Research Article

Lateral Stability Control of 4WD Vehicle considering Ride Performance

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In order to improve the lateral stability of 4WD vehicle and guarantee its ride performance, this study designs target yaw control system based on sliding mode variable structure control, proposes driving torque distribution control algorithm based on the response of vehicle stable steering characteristic, and designs semiactive suspension control system based on double-fuzzy controller. The control performance is analyzed during the step steering and the split-μ road turning. The results indicate that the combined control system is able to improve the lateral stability of this 4WD vehicle during turning, reduce its body vibration acceleration and suspension dynamic deflection during travelling on bumpy and uneven road, guarantee its dynamic property, and improve its comprehensive performance.

1. Introduction

Four-wheel drive (4WD) vehicle in terms of power performance is excellent, because it can take full advantage of road adhesion. Generally, the 4WD vehicles are equipped with open differential on driving axle and transfer case or central differential between axles. This limits their comprehensive performance when they travel on the low adhesion road. In recent years, a great number of studies focus on power performance and stability of 4WD vehicles. They mostly concentrated on researching traction control system (TCS) of 4WD vehicle by combining engine torque control with interwheel limited slip differential (LSD) control or central LSD control. In [1], a hierarchical controller was designed to improve the acceleration performance while ensuring the vehicle stability with braking or driving torque distribution control and engine torque control. Song and Boo investigated the dynamic response of front wheel drive (FWD) and 4WD vehicle with brake or TCS which combines the brake pressure control with engine torque control [2]. In order to improve the vehicle’s traction and fuel economy, a traction force and wheel slip control algorithm for a 4WD vehicle was presented in [3]. Gao et al. proposed a traction force control strategy integrating engine throttle control, driving wheel brake control, and interaxle torque distribution control [4]. Yang et al. designed an optimum driving torque controller to improve the acceleration ability when the vehicles travel on slippery road with engine PID control [5].

At present, people require that 4WD vehicles should have good power performance, stability, ride performance, and ride comfort. Road conditions have great influence on the ride performance and the ride comfort. Current researches on 4WD vehicles mostly ignore this influence, so the improvement of power performance and stability are limited. Studies show that interwheel torque distribution control responds more rapidly than central LSD torque distribution control. Moreover, not only can the interwheel torque distribution device adjust the wheel speed, but can also distribute the torque between both sides of driving wheels actively. Compared with current electronic stability control (ESC), the interwheel torque distribution control technology can improve the vehicle lateral stability without reducing engine output torque or active braking and guarantee the vehicle power performance during turning. This paper researches the ride performance of a 4WD vehicle by integrating target yaw moment control with semiactive suspension control and driving wheels torque actively distributing control. This research initially realizes the chassis integrated control and
torque of left clutch and right clutch, respectively; GL = \( Z_i Z_6 / (2Z_3Z_4) \), GR = \( Z_i Z_5 / 2Z_2Z_4 \), \( Z_i (i = 1 \sim 6) \) are the number of gear teeth.

### 3. Control Strategy for Driving

**Force Distribution Based on Target Yaw Moment Control**

#### 3.1. Target Yaw Rate and Side-Slip Angle

This paper regards a single track model as the ideal reference model; the yaw rate and side-slip angle in ideal state can be calculated by this model [8]. The results, as follows, are nominal yaw rate \( \gamma_n \) and nominal side-slip angle \( \beta_n \), which are the target reference variables of the controller:

\[
\gamma_n = \frac{\delta_f v_x}{L + (mv_x^2/L) \left( \frac{b}{K_f} - a/K_r \right)},
\]

\[
\beta_n = \frac{\delta_f (b - amv_x^2/LK_r)}{L + (mv_x^2/L) \left( \frac{b}{K_f} - a/K_r \right)},
\]

where \( K_f \) and \( K_r \) are cornering stiffness of front and rear axle, respectively.

The vehicle lateral acceleration and yaw acceleration are constrained by the tire-road friction coefficient, so it is necessary to limit the range of \( \gamma_n \) and \( \beta_n \) according to the real road surface [9]:

\[
\gamma_{\text{max}} = 0.85 \mu g, \quad v_x,
\]

\[
\beta_{\text{max}} = \tan^{-1}(0.02 \mu g).
\]

Above all, the reference side-slip angle and yaw rate are defined as follows:

\[
\gamma_i = \gamma_n, \quad |\gamma_i| \leq \gamma_{\text{max}},
\]

\[
\gamma_i = \gamma_{\text{max}} \cdot \text{sgn} (\gamma_n), \quad |\gamma_i| > \gamma_{\text{max}},
\]

\[
\beta_i = \beta_n, \quad |\beta_i| \leq \beta_{\text{max}},
\]

\[
\beta_i = \beta_{\text{max}} \cdot \text{sgn} (\beta_n), \quad |\beta_i| > \beta_{\text{max}}.
\]

