A STUDY OF PLUNGER-TYPE PRESSURE CONTROL VALVE

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A Study of Plunger-Type Pressure Control Valve

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In this paper, we propose a plunger-type pneumatic pressure control valve which is different from the diaphragm-type pressure reducing valve used widely. The feature of this valve is to hold a constant pressure by controlling the position of the plunger equipped with a spool valve through the oil pressure in the feedback circuit. Also, in this valve, it is easy to obtain an arbitrary constant pressure by the simple manipulation.

Firstly, both the structure and its principle of the action of the plunger-type pressure control valve are explained. Secondly, the dynamics of the plunger is analyzed linearly and the stability conditions are derived for a set of parameter values of the control valve made on trial. Finally, experimental studies are performed and both the pressure-flow characteristics and the pressure-adjustment characteristics are examined.

Throughout these analytical and experimental studies, the validity of the plunger-type pressure control valve is verified.

I. Introduction

Pressure control valve is a self-actuated regulator which preserves a constant secondary pressure lower than the primary pressure, in spite of the change of primary pressure or the condition of air consumption. Usually, the diaphragm-type pressure reducing valve has been widely used as a pressure control valve. However, this valve does not always have good characteristics, because the regulation of the valve to obtain a prescribed pressure is not

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easy and also the secondary pressure fluctuates by the variation of flow rate.[1]

In this paper, we propose a pressure control valve with the plunger equipped with a spool valve instead of the diaphragm. The outline of pressure control valve proposed here is illustrated as follows. That is, the secondary pressure of this valve is transformed into the oil pressure in the feedback circuit, and the oil pressure causes the plunger to move vertically. From this vertical motion of the plunger, the spool valve opens or shuts and adjusts the volume of air supply. As a result, the secondary pressure keeps a constant rate irrelevant to the air consumption. This valve has also the function of the relief valve. Then, when the secondary pressure becomes higher than the prescribed pressure, the valve opens to the atmosphere by rising of the plunger. As the prescription of the secondary pressure may be determined by a counter weight added to the top of the plunger, it is easy to obtain precisely a prescribed constant pressure by choosing arbitrarily weight. Moreover, the plunger goes smoothly up and down, because the mechanical rotation is continually given for the plunger and then the static friction force is eliminated for the vertical motion. In this way, the plunger-type pressure control valve possesses structually an excellent pressure adjusting characteristics.

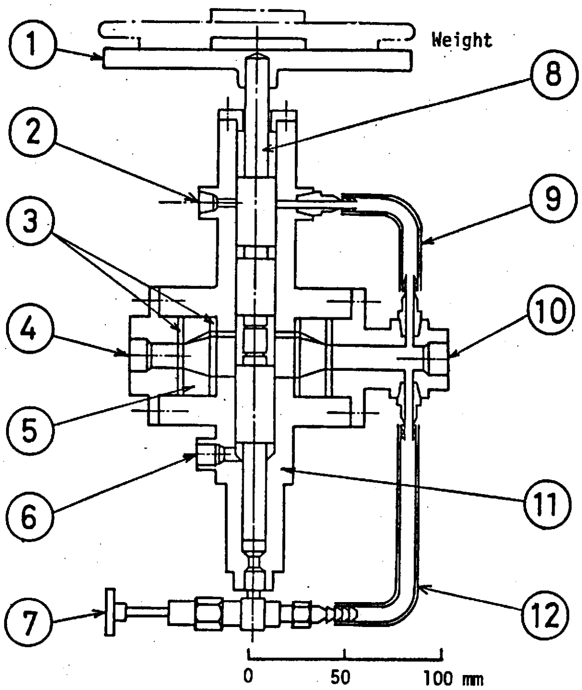
Chapter 2 explains both the structure and its principle of action of the plunger-type pressure control valve. In chapter 3, the dynamics of the valve is analyzed linearly, taking the change of the load into consideration and stability conditions are especially derived for a set of parameter values of the control valve. In Chapter 4, experimental studies are performed for the control valve made on trial and both the pressure-flow characteristics and the pressure-adjustment characteristics are examined from the viewpoint of practical use.

2. Structure and Principle of Action

In the proposed pressure control valve, the prescribed secondary pressure can be determined by selecting a moderate counter weight on the top of the plunger with a spool valve and its structure is shown in Fig.1.

