Tuning Methods of a Smith Predictor for Pneumatic Active Anti-Vibration Apparatuses*

Yukinori NAKAMURA**, Satoru GOTO** and Shinji WAKUI**

**Tokyo University of Agriculture and Technology
2–24–16 Nakacho, Koganei, Tokyo, 184-8588 Japan
E-mail: yukino-n@cc.tuat.ac.jp

Abstract

Pneumatic active anti-vibration apparatuses (AVAs) are widely employed for the high-precision positioning of wafer stages in semiconductor lithography equipment. However, there exists delay of transmission of air supplied to air springs. In this paper, we consider the suppression of vibration caused by the time delay (i.e., dead-time) of AVAs. The dead-time is compensated by using a Smith predictor, which is well-known as a control method for time delay systems. In the implementation of the predictor, the plant model and the dead-time are needed. For this reason, this paper focuses on the tuning of Smith predictor effective for the control of AVAs. To improve the model accuracy of AVAs, the approximation model of pipe resonance is utilized. Moreover, to determine suitable time delay in the Smith predictor, our methods utilize the resonance frequency of the control system and the tracking error of the isolated table. Conventionally, the nominal value of the dead-time can be obtained by using step response of the control system. On the other hand, it is difficult to exactly estimate the dead-time of AVAs due to position sensor noise. Therefore, the time delay is precisely tuned by proposed methods instead of the step response approach in order to improve the performance of the Smith predictor. Experimental results show that when the error of the time delay in the predictor is zero, the resonance frequency and the time integral become minimized.

Key words: Anti-Vibration Apparatus, Smith Predictor, Resonance Frequency, Time Integral

1. Introduction

In semiconductor manufacturing, wafer scanners are mounted on pneumatic anti-vibration apparatuses (AVAs)1)–6). The apparatuses are employed to suppress disturbances such as ground vibration, machine vibration in lithography equipment, and so on. For this reason, high-precision positioning of wafer stages can be guaranteed by using AVAs. In active type of pneumatic AVAs, the amount of air in springs is adjusted by feedback control so that the position of the isolated table can be kept constant. To supply air to the springs, compressed air is sent through servo valves. However, since semiconductor lithography equipment is large and complex, it is difficult to inspect servo valves located near air springs. For ease of maintenance, the servo valves used in the industry of semiconductor manufacturing are set apart from air springs (see Fig. 1). Because of this, the length of pipes from servo valves to air springs becomes long. Consequently, the time delay in the response of air springs becomes large. It leads to the performance degradation of the vibration control.

As one approach for the compensation of the delay, a Smith predictor7) has been well known. This predictor is used to various applications. Reference (8) employed the Smith predictor and a proportional controller for the TCP congestion control in the presence of the communication delay. In Ref. (9), it was experimentally shown that the production of aminoacids increased by using the filtered Smith predictor. Reference (10) utilized the predictor so as
to control the water level of irrigation canal pool. However, in the case of pneumatic active AVAs, there are two practical issues for the implementation of the Smith predictor.

The first issue is to consider pipe resonance, which arises in high frequency region. The reason is that in the Smith predictor, the plant model is utilized for the delay compensation. In our previous work\(^{(11)}\), the nominal value of the AVA parameters was identified to obtain the plant model of AVAs, whereas effects of pipe length were not considered. In addition to the plant model, the dead-time is also utilized for the implementation of the Smith predictor. To estimate the dead-time, the experimental result on the step response of the control system is often employed\(^{(12)}\). However, in the case of the AVA, the error is present in the estimate of the dead-time due to the position sensor noise. In Refs. (1)-(6), the exact measurement method of the dead-time in AVAs was not discussed sufficiently. Hence, the second issue is to estimate the dead-time in order to improve the control performance of AVAs.

In this paper, we consider the implementation of the Smith predictor for AVAs. First, in our methods, the approximation model of the pipe resonance is utilized in the implementation. The accuracy of the plant model considering the pipe resonance is evaluated based on frequency characteristics of AVAs. Then, the time delay in the Smith predictor are tuned by using the following two methods: resonance frequency based approach and tracking error based approach. In the former (latter) one, the time delay is decided so that the resonance frequency of the control system (the tracking error of the isolated table) for AVAs is minimized.

The rest of the paper is organized as follows. Section 2 explains the AVA used in experiments and its control system. Section 3 evaluates effects of the pipe length. Section 4 presents the resonance frequency based tuning approach of the time delay in the Smith predictor. In Section 5, the tracking error based tuning approach is explained. Finally, Section 6 contains concluding remarks.

