The Dynamics and Sensitivity Analysis of a Plunger-type Pressure Control Valve

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The Dynamics and Sensitivity Analysis of a Plunger-type Pressure Control Valve

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This paper is concerned with an analytical study of the dynamics of a plunger-type pneumatic pressure control valve, already proposed by the authors, which maintains a constant secondary pressure lower than the primary pressure, in spite of the change of the primary pressure or the fluctuation of the load connected with this valve. At first, the dynamics of the secondary pressure and the plunger movement, caused by changes of the primary pressure and the load, is analyzed by means of the state variable method. Furthermore, for improvement of responsibility, the conditions for adjusting system parameters are found through the root locus method. Next, the sensitivity analysis is applied to the investigation of the influences of parameter variations on the dynamics of the control valve. By means of simulation studies of sensitivity functions, we have shown which parameters are really significant and also which are not. Throughout these analytical and simulation studies, guidelines for improving the performance of a plunger-type pressure control valve are obtained.

1. Introduction

The plunger-type pressure control valve is a pneumatic control valve which holds a constant pressure by controlling the position of the plunger equipped with a spool valve instead of the diaphram, to improve the characteristics of the diaphram-type pressure reducing valve used widely. The authors[1] had already made this control valve on trial, and examined the performance, especially,

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the static characteristics from the theoretical and experimental viewpoints. As a result, we had obtained both the excellent pressure-flow characteristics and the pressure-adjustment one, comparing with the diaphram-type control valve. In addition, it had been shown that the regulation was easy to obtain a prescribed pressure in this valve.

In this paper, we present the analytical study for the dynamics of a plunger-type pressure control valve by means of the state variables method, in which the dynamic behaviors of both the secondary pressure and the plunger movement, caused by changes of the primary pressure and the load, are analyzed. Furthermore, the sensitivity analysis[(2)(3)] is applied to the investigation of the influences of parameter variations on the dynamics of the control valve. Throughout these studies, the guidelines for improving the performance of a plunger-type pressure control valve are shown.

Both the structure and its principle of action of the plunger-type pressure control valve (which is simply called the control valve, hereafter) were already explained in details[(1)]. Then, we will give only the simple explanation for the principle of the control valve, which is necessary to understand the contents of this paper. The schematic diagram of the control valve is shown in Fig.1. If the secondary pressure (the prescribed pressure) P_2 may be changed by the load fluctuation, the change of the pressure is transmitted to the bottom of the plunger through the oil-filled

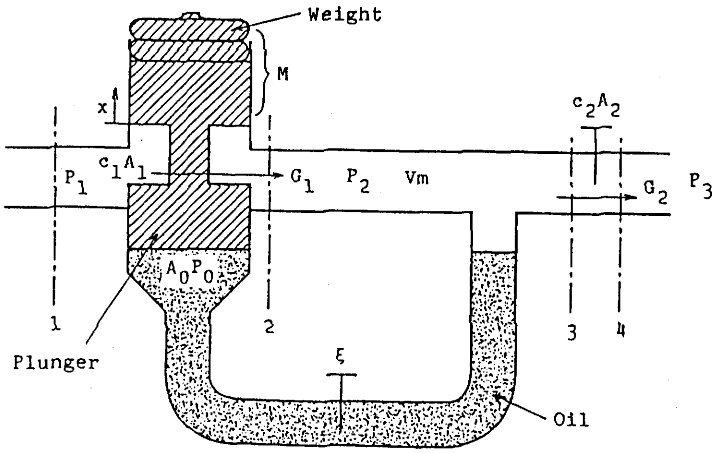

Fig.1 Schematic diagram of Plunger-type pressure control valve

pipe (feedback pipe) and the plunger goes up or down from the equilibrium position. Hence, as the spool valve of the plunger becomes extended or narrow quickly and then the volume of the air supply increases or decreases, the secondary pressure P_2 returns to the prescribed one. Thus, this control valve has the function which maintains a constant secondary pressure. As the alteration of the command pressure can be determined by selecting a moderate counter weight on the top of the plunger, it becomes possible to realize a remote control by using the moderate alterating mechanism of the counter weight. The pressure-adjustment sensitivity is always raised by giving the slow rotation to the plunger. Principal symbols used here are listed below:

A_0 : bottom area of plunger m^2
A_1 : openning area of spool valve m^2
A_2 : openning area of outlet throttle valve m^2
ξ : coefficient of feedback pipe $m^5/(N\cdot s)$
G_1 : mass flow rate of air through spool valve kg/s
G_2 : mass flow rate of air through outlet throttle valve kg/s
M : mass of movable portion kg
P_0 : pressure acted to the bottom of plunger MPa
P_1 : primary pressure MPa
P_2 : secondary pressure MPa
P_3 : output pressure of outlet throttle valve MPa
Vm : volume of air chamber at the secondary side m^3
f : friction coefficient of movable portion $N\cdot s/m$
c_1 : flow coefficient of plunger port
c_2 : flow coefficient of outlet throttle valve

2. Description of the System Model

With respect to the secondary pressure P_2 and the plunger displacement x as shown in Fig.1, the dynamics of each portions can be represented by linearized equations, taking the small deviations around stationary states into consideration.

At first, the plunger shifts from x to x+Δx due to the small deviation ΔP_0 of the pressure acted under the bottom of the plunger. Hence, associated with the dynamics of the movable portion, the following equation yields,

$$M \frac{d^2(\Delta x)}{dt^2} + f \frac{d(\Delta x)}{dt} = A_0 \Delta P_0 \qquad (1)$$

Nextly, from the material balance of air, the following relationship holds,

$$K_1\frac{d(\Delta P_2)}{dt^2} + \Delta P_2 = K_2\Delta x - K_3\Delta P_3 + K_4\Delta P_1 - K_5\Delta c_2 A_2 \tag{2}$$

Furthermore, it follows that the relationship between the displacement of the plunger and the oil pressure holds as

$$\frac{d(\Delta x)}{dt} = \frac{\xi}{A_0}(\Delta P_2 - \Delta P_0)\,. \tag{3}$$

In Eqs.(1) to (3), "Δ" denotes the small deviation from the stationary states. Also, $K_1 \sim K_5$ are coefficients determined by mass flow rates of air G_1, G_2, pressures P_1, P_2, P_3, and the openning areas of spool valve or throttle valve A_1, A_2. (see Appendix)

Here, in the case where the ratio between the primary pressure P_1 and the secondary one P_2 satisfies $P_2/P_1 \leqq 0.528$, the velocity of a flowing air becomes supersonic. Then, it must be careful about the treatments of K_1 to K_5. In this study, we treat only the dynamics in the case where the velocity of flowing air is subsonic, that is, $P_2/P_1 > 0.528$.

In order to express the system in state variable form, the state vector can be taken as

$$x_1 = \Delta x(t), \quad x_2 = \dot{x}_1, \quad x_3 = \Delta P_2(t)$$
$$u_1 = \Delta P_3, \quad u_2 = \Delta P_1, \quad u_3 = \Delta c_2 A_2\,.$$

Using the above notation, Eqs.(1) to (3) can be rewritten by

$$\frac{dx_1}{dt} = x_2 \tag{4}$$

$$\frac{dx_2}{dt} = -\left(\frac{f}{M} + \frac{A_0^{\,2}}{M\xi}\right)x_2 + \frac{A_0}{M}x_3 \tag{5}$$

$$\frac{dx_3}{dt} = \frac{K_2}{K_1}x_1 - \frac{1}{K_1}x_3 - \frac{K_3}{K_1}u_1 + \frac{K_4}{K_1}u_2 - \frac{K_5}{K_1}u_3\,. \tag{6}$$

From Eqs.(4) to (6), the state equation of the pressure control valve can be represented as

$$\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{u}\,, \tag{7}$$

where

$$\mathbf{x} = \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix}, \quad \mathbf{A} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & -(f+A_0^{\,2}/\xi)/M & A_0/M \\ K_2/K_1 & 0 & -1/K_1 \end{bmatrix}$$

$$u = \begin{bmatrix} u_1 \\ u_2 \\ u_3 \end{bmatrix}, \quad B = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ -K_3/K_1 & K_4/K_1 & -K_5/K_1 \end{bmatrix} .$$

Through finding out the solution of the differential equation (7), we can obtain the transient responses for both the variation of the secondary pressure $\Delta P_2(=x_3)$ and the displacement of the plunger $\Delta x(=x_1)$, caused by the disturbance to the system. Furthermore, for the purpose of the improvement of responsibility, the conditions for adjusting system parameters are found through the root locus method.