The secondary pressure at the outlet port 10 is transmitted to the bottom of the plunger 8 through the pipe 12. The portion 4 is the inlet port, 2 the relief port and 1 the support of a counter weight. The pipe 12 is filled with oil and its fow rate can be controlled by a needle valve 7 .

In the static behavior of this equipment, the flow rate passing through the pressure control valve is equal to the one passing through the outlet port. Hence, plunger comes to a standstill at the fixed position, as the secondary pressure becomes equal to the pressure in the feedback pipe. But, if the flow rate in the outlet port increases, the secondary pressure becomes lower than the prescribed pressure. Then, the plunger goes down from the equilibrium state because of the drop of oil pressure acted under the bottom of the plunger, and then the spool valve of the plunger spreads quickly. From these reasons, the secondary pressure returns to the prescribed one as the volume of the air supply increases and the pressure in the outlet port raises. On the other hand, in the case where the secondary pressure becomes higher than the prescribed pressure, the pressure acted under the bottom of the plunger heightens, the plunger rises and the openning area of a spool valve becomes narrow. As a result, the secondary pressure returns to a prescribed one, as the volume of the air supply decreases and the pressure in the outlet port falls. Furthermore, in the case where the secondary pressure becomes too high, the plunger goes up still more and the air in the secondary side is escaped through the relief port equipped with the upper part of the plunger.



1. Weight support
2. Relief port
3. Packing
4. Inlet port
5. Teflon packing
6. Oil port
7. Needle valve
8. Plunger
9. Vinyl pipe
10. Outlet port
11. Casing
12. Feedback pipe

Fig.1 Structure of plunger-type control valve

3. Analysis of Dynamics

Figure 2 shows the schematic drawing of the plunger-type pressure control valve analyzed here. Principal symbols 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
A_3 : openning area of throttle valve in the feedback pipe m^2
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 [plunger + support + counter weight + rotary pully] kg
P_0 : pressure in feedback pipe MPa
P_1 : primary pressure (pressure of inlet port) MPa
P_2 : secondary pressure (pressure of outlet port) MPa
P_3 : output pressure of outlet throttle valve MPa
V_m : volume of air chamber at the secondary side m^3
f : friction coefficient N·s/m
T : absolute temperature of air K
$\dot{R}$: gas constant of air J/(kg·K)
α : flow coefficient
κ : adiabatic index number
γ : density of air kg/m^3
g : acceleration of gravity m/s^2

Since the pressure control valve is a constant-valued control system, the dynamics of each portions can be represented by linearlized equations, taking the small deviations around equilibrium states into consideration. Due to the small deviation ΔP_0 of the pressure acted under the bottom of the plunger, the plunger shifts

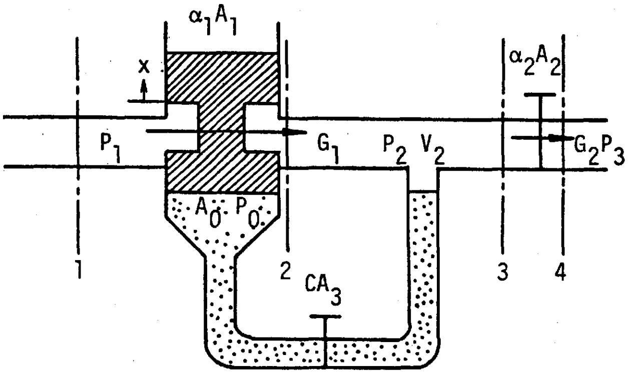


Fig.2 Sketch of plunger-type control valve

from x to x+Δx. Hence, associated with the dynamics of the movable portion, the following equation yields,

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

Assuming zero initial conditions, the Laplace transform may be written by

$$(Ms^2 + fs)\Delta x(s) = A_0 \Delta P_0(s) \tag{2}$$

Nextly, we consider the flow of air through the throttle valve. Supposed that the air is an adiabatic flow passing through both the spool valve of the plunger and the throttle valve in the outlet port, the mass flow rate of air G_1 and G_2 for each valves (section 1 to 2 and section 3 to 4 in Fig.2) are given by

$$G_1 = \frac{\alpha_1 A_1 P_1}{\sqrt{RT}} \sqrt{\frac{2\kappa}{\kappa-1}\{(\frac{P_2}{P_1})^{2/\kappa} - (\frac{P_2}{P_1})^{(1+\kappa)/\kappa}\}} \tag{3}$$

$$G_2 = \frac{\alpha_2 A_2 P_2}{\sqrt{RT}} \sqrt{\frac{2\kappa}{\kappa-1}\{(\frac{P_3}{P_2})^{2/\kappa} - (\frac{P_3}{P_2})^{(1+\kappa)/\kappa}\}} \tag{4}$$

