Anti-resonance Vibration Suppression Control in Full-closed Control System

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External position sensors such as linear scales are frequently used in motion control systems of industrial machines, and they form a full-closed position feedback loop for accurate positioning. When the velocity signal calculated with the position signal from the motor encoder is used for the velocity control loop, resonance is less likely to occur than when using the velocity signal calculated with the position signal from the external encoder. If the position loop gain is increased, vibration is generated at the anti-resonant frequency, even if the velocity loop is stably controlled with respect to the resonance frequency. Since there is no control method for suppressing the vibration, such as the vibration control method in semi-closed control, the proportional gain of the position feedback loop cannot be made sufficiently high. In this paper, we propose a full-closed control method that suppresses the vibration at anti-resonance frequency and confirm its effectiveness through simulation and experiment.

Keywords: anti-resonance, vibration suppression, full-closed control

1. Introduction

In the control system of a general industrial application servo drive that drives a linear motion mechanism by a ball screw etc., it is common to configure a double loop that feeds back the motor angle signal and the motor angular velocity signal. Servo motors often only have an angle detector such as a rotary encoder, and the angular velocity is obtained by calculation from the angle signal of the detector. This kind of control method that configures a closed loop based on the motor rotation angle is called semi-closed control (1)-(3), and is often used in industrial applications because it can be driven with a simple configuration.

Meanwhile, with mechanisms such as semiconductor inspection equipment, positioning control using full-closed control with an external sensor such as a linear scale may be performed as the target of inspection becomes smaller. Such mechanisms start and stop frequently, and the angular velocity of the mechanism itself is often not ensured in order to reduce costs, and the machine base and ball screw often vibrate due to the excitation force accompanying the acceleration and deceleration of the motor. In the case of ball screw vibration, the motor generates vibration due to resonance, but in the case of machine base vibration, the anti-resonance vibration of the machine base vibrates the motor according to the law of action and reaction.

When mechanical resonance characteristics are included in the control loop, the control gain is limited to prevent oscillation of the control system. In particular, the proportional gain of the velocity controller is set higher than the proportional gain of the position controller for the sake of stability of the control loop, and therefore is significantly affected by resonance characteristics.

There are two methods for configuring the velocity control system in fully-closed control: a method that calculates the velocity signal from the motor encoder signal and a method that calculates it from the external position sensor signal. In the case of the former, the start-up of the machine can first be commenced as a semi-closed control system, and then the position loop can be switched to an external sensor signal thereafter to improve the position control accuracy of load position (4).

When setup a servo drive on the manufacturing site, it is often the case that the first step is to confirm operation by velocity control. At that time, the mechanism is often under assembly, so it is often operated by semi-closed control using a motor encoder. In applications that require high-accuracy positioning after completion of operation confirmation, position control is performed by switching to full-closed control, but the velocity control system often remains in semi-closed control for the purpose of reducing man-hours.

Examples of measures to stabilize the mechanical resonance characteristics in the velocity loop include a method that uses a low-pass, notch or other filter; a method that inputs the same signal as in a normative model into the actual system, processes the output difference between the two and returns it to the input with the aim of a response similar to the model (5), and a method of stabilizing the control system by feedback back the state quantity estimated using an observer (6)-(13).

In the above full-closed control system, after stabilizing the velocity loop and increasing the gain by such a method, vibration may occur at the anti-resonance frequency of the
mechanism when the proportional gain of the position loop is increased.\(^{15}\)

In the following, the method for calculating the speed from the motor encoder signal is defined as P (Position) type full-closed control, and the method for calculating the speed from the external position sensor signal is defined as PS (Position-velocity) type full-closed control.

For PS type full-closed control, a method of stabilizing by the differential velocity feedback between the encoder attached to the motor and the external sensor (15) and a method of stabilizing only by the external sensor (16) have been proposed. However, none of these methods are intended to suppress anti-resonance of the mechanism. Therefore, PS type full-closed control (with an external position sensor) can be applied to P type full-closed control. Detected speed:

In this study, we propose anti-resonance for P type full-closed control.

