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Selection and Simulation of Electric Steering Gear Control System

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Abstract

For the electric steering gear control systems, the principle of the system is introduced. Specific model and parameters are selected to deduce the optimal combination of control system and choices method of the steering system model is determined. The transfer function of motor according to the parameters is achieved and simulation results are gained based on MATLAB. Simulation results show that this method can meet the performance requirements of electric steering gear, the control method is simple and easy to work to achieve.

Keywords*: electric steering gear, selection, simulation*

1. Introduction

With the rapid development of aerospace and all kinds of advanced precision guided weapons, the requirements of performance for missile's rudder system are becoming increasingly high. This paper is mainly on the research of actuator system selection of the trajectory correction projectile.

The steering gear is the key part of the flight-controlling system in the trajectory correction projectile, also is the actuator of the control system. In flight, the trajectory correction projectile achieves control surface deflection through the actuator, changing its projectile flight attitude to control the flight path of the projectile. The performance of the projectile's flight control system is directly influenced by the performance of the steering gear.

Here, we mainly study on the steering system in the following aspects of the model selection problem, then The schematic diagram of the electric actuator control system is shown in Figure 1 [1-4].

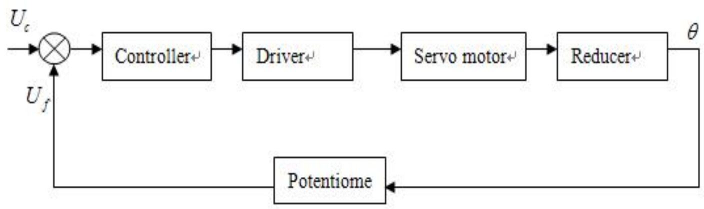

Figure 1. Electric Steering Control System Diagram

2. Selection of Electric Power Steering System

2.1. Selection of Reducer

Gear reduction unit, combined by bevel gear and spur gear, not only has good anti-overloading performance and economic application, but also has transmission efficiency as high as 80%, completely in accordance with the design requirements of electric actuator control system.

The steering gear speed reducer is composed of multi-stage gear group as the first stage reduction device, with the star wheel and the rudder connected to form a deceleration system and set the shifting fork. The reduction ratio of the part connected by fork and rudder

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surface is k_1; the tooth reduction ratio of the first gear and the shifting fork is k_2; the tooth reduction ratio of the first gear and the second gear is k_3; the tooth reduction ratio of the second gear and the third driven wheel is k_4; the tooth reduction ratio of the third driven wheel and the drive wheel is k_5;and the third drive wheel is directly connected with a motor shaft. According to the overall dimension of the steering gear system, the speed reducing gear group modulus size and requirements for design of structure machining process, we can pre-distribute reduction ratio of 100 for each stage, total reduction ratio of 500 [5, 6].

In the figure above, the moment of inertia of each stage is shown respectively:

$$M_{li} = k_i \times J_i \times \frac{d^2\theta q}{dt^2}$$

where k_i is the reduction ratio of each stage, J_i is the moment of inertia of each stage.

Moment of inertia is calculated as $M_l = \frac{d^2 q}{dt^2}$, the oscillation angle function of rudder wing varied with time is $\theta(t) = \theta_0 \sin 2\pi ft$, the swing frequency of the steering gear system is set to 6Hz, the swing range of declination is $\pm 15^0$.So we can get the important parameters for the follow-up calculation: the total moment of inertia of the steering gear.

$$M_l = M_{l1} + M_{l2} + M_{l3} + M_{l4} + M_{l5} = 0.1245 N \cdot m$$

2.2. Selection of Servo Motor

The servo motor is the key part of directly-controlled electric servo mechanism. Its performance directly determines the performance of servo mechanism. Therefore, in the scheme demonstration, we must choose it very carefully.

