Static and Dynamic Performance Simulation of Direct-Acting Force Motor Valve

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Abstract: This work focuses on static and dynamic characteristics of direct-acting force motor valve. First, we analyzed the structure features and operating principle of the Mitsubishi-Hitachi force motor valve (FMV) and the operating principle of its internal permanent-magnet moving-coil force motor magnetic circuit, determined the transfer function of the FMV force motor system, and established a mathematical model for the system. Secondly, we established a static performance analysis model using the AMESIM software and utilized the model in combination with experimental results to analyze the effects of electro-hydraulic servo valve structural parameters on static characteristics. Lastly, we deduced the trajectory equation of the system, established the relationship between dynamic characteristic indexes and structural parameters, and analyzed the effects of different parameter values on the dynamic characteristics of the system. This research can provide a theoretical guidance for designing and manufacturing the FMV body.

Keywords: Force motor valve; Mathematical model; Simulation model; Static and dynamic performance; Parameters design

1. Introduction

Compared with traditional mechanical transmission, hydraulic transmission uses hydraulic fluid as the working medium for energy transfer and control. With its unique advantages as strong anti-interference ability and high control accuracy, it has become one of the key technologies in modern transmission and control systems [1]. In the hydraulic system, a hydraulic valve, by adjusting the flow, direction and pressure of liquid, controls the actuator’s motion direction, motion velocity, thrust, etc. [2]. If it malfunctions, the whole system can not work properly, resulting in incalculable losses. Hitachi force motor valves (FMV) are widely used in mill AGC systems [3] to control the mill work roll gap due to its characteristic of fast response, high control precision, and strong anti-pollution capacity, all which could affect the accuracy of machining precision of steel plates directly. Thus, real-time and effectively monitoring its various performance indicators is one of the important aspects to control the cost of steel production. Researchers [4-7] have performed some related researches on direct drive servo valves. Due to the limitations of theory and actual conditions, studies on FMV and relevant simulation tests were rarely reported. In addition, Mitsubishi-Hitachi hasn’t made its related technical parameters open to the public [8-12]. In this study, we first illustrated the structure and principle of FMV servo valve, then established a simulation model and applied simulation software to obtain its optimal structure parameters, analyze its main structure parameters affecting FMV static and dynamic properties, set up a relevant database and last experimentally verified the simulation model.
This research is of guidance for the design, manufacture, and measurement of similar direct-acting servo valves.

2. **FMV structural features and working principle**

Figure 1 shows the structure and principle diagram of the FMV. The spool connecting to the force motor coil together with the hydraulic part lie in the same working space, as a wet motor. The displacement sensor consists of the anti-magnetic sensor shield, differential pressure transmitter (coil), differential pressure transmitter (mandrel) and other components. The valve body consists of the spool, sleeve, etc. The force motor consists of the coil, permanent magnet, magnet guide, centralizing spring, etc. The adjustment mechanism for centralization consists of the lock bolt, lock ring, adjusting bolt, lock nut, intermediate adjust seat, key, seat ring, position ring, etc..

![Figure 1. FMV structure principle diagram.](image)

1, spool; 2, valve sleeve; 3, valve body; 4, O-ring; 5, connector; 6, wire; 7, bolt; 8, positioning ring; 9, key; 10, seat ring; 11, lock ring; 12, lock bolt; 13, adjust bolt; 14, lock nut; 15, median adjust seat; 16, lid; 17, spring; 18, permanent magnet; 19, guide magnet; 20, coil; 21, lid; 22, anti-magnetic shield of displacement sensor; 23 differential pressure transmitter (coil); 24, differential pressure transmitter (mandrel); 25, connector; 26, conductor.

