Modeling and optimization of B-axis hydraulic delay for gantry-type CNC machine tool

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Abstract. Due to the limitation of dimension, structure and material, aviation structural workpiece is usually processed by AB rotary axis five-axis CNC gantry-type machine tools with large torque. However, the B-axis mechanical structure of gantry-type machine tools is complex and bulky, and hydraulic system is adopted for gravity balance. After the long production period of the machine tool, the hydraulic balance force of the B-axis is lost immediately and the spindle goes out of control when it emergency stops, which seriously endangers the quality of workpiece. In view of this phenomenon, this paper takes the hydraulic control system of machine tool as the breakthrough point, and analyzes the hydraulic system without adjusting the mechanical structure. And the mathematical model of hydraulic system is established. The hydraulic control system of B-axis is optimized by the simulation of MATLAB/Simulink. On the premise of improving the stability of the system, the duration of hydraulic balance force is extended, and the deflection time of 2 seconds is delayed, which effectively reduces the processing risk of workpiece.

1. Introduction

CNC machining of aviation structural component is the most widely representative advanced manufacturing technology. With the rapid development of aviation science, the application proportion of large size complex aviation structural components composed of various new materials that are difficult to process are increasing in aviation equipments [1, 2]. Therefore the requirements on the overall performance of CNC machining equipment, including the size of the processing area, cutting torque, stiffness and others are becoming more stringent. The five-axis CNC gantry-type machine tools are ideal for processing such components due to their excellent space accessibility, large cutting force, outstanding whole stiffness and high machining accuracy [3-7]. The whole machining accuracy of the machine tool is the core to ensure the final quality of workpiece [8, 9]. As the service life of machine tool increases, the components are wearing out. The abnormal positioning phenomena of out-of-control deflection often occurs in rotary axis, which seriously endangers the quality of workpiece. Mechanical adjustment is the most direct way to solve the problem. However, this method needs to dismantle the B-axis mechanical structure wantonly and the maintenance cost is very expensive, so the hydraulic analysis method is a better method.

For hydraulic system optimization, Chen [10] proposed a new variable boundary layer sliding mode control method, comparatively analyzed the control performance of different control schemes through the mathematical modeling of the hydraulic system of the nonlinear electro-hydraulic position servo system. Qin-Man [11] used the SimMechanics toolbox and Simulink toolbox to establish the mechanical system simulation model and hydraulic system simulation model of the lifting mechanism of the
engineering vehicle, and realized the dynamic simulation analysis of the machine-liquid coupling in the same frame. Jin [12] established the mathematical model of the plunger hydraulic cylinder and the simulation model under the Simulink environment, and proposed the influence of the design parameters and medium characteristics of the hydraulic cylinder on its unit step response, and summarized the influence of design parameters and medium characteristics on its unit step response for plunger hydraulic cylinder, which provided a theoretical basis for optimizing the dynamic characteristics of hydraulic cylinders.

2. B-axis mechanical and hydraulic structure

A certain large five-axis CNC gantry-type machine tool is mainly consists of the body, worktable, column, beam and coordinate axis driven system. The overall mechanical structure is shown in Figure 1. The entire gantry column and beam of the machine tool runs on the X-axis rail. The beam carries the B-axis housing and the Y-axis slider. The end of the structure is the A-axis and the spindle.

![Figure 1. Mechanical structure](image)

When the B-axis rotates, it needs to drive the entire B-axis housing. Since the B-axis has relatively large self-weight, it is required to balance the gravity of the B-axis by a hydraulic device. Figure 2 is a schematic diagram of the B-axis hydraulic balance system. When the B-axis moves in the positive direction, the system pressure oil overflows the pressure reducing valve for pressure reduction, then flows through the proportional relief valve to adjust the pressure, and then enters the hydraulic cylinder in the rod cavity to push the piston rod to provide the balance force for the rotary axies. At the same time, the hydraulic oil in the rodless cavity of the hydraulic cylinder overflows back to the oil tank through the overflow valve; when the B-axis moves in the reverse direction, the system pressure oil is depressurized by the pressure relief valve. Then flows into the rodless cavity of the hydraulic cylinder through the one-way valve. Meanwhile, the hydraulic oil in the rod cavity of the hydraulic cylinder overflows back to the tank through the proportional relief valve.

![Figure 2. Schematic diagram of B-axis hydraulic balance system](image)
3. Mathematical modeling of hydraulic system
The spindle of CNC gantry-type machine tool is the motorized spindle. The spindle speed is high, which can up to maximum of 10000r/min. After the spindle stopped at high speed, the tool's speed will not stop immediately, but stop after a small while instead. If the spindle is not stopped, it will continue to cut workpiece under the condition of deflection, resulting in the phenomenon of overcutting. According to statistics data of the machine tool, when it works at the highest speed of 10000r/min, it takes 2.2s for the spindle to stop completely.