2. AVA and Its Control System

2.1. Pneumatic Active Anti-Vibration Apparatuses

Figure 2 shows the structure of the pneumatic AVA used in experiments. At the upper side of the apparatus, the isolated table of 120.22 kg is mounted. Under the table, an air spring is equipped. Furthermore, in order to suppress the vibration in the horizontal direction, an air slider is also equipped. To measure the vertical displacement of the isolated table and the inner pressure of the air spring, a position sensor (SDL, NS2300/A) and a pressure sensor (Setra, 204) are attached to the apparatus. As shown in Fig. 3, the compressed air is supplied to the air spring through the servo valve. The measured signal on the displacement of the isolated table is fed back to the digital signal processor (DSP). In the DSP, the voltage to the servo valve is determined based on the control law. By using the voltage, the servo valve is opened/closed.

Figure 4 shows the schematic diagram of the AVA used in experiments. Variables and parameters are summarized in Table 1. Now, we consider the model linearized around equilibrium position of the isolated table. When \(x_0 = 0\), the transfer function from the input voltage \(u\) to the displacement \(x\) of the isolated table is given by

\[
\frac{x}{u} = \frac{1}{Ms^2 + Ds + K + \frac{A_0 G_q}{\beta_0 V_0 s + c}} e^{-T_0 s}. \tag{1}
\]
The block diagram of the A VA is shown in Fig. 5. In this figure, \( k_{\text{pres}} \) denotes the pressure sensor sensitivity [V/Pa]. The A VA consists of a pneumatic system and mechanical system. The pneumatic system in Fig. 5 corresponds to \( A_0G_qe^{-T_d/s}/(\beta_0V_0s + c) \) in (1). Since the servo valve is located some distance from the air spring, the time delay of input exists. Thus, the dead-time is included in the model of the A VA.

2.2. Smith Predictor

The control system with the Smith predictor for the A VA is shown in Fig. 6. \( \tilde{G} \) denotes the
model of the AVA, and \( T_d \) denotes the time delay [s] in the Smith predictor. \( r \) and \( k_{pos} \) denote reference [V] and position sensor sensitivity [V/m], respectively. To levitate the isolated table, the position PI controller

\[
C_{pos}(s) = k_s \frac{T_{pos} s + 1}{T_{pos} s}, \tag{2}
\]

is utilized. \( k_s \) and \( T_{pos} \) denote gain [-] and time constant [s]. In the development of pneumatic AVAs for lithography equipment, the compensator for the position control is designed such that the low-frequency vibration can be suppressed. For this reason, PI compensator (2) is employed for the position feedback control instead of the controller including differentiator.

By denoting

\[
G(s) = \frac{A_0 G_q}{A_0 s^2 + (M s^2 + D s + K)(\beta_0 V_0 s + c)}, \tag{3}
\]

we obtain the following closed-loop transfer function from the reference \( r \) to the displacement \( x \):

\[
x \frac{r}{x} = C_{pos} G e^{-T_d s} \left( 1 + k_{pos} C_{pos} G e^{-T_d s} - \tilde{G} e^{-T_d s} \right). \tag{4}
\]

By choosing \( \tilde{G} = G \) and \( T_d = T_d \), (4) can be rewritten as

\[
x \frac{r}{x} = C_{pos} G e^{-T_d s} \left( 1 + k_{pos} C_{pos} G e^{-T_d s} \right). \tag{5}
\]

Note that the dead-time element is not present in the denominator of (5). Hence, to compensate for the dead-time by Smith predictor, it is needed to suitably select the plant model and the time delay.

3. Tuning of the Model Considering Pipe Length

In this section, we consider the tuning of the model for the AVA. As mentioned in Section 1, the nominal value of the AVA’s parameters was obtained\(^{(1)}\). On the other hand, effects of pipe length were not investigated. To improve the accuracy of the plant model, we evaluate frequency characteristics of the AVA under different pipe lengths. Figure 7 shows the experimental result on Bode plots of the transfer function from the input voltage \( u \) to the pressure sensor output \( k_{pres} p \). The sensitivity \( k_{pres} \) of the pressure sensor is \( 1.42 \times 10^{-5} \text{ V/Pa} \). From this figure, it is found that in the frequency region from 10 to 100 Hz, resonance is observed. Moreover, the resonance frequency decreases as the pipe length increases. When the air supplied by compressor is reflected to the air spring, the standing wave is created by the sum of two travelling waves in opposite directions (see Fig. 8). Due to the standing wave, column resonance\(^{(13)}\) occurs in the pipe. As a result, the high-frequency resonance is observed in Fig. 7. The frequency \( f_{pipe} \) [Hz] of the first mode of the column resonance is written as\(^{(13)}\)

