Application of Pressure Control Type Quasi-Servo Valve to Force Control System

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***Abstract*—Today, the aged people are rapidly increasing and the number of children is decreasing in Japan. This social problem causes the demand of the care and welfare equipment to support a nursing and a self-reliance for the senior. For example, a power assist device for reducing the burden of the user has been researched and developed. The purpose of this study is to develop a small and light-weight pneumatic control valve and to apply it to the care and welfare equipment. In our previous study, the small-sized quasi-servo valve using two inexpensive on/off valves was developed and tested. The pressure control type quasi-servo valve was also proposed and tested by using the quasi-servo valve, a pressure sensor and an embedded controller. In this paper, the pressure control type quasi-servo valve is applied to a force control of the pneumatic cylinder, and its control performance is investigated.**

***Index Terms*—quasi-servo valve, pneumatic cylinder, force control**

I. INTRODUCTION

Today, the care and welfare pneumatic devices to support a nursing care and a self-reliance of the senior and the disabled are actively researched and developed by many researchers [1], [2]. These wearable devices require many control valves for multi degrees of freedom and precise control performance of the wearable actuator. However, by increasing the degree of freedom, the total weight load of the wearable devices increases too. Therefore, we aim to develop a small-sized, light-weight and low-cost quasi-servo valve using on/off valves to decrease the burden of the user instead of expensive and bulky conventional electro-pneumatic servo valves. In our previous study [3], an inexpensive pressure control type quasi-servo valve using a low-cost embedded controller and a pressure transducer was proposed and tested. In addition, the compensation for decrease of output flow rate was proposed to improve the pressure control performance of the valve. An analytical model of the pressure control type quasi-servo valve including the embedded controller was also proposed. The control performance of the valve using P and PD controller was investigated theoretically [4]. We also investigated the optimal control parameter of the PD controller by means of simulation. It is easier to realize the force control when the pressure control type valve is used. In this paper, as an application of the pressure control type quasi-servo valve, the force control system is built and tested by using a pneumatic cylinder. The force control system consists of the pressure control type valve, a pneumatic cylinder and an electric linear actuator.

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II. CONSTRUCTION AND OPERATING PRINCIPLE OF QUASI-SERVO VALVE

Fig. 1 shows the schematic diagram of the quasi-servo valve developed before [5]. The valve consists of two on/off type control valves (Koganei Co. Ltd., G010HE-1) whose both output ports are connected to each other. One valve is used as a switching valve to exhaust or supply, and the other is used as a PWM control valve that can adjust output flow rate like a variable fluid resistance. The valve connected with the actuator is a two-port valve without exhaust port. The other is a three-port valve that can change the direction of fluid flow from the supply port to the output port or the fluid flow from the output port to the exhaust port. The two-port valve is driven by pulse width modulation method in order to adjust the valve opening per time. The size of the on/off valve is $33\times19.6\times10$ mm, and the mass is only 15 g. The maximum output flow rate is 38 liter/min at 500 kPa.

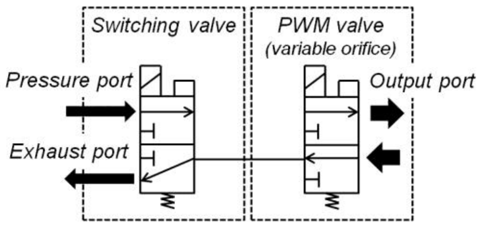

Figure 1. Schematic diagram of quasi-servo valve.

III. PRESSURE CONTROL TYPE QUASI-SERVO VALVE

A. Construction

Fig. 2 shows the schematic diagram of the pressure control type quasi-servo valve. The valve system consists of the above quasi-servo valve, a pressure sensor (Matsushita Electronics Co. Ltd., ADP5160) and an embedded controller (Renesas Co. Ltd. R8C12M). The pressure control is done as follows. The embedded controller gets the sensor output voltage and the reference voltage through an inner 10 bit A/D converter. The manipulated value for the PWM valve, duty ratio, is

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calculated based on a control scheme by using these AD values.