In order to reduce the damage accident of workpiece, the mathematical modeling and analysis of the control system of the B-axis the hydraulic system have been carried out. The hydraulic components parameters is optimized to improve the system dynamic performance and delay start time of the B-axis emergency-stop deflection until the cutting force of the spindle is insufficient or no cutting force.

3.1. Hydraulic Control System Mathematical Modeling
The mathematical equations describing the dynamic performance or static performance of hydraulic system are based on the three basic equations of the mechanical equilibrium equation, the flow equation of the fluid and the continuous flow equation.

3.1.1. Mechanical equilibrium equation of hydraulic counterbalance cylinder
According to Newton's second law, ignoring Coulomb friction and oil quality, the following can be obtained:

\[ A \Delta P_p = m \frac{d^2 x_p}{dt^2} + B_p \frac{dx_p}{dt} + K \Delta x_p + F_L \]  

Where, \( A \) is the effective area of the piston of the control cavity of the hydraulic counterbalance cylinder; \( \Delta P_p \) is the pressure difference between the rod cavity and the rodless cavity of the hydraulic counterbalance cylinder; \( m \) is the equivalent mass of the piston and the load converted to the piston; \( x_p \) is the displacement of the piston rod; \( B_p \) is the viscous damping coefficient of the piston rod and the load; \( K \) is the stiffness coefficient of the equivalent spring; \( F_L \) is the external force load of the hydraulic cylinder.

3.1.2. Flow equation of the oil in the pressure relief valve
Assuming that the valve port of the two-way pressure relief valve is a thin-walled small hole, the flow equation of the oil passing through the two-way pressure relief valve is:

\[ Q_v = K_v \frac{\partial x_v}{\partial t} = K_v \Delta P_v \]  

Where, \( K_v \) is the flow coefficient at the valve port. For thin-walled small holes, \( K_v \) is generally taken from 0.61~0.65; \( w \) is the flow area gradient of the valve port; \( x_v \) is the displacement of the spool; \( \rho \) is the liquid density, \( \Delta P_v \) is the pressure difference of the valve port, \( \Delta P_v = p_2 - p_1 \), \( p_1 \) and \( p_2 \) are the inlet pressure and outlet pressure of the pressure relief valve respectively; \( K_q \) is the flow gain at the valve port \( K_q = \frac{C_i w \rho}{2} \); \( K_v \) is the flow-pressure coefficient \( K_v = \frac{C_i w \rho}{2} \sqrt{\frac{2}{\rho \frac{\partial P_v}{\partial P_v}}} \).

3.1.3. Flow continuity equation of the hydraulic counterbalance cylinder
Since the flow rate of the oil fluid continuously flowing through the two-way pressure relief valve is equal to the oil flow rate in the hydraulic counterbalance cylinder, according to the flow conservation principle, the fluid continuity equation can be obtained:

\[ \dot{Q}_c = A \frac{dx_p}{dt} + V_t \frac{d\Delta P_p}{dt} + C_i \Delta P_p \]  

Where, \( V_t \) is the total working volume of the oil passage; \( \beta_k \) is the hydraulic oil elastic modulus; \( C_{ip} \) is the total leakage coefficient of the hydraulic counterbalance cylinder, \( C_{ip} = C_{ip} + C_{ep} / 2 \); \( C_{ip} \) is the internal leakage coefficient of the hydraulic counterbalance cylinder, and \( C_{ep} \) is the external leakage coefficient of the hydraulic counterbalance cylinder.
3.1.4. Transfer function of hydraulic system

After the performance of Laplace transform and linearization of (1) ~ (3), the transfer function of the hydraulic cylinder output displacement and the two-way pressure relief valve spool displacement can be obtained:

\[
x_p = \frac{K_e \omega_n^2 (s + \frac{1}{\xi \omega_n})(1 + \frac{V_{in}}{4 \beta \omega_n s})F_p}{(s + \frac{K_e \omega_n^2}{\omega_n^2})(s + \frac{1}{\xi \omega_n})(s + \frac{2 \xi \omega_n}{\omega_n} + 1)}
\]

Where, the natural frequency of the two-way pressure relief valve \(\omega_n = \sqrt{\frac{4 \beta \omega_n^2}{m_f}}\); the hydraulic damping ratio \(\xi = \frac{4 \beta \omega_n^2}{m_f} \omega_n\); the total flow rate - pressure coefficient \(K_{ce} = K_e + C_p\).

3.2. System Simulation Model Establishment and Test Analysis

Establish the Simulink system simulation model according to the above analysis, and initialize each parameter in the model as shown in Table. 1, as shown in Figure. 3.

![Simulink model](image)

Figure. 3 B-axis Simulink system simulation model

Input the above parameters into the simulation model, and use the ode45 explicit Runge-Kutta (4, 5) adaptive step size solver [13]. The fourth-order method is used to provide the candidate solution, the fifth-order method is used to simulate the control error, and the influence of the parameter change on its dynamic characteristics is investigated, which provides the basis for the optimal design of the hydraulic system.

