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Design of High Speed Solenoid Actuator for Hydraulic Servo Valve Operation

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Abstract – Modern electric controlled valves are demanded that its solenoid actuator should be smaller size, lighter weight, lower consumption power, and higher response time. For achieving these purposes, the major design factors of solenoid actuator such as magnetic flux density, coil turn numbers, plunger size, bobbin dimension, and etc. are must be optimized. In this study, for optimal design of high speed solenoid actuator for hydraulic servo valve operation, we draw up governing equations which are composed by combination of electromagnetic theories and empirical knowledge, and deduct the values of major design factors by use of them. For more increase the operating speed, voice coil are used as main armature in manufacturing of prototype actuator. And, we have proven the propriety of the governing equations and speed increasing method by experiments using the hydraulic valve assembly adopted the prototype of solenoid actuator.

Keywords: Solenoid Actuator, Permanent Magnet, Governing Equation, Design Factor, Voice Coil, Frequency Response Test

1. Introduction

Solenoid actuator is a very economical motion converter due to its simple structure. The important items for design of solenoid actuator are calculation and analysis of magnetic flux density, size of plunger, optimal design of bobbin, analysis of adopted magnetic materials, determination of space factor, and calculation of turn numbers considering the temperature rising for design of solenoid actuator.

For optimal design, theoretical and empirical knowledge are simultaneously needed. Theoretical knowledge governs the operational characteristics of the solenoid actuator, and empirical knowledge compensates for the theoretical limitation obtained from the designer's design and manufacturing experiences for various kinds of solenoid actuators [1].

In particular, empirical knowledge is more essential than theoretical knowledge for determination of the plunger shape and value of the space factor because the varieties of plunger shape are very versatile and the space factor has a subjective property. As such, they cannot be determined solely by calculation or simulation. When designer's accumulated experiences and expertise are added to theses, the most proper shape and value of them can then be obtained. Especially, the research for higher response and precise solenoid actuator has been actively studied. The representative world-wide makers like Yuken, Moog, and Rexroth are still studied about life test for linear actuator to ensure reliability of the durability of it[11-13]

Permanent magnet is also an important component effecting on operation of solenoid actuator. It needs the determination of optimal operating point and selection of magnetic material. But, in this study, we deal with the permanent magnet as a only flux density body having a constant value for simplification of design. So, the overall needed attraction force is controlled by current adjustment.

The solenoid actuator performs the energy converting role in the valve operation mechanism. It converts the input electrical energy of coil to mechanical reciprocation movement of spool. Theoretically, the improvement of spool operational speed can be realized by change of some design parameters as like reduction of the spool weight, increase of coil turn numbers, and increase of the input current. But, realistically, it is very difficult to get the needed design values with only them, because the design parameters are mutually connected each other in the mathematical design equations. As it were, we can not expect the high response of spool by only electromagnetic force of the coil based on theoretical knowledge.

In this study, we derived the governing equations composed by a combination of electromagnetic equations and empirical coefficients that represent the operating principle of high speed solenoid actuator for hydraulic servo valve operation. We additionally introduced magnetic force of permanent magnets adding to coil current for overcoming the theoretical limitation and getting higher operational response in low input current. As well, we

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manufactured a prototype of a solenoid actuator based on final results of the design, and proved the propriety of the governing equations and empirical coefficients by experiments.

2. High Speed Solenoid Actuator

2.1 Structure of High Speed Solenoid Actuator

In this study, we applied the voice coil type actuator as high speed solenoid actuator. Voice coil type actuator is called moving coil type actuator, coiled coil bobbin is operated forward backward movement unlike the typical direct drive type actuator. Fig.1 is represented the simple structure of voice coil type solenoid actuator.