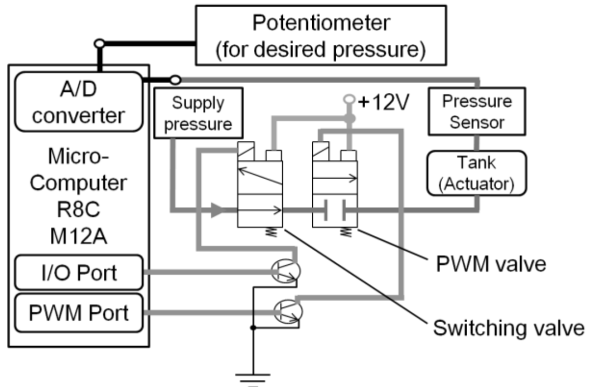

Figure 2. Schematic diagram of pressure control type quasi-servo valve

B. Compensation of Flow Rate

The supply and exhaust flow rate are not same even if the valve opening is same, because the pressure difference between upstream and downstream at the supply port is different from that at the exhaust port. In order to compensate the flow rate, the following compensation method was proposed [3]. This is a natural phenomenon of fluid flow. This method leads to a linearization of the valve characteristics to the controller.

$$u_s = |u| \frac{f(z)_{\max}}{f(z)} + 47.5 \quad , \quad z = \frac{P_L}{P_S} \tag{1}$$

$$u_e = |u| \frac{P_s}{P_L} \frac{f(z)_{\max}}{f(z)} + 47.5 \quad , \quad z = \frac{P_a}{P_L} \tag{2}$$

where u_s and u_e represent the input duty ratio for supply and exhaust with the compensation of flow rate, respectively. *f(z)* is the function which expresses the state of fluid flow; sonic flow and subsonic flow [5]. $f(z)_{max}$ represents the maximum value of *f(z)* when the flow is sonic flow which is $f(z)_{max}$=0.484. As the valve has a dead zone, the duty ratio of PWM valve is always added by 47.5%. The duty ratio *u* for PWM valve and the state of switching valve is given by the following equation.

$$u = K_p e_{c(i)} + K_d (e_{c(i)} - e_{c(i-1)}) / T_m \tag{3}$$

$$u > 0 \Rightarrow \text{ON (Supply)}, \; u \leq 0 \Rightarrow \text{OFF (Exhaust)} \tag{4}$$

where $e_c(i)$, K_p, K_d and T_m represent the error from the reference pressure, the proportional gain (0.59%/AD), derivative gain (= 4.73×10^{-3} %・s/AD) and the sampling time of control(=3.2 ms), respectively.

IV. Force Control of Pneumatic Cylinder

A force control is needed in industrial robots, power-assisted systems and rehabilitation devices [6]. The force control using a fluidic actuator is easier and more inexpensive than that using an electric actuator. This is because the force can be controlled directly by the fluid pressure, and its control system does not need a force sensor. In this section, the force control of the pneumatic cylinder using the pressure control type quasi-servo valve is described.

A. Control System

Fig. 3 shows the schematic diagram and a view of the proposed force control system. The system mainly consists of the double action type pneumatic cylinder (Koganei Co., Ltd., PBDA 16x100-1A), an electric linear actuator (SUS Co., XA-50H-300E), a load cell (KYOWA ELECTRONIC INSTRUMENTS Co., Ltd., LUR-A-SA1, maximum force: 100N) for measuring the controlled force and a potentiometer (MIDORI PRECISIONS Co., Ltd, LP-150F-C, stroke: 150 mm) for measuring the position of the cylinder. The tested cylinder has an inner diameter of 8 mm and a stroke of 100 mm. The end of the piston rod of the cylinder is connected with the slide table of the electric liner actuator through the load cell, and the displacement of the piston is given by the electric liner actuator. The output signals from the load cell, the pressure sensor and a potentiometer are recorded by the data logger (HIOKI E.E. Co., MEMORY HiLOGGER 8430).

