Research on Disturbance Rejection Control Method of Mechanical Transmission System Based On Torque Compensation

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Abstract. In order to improve the immunity ability of mechanical transmission system and reduce the dynamic load of transmission system caused by impact load, a disturbance rejection control method for mechanical transmission system is proposed based on torque compensation. The servo control modal parameters of mechanical transmission system are identified by fuzzy identification and inertial fusion. The inverse kinematics solution of the control structure of the mechanical transmission system is carried out by using the inverse kinematics decomposition method of the end effect, and the torque compensation and the optimization identification of the mechanical parameters of the mechanical transmission system are carried out by the modified DH parameter method. The dynamic load suppression problem is transformed into the torque parameter optimization problem of the mechanical transmission system. The motor torque is controlled to suppress the dynamic load of the mechanical transmission system. The Kalman filter is used to estimate the aerodynamic torque load of the system. The optimization of anti-disturbance control of mechanical transmission system is realized. The simulation results show that the proposed method has good stability and can effectively attenuate the dynamic load of the transmission system caused by the external abrupt load.

1. Introduction
The transmission system of the mechanical equipment is one of the main equipment of the complete set of fully mechanized mining equipment, which plays a decisive role in improving the productivity and efficiency of the working face. The electromechanical coupling system composed of gear transmission system and cutting drum takes up 80%–90% of the motor power of the whole mechanical equipment. Because of the bad cutting environment of the transmission system of the mechanical equipment, the external load acting on the drum has the characteristics of randomness and strong impact, which makes the transmission system of the mechanical equipment become one of the weakest parts of the whole machine [1]. In order to improve the anti-disturbance ability of the mechanical transmission system, In order to reduce the dynamic load of transmission system caused by impact load, it is necessary to optimize the anti-disturbance control design of mechanical transmission system [2]. It has great significance to study the anti-disturbance control method of mechanical transmission system.
At present, more and more scholars pay attention to the performance of mechanical transmission system, but the main research focus is on the reliability optimization of transmission system, the optimization of drum cutter arrangement, the innovative design of transmission system, and so on [3, 4]. The undamped nonlinear mechanical drive control of mechanical transmission system mainly includes PID neuron adaptive control, mechanical transmission control model based on virtual prototyping technology, mechanical transmission control based on fuzzy control theory, etc. The mechanical transmission control system with PID neuron adaptive control is the most common. By establishing the PID neuron adaptive control scheme, the fuzzy control law is established. When the transmission control voltage of the mechanical transmission system rises suddenly, the output power appears beat resonance. The performance is not good. In reference [5], a nonlinear control model of mechanical transmission system based on nonlinear control design of static non-power compensator is proposed, in which linear feedback linearized superconducting moment of inertia is designed. A damping controller based on the controlled object model is established. The method relies on more accurate traditional system parameters. In fact, the structure of electromechanical transmission system is complex and the parameters are time-varying, so it is difficult to obtain accurate system parameters [6].

In order to solve the above problems, a disturbance rejection control method for mechanical transmission system based on torque compensation is proposed in this paper. The modal parameters of servo control of mechanical transmission system are identified by fuzzy identification and inertial fusion. The dynamic load suppression problem is transformed into the torque parameter optimization problem of the mechanical transmission system. The motor torque is controlled to suppress the dynamic load of the mechanical transmission system. The Kalman filter is used to estimate the aerodynamic torque load of the system. The optimization of anti-disturbance control of mechanical transmission system is realized. Finally, the simulation results show that the proposed method can improve the anti-disturbance control ability of mechanical transmission system.

