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Development of direct drive type pneumatic servo valve

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ABSTRACT

: In this study, firstly, we have modeled a system consisting of various electro-mechanical and mechanical subsystems. The appropriateness of the model has been verified by simulation. The simulation model resolves the motion of spool, the winding current and the magnetic force. Also, we have calculated the displacement and velocity of the spool, flux contour line, b vector, flux density, flux linkage and back EMF etc. Secondly, a program for analysis the flow in the spool was developed, and a study was conducted on the flow rate of the nozzle depending upon the pressure ratio between the upstream pressure and the downstream pressure, when the valve is fully open in the spool and the flow force depending upon a displacement of the spool in the valve. Further, the performance of pneumatic servo valve has been verified through an overall performance test on the developed product.

KEY WORDS

Servo Valve, Fluid Power, Servo Solenoid, Spool Commutation Mechanism, PWM

NOMENCLATURE

B_r : Residual magnetic flux density[T]
B_g : Magnetic flux density at operating point of permanent magnet [T]
μ_r: Specific permeability of magnet
P_d : Thickness of magnet [mm]
g : Pore length [mm]
n : Number of windings of coil
P_H : Pressure at valve inlet [N/m^2]
P_L :Pressure at valve outlet [N/m^2]
$\dot{m}$: Mass flow rate of air passing through valve [kg/s]
b : Critical pressure ratio
R : Gas constant [N m/kg K]
T_H : Temperature at valve inlet [℃]
A_e : Effective sectional area of valve [m^2]
k : Specific heat ratio of air

1. INTRODUCTION

A pneumatic actuator provides a power of which the level stands between the power level of an electric actuator and a hydraulic actuator, and it is steadily used in the industry. The pneumatic servo system has a competitive power in the price aspect and its maintenance is simple. Also, its work environment is clean and its reliability is high. So, it has been applied to a manipulator and further recent new fields, such as a bio-machine, and the like. In these systems, a servo valve is a very important element to control the power [1]. In this study, a pneumatic servo valve was designed, and each simulation was conducted on any variation in the flow rate depending upon the magnetic force of the solenoid and the displacement of the spool. And ferromagnetic permanent magnet was used as a material for the plunger of the servo valve. Thereby, a low power consumption-type coil was designed, and a modeling for the coil design was conducted by using the magnetic circuit. Also, the feasibility of the modeling was verified by using a commercial magnetic field analysis program. The designed and fabrication of spool and sleeve, position sensor, servo controller and the dynamic characteristic verified by the experiment.

2. DESIGN OF SERVO SOLENOID

2.1 Configuration and design specifications

A configuration of the servo valve developed in this study is as shown in Fig. 1. An actuating system of the servo valve comprises a solenoid having coil wound on a plastic bobbin, a yoke for preventing any leakage flux and a mover using a permanent magnet. As the permanent magnet used for the mover, a neodymium-series magnet (NdFe35) which is recently widely applied to various fields was used [2]. And, the permanent magnet was fixed to the right side and the left side, and a pure iron washer was made to cling to both sides to which the permanent magnet was to be fixed. In order to enhance the magnetic force, the washer was designed so that its outer diameter might be thicker than its inner diameter and further so that it might be built in the aluminum spool [3][4].
And, as a precision valve for controlling flow rate, this valve is basically operated with a 5/3 way structure, and the operating signal is in the range of 0～10 [V]. Specification of the servo valve to be developed is as shown in Table 1.

Table 1 Specification of a servo valve

Item		Specification
Operating voltage		24 [V]
Control voltage		0～10 [V]
Current consumption	mid-position	0.05 [A]
	maximum	1.5 [A]
Turns per coil		260 [N]
Diameter of coil		0.18 [mm]
Resistance of coil		20 [Ω]
Permanent Magnet	Series	NdFe35
	Residual flux density	1.23 [T]
	Size of magnet	O.D.: ψ15, I.D.: ψ4, Width:6 [mm]
Stroke		±1.3 [mm]
Air gap		0.7 [mm]
Supply pressure		6 [bar]
Standart nominal flow rate		700 [l/min]
Connections		1/8 [inch]
Effective diameter		6 [mm]
Diameter of spool		11 [mm]

