Development of the Pneumatic Servo Valve

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Abstract: Pneumatic servo valve is an electro-mechanical device which change electric signals to a proper pneumatic signals, that is, flowrate and pressure. In this study, a pneumatic servo valve was designed and each simulation was conducted on any variation in the flowrate depending upon the magnetic force of the linear force motor and the displacement of the spool. And permanent magnet was used as a material for the plunger of the servo valve. Thereby, a low electrical power consumption type coil was desinged. 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 the spool and sleeve, position sensor, servo controller and the dynamic characteristic verified by the experiment.

Keywords: servo valve, fluid power, servo solenoid, spool commutation mechanism, PWM(Pulse Width Modulation), magnetic force, flow rate, flow force, hall

1. INTRODUCTION

A pneumatic actuator provides a power of which the level stands between the power level of an electric actuator and an 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].

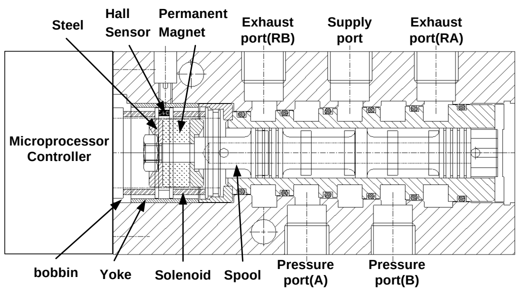

Fig. 1 Scheme of the servo valve

Table 1 Spcifiation 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 [ ]
Permanent Magnet	Series	NdFe35
	Residual flux density	1.23 [T]
	Size of magnet	O.D.: 15, I.D.: 4, Width:6 [mm]
Stroke		1.3 [mm]
Air gap		0.7 [mm]
Supply pressure		6 [bar]
Standard nominal flow rate		700 [l/min]
Connections		1/8 [inch]
Effective diameter		6 [mm]
Diameter of spool		11 [mm]

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 0～10V.

Also, in the range of 0～5V, the ahead stage port (A) gets to be opened, and in the range of 5～10V, the astern stage port (B) gets to be opened. Specification of the servo valve to be developed is as shown in Table 1.

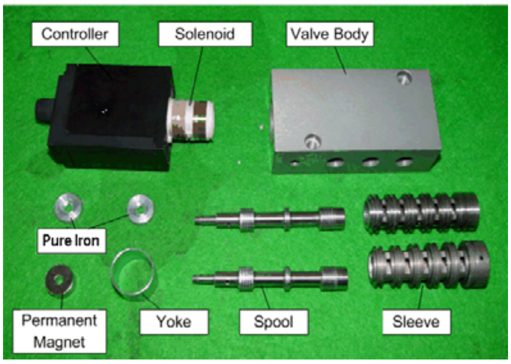

Fig. 2 Developed servo valve

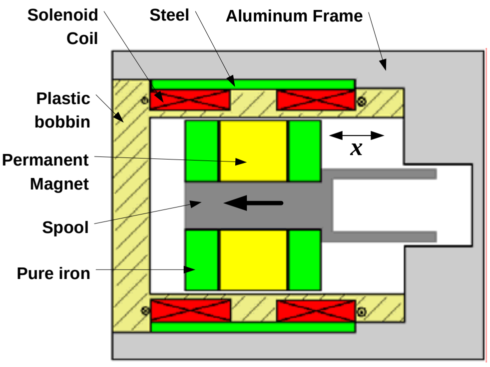

Fig. 3 Model of the servo valve (solenoid part)

2.2 Design and Modeling

Such modeling as shown in Fig. 3 was conducted for analysis of the solenoid of the servo valve. FT [N/m], a total force acting between the solenoid and the permanent magnet is as shown in the following Eq. (1).

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

Where, F_L and F_R are the forces 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.

$$\begin{aligned} F_L &= ln B_g i, \\ F_R &= ln B_g i \times (-1). \end{aligned} \tag{2}$$

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).

$$e = Ri + L\frac{di}{dt} + k_e \frac{dx}{dt} \; . \tag{3}$$

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

$$M\frac{d^2x}{dt^2} = k_e i - C_d \frac{dx}{dt} \; . \tag{4}$$

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

$$B_g = k \times \frac{B_r}{1 + \dfrac{\mu_r}{P_d / g}} \; . \tag{5}$$

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. 4. R1 and R2, the coil resistance are 10 respectively. And the supply voltage was applied at +/-12[V] by square wave with 35kHz. Offset voltage was 1 volt. Also time step for simulation was 0.001 milli second, total time of simulation was 0.5 second.

