Modeling and Analysis of Dual Motor Precision Transmission Mechanism

Jieji Zheng, Ruoyu Tan, Bin Yu, Dapeng Fan, Xin Xie *
(College of Intelligence Science and Technology, National University of Defense Technology, Changsha, 410073)

Corresponding author’s e-mail: xiexin12@nudt.edu.cn

Abstract. Aiming at the difficulty of fine modeling of dual-motor precision transmission mechanism, the modeling of friction and clearance of the system is studied. Firstly, the structure of the double-motor precision transmission mechanism is introduced, the system is simplified to a three-mass system, and a linear model of the system is established. On this basis, the improved stirbeck model is used to describe the friction nonlinearity inside the two-way transmission chain and the large ring gear, and the dead zone model is used to describe the gap nonlinearity between the pinion and the large ring gear, thus forming a complete dual-motor precision transmission mechanism kinetic model. An experimental device was built, and different excitation signals were used to verify the accuracy of the model. The experimental results show that the Pearson correlation coefficient between the established dynamic model and the actual system is >99% under various excitation signals, which shows the accuracy of the modeling method.

1. Introduction

In high-precision servo devices such as aerospace equipment, precision machine tools, industrial robots, and pointing mechanisms, transmission clearance has always been one of the most important factors affecting system performance [1, 2]. When the drive part loses direct contact with the load, backlash occurs in the servo system, causing the movement of the load to be autonomous, i.e. "uncontrollable". Therefore, the nonlinearity of the gap often leads to steady-state errors and even oscillation instability, which seriously reduces the control performance and stability of the equipment. In recent years, more and more researchers have used dual-motor precision transmission mechanisms to eliminate backlash in high-precision servo devices. This method can completely eliminate the gap on the basis of ensuring the system servo accuracy [3, 4].

Before 2010, most scholars focused on the anti-backlash method and synchronous control of the dual-motor transmission mechanism, and most of them stayed in the linear model in terms of modeling analysis, which could not meet the requirements of system characteristic analysis. In recent years, some scholars have begun to explore the modeling method of dual-motor precision transmission mechanism. Wen[5] established the dynamic model of the double pinion system considering the complex influence of nonlinear mesh stiffness and reverse impact between the pinion and the bull gear. The obtained model is helpful to design the intelligent controller of the double pinion system. Masahiko [6] established a four-mass model of a dual-motor mechanism. The model considered the rigidity of the connecting shaft between the large ring gear and the load, and used the model to estimate the load speed converted to the motor shaft, and dynamically calculated the relationship between the load speed and the motor speed. The difference value is added to the speed command to suppress the transient vibration generated at the load, and a better control effect is achieved. Zhao[7] proposed a model of the linear part of the system.
for the control problem of a dual-motor drive servo system with dead-band nonlinearity and used the description function method to analyze the dead-band nonlinearity. The simulation results show that due to the existence of dead-band nonlinearity, the error curve peaks at the commutation instant of the sinusoidal response. FENG[8] established a complex dynamic model of the dual-motor system by considering the effects of backlash nonlinearity, periodic time-varying stiffness of the gear train, random wind disturbance torque and motor cogging torque, and designed for the dynamic optimization of the bias torque. Provides a model reference. Wei[9] established the overall dynamic model of the dual-motor coupling drive system (DCDS) considering backlash nonlinearity, periodic time-varying stiffness of the gear train, random wind disturbance torque and motor cogging torque, and designed for the dynamic optimization of the bias torque. Provides a model reference. The above research provides a reference for the optimization of the model of the dual-motor transmission mechanism, but ignores the importance of the nonlinear friction of the system, and simplifies the friction as viscous damping for analysis, resulting in a low degree of fitting between the model and the actual system, which cannot be truly described The influence of nonlinearities such as clearance and friction on system characteristics. In this paper, the friction characteristics of the two-way transmission chain and the large ring gear of the dual-motor precision transmission mechanism are described by the improved stribeck model, and the dead zone model is used to describe the gap nonlinearity of the system. The dynamic model of the dual-motor precision transmission mechanism was established, and finally an experimental device was built to verify the high accuracy of the model.

