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Oscillation Control of a Contact Scanning Scanners System

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Abstract: In the inspection of the deterioration state of the structures such as plants, contact scanning type sensors are useful devices. Using the sensors, we can check deterioration of structure and matter inside pipes during the production process. However, in case of the high-speed scanning, oscillation occurs in the sensor part due to unevenness and friction on the measured surface. In this paper, to overcome the problem, we propose an oscillation controller for scanning sensor systems. At last, to show the effectiveness of the proposed controller, numerical simulations are carried out.

Keywords: The scanning system of a sensor, Contact type sensor, Oscillation control, Accelerometers, Ultrasonic sensor

1 INTRODUCTION

In the inspection of the deterioration state of the structures such as plants, the manufacturing inspection of metals, inspection of the welded junction part and so on, usually, contact type scanning sensor systems [1-5] with ultrasonic are used. In case of the high-speed scanning, oscillation occurs in the sensor part due to unevenness and friction on the measured surface. If oscillation occurs in the sensor part, measurement accuracy may worsen remarkably caused by the fluctuation of the ultrasonic incident angle to inspection surface. In addition, the durability of the mechanism of inspection systems is also reduced. For the reasons, in the real inspection, a slow scanning speed is used to achieve the measurement accuracy. Moreover, since the amplitude of oscillation varies according to surface conditions of materials, the operators need to determine the scanning speed by using a trial and error technology so that the occurred oscillation in the sensor part can be reduced in the permissible level. To address the problem, there are the following two methods. One is to increase the number of sensors. Another one is to control oscillation of the sensor part with accurate measurements of the oscillation. However, both methods require expensive measuring instruments and sensors.

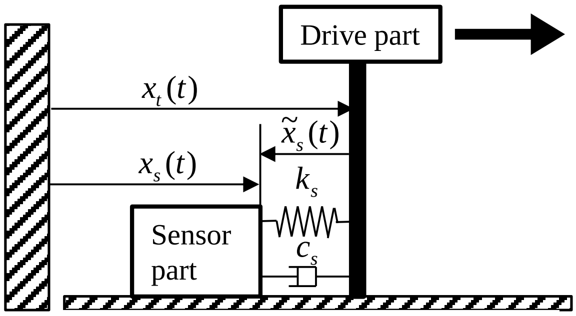


Fig.1 Simplified Model

In this research, in order to overcome the problem mentioned above, we propose a new oscillation controller. One of the main features of the controller is that cheap accelerometers are only used for measurements of the sensor part.

2 CONTROL OBJECT AND CONTOROL PURPOSE

Fig. 1 shows the simplified model of a scanning type sensor system. The dynamic equation of this model is expressed by the following equation. The symbol $f(t)$ denotes the dead zone and shows friction occurring in the sensor part. In addition, explanations of the symbols used in the following equations are shown in Table.1, and the nominal values of parameters are shown in Table.2.

$$\ddot{x}_t(t) = -\frac{c_t}{m_t}\dot{x}_t(t) + \frac{b_t}{m_t}u(t) \tag{1}$$

$$\ddot{\tilde{x}}_s(t) = f(t) - \ddot{x}_t(t) \tag{2}$$

$$f(t) = \begin{cases} -\frac{c_s}{m_s}\dot{\tilde{x}}_s(t) - \frac{k_s}{m_s}\tilde{x}_s(t) - d & -\frac{c_s}{m_s}\dot{\tilde{x}}_s(t) - \frac{k_s}{m_s}\tilde{x}_s(t) > d \\ 0 & \left|-\frac{c_s}{m_s}\dot{\tilde{x}}_s(t) - \frac{k_s}{m_s}\tilde{x}_s(t)\right| \le d \\ -\frac{c_s}{m_s}\dot{\tilde{x}}_s(t) - \frac{k_s}{m_s}\tilde{x}_s(t) + d & -\frac{c_s}{m_s}\dot{\tilde{x}}_s(t) - \frac{k_s}{m_s}\tilde{x}_s(t) < -d \end{cases} \tag{3}$$

The control objective is to reduce the oscillation of $|\tilde{x}_s(t)|$. To achieve the objective, we make the following assumptions for the controlled object.

A1: The position $x_t(t)$ and speed $\dot{x}_t(t)$ of the drive part are measurable.

A2: The position $x_s(t)$ and speed $\dot{x}_s(t)$ of the sensor part can not be measured.

A3: The acceleration $\ddot{x}_t(t)$ and the relative acceleration $\ddot{\tilde{x}}_s(t)$ are measurable.