The working principle of the FMV is as follows: When the current signal is not input, the centralizing spring has no deformation, the spool is at the zero position, the oil-supply chamber (Ps) is not connected to the oil-control chamber (Pc1), the oil control chamber (Pc2) is not connected to the oil-return chamber (R), while mutually-connected Pc1 and Pc2 turn into the oil-control chamber (Pc). When the forward current flows, the coil acted by force drives spool to move to the left side. When the force reaches equilibrium with damping force from the centralizing spring, the spool will stop at some positions on the left side. At this time, Ps and Pc1 are closed, Pc2 connects with oil-return chamber (R), Pc1 connects with Pc2, oil-supply chamber (Ps) and oil-control chamber (Pc) are closed, oil-return chamber (R) connect with oil-control chamber (Pc). When the opposite current flows, the system state is just opposite.

3. **Setup of FMV mathematical model**

3.1. **Force motor magnetic circuit of permanent magnet moving coil**

As shown in figure 2, the permanent magnet moving-coil force motor mainly consists of the working coil, guide magnet, permanent magnet, centralizing spring, etc. [13]. In the magnetic circuit, the ampere force is
\[ F = B_a \pi D N I_c \sin \theta \]  

(1)

where \( F \) is the ampere force, namely, the electromagnetic force applied on the moving coil, [N]; \( B_a \) is the magnetic induction intensity around the charged coil [T]; \( I_c \) is the current through the charged coil [A]; \( N_c \) is the number of turns of the charged coil; \( D \) is the average diameter of the charged coil [m]; and \( \theta \) is the acute angle between the charged coil and the magnetic induction [°].

Figure 3 shows the magnetic path of the permanent magnet moving-coil force motor. Let \( e \) stands for the output voltage of the direct current (DC) servo amplifier at the front of the permanent magnet moving-coil force motor, \( r_a \) stands for the internal resistance of the DC servo amplifier, and \( r_c \) stands for the resistance of the control coil, the equation of electric circuit of the working coil is:

\[ e = i(r_a + r_c) + L_c \frac{di}{dt} + K_{d} \frac{dX_v}{dt} \]  

(2)

where \( L_c \) is the self-induction coefficient of the the coil; \( K_d \) is the counter electromotive force constant, \( K_d = B_a \pi D N_c \). Taking Laplace transform of equation (2) finds:

\[ E(s) = I(s)(r_a + r_c) + L_c sI(s) + K_{d}sX_v(s) \]  

(3)

Further simplifying the equation obtains:

\[ I(s) = \frac{E(s) - K_{d}sX_v(s)}{(r_a + r_c) + L_c s} \]

\[ = \frac{[E(s) - K_{d}sX_v(s)]/(r_a + r_c)}{1 + s/\omega_a} \]

(4)

Where \( \omega_a = L_{cs}/(r_a + r_c) \).

**Figure 2.** Schematic of charged coil structure.  
**Figure 3.** Schematic of magnetic circuit.

### 3.2. Setup of mathematical model for FMV force motor system

Inside the FMV spool, when the electromagnetic drive force, hydrodynamic force, elastic force, and frictional resistance reach an equilibrium, the relationship between the force generated by the force motor and the corresponding displacement is

\[ F = M_v \frac{d^2X_v}{dt^2} + (B_n + B_s) \frac{dX_v}{dt} + k_v X_v + k_f X_v \]  

(5)

where \( M_v \) is the total mass of the FMV spool, charged coil, and other parts [kg]; \( X_v \) is the displacement of the FMV spool [m]; \( B_n \) is the viscous damping coefficient of the system; \( B_s \) is the damping coefficient generated by the transient hydrodynamic force; \( K_v \) is the stiffness of the FMV centralizing spring [N/m]; and \( K_f \) is the stiffness of the stable hydrodynamic force from which the FMV spool suffers [N/m].