Accordingly, from Eqs.(3) and (4), the small derivatives ΔG_1 and ΔG_2 can be obtained as

$$\Delta G_1 = (\frac{\partial G_1}{\partial P_1})\Delta P_1 + (\frac{\partial G_1}{\partial P_2})\Delta P_2 + (\frac{\partial G_1}{\partial \alpha_1 A_1})\Delta \alpha_1 A_1 \tag{5}$$

$$\Delta G_2 = (\frac{\partial G_2}{\partial P_2})\Delta P_2 + (\frac{\partial G_2}{\partial P_3})\Delta P_3 + (\frac{\partial G_2}{\partial \alpha_2 A_2})\Delta \alpha_2 A_2 \tag{6}$$

As the mass of air W_m in the air chamber of the secondary side is given by

$$W_m = (V_m/RT)P_2 \ , \tag{7}$$

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

$$\Delta G_1 - \Delta G_2 = \frac{dW_m}{dt} = \frac{V_m}{RT}\frac{d(\Delta P_2)}{dt} \ . \tag{8}$$

Substituting Eqs.(5) and (6) into Eq.(8), and taking the Laplace transform, we have

$$\Delta P_2(s) = \frac{1}{1+K_1 s}\{K_2 \Delta x(s) - K_3 \Delta P_3(s) + K_4 \Delta P_1(s) - K_5 \Delta \alpha_2 A_2(s)\} \tag{9}$$

where

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

$$K_2 = (\frac{\partial G_1}{\partial \alpha_2 A_1})(\frac{\partial \alpha_1 A_1}{\partial x}) / \{(\frac{\partial G_2}{\partial P_2}) - (\frac{\partial G_1}{\partial P_2})\}$$

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

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

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

Furthermore, let us consider the dynamics of oil flow in the feedback pipe. Since the feedback pipe is filled with oil and the fluid velocity of oil flowing in the pipe is slow, it can be considered that the oil flow is a laminar one. Hence, as the pressure drop due to the pipe friction is given by the rule of Hagen Poiseuille, the average fluid velocity is proportional to the pressure drop. Accordingly, the quantity of oil flow q is

$$q = CA_3(\Delta P_2 - \Delta P_0), \tag{10}$$

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

$$\frac{d(\Delta x)}{dt} = \frac{CA_3}{A_0}(\Delta P_2 - \Delta P_0). \tag{11}$$

The Laplace transform of Eq.(11) is given by

$$\Delta x(s) = \frac{CA_3}{A_0 s}\{\Delta P_2(s) - \Delta P_0(s)\}. \tag{12}$$

By referring to Eqs.(2),(9) and (12), the block diagram is reduced to the form of Fig.3 by the block diagram reduction technique.

From the block diagram as shown in Fig.3, the characteristic equation of this system can be obtained as

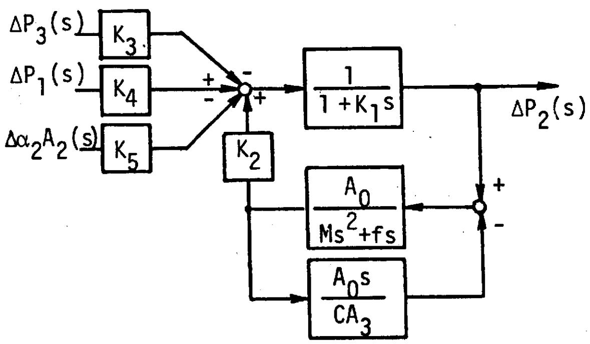


Fig.3 Block diagram

$$a_0s^3 + a_1s^2 + a_2s + a_3 = 0, \tag{13}$$

where $a_0=MK_1CA_3$, $a_1=fK_1CA_3+A_0^2K_1+MCA_3$, $a_2=fCA_3+A_0^2$ and $a_3=K_2A_0CA_3$.