\[
f_{pipe} = \frac{v}{4L}, \tag{6}
\]

where \( v \) denotes the air sound velocity [m/s] and \( L \) denotes the pipe length [m]. Figure 9 shows effects of the pipe length \( L \) on the resonance frequency \( f_{pipe} \) of the first mode. In this figure,
$v$ is set to 340 m/s. Theoretical value of the resonance is obtained from (6). From this figure, it can be seen that the experimental result is similar to theoretical value. Thus, it is probable that by considering the column resonance, the accuracy of the plant model can be improved. The control system considering the pipe resonance is shown in Fig. 10. In this figure, $G_{pipe}$ denotes the transfer function corresponding to the resonance, and $\tilde{G}_{pipe}$ denotes the model of the resonance. Based on the results shown in Fig. 7, the pipe resonance is approximately modelled as the following second order lag system:

$$G_{pipe}(s) = \frac{\tilde{\omega}_{pipe}^2}{s^2 + 2\tilde{\zeta}_{pipe}\tilde{\omega}_{pipe}s + \tilde{\omega}_{pipe}^2},$$  

where $\tilde{\omega}_{pipe}$ and $\tilde{\zeta}_{pipe}$ denote the natural angular frequency [rad/s] and the damping ratio [-]. Figure 11 shows Bode plots of the plant model. In this figure, the AVA’s model parameters used in the experiment are the same as those used in Ref. (11). The natural angular frequency $\tilde{\omega}_{pipe}$ is set to 119.38 rad/s and the damping ratio $\tilde{\zeta}_{pipe}$ is set to 0.12. The pipe length $L$ is 3.0 m. By using the model of the pipe resonance, frequency characteristics of the plant model are almost same as those of the AVA around frequency of the column resonance.

![Fig. 7 Effects of the pipe length.](image)

![Fig. 8 Principle of column resonance.](image)

![Fig. 9 Resonance frequency $f_{pipe}$ vs. pipe length $L$.](image)

![Fig. 10 Control system considering column resonance.](image)

![Fig. 11 Bode plots of the plant model with pipe resonance.](image)
4. Tuning of the Time Delay –Resonance Frequency Based Approach–

In this section, we propose the first tuning method of the time delay $\tilde{T}_d$. In this method, the time delay is adjusted by using the resonance frequency of the closed-loop control system for the AVA. In the following subsection, the issue of step response based tuning method is considered. Moreover, the detail of our method is presented.

4.1. Issue of Step Response Based Approach

To evaluate the nominal value of the dead-time, we measure the step response of the displacement for the isolated table. In this experiment, the gain and the time constant of the PI controller (2) are chosen as $k_s = 0.2$ and $T_{pos} = 17.68$ s, respectively. The sensitivity $k_{pos}$ of the position sensor is 6667 V/m. Figure 12 shows the experimental result of the step response. In this figure, the equilibrium position of the isolated table is set to zero. From the result, we can find that the dead-time of about 0.4 s is observed. However, it is difficult to exactly measure the dead-time due to the sensor noise. For this reason, it is needed to adjust the time delay $\tilde{T}_d$ in the Smith predictor by using alternative methods.

4.2. Time Delay Adjustment Using Resonance Frequency

To precisely tune the time delay, we focus on the resonance frequency of the closed-loop control system. From Fig. 10, the transfer function from the reference $r$ to the displacement $x$ is given by

$$\frac{x}{r} = \frac{C_{pos}GG_{pipe}e^{-Ts}}{1 + k_{pos}C_{pos}\tilde{G}G_{pipe} + k_{pos}C_{pos}(GG_{pipe}e^{-Ts} - \tilde{G}G_{pipe}e^{-Ts})}.$$  \hspace{1cm} (8)

We assume that the parameters of the plant model $\tilde{G}G_{pipe}$ are same as those of the plant $GG_{pipe}$. Then, (8) is rewritten as

$$\frac{x}{r} = \frac{C_{pos}GG_{pipe}e^{-Ts}}{1 + k_{pos}C_{pos}GG_{pipe}(1 + e^{-Ts} - e^{-Ts})}.$$  \hspace{1cm} (9)