2. Dynamic Model of Double Motor Precision Transmission Mechanism
The dual-motor precision transmission mechanism is usually driven by two groups of permanent magnet synchronous motors and planetary reducers with the same nominal parameters, respectively, to drive pinions with the same module and number of teeth. "Large gear ring") on both sides of the outer ring, the outer ring of the large gear ring is fixedly installed on the base, and the inner ring is fixedly connected with the rotating tower base. After the two groups of pinions mesh with the outer ring gear of the large ring gear, they complete the rotation and drive the rotating tower with the load to revolve. The schematic diagram of the transmission structure is shown in Figure 1. The planetary reducer selected for the first-stage transmission chain of the dual-motor precision transmission mechanism is a precision backlash, and the transmission accuracy is high. The transmission with the large ring gear is analyzed emphatically.

According to the dual-motor precision transmission mechanism shown in Figure 1, it is simplified to the three-inertia dynamic model shown in Figure 2. The meaning of the variables in the figure will be explained later. The two-way transmission chain in the dual-motor precision transmission mechanism belongs to parallel connection. relationship, the meaning of the parameters is exactly the same, therefore, one of the transmission chains can be analyzed as an example.
First, the armature loop equation of the motor is:

\[ i = iR + L \frac{di}{dt} + Ke \theta_m \]  

(1)

where \( u \) is the armature voltage of the motor; \( i \) is the armature current of the motor; \( R \) is the armature resistance of the motor; \( L \) is the armature inductance of the motor; \( eK \) is the back EMF coefficient of the motor; \( m\theta \) is the rotation angle of the motor.

The drivers of the two motors are set to the current loop mode, and the bandwidth of the current loop of the driver is more than 1KHz, which is much higher than the response bandwidth of the system speed loop, so the driver can be approximately regarded as a proportional link.

\[ i = K_d u \]  

(2)

where, \( K_d \) is the driver conversion factor.

The electromagnetic torque of the permanent magnet synchronous motor is proportional to the armature current, so there is

\[ T_m = K_M i \]  

(3)

where, \( T_m \) is the electromagnetic torque of the motor; \( K_M \) is the torque coefficient of the motor.

Assuming that the planetary reducer is an ideal transmission link, the force analysis of the pinion can be obtained:

\[ N_1T_m = (N_2J_m + J_1)\dot{\theta}_x + T_f + \tau \]  

(4)

where, \( N_1 \) is the reduction ratio of the planetary reducer, \( J_m \) is the moment of inertia of the motor, \( J_1 \) is the moment of inertia of the planetary reducer, \( \theta_x \) is the rotation angle of the pinion at the output end of the planetary reducer, and \( \tau \) is the output torque of the gear at the output end of the planetary reducer. \( T_f \) is the friction torque of the motor and planetary reducer equivalent to the pinion end, and the friction torque is represented by the improved stribeck model [11]:

---

Fig. 1. Schematic diagram of the double-motor precision transmission mechanism

Fig. 2 Three-inertia dynamic model framework
Fig. 3 Improved striebeck model

\[
T_f = \begin{cases} 
T_c^+ + (T_s^+ - T_c^+)e^{-(w/\Omega_+)} + B^+ w & w \geq w_{slip} \\
T_c^- + (T_s^- - T_c^-)e^{-(w/\Omega_-)} + B^- w & w \leq -w_{slip} \\
(w/w_{slip})^2 T_p & 0 \leq w \leq w_{slip} \\
(w/w_{slip})^2 T_n & -w_{slip} \leq w \leq 0 \\
0 & \text{else}
\end{cases}
\]  

(5)

where

\[T_p = T_c^+ + (T_s^+ - T_c^+)e^{-(w/\Omega_+)} + B^+ w\]

\[T_n = T_c^- + (T_s^- - T_c^-)e^{-(w/\Omega_-)} + B^- w\]

(6)

where \(T_s^+, T_s^-, T_c^+, T_c^-, \Omega_+, \Omega_-, B^+, B^-, w_{slip}\) are the positive and negative static friction, Coulomb friction, striebeck velocity, viscous damping coefficient and pre-slip zone, respectively.