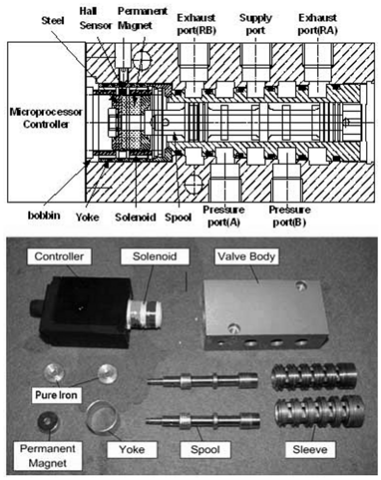

Fig. 1 The Developed of Servo valve

2.2 Design and modeling

Such modeling as shown in Fig. 2 was conducted for analysis of the solenoid of the servo valve. F_T [N/m], a total force acting between the solenoid and the permanent magnet is as shown in the following Eq. (1).

$$F_T = F_L + F_R \tag{1}$$

$$F_L = \ln B_g i,\ F_R = \ln B_g i \times (-1) \tag{2}$$

Where, F_L and F_R are force acting on the right-hand coil and the left-hand coil respectively, which are as shown in Eq. (2). And the coil resistance and the number of windings are the same between both coils, and only the coil winding direction is opposite to each other. Where, l is an axial effective length of the coil, and i is the current flowing on the coil. And the voltage equation at the current coil of the solenoid is as shown in Eq. (3).

$$e = Ri + L\frac{di}{dt} + k_g\frac{dx}{dt} \tag{3}$$

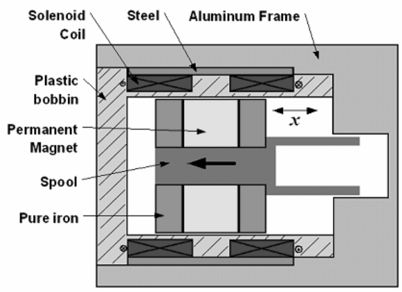

Fig. 2 Model of the servo valve (solenoid part)

In the above equation, the third term of the right-hand is a electromotive force caused by the motion of the permanent magnet, and the mechanical equation of motion is as shown in Eq. (4).

$$M\frac{d^2x}{dt^2} = k_g i - Cd\frac{dx}{dt} \tag{4}$$

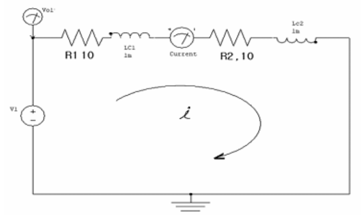

Fig. 3 Scheme of the solenoid circuit

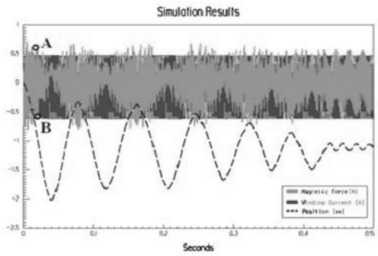

Fig. 4 Scheme of the solenoid circuit

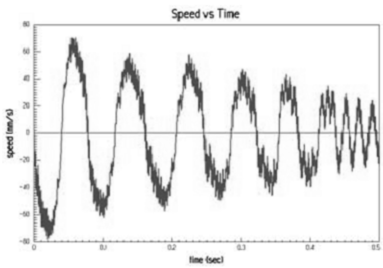

Fig. 5 Speed of spool

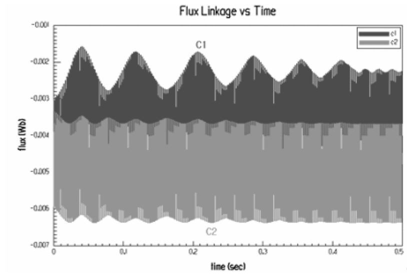

Fig. 6 Flux linkage of solenoid coil (C1, C2)

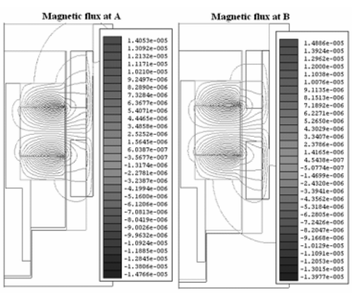

Fig. 7 Flux density diagram in A, B point of Fig. 4

The pore flux density caused by the permanent magnet, the mover at the solenoid of the servo valve is as shown in Eq. (5).