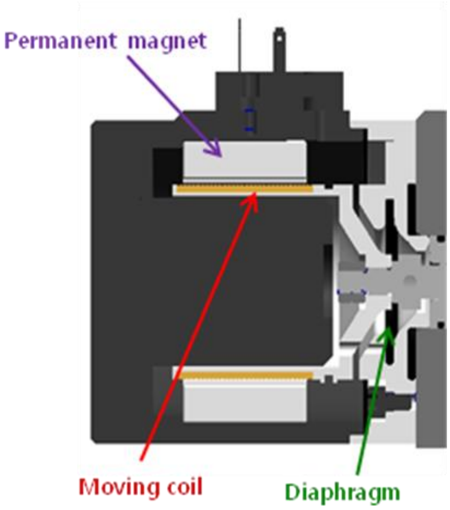


Fig. 1. Structure of voice coil type solenoid actuator

3. Governing Equation

3.1 Magnetic Flux Density and Magnetic Motive Force in Air Gap

The attraction force F is shown in equation (1).

$$F = \frac{B^2 \cdot S}{2\mu_0} [N] \tag{1}$$

Where, B is magnetic flux density, S is cross sectional area of plunger, as it were, it is $\pi\left(\frac{d_l}{2}\right)^2$ when d_l is radius of plunger, and μ_0 is permeability in the air

Therefore, from equation (1), the magnetic flux density B needed in air gap is expressed as equation (2).

$$B = \sqrt{\frac{F \cdot 2\mu_0}{S}} \tag{2}$$

And, theoretical magnetic motive force U_m are shown in equation (3), Where, d is maximum distance between plunger and stationary.

$$U_m = \frac{B \cdot d}{\mu_0} \tag{3}$$

Equation (4) is obtained from equations (1) and (3), and also, the design coefficient K_f can be expressed as equation (5).

$$F = \frac{K_f}{d^2} \tag{4}$$

$$K_f = \frac{\mu_0 \cdot S \cdot U_m^2}{2} \tag{5}$$

When the length of fixed air gap is S_f, the maximum distance d between plunger and stationary is given to equation (6) that is represented by the sum of fixed air gap S_f and plunger stroke S_e. So, the maximum attraction force $F_{\max}$ and, the minimum attraction force $F_{\min}$ become equations (7) and (8), respectively.

$$d = S_f + S_e \tag{6}$$

$$F_{\max} = \frac{K_f}{S_f^2} \tag{7}$$

$$F_{\min} = \frac{K_f}{d^2} \tag{8}$$

And therefore, form equations (7) and (8), the relation between maximum attraction force and minimum attraction force become equation (9).

$$F_{max} = \left(\frac{d}{S_f}\right)^2 \cdot F_{min} \tag{9}$$

Magnetic flux density of equation (3) is equal to equation (10) by substitution of equations (1) and (8) to equation (9).

$$B = 2 \cdot \frac{\sqrt{2 \cdot \mu_0 \cdot F_{min}}}{d_l \cdot \sqrt{\pi}} \tag{10}$$

And, from equations (3), the actual magnetic motive force U, needed by the solenoid actuator, is obtained by equation (11).

$$U = \frac{C_m \cdot B \cdot d}{\mu_0} \tag{11}$$

Where, C_m is and empirical compensation coefficient for magnetic motive force [1], [6]. It is needed for compensating the loss portion of magnetic force in the actual magnetic circuit.

3.2 Permanent Magnet and Flux Density in Air Gap

The magnetic flux density generated by permanent magnet in the air gap is as equation (12).

$$B_g = \frac{B_r \cdot h_m}{\frac{A_g}{A_m} h_M + \mu_M S_f} \tag{12}$$

Where, A_g is cross sectional area of air gap, A_m is pole area of permanent magnet, B_r is residual magnetic flux density, h_M is length of permanent magnet in magnet sticking direction.

And, the permeability of permanent magnet μ_M is like equation (13).

$$\mu_M = \frac{B_r}{H_c \mu_0} \tag{13}$$

If the cross sectional area of air gap and permanent magnet is same, equation (12) becomes to equation (14).

$$B_g = \frac{B_r}{1 + \frac{S_f}{h_M} \mu_M} \tag{14}$$

From equation (14), we can know that the magnetic flux density of air gap approaches to the residual magnetic flux density when the length of permanent magnet is long and the length of air gap is completely short.