The Hurwitz's stability criteria is applied to Eq.(13). Here, coefficients a_0, a_1, a_2 and a_3 are always positive. Hence, the condition of stability is

$$\begin{aligned} H &= a_1a_2 - a_0a_3 \\ &= \{(fK_1 + M)CA_3 + A_0^2K_1\}(fCA_3 + A_0^2) - MK_1K_2A_0(CA_3)^2 > 0 . \end{aligned} \tag{14}$$

For the step-like displacement $\Delta\alpha_2A_2$ in the outlet throttle valve, the moving rate ε_x[m] of the plunger displacement x is derived as follows, referring to the block diagram in Fig.3,

$$\varepsilon_x \triangleq \lim_{t\to\infty} \Delta x(t) = -\frac{K_5}{K_2}\Delta\alpha_2 A_2. \tag{15}$$

Table 1 shows a set of parameters for various masses M of the movable portion of the plunger-type pressure control valve made on trial. Utilizing a set of parameters in Table 1 for Eq.(14), the critical values $CA_{3cr.}$ of stability region are calculated.

Table 1 Parameters of control valve made on trial

Parameters		1	2	3	4
M	kg	2.11	2.61	3.11	3.61
Q	m^3/s (10^{-3})	0.68 ~7.3	0.67 ~6.7	0.67 ~6.3	0.67 ~6.0
P_1	MPa	0.49			
P_2	MPa	0.30 ~0.26	0.34 ~0.32	0.40 ~0.37	0.44 ~0.42
P_3	MPa	0.10			
A_0	m^2	1.0×10^{-4} ($\phi=11.5\times10^{-3}$m)			
b	m	0.6×10^{-2}			
A_1	m^2 ($\times10^{-4}$)	0.01 ~0.28	0.01 ~0.18	0.02 ~0.26	0.03 ~0.37
a_1		0.70 ~0.27	0.53 ~0.40	0.42 ~0.29	0.36 ~0.22
α_2A_2	m^2 ($\times10^{-7}$)	11.6 ~148	9.98 ~108	8.77 ~88.3	7.82 ~71.8
V_m	m^3	223×10^{-6}			
f	N·S/m	49.0			
CA_3	$m^5/(N\cdot S)$	$0.0337\sim5.10\times10^{-10}$			
K_1	s	0.820 ~0.077	0.712 ~0.068	0.548 ~0.066	0.324 ~0.052
K_2	N/m^3 ($\times10^9$)	1.49 ~0.055	0.924 ~0.079	0.496 ~0.044	0.196 ~0.008
K_5	N/m^4 ($\times10^{10}$)	21.4 ~1.72	21.6 ~2.18	18.9 ~2.16	12.5 ~1.93

(note)P_1&P_2:gauge pressure

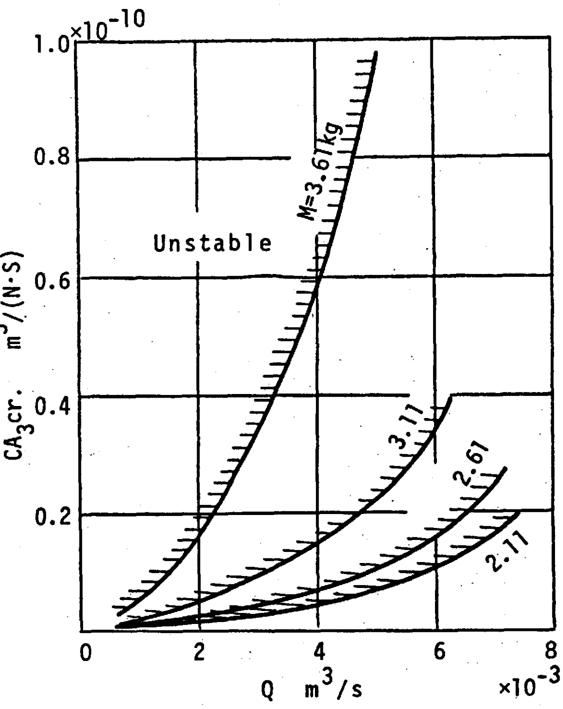


Fig.4 Stability region

Figure 4 shows the critical curve of the stability for various masses M of the movable portion, in the case where both the ordinate and the abscissa represent the coefficient $CA_{3cr.}$ of the throttle valve in the feedback circuit and the flow rate Q, respectively. This implies that, for parameters in the upper region of critical curve, the system diverges and becomes unstable. On the other hand, for parameters in the lower region of critical curve, the system converges and becomes stable. The stability region spreads out according to the increases of both the flow rate Q and the mass M of the movable portion. Consequently, it can be understood that the use of valve is advantageous to the cases of both the small magnitude of reducing pressure and much flow rate.