2. Electromechanical coupling model and modal parameter analysis of transmission system

2.1. Electromechanical coupling model of transmission system

The mechanical transmission device studied in this paper is long chain parallel shaft gear cutting machine, including cutting motor, gear transmission system and cutting drum. Gear transmission system includes multistage parallel shaft gear and first grade planetary gear. The asynchronous motor used in the cutting part of the shearer is a high order nonlinear, strongly coupled multivariable system [7]. The mathematical model of the asynchronous motor under d-q axis coordinates is expressed as follows:

\[ f_0(X) = w_p P_1(X) + w_v V(X) + w_c C(X) + \frac{1}{\delta} \left[ f_n \left( 1 - \frac{T_\text{max}}{\omega_n} \right) + f_n \left( 1 - \omega_{T,\text{max}} \right) + f_n (B_n / B_{y,\text{max}} - 1) \right] \]

The flux equation is:

\[ \frac{\partial J_{\text{macro}}(n)}{\partial f_j(n)} = \frac{1}{4} \left[ \sum_{p=1}^{n} \frac{\partial J_{R,p}(n)}{\partial f_j(n)} + \sum_{p=1}^{n} \frac{\partial J_{I,p}(n)}{\partial f_j(n)} \right] \]

Fuzzy identification and inertial fusion of the servo control modal parameters of the mechanical transmission system are carried out. According to the principle of Lyapunov stability [8], the error function is obtained as follows:

\[ e_{R,j} = (y_{R,j}(n) - R_{2,0}) \times y_{R,j}(n)^* \]
The iterative equation for dynamic load disturbance rejection control of mechanical transmission system is described as follows:

\[
f_f(n+1) = f_f(n) + \mu_{HCM} \times \left( \left| y_{R,j}(n) \right|^2 - R_{z,k} \right) y_{R,j}(n) + j \left| y_{I,j}(n) \right|^2 - R_{z,I} \right) y_{I,j}(n) \times x_j(n)
\]

(4)

The electromechanical coupling dynamic model of transmission system of long chain parallel shaft gear cutting machine is established. The initial value of \((u_0, u_t) \in H^n \times H^n\) satisfies:

\[
\| (u_0, u_t) \|_{H^n \times H^n} \leq A \ 
\| w(t)(u_0, u_t) \|_{H^n \times H^n} \leq \eta
\]

(5)

The Hankel matrix of input transmission system dynamic load and output converges to:

\[
E[\tilde{X}(n)y_j(n-k)] = 0
\]

(6)

Where \(k = K - 1\), \(X(n)\) is Toeplitz matrix of \(\tilde{X}(n)\), ADRC control is introduced into the field of dynamic load suppression of transmission system. The electromechanical coupling model of mechanical transmission system is established by considering the dynamic characteristics of motor, the time-varying meshing stiffness of gear and the load characteristics of roller.

### 2.2. Modal parameter analysis

Fuzzy identification and inertial fusion of servo control modal parameters of mechanical transmission system are carried out. The kinematics inverse solution of the control structure of mechanical transmission system is carried out by using the method of end effect inverse kinematics decomposition, and the target position are obtained. The distribution of common mode control constraint parameters of the initial joint is expressed as follows:

\[
u_{i,j}(k-1/k-1) = P(m_i(k-1)/m_j(k), z^{k-1}) = \frac{1}{e^{\eta}} P_{\eta}(k-1)
\]

(7)

The modified DH parameter method is used to adjust and fuse the end pose parameters of the mechanical transmission system. The mechanical control model of the end pose of the mechanical transmission system is established as:

\[
\tilde{x}^{0}(k-1/k-1) = \sum_{i} \tilde{x}^{0}(k-1/k-1)u_{i,j}(k-1/k-1)
\]

(8)

\[
P^{0}(k-1/k-1) = \sum_{j} u_{i,j}(k-1/k-1)[P^{0}(k-1/k-1) + [\tilde{x}^{0}(k-1/k-1) - \tilde{x}^{0}(k-1/k-1)][\tilde{x}^{0}(k-1/k-1) - \tilde{x}^{0}(k-1/k-1)]^{T}]^{-1}
\]

(9)

The optimal characteristic solution of the end position and attitude estimation of the mechanical transmission system is described as follows:

\[
x_i(t) = \sum_{k=0}^{n} a_{i,k} x_i(t-k) + \varepsilon_i(t)
\]

(10)
The motor has strong robustness to parameter variation. The DTC control with the advantages of fast torque dynamic response and so on. The stator flux and torque are controlled by the flux hysteresis comparator and the torque hysteresis comparator respectively. The error is stabilized at the hysteresis width, and the electromagnetic torque of the motor can be controlled according to the stator magnetic field. The chain and stator current are estimated to obtain the meshing force of the inner and outer meshing pairs.

