Dynamic Characteristics Analysis of Cover System

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Abstract: Up to now, although a lot of research has been done on the control algorithm of cover motion, no systematic analysis has been carried out on the trajectory planning of cover opening yet. In order to improve the dynamic characteristics of the cover in the process of cover opening, a velocity planning algorithm based on trigonometric function is proposed in this paper, which improves the poor smoothness of the acceleration curve of the traditional S-shaped feedrate planning algorithm. The simulation model of the cover system is established to verify the effectiveness of the algorithm, and the results show that the algorithm is effective.

1. Introduction

The cover system is an electro-hydraulic servo system with large inertia. Due to the large load inertia and poor dynamic performance of the system, the uneven acceleration and deceleration in the process of cover opening will produce impact, which may further affect the control effect of opening the cover. In order to ensure the smoothness of the cover opening process, besides improving the control algorithm, we need a reasonable velocity planning to reduce the impact during the cover opening process[1]. At present, the research on the cover opening process mainly focuses on the control algorithm, but it is necessary to study the trajectory of cover opening in order to achieve smooth and highly reliable cover opening.

The motion process of large inertia servo system is generally divided into three stages: acceleration, uniform velocity and deceleration. When planning the motion trajectory, we usually focus on the research design of the stages of acceleration and deceleration. Commonly used acceleration and deceleration methods include trapezoidal acceleration and deceleration, exponential acceleration and deceleration, S-shaped acceleration and deceleration, etc[2]. Trapezoidal acceleration and deceleration is a curve based on time optimization. With simple algorithm, it is widely used, but there is sudden change of velocity in it[3]. S-shaped acceleration and deceleration is another widely used velocity curve. The velocity changes smoothly, and both velocity and acceleration have continuity, but the acceleration curve is not smooth[4].

This study has firstly modeled the cover system, and established the simulation model of the cover system. Then, it has discussed the design idea of velocity planning. Based on the commonly used velocity planning algorithm, a trigonometric velocity planning algorithm for cover opening is proposed, and the effectiveness of the algorithm is analyzed by simulation.

2. Cover system model

The cover system is composed of hydraulic units, transmission mechanism and cover, and the structural diagram is shown in Figure 1. One end of the push rod of the hydraulic cylinder is connected to the transmission mechanism of the cover system. When the cover is opened, the piston of the
hydraulic cylinder moves upward to push the crank-slider mechanism to move. The crank-slider mechanism converts the linear movement of the hydraulic cylinder into the rotation of the cover and pushes the cover to rotate around the axis.

2.1. Kinematic analysis of the cover system
As shown in Figure 1, it can be considered that the cover rotates around point O2, and a coordinate system with O2 as the origin and the vertical direction as the x axis is established. Point O1 is the particle of the cover, point O3 and O4 are the kinematic pair of crank-slider mechanism, and point O5 is the projection of point O4 on the x axis. The distance between O2 and O3 is $l_1$, the distance between O3 and O4 is $l_2$, the distance between O2 and O5 is $SC$, and the distance between O3 and O4 is $e$. $\angle O_5O_2O_3$ is $\alpha_1$. The angle between the vertical direction through point O3 and O3O4 is $\alpha_2$.

From the vector method:

$$C_S = l_1 \cos \alpha_1 + l_2 \cos \alpha_2 = S_C$$  \hspace{1cm} (1)
$$C_e = l_1 \sin \alpha_1 + l_2 \sin \alpha_2 = e$$  \hspace{1cm} (2)

By getting rid of $\alpha_2$ through equations (2) and (3):

$$l_1 \cos \alpha_1 - S_C)^2 + (l_1 \sin \alpha_1 - e)^2 = l_2^2$$  \hspace{1cm} (4)

By solving the quadratic equation with one unknown:

$$S_C = l_1 \cos \alpha_1 + \sqrt{l_1^2 - (e - l_1 \sin \alpha_1)^2}$$  \hspace{1cm} (5)
$$x = L - S_C$$  \hspace{1cm} (6)

Where X is the displacement of the piston motion of the hydraulic cylinder, and L is the length of SC when the opening angle of the cover is 0 degree.