The dead-band model is widely used to describe the backlash nonlinearity of the control system [12-16]. Let the backlash width be 2, as shown in Figure 3, then the gap between the output gear of the planetary reducer and the outer ring gear of the large ring gear. The gap can be described as:

\[
\tau = \begin{cases} 
K(z - \Delta) + c\dot{z} & z > \Delta \\
0 & |z| \leq \Delta \\
K(z + \Delta) + c\dot{z} & z < -\Delta 
\end{cases}
\]  

(7)

where \(K\) is the meshing stiffness of the pinion gear and the large ring gear; \(z = \theta_e - N_1 \theta_L\) is the transmission error between the output gear of the planetary reducer and the large ring gear; \(N_1\) is the gear ratio of the pinion gear and the large ring gear; \(\theta_L\) is the rotation angle of the rotating tower base.

According to the force of the rotating tower, the dynamic equation of the rotating tower is:

\[
\tau_1 + \tau_2 = J_L \ddot{\theta}_L + T_{fL}
\]  

(8)

where \(\tau_1, \tau_2\) are the torques output by the two transmission chains to the large ring gear, \(J_L\) is the moment of inertia of the rotating tower base, and \(T_{fL}\) is the friction of the large ring gear, which can also
be represented by an improved stribeck model. The simultaneous equations (2)-(8) can obtain the dynamic model of the dual-motor precision transmission mechanism including the gap between the output gear of the planetary reducer and the large ring gear, and draw the model block diagram as shown in Figure 4.

![Fig. 5. Block diagram of the dynamic model of the dual-motor precision transmission mechanism](image)

### 3. Model Experiment Verification

In order to verify the accuracy of the established model, an experimental test device for a dual-motor precision transmission mechanism was built, and the dSPACE hardware-in-the-loop simulation platform was used to carry out model verification experiments.

#### 3.1. Experimental device

A dual-motor precision transmission mechanism experimental device as shown in Figure 6 was built, mainly consisting of 2 permanent magnet synchronous motors (model: SPALY80), 2 motor drivers (model: Elmo P/N: SOL-WHI 20/100PYE), 2 L-shaped planetary reducers (model: FABR060-25-S2-P1), weapon station azimuth platform, absolute encoder (model: CAPRO-B112050), fiber optic gyro (model: FOG-118), 24V power supply, 48V It consists of power supply, dSPACE1104 and industrial computer.

![Fig. 6 Experimental setup](image)
By referring to the specific model and key parameters of the experimental equipment, the device was tested and identified, and the system parameter values shown in Table 1 and the friction model parameter values shown in Table 2 were obtained.

Table 1 System parameter values

| Parameter                      | Symbol  | Value |
|--------------------------------|---------|-------|
| Drive conversion factor       | $K_d (A/V)$ | 3     |
| Motor torque coefficient      | $K_m (N\cdot m \cdot A^{-1})$ | 0.13  |
| Motor inertia                 | $J_m (kg \cdot m^2)$ | 2e-4  |
| Planetary reducer inertia     | $J_r (kg \cdot m^2)$ | 9e-6  |
| Inertia of rotating tower     | $J_t (kg \cdot m^2)$ | 2.5   |
| Planetary reducer reduction ratio | $N_1$ | 25    |
| Ratio of pinion to large ring gear | $N_2$ | 8.25  |
| Meshing stiffness of pinion and large ring gear | $K (N\cdot m\cdot rad^{-1})$ | 1.3e5 |
| Mesh damping between pinion gear and large ring gear | $c (N\cdot m\cdot rad^{-1}\cdot s)$ | 1     |
| Axis 1 pinion and large ring gear clearance | $\Delta_{1} (arcmin)$ | 14    |
| Axis 2 pinion and large ring gear clearance | $\Delta_{2} (arcmin)$ | 10    |