3. Optimization of disturbance rejection control model for traditional systems

3.1. Torque compensation and mechanical parameter optimization identification of mechanical transmission system

On the basis of establishing the electromechanical coupling model and modal parameter analysis of the transmission system, fuzzy identification and inertia fusion of the servo control modal parameters of the mechanical transmission system are carried out, and the optimal design of the control model is carried out. The disturbance rejection control method of mechanical transmission system based on torque compensation, the modified DH parameter method for torque compensation and mechanical parameter optimization identification of mechanical transmission system [9], and the Newton mechanics of non-inertial system are used to obtain the planetary gear motion. Mathematical model of mechanics, curvature fitting equation of neutral axis of section are expressed as:

\[ K_{k+1/k+1} = \rho_k \frac{m_k}{m_{k+1}} \Phi_{k} K_{k/k} \Phi_{k}^T + \frac{1}{m_{k+1}} \delta K \]  

(12)

Where, the oxy inverse kinematics characteristics of solar wheel and inner gear ring in the dynamic coordinate system are expressed as follows:

\[ \delta K = \begin{bmatrix} S - \sigma I \n z \end{bmatrix} \varepsilon \]

(13)

\[ m_k = \sum_{i=1}^{n} a_i, \quad m_{k+1} = m_k + \delta m_{k+1} \]  

(14)

\[ \sigma = \frac{1}{m_k} \sum_{i=1}^{n} a_i b_i r_i, \quad B = \frac{1}{m_k} \sum_{i=1}^{n} a_i b_i r_i^T \]  

(15)

According to the meshing frequency, the comprehensive stiffness of gear teeth is simplified to the periodic function of rectangular wave variation, and the torque compensation characteristic of mechanical transmission system satisfies the attitude fusion inertial parameter set \( p_{obj} \in \mathbb{R}^6 \), mechanical transmission of \( g_c = \{ g_0, ..., g_N \} \), mechanical transmission system in C-space. The path search direction of the system is: \( \tau : [0,1] \rightarrow C_{free} \). When the shaft length between the two gears of \( \tau[1] = \theta_{goal} \) is satisfied, the dynamic load of the \( (p_{obj}, g_c) \rightarrow f(\theta_{goal}) \), mechanical transmission system satisfies the following constraints:
When the mechanical parameters of the electromechanical transmission system are fixed, the superposition of the compensation torque on the basis of the original motor torque can be suppressed, so the dynamic load suppression problem of the transmission system can be converted into the torque parameter optimization problem of the mechanical transmission system [10].

3.2. Disturbance rejection control law for mechanical transmission system

Under the condition of adaptive error adjustment, the kinematics and dynamics characteristic solutions of attitude adaptive adjustment of mechanical transmission system are expressed as follows:

\[
\eta_i(t) = \begin{bmatrix} x^T(t) & x^T(t+r) \end{bmatrix}^T
\]

\[
\eta_s(t) = \begin{bmatrix} y^T(t) & y^T(t-\sigma) \end{bmatrix}^T
\]

The Lyapunov stability of the controlled model of the mechanical transmission system is analyzed by combining the above formulas. The state variables of the truncated transmission system are tracked by ESO, and the total disturbances (including non-linearity and time-varying parameters) are estimated in real time. The real-time action of the total disturbances is compensated to the actual control variables to satisfy the following requirements:

\[
\begin{bmatrix} r_{11} & 0 & 0 & 0 & p & -\tau Z_i & 0 \\
0 & r_{22} & r_{23} & r_{24} & 0 & 0 & Y_{26} \\
0 & 0 & 0 & 0 & 0 & 0 & \sigma D Z_i \\
0 & 0 & 0 & 0 & 0 & 0 & \tau Z_i \\
0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & -\sigma Z_i \\
0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 \\
\end{bmatrix} < 0
\]