3. Analysis of System Dynamics

For the convenience of analysis, the insight into the free response of the vector state equation (7) can be considered by setting $u=0$ in Eq.(7), i.e.,

$$\dot{x} = \frac{d}{dt}\begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ a_{31} & a_{32} & a_{33} \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} = Ax, \tag{8}$$

where $a_{11}=0$, $a_{12}=1$, $a_{13}=0$, $a_{21}=0$, $a_{22}=-(f+A_0{}^2/\xi)/M$, $a_{23}=A_0/M$, $a_{31}=K_2/K_1$, $a_{32}=0$, and $a_{33}=-1/K_1$. The eigenvalues of the matrix A, letting λ be eigenvalue, can be obtained by

$$\det(\lambda I - A) = \det\begin{bmatrix} \lambda & -a_{12} & 0 \\ 0 & \lambda-a_{22} & -a_{23} \\ -a_{31} & 0 & \lambda-a_{33} \end{bmatrix} = 0. \tag{9}$$

From Eq.(9), the characteristic equation is as follows,

$$\lambda^3 - (a_{22}+a_{33})\lambda^2 + a_{22}a_{33}\lambda - a_{12}a_{23}a_{31} = 0 . \tag{10}$$

By finding the eigenvalues of Eq.(10),[4] we can construct the solution of Eq.(7).

The general solution of Eq.(7) can be expressed as follows,

$$x(t) = e^{A(t-t_0)}x_0 + \int_{t_0}^{t} e^{A(t-\tau)}Bu(\tau)d\tau. \tag{11}$$

The state transition matrix $\Phi(t)$ at $t_0=0$ is given by

$$\Phi(t) = e^{At}. \tag{12}$$

Since the determinant (12) is a function with infinite power series of the matrix A, it can be, applying the Sylvester's expansion theorem,[5] obtained in the following form,

$$\Phi(t) = \sum_{i=1}^{3} f(\lambda_i)F(\lambda_i), \tag{13}$$

where

$$f(\lambda_i) = e^{\lambda_i t} \tag{14}$$

$$F(\lambda_i) = \prod_{\substack{j=1 \\ j\neq i}}^{3} \left[\frac{A-\lambda_i I}{\lambda_i-\lambda_j}\right], \tag{15}$$

where λ_i are roots of Eq.(10). Let the state transition matrix $\Phi(t)$ be

$$\Phi(t) = \begin{bmatrix} \phi_{11} & \phi_{12} & \phi_{13} \\ \phi_{21} & \phi_{22} & \phi_{23} \\ \phi_{31} & \phi_{32} & \phi_{33} \end{bmatrix}. \tag{16}$$

Then, the (i,j) elements of Eq.(16) can be determined as follows.

(1) The case where eigenvalues are three real ones, i.e., $\lambda=(\eta_1, \eta_2, \eta_3)$.

$$\phi_{ij} = \sum_{\ell=1}^{3} \frac{\{a^*_{ij} - a_{ij}\sum\limits_{\substack{m=1 \\ m\neq\ell}}^{3} \eta_m\} e^{\eta_\ell t}}{\prod\limits_{\substack{n=1 \\ n\neq\ell}}^{3} (\eta_\ell - \eta_n)} \qquad (i\neq j) \tag{17}$$

$$\phi_{ij} = \sum_{\ell=1}^{3} \frac{\{a^*_{ii} - a_{ii}\sum\limits_{\substack{m=1 \\ m\neq\ell}}^{3} \eta_m + \prod\limits_{\substack{n=1 \\ n\neq\ell}}^{3} \eta_n\} e^{\eta_\ell t}}{\prod\limits_{\substack{q=1 \\ q\neq\ell}}^{3} (\eta_\ell - \eta_q)} \qquad (i=j) \tag{18}$$

(2) The case where one is a real and others are complex conjugate, i.e., $\lambda=(\eta, \mu\pm j\nu)$.

