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CONTROL SYSTEM DESIGN OF A PNEUMATIC SERVO SYSTEM CONSIDERING THE DYNAMIC CHARACTERISTICS OF THE SERVO VALVE

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ABSTRACT

Pneumatic servo systems are applied in several industrial applications and have also been studied extensively for the purposes of compensating the nonlinearity of pneumatic servo systems and improving the controllability with robust control. In their control system design, the transfer function of a pneumatic servo system has been a third-order time lag system, based on the assumption that servo valve dynamics is negligible because it is sufficiently faster than cylinder dynamics. However, the characteristic of a servo valve greatly influences the dynamic characteristic of the pneumatic servo and should be taken into account in the control system design. In this research, a new method of control system design, which deals with the pneumatic servo system as a fifth-order time lag system including a second-order time lag system of a servo valve, has been proposed. The experiment shows that the proposed method is effective. As a result, we can decide all of the control parameters including the proportional gain without trial and error. The obtained proportional gain can be larger, so that the performance of a pneumatic servo has been improved.

KEY WORDS

Pneumatics, Control System Design, Pneumatic Servo System, Servo Valve, Cylinder

NOMENCLATURE

K_{sv}	: Servo valve gain	[m^2/V]
K_n	: Flow gain of servo valve	[m/s/mm^2]
ω_v	: Natural frequency of servo valve	[rad/s]
ω_n	: Natural frequency of pneumatic cylinder	[rad/s]
ζ_v	: Damping coefficient of servo valve	[-]
s	: Laplace operator	[-]
s^*	: Dimensionless Laplace operator	[-]

INTRODUCTION

Since the late '90s, pneumatic servo systems have been used in many fields, such as the active suspension system on the Shinkansen bullet train, molding machines for glass lenses, and amusement robots, because of the numerous advantages of high power, compliant property, and good force controllability. However, the characteristic of a pneumatic servo is nonlinear, which make control difficult. Therefore, control methods of a pneumatic servo, applying advanced control theories such as fuzzy control or robust control, have been investigated [1][2]. In these cases, the transfer function of third-order time lag system derived only from equations on cylinder dynamics is utilized for the control system design, because the pneumatic servo is usually driven by a servo valve that is sufficiently faster than the cylinder dynamics. However, the authors previously clarified that the dynamic characteristic of a servo valve decides the entire dynamic characteristic of a pneumatic servo. Therefore, the transfer function should be a fifth-order time lag system including the dynamic characteristic of a servo valve in the control system design of a pneumatic servo system. The purpose of research is to establish a new control system design method on a pneumatic servo system considering the dynamic characteristic of a servo valve.

Control system design procedure considering the dynamic characteristic of a servo valve

Mathematical model of a pneumatic servo system [3]

In general, the transfer function of a spool type servo valve is treated as the following second-order transfer function:

$$G_v(s) = \frac{K_{sv}\omega_v^2}{s^2 + 2\zeta_v\omega_v s + \omega_v^2} \tag{1}$$

The mathematical model from the flow rate of a servo valve to the displacement of a pneumatic cylinder, which is an actuator in the pneumatic servo system, is described by the flow rate equation on the servo valve, a gas state equation, and the dynamic equation of the cylinder. By linearizing these equations, where it is assumed that the change in the state of air is an isothermal change and that the friction of a cylinder is negligible, the transfer function is obtained as follows:

$$P_n(s) = \frac{K_n\omega_n^2}{s(s^2 + \omega_n^2)} \tag{2}$$

Conventional control system design method

If the assumption that the dynamics of the servo valve $G_v(s)$ is sufficiently fast compared to the dynamics of the cylinder $P_n(s)$ is true, then the transfer function of the controlled system is a third-order time lag system because $G_v(s)$ is negligible. Therefore, a control method in which the feedback signals are velocity and acceleration, in addition to position, is adopted. A block diagram of this control method is shown in Fig.1, and the non-dimensional transfer function of a closed-loop system transformed by the non-dimensional time $\tau = (K_pK_{sn}K_n\omega_n)^{1/3}$ is given as follows:

$$Gc(s^*) = \frac{1}{s^{*3} + \alpha' s^{*2} + \beta' s^* + 1} \tag{3}$$

The velocity or the acceleration feedback gain are then respectively given as follows:

$$K_a = \left(\alpha' K_p^{1/3}\right) / \left(K_{sn}K_n\omega_n^2\right)^{2/3} \tag{4}$$
$$K_v = \left\{\beta'\left(K_{sn}K_nK_p/\omega_n\right)^{2/3} - 1\right\} / \left(K_{sn}K_n\right) \tag{5}$$