$$B_g = k \times \frac{B_r}{1+\frac{\mu r}{P_d / g}} \tag{5}$$

2.3 Simulation

A simulation was conducted on the solenoid of the designed servo valve, and the internal electric circuit of the solenoid is as shown in Fig. 3. R1 and R2, the coil resistance are 10Ω respectively. And the supply voltage was applied at +/-12[V] by square wave with 35 kHz. Offset voltage was 1 volt. Also time step for simulation was 0.001 second, total time of simulation was 0.5 second. Therefore, simulation results are as shown in from Fig. 4 to Fig. 8. Fig. 4 is magnetic force, winding current, and position of spool and Fig. 5 is speed of spool. Maximum magnetic force is 0.8[N], winding current at normal state is 500[mA], and maximum velocity of spool is 80[mm/s].
And Fig. 6 is a flux linkage of solenoid coil. Lastly, Fig. 7 is a flux density diagram (A, B) when the magnetic force is positive and negative.

3. CHARECTERISTIC ANALYSIS OF SPOOL COMMUTATION MECHANISM

3.1 Analysis of flow rate in spool

Fig. 8 shows model of a spool and a sleeve. For analysis the flow in the spool of the pneumatic servo valve, the valve was considered as a general compressible nozzle flow, and the energy equation of the compressible fluid is as follows:

$$u_1 + \frac{P_1}{\rho_1} + \frac{V_1^2}{2} + gz_1 = u_2 + \frac{P_2}{\rho_2} + \frac{V_2^2}{2} + gz_2 - q + w \tag{6}$$

If any potential energy of the nozzle is ignored and any inflow of heat and work is not made from outside, defined as an ideal gas, the following equation is satisfied;

$$P = \rho RT, \; k = \frac{C_p}{C_v}, \; R = C_p - C_v \tag{7}$$

If the above ideal gas equation and the above isentropic equality are applied to the energy equation, the result is as follows;

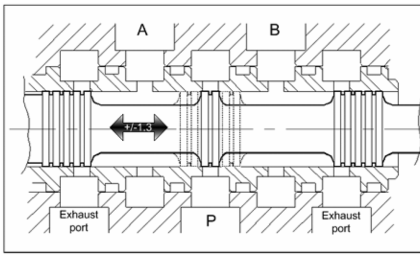

Figure8. Model of servo valve (spool and sleeve parts)

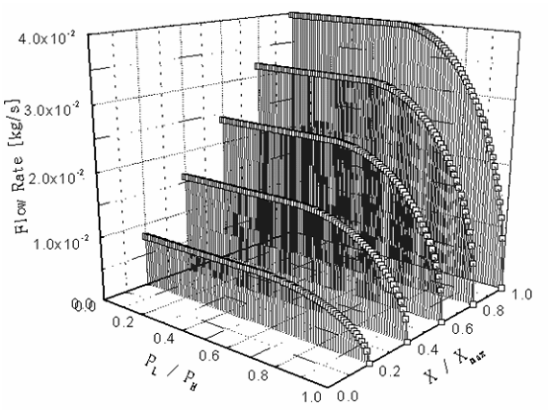

Fig. 9 Relations of flow rate to pressure ratio and full opened ratio.

$$V_2^2 = \left(\frac{2k}{k-1}\right)\frac{P_1}{\rho_1}\left[1-\left(\frac{P_2}{P_1}\right)^{k-1/k}\right] \tag{8}$$

Where, if z=p2/p1 is defined, the mass flow rate passing through the nozzle is as shown in Eq. (9).

$$\begin{aligned} \dot{m} &= \frac{p_1}{RT_1}\sqrt{\left(\frac{2k}{k-1}\right)(RT)\left(z^{2/k}-z^{k+1/k}\right)} \\ &= \frac{A_{12}P_1}{\sqrt{T_1}}\sqrt{\left(\frac{2k}{k-1}\frac{1}{R}\left(z^{2/k}-z^{k+1/k}\right)\right)} \end{aligned} \tag{9}$$