Therefore, simulation results are as shown in from Fig. 5 to Fig. 9. Fig. 5 is magnetic force, winding current, and position of spool and Fig. 6 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. 7 is a flux linkage of solenoid coil, Fig. 8 is a back EMF. Lastly, Fig. 9 is a flux density diagram(A,B) when the magnetic force is positive and negative.

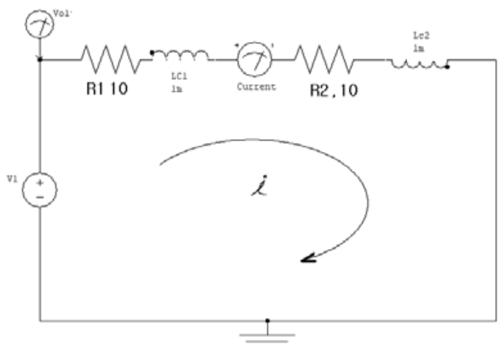

Fig. 4 Scheme of the solenoid circuit

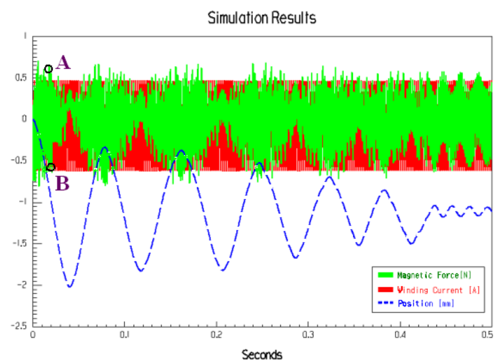

Fig. 5 Magnetic force, winding current, position

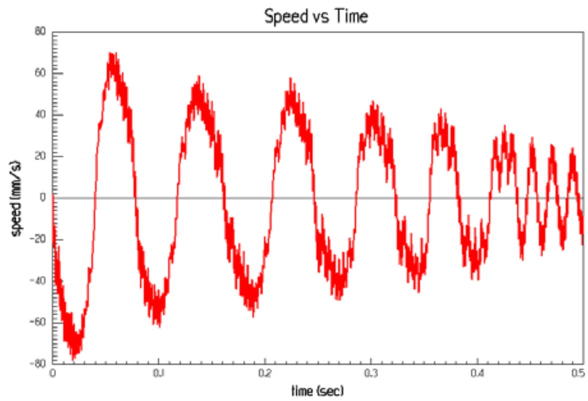

Fig. 6 Speed of spool

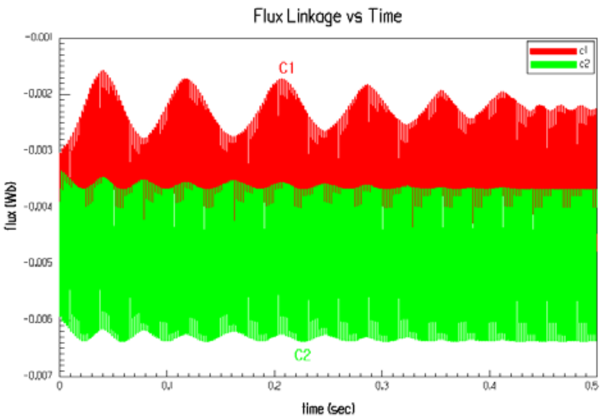

Fig. 7 Flux linkage of solenoid coil(C1, C2)

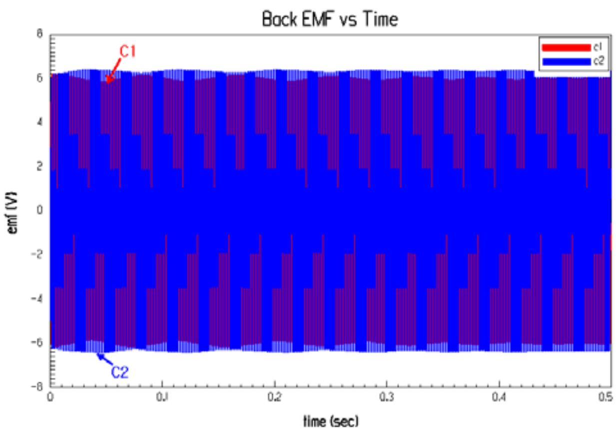

Fig. 8 Back EMF of solenoid coil(C1, C2)