Table 2 Friction model parameter values

| Friction of Motor and planetary reducer | Friction of rotating tower |
|-----------------------------------------|---------------------------|
| Parameter                               | Symbol | Value | Parameter | Symbol  | Value |
| Positive static friction                | $T_s^+ (N\cdot m)$ | 3     | Positive static friction | $T_s^- (N\cdot m)$ | 23    |
| Negative static friction                | $T_s^- (N\cdot m)$ | -2    | Negative static friction | $T_s^- (N\cdot m)$ | -20   |
| Positive Coulomb friction               | $T_c^+ (N\cdot m)$ | 2.8   | Positive Coulomb friction | $T_c^- (N\cdot m)$ | 20    |
| Negative Coulomb friction               | $T_c^- (N\cdot m)$ | -1.8  | Negative Coulomb friction | $T_c^- (N\cdot m)$ | -17   |
| Positive stribeck speed                 | $\Omega_s^+ (deg\cdot s^{-1})$ | 2     | Positive stribeck speed | $\Omega_s^- (deg\cdot s^{-1})$ | 1     |
| Negative stribeck speed                 | $\Omega_s^- (deg\cdot s^{-1})$ | -2    | Negative stribeck speed | $\Omega_s^- (deg\cdot s^{-1})$ | -1     |
| Positive Viscous Damping Coefficient   | $B_s^+ (N\cdot m\cdot deg^{-1}\cdot s)$ | 0.01  | Positive Viscous Damping Coefficient | $B_s^- (N\cdot m\cdot deg^{-1}\cdot s)$ | 1.2    |
| Negative Viscous Damping Coefficient   | $B_s^- (N\cdot m\cdot deg^{-1}\cdot s)$ | 0.008 | Negative Viscous Damping Coefficient | $B_s^- (N\cdot m\cdot deg^{-1}\cdot s)$ | 1     |
| Pre-slipe area                          | $\omega_{slipe} (deg\cdot s^{-1})$ | 0.2   | Pre-slipe area | $\omega_{slipe} (deg\cdot s^{-1})$ | 0.05 |
3.2. Experimental results and analysis

In order to fully verify the accuracy of the model, the difference between the model response and the actual speed signal under different excitation signals is compared. The excitation signals include sinusoidal signals and square wave signals with different amplitudes and frequencies. The experimental results are shown in Figure 6.

It can be seen from the figure that the overall fit between the model response and the actual speed curve is high, but there is a large jitter in the model response during the commutation process, while the actual system is relatively smooth. The reason is that the model adopts the dead zone model. Describing the gap nonlinearity, the disadvantage of this model is that the driving part and the driven part do not transmit torque at all in the range of the gap. During the simulation analysis, it is found that the output torque of the dead zone model in the low-speed section of the driving part and the driven part after crossing the gap is not equal. This phenomenon is different from the actual system, so it is necessary to correct the dead zone model to correctly describe the nonlinear characteristics of the system gap.

In order to quantify the accuracy of the model, the Pearson correlation coefficient was used to evaluate the fitting degree of the model response curve and the actual speed signal curve. The Pearson correlation coefficient is the ratio of the product of the covariance and the standard deviation of two variables, which is used to describe the degree of correlation between the two sets of data X and Y. The more correlated the data, the closer the value of the Pearson correlation coefficient is to 1. The expression for the Pearson correlation coefficient is:

\[ r = \frac{\text{cov}(X, Y)}{\sqrt{\text{var}(X) \cdot \text{var}(Y)}} \]
\[ \gamma_{\text{Pearson}} = \frac{\sum_{i=1}^{n} (x_i - \bar{x})(y_i - \bar{y})}{\sqrt{\sum_{i=1}^{n} (x_i - \bar{x})^2 \sum_{i=1}^{n} (y_i - \bar{y})^2}} \]  

The statistical results of the Pearson correlation coefficient are calculated according to Figure 7, as shown in Table 3. From the statistical results, it can be seen that the fitting degree between the model response and the actual speed is >99%, which verifies the accuracy of the model.