The structure of this paper is shown below.

Section 2 shows the mechanism of anti-resonance vibration generation in P type full-closed control, and Section 3 explains vibration suppression when the conventional method is applied. Section 4 explains vibration suppression by the proposed method, and Sections 5 and 6 confirm the effectiveness of the proposed method by simulation and experimental results. Finally, the conclusion of this paper is given.

2. Mechanism of Anti-resonance Vibration Generation

Assuming that a motor control system including mechanical resonance elements such as ball screws is a 2-inertial resonance system as shown in Fig. 1, the equations of motion from the motor torque to the velocity of the control when operating controlled object when operating by full-closed control are Equations (1) and (2).\(^{17}\)

\[ J_m \ddot{\theta}_m + D(\dot{\theta}_m - \dot{\theta}_L) + K(\theta_m - \theta_L) = T_m \]  
\[ J_L \ddot{\theta}_L + D(\dot{\theta}_L - \dot{\theta}_m) + K(\theta_L - \theta_m) = 0 \]

Here, J, D, and K represent the moment of inertia, viscous damping coefficient, and spring constant, respectively. T, \( \dot{\theta} \), and \( \theta \) represent torque, position, and velocity respectively. Subscripts m and L represent the motor side and load side constants, respectively.

When Equations (1) and (2) are made transfer functions by Laplace transform, they become Equations (3) and (4). P-type full-closed control (detected by motor encoder speed):

\[ G_{lp}(s) = \frac{\Theta_m(s)}{T_m(s)} = \left( \frac{1}{J_M + J_L} \cdot \frac{1}{s} \right) \cdot \left( \frac{s^2 + \frac{D}{J_L} s + \frac{K}{J_L}}{s^2 + \left( \frac{1}{J_m} + \frac{1}{J_L} \right) K} \right) \]

\[ = \left( \frac{1}{J_M + J_L} \cdot \frac{1}{s} \right) \cdot \left( \frac{s^2 + 2\zeta_c \omega_n s + \omega_n^2}{s^2 + 2\zeta_c \omega_n s + \omega_n^2} \right) \quad \cdots (3) \]

PS type full-closed control (velocity detected by external encoder):

\[ G_{lp}(s) = \frac{\Theta_m(s)}{T_m(s)} \cdot \left( \frac{1}{J_M + J_L} \cdot \frac{1}{s} \right) \cdot \left( \frac{2\zeta_c \omega_n s + \omega_n^2}{s^2 + 2\zeta_c \omega_n s + \omega_n^2} \right) \]

\[ = \left( \frac{1}{J_M + J_L} \cdot \frac{1}{s} \right) \cdot \left( \frac{s^2 + \frac{D}{J_L} s + \frac{K}{J_L}}{s^2 + 2\zeta_c \omega_n s + \omega_n^2} \right) \quad \cdots (4) \]

However,

\( \zeta_c \): damping coefficient of resonance
\( \omega_n \): Angular vibration frequency of resonance
\( \zeta_{ar} \): Damping coefficient of anti-resonance
\( \omega_{ar} \): Angular frequency of anti-resonance

Here, the first-order term of s in the numerator determines the characteristics of frequencies higher than the resonance peak, but the generation of vibration is considered to have no effect on the study of vibration suppression because of the resonance peak. Therefore, in this study, for simplicity, the first order term of s in the numerator is ignored and the PS-type full-closed control is expressed as

\[ G_{lp}(s) = \left( \frac{1}{J_M + J_L} \cdot \frac{1}{s} \right) \cdot \left( \frac{\omega_n^2}{s^2 + 2\zeta_c \omega_n s + \omega_n^2} \right) \quad \cdots (5) \]

The transfer function from the motor torque to the position signal of the external position sensor can also be described as follows using Equations (3) and (5).