3 OSCILLATION CONTROL SCHEMES

In order to simplify development of an oscillation controller, consider the linearized model of the inspection system given as

$$\ddot{x}_t(t) = -\frac{c_t}{m_t}\dot{x}_t(t) + \frac{b_t}{m_t}u(t) \tag{4}$$

$$\ddot{\tilde{x}}_s(t) = -\frac{c_s}{m_s}\dot{\tilde{x}}_s(t) - \frac{k_s}{m_s}\tilde{x}_s(t) - \ddot{x}_t(t) \tag{5}$$

where c_s is a small damping constant and k_s is a spring constant.

In the equation (5), if it is assumed that the signal $\ddot{x}_t(t)$ can be used as a control input, we may use the controller given by

$$\ddot{x}_t(t) = \alpha\dot{\tilde{x}}_s(t) \tag{6}$$

where α is a design parameter. By using this control input, the oscillation can be reduced in a relative position $\tilde{x}_s(t)$. However, according to the assumption A2, the relative speed $\dot{\tilde{x}}_s(t)$ is unavailable. Moreover, in fact, $\ddot{x}_t(t)$ cannot be used as an input signal.

In order to overcome the problem, the following new signal is defined.

$$\ddot{\tilde{x}}_t(t) = \ddot{x}_t(t) - \alpha L^{-1}\left[\frac{Ts}{Ts+1}\frac{T}{Ts+1}\ddot{\tilde{X}}_s(s)\right] - \dot{v}_{td}(t) \tag{7}$$

Here, T is the design parameter, and $v_{td}(t)$ is the ideal speed of the drive part given by

$$V_{td}(s) = \frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2}V_d(s) \tag{8}$$

where ω_n, ς are design parameters and v_d is the final constant speed.

$$\frac{Ts}{Ts+1}\frac{T}{Ts+1}\ddot{X}_s(s) = \left(\frac{Ts}{Ts+1}\right)^2\dot{X}_s(s) \tag{9}$$

The right hand side denotes the high frequency signals of $\dot{\tilde{x}}_s(t)$. Using the new signal (7) in (5) and rewriting, we have

Table.1 Notation of Board Thickness Scanners

x_t	position of drive division
$\tilde{x}_s$	relative position of sensor division to drive system
m_t , m_s	mass of drive part and sensor part
b_t , c_t	gain of input and viscosity resistance of motor
k_s , c_s	spring constant and damper constant
u	control input

Table.2 Simulation parameter

$\frac{b_t}{m_t}$	0.55	$\frac{k_s}{m_s}$	2518	d	0.9
$\frac{c_t}{m_t}$	5.05	$\frac{c_s}{m_s}$	1.88		

$$\tilde{X}_s(s) = G_s(s)\left(\ddot{\tilde{X}}_t(s) + \dot{V}_{td}(s)\right) \tag{10}$$

$$\left.\begin{aligned} A_{z41} &= \frac{k_s}{m_s}\frac{1}{T^2}, \quad A_{z42} = 2\frac{k_s}{m_s}\frac{1}{T} + \frac{c_s}{m_s}\frac{1}{T^2} \\ A_{z43} &= \frac{k_s}{m_s} + 2\frac{c_s}{m_s}\frac{1}{T} + \frac{1}{T^2} \\ A_{z44} &= \frac{c_s}{m_s} + \alpha + \frac{2}{T} \end{aligned}\right\} \tag{11}$$

where the design parameter α and T are set so that the transfer function $G_s(s)$ can become asymptotically stable. Since the ideal acceleration $\dot{v}_{td}(t)$ of the drive pert converges to zero, if $\ddot{\tilde{x}}_t(t)$ can be designed so as to converge to zero, then, $\tilde{x}_s(t)$ becomes also asymptotically stable. In order to realize this, a new state

$$\boldsymbol{x}(t) = \left[\dot{\tilde{x}}_t(t) \quad \ddot{\tilde{x}}_t(t)\right]^T \tag{12}$$

is defined. Using the state, we have

$$\dot{\boldsymbol{x}}(t) = A\,\boldsymbol{x}(t) + B\,\boldsymbol{z}(t) + D\boldsymbol{v}_d(t) + \boldsymbol{b}\left(\dot{u}(t) + u(t)\right) \tag{13}$$