For decision of operating point of permanent magnet, we must consider the maximum energy area of permanent magnet and the reduced magnetic flux due to reaction of magnetic field by solenoid coil. But, in case of this paper, the change of characteristic of permanent magnet may not be occurred because the operating point of permanent magnet resulted from completely short length of air gap and path of magnetic flux.

3.3 Estimation of Yoke Thickness

Inner diameter d_{yi} and outer diameter d_{yo} of yoke are as equation (15) and (16).

$$d_{yi} = d_{bo} + C_g \tag{15}$$

$$d_{y0} = \sqrt{d_{yi}^2 + C_p \cdot d_l^2} \tag{16}$$

$$\text{Yoke thickness} = (d_{yo} - d_{yi}) / 2 \tag{17}$$

The empirical constant C_g in equation (15) is the length margin for smooth heat dissipation of the coil, and the empirical constant C_p in equation (16) is the length margin for smooth passing of magnetic flux [6], [8-9].

3.4 Temperature Rising and Bobbin Length

Heat dissipation coefficient λ is the amount of heat energy radiated form the coil surface. It can be founded in Fig.2.

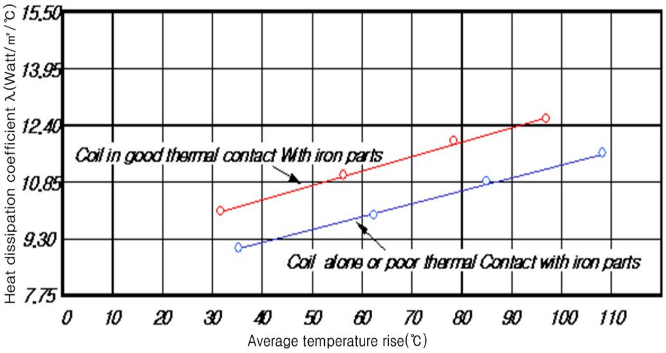


Fig. 2. Heat dissipation coefficient according to temperature rising

Coil resistance R and current I passing through it produce the rising temperature represented as T_f in equation (18). By substituting equations (19) and (20) into equation (18), we can make the constructive equation of final temperature rising as equation (21).

$$T_f = \frac{W}{2 \cdot \lambda \cdot S} = \frac{I^2 \cdot R}{2 \cdot \lambda \cdot S} \tag{18}$$

$$R = \rho \frac{(l_m \cdot N^2)}{h \cdot w \cdot X_i} \tag{19}$$

$$X_i = \frac{\pi}{4} \left(\frac{d_s}{d_0} \right)^2 \tag{20}$$

$$T_f = \frac{q \cdot p}{d \cdot \lambda \cdot X_{i \cdot w}} \cdot \left(\frac{N \cdot W}{h \cdot V} \right)^2 \tag{21}$$

Where, S : area of heat dissipation ($= l_m \cdot h$), W :consumption power, ρ : relative resistance, l_m : coil mean length per single turn, N : total turn numbers, h : coil height, w: coil layer thickness, q : duty ratio, d_s : diameter of bare wire, d_0: diameter of insulated coil, and V : supply voltage

Bobbin length (coil height) used in equation (19) is calculated by equation (22).

$$h = \sqrt[3]{\frac{(q \cdot \beta \cdot \rho \cdot U^2)}{2 \cdot \lambda \cdot X_i \cdot T_f}} \tag{22}$$

Where, β is the ratio of bobbin height h and coil layer thickness w. That is, β is equal to $\frac{h}{w}$.