And, at a given nozzle area, a maximum mass flow rate depending upon a total pressure and a temperature exists, wherein, the value of the pressure ratio (z) is referred to as the critical pressure ratio. This is a relational equation of the mass flow rate of the compressible air passing through the valve nozzle, applying Anderson's experimental method, and the critical pressure ratio of the valve is shown as follows;

$$\dot{m} = \dot{m}_{cr} f(z) \tag{11}$$

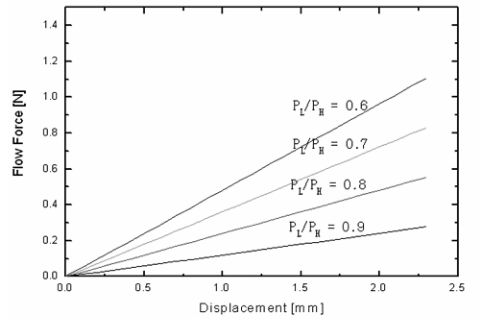

Fig. 10 Relation of flow force to displacement at various pressure ratios

$$f(z) = \left\{ 1 / \sqrt{\left(1-\left(\frac{z-b}{1-b}\right)^2\right)} \right\} \quad \begin{matrix} z \le b \\ z > b \end{matrix} \tag{12}$$

$$z = \frac{P_L}{P_H}$$

$$\dot{m}_{cr} = \left(\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}} \bullet \frac{k}{R}\right)^{1/2} \bullet A_e \frac{P_H}{\sqrt{T_H}} \tag{13}$$

Fig. 9 shows the flow rate of the nozzle depending upon the opening ratio of the valve in the spool and the pressure ratio between the upstream pressure and the downstream pressure [5].

3.2 Flow Force on Spool

Bernoulli's equation is applied under the assumption of the compressible and non-viscous flow, the jet velocity (U) at vena contractor (The point at which the flow area reaches its minimum is called Vena Contractor) of the nozzle is as follows;

$$U = \sqrt{(2\Delta P)/\rho} \tag{14}$$

$$F_{AB} = F_{fg} - F_{hi} = QU\rho\cos\theta \tag{15}$$

The net axial force, F_{AB} is as shown in Eq. (14). In the above equation, if C_V (discharge coefficient) of the nozzle flow is applied, the result is as shown in Eq. (15). If the flow force against the axial length is shown under the following condition, it is as shown in Fig. 10.

$$\begin{aligned} F_{AB} &= SC_V\omega z\Delta P\cos\theta \\ &= \rho\frac{Q^2}{\omega z C_V}\cos\theta \end{aligned} \tag{15}$$

Where, $$C_v = \frac{Q}{\omega z\sqrt{(2\Delta p)/\rho}}$$

ω = Peripheral width of nozzle

x = Axial length of nozzle

z = Diagonal length of nozzle $(\sqrt{x^2 + C_v^2})$

4. DESIGN OF MICROPROCESSOR CONTROLLER

As described in Fig. 11, this system is an apparatus to enable an amount of the fluid flowing in the pneumatic cylinder to be controlled linearly in proportion to the magnitude of the reference signal to be inputted from outside. Wherein, it is necessary to build up a closed loop system that can detect a difference between the reference signal and the actual response of the control system and further control by such difference value consecutively in order to control the flow rate so that it may be in conformity with the reference signal inputted from outside.

4.1 Detection of a position

Any adjustment of the flow rate of the pneumatic servo valve is made by controlling a position of the spool located in the valve. A hall sensor was used for detecting a position of the spool, and Fig. 12 is a schematic view of the position detection system

4.2 Control method

The opening amount of the pneumatic servo valve is determined by the electromagnetic force acting between the solenoid coil and the plunger. The electromagnetic force on the solenoid is in proportion