$$\begin{aligned}\phi_{ij} = {} & [(a^*_{ij} - 2\mu a_{ij})(e^{\eta t} - e^{\mu t}\cos\nu t) + \{(\mu-\eta)a^*_{ij} \\ & + (\nu^2-\mu^2+\eta^2)a_{ij}\}e^{\mu t}\sin\nu t/\nu]\{\nu^2+(\mu-\eta)^2\} \qquad (i\neq j)\end{aligned} \tag{19}$$

$$\begin{aligned}\phi_{ij} = {} & [\{a^*_{ii} - 2\mu a_{ii} + (\mu^2+\nu^2)\}e^{\eta t} - \{a^*_{ii} - 2\mu a_{ii} + \eta(2\mu-\eta)\}e^{\mu t}\cos\nu t \\ & + \{(\mu-\eta)a^*_{ii} + (\nu^2-\mu^2+\eta^2)a_{ii} + \eta(\mu^2-\eta\mu-\nu^2)\}e^{\mu t}\sin\nu t/\nu] \\ & /\{\nu^2+(\mu-\eta)^2\} \qquad (i=j)\end{aligned} \tag{20}$$

where

$$a_{ij}^{*} \triangleq \sum_{k=1}^{3}(a_{ik}a_{kj}) \ . \tag{21}$$

Using Eqs.(17) to (20), we can represent the solution form of Eq.(11). Now, we shall consider the transient responses of both the variation of the secondary pressure ΔP_2 and the plunger displacement Δx, due to the fluctuation of the load pressure ΔP_3. Letting initial condition be $x_0=0$ at $t_0=0$, Eq.(11) becomes,

$$x(t) = \int_0^t \Phi(t-\tau)\, Bu(\tau)d\tau \ . \tag{22}$$

Hence, the transient response of the plunger displacement can be given by

$$x_1(t) = \int_0^t \{-\frac{K_3}{K_1}u_1\phi_{13}(t-\tau)+\frac{K_4}{K_1}u_2\phi_{13}(t-\tau)-\frac{K_5}{K_1}u_3\phi_{13}(t-\tau)\}d\tau, \tag{23}$$

and the transient response for the variation of the secondary pressure can be obtained by

$$x_3(t) = \int_0^t \{-\frac{K_3}{K_1}u_1\phi_{33}(t-\tau)+\frac{K_4}{K_1}u_2\phi_{33}(t-\tau)-\frac{K_5}{K_1}u_3\phi_{33}(t-\tau)\}d\tau. \tag{24}$$

Here, $\phi_{13}(t-\tau)$ and $\phi_{33}(t-\tau)$ are obtained by Eqs.(17) to (20).

The practical responses for the control valve made on trial can be calculated utilizing a set of parameters in Table 1 of the reference (1). Here, the resistance coefficient ξ for the throttle of the valve in feedback pipe, which gives an important effect for the transient response, can be determined through the root locus based on Eq.(10).

Figure 2 shows the root locus in each cases where the prescribed values of the secondary pressure are P_2=0.46(M=3.61kg), 0.41 (M=3.11kg) and 0.36(M=2.61kg)MPa. Here, as the gain parameter, the resistance coefficient ξ was taken for the throttle of the valve in feedback pipe. Although the alteration of the prescribed secondary pressure P_2 is done by the change of the mass M, then the system parameters take all different values. Hence, the root locus draws different curves in each cases of P_2, as shown in Fig.2. Now, we shall examine the dynamic behaviors in the case where P_2=0.41MPa ($\cong$4.2kg/cm^2). As examples, the transient responses for the variation of the secondary pressure ΔP_2 were calculated by using Eqs.(17) to (21) and Eq.(24), subjected to the load fluctuation ΔP_3=50000Pa. These results are shown in Fig.3, where the damping constants are set as ζ=0.3 and 0.7. From Fig.3, it can be understood that the favorable responses are, by experience, obtained at $\xi=0.55\times10^{-10}$ $m^5/(N\cdot s)$ ($\zeta=0.3$) if the speed responsibility is important and at $\xi=0.17\times10^{-10}$ $m^5/(N\cdot s)$ ($\zeta=0.7$) if the damping one is important.