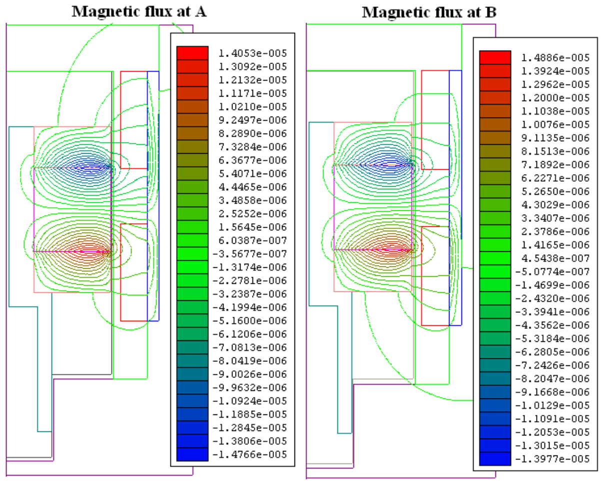

Fig. 9 Flux density diagram in A and B point of Fig. 5

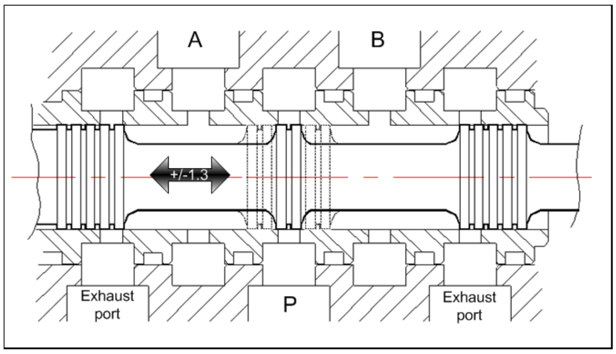

Fig. 10 Model of servo valve (spool and sleeve parts)

3. Characteristic Analysis of Spool Commutation Mechanism

3.1 Analysis of Flow Rate in Spool

Fig. 10. 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;

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

Where, if $z=P_2/P_1$ is defined, the mass flow rate passing through the nozzle is as shown in Eq. (9).

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

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.

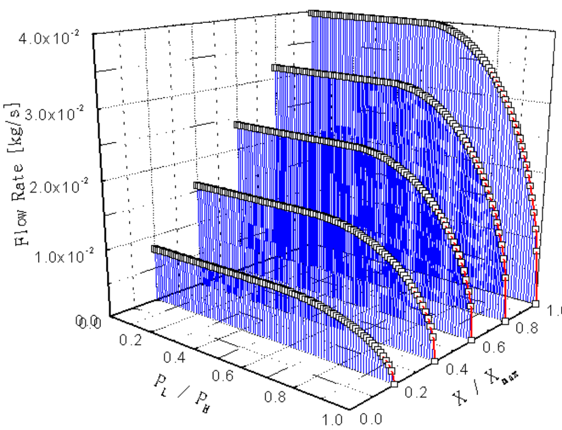

Fig. 11. Relation of flow rate to pressure ratio, full opened ratio

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;

$$\dot{m} = \dot{m}_{cr}\, f(z) \qquad (10)$$

$$f(z) = \begin{cases} 1 & z \le b \\ \sqrt{1 - \left(\dfrac{z-b}{1-b}\right)^2} & z > b \end{cases} \qquad (11)$$

where, $z = \dfrac{P_L}{P_H}$.

$$\dot{m}_{cr} = \left(\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}} \cdot \frac{k}{R}\right)^{1/2} A_e \frac{P_H}{\sqrt{T_H}} \qquad (12)$$

Fig. 11 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 contracta (The point at which the flow area reaches its minimum is called Vena Contracta) of the nozzle is as follows;

$$U = \sqrt{(2\Delta P)/\rho} \qquad (13)$$

The net axial force, F_{AB} is as shown in Eq. (14).

$$F_{AB} = F_{fg} - F_{hi} = QU\rho\cos\theta \qquad (14)$$

In the above equation, if Cv (discharge coefficient) of the nozzle flow is applied, the result is as shown in Eq. (15).