Combining equation (1) with (5) finds:
\[ B_{z} D M I = M \left( \frac{d^{2} X}{dt^{2}} + (B_{z} + B_{v}) \frac{d X}{dt} + K_{v} X_{*}, \right) \]  

\[ B_{z} D M I(s) = M X(s) S^{2} + (B_{z} + B_{v}) X(s) S + K_{v} X(s), \]

and after simplification, one obtains the transfer function between the FMV spool displacement and the driving current:

\[ G(s) = \frac{X_{*}(s)}{I_{*}(s)} = \frac{B_{z} D M N_{f}}{M_{v} S^{2} + (B_{z} + B_{v}) S + K_{v} + K_{f}} \]

\[ = \frac{B_{z} D M N_{f}}{S^{2} + (B_{z} + B_{v}) S + K_{v} + K_{f}} \]

(8)

where \( K \) is the transfer function gain of the FMV force motor system; \( Z_{i} (i = 1,2,3 \ldots m) \) and \( P_{j} (j = 1,2,3 \ldots n) \) represent all the zeros and all the poles of the system, respectively.

Through equation (8), one can separately find the FMV’s damping ratio \( \delta = \frac{B_{z} + B_{v}}{2 \sqrt{M_{v}(K_{v} + K_{f})}} \) and inherent frequency \( \omega_{n} = \sqrt{\frac{K_{v} + K_{f}}{M_{f}}} \) with its the frequency characteristic equation of

\[ G(j\omega) = \frac{B_{z} D M N_{f}}{M_{v} \left( B_{z} + B_{v} \right) j\omega M_{v} + K_{v} + K_{f}} \]

(9)

Through equation (10), one can respectively find the amplitude and the phase angle of the transfer function of the FMV force motor, \( A(\omega) \) and \( \varphi(\omega) \):

\[ A(\omega) = \sqrt{\left( \frac{B_{z} D M N_{f}}{M_{v}} \right) \left( \frac{K_{v} + K_{f} - \omega^{2}}{M_{v}} \right) + \left( \frac{\omega (B_{z} + B_{v})}{M_{v}} \right)^{2}} \]

(11)

\[ \varphi(\omega) = \tan^{-1} \left( \frac{-(B_{z} + B_{v}) \omega}{K_{v} + K_{f} - \omega^{2} M_{v}} \right) \]

(12)

Thus, the log amplitude-frequency characteristic of the FMV force motor system, \( L(\omega) \) is

\[ L(\omega) = 20 \log(A(\omega)) = 20 \log \left( \sqrt{\left( \frac{B_{z} D M N_{f}}{M_{v}} \right) \left( \frac{K_{v} + K_{f} - \omega^{2}}{M_{v}} \right) + \left( \frac{\omega (B_{z} + B_{v})}{M_{v}} \right)^{2}} \right) \]

(13)

The dynamic characteristics of a second-order system are represented in a series of indexes under the unit step response including the overshoot, adjustment time, peak time, rising time, stable state error, etc. [14]. The dynamic characteristics of FMV depend on its damping ratio and the inherent frequency, both of which in turn depend on the viscous damping coefficient, the damping coefficient generated by the transient hydrodynamic force, the mass of FMV’s moving parts, the stiffness of the centralizing
spring, and others. Therefore, in order to find the parameter values making the dynamic characteristics of the FMV force motor be more optimal, it is necessary to analyze the effects of these factors on FMV’s dynamic characteristics.

4. Simulation analysis of FMV static characteristics

The static characteristics of the servo valve can be assessed by the null bias, hysteresis loop, internal leakage, etc. under the unloaded state [15]. This paper applies AMESIM, the hydraulic system simulation software, to analyze the static characteristics of the FMV.

4.1 FMV static simulation model

Tables 1 and 2 list the structure parameter values used in the AMESIM simulation.