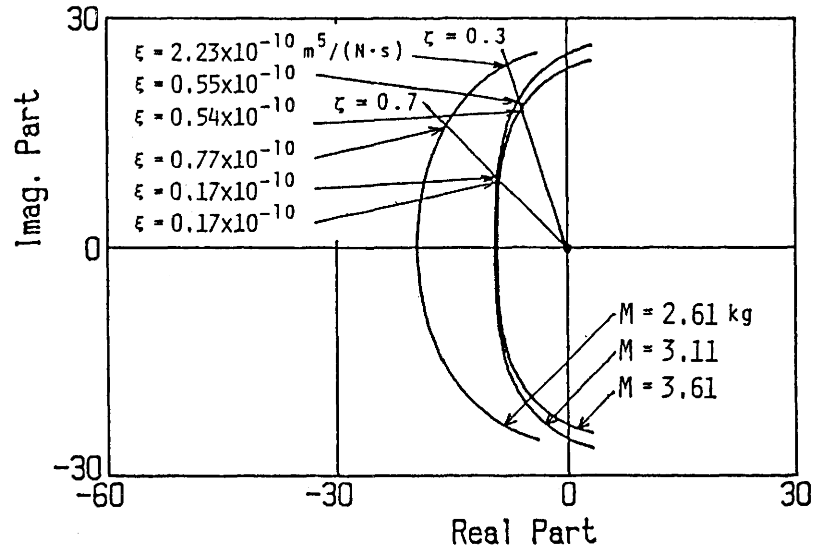

Fig.2 Root-locus for eigenvalue of system

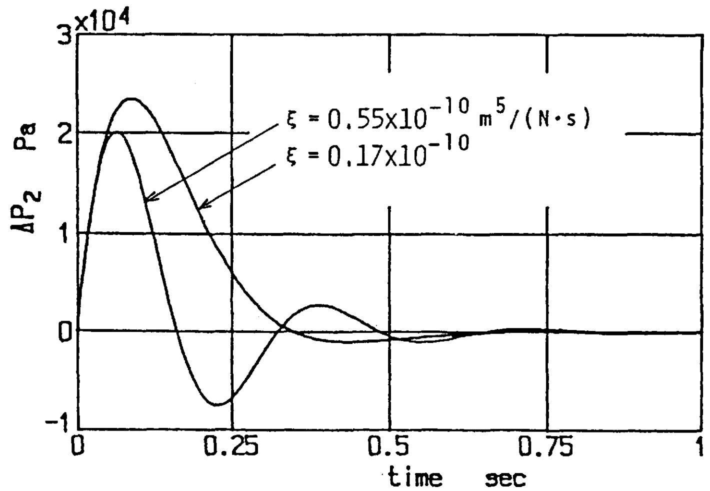

Fig.3 Transient response of ΔP_2 to the deviation, ΔP_3=50000 Pa

4. Sensitivity Analysis

The pressure control valve is normally designed through the analysis of dynamics by mathematical models expressed as constant coefficient state equations, as described in the above chapter. However, the parameters of models based on physical laws or the technical characteristics given by the manufacturers are quite inaccurately known in the design situation. On the other hand, the parameters change with the external conditions and with time. For instance, the environmental conditions such as the minute variations of the primary pressure or secondary one, temperature fluctuation and air leakages change the dynamic characteristics of the system. In order to obtain a reliable system agreeing with the design, it is important to be able to estimate the influences of parameter variations on the dynamic behaviors of the system. In this section, parameter influences can be studied by sensitivity analysis.[2]

Now, we set system parameters as

$$\alpha_1=M,\quad \alpha_2=f,\quad \alpha_3=A_0,\quad \alpha_4=\xi,\quad \alpha_5=K_1$$
$$\alpha_6=K_2,\quad \alpha_7=K_3,\quad \alpha_8=K_4,\quad \alpha_9=K_5. \qquad (25)$$