The displacement and rotation angle of hydraulic cylinder can be seen from Figure (1):

$$\alpha_1 = \arccos \left( \frac{S_C^2 + e^2 + l_1^2 - l_2^2}{2l_1 \sqrt{S_C^2 + e^2}} \right) + \arctan \left( \frac{e}{S_C} \right)$$  \hspace{1cm} (7)
$$\theta = \alpha_1 - \alpha$$  \hspace{1cm} (8)

Where $\theta$ is the opening angle of the cylinder cover, and $\alpha$ is the degree of $\alpha_1$ when the opening angle of the cover is 0 degree.
2.2. Modeling of power mechanism of the cover system

Let the angle between O1O2 and the y axis be γ, the cover be equivalent to the center of mass O1, the moment of inertia of the cover be I, the angular acceleration of the cover be A, the gravity of the cover be G, \( \angle O2O3O4 \) be \( \rho \), and the resolution of force of load force F of hydraulic cylinder along O3O4 be \( F_L \). From Newton’s second law of motion:

\[
F_L = \frac{JA + GR \cos(\theta + \gamma)}{\sin \alpha_3} \\
\cos \alpha_3 = \frac{l_1^2 + l_2^2 - (S^2_c + e^2)}{2l_1l_2} \\
F = -\frac{F_L}{\cos(\alpha_1 + \alpha_3)}
\]

2.2.1 Flow equation of slide valve

The electro-hydraulic servo system provides power for the cover system. The control valve block of the electro-hydraulic servo system is a zero-opening four-way slide valve, and the hydraulic cylinder is a double-rod symmetrical hydraulic cylinder. In order to simplify the model, the following assumptions are made: (a) the hydraulic power source is an ideal constant voltage source, the charge oil pressure is constant, and the return oil pressure is assumed to be zero; (b) ignore the pressure loss in pipes and valve chambers; (c) assume that the flow coefficient of each throttle of the valve is the same. Load flow \( Q_L \) can be expressed as:

\[
Q_L = C_d \omega \left( \frac{1}{\rho} (P_S - x_v P_L) \right)
\]

Where \( C_d \) is the flow coefficient of throttle, \( \omega \) is the throttling area gradient of servo valve, \( x_v \) is the valve core displacement, \( \rho \) is the density of hydraulic oil, \( P_S \) is the charge oil pressure, and \( P_L \) is the load pressure.

2.2.2 Flow continuity equation of hydraulic cylinder

Ignoring pipeline loss, pipe effect, uniform pressure distribution, constant oil temperature and bulk elastic modulus, and laminar flow. Consider leakage and compressibility, and define the load flow as:

\[
Q_L = \frac{(Q_1 + Q_2)}{2}
\]

Where the flow capacity of the oil inlet chamber \( Q_1 \) is:

\[
Q_1 = A_p x' + C_{ip} (P_1 - P_2) + C_{ep} P_1 + \frac{V_1 P_1'}{\beta_e}
\]

Flow capacity of oil return chamber \( Q_2 \) is:

\[
Q_2 = A_p x' + C_{ip} (P_1 - P_2) - C_{ep} P_2 - \frac{V_2 P_2'}{\beta_e}
\]

\( A_p \) is the effective area of hydraulic cylinder piston, \( x \) is piston displacement, \( C_{ip} \) is the internal leakage coefficient in hydraulic cylinder, \( C_{ep} \) is the external leakage coefficient of hydraulic cylinder, \( \beta_e \) is the effective bulk elastic modulus of hydraulic oil, \( V_1 \) is the volume of the oil inlet chamber of the hydraulic cylinder, and \( V_2 \) is the volume of the oil return chamber of the hydraulic cylinder.

According to:

\[
V_1 = V_{10} + A_p x \\
V_2 = V_{20} - A_p x \\
V_{10} = V_{20} = V_0 = \frac{V}{2} \\
A_p x << V_0
\]

\( Q_L \) can be obtained:
\[ Q_L = A_p \dot{x} + C_{pL} P_L + \frac{V_t P_L}{4 \beta_e} \]  
(17)

Where \( C_{pL} \) is the total leakage coefficient:
\[ C_{pL} = C_p + \frac{C_{op}}{2} \]  
(18)

2.2.3 Balanced force equation of hydraulic system
The balanced force equation of hydraulic cylinder and load can be expressed as:
\[ A_p P_L = m \ddot{x} + B_p \dot{x} + F \]  
(19)

In this equation, \( m \) is the piston mass of hydraulic cylinder, \( B_p \) is the viscous damping coefficient, and \( F \) is the external load force of hydraulic cylinder.