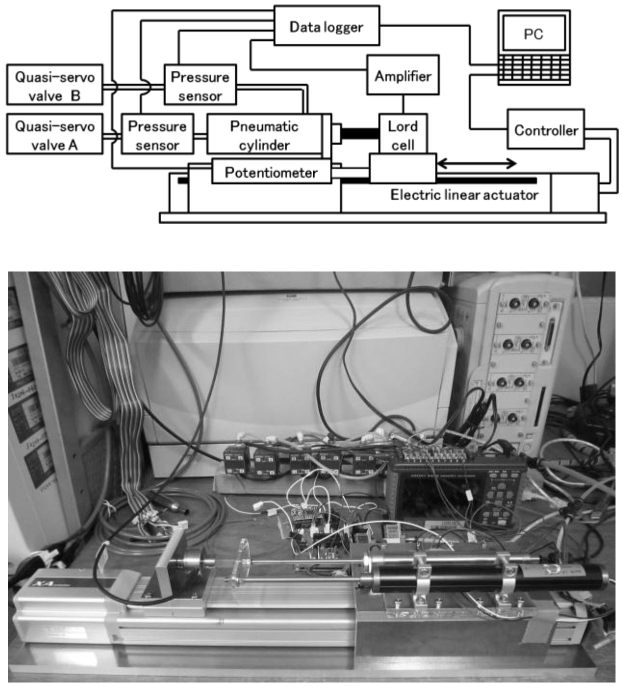

Figure 3. Schematic diagram of experimental equipment.

B. Control Procedure

Fig. 4 shows the block diagram of the control system. The pneumatic cylinder is controlled by using two tested pressure control type quasi-servo valves. Each chamber pressure of the valve is controlled independently by PD control scheme. One of the valves (Quasi-servo valve B) regulates the constant pressure of 50kPaG, and the other (Quasi-servo valve A) controls the pressure to generate the desired force.

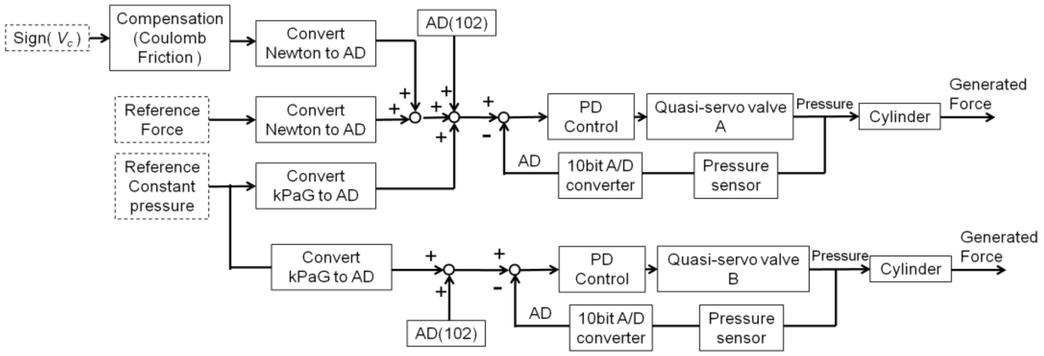

Figure 4. Block diagram of tested system.

C. Control Results and Discussion

Fig. 5 shows the control result of the cylinder force. The reference force is 5N. In the figure, the solid and dotted lines show the measured force and displacement of piston, respectively. The displacement of triangle wave with an offset of 40 mm and an amplitude of 20 mm was applied to the cylinder. The piston speed is plus or minus 16 mm/s. From the figure, it is observed that there is a big difference between reference force and measured force. The constant force opposite to the moving direction of the slide table can be found. This is caused by Coulomb friction between the piston and the cylinder. Therefore, the friction characteristic of the cylinder was investigated by the experiment.