3.5 Turn Numbers and Consumption Power of Coil

Coil mean length l_m turn is represented as equation (23).

$$l_m = \frac{\pi(d_{bo} + d_{bi})}{2} \tag{23}$$

And, the relation between equivalent resistance R_t of solenoid circuit using copper wire, supply voltage V, current I, and relative resistance ρ and be expressed by equation (23).

$$R_t = \frac{V}{I} = 4\rho \left[\frac{(l_m \cdot N)}{\pi \cdot d_s^2} \right] \tag{24}$$

Diameter of bare wire, d_s is induced to equation (25) from equation (24).

$$d_s = \sqrt{\left(\frac{2 \cdot \rho \cdot (d_{b0} + d_{bi}) \cdot U}{V} \right)} \tag{25}$$

If it is assumed that insulated wire diameter is d_0 and the winding loss of a winding layer is 1 turn, the total turn number to be winded n_c in shaft direction given in equation (26). And, the total layer number m_c of coil in the radial direction is given by equation (27).

$$n_c = \left(\frac{h}{d_0} \right) - 1 \tag{26}$$

$$m_c = \frac{w}{d_0} \tag{27}$$

Therefore, the total turn number N to be winded on the bobbin can be given by equation (28).

$$N = n_c \cdot m_c \tag{28}$$

By combining equations (24) and (25), the equivalent resistance R_t, which represents the total resistance of coil, is fully obtained by equation (29).

$$R_t = \frac{2 \cdot \rho \cdot (d_{b0} + d_{bi}) \cdot N}{\pi d_s^2} \tag{29}$$

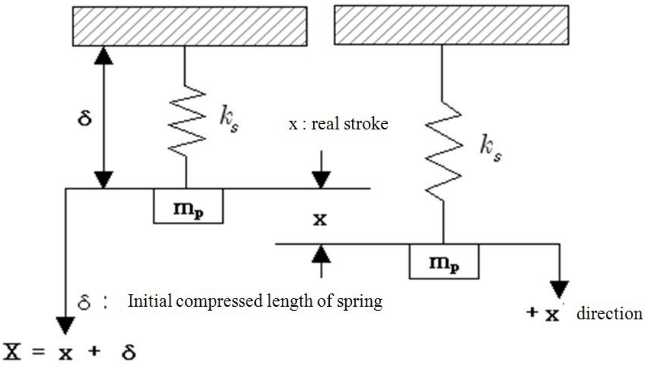
According to determination of R_t, the equations of coil current I and consumption power W are determined by equations (30) and (31), respectively.

$$I = \frac{V}{R_t} \tag{30}$$

$$W = V \cdot I \tag{31}$$

3.6 Operating Frequency

The solenoid actuator can be expressed as return spring – mass system like Fig. 3. After applying power, plunger displacement is equivalent to the displacement of mass m_p[8].



Before connection of mass m_p to the spring

After connection of mass m_p to the spring

Fig. 3. Mechanical model of solenoid actuator

The state equation of mass m_p to the x-direction is equation (32). By substitution of $X = x + \delta$ to equation (32), we can achieve equation (33).

$$m_p \ddot{X} + k_s X = m_p g \tag{32}$$

$$m_p \ddot{x} + k_s x = 0 \tag{33}$$

Therefore, mathematical model about the system of Fig. 4 become to equation (34).

$$\ddot{x} + \omega^2 x = 0 \tag{34}$$

Here, the operating speed ω and operating frequency f_p of the actuator can be expressed by equation (35) and (36), respectively[8].

$$\omega = \sqrt{\frac{K_s}{m_p}} \tag{35}$$

$$f_p = \frac{\omega}{2\pi} \tag{36}$$

4. Results of Design and Manufacturing of Prototype

We obtained the results of design in table 1 through the governing equations and input parameters. The prototype high speed solenoid actuator is represented Fig. 4. And also, Fig. 5 is represented the hydraulic servo valve which is adopted the high speed solenoid actuator prototype.

Table 1. Results of design

Parameters	Valuse
Attraction force	60 (N)
Magnetic flux density	2.3 ~ 2.4 (T)
Permanent magnet width	8 (mm)
Coil turn number	330 (No.)