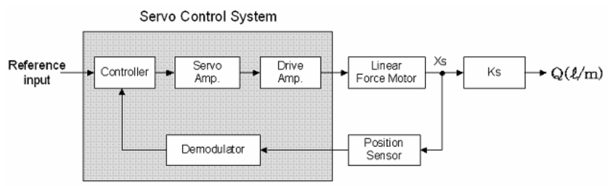

Fig. 11 Scheme of control block

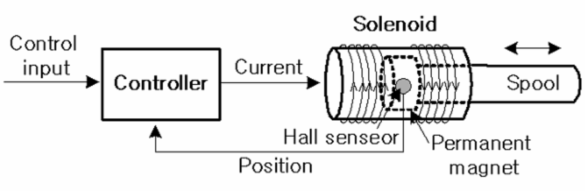

Fig. 12 Scheme of position detection system

to the current applied to the coil and the number of windings. Wherein, as the number of windings and the coil resistance are fixed, the current amount can be controlled by the input voltage. Therefore, in this study, the PWM (pulse width modulation) method was used as a method for controlling any voltage. The offset voltage variation method was used as a method for controlling the voltage against the control input, and the PWM control method of high frequency was used for controlling any displacement of the spool caused by pneumatics or other disturbance rapidly and precisely [6][7].

5. PERFRORMANCE TEST

5.1 Design of performance tester

In this study, in order to develop the technology for evaluating the overall performance and reliability of the pneumatic servo valve, the following tester was directly designed and manufactured, and Fig. 13 show the developed overall performance tester. First, the performance tester as shown in Fig. 13(left) was designed and manufactured so that the performance test might be conducted on the servo valve by using a low-friction high-speed cylinder. Then, an air filter, a flow rate sensor, a pressure sensor, a temperature sensor, a displacement sensor and a velocity sensor were fixed to it, and for a dynamic characteristic test on the servo valve, a length of the pipe connecting between the valve and the actuator was designed so that it might be the shortest.

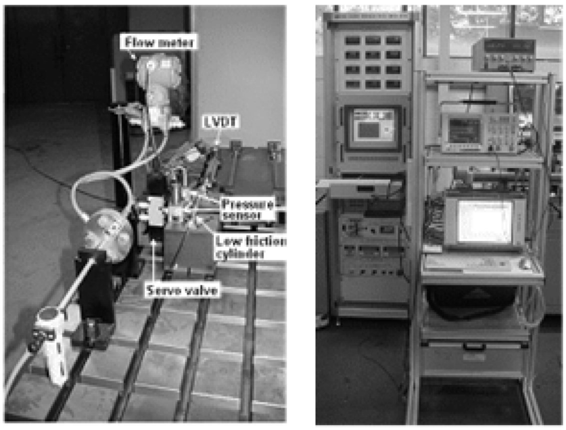

Fig. 13 Pneumatic servo valve performance tester

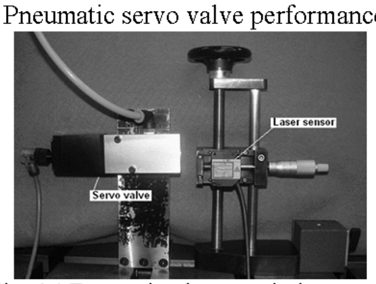

Fig. 14 Dynamic characteristic tester

And the controller for conducting the performance test comprises an indicator and a PC system for control enabling data from each sensor to be converted and real-time processed. The PC system for control is driven by the software dedicated to the dynamic characteristic test on the servo valve as developed in this study, having a function generating function, a noise filtering function and various signals processing function including real-time FFT. Also, as an interface device between sensors and the PC system for control, the NI-DAQ board having a signal conditioner built instrument for measuring each displacement of the therein was used, and thereby, the reliability of the experimental data was enhanced. Fig. 14 shows an spool of the servo valve directly by using a laser displacement sensor. In the case of the pneumatic servo valve, unlike a hydraulic servo valve, when a displacement of the cylinder is measured by using a general cylinder, the dynamic characteristic of the servo valve cannot be exactly measured due to compressibility of the working fluid. Therefore, in this study, a method for measuring the spool's displacement directly was used in order to avoid such problem.

5.2 Flow rate and pressure test

This test is the maximum outward flow rate test on the servo valve, and in order to test an outward flow rate depending upon the supply pressure, the supply pressure is first set. Then, the outward flow rate depending upon a change in the spool's position is measured by applying the control input to the servo valve at 0~10 [V].

As a result of conducting such tests, it could be identified that the outward flow rate of the servo valve met 700[l/min], the target value for development, and further that the pressure also met 0~10 bar, the target value for development.