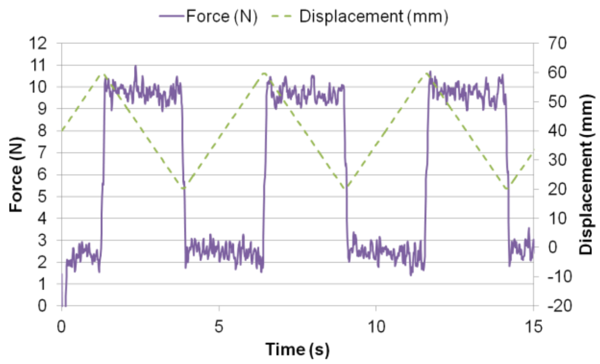

Figure 5. Control result (without friction compensation).

Fig. 6 shows the relation between velocity of the piston and frictional force. The experiment was carried out three times under the constant velocity of the slide table, and the force was measured at the certain position. From the experimental results, the following relation is obtained.

$$F_c = 13.8V_c + 3.21\mathrm{sgn}(V_c) \tag{5}$$

where F_c [N] and V_c [m/s] represent the frictional force and the velocity of the piston, respectively. From this equation, the coulomb friction of 3.21 N and the coefficient of viscous resistance of 13.8 N/(m·s) are obtained. In the following experiment, based on this result, the force control with friction compensation was tried.

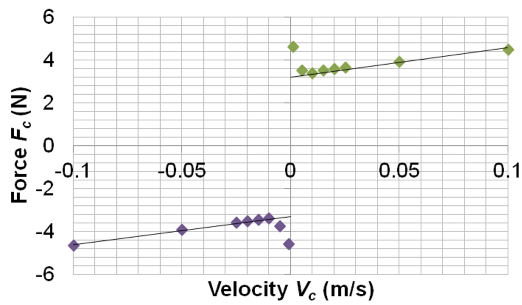

Figure 6. Friction characteristics.

Fig. 7 shows the force control result using tested valve with friction compensation. The compensation method is as follows. The sign of piston velocity is detected, and the pressure corresponding to the frictional force of 3.21 N is added or subtracted based on the sign. This control procedure is also shown in the block diagram in Fig. 4. From Fig. 7, it is found that there still exists an error of 1.8 N between reference and measured force. It is also observed that there is a sudden change of measured force when the piston displacement is the maximum and the moving direction is changed. At this position, the cylinder is extended largely and the chamber volume becomes maximum. It is considered that the sudden change is caused by the time delay of the pressure response due to the larger chamber volume. We think that these problems can be solved by improving the control scheme.

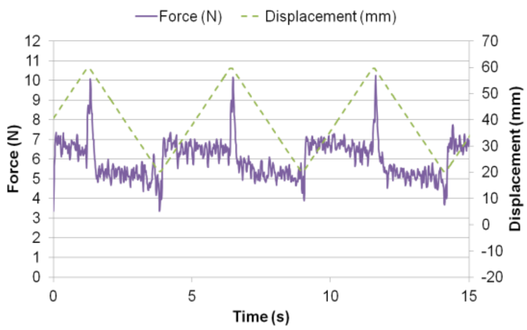

Figure 7. Control result (with friction compensation).

V. Conclusions

The purpose of this study is to develop a small and light-weight pneumatic control valve and to apply it to the care and welfare equipments. This study can be summarized as follows.

The small-sized quasi-servo valve which consists of two inexpensive on/off valve is explained. The pressure control type quasi-servo valve is built by the quasi-servo valve, a pressure sensor and an embedded controller. The force control using the pressure control type quasi-servo valve is easier and more inexpensive than others. Therefore, as an application of the tested pressure control type quasi-servo valve, the force control system of the cylinder is built and tested. The force control system consists of a pneumatic cylinder, an electric linear actuator, a load cell and a potentiometer. As a result, a large error between reference and measured force was observed. This is because of Coulomb friction in the cylinder. Then, the friction characteristics were investigated and the control performance was improved by compensating the friction.

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