$$\left.\begin{aligned} \mathbf{z}(t) &= [z_1 \quad z_2 \quad z_3 \quad z_4]^T \\ z_i &= L^{-1}\left[\frac{s^{i-1}}{(Ts+1)^2}\tilde{X}_s(s)\right] \ (i = 1, 2, 3, 4) \\ \mathbf{v}_d(t) &= [v_{td}(t) \quad \dot{v}_{td}(t) \quad \ddot{v}_{td}(t)]^T \\ A &= \begin{bmatrix} 0 & 1 \\ -\frac{c_t}{m_t} & -\left(\frac{c_t}{m_t} + 2\right) \end{bmatrix}, \quad \mathbf{b} = \begin{bmatrix} 0 \\ \frac{b_t}{m_t} \end{bmatrix} \\ B &= \begin{bmatrix} 0 & 0 & 0 & 0 \\ B_{21} & B_{22} & B_{23} & B_{24} \end{bmatrix} \\ D &= \begin{bmatrix} 0 & 0 & 0 \\ -\frac{c_t}{m_t} & -\left(\frac{c_t}{m_t} + 2\right) & -1 \end{bmatrix} \\ B_{21} &= -\left(\frac{c_s}{m_s} + \frac{c_t}{m_t} + 1\right)T^2 + 2T \\ B_{22} &= -\left(\frac{c_t}{m_t} + \frac{k_s}{m_s}\right)T^2 - 2\frac{c_s}{m_s}T + 1 \\ B_{23} &= -\alpha T^2 + 2\frac{k_s}{m_s}T + \frac{c_s}{m_s}, \quad B_{24} = \frac{k_s}{m_s} \end{aligned}\right\} \tag{14}$$

$$\dot{\boldsymbol{z}}(t) = A_z\boldsymbol{z}(t) + B_z\boldsymbol{x}(t) + D_z\boldsymbol{v}_d(t) \tag{15}$$

$$\left.\begin{aligned} A_z &= \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ A_{z41} & A_{z42} & A_{z43} & A_{z44} \end{bmatrix} \\ B_z &= \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & -\frac{1}{T^2} \end{bmatrix}, \quad D_z = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & -\frac{1}{T^2} \\ 0 & 0 & 0 & 0 \end{bmatrix} \end{aligned}\right\} \quad (16)$$

where A_z is an asymptotically stable matrix. Based on the system, we developed the following controller.

$$\dot{u}(t) = -u(t) - \boldsymbol{k}^T \boldsymbol{x}(t) \ , \boldsymbol{k}^T = \begin{bmatrix} \beta^2 \frac{m_t}{b_t} & 2\beta \frac{m_t}{b_t} \end{bmatrix} \quad (17)$$

Where β is the design parameter. Using the new state

$$\mathbf{x}_\beta(t) = \mathrm{diag}[1 \quad \beta^{-1}]\mathbf{x}(t) \quad (18)$$

we have

$$\begin{aligned} \dot{\mathbf{x}}_\beta(t) &= \beta A_\beta \mathbf{x}_\beta(t) + \left(A_{\beta 1} + \beta^{-1} A_{\beta 2}\right)\mathbf{x}_\beta(t) \\ &\quad + \beta^{-1} B\, \mathbf{z}(t) + \beta^{-1} D\, \mathbf{v}_d(t) \end{aligned} \quad (19)$$

$$\dot{\mathbf{z}}(t) = A_z \mathbf{z}(t) + \beta B_z \mathbf{x}_\beta(t) + D_z \mathbf{v}_d(t) \quad (20)$$

$$\left.\begin{aligned} A_\beta &= \begin{bmatrix} 0 & 1 \\ -1 & -2 \end{bmatrix}, \ A_{\beta 1} = \begin{bmatrix} 0 & 0 \\ 0 & -\left(\frac{c_t}{m_t} + 2\right) \end{bmatrix} \\ A_{\beta 2} &= \begin{bmatrix} 0 & 0 \\ -\frac{c_t}{m_t} & 0 \end{bmatrix} \end{aligned}\right\} \quad (21)$$

The stability of the control system (19)-(21) using the controller (17) is shown below.

Theorem: In the control system (19)-(21), the design parameter β is set such that the following inequalities are satisfied.

$$\rho_{x1} = \frac{1}{2}\rho_1 - \beta^{-1}(\rho_2 + \rho_3 + \rho_4) > 0 \quad (22)$$

$$\rho_{z1} = \frac{1}{4}\rho_{vz}\rho_5 - \beta^{-1}\rho_3 > 0 \quad (23)$$

$$\left.\begin{aligned} &\rho_1 = \lambda_{\min}[Q_\beta], \rho_2 = 2\left(\|P_\beta A_{\beta 1}\| + \|P_\beta A_{\beta 2}\|\right), \\ &\rho_3 = \|P_\beta B\|, \rho_4 = \|P_\beta D\|, \rho_5 = \lambda_{\min}[Q_z] \\ &\rho_{vz} = \frac{\rho_1 \rho_5}{\|P_z B_z\|^2} \end{aligned}\right\} \quad (24)$$