3. Trajectory planning of cover opening
Displacement or velocity control is adopted in the opening of the cover, so the continuity of displacement and velocity should be guaranteed first when planning the opening trajectory of the cover. Theoretically speaking, in order to reduce the impact on the system, we should try our best to ensure that there is no sudden change in velocity and acceleration when planning the trajectory.

In this paper, three acceleration and deceleration methods including trapezoidal velocity model, S-curve velocity model and trigonometric velocity model are used for velocity planning. In order to measure the performance difference between various planning programs, in the study, the opening angle is 115°, the opening duration is 5 seconds, the acceleration duration is 2 seconds, the uniform velocity duration is 1 second, and the deceleration duration is 2 seconds.

3.1. Trapezoidal velocity model
Trapezoidal velocity planning is a classical velocity planning method. In trapezoidal velocity planning, the acceleration in stages of acceleration and deceleration is constant, and the stage of uniform velocity is in the middle part, so the expression of angular velocity of cover opening using trapezoidal acceleration and deceleration planning is as follows:
\[
\omega = \begin{cases} 
\frac{a t}{2} & 0 \leq t < 2 \\
-a & 2 \leq t < 3 \\
\frac{-a t}{2} + \frac{5a}{2} & 3 \leq t \leq 5 
\end{cases} \]  
(20)

Where \( a \) is the maximum angular velocity of trapezoidal velocity planning:
\[ a = \frac{115}{\int_0^2 \frac{t}{2} dt + a + \int_3^5 \frac{t+5}{2} dt} \]  
(21)

The position, velocity and acceleration curve of trapezoidal velocity model are shown in Figure 2.
3.2. *S-curve velocity model*

From the acceleration curve of trapezoidal velocity model, it can be seen that there is a sudden change of acceleration. Discontinuity of acceleration often leads to impact and induces residual vibration.

*S-curve velocity model* is another widely used trajectory planning model, which is characterized by continuity of position, velocity and acceleration curve. *S-curve velocity planning* covers such stages as acceleration stage, constant acceleration stage, reduced acceleration stage, uniform velocity stage, increased deceleration stage, constant deceleration stage and reduced deceleration stage. Since the jerk values of all stages are constant, when the duration and total distance of each stage are determined, the trajectory planning curve can be obtained by integrating acceleration with time.

In this paper, the *S-curve velocity planning* without constant acceleration and constant deceleration stages is used to plan the cover opening trajectory curve. Accordingly, the expression of the opening angular acceleration planned by *S-curve velocity model* is as follows:

\[
\alpha = \begin{cases} 
bt & 0 \leq t < 1 \\
-bt+2b & 1 \leq t < 2 \\
0 & 2 \leq t < 3 \\
-at+3a & 3 \leq t < 4 \\
at-5a & 4 \leq t \leq 5 
\end{cases}
\]  

(22)

By integrating the angular acceleration, the angular acceleration can be obtained as follows:

\[
\omega = \begin{cases} 
\frac{bt^2}{2} & 0 \leq t < 1 \\
-\frac{bt^2}{2}+2bt-b & 1 \leq t < 2 \\
b & 2 \leq t < 3 \\
-\frac{bt^2}{2}+3bt-\frac{7b}{2} & 3 \leq t < 4 \\
\frac{at^2}{2}-5at+\frac{25a}{2} & 4 \leq t \leq 5 
\end{cases}
\]  

(23)

Where \( b \) is the maximum angular velocity of *S-curve velocity planning*:

\[
b = \frac{115}{\int_0^1 \frac{t^2}{2} \, dt + \int_1^2 \frac{t^2}{2} + 2t-1 \, dt + \int_2^3 \frac{t^2}{2} + 3t-\frac{7}{2} \, dt + \int_3^4 \frac{t^2}{2} - 5t+\frac{25}{2} \, dt}
\]  

(24)

The position, velocity and acceleration curve of *S-curve velocity model* are shown in Figure 3.
3.3. **Trigonometric velocity model**

It can be seen from Figure 3 that in the S-curve velocity model, the velocity is continuous and smooth. Although the acceleration changes continuously without sudden change, the smoothness is not good, which indicates that the jerk curve has sudden change, and the discontinuous jerk curve will cause the system to be impacted and produce vibration.