Table 1. FMV’s electric and hydraulic parameter values

| Hydraulic parameter | Value            | Name                      |
|---------------------|------------------|---------------------------|
| P_s                | 5 MPA            | Oil-supply pressure       |
| P                  | 850 kg/m³        | Density of oil            |
| M                  | 0.00432 N·s/m²   | Dynamic viscosity         |
| A                  | 69°              | Jet angle                 |
| E                  | 2.1X10⁷ N/m²     | Elastic modulus of oil    |
| f                  | 0.1 Hz           | Frequency of sine wave    |
| U                  | 0.002V           | Amplitude of sine wave    |

Table 2. Parameter values of FMV structure

| Structure parameter | Value            | Name                      |
|---------------------|------------------|---------------------------|
| d                   | 15 mm            | Spool diameter            |
| uLap                | -0.001~0.001     | Cover amount of valve orifice |
| M                   | 0.06 kg          | Spool mass                |
| δ                   | 0.01~0.3 mm      | Radial gap between spool and valve body |
| ε                   | 0.1~1            | Damping coefficient       |
| w                   | 0.002 mm         | Throttle orifice width    |

Figures 4 and 5 show the simulation models for FMV flow and pressure characteristics established through AMESIM.

**Figure 4.** Simulation model for FMV flow characteristics. **Figure 5.** Simulation model for FMV pressure characteristics.
4.2 Analysis of FMV static characteristics
Figures 6 and 7 show the simulation curves of FMV flow and pressure characteristics, respectively.

![Figure 6. Simulation curves of FMV flow characteristics.](image1)

![Figure 7. Simulation curves of FMV pressure characteristics.](image2)

In order to test the FMV’s static state operation curve, a static state test system was established according to the hydraulic principle schematic of FMV static state test platform, as shown in figure 8.

![Figure 8. Schematic of the hydraulic principle of FMV static test platform.](image3)

The current required for the FMV servo valve was 6 A, and the set pressure was 1.5 MPa. Figures 9 and 10 show the curves plotted based on the static tester-measured flow and pressure characteristics of the FMV servo valve.

![Figure 9. Characteristic curve of FMV flow.](image4)

![Figure 10. Characteristic curve of FMV pressure.](image5)

Simulation curves were obtained based on the relationships of the FMV’s flow and pressure to the spool displacement and corresponded one by one to the coordinate parameters of experimentally tested curves. In practical engineering, the size of current signal is not necessarily given in accordance with the size of rated current, it is enough to meet applications only using the actually tested current range.
From figures 6, 7, 9 and 10, it is obvious that experimentally test curves are roughly consistent with the simulation curves, with a smaller difference, thus indicating that the static research result obtained in this paper is good enough and can be used to detect static properties of the FMV.

4.3 Analysis of valve structure parameters impacting FMV static characteristics

The valve structure parameters impacting FMV static characteristics include the radial gap between its spool and sleeve, the cover amounts of spool and sleeve, null bias, and the form of the valve orifice [16]. Analytic results are given as follows:

1) The gap radial gap between spool and sleeve affects the FMV’s internal leakage, that is, increased radial gap will enlarge the internal leakage and reduce the control flow.

2) The amount of lap between spool and sleeve of the electro-hydraulic servo valve orifice includes the amount of zero, plus and minus laps. Their characteristics have been analyzed previously [17] and are not discussed in the study.

3) When no signal is inputted due to machining sizes, electromagnetic properties, and other causes, the output flow from the servo valve is not equal to zero. The FMV’s zero bias is set at about -0.8 A to prevent sudden disruption of power supply-resulted interconnection of the oil-supply chamber with the oil-control chamber, and further result acting-resulted roll damage.

4) The cross-section type of valve orifice comprises the cylindrical spool valve type, cone valve type, ball valve type, and axial triangular trough valve type. Different cross-section types have different section areas of flow, resulting in different gain characteristics of control flow.

