ; 90^\circ \leq \theta_j \leq 270^\circ. \tag{10}$$

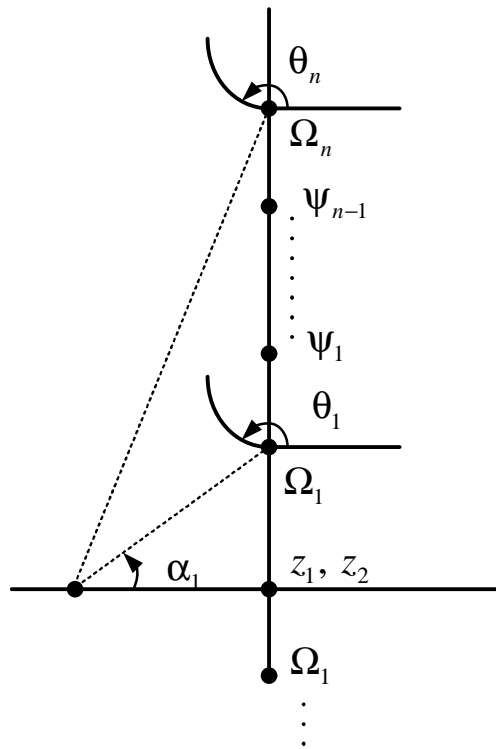


Fig. 7. Root locus of Fig. 6

The above equations mean that the controller based on the external force feedback makes the oscillation poles stable. This is 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”.

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:

$$\begin{cases} \theta_m = \frac{(s^2 + \omega_a^2)}{\omega_a^2} \cdot G_1(s) \cdot G_2(s) \cdot \theta^{cmd}; \theta_a = G_1(s) \cdot G_2(s) \cdot \theta^{cmd}; \\ \omega_a = \sqrt{\frac{K_f}{I_a}}; \omega_m = \sqrt{\frac{K_f}{I_a} \cdot (1 + K_r \cdot I_a)} = K \cdot \omega_a; K = \sqrt{1 + K_r \cdot I_a}. \end{cases} \quad (11)$$

Here ω_a and I_a are the equivalent frequency and inertia of the load side in Fig 4, ω_m and K_f is the natural frequency of the motor side and the equivalent stiffness 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 $G_1(s) \cdot G_2(s)$ is given as follows:

$$D(s) = s^4 + K_v \cdot s^3 + (K_p + \omega_m^2) \cdot s^2 + K_v \cdot \omega_a^w \cdot s + K_p \cdot \omega_a^2. \quad (12)$$

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:

$$D(s) = (s^2 + 2\zeta_1 \cdot \omega_1 \cdot s + \omega_1^2) \times (s^2 + 2\zeta_2 \cdot \omega_2 \cdot s + \omega_2^2). \quad (13)$$

From (12) and (13), the following relations are obtained:

$$\begin{aligned} K_v &= 2 \cdot (\zeta_1 \cdot \omega_1 + \zeta_2 \cdot \omega_2); K_p = \frac{\omega_1^2 \cdot \omega_2^2}{\omega_a^2}; \\ \omega_m^2 &= \sqrt{-\frac{\omega_1^2 \cdot \omega_2^2}{\omega_a^2} + \omega_1^2 + \omega_2^2 + 4\zeta_1 \cdot \zeta_2 \cdot \omega_1 \cdot \omega_2}. \end{aligned} \quad (14)$$

The important goal in the vibration control is to suppress the vibration, so that $\zeta_1 = \zeta_2 = 1,0$ in (14). 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 $\sqrt{5}$.

$$\begin{cases} K_r = \frac{4}{I_a}; K_p = \omega_a^2; \\ K_v = 4 \cdot \omega_a. \end{cases} \quad (15)$$

Experimental Examples of Vibration Control

By using a set of the gains shown in (15), the vibration of the mechanical resonance system is well suppressed.

Fig. 8 and 9 are the experimental results of PD control and resonance ratio control for step-motor, respectively. These results clearly show that the resonance ratio control is effective for the vibration suppression in the mechanical resonance system.