\[ G(s) = \left( \frac{1}{J_M + J_L} \cdot \frac{1}{s} \right) \cdot \left( \frac{s^2 + 2\zeta_c \omega_n s + \omega_n^2}{s^2 + 2\zeta_c \omega_n s + \omega_n^2} \right) \]

\[ \cdot \left( \frac{2\zeta_c \omega_n s + \omega_n^2}{s^2 + 2\zeta_c \omega_n s + \omega_n^2} \right) \quad \cdots (6) \]

When describing the control target in Equation (6), the block diagram of Fig. 2 is obtained in the case the motor velocity and position signal of the external position location sensor are fed back.

The superscripts ref and res indicate reference value and response value, respectively. In Fig. 2, the proportional gain of the velocity controller can be made sufficiently larger than the proportional gain of the position controller by reducing
the effect of mechanical resonance using filters and various vibration suppression control methods, so for simplicity, the transfer function of the velocity control system is made “1”.

At this time, the loop transfer function from the position error to the detected position signal is

\[
G(s) = \frac{K_p \omega_a^2}{s^3 + 2J_a \omega_a s^2 + \omega_a^2 + \omega_d^2} \quad \ldots (7)
\]

Therefore, if the proportional gain of the position controller is increased, the controller may vibrate at a frequency near \(\omega_a\). The transfer function from the position reference to the load position is as follows:

\[
G(s) = \frac{K_p \omega_a^2}{s^3 + 2J_a \omega_a s^2 + \omega_a^2 + \omega_d^2} \quad \ldots (8)
\]

Substituting \(s = j\omega\) into Equation (7) yields the following equation:

\[
G(j\omega) = \frac{K_p \omega_a^2}{j\omega (\omega_a^2 - \omega^2) + 2J_a \omega_a} \quad \ldots (9)
\]

If we rationalize the above equation and find \(\omega\) with an imaginary part of 0, we can obtain Equation (10) from \(\omega_a > 0\).

\[
\omega = \omega_a \quad \ldots (10)
\]

Equation (10) shows that the vibration frequency in the case the position control system becomes unstable is the anti-resonance frequency of the mechanism (16). For this reason, a method for suppressing vibration at the anti-resonance frequency is necessary.

3. Vibration Suppression by Conventional Method

Figure 4 (15) shows an example of a conventional method that feeds back the difference between the motor velocity and the load velocity calculated from the external position sensor. Defining the difference between the motor velocity and the load velocity calculated from the external position sensor as the differential velocity, the open-loop transfer function from the position reference to the differential velocity for the velocity control system of Fig. 4 is given by the following equation. However, for simplicity, the integral is excluded and it was made velocity \(P\) control, such that \(J_m + J_L = 1\).

\[
G(s) = \frac{b_2 s^2 + b_1 s + b_0}{s^3 + a_2 s^2 + a_1 s + a_0} \left( 1 - \frac{\omega_a^2}{s^3 + 2J_a \omega_a s + \omega_a^2} \right) \quad \ldots (11)
\]

\[
G(s) = \frac{b_2 s(s + 2\omega_a \omega_d)}{s^3 + a_2 s^2 + a_1 s + a_0} \quad \ldots (12)
\]

The coefficients \(a_2\) to \(a_0\) and \(b_2\) to \(b_0\) in Equation (11) are as follows:

\[
a_2 = 2\omega_a \omega_d + \frac{\omega_a^2 K_v}{\omega_d^2} \quad \ldots (13)
\]

\[
a_1 = \omega_d^2 + \frac{2\omega_a^2 K_v}{\omega_d} \quad \ldots (14)
\]

\[
a_0 = \omega_a^2 K_v \quad \ldots (15)
\]

\[
b_2 = \omega_a^2 K_v \quad \ldots (16)
\]

Fig. 3. Comparison of simulation results of conventional method and proposed method

Fig. 4. Control block diagram of conventional method with vibration suppression control using differential velocity feedback

\[
b_1 = \frac{2\omega_a \omega_d^2 K_v}{\omega_d} \quad \ldots (17)
\]

\[
b_0 = \omega_a^2 K_v \quad \ldots (18)
\]

The configuration of Equation (12) contains the characteristics of the velocity feedback control system, which includes mechanical resonance and anti-resonance characteristics. Since the form of the equation is different from the transfer function of Equation (7), it is not possible to detect only the anti-resonance frequency that occurs when the position control system becomes unstable.