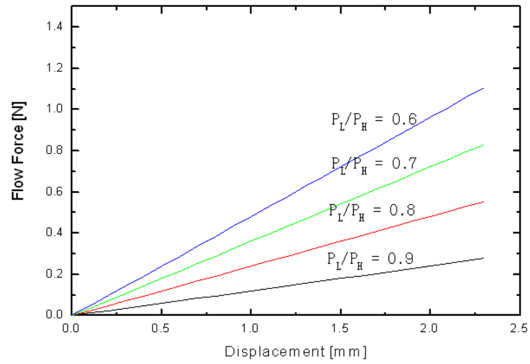

Fig. 12. Relation of flow force to displacement at various pressure ratio

$$F_{AB} = 2Cv\omega z\Delta P\cos\theta = \rho\frac{Q^2}{\omega z Cv}\cos\theta \qquad (15)$$

Where, $Cv = \dfrac{Q}{\omega z\sqrt{(2\Delta P)/\rho}}$

ω = peripheral width of nozzle

x = axial length of nozzle

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

If the flow force against the axial length is shown under the following condition, it is as shown in Fig. 12

4. Design of Microprocessor Controller

Fig. 13 shows the configuration of the control system of the pneumatic servo valve. As described in Fig. 13, 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.

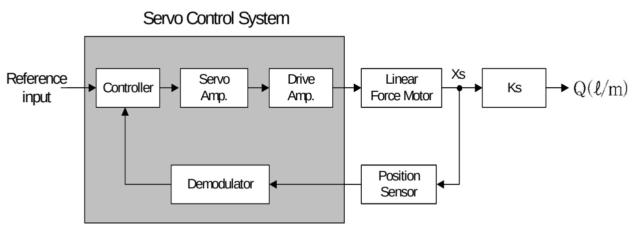

Fig. 13. Scheme of control block

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. 14 is a schematic view of the position detection system. The hall sensor has the current (I_H) flowing in one direction of a metal or a semiconductor, and if applies a flux density (B) in the direction perpendicular to the current, the electromotive force (V_H) gets to be induced from the direction orthogonal to them. This is referred to as Hall effect, which is as shown Eq. (16).

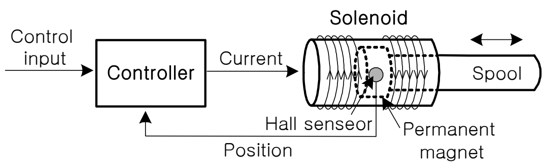

Fig. 14. Scheme of position detection system

$$V_H = \frac{R_H}{d} I_H B\cos\theta + K_e I_H \qquad (16)$$

R_H : : Hall coefficient,
d : Thickness of element
θ : Gradient of flux incident to the face of hall element
K_e : Unbalance coefficient

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 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].

4.3 Test Results of Controller

The experiments conducted in several ways with differently composed modules. The module is consists of controller, valve, and sensor system. The digital type controller was made for experiments. And we made valves of two kinds. With these, experiments conducted and results are shown in Fig. 15.

Table 2. Module Composing

<table>
<tr><th></th><th>Controller</th><th>Valve</th><th>Sensor System (Solenoid+Hall Sensor)</th></tr>
<tr><td>Data 1</td><td>Analogue</td><td rowspan="3">conventional Model</td><td rowspan="3">Conventional Model</td></tr>
<tr><td>Data 2</td><td rowspan="4">Digital Type Controller</td></tr>
<tr><td>Data 3</td></tr>
<tr><td>Data 4</td><td>Model 1</td><td rowspan="2">Model 1</td></tr>
<tr><td>Data 5</td><td>Model 2</td></tr>
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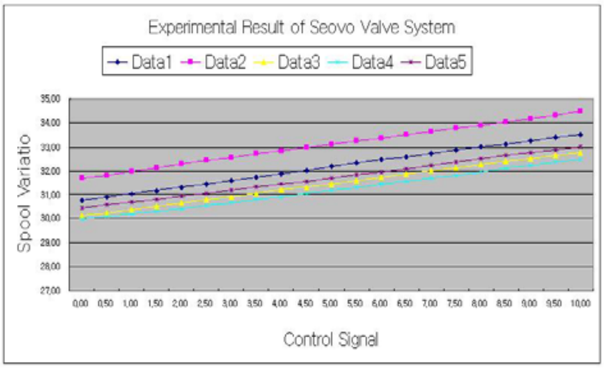