5. Simulation analysis of FMV dynamic performance

The indexes used to measure the dynamic performance of the control system include stability, accuracy and rapidity of the system [18]. According to the transfer function of the FMV force motor system, i.e., equation (8), the root locus equations of the system can be found as follows:

\[
\prod (s+P)=M_s^2+(B_s+B_b)s+K_s+K_f=0
\]  

(14)

After theoretical analysis, the simulation software Matlab was used to make codes, plot diagram of the FMV force motor system, calculate the dynamic performance indicators, and assess the dynamic performance of the control system. After setting the parameters, the variable method was used to analyze the effects of different parameter values on the dynamic characteristics of the FMV force motor system. Let \( K_i=B_s+B_b \), \( K_2=K_v+K_f \) and \( K_3=B_a\pi DB_C \) and according to the above FMV structure parameter values, let \( M_s=0.02 \) kg, \( K_3=28800 \) N mA, \( K_2=28800 \) N/m, and let \( K_i \) be 4.8, 14.4, 24, 33.6, and 43.2. Simulation by bringing these values in the Matlab codes obtained the root locus plot (Figure 11), phase angle margin diagram (Figure 12), unit step response curve (Figure 13), unit pulse response curve (Figure 14) and Bode diagram (Figure 15) of the system at different \( K_1 \) values.

![Figure 11. FMV’s root locus plot at different \( K_1 \) values.](image)
Figure 12. FMV’s phase angle margin at different $K_1$ values.

Figure 13. FMV’s unit step response curve at different $K_1$ values.

Figure 14. FMV’s unit pulse response curve at different $K_1$ values.

Figure 15. FMV’s Bode plot at different $K_1$ values.

Figure 11 clearly shows that all the closed-loop poles of the system at different $K_1$ values lie in the left-half part of the complex plane, indicating the system is stable. From figure 12, figure 14 and figure 15, the stable phase angle margin, the peak time and the unit pulse response curve, and the frequency width of the FMV force motor system at different $K_1$ values can be obtained, respectively. From figure 13, the overshoot ($\sigma$), adjust time ($t_a$), peak time ($t_p$), rise time ($t_r$), etc. of the unit step signal can be found.
Table 3 lists all the data calculated based on the above-mentioned analysis. From the table, it can be seen that when other parameters of the FMV force motor system remain invariable, the dynamic performance of the system at $K_1=24$ or so best satisfies the actual requirement of production.

**Table 3. Effect of different $K_1$ values on FMV system performance**

| $K_1$ (unit step signal) | 4.8  | 14.4 | 24   | 33.6 | 43.2 |
|--------------------------|------|------|------|------|------|
| Phase angle margin (°)   | 16.26| 50.22| 90   | 163.7| 180  |
| Overshoot σ (%)          | 72.9 | 36.56| 15.83| 4.81 | 0    |
| Adjust time $t_a$ (s)    | 0.032| 0.0094| 0.0067| 0.0049| 0.0039|
| Rise time $t_r$ (s)      | 0.00094| 0.0011| 0.0014| 0.0018| 0.0024|
| Frequency width (Hz)     | 200  | 198  | 191  | 190  | 185  |
| Damping ratio δ          | 0.1  | 0.3  | 0.5  | 0.7  | 0.9  |
| Inherent frequency $W_n$ (rad/s) | 1200 | 1200 | 1200 | 1200 | 1200 |

6. Conclusions
This study investigates the FMV’s static and dynamic characteristics and optimizes its performance parameters using combined theoretical analysis and experimental verification approach as well as an integrated static and dynamic simulation method. Firstly, it analyzes the structural characteristics and working principle of Mitsubishi-Hitachi FMV, which laid a basis for establishing its theoretical model; Secondly, it explores the working principle of the internal permanent-magnet moving-coil force motor magnetic circuit, determined the transfer function based on the automatic control principle, established the mathematical model of the FMV force motor system, and revealed the influencing laws of these structural parameters on its performance, all of which provide references for the servo valve design; Thirdly, it establishes a simulation model of FMV’s static characteristics using AMESIM. The simulation results are roughly consistent with the experimental results; and fourthly, it sets up the database of FMV’s structure parameters, determines the dynamic characteristic indexes, and obtains the curves of root locus, phase angle margin, unit step response and unit pulse response with variable parameter $K_1$ and Bode plots, further analyzes its dynamic characteristics and influencing factors, and provides a basis for the improvement of FMV.

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