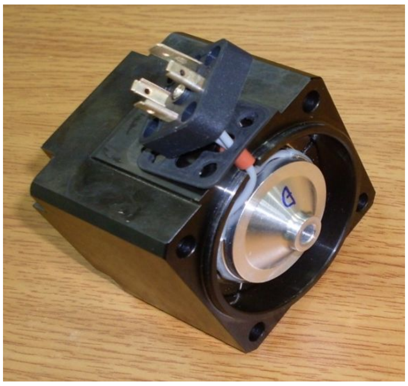


Fig. 4. Prototype of high speed solenoid actuator

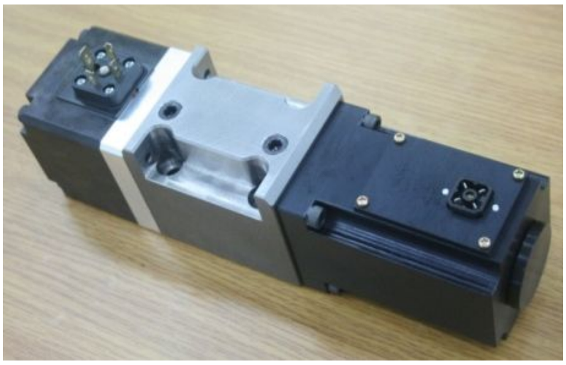


Fig. 5. Hydraulic servo valve assembly

5. Experiment Results

5.1 Test of Step Response

This test is to measure the time difference between supplying time of input step signal and reaction time of plunger.

At here, the 100 % control signal (10 V) to controller is used as input step signal, and reaction of plunger is detected by output signal of LVDT.

Fig.6 and 7 show the step response characteristics of before and after design improvement, respectively. By comparing of these two results, we can know that the step response characteristic is improved by 24ms. Referring to Fig.6, the control signal 10 V is applied to controller at point of time 56 ms, and the reaction signal 2.5 V of LVDT is detected at point of time 62 ms. So, we can know that the step response time is 6 ms.

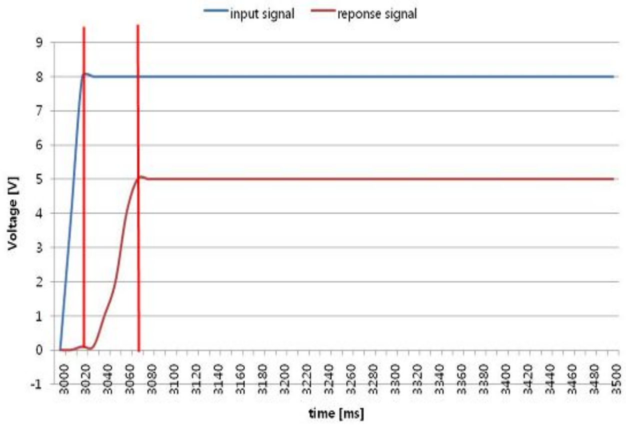


Fig. 6. Test result of step response before improvement

This test is similar to the test of step response. The input is control signal of controller and output is reaction signal of LVDT. This test performed at 25 % magnitude of input signal with 0.01 Hz ~ 500 Hz carrier frequency region.

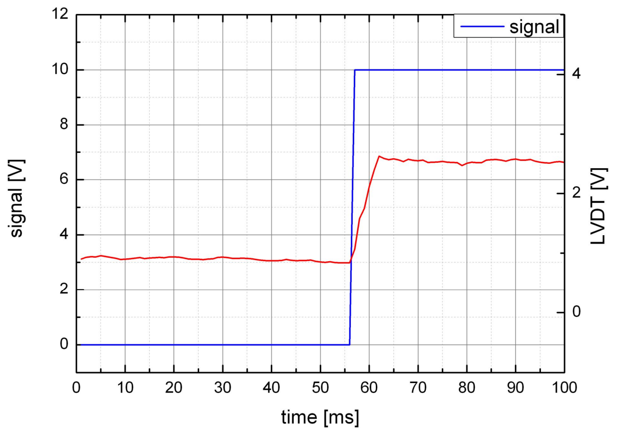


Fig. 7. Test result of step response after improvement

5.2 Test of Frequency Response

Fig.8 and 9 show the frequency response characteristics of before and after design improvement, respectively. By comparing of these two results, we can know that the frequency response characteristic is improved by 87 Hz. Fig.9 is the test results of after improvement for 25 % control signal of controller. It shows that the -3dB frequency is about 189 Hz in gain and 291 Hz in phase.