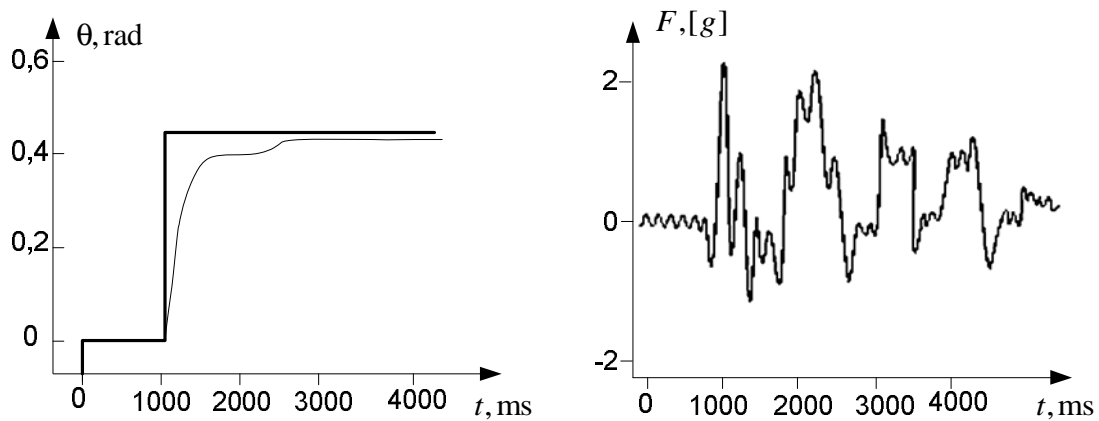


Fig. 8. PD control in mechanical resonance system

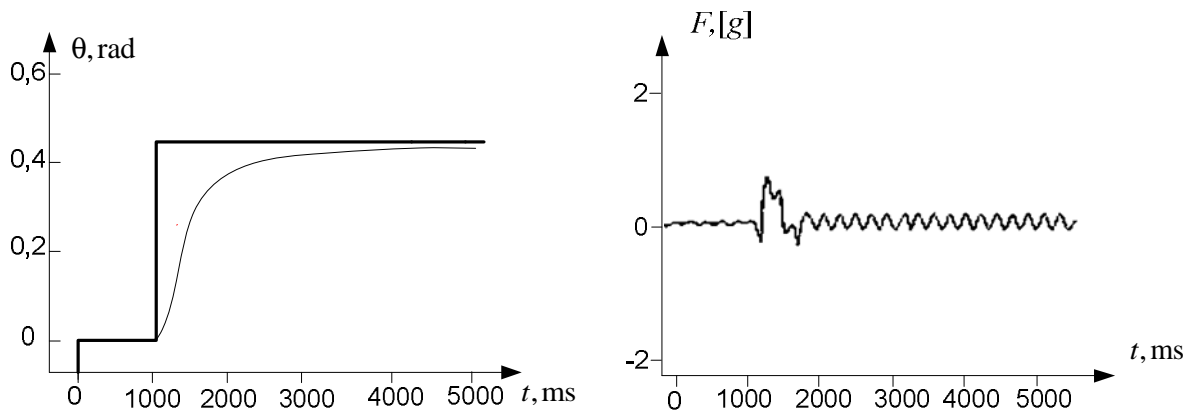


Fig. 9. Resonance ratio control in mechanical resonance system

Conclusions

The robustness of the motion control makes the system more flexible. The stiffness of the motion, which corresponds to the forward gain of the position, is defined to be a good index of robustness. The motion controller acquires robustness concerning estimating disturbance. The robustness and the identification are both sides of a motion which control each other. The recent modern technique, including two-degrees-of freedom control, H^∞ control, has proved the acceptance of the same structure from physical point of view [8].

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.

For further development, particularly in the connection of antivibration controller of motion, the antivibration control will be expected. Such controller is very important for the modern agricultural machines with advanced mechatronic devices in order to stabilize their course of motion.

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