From Eqs.(4) to (6), a mathematical model of the system is described by

$$\dot{\mathbf{x}} = \mathbf{f}(\mathbf{x},\mathbf{u},\boldsymbol{\alpha}) \qquad (26)$$

where $\mathbf{x}=[x_1,x_2,x_3]^T$,

$\mathbf{u}=[u_1,u_2,u_3]^T$,

$\boldsymbol{\alpha}=[\alpha_1,\alpha_2,\alpha_3,\alpha_4,\alpha_5,\alpha_6,\alpha_7,\alpha_8,\alpha_9]^T$,

and where

$$\mathbf{f}(\mathbf{x},\mathbf{u},\boldsymbol{\alpha}) = \begin{bmatrix} x_2 \\ -\left(\frac{\alpha_2}{\alpha_1}+\frac{\alpha_3^2}{\alpha_1\alpha_4}\right)x_2+\frac{\alpha_3}{\alpha_1}x_3 \\ \frac{\alpha_6}{\alpha_5}x_1-\frac{1}{\alpha_5}x_3-\frac{\alpha_7}{\alpha_5}u_1+\frac{\alpha_8}{\alpha_5}u_2-\frac{\alpha_9}{\alpha_5}u_3 \end{bmatrix}. \qquad (27)$$

Denote the vector sensitivity functions

$$z^j = \left(\frac{\partial \mathbf{x}}{\partial \alpha_j}\right)_n, \qquad j=1,\cdots\cdots,9, \qquad (28)$$

where the suffix n represents the normal value and j the order of parameter vector. Assuming that u is independent on α and differentiating equation (26) partially with respect to α, we obtain the sensitivity equations in the form,

$$\dot{z}^j = \left(\frac{\partial f}{\partial x}\right)_n z^j + \left(\frac{\partial f}{\partial \alpha_j}\right)_n, \qquad j=1,\cdots\cdots,9 \tag{29}$$

where $(\partial f/\partial x)_n$ is the Jacobian matrix evaluated on the nominal solution. From Eq.(27), the first term of Eq.(29) is obtained as follows.

$$\frac{\partial f}{\partial x} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & -\frac{\alpha_2}{\alpha_1}-\frac{\alpha_3^2}{\alpha_1\alpha_4} & \frac{\alpha_3}{\alpha_1} \\ \frac{\alpha_6}{\alpha_5} & 0 & -\frac{1}{\alpha_5} \end{bmatrix} . \tag{30}$$

Also, the second term of Eq.(29) is obtained by differentiating Eq.(27) partially with respect to $\alpha_j$$(j=1,\cdots\cdots,9)$. The initial conditions for Eq.(29) are

$$z_0^j = \left(\frac{\partial x_0}{\partial \alpha_j}\right)_n, \qquad j=1,\cdots\cdots,9. \tag{31}$$

Now, the initial state is assumed to be zero state

$$x_0 = 0 , \tag{32}$$

so that the initial conditions of the sensitivity equations are

$$z_0^j = 0, \quad j=1,\cdots\cdots,9. \tag{33}$$

Using Eqs.(26) to (33), we can solve, with the computer simulation, the vector sensitivity functions $z^j$$(j=1,\cdots\cdots,9)$.

In addition to the sensitivity functions, we often have to know the differential variation δx of state variables, which is due to the parameter variation,

$$\delta\alpha = \alpha - \alpha_n. \tag{34}$$

Using Taylor's expansion theorem, the first approximation of the variation δx may be written

$$\delta x \cong \left(\frac{\partial x}{\partial \alpha}\right)_n \delta\alpha , \tag{35}$$

where $(\partial x/\partial\alpha)_n$ is the $m\times j$ matrix of the sensitivity functions. Here, m is the order of the state vector. Once we know the vector sensitivity functions, we can, according to Eq.(35), calculate the first order approximation of the variation δx.

5. Simulation Experiments

Simulation Experiments have been done, taking notice of the variation of the secondary pressure ΔP_2 (described by the state variable x_3), which is the most interesting dynamic behavior in this control valve. The sensitivity analyses were performed by selecting the step-like variation of the load u_1 as the input signal. As the values of parameters in Eq.(27), the physical constants of the control valve made on trial were used.[1] Figure 4 shows the transient response $\delta\Delta P_2$ and the sensitivity functions Z^j to a step input ΔP_3=50000Pa, which were obtained from computations of Eq.(29). To simplify the comparison, each sensitivity functions are shown as non-dimensional form from 1 to -1, by dividing with each maximum values. The size of the variation of the nominal step response of ΔP_2 is given, taking into account Eq.(35), by