Therefore, it is considered that anti-resonance vibration cannot be suppressed even if the gain \(K_v\) is simply applied to the differential velocity and fed back to the motor torque. Figure 3 shows the simulation results of Fig. 4. In the end, the method of Fig. 4 does not suppress vibration at all.

However, if the transfer function of the velocity control system could be set to “1”, Equation (11) changes as follows:

\[
G(s) = \frac{s^2 + 2\omega_a \omega_d s + \omega_a^2}{s^3 + 2J_a \omega_a s + \omega_a^2} \quad \ldots (19)
\]

4. Vibration Suppression by the Proposed Method

With Equation (19), it is possible to detect only the anti-resonance frequency, however in reality, the transfer function
of the velocity control system cannot be set exactly to “1”. As a countermeasure, if the velocity reference is used instead of the motor velocity, the transfer function of the velocity control system can be considered to be exactly “1”. Thereafter, the difference between the velocity reference and the load velocity calculated from the external position sensor is used as the differential velocity.

Since \( \zeta_a \ll 1 \) frequently occurs, to simplify the explanation, in the numerator of Equation (19), at frequencies near \( \omega_a \), considering \( s = j\omega_a \) allows the approximation

\[
s + 2\zeta_\omega \omega_a \approx s \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots (20)
\]

From Equation (20), Equation (19) can be approximated as

\[
G(s) = \frac{s^2}{s^2 + \omega_a^2} \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots (21)
\]

Since the numerator of the Equation (21) is a 2nd order equation of \( s \), the signal obtained by integrating the velocity difference and multiplying by proportional gain \( K_f \) is negatively fed back to the velocity reference. At this time, the transfer function from the velocity reference to the load velocity is as follows:

\[
G(s) = \frac{\omega_a^2}{s^2 + K_f s + \omega_a^2} \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots (22)
\]

In Equation (22), since the coefficient of the first-order term of \( s \) of the denominator polynomial can be increased by the proportional gain \( K_f \), an effect equivalent to that of increasing the damping coefficient \( \zeta_a \) of the controlled object can be obtained.

When a full-closed controller that feeds back the position of the external position sensor is constructed outside the velocity control system expressed by the transfer function in Equation (22), the transfer function from the position reference to the load position is as follows:

\[
G(s) = \frac{K_p \omega_a^2}{s^3 + K_f s^2 + \omega_a^2 s + K_p \omega_a^2} \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots \cdots (23)
\]

Comparing Equations (8) and (23), \( \zeta_a \omega_a^2 \), which cannot be directly controlled, is replaced by proportional gain \( K_f \), and it the stability of the position control system can be expected to be restored by increasing \( K_f \).

5. Simulation Results

Figure 5 is a control block diagram that assumes a linear motion mechanism that drives a ball screw with a servo motor. In general, however, the spring element of the mechanism is assumed to be a coupling and ball screw nut, however in Fig.5, the nut alone is assumed to be the spring element. Figure 7 shows the frequency characteristics from the motor torque to the motor velocity for the control object of Fig.5. The anti-resonance frequency corresponding to the natural vibration frequency of the mechanism is about 54 Hz, and the resonance frequency is 300 Hz.

In the case proportional gain of the position controller \( K_p = 300/s \), proportional gain of the velocity controller \( K_v = 1508/s \) and integration time \( T_i = 2.7 \) ms for semi-closed control, the motor rotation angle and the motor torque become as in Fig. 8.