4. Experimental Studies

4.1 Description of Apparatus : In order to examine both the pressure flow characteristics and the pressure adjustment charcteristics, experimental apparatus is constructed as shown in Fig.5. The air stored in compressor 5 passes through the filter 7 and may be applied to the plunger-type pressure control valve 14 through both the diaphragm-type pressure adjusting valve 8 and the air

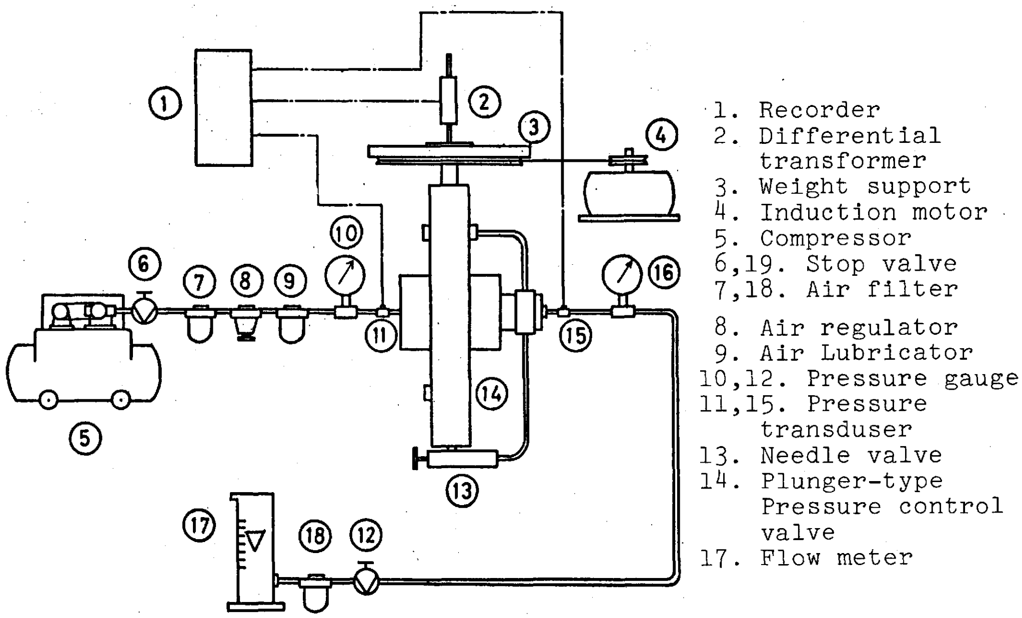


Fig.5 Experimental apparatus

lubricator 9 . The flow from the valve 14 passes through the valve 12 and the filter 18, and then is released to the atmosphere after measurement of the air flow rate by the area flowmeter 17 . The primary nad secondary pressures are detected by the semiconductor pressure detector 11 , 15 and their time responses are recorded on the recorder 1 . The rough values of their pressures can be estimated by Bourdon gauges 10 and 16. The plunger is rotated slowly by the small-sized electric motor 4 through the pully 3, and then the plunger goes smoothly up and down. The desirable pressure can be obtained structurally by setting up the counter weight on the pully. The action of plunger is transformed into electric signal through the differential transformer and recorded on the recorder 1. The flow rate of the outlet port is varied by the throttle valve 12 .

4.2 Flow Coefficient : The throttle of the needle valve in the feedback pipe plays an important role for this device to keep equilibrium balance. Then, the flow coefficient of this needle valve was first examined by the experiment. Though the oil flow rate is given by Eq. (10), it is difficult to measure the area of throttle port A_3. Hence, the flow coefficient may be treated as CA_3, including both C nad A_3. It may be considered for the difference of pressure $\Delta P_2-\Delta P_0$ to be practically less than 0.05 MPa. Then, the flow rates per unit time were measured for every 0.01 MPa from 0.01 MPa to 0.05 MPa in the gauge pressure. From these results, values of CA_3 were obtained and then the relationship between rotating number of the needle valve and CA_3 was shown in Fig.6. Figure 6 illustrates that the value CA_3 can be obtained by the rotating number of the needle valve.