$$\delta\Delta P_2 = \sum_{j=1}^{7} z_3{}^j \delta\alpha_j \ . \tag{36}$$

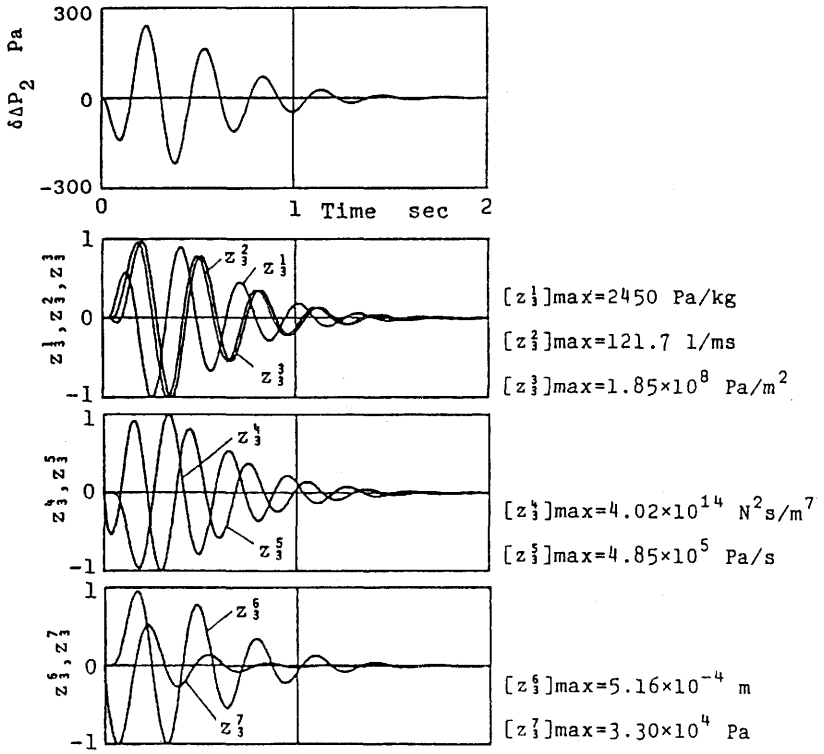

Fig.4 Transient response $\delta\Delta P_2$ and sensitivity function to a step input ΔP_3=50000 Pa

This implies that the variation of the secondary pressure ΔP_2 is represented as the total sum of the product of each parameter variations by the sensitivity function. In order to examine the rate of any influence of parameter variation on the response of the secondary pressure ΔP_2, we get the equation for comparisons,

$$\left|\frac{\delta \Delta P_2}{P_{2s}}\right|_{max,j} = \frac{|Z_3{}^j|_{max} \cdot |\delta\alpha_j|}{|P_{2s}|}, \quad j=1,\cdots\cdots,9, \tag{37}$$

where P_{2s} is the value of the secondary pressure in the steady state. In this case, we set P_{2s}=0.41MPa.

By means of simulation studies, it has been found that the first order sensitivity model is still very accurate even when the variations in the parameter vector α are 10 percent.[6] So, if we are looking at the influences of 1 percent parameter changes, we can be sure that the first order sensitivity model gives results accurate enough for comparisons. Giving 1 percent change for the parameters, i.e.,

$$|\delta\alpha_j| = 0.01\alpha_j \tag{38}$$

and taking the maximum values $|Z_3{}^j|_{max}$ according to Fig.4, we can compute maximum variations $|\delta\Delta P_2/P_{2s}|_{max,j}$ by Eq.(37). The result is shown in Fig.5 with the aid of histograms, in which one column represents a maximum variation in the nominal step response of ΔP_2