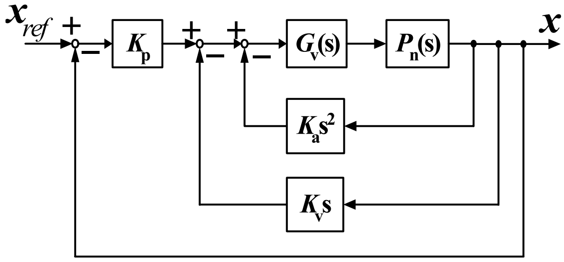

Figure 1 Block diagram of the pneumatic servo system

The transient response of Eq. (3) is determined by the values of α' and β'. These values, usually α'=2 and β'=3, are chosen so as to enable adequate damping in the response. The proportional gain K_p cannot be determined in this conventional method. Consequently, the maximum proportional gain K_p is determined in the stable region by trial and error, because the controllability becomes better as K_p increases.On the servo valve, the control parameters, the natural frequency ω_v, and the damping coefficient ζ_v, are adjusted independently of the pneumatic servo system. The natural frequency ω_v is decided by the maximum force of the actuator driving the valve. In many cases, the damping coefficient of the servo valve ζ_v is set to 0.7 - 0.8 in order to realize a suitable response in second-order time lag system.

Proposed control system design method

In the conventional method, poles of the closed-loop system is bigger and the system response is faster, when the proportional gain K_p becomes large, so that the above assumption that the dynamics of the servo valve $G_v(s)$ is sufficiently fast compared to the dynamics of the cylinder $P_n(s)$, is not true. The closed-loop system should be dealt with as a fifth-order time lag system including the transfer function of the servo valve. In addition, parameters of ω_v and ζ_v on the servo valve should also be included in the adjusted control parameters in the control system design procedure.

The fifth-order time lag system of Eq. (6) is obtained when the closed-loop transfer function of Fig.1 is formed into a non-dimension by the dimensionless time $\tau = (K_cK_p)^{1/5}t$.

$$Gc(s^*) = \frac{1}{s^{*5} + \alpha s^{*4} + \beta s^{*3} + \gamma s^{*2} + \delta s^* + 1} \tag{6}$$

The relationship between α, β, γ, and δ in the above equation and each feedback gain are given by the following equations:

$$K_p = \left\{\left(\omega_{v\max}^2 + \omega_n^2\right)/\beta\right\}^{5/2} / K_c \tag{7}$$
$$\zeta_v = \alpha\left(K_cK_p\right)^{1/5} / \left(2\omega_{v\max}\right) \tag{8}$$
$$K_a = \left\{\gamma\left(K_cK_p\right)^{3/5} - 2\zeta_v\omega_{v\max}\omega_n^2\right\} / K_c \tag{9}$$
$$K_v = \left\{\delta\left(K_cK_p\right)^{4/5} - \omega_{v\max}^2\omega_n^2\right\} / K_c \tag{10}$$

As stated previously, ω_V is decided by the maximum force of the voice coil motor driving a servo valve. That is to say, it can be assumed that this value is known. Then, all of the adjusted control parameters, ω_V and ζ_V, on the servo valve and each feedback gain including a proportional gain K_p can be calculated by giving α, β, γ, and δ in Eqs. (7) - (10). A desired response is obtained when a suitable value of α, β, γ, and δ is selected. There are a number of selection methods, and, for example, the integral of time multiplied by the absolute value of error (ITAE) criterion provides these values.