The aerodynamic torque load of the system is estimated by Kalman filter. The parameters of ESO in the dynamic load ADRC controller of transmission system are estimated as follows:

\[
\begin{bmatrix} R_1 & E \end{bmatrix} > 0, \quad \Psi_1 = \begin{bmatrix} X & N \\ * & Z_i \end{bmatrix} \geq 0, \quad \Psi_2 = \begin{bmatrix} Y & M \\ * & Z_j \end{bmatrix} \geq 0
\]

The real time action of the total disturbance is compensated to the actual control by using disturbance compensation method. The actual output torque of ADRC after disturbance compensation is expressed as follows:

\[
\begin{align*}
\tau_d &= C \cdot \tau_i = D \sin \theta \cos \phi + \frac{D}{2r} \sin^2 \theta \cos \phi \sin \phi \\
\tau_d &= C \cdot \tau_i = D \sin \theta \cos \phi + \sin \phi \\
\tau_d &= C \cdot \tau_i = D \sin \theta \sin \phi + \frac{D}{2r} \sin^2 \theta \cos \phi \sin \phi
\end{align*}
\]

The aerodynamic torque load of the system is estimated by Kalman filter. The parameters of ESO in the dynamic load ADRC controller of transmission system are estimated as follows:
\[ T_i(q_i) = \prod_{j=1}^{7} (1 - a_{ij}) \]

\[ = \left[ \begin{array}{ccc} n & o & a & p \\ 0 & 0 & 0 & 1 \end{array} \right] \left[ \begin{array}{c} \sum_{j=0}^{i} \sum_{j=0}^{i} \sum_{j=0}^{i} b_{ij} \cdot x_{ij} \\ \sum_{j=0}^{i} b_{ij} \cdot \Pi_{ij} \end{array} \right] \]

Based on the above analysis, torque compensation method is used to optimize the disturbance rejection control of mechanical transmission system.

4. Simulation experiment and result analysis

In order to test the application performance of this method in anti-disturbance control of mechanical transmission system, the simulation experiment is carried out. The simulation is based on Matlab Simulink software and the control parameters are designed as \( N_p = N_r = 20 \), \( f = 24.5 \text{KHz} \), \( C_p = C_r = 0.9 \mu F \). The input voltage is 24 V. The stator resistance of the undamped nonlinear mechanical transmission system of the mechanical transmission system is \( R_s = 0.7348 \Omega \), rotor resistance \( R_r = 0.7402 \Omega \). Other parameter settings are shown in Table 1.

| Mechanical transmission system parameters | Values |
|------------------------------------------|--------|
| Transmission inertia /(kg.m²)            | 0.342  |
| Speed difference of torque compensation control | 0.21  |
| Electromagnetic torque /KN.s             | 1200   |
| Driving wheel diameter of mechanical transmission system /m | 3.0    |
| Active wheel speed ratio                 | 12.3   |
| Rotational speed at both ends            | 0.28   |

Table 1. Main parameters of transmission control of mechanical transmission system.

![Figure 1. Dynamic load curve of high-speed end gear of mechanical transmission system.](image-url)
According to the above simulation environment and parameter setting, the anti-disturbance control of mechanical transmission system is carried out, and the dynamic load curve of high-speed end gear of mechanical transmission system is obtained as shown in figure 1.

5. Conclusion
In this paper, a disturbance rejection control method for mechanical transmission system is proposed based on torque compensation. The servo control modal parameters of mechanical transmission system are identified by fuzzy identification and inertial fusion. The inverse kinematics solution of the control structure of the mechanical transmission system is carried out by using the inverse kinematics decomposition method of the end effect, and the torque compensation and the optimization identification of the mechanical parameters of the mechanical transmission system are carried out by the modified DH parameter method. The dynamic load suppression problem is transformed into the torque parameter optimization problem of the mechanical transmission system. The motor torque is controlled to suppress the dynamic load of the mechanical transmission system. The Kalman filter is used to estimate the aerodynamic torque load of the system. The optimization of anti-disturbance control of mechanical transmission system is realized. The simulation results show that the proposed method has good stability and can effectively attenuate the dynamic load of the transmission system caused by the external abrupt load. This method has a good application value in the traditional mechanical system disturbance rejection control.

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