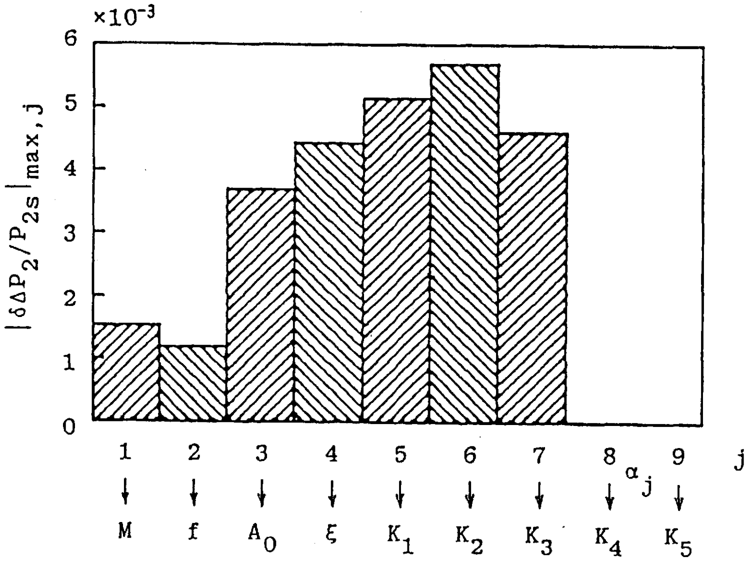

Fig.5 Comparison of parameter influences on the secondary pressure ΔP_2

scaled by the steady-state value P_{2s} and caused by a one percent variation in one parameter at a moment of time when the sensitivity function corresponding to the parameter reaches its maximum absolute value. It may be noted that different sensitivity functions reach their maximum absolute value at different moments of time. However, the histograms clearly show which parameters are really significant and show are not.

From Fig.5, it can be seen that the coefficient $K_2(\alpha_6)$ with respect to the open area of the plunger port gives the maximum influence for the variation of the secondary pressure ΔP_2. Also, the coefficients $K_1(\alpha_5)$ and $K_3(\alpha_7)$ reflect strongly, which are related to the volume of air chamber at the secondary side and the variation of the load pressure, respectively. Furthermore, the bottom area $A_0(\alpha_3)$ of plunger in the feedback pipe influences fairly, similarly to the coefficient $\xi(\alpha_4)$ of the throttle valve in the feedback pipe. On the otherhand, it can be understood that the mass $M(\alpha_1)$ and the viscous resistance coefficient $f(\alpha_2)$ of the movable portion are almost non-sensitive.

From results described above, the rate of the influence of each parameters was clarified from the viewpoints of the system design. In order to obtain the favorable system dynamics, since it is difficult to regulate K_1, K_2 and K_3 which depend strongly on the air flow or the air pressure, it is especially important to regulate the resistance coefficient ξ, that is, the throttle of the valve in the feedback pipe.

6. Conclusions

In this research, for the plunger-type pressure control valve proposed already by authors, we derived the system equation by the state space model and clarified the system dynamics. Especially, we analized both the dynamics of the secondary pressure and the plunger displacement by the load fluctuation and examined their dynamic behaviors by simulation experiments. Also, we have found, by root locus method, the adjusting conditions of parameters to obtain the favorable response characteristics. Furthermore, for the trial of the control valve practically, we can not necessarily design a set of parameters precisely. Taking into account these facts, the sensitivity analysis was applied to examine the influence of parameters to the system dynamics, when the parameters change with the external conditions and with time. From this analysis, it was clarified that the coefficients with respect to the volume of air chamber in the secondary side and the bottom area of the plunger give strongly effects on the system dynamics. In order to improve the responses of the control valve in the case of the practical use, it has been known that it was mostly effective to adjust the throttle of the valve in the feedback pipe.

From the analytical results presented in this paper, the guideline for the design could be obtained to improve the performance of the plunger-type pressure control valve.

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Appendix

The coefficients from K_1 to K_5 are given below;

$$K_1 = \frac{V_m}{RT} / \{(\frac{\partial G_2}{\partial P_2}) - (\frac{\partial G_1}{\partial P_2})\}$$

$$K_2 = (\frac{\partial G_1}{\partial \alpha_1 A_1})(\frac{\partial \alpha_1 A_1}{\partial x}) / \{(\frac{\partial G_2}{\partial P_2}) - (\frac{\partial G_1}{\partial P_2})\}$$

$$K_3 = (\frac{\partial G_2}{\partial P_3}) / \{(\frac{\partial G_2}{\partial P_2}) - (\frac{\partial G_1}{\partial P_2})\}$$

$$K_4 = (\frac{\partial G_1}{\partial P_1}) / \{(\frac{\partial G_2}{\partial P_2}) - (\frac{\partial G_1}{\partial P_2})\}$$

and

$$K_5 = (\frac{\partial G_2}{\partial \alpha_2 A_2}) / \{(\frac{\partial G_2}{\partial P_2}) - (\frac{\partial G_1}{\partial P_2})\}.$$