Because the trigonometric function is continuous and differentiable and there are two points with derivative of zero in a cycle. The trigonometric function can be used to plan angular acceleration. In the acceleration stage and deceleration stage, the acceleration curve is planned by a complete cycle between two adjacent minimum points of sine function. The acceleration curve can be determined by determining the duration and total distance of acceleration stage, uniform velocity stage and deceleration stage, and the velocity curve can be obtained by integrating the acceleration curve. Therefore, the opening angular acceleration expression of trigonometric acceleration and deceleration algorithm is:

\[
\alpha = \begin{cases} 
-\frac{c}{2} \cos(\pi t) + \frac{c}{2} & 0 \leq t < 2 \\
0 & 2 \leq t < 3 \\
-\frac{c}{2} \cos(\pi t) - \frac{c}{2} & 3 \leq t \leq 5 
\end{cases}
\] (25)

By integrating the angular acceleration, the angular acceleration can be obtained as follows:

\[
\omega = \begin{cases} 
\frac{c}{2} t - \frac{c \sin(\pi t)}{2\pi} & 0 \leq t < 2 \\
c & 2 \leq t < 3 \\
-\frac{c}{2} t - \frac{c \sin(\pi t)}{2\pi} + \frac{5c}{2} & 3 \leq t \leq 5 
\end{cases}
\] (26)

Where c is the maximum angular acceleration of trigonometric velocity planning:

\[
c = \frac{115}{\int_0^2 \frac{\sin(\pi t)}{2\pi} dt + c + \int_2^5 \frac{t - \frac{\sin(\pi t)}{2\pi} + \frac{5c}{2} dt}{2}}
\] (27)

The position, velocity and acceleration curve of trigonometric velocity model are shown in Figure 4.
4. Simulation and Analysis

In order to compare the performance of the above-mentioned three trajectory planning methods, the simulation model of the cover system is established. Then the external load force curves of the hydraulic cylinder and the tracking error curves based on three cover opening trajectories are obtained. Please see Table 1 for parameters of cover system simulation model.

![Figure 4. Trigonometric velocity model](image)

| Parameter                              | Symbol | Value   | Measure |
|----------------------------------------|--------|---------|---------|
| Oil effective bulk modulus            | $\beta_e$ | $1 \times 10^9$ | N/m²   |
| Viscous damping coefficient           | $B_P$  | 7500    | N·s/m   |
| Effective action area of hydraulic cylinder | $A_P$  | 0.0019  | m²      |
| Piston mass                           | $m$    | 30      | Kg      |
| Volume of hydraulic cylinder          | $V_t$  | $0.38 \times 10^{-3}$ | m³     |
| Leakage coefficient of hydraulic cylinder | $C_{tp}$ | $9.2 \times 10^{-13}$ | m³/s·Pa |
| The length of O₁O₂                   | $R$    | 0.72    | m       |
| The length of O₂O₃                   | $l_1$  | 0.3     | m       |
| The length of O₃O₄                   | $l_2$  | 1.8     | m       |
| The length of O₄O₅                   | $e$    | 0.2     | m       |
| Quality of covering device           | $G$    | $2.3 \times 10^4$   | N      |

From the balanced force equation of hydraulic system, it can be seen that when the external load force is large, its change has great impact on the motion control of hydraulic cylinder. Through the dynamic equation of the cover system, the change curves of the external load force $F$ of the hydraulic system corresponding to different trajectories can be obtained, as shown in Figure 5. It can be seen from the figure that there are sudden changes in external load force curves of hydraulic cylinder in trapezoidal acceleration and deceleration algorithm, which will cause shaking and hydraulic impact in the process of opening the cover. The external load force curves of hydraulic cylinder in S-curve velocity model are continuous but the external load force of hydraulic cylinder in S-curve velocity model is not smooth when the time is 1s and 2s, which will cause the system to be impacted and produce vibration. The external load force curves of hydraulic cylinder in trigonometric velocity model are continuous and smooth, so the trigonometric velocity model has better performance.
In order to verify the final control effect of the three trajectory planning methods, position following control is adopted, and the position following error is shown in Figure 6. It can be concluded from Figure 4 that the position following error of the trigonometric velocity model is smaller and thus better control effect can be achieved.

5. Conclusions
In this paper, a trigonometric model is proposed for trajectory planning in the process of cover opening, and a simulation model of the cover system is established. The effectiveness of the trigonometric velocity model is evaluated by comparing the control effects of the trigonometric velocity model, with that of the commonly used trapezoidal velocity model and the trigonometric velocity model.

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