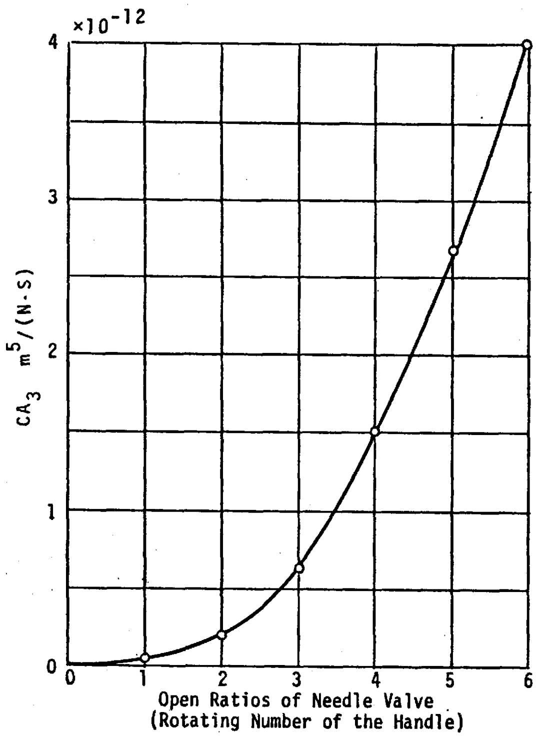


Fig.6 Relationship between rotating number of the handle and flow coefficient

4.3 Pressure-Flow Coefficient : Let the prescribed value of the secondary pressure be the pressure in the case where the flow rate is zero. The pressure-flow characteristics expresses the rate of fluctuation of the secondary pressure, when the primary pressure is constantly kept and the flow rate varies.(2)(3) In this equipment, the prescription of the secondary pressure may be determined by the weight added on the top of the plunger. The mass of the plunger used here was 2.11kg, and five kinds of weights such as 0.5kg, 1.0kg, 1.5kg, 2.0kg and 2.5kg were added on the top of the plunger. In addition, experiments were performed with regards to the case that the primary pressure was varied from 0.24 MPa to 0.49 MPa in the gauge pressure. An example of the results is shown in Fig.7, in the case where the primary pressure is 0.45 MPa. The solid line shows the characteristics of the plunger-type pressure control valve and the broken line shows, for the comparative study, the experimental result of the diaphragm-type pressure control valve under the same conditions.

As shown in Fig.7, the deviation of the secondary pressure is less than 10 percent of its prescribed value, under that the flow-rate is at about 5×10^{-3} m^3/s. Accordingly, this valve can hold the accurate secondary pressure. On the contrary, the diaphragm-type reducing valve tested here is less than 10 percent at the flow-rate 0.83×10^{-3} m^3/s and, in addition, less than 50 percent at about 5×10^{-3} m^3/s. This tendency is similar to the cases for another primary pressures. Thus, the valve proposed here has an excellent pressure-flow characteristics.