Effectiveness of the proposed control system design method

Experimental method

The proposed control system design method was tested experimentally using an actual pneumatic servo system, as shown Fig. 2. Figure 3 shows the apparatus of its pneumatic servo system [3]. This pneumatic servo system is general and consists of a pneumatic cylinder, a servo valve, and a controller. However, the pneumatic cylinder is special. An air bearing supports the piston of a pneumatic cylinder and the frictional force is very small. The piston is driven by the supply pressure of 0.4 [MPa]. The cross sectional area of the cylinder is 380 [mm^2], and the stroke of the cylinder is 20 [mm]. The personal computer is used for the controller, and the control algorithm was programmed by xPC-Target of Matlab/Simulink. The cylinder and the servo valve were simultaneously controlled by a single computer. The sampling frequency is 10 [kHz]. The spool position of the servo valve and the cylinder displacement were measured by two digital position sensors and were input to a controller through the I/O-board. The resolution of a position sensor is 0.128 [μm]. The velocity and acceleration signals required for feedback control are obtained from the displacements by using the observer. The observer's poles were set to be sufficiently higher than the pole of the closed-loop system. The output of the controller is output to the servo valve through the D/A-board. Finally, the pneumatic cylinder is positioned.

Validity of a mathematical model

In order to verify the validity of the mathematical model of a pneumatic servo system, a step response test was conducted. The results for the servo valve, where $\omega_V = 140$ [Hz] and $\zeta_v = 1$, are shown in Fig. 4. The origin is the null position of the servo valve. This figure shows that the response of the mathematical model is approximately the same as the experimental response.The response of the closed-loop system is shown in Fig. 5. The step width is 5 [mm], and the one of the stroke ends of the cylinder is the origin point of the cylinder position. The values of ω_V and ζ_v for the servo valve were same as those shown in Fig. 4. The proportional gain $K_p = 0.03$ is chosen, and the feedback gains on velocity and acceleration are calculated from Eqs. (4) and (5) as $\alpha' = 2$ and $\beta' = 3$, respectively. The experimental results show good agreement with the response of the mathematical model shown in Eq. (6), although the pneumatic servo system has some nonlinearity. The mathematical model including the servo valve is therefore valid.

Figure 2 Photograph of the tested pneumatic servo system

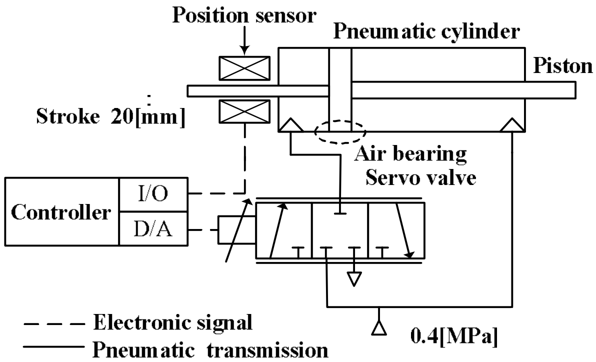

Figure 3 Apparatus of the pneumatic servo system

Effectiveness of the proposed control design method

The step response obtained by the proposed control system design method is shown in Fig. 6. In this design, α, β, γ, and δ are decided by the ITAE criterion, and the natural frequency of the servo valve ω_V is 140 [Hz], considering the validity of the mathematical model. Control parameters, such as the velocity or the acceleration feedback gain, were calculated directly by Eqs. (7) - (9) without the need for a trial-and-error process. The proportional gain then became $K_p = 0.13$ [V/m]. In Fig.5, the dash-dotted line shows the response of Eq. (6) used in the control system design. The experimental results are similar to those of Eq. (6) with respect to the overshoot or undershoot of the response upon settling. This difference may be due to a nonlinearity of the pneumatic system. We concluded that the proposed control design method is effective and that the pneumatic servo system should be dealt with as a fifth-order time lag system.

Comparison with the conventional method

The results of the conventional control system design method are compared with those of the proposed system in Fig. 7. In the conventional method, the servo valve dynamic characteristic of $\omega_v = 140$ [Hz] is the same as that of the proposed method. The damping coefficient of a servo valve was adjusted to $\zeta_v = 0.75$ in a second-order time lag system.