Since the motor torque in Fig. 8 oscillates at about 58 Hz, the velocity controller can sufficiently suppress the resonance, and it can be considered that only the anti-resonance frequency is generated.

Consider a full-closed control system with a linear scale or other position sensor attached to the movable table of a linear motion mechanism.

Regarding full-closed control, we assume a P type full-closed loop control in which only the load position that is used in industrial applications is fed back as shown in Fig. 6, and the velocity is fed back from the motor encoder.
First, consider the case where the proposed method is not used ($K_f = 0$). When operating it with the same gain setting as in semi-closed control, the load position and motor torque vibrate at approximately 58 Hz as shown in Fig. 9. The Nyquist diagram for the velocity closed loop/position open loop is as shown in Fig. 11, indicating that the control system is unstable.

When the proposed method is applied to this state with a proportional gain $K_f = 500$, vibration can be suppressed as shown in Fig. 10, and the proportional gain of the position controller can be set to the same level as in semi-closed control. Figure 12 shows the Nyquist diagram of the velocity closed loop/position open loop when the proportional gain is changed to $K_f = 200$, $K_f = 300$, and $K_f = 500$. In Fig. 12, the proportional gain increases in the order of yellow, blue, and red. As the proportional gain $K_f$ is increased, the stability gradually recovers.

6. Experimental Results

Since the effectiveness of the proposed method was confirmed by simulation, experiments were performed on two types of mechanisms to confirm its effectiveness. Experiments were conducted using two mechanisms: A rotary two-inertial system using a low stiffness spring coupling and HEIDENHAIN K.K. rotary encoder, and a linear motion system connecting a THK CO., LTD. ball screw slider (lead 20 mm) to a HEIDENHAIN K.K. linear scale (measurement resolution 0.1 μm).

Figures 13 and 16 show the appearance of the experimental apparatus, and Figs. 14 and 17 show the respective frequency characteristics.

Table 1 shows the servo parameters applied during the experiment and the load moment of inertia of the experimental equipment.

In the rotary two-inertial mechanism shown in Fig. 13, three couplings are connected. Couplings 2 and 3 exhibit spring characteristics at a frequency much higher than the frequency range of the controller used in Table 1. This results in the two-inertial characteristics of Coupling 1 only as in the frequency characteristics of Fig. 14.

Meanwhile, in the ball screw mechanism of the linear motion system of Fig. 16, the resonance characteristic of Coupling 1 exists near 1300 Hz, and the resonance characteristic of the ball screw nut exists near 200 Hz. Furthermore, because it is a linear motion system, the natural vibration of the machine base exists in the frequency characteristics as an anti-resonance characteristic of about 30 Hz. Resonance generated at 1300 Hz when the proportional gain of the velocity controller is increased is suppressed by a notch filter.

When the proposed method is not applied, the two-inertial mechanism in Fig. 13 generates 8.2 Hz vibration in the position error and motor torque as shown in Fig. 15, and the ball screw mechanism in Fig. 16 generates 31 Hz vibration in the position error and motor torque shown in Fig. 18.
Since all the vibrations that occur are almost the same as the anti-resonance frequency of the mechanism, they are found to be generated by the natural vibration of the mechanism. In particular, when looking at the position error graph, the vibration damping time is long, so the positioning time of the mechanism is long, which is a problem for industrial use.

When the proposed method is applied to a state in which vibration occurs, the results are as shown in Figs. 15 and 18, respectively.

In all cases, the vibration was clearly reduced, and the effectiveness of the proposed method in industrial applications was confirmed.
7. Conclusion

Simulations and experiments demonstrated that the method of suppressing vibration by feeding back the difference between the velocity reference and the velocity calculated from the external sensor position is effective against vibration caused by the natural vibration of a mechanism that occurs during full-closed control.

The proposed method is considered to be effective for industrial applications because the configuration and adjustment of the control system are simple.

References


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