#### 3.2. Target Yaw Moment Control System Based on Sliding Mode Variable Structure Control

When the vehicle yaw motion is controlled by distributing driving torque or braking, an additional yaw moment \( M_t \) is generated between rear driving wheels. So the vehicle nonlinear dynamics equations are as follow [10]:

\[
mv_x (\dot{\beta} + \gamma) = (F_{yfl} + F_{yfr}) \cos \delta + F_{yrl} + F_{yrr} + (F_{xfl} + F_{xfr}) \sin \delta,
\]

\[
I_x \ddot{\gamma} = a \left[ (F_{yfl} + F_{yfr}) \cos \delta + (F_{xfl} + F_{xfr}) \sin \delta \right]
\]

\[
- b (F_{yrl} + F_{yrr})
\]

\[
+ d \left[ (F_{xfr} - F_{xfl}) \cos \delta + (F_{yfr} - F_{yfl}) \sin \delta \right]
\]

\[
+ F_{xrr} - F_{xrl} \right] + M_t.
\]
In this paper, the sliding mode variable structure control algorithm is adopted to design the target yaw moment control system \[11,12\], so that yaw rate and side-slip angle can track the ideal value well. The sliding surface \(s = 0\) is defined as the weighting of yaw rate deviation and side-slip angle deviation:

\[
s = \gamma - \gamma_n - \varepsilon (\dot{\beta} - \dot{\beta}_n),
\]

where \(\varepsilon\) is a positive constant; the greater the \(\varepsilon\), the faster the convergence.

The dynamic characteristic of the system is as follows:

\[
\dot{s} = \dot{\gamma} - \dot{\gamma}_n - \varepsilon (\dot{\beta} - \dot{\beta}_n).
\]

In order to reduce system disturbances, the switching control rate is defined as

\[
\dot{s} = -k \text{sgn}(s), \quad k > 0,
\]

where \(k\) is design parameter of controller, which determines the speed of system reaching the sliding surface.

Sign function may cause system chattering because of discontinuous switching, so this paper uses the saturation function with continuous switching property to avoid system chattering. The saturation function is defined as

\[
\text{sat}(s) = \begin{cases} 
\frac{s}{\tau}, & |s| \leq \tau \\
\text{sgn}(s), & |s| > \tau 
\end{cases}
\]

where \(\tau\) is the thickness of the boundary layer.

With \((6), (7), (9), (10),\) and \((11)\), the final target yaw moment is obtained as follows:

\[
M_t = I_z \ddot{\gamma}_n + I_z \varepsilon \left\{ \frac{1}{m v_x} \left[ (F_{yfl} + F_{yfr}) \cos \delta + F_{yrl} + F_{yrr} 
+ (F_{xfl} + F_{xfr}) \sin \delta \right] - \gamma - \dot{\beta}_n \right\} 
- k I_z \text{sat}(s) 
- a \left[ (F_{yfl} + F_{yfr}) \cos \delta + (F_{xfl} + F_{xfr}) \sin \delta \right] 
- b (F_{yrl} + F_{yrr}) 
+ d \left[ (F_{xfr} - F_{xfl}) \cos \delta + (F_{yfl} - F_{yfr}) \sin \delta 
+ F_{xrr} - F_{xrl} \right].
\]

3.3. Control Strategy Based on Target Yaw Moment

3.3.1. Logical Judgment of Vehicle Stable Steering Characteristic. When the front wheels steering angle \(\delta\) is constant, the vehicle steering characteristic can be judged by using \(\delta\) and the difference of yaw acceleration \(\Delta \ddot{\gamma}(t)\) \[8\].

\(\Delta \ddot{\gamma}\) is defined in time interval \([t_0, t_f]\). Assume that the longitudinal adhesion increment of rear wheels is \(\Delta F_{xrl}\) and \(\Delta F_{xrr}\); the lateral adhesion increments are \(\Delta F_{yrl}\) and \(\Delta F_{yrr}\). From \((6)\), the difference of yaw accelerations is derived as follows:

\[
\Delta \ddot{\gamma} = \ddot{\gamma}(t_f) - \ddot{\gamma}(t_0) 
= \frac{-d (\Delta F_{xrl} - \Delta F_{xrr}) - b (\Delta F_{yrl} + \Delta F_{yrr})}{I_z}. \tag{13}
\]

When \(\Delta \ddot{\gamma} = 0\), the vehicle is in the neutral steering condition. When \(\Delta \ddot{\gamma} < 0\), the vehicle is in the under steering condition. When \(\Delta \ddot{\gamma} > 0\), the vehicle is in the over steering condition.