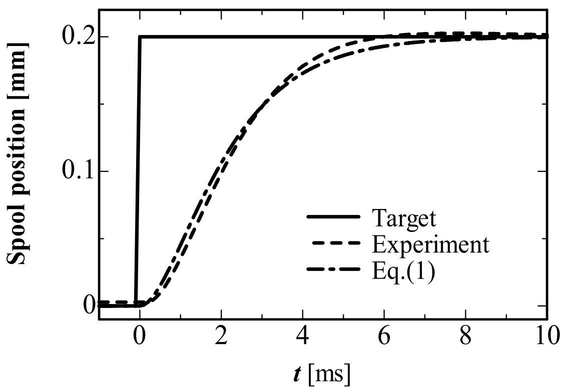

Figure 4 Step response of the mathematical model on the servo valve compared with the experiment

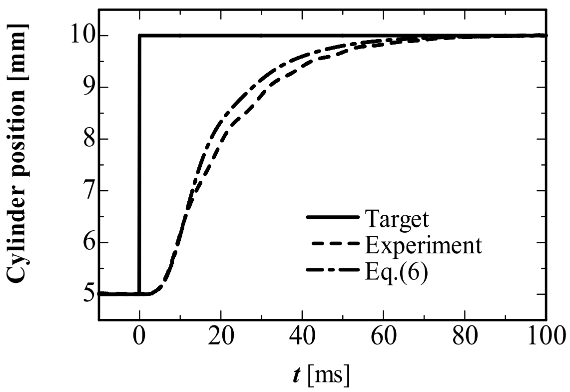

Figure 5 Step response of the mathematical model on the pneumatic servo system compared with the experiment

The velocity and acceleration gains were calculated from Eqs. (4) and (5) by setting $\alpha' = 2$ and $\beta' = 3$. The proportional gain K_p was obtained by trial and error, keeping the velocity and the acceleration gains constant, and $K_p = 0.10$ [V/m] was selected. When the proportional gain K_p is set to be greater, a small fluctuation occurred in the response. Consequently, the proportional gain is smaller in the case of the conventional control system design method than in the case of the proposed method. The rise time and the settling time of the proposed method are faster, as compared to the conventional method. Figure 7 reveals the superiority of the proposed technique.

Conclusion

A new control system design method was proposed in which the transfer function of the pneumatic servo system is considered to be a fifth-order time lag system by dealing with the dynamic characteristics of servo valve as a second-order time lag system explicitly. The results of an experiment indicate that the proposed method is very effective. In particular, the proposed

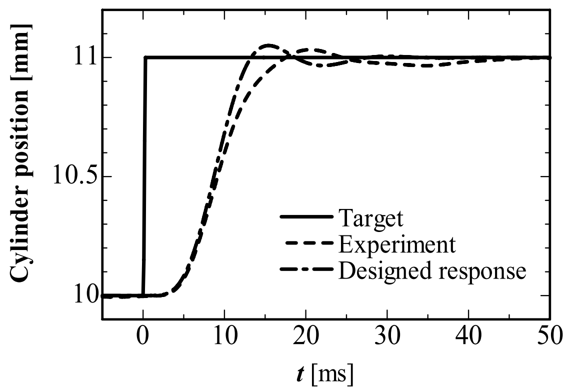

Figure 6 Step response designed by the proposed method

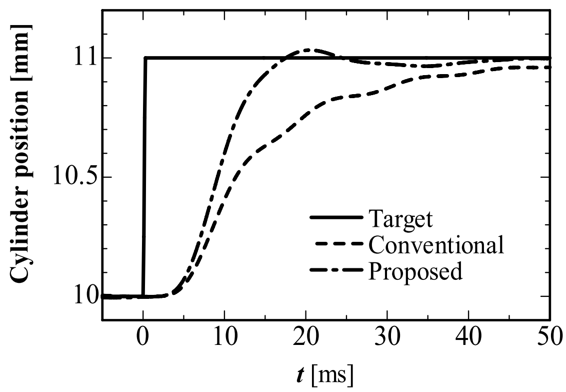

Figure 7 Comparison of the proposed method with the conventional method

method can be used to decide all of the control parameters without the need for a trial-and-error process. Furthermore, the proportional gain obtained by the proposed method is larger than that obtained by the conventional method through trial and error and is near maximum in the stable region. Therefore, the controllability of the pneumatic servo has been improved.

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