Study on the Stability of an Electronically Controlled Hydraulic Drive Vehicle Based on Sliding-mode Control and Fuzzy Control Algorithm

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Abstract. In order to improve the stability of an electronically controlled hydraulic drive four-wheel drive vehicle, a two-degree-of-freedom and seven-degree-of-freedom vehicle mathematical model was established and analyzed. The entire vehicle was obtained through the combination of sliding-mode control control and fuzzy algorithm Control, according to the dynamic force of the wheels, the torque distribution between the different wheels of the vehicle is adjusted. The stability control strategy of the electronically controlled hydraulic drive vehicle is verified through the joint simulation of MATLAB/simulink and Carsim. Analysis of working conditions shows the effectiveness of the proposed stability.

1. Introduction
In recent years, with the continuous development of control algorithms and technologies, more and more researches have been made on four-wheel independent driving vehicles[1-3].The Four Wheel Independent Steering/Driving (4WIS/4WID) vehicle has the advantage that the rotation angle and driving torque of each wheel can be independently and accurately controlled, it is conducive to the stability control and precise control of vehicles[4-6].Both hierarchical and centralized control structures are used to control vehicle stability. In general, hierarchical control structures have proven to be more flexible and effective in respect to centralized control structures[7].In this paper, the stability of the electronically controlled hydraulic drive vehicle is studied, combined the sliding mode control algorithm and the fuzzy control algorithm for stability design, and verified by joint simulation.

2. Theoretical analysis
2.1. Two-degree-of-freedom vehicle model
Through the establishment of a two-degree-of-freedom vehicle model, the overall state of the vehicle is estimated and used as a comparative analysis. Both experiments and analyses have proved the accuracy of the two-degree-of-freedom model. The 2DOF vehicle model is widely used as a reference for the design of automotive controller model [8-10].
From the analysis of Figure 1, it can be seen that the resultant force along the y-axis direction of the external force received by the two-degree-of-freedom vehicle and the moment around the center of mass are:

\[ \sum F_y = F_{y1} \cos \delta + F_{y2} \]  
\[ \sum M_z = a F_{y1} \cos \delta - b F_{y2} \]

The acceleration \( a_x \) and \( a_y \) of the vehicle along the x-axis and y-axis are respectively:

\[ a_x = \ddot{u} - v \dot{w}_r \]  
\[ a_y = \dot{v} + u \dot{w}_r \]

\[ F_{y1} = k_1\alpha_1 \]  
\[ F_{y2} = k_2\alpha_2 \]

According to the coordinate regulations, the slip angle \( \alpha_1 \), \( \alpha_2 \) of the front and rear wheels are:

\[ \alpha_1 = \beta + \frac{aw_r}{u} - \delta \]  
\[ \alpha_2 = \beta - \frac{bw_r}{u} \]

Combining the above equations to obtain the differential equation of two-degree-of-freedom vehicle motion is:

\[ (k_1 + k_2)\beta + \frac{1}{u}(ak_1 - bk_2)w_r - k_1\delta = m(\dot{v} + u\dot{w}_r) \]  
\[ (ak_1 - bk_2)\beta + \frac{1}{u}(a^2 k_1 + b^2 k_2)w_r - ak_1\delta = I_\gamma \dot{w}_r \]

\( F_{y1} \) and \( F_{y2} \) are the lateral reaction forces of the ground against the front and rear wheels, that is, the cornering force; \( \delta \) is the front wheel angle; \( a \) and \( b \) are the distance between the center of mass and the center of the front and rear wheels; \( u, v \) are the center of mass of the vehicle The speed along the x and
y axis; $k_1$ and $k_2$ are the cornering stiffness of the front and rear wheels; $w_r$ is the yaw rate; $I_z$ is the moment of inertia of the vehicle around the z axis.

The steady-state yaw rate is:

$$w_{r1} = \frac{u}{L} \frac{1}{1 + Ku} \delta$$

(11)

$$K = \frac{m}{L^2} \left( \frac{a}{k_1} - \frac{b}{k_2} \right)$$

(12)

$w_{r1}$ is the steady-state yaw rate of the vehicle; $L$ is the front and rear track of the vehicle.

2.2. Seven-degree-of-freedom nonlinear vehicle model

![Vehicle model with seven degrees of freedom](image)

According to Figure 2, the force analysis of the vehicle is as follows:

$$m \ddot{x} = (F_{sfr} + F_{sfr}) \cos \delta + (F_{yfr} + F_{yfr}) \sin \delta + F_{xrl} + F_{yrl}$$

(13)

$$m \ddot{y} = (F_{sfr} + F_{sfr}) \sin \delta + (F_{yfr} + F_{yfr}) \cos \delta + F_{xry} + F_{yry}$$

(14)

$$I_z \ddot{\omega} = ((F_{xrl} + F_{yrl}) \cos \delta - F_{yfr} \sin \delta) - (F_{xry} + F_{yry}) \cos \delta - F_{yfr} \sin \delta) \frac{d}{2}$$

$$+ a(F_{yfr} \sin \delta + F_{yfr} \cos \delta + F_{yfr} \sin \delta + F_{yfr} \cos \delta) - b(F_{yrl} + F_{yry})$$

(15)

3. Experiments and analysis

3.1. Design of sliding mode control algorithm

Set the yaw angle of the vehicle as $\theta$ and consider the uncertain system:

$$\ddot{\theta} = f(\theta, \dot{\theta}) + b(u(t) + E(t))$$

(16)

Among them, $f(\theta)$ is known, $b > 0$, $E(t)$ is interference.

The design synovial function is:

$$s = e + c \dot{e}$$

(17)
$e$ is the tracking error, $e = \int_{0}^{t} w_r - w_A dt$ where $w_r$ is the actual yaw rate, $w_A$ is the ideal yaw rate.