Let $\tilde{T}_e$ denote the error in time delay, i.e., $\tilde{T}_e := \tilde{T}_d - T_d$. Then, (9) is expressed as

$$\frac{x}{r} = \frac{C_{pos}GG_{pipe}e^{-T_e}}{1 + k_{pos}C_{pos}GG_{pipe}(1 + (1 - e^{-T_e})e^{-T_e})}.$$  \hspace{1cm} (10)

Since the AVA has a mechanical resonance at about 2.0 Hz\textsuperscript{(11)}, it is probable that the resonance frequency of the closed-loop control system is similar to the mechanical resonance frequency of the AVA. On the other hand, the breakpoint frequency of the PI compensator (2) is $1/(2\pi T_{pos}) = 0.009$ Hz. In other words, the time constant $T_{pos}$ is large enough. For this reason, the PI compensator in the frequency region including the resonance of the closed-loop control system is approximately expressed as $C_{pos} \approx k_s$. In addition, the natural frequency of the pipe resonance model (7) is $\tilde{\omega}_{pipe}/(2\pi) = 19.9$ Hz, which is larger than the mechanical resonance frequency (i.e., about 2.0 Hz). Thus, the pipe resonance model in the frequency region including the resonance of the closed-loop control system can be written as $G_{pipe} \approx 1$.\hspace{1cm}
Under the approximation, the closed-loop system (10) is expressed as

$$\frac{x}{r} \approx \frac{1}{MS^2 + DS + K + K_{air} + k_{pos}K_{air}G_q/\beta_0V_0s + e} \frac{k_cA_0G_qe^{-T_{es}}}{\beta_0V_0s + e}. \tag{11}$$

Since the flow conductance $c$ is sufficiently small, (11) is rewritten as

$$\frac{x}{r} \approx \frac{1}{MS^2 + DS + K + K_{air} + k_{pos}K_{air}G_q/\beta_0V_0s} \frac{k_cA_0G_qe^{-T_{es}}}{\beta_0V_0s + e}. \tag{12}$$

where $K_{air} := A_0^2/(\beta_0V_0)$ denotes the stiffness of the air spring (5), (6). Since the second term $(k_cA_0G_qe^{-T_{es}})/(\beta_0V_0s + e)$ is a first order lag system including the dead-time, the resonance frequency of closed-loop control system appears by effects of the first term in (12).

Let

$$M(s) = \frac{1}{MS^2 + DS + K + K_{air} + k_{pos}K_{air}G_q/\beta_0V_0s} \frac{k_cA_0G_qe^{-T_{es}}}{\beta_0V_0s + e}. \tag{13}$$

Then, the resonance frequency can be obtained by solving the following equation:

$$\frac{d[M(j\omega)]}{d\omega} = -\frac{1}{(K + K_{air})\omega^3} \left\{ \omega^2 - \frac{2L_1L_2(\omega)}{\omega^2} \right\}^2 + \left\{ \frac{-2\omega^3 \omega_n^2 - L_1L_3(\omega)}{\omega_n^2} \right\} \left[ L_1[1 - 2L_3(\omega)] - \omega L_1[2T_{d}L_3(\omega) + \tilde{T}_eL_3(\omega)] \right]$$

$$= 0, \tag{14}$$

where

$$\omega_n = \sqrt{\frac{K + K_{air}}{M}}, \quad \zeta = \frac{D}{2 \sqrt{M(K + K_{air})}}, \quad L_1 = \frac{k_{pos}k_{air}G_q}{A_0M},$$

$$L_2(\omega) = \cos \left( \frac{2T_d + \tilde{T}_e}{2} \omega \right) \sin \left( \frac{-\tilde{T}_e}{2} \right), \quad L_3(\omega) = \sin \left( \frac{2T_d + \tilde{T}_e}{2} \omega \right) \sin \left( \frac{-\tilde{T}_e}{2} \right),$$

$$L_4(\omega) = \cos \left( \frac{2T_d + \tilde{T}_e}{2} \omega \right) \sin \left( \frac{\tilde{T}_e}{2} \right), \quad L_5(\omega) = \cos((T_d + T_q)\omega) \sin((T_d + T_q)\omega).$$