2.2.1. Load Analysis of Servo Mechanism

Static load torque of servo mechanism, mainly composed of hinge moment on the missile rudder M_h、friction torque of reducer M_f and moment of inertia on the output shaft of the servo mechanism $J\frac{d^2q}{dt^2}$, i.e. $M = M_h + M_f + J\frac{d^2q}{dt^2}$. The calculation formula for hinge moment is $M_h = \frac{1}{2} rV^2 m_h Sb = qm_h Sb$, where q --velocity head, and $q = \frac{1}{2} rV^2$, S - rudder area; V -Projectile velocity, b - average pneumatic wing chord, m_h - coefficient of hinge moment; $m_h = m_h^{\alpha}\alpha + m_h^{\phi}\phi$, where m_h^{α} -partial derivative of coefficient of hinge moment to α; m_h^{ϕ} - partial derivative of hinge moment coefficient to ϕ. Suppose attack angle of the rudder α is very small, then $M_h = qSbm_h^{\phi}\phi$. From this formula, we can conclude that hinge moment varies with the change of projectile's flight state, so the load torque of the servo mechanism varies and the rudder system's load is continuously changing [7, 8].

Conversion from load torque to the motor shaft: Suppose the output torque of the servo mechanism is M, the rotational speed is ω, the output power is P, the motor torque is M_m, the rotational speed is ω_m, the output power is P_m, reduction ratio is i, when the servo mechanism works in steady-state: $P=\eta P_m$, $M\omega=\eta M_m\omega_m$, $M_m=\frac{M\omega}{\eta\omega_m}=\frac{M}{\eta i}$; When the speed reducer works for multi-stage deceleration $M_m=\frac{M}{\eta_1\eta_2...\eta_n i_1 i_2...i_n}$.

○,1Conversion from moment of inertia to the motor shaft: the total kinetic energy converted should be equal to each kinetic energy of all moving parts in the servo mechanism, i.e. $\frac{1}{2}J^*\omega_m^2=\frac{1}{2}(J_m\omega_m^2+J_1\omega_1^2+J_2\omega_2^2+...J_n\omega_n^2)$.

Where J^* -total moment of inertia converted to the motor shaft; $J_1,J_2,...J_n$ -the moment of inertia to each rotation axis; $\omega_1,\omega_2...\omega_n$ - the corresponding rotational speed; J_m - The moment of inertia of the motor. When the transmission efficiency is taken into consideration, equivalent moment of inertia on the motor shaft is $J^*=J_m+\frac{J_1}{i_1^2\eta_1}+\frac{J_2}{i_2^2\eta_2}+...\frac{J_n}{i_n^2\eta_n}$, from this formula we can conclude that: Load torque of steering gear is the sum of the three, i.e. motor load torque equals the sum of hinge moment, friction torque and moment of inertia. Steering gear power is determined by the load torque and the maximum angular velocity, which can be generally considered as other load torques converted according to 20% of the hinge moment [9,10].

Here, we suppose that the structure size of tail is shown in Figure2.

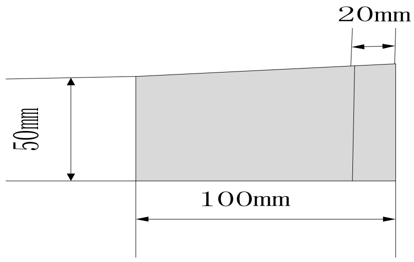

Figure 2. Tail Structure Size

From the design index of guided munitions we know that the flight velocity is 50-100m/s, here we calculate the power parameters of the motor by taking the limit velocity as 100m/s. Airflow velocity is taken by the maximum value V=300m/s, air density is taken as $\rho=1.293kg/m^3$, the maximum angle of rudder is $\omega_{\max}=15^0$. According to the principle of aero dynamics, when the rudder works at maximum deflection angle position, the resistance is: $F=\frac{1}{2}\rho V^2 S\cdot\frac{\omega_{\max}}{90^0}$, where S is the area of the rudder. For the tail of guided munitions, which usually takes the structure of one horizontal tail and two fins, the calculation of the area of tail is as follows: horizontal tail: $S_{HT}=\frac{c_{HT}\bar{c}S_W}{L_{HT}}$, fin: $S_{VT}=\frac{c_{VT}b_W S_W}{L_{VT}}$, where c_{HT}, c_{VT}, b_W, S_W L_{HT}, L_{VT}, $\bar{c}$ is pitch agility of the horizontal tail, pitch agility of the fin, wing span, Wing reference area, arm of force of the horizontal tail, arm of force of the fin, and the average pneumatic wing chord respectively. L_{HT}, L_{VT} are usually calculated as 40-50% of the vehicle length , here we take: $L_{HT}=L_{VT}=0.55m$; according to the table we can get $c_{HT}=0.7$, $c_{VT}=0.04$, $\bar{c}=0.0844$, $S_W=0.13$, $b_W=1.6$,According to the formula ,we calculated : $S_{HT}=0.014m^2$, $S_{VT}=0.015m^2$ $F=\frac{1}{2}\rho V^2 S\cdot\frac{15^0}{90^0}=16.1625N$.