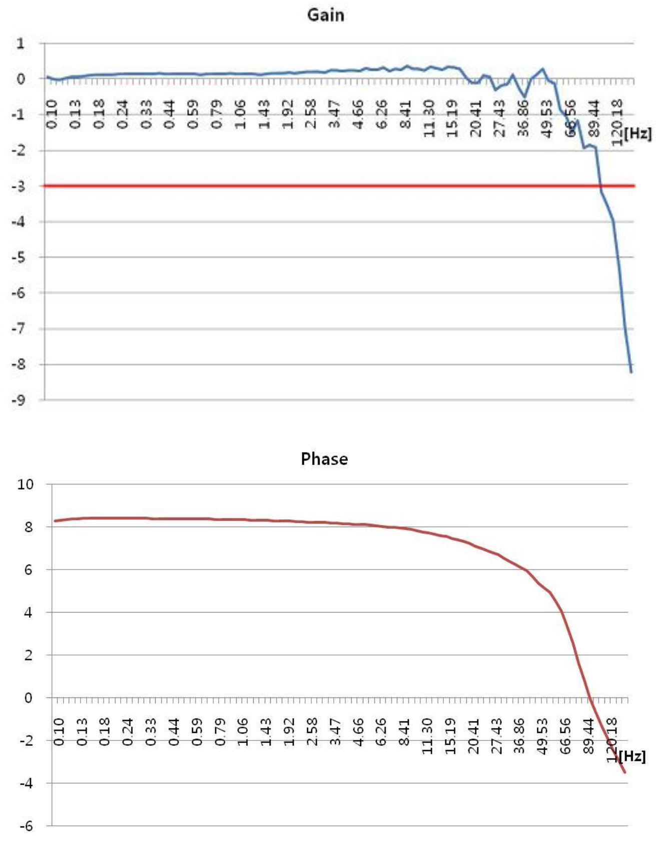


Fig. 8. Test result of frequency response before improvement

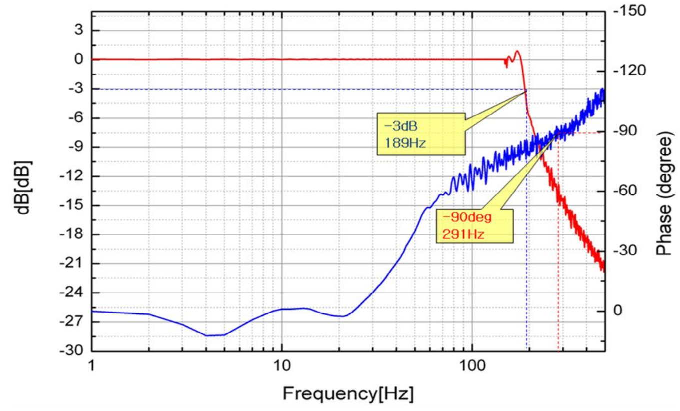


Fig. 9. Test result of frequency response after improvement

6. Conclusion

In this paper, we introduced all design courses of high speed solenoid actuator for hydraulic servo valve operation. The final results are as follows:

1) The governing equations are induced for design using between electromagnetic theories and empirical knowledge.
2) The important values of the design factors are decided as the results of optimal design through the governing equations.
3) For experiments, we manufactured a prototype of the high speed solenoid actuator using the above design results.
4) As results of experiments, we can know that the step response of prototype solenoid actuator approaches 6 ms in Fig.6, the operating frequency is about 189 Hz in -3dB gain and 291 Hz in -90 degree at 25 % input signal in Fig.7.

These test results mean that the performance of the prototype solenoid actuator was satisfactory for the specifications of general high speed actuator for valve operation, and the induced governing equations are propriety for optimal design of high speed solenoid actuator for hydraulic servo valve operation.

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