Since (14) is a nonlinear equation, it is difficult to obtain an analytical solution. Alternatively, we solve the equation (14) by numerical calculation. The simulation result is shown in Fig. 13. From this figure, we can find that the resonance frequency changes depending on the error $\tilde{T}_e$. The experimental result is shown in Fig. 14. From this figure, we can find that when the error $\tilde{T}_e$ is 0 s, the resonance frequency is smallest. On the other hand, the resonance frequency in the case of simulation is not minimized at $\tilde{T}_e = 0$ s. This is because (13) is the approximation model in the frequency region including the mechanical resonance of the AVA. Figure 15 shows Bode plots of non-approximated model (i.e., the closed-loop transfer function (10)). From this figure, we can see that the resonance frequency is smallest at $\tilde{T}_e = 0$ in the case with non-approximation model. Thus, it is probable that the difference between the simulation and the experiment is due to the approximation. The resonance frequency can be found by using Bode plots of the non-approximation model. Moreover, the diagram as in Fig. 14 can be drawn by observing the resonance frequency under the presence of the time error $\tilde{T}_e$. The approximation model does not need to be derived. However, due to the double check of effects of the time error $\tilde{T}_e$ on the resonance frequency, we derived the approximation model and performed the simulation shown in Fig. 13. From the experimental result shown in Fig. 14, the time delay in the Smith predictor should be determined so that the resonance frequency becomes minimized.
5. Tuning of the Time Delay – Tracking Error Based Approach –

In this section, we explain another tuning method of the time delay $\tilde{T}_d$. To tune the time delay, this method utilizes the time integral of table’s tracking error. In the following subsection, at first, it is shown that the control performance changes depending on the error $\tilde{T}_e$ by evaluating the frequency characteristics of the open-loop transfer function for the control system. Secondly, the tuning method and its experimental results are presented.

5.1. Effects of Error on Control Performance

To show the relationship between the error in the time delay and the control performance, we derive the open-loop transfer function of the control system by using the approach of Ref. (14). From Fig. 10, the open-loop transfer function $H(s)$ can be expressed as

$$H(s) = \frac{k_{pos} C_{pos} GG_{pipe} e^{-\tilde{T}_d s}}{1 + k_{pos} C_{pos} GG_{pipe} (1 - e^{-\tilde{T}_d s})}.$$  \hspace{1cm} (15)

Now, Fig. 10 is transformed equivalently to Fig. 16. From Fig. 16, the open-loop transfer function (15) can be rewritten as

$$H(s) = \frac{k_{pos} C_{pos}}{1 + k_{pos} C_{pos} GG_{pipe} (1 - e^{-\tilde{T}_d s})} (GG_{pipe} e^{-\tilde{T}_d s} - \tilde{G} GG_{pipe} e^{-\tilde{T}_d s}).$$ \hspace{1cm} (16)

Under the assumption $GG_{pipe} = \tilde{G} GG_{pipe}$, the open-loop transfer function (16) is rearranged into

$$H(s) = \frac{k_{pos} C_{pos} GG_{pipe} e^{-\tilde{T}_d s}}{1 + k_{pos} C_{pos} GG_{pipe}} (1 - e^{-\tilde{T}_d s}).$$ \hspace{1cm} (17)

Then, the frequency transfer function of (17) can be expressed as

$$H(j\omega) = \frac{k_{pos} C_{pos} GG_{pipe} e^{-j\tilde{T}_d \omega}}{1 + k_{pos} C_{pos} GG_{pipe}} (1 - e^{-j\tilde{T}_d \omega}).$$ \hspace{1cm} (18)
Using Euler’s formula, $1 - e^{-j\tilde{T}_e\omega}$ in (18) can be written as

$$1 - e^{-j\tilde{T}_e\omega} = \left| 2\sin\left(\frac{\tilde{T}_e}{2}\omega\right)\right| \angle \tan^{-1}\left(\frac{1}{\tan\left(\frac{\tilde{T}_e}{2}\omega\right)}\right).$$

(19)

From (18) and (19), we obtain the following gain and phase:

$$|H(j\omega)| = \left| \frac{k_{pos}C_{pos}GG_{pipe}}{1 + k_{pos}C_{pos}GG_{pipe}} \right| \left| 2\sin\left(\frac{\tilde{T}_e}{2}\omega\right)\right|,$$