The total moment together on the two rudders is $M_h = 4FL_1 = 3.6N\cdot m$, the minimum load force on the motor shaft is $M = 4.5165N\cdot m$.

2.2.2. Speed Selection of Servo Motor

The power of the motor is related to its structure parameters: $P_2 = c_1 D^2 L n_m$, where D - effective diameter of the motor(m); L - effective length of the motor m ; c_1 -constant.

From the above relations we know that if the motor works under certain power, the higher the rated speed, the smaller the structure size, as well as the smaller the moment of inertia. But if the speed of the motor is too high, the stages of reducer will be increased, and the structure will be more complex. We can see that the selection of the motor and the reducer is restricted to each other, so in the selection of the motor, it must be considered comprehensively the matching condition of parameters of both sides. The relationship of the rotation speed of the motor and servo speed is : $n_m = i \times n$. When the speed of the servo mechanism n is given, the speed of motor n_m or deceleration ratio can be determined according to the above formula, with the total reduction ratio of predetermined reducer numbering 500.

2.2.3. Selection of the Maximum Output Torque of Servo Motor

When the reducer ratio and efficiency is determined, according to the given maximum output torque of the servo mechanism, use the following formula to determine the maximum motor output torque, i.e. $M_{m\max} = \dfrac{M_{\max}}{in}$. From the above analysis and combined with the existing motor, we initially choose permanent magnet DC motor with the type 55ZYT06. Its main parameters are as follows: rated power: 35w, rated voltage: 27V, rated speed: 6000r/min, rated current; 1.4A, actuator output torque $3.6N\cdot m$, maximum load torque converted to the motor shaft is $4.5165N\cdot m$, the maximum swinging angular velocity of the rudder plate: $\omega \geq 250^0 rad/s$.

From the empirical formula, the motor armature resistance $R_a = \dfrac{U_N I_N - P_N}{2{I_N}^2} = 0.714\Omega$, where the rated voltage of the motor U_N is 27V, rated current I_N is 1.4A, rated power P_N is 35W, rated speed N_N is 6000r/min. The number of pole pairs P=1, according to the formula we can calculate the armature inductance $L_a = 3.82 \times \dfrac{U_N}{P \times N_N \times I_N} = 0.0127mH$, then $T_a = \dfrac{L_a}{R_a} = 0.018ms$.

According to the selection of motor, motor rotor is a cylinder with the height 96mm，and the diameter length 48mm, from where the rotor moment of inertia J^* can be calculated. We often take $\rho = 7.8 \times 10^3 kg/m^3$.

$$J^* = \frac{mr^2}{2} = \frac{1}{2}\rho v r^2 = \frac{1}{2}\pi\rho r^2 h r^2 == 3.9 \times 10^{-4} kg / m^2$$

Considering the influence of other factors, the actual moment of inertia of the motor rotor is 1.5 times more than J^* , i.e. $J_m = 5.85 \times 10^{-4} kg / m^2$.

The reverse electromotive constant of the motor: $K_t = 0.2$, $K_m = 0.2$, then $T_m = \dfrac{R_a J}{{K_t}^2} = 10ms$, from this we know the mechanical and electrical time constant is much larger than the electric time constant, so the electric time constant can be ignored.