3.2.1. Damping coefficient \(\zeta_h\)

The damping coefficient \(\zeta_h\) of the hydraulic system characterizes the relative stability of the system. For satisfactory performance, \(\zeta_h\) should have appropriate values. The general hydraulic servo system is underdamped [14]. Therefore, the damping coefficient \(\zeta_h\) is taken as 0.7, 0.5 and 0.4 as the simulation test parameters. The dynamic characteristic diagram of the displacement step response of the B-axis of the hydraulic counterbalance cylinder is shown in Figure.4. The smaller the damping coefficient \(\zeta_h\), the slower the amplitude attenuation of the curve, the larger the oscillation frequency, and the longer the steady-state response time. It can be seen from the figure that the damping coefficient \(\zeta_h\) has a great influence on the oscillation performance of the system. To ensure sufficient stability of the system, it is necessary to select a larger \(\zeta_h\). It can be known from the above formula that \(\zeta_h\) can be increased by appropriately increasing \(K_{ce}\) and \(B_p\).

| Table. 1 System simulation parameters | parameter | value |
|-------------------------------------|----------|-------|
| \(K_e (m^2 \cdot s^{-1})\)         | 0.361    |
| \(A (m^2)\)                        | 4.5 \times 10^{-5} |
| \(K_{ce} (m^2 \cdot s^{-1})\)     | 0.4 \times 10^{-3} |
| \(C_p (m^2 \cdot s^{-1} \cdot Pa^{-1})\) | 6.7 \times 10^{-11} |
| \(B_p (N \cdot m^{-1} \cdot s^{-1})\) | 1.2 \times 10^{7} |
| \(m(kg)\)                          | 300      |
| \(\beta (Pa)\)                     | 7 \times 10^{8} |
| \(F_t(kg)\)                        | 4600     |
| \(V_{in}(m^3)\)                   | 0.0294   |
3.2.2. Natural frequency $\omega_h$

When the natural frequency $\omega_h$ is taken as 100, 150, and 200, the system step response curves are as shown in Figure 5 respectively. As the dynamic response speed is increased, the rise time, peak time, and adjustment time are reduced. Therefore, to delay the deflection of the B-axis and reduce the response speed of the system, it is necessary to appropriately reduce $\omega_h$, however, to reduce the rise time, the peak time, and the adjustment time, $\omega_h$ cannot be too small. According to the above calculation formula of $\omega_h$, $\omega_h$ can be reduced through reducing $A$ or increasing $V_t$.

3.2.3. Flow gain $K_q$

When the sliding valve flow gain $K_q$ is taken as 0.1, 0.3, and 0.5 respectively, the step dynamic response of the system is as shown in Figure 6. It can be clearly seen that as the sliding valve flow gain $K_q$ increases, the oscillation amplitude of the system increases, and the adjustment time to reach the steady state increases. When the spool is at zero position, the sliding valve flow gain $K_q$ is the largest, which means the system stability is the worst; when there is load, the $K_q$ is reduced, the response speed of the system is reduced, and the adjustment time is increased. Therefore, the appropriate type of valve port can be used to reduce the flow area gradient the valve port, or connect the throttle in tandem to suppress the flow rate, to reduce the value of $K_q$ to delay the corresponding speed of the counterbalance cylinder.

3.2.4. Optimization results

Through the above simulation test analysis, in order to delay the deflection time of the B-axis to ensure certain system stability, the components of the hydraulic system of the B-axis can be selected according to the configuration of the damping parameter $\zeta_h=0.5$, the natural frequency $\omega_h=150$, and the flow gain $K_q=0.3$, and the dynamic response of the system step before and after optimization is shown in Figure 7. It can be seen from the figure that the oscillating performance of the system is weakened and the stability is improved after optimization; the adjustment time increased from $t_s$ to $t'_s$. The deflection time of the B-axis before and after optimization is shown in Figure 8. As can be seen, after optimization, each angle has an average delay of $\Delta \approx 2s$. 
4. Conclusion
To solve the problem of the emergency-stop deflection phenomenon of the critical rotary axis of the large five-axis CNC gantry-type machine tool and improve the precision performance of the machine tool, the research work has been carried out from the aspect of hydraulic system in this paper. In order to reduce the processing risk of workpiece, the mathematical model of the hydraulic control system is established combining with the structure of hydraulic balance system of the B-axis. In addition, the simulation analysis on the three parameters $\zeta_h$, $\omega_h$, $K_q$ that are affecting its dynamic characteristics are carried out. Optimizing the system parameter $\zeta_h$, $\omega_h$ and $K_q$ are to 0.5, 150 and 0.3 respectively. The oscillation characteristic of the system is stabilized, and the start time of deflection is delayed 2s in the whole range of B-axis rotate angle.

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