Where matrixes P_β and P_z are the positive definite matrix solutions of the Liapounov equations as

$$\begin{aligned} &A_\beta{}^T P_\beta + P_\beta A_\beta = -Q_\beta, \quad Q_\beta > 0 \\ &A_z{}^T P_z + P_z A_z = -Q_z, \quad Q_z > 0 \end{aligned} \quad (25)$$

If the initial states are $\boldsymbol{x}_\beta(0) = 0$ and $\boldsymbol{z}(0) = 0$, there exists a positive constant ρ_6 independent of the design parameter β such that

$$\|\boldsymbol{x}_\beta(t)\|^2 \le \beta^{-4}\rho_6 \quad (26)$$

Proof: Consider the positive definite function $V_1(t)$.

$$V_1(t) = \mathbf{x}_\beta^T(t) P_\beta \mathbf{x}_\beta(t) + \beta^{-1}\rho_{vz}\mathbf{z}^T(t) P_z \mathbf{z}(t) \quad (27)$$

The time derivative of $V_1(t)$ satisfies the following relation.

$$\dot{V}_1 \le -\beta\rho_{x1}\|\mathbf{x}_\beta(t)\|^2 - \beta^{-1}\rho_{z1}\|\mathbf{z}(t)\|^2 + \beta^{-1}\rho_7 \quad (28)$$

The parameter ρ_7 is a positive constant independent of the design parameter β. Consider the inequalities

$$\left.\begin{aligned} &2\beta^{-1}\rho_3\|\mathbf{x}_\beta(t)\|\|\mathbf{z}(t)\| \le \rho_3\left(\|\mathbf{x}_\beta(t)\|^2 + \beta^{-2}\|\mathbf{z}(t)\|^2\right) \\ &2\beta^{-1}\rho_4\|\mathbf{x}_\beta(t)\|\|\mathbf{v}_d(t)\| \le \rho_4\|\mathbf{x}_\beta(t)\|^2 + \beta^{-2}\rho_8 \\ &2\rho_{vz}\|P_z B_z\|\|\mathbf{z}(t)\|\|\mathbf{x}_\beta(t)\| \le \frac{1}{2}\beta^{-1}\rho_{vz}\rho_5\|\mathbf{z}(t)\|^2 \\ &\qquad + \frac{1}{2}\beta\rho_1\|\mathbf{x}_\beta(t)\|^2 \\ &2\beta^{-1}\rho_{vz}\|P_z D_z\|\|\mathbf{z}(t)\|\|\mathbf{v}_d(t)\| \le \frac{1}{4}\beta^{-1}\rho_5\|\mathbf{z}(t)\|^2 + \beta^{-1}\rho_9 \end{aligned}\right\} \quad (29)$$

where ρ_8 and ρ_9 are positive constants independent of the design parameter β. Using (29), (23) and (24), we have

$$\dot{V}_1(t) \le -\rho_v V_1(t) + \beta^{-1}\rho_{10} \quad (30)$$

where ρ_{10} is a positive constant independent of the design parameter β. From (30), it can be seen that the control system becomes stable. Moreover, it can be obtain that there exists a positive constant ρ_z independent of the design parameter β such that $\|\mathbf{z}(t)\| \le \rho_z$.

In order to analyze the detail control performance of the control system, consider the following positive definite function $V_2(t) = \boldsymbol{x}_\beta^T(t) P_\beta \boldsymbol{x}_\beta(t)$. Using the relations

$$\begin{aligned} &2\beta^{-1}\rho_3\|\mathbf{x}_\beta(t)\|\|\mathbf{z}(t)\| \le \frac{1}{4}\beta\rho_1\|\mathbf{x}_\beta(t)\|^2 + \beta^{-3}\rho_{11}\|\mathbf{z}(t)\|^2 \\ &2\beta^{-1}\rho_4\|\mathbf{x}_\beta(t)\|\|\mathbf{v}_d(t)\| \le \frac{1}{4}\beta\rho_1\|\mathbf{x}_\beta(t)\|^2 + \beta^{-3}\rho_{12} \end{aligned} \quad (31)$$

we obtain

$$\dot{V}_2(t) \le -\beta\rho_{x2}\|\mathbf{x}_\beta(t)\|^2 + \beta^{-3}\rho_{13} \quad (32)$$

$$\rho_{x2} = \frac{1}{2}\rho_1 - \beta^{-1}\rho_2 \quad (33)$$

where ρ_{11}, ρ_{12} and ρ_{13} are positive constants independent of the design parameter β. According to (22), ρ_{x2} is a positive value. Therefore, from (32), we can derive the inequality (26).