Table 3 Statistical results of Pearson correlation coefficient

| Experiment | Results |
|------------|---------|
| (a) 0.6\sin(2\pi t)[V] | 0.9941 |
| (b) 0.6\sin(2\pi t)[V] | 0.9935 |
| (c) 0.75\sin(\pi t)[V] | 0.9957 |
| (d) 0.75\sin(2\pi t)[V] | 0.9967 |
| (e) 0.6\text{square}(0.8\pi t)[V] | 0.9968 |
| (f) 0.75\text{square}(0.8\pi t)[V] | 0.9973 |

4. Conclusion

Aiming at the low accuracy of the dynamic modeling of the dual-motor precision transmission mechanism, this paper simplifies the mechanism into a three-mass system, focusing on two key nonlinear factors, system friction and clearance. The friction between the inside of the two-way transmission chain and the three parts of the large ring gear is represented by the improved stribeck model, and the gap between the pinion and the large ring gear is described by the dead zone model, forming a mechanism dynamic model, and finally the model is verified by experiments. The established model can lay a theoretical foundation for the subsequent nonlinear dynamic analysis and high-precision control of the dual-motor precision transmission mechanism.

Acknowledgments

This work was funded by National Key R&D Program of China (Grant No. 2019YFB2004700) and Young Teacher Innovation Research Project (ZN2019-7).

References

[1] Zhao, W., X. Ren and L. Li, Synchronization and Tracking Control for Dual-motor Driving Servo Systems with Friction Compensation. Asian journal of control, 2019. 21(2): p. 674-685.
[2] Zhao, W., X. Ren and X. Gao, Synchronization and tracking control for multi-motor driving servo systems with backlash and friction. International journal of robust and nonlinear control, 2016. 26(13): p. 2745-2766.
[3] Fang L, Sun L. Backlash Elimination Characteristics of Dual Motor Driving Systems[J]. China Mechanical Engineering, 2012. (in Chinese)
[4] Liang R, Fang Q. Dual-drive anti-backlash system based on torque compensation[J]. Journal of Mechanical & Electrical Engineering, 2010. (in Chinese)
[5] Wen, C., Modelling and Simulation of Dual-Pinion Driving Systems for Backlashes Elimination. International Journal of Modelling & Simulation, 2010. 30(2): p. 178-158.
[6] ITOH, M., Torsional Vibration Suppression of a Twin-drive Geared System Using Model-based Control, in 10th IEEE International Workshop on Advanced Motion Control (AMC’08), vol.1. 2008: Trento, Italy. p. 176-181.
[7] Zhao, H., A Compensation Approach of Dead-Zone Nonlinearity in Dual-Motor Driving Servo System. Applied Mechanics and Materials, 2013.
[8] Feng J P, Ma W L, Huang J L. Anti-backlash Research of Telescopes Driven by Dual-bias Motors Using Dynamic Method[J]. Opto-Electronic Engineering, 2009.
[9] Fan, W., Y. Yang and X. Su, Dynamic Modeling and Vibration Characteristics Analysis of Transmission Process for Dual-Motor Coupling Drive System. Symmetry (Basel), 2020. 12(7): p. 1171.
[10] Jiang, H., et al., Elimination of Gear Clearance for the Rotary Table of Ultra Heavy Duty Vertical Milling Lathe Based on Dual Servo Motor Driving System. Applied sciences, 2020. 10(11): p. 4050.

[11] Zhi-qiang, L., et al., Prestiction friction compensation in direct-drive mechatronics systems. Journal of Central South University, 2013. 20(11).

[12] Sun, G., J. Zhao and Q. Chen, Observer-based compensation control of servo systems with backlash. Asian journal of control, 2021. 23(1): p. 499-512.

[13] Kang, M.J., et al., A disturbance observer design for backlash compensation. International Journal of Control Automation & Systems, 2011. 9(4): p. 742.

[14] Villwock, S. and M. Pacas, Time-Domain Identification Method for Detecting Mechanical Backlash in Electrical Drives. IEEE transactions on industrial electronics (1982), 2009. 56(2): p. 568-573.

[15] Nordin, M. and P.O. Gutman, Controlling mechanical systems with backlash—a survey. 2002. 38(10): p. 1633-1649.

[16] Zhao G F, Fan W H, Chen Q W, et al. Survey on backlash nonlinearity[J]. Binggong Xuebao/Acta Armamentarii, 2006, 27(6):1072-1080. (in Chinese)