As a result of conducting the flow rate test and the pressure test on the 1st pilot model, it was identified that the blind zone of the neutral point was great. When designing the 2nd pilot model, in order to reduce such blind zone, the range of the neutral point regime of the spool was reduced so that the blind zone could be reduced in the 2nd pilot model.

5.3 Dynamic characteristic test

A test to identify dynamic characteristics of the developed pneumatic servo valve was conducted in several ways, but there was a difficulty resulting from the time delay problem caused by the basic compressibility of air. In this study, in order to solve the time delay problem, a displacement of the servo valve spool was directly measured by using a laser sensor, and thereby the problem could be solved. As experimental conditions, the supply pressure was 6 bar, the flow rate was 700ℓ/min and a sine wave was used for a control input. Also, an experiment was conducted

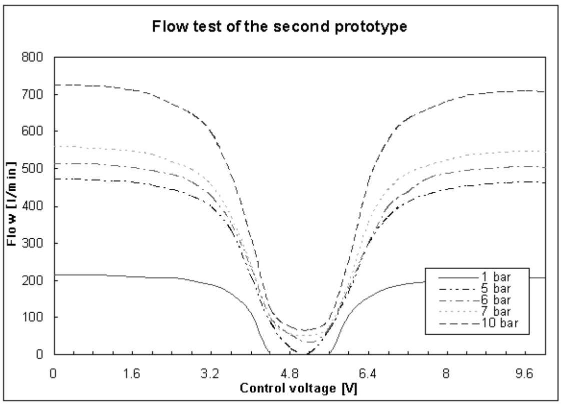

Fig. 15 Results of flow rate test on 2nd pilot model

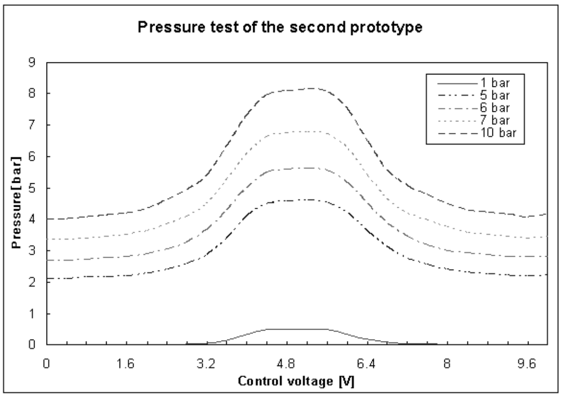

Fig. 16 Results of pressure test on 2nd pilot model

on a gain and a phase delay when the sine wave input signal was used as the 80% input of the experimental rated input signal of the valve and the frequency was varied in the range of 1Hz~100Hz. And, as a result of conducting the dynamic characteristic test by respective frequencies, the gain and the phase delay were shown in the frequency response graph as shown in Fig. 17.

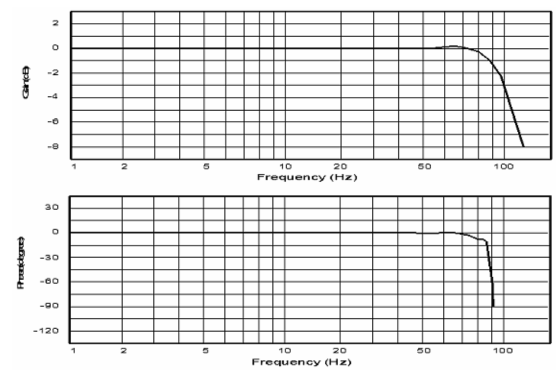

Fig. 17 Frequency response for gain and phase

6. CONCLUSION

In this study, the pneumatic servo valve was developed, and the study results could be summarized as follows;

1. A servo solenoid was designed, and its electromagnetic field was interpreted and the system's transient response was identified by using a commercial analysis program.

2. A program for analysis the flow in the spool was developed, and a study was conducted on the flow rate of the nozzle depending upon the pressure ratio between the upstream pressure and the downstream pressure, when the valve is fully opened in the spool and the flow force depending upon a displacement of the spool in the valve.

3. A PWM analogue controller was designed and manufactured in order to actuate the servo solenoid and further control a position of the spool by using a hall sensor.

4. In order to solve the time delay problem in dynamic characteristic test, a displacement of the servo valve spool was directly measured by using a laser sensor, and thereby the problem could be solved.

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