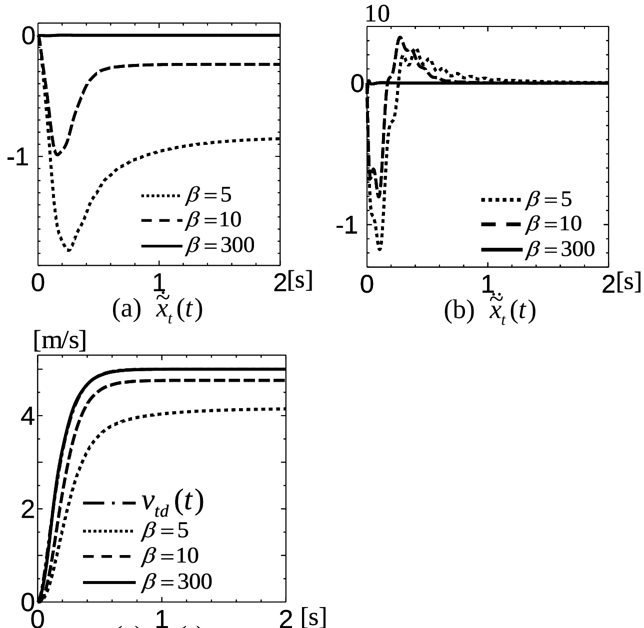
From the theorem, it can be concluded that the absolute values of $\dot{\tilde{x}}_t(t)$ and $\ddot{\tilde{x}}_t(t)$ can be reduced by setting the value of the design parameter β.

4NUMERIC SIMULATIONS

Carrying out numerical simulations, the effectiveness of the developed controller is shown. The system (1) and (2) are used as the dynamic equation of the controlled object. The values of system parameters are shown in Table 2. The design parameters except β are set as $\omega_n = 11$, $\xi = 1$, $T = 0.1$ and $\alpha = 60$. The final ideal speed of the drive part is set as $v_d = 5$. Fig. 2 shows responses for the design parameter $\beta = 5, 10, 300$. Fig.2 (a) and (b) show the responses of the signals $\dot{\tilde{x}}_t(t)$ and $\ddot{\tilde{x}}_t(t)$. Fig.2 (c) shows the responses of the ideal speed $v_{td}(t)$ of the drive part and the speed of the drive part $\dot{x}_t(t)$.

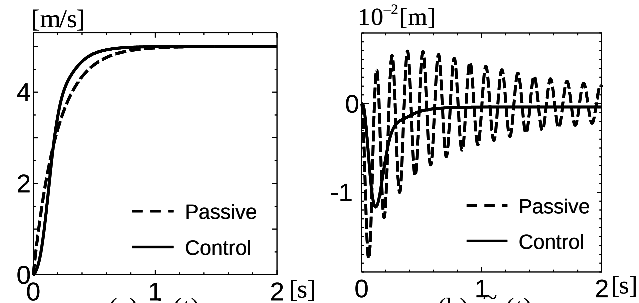
As shown in Fig. 2 (a) and (b), by increasing the value of β, the maximum values of the absolute values of $\dot{\tilde{x}}_t(t)$ and $\ddot{\tilde{x}}_t(t)$ become smaller. As shown in Fig.2 (a) and (c), when β is set as 300, the absolute value of $\dot{\tilde{x}}_t(t)$ become almost zero, and $\dot{x}_t(t)$ follows $v_{td}(t)$.

Fig.3 shows the responses of the controlled system with $\beta = 300$ and the passive system with a constant input ($u = 45.9$). As shown in Fig.3, compared with the passive system, the speed of drive part of the control system converges to the ideal value 5 m/s quickly. Moreover, as shown in Fig.3 (b), it can be seen that oscillation occurs in the passive system but the oscillation can be controlled in the control system.



(a) $\dot{\tilde{x}}_t(t)$ (b) $\ddot{\tilde{x}}_t(t)$ (c) $\dot{x}_t(t)$

Fig.2 Response of $\dot{\tilde{x}}_t(t), \ddot{\tilde{x}}(t)_t, \dot{x}(t)_t$



(a) $\dot{x}_t(t)$ (b) $\tilde{x}_s(t)$

Fig.3 Respnse of $\dot{x}_t(t), \tilde{x}_s(t)$

5CONCLUSION

In this paper, based on the linearlized dynamic equation, we proposed the oscillation control technique for the contact type scanning sensor systems using the accelerometer. Carrying out numerical simulations for the nonlinear controlled object, effectiveness of the developed controller was confirmed. As a result, it has been shown that the oscillation of the sensor part can be prevented in high-speed scanning.

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