(20)

$$\angle H(j\omega) = \tan^{-1}\left(\frac{\text{Im}\left[\frac{k_{pos}C_{pos}GG_{pipe}e^{-j\tilde{T}_e\omega}}{1+k_{pos}C_{pos}GG_{pipe}}\right]}{\text{Re}\left[\frac{k_{pos}C_{pos}GG_{pipe}e^{-j\tilde{T}_e\omega}}{1+k_{pos}C_{pos}GG_{pipe}}\right]}\right) - \tan^{-1}\left(\frac{1}{\frac{\tilde{T}_e}{2}\omega}\right) - \omega T_d.$$

(21)

The gain (20) and the phase (21) are utilized so as to investigate the effects of the error $\tilde{T}_e$ by Bode plots. Figure 17 shows simulation results on Bode plots of the open-loop transfer function $H$. From this figure, we can see that the gain margin decreases as the error $\tilde{T}_e$ in the time delay increases. This is because when the error $\tilde{T}_e$ changes, the gain $\left| 2\sin\left(\frac{\tilde{T}_e}{2}\omega\right)\right|$ in (20) also changes (see blue line in Fig. 17). Therefore, the control performance of AVAs is affected by the error $\tilde{T}_e$ in time delay.

5.2. Time Delay Adjustment Using Tracking Error

As mentioned in Subsection 5.1, AVA’s control performances (e.g., settling time, over-shoot, etc.) changes depending on the error $\tilde{T}_e$. For this reason, it is probable that the tracking error of the isolated table is affected by the error $\tilde{T}_e$ in time delay. Hence, in our method, the time delay $\tilde{T}_d$ is determined by using the time integral of the tracking error $e := r - k_{pos}\hat{x}$. The time integral is given by

$$S = \int_{T_1}^{T_2} |e(t)|dt,$$

(22)

where $T_1$ and $T_2$ are constant. Figure 18 shows experimental results of the time integral $S$. The time integral $S$ is calculated by using the results concerning the step response of the isolated table. Since the step signal for the reference $r$ is provided at $t = 2$, the time $T_1$ in (22) is set to 2 s. The time $T_2$ in (22) are set to 5 and 8 s. In this figure, blue (red) solid circle denotes the
time integral $S$ at the time $T_2 = 8$ ($T_2 = 5$). The other parameters used in the experiments are the same as those used in Section 4.

First, from this figure, we can see that the time integral $S$ increases with the time $T_2$. This is because when the time $T_2$ becomes large, the interval $[T_1, T_2]$ of integration becomes wide, resulting in the increase of the time integral $S$.

Then, we can also see that when the time $T_2$ is set to 8 s, the time integral $S$ is minimized at $\tilde{T}_e = 0$ (see blue solid circle in Fig. 18). As mentioned in Subsection 5.1, the gain margin of the open-loop transfer function $H$ becomes small due to the error $\tilde{T}_e$. Consequently, the displacement of the isolated table oscillates, resulting in the increase of time integral $S$. For example, Fig. 19 shows the time response of the control system for the AVA. In the case of the error $\tilde{T}_e = -0.1$ s, the oscillation of the displacement $x$ is observed. Hence, the time integral of the tracking error $e$ increases due to the error $\tilde{T}_e$.

Finally, we can find that when $T_2$ is set to 5 s, the time integral $S$ is almost same regardless of the error $\tilde{T}_e$. Although the displacement $x$ of the isolated table oscillates due to the error $\tilde{T}_e$, the amplitude of the oscillation in the case of $T_2 = 5$ s is smaller than that in the case of $T_2 = 8$ (see Fig. 20). As a result, the time integral $S$ at $T_2 = 5$ s does not change so much.

Therefore, from these results, the large time $T_2$ is effective for the tuning of the time delay $\tilde{T}_d$ in the Smith predictor. Moreover, the time delay $\tilde{T}_d$ should be adjusted so that the time integral $S$ becomes minimized in the case of large time $T_2$.

6. Conclusion

In this paper, we considered the tuning of the Smith predictor for AVAs. In our methods, the approximation model of column resonance is utilized for improving the model accuracy. The time delay in the Smith predictor is determined based on the resonance frequency of closed-loop control system and the time integral of the tracking error. The effectiveness of our methods was demonstrated through experiments.

It is probable that our tuning methods can be used in other devices including the mechanical resonance. Our future work is to employ the proposed methods in other devices so as to show the versatility of our approach. Moreover, we will investigate effects of the pipe length on the resonance frequency and the time integral.
Acknowledgement

This work is supported in part by JSPS Grant-in-Aid for Scientific Research (C) Grant Number 25420178.

References