The design synovial controller is:

$$u = \frac{1}{b} (f(\theta) + c\dot{e} - \dot{\Theta} - K(t) \text{sgn}(s))$$  \hspace{1cm} (18)

$$K(t) = \max\{|E(t)| + n(n > 0)\}$$  \hspace{1cm} (19)

Take the Lyapunov function as:

$$V = \frac{1}{2} s^2$$  \hspace{1cm} (20)

$$\dot{V} = s\ddot{s} = s(\ddot{e} + c\dot{e}) = s(-K(t)\text{sgn}(s) - E(t)) = \dot{E}(t)|s| - E(t)s \leq -n|s|$$  \hspace{1cm} (21)

It can be concluded that the system is stable. In the SMC, the switching gain $K(t)$ is the main cause of system chattering. Due to the complex changes of electronically controlled hydraulic vehicle parameters and the delay and error in sensor measurement, to improve the robustness of the stability control of the vehicle is very important.

### 3.2. Design of fuzzy control algorithm

Combining the vehicle’s yaw rate deviation and the vehicle’s slip rate $s$, design a fuzzy control algorithm. Fuzzy control reduces the control when the vehicle is in a stable and relatively stable state, and more accurately control the unstable vehicle. The vehicle control rules are shown in Table 1.

| $s$  | LL | L  | M  | H  | HH |
|------|----|----|----|----|----|
| LL   | LL | L  | ML | MH | MH |
| L    | L  | L  | ML | MH | H  |
| M    | ML | ML | MH | H  | H  |
| H    | MH | MH | H  | H  | HH |
| HH   | MH | H  | H  | HH | HH |

The output weighting coefficient $Z$ is shown in Figure 3.

![Figure 3. Weighting coefficient under fuzzy algorithm](image-url)
3.3. Torque distribution of each wheel

According to the difference in the reaction force of the ground to the wheels, in order to make full use of the ground adhesion, an additional yaw moment is applied to each wheel through a distribution algorithm to improve the stability of the vehicle. The calculation formula is as follows:

\[ \xi_i = \frac{F_z}{F_\mu} \quad (i = fl, fr, rl, rr) \]  

\[ \Delta T = \frac{T_\mu}{r} \cos \delta \frac{d}{2} + \frac{T_\mu}{r} \sin \delta a + \frac{T_{fr}}{r} \cos \delta \frac{d}{2} + \frac{T_{fr}}{r} \sin \delta a + T_{vl} \frac{d}{2} + T_{vl} \frac{d}{2} \]  

\[ T_i = \xi_i T \quad (i = fl, fr, rl, rr) \]  

\[ -T_{b_{max}} \leq T_{ai} + T_i \leq T_{d_{max}} \]

\( F_{zi} \) is dynamic vertical force of the wheel, \( \xi_i \) is the additional torque applied coefficient of the wheel; \( T_i \) is the calculated applied torque of the wheel; \( T_{ai} \) is the additional torque of the wheel; \( T_{b_{max}} \) is the maximum torque when the wheel is braking; \( T_{d_{max}} \) is the maximum torque when the wheel is driving.

4. Simulation and analysis

A joint simulation model is established by MATLAB/simulink and carsim. In the joint simulation, the stability of the vehicle is analyzed by double-shifting line surface simulation to verify the effectiveness of the proposed strategy. Set the adhesion coefficient of the test road to 0.3 and the target speed of the vehicle to 80km/h to obtain the test results under different control modes.

![Figure 4: Variation curve of vehicle longitudinal displacement](image)

It can be seen from Figure 4 that in the case of no control of the vehicle, the maximum offset is -1.5m at 8.4s, and at 7-12s, the vehicle has been in a state of deviating from the target trajectory; In the simulation test of dual-line shifting, the longitudinal displacement trajectory of the electronically controlled hydraulic drive vehicle under fuzzy control and PID control is closer to the target path of the vehicle, and the vehicle motion trajectory under fuzzy control is closest to the target path.
Figure 5. Variation curve of yaw rate

It can be seen from the change curve of the yaw rate in Figure 5 that the yaw rate changes of the three control modes are not much different at 0~6s. The yaw rate of the vehicle without control undergoes a sudden change after 6s, reaching 24 °/s, and maintained at around 20°/s, indicating that the vehicle is under poor stability control. The vehicle yaw rate under fuzzy control and PID control showed a decreasing trend after 7s, indicating that the vehicle was in a stable state under its stability control strategy. The overall value of the yaw rate change curve under the fuzzy control is lower than the yaw rate under the PID control, indicating that the fuzzy control presents better stability control.

Figure 6. Variation curve of center of mass slip angle

It can be seen from the curve of mass center slip angle in Figure 6 that the vehicle loses stability without control, while the vehicle is in a stable state under control. Under PID control, the vehicle’s mass center slip angle is 5.6°, it reaches the maximum value of 4.5° at 5.3s, and the side slip angle of the center of mass under the fuzzy control strategy reaches a peak value of 2.7° at 5.4s, which is nearly half of that under PID control, indicating that the vehicle under fuzzy control has good heading following performance.

5. Conclusion
Aiming at the stability of electronically controlled hydraulic drive vehicles, this paper proposes a stability control strategy for electronically controlled hydraulic drive vehicles that combines a sliding film control algorithm and a fuzzy control algorithm. The vehicle double-shifting low-adhesion road surface is implemented through MATLAB/simulink and carsim. Simulation experiments on working conditions show that the proposed stability control strategy can effectively improve the handling stability of the vehicle, bring the vehicle closer to the target path, reduce the peak value of yaw rate...
and the peak value of side slip angle of the vehicle, and greatly reduce the slip rate of the vehicle verifies the effectiveness of the control strategy on the stability of the vehicle.

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