Fig. 15. Controller test result

5. Configuration of Test System

Fig. 15 and Fig 16 show the dynamic characteristic tester of pneumatic servo valve and the controller thereof as used in this study, respectively. The dynamic characteristic tester of Fig. 15 has an air filter, a flow -rate sensor, a pressure sensor and a laser displacement with high frequency. And, the controller for a performance and dynamic characteristic test comprises an indicator and a PC control system enabling data of various sensors to be converted and processed on the real time basis. The PC control system is run by the software dedicated to the test to identify dynamic characteristics of the servo valve, which was developed in this study, and it has various signal processing functions, such as a function to generate functions, noise filtering, real-time FFT, and the like. Also, the reliability of the experimental data was enhanced by using the NI-DAQ board having a built-in signal conditioner as an interface device between sensors and the PC control system.

5.1 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 700 /min and a sine wave was used for a control input. Also, an experiment was conducted 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.

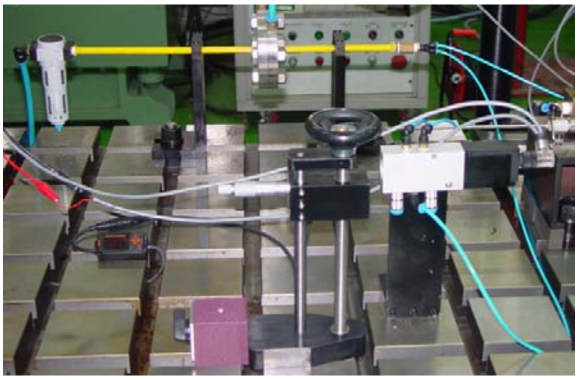

Fig. 15 Dynamic characteristic tester

Fig. 16. Controller for tester

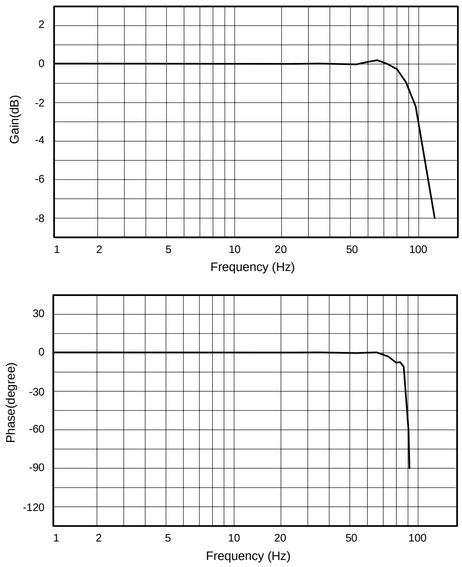

Fig. 17. Frequency response for gain and phase

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 analog 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. A new controller, valve, and sensor system was manufactured and experimented. As a result, servo valve system with manufactured modules (digital type controller, manufactured valve(model 2), and manufactured sensor system) shows good result comparing with conventional servo valve system.

5. 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.

Acknowledgements

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Reference

[1] S. H. Choi, *“A Study on analysis and desin of a poppet type electropneumatic servovalve*", KAIST, 2000.

[2] Yoon Seok-Gil, "*Rare Earth Material for Permanent Magnet and Application thereof*", UUP, 1999.

[3] W.J, Duffin, "*Electricity and magnetism*", McGraw -Hill, p162-193, 1990.

[4] John D. Karaus, "*Electromagnetics*", McGraw-Hill, Third edition, 1984.

[5] Anderson, "*The Analysis and Design of Pneumatic System*", 1967.

[6] N. Ye, S. Scavarda, M. Betemps and A. Jutard, "Models of a pneumatic PWM solenoid valve for engineering applications", *Transactions of the ASME*, Vol. 114, p680-688, 1992.

[7] K. Ichiryu, H. Tamura, and S. Kogarashi, "Development of linear motor servo valve", *Fluid Power, Third JHPS,* p391-396, 1996.