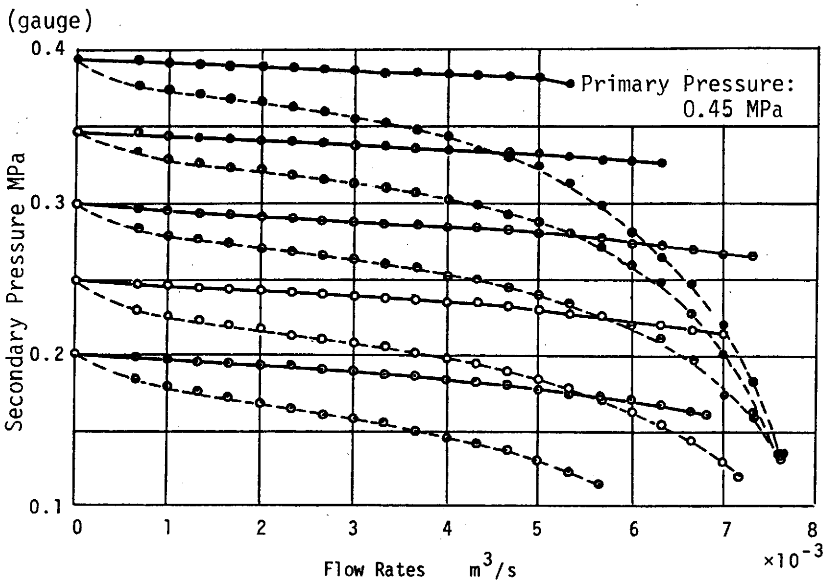


Fig.7 Pressure-flow characteristics

4.4 Pressure-Adjustment Characteristics : The Pressure-adjustment characteristics expresses the rate of the deviation of the secondary pressure for the range of the primary pressure, in the case where the flow rate is constantly kept. In this experiment, when the flow rate is varied from 0 to 5×10^{-3} m^3/s, the relationships between the primary and secondary pressures were measured, associated with the mass of movable portions, for every 0.5kg from 2.11kg to 5.11kg. An example of results is shown in Fig.8, in which the ordinate and the abscissa are the primary and secondary pressures, respectively. Figure 8 shows the case where the flow rate is at 0.83×10^{-3} m^3/s, taking the mass M of movable portion as a parameter. From this result, it is clearly seen that the secondary pressure keeps constant for various primary pressures. This fact holds for the range of 1.67×10^{-3} m^3/s. However, for the flow rate beyond this limit value, the secondary pressure becomes slightly lower along with the increase of the primary pressure, although the magnitude is less than 5 percent.

5. Conclusions

For the plunger-type pressure control valve made on trial, the performance, especially, the static characteristics has been examined theoretically and experimentally. In consequence, the guideline for the design of control valve can be summarized as follows.

(1) **The effect for the mass of movable portion :** From the analytical result, the stability region expands, if the mass of movable

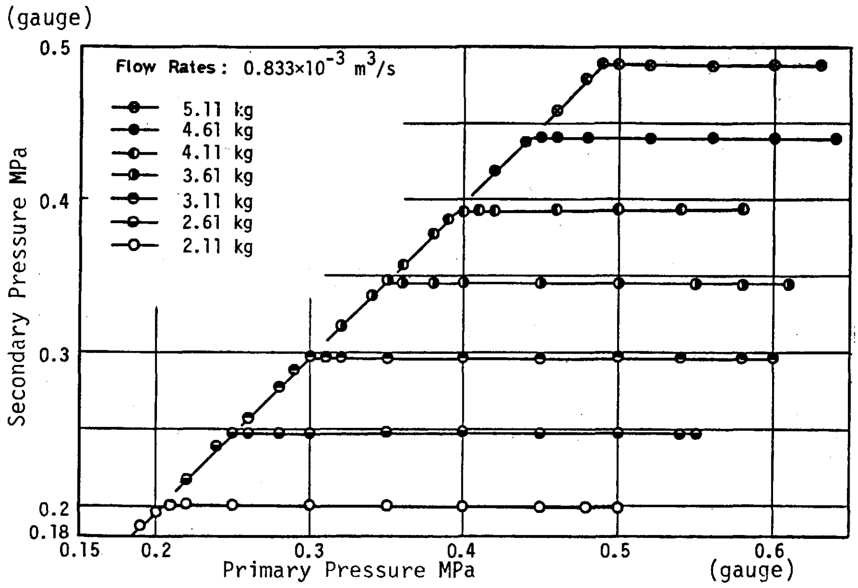


Fig.8 Pressure-adjustment characteristics

portion M is larger. However, if the mass M is too much large, the settling time becomes long and the responsibility grows worse. Inversely, in the case where the mass M is too much small, the plunger arises the hunting phenomenon and becomes unstable. Accordingly, it is desirable that the mass of movable portion M is somewhat large.

(2) **The effect for the throttle of the valve in feedback pipe :** In the case where the throttle of the valve in feedback pipe has narrowed, the fluid resistance of oil becomes large and then the device becomes stable as the damping force acts on properly. However, when the throttle has too much narrowed, the settling time becomes longer and then the responsibility grows worse. On the contrary, when the throttle has too much opened, the plunger gives rise to the vibration and becomes unstable. From the experimental result, we can obtain the favorable responsibility at about $CA_3 = 1.0 \times 10^{-10}\ m^5/(N \cdot s)$.

(3) **The effect for the pressure adjustment :** For a range of the flow rate performed in the experiment, it has been verified that the secondary pressure keeps a constant prescribed value, comparing with the diaphragm-type reducing valve used in general. Consequently, if this valve may be used for a relatively large flow rate, it can be expected that a stable action and an effective pressure adjustment are obtained, although the velocity performance reduces to lower efficiency.

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