3. Simulation Block Diagram

According to related equations of DC motor and the above parameters, we can get the following transfer function: $G_1(s) = \frac{78}{s+55}$, $G_2(s) = \frac{71.4}{s}$, $G_3(s) = \frac{500}{s^2+s+100}$. According to the above transfer function we can draw the DC motor dynamic structure graph, as shown in Figure 3.

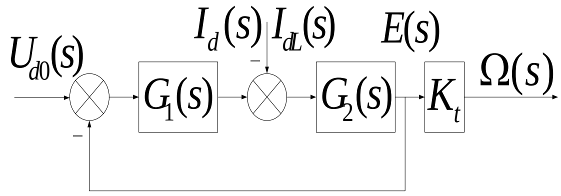

Figure 3. DC Motor Dynamic Structure Diagram

From Figure 3 we can see the servo motor has two inputs, one is the motor terminal voltage.

U_{d0} , the other is a load current I_{dL} , the former is the control input, the latter is the disturbance input. If the motor is under the ideal no-load, then I_{dL} =0. Under the circumstances ofcontinuous current, the transfer function between terminal voltage of the servo motor and speed can be obtained as:

$$G(s) = \frac{\Omega(s)}{U_{d0}(s)} = \frac{1/K_t}{T_m T_a s^2 + T_m s + 1}$$

Through the establishment and related calculation of mathematical model of the motor system, we can make the Simulink simulation block diagram of the steering control system in an actuator motor, which is as shown in Figure 4.

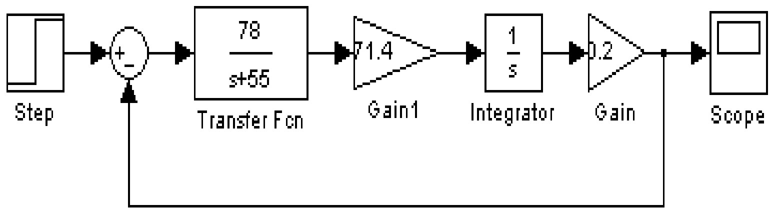

Figure 4. Simulation Diagram of Steering Control Actuator's System

4. Simulation Result

Use the Simulink software to get the simulation of the actuator in an electric steering gear, the input for the system is unit- step signal , then the simulation curve and Bode figure can be obtained , as shown in Figure 5 (the horizontal axis is the axis (s), the vertical axis is the amplitude) and Figure 6 respectively.

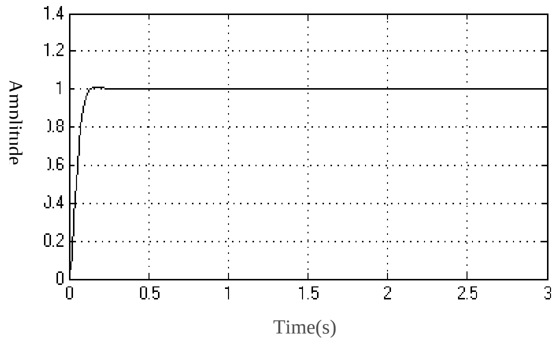

Figure 5. Simulation Curve of the Motor under the Unit Step Response

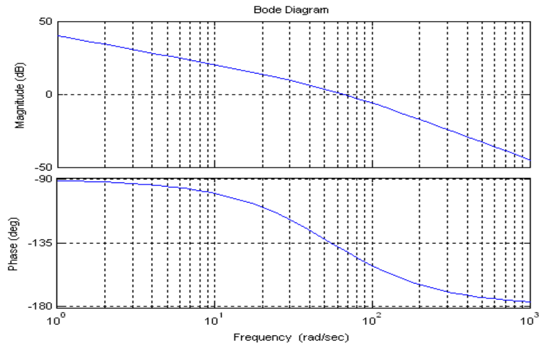

Figure 6. Open-loop Potter Figure of the Motor

5. Conclusion

After the model selection for electric actuator control system and relevant calculation, we have designed an actuator simulation controlling frame. Then through the analysis of system Bode chart , and further proved the accuracy of the mathematical model. It is worth pointing out, in this paper, the model selection and simulation methods has a certain reference to the study of other types of control system of random position.

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