3.3.2. Control Flowchart of Driving Torque Distribution. The control logical flowchart of driving torque distribution is presented as in Figure 2. Firstly, the system judges the steering characteristic by \(\delta\) and \(\Delta \ddot{\gamma}(t)\) and then delivers command signals to actuators of the target yaw moment control system. The interwheel torque distribution device transfers driving
torque to each side of wheel by engaging the corresponding clutch. This can avoid deviating from the ideal track caused by vehicle oversteering or understeering.

Assume that the longitudinal and lateral force of front tires stay constant during travelling. If the rear wheels are without braking, the dynamics equation of rear wheels can be simplified as follows:

\[ T_{rl} = F_{xrl} R_w + f_{rl} F_{zrl} R_w \]
\[ T_{rr} = F_{xrr} R_w + f_{rr} F_{zrr} R_w. \] (14)

The relationship between the generated yaw moment and the difference of \( F_{xrr} \) and \( F_{xrl} \) is as follows:

\[ M_t = d (F_{xrr} - F_{xrl}) \]
\[ = \frac{d (T_{rr} - T_{rl} + f_{rl} F_{zrl} R_w - f_{rr} F_{zrr} R_w)}{R_w}. \] (15)

Then the difference of driving torque generated by the additional yaw moment is calculated as follows:

\[ \Delta T_w = \frac{M_t R_w}{d} - (f_{rl} F_{zrl} - f_{rr} F_{zrr}) R_w. \] (16)

When the right clutch is engaged, \( T_{cr} = \Delta T_w \); when the left clutch is engaged, \( T_{cl} = \Delta T_w \).

4. Fuzzy Control System of Semiactive Suspension

4WD vehicles usually travel on bumpy and uneven road, on which their ride performance is deteriorated. The interwheel driving torque distribution active control is solely to improve the vehicle lateral stability, and it ignores the research of ride performance. So this paper designs a semiactive suspension control system based on fuzzy controllers and formulates the fuzzy control rules to improve vehicle ride performance and ride comfort.

\begin{table}[h]
\centering
\caption{The fuzzy control rule.}
\begin{tabular}{|c|c|c|c|c|c|c|c|c|}
\hline
\textbf{de/dt} & \textbf{NB} & \textbf{NM} & \textbf{NS} & \textbf{ZE} & \textbf{PS} & \textbf{PM} & \textbf{PB} \\
\hline
\textbf{e} & NB & PB & PM & PS & PS & ZE & ZE \\
\hline
\textbf{NB} & PB & PB & PM & PS & PS & ZE & ZE \\
\textbf{NM} & PB & PM & PS & PS & PS & ZE & ZE \\
\textbf{NS} & PM & PM & PS & PS & ZE & NS & NM \\
\textbf{ZE} & PM & PM & PS & PS & ZE & NS & NM \\
\textbf{PS} & PS & PS & ZE & NS & NM & NB & NB \\
\textbf{PM} & ZE & ZE & NS & NS & NM & NB & NB \\
\textbf{PB} & ZE & ZE & NM & NM & NB & NB & NB \\
\hline
\end{tabular}
\end{table}

The semiactive suspension control system, as shown in Figure 3, consists of double two-dimension fuzzy controllers, which control the rear-left and rear-right suspension, respectively. Take the design of the rear-left suspension controller for instance.

The deviation of rear-left suspension dynamic deflection \( (Z_{zrl} - Z_{zrl}) \) and their variance rate \( (\dot{Z}_{zrl} - \dot{Z}_{zrl}) \) are inputs, and the adjustable damping force \( f_3 \) is output [13, 14]. Seven-language fuzzy subset is usually used to represent the fuzzy states of a system with two inputs and one output. The subset contains negative big (NB), negative middle (NM), negative small (NS), zero equal (ZE), positive big (PB), positive middle (PM), and positive small (PS). The fuzzy domains of input and output are \([-6, 6]\). This paper uses the "Mamdani" fuzzy logical reasoning method, in which the triangular function with higher sensitivity is used as the membership function of fuzzy subset. \( K_e \) and \( K_{ec} \) are the quantification factor of the inputs, \( K_f \) is the scale factor of the output, where \( K_e = 30 \), \( K_{ec} = 1.5 \), and \( K_f = 60 \). Similarly, the fuzzy controller of rear-right suspension is designed in the same way.

According to experience and the rules to choose control variables, the fuzzy control rule of 4WD vehicle rear axle semiactive suspension is formulated in Table 1.
5. Structure of 4WD Vehicle Combined Control System

So far, the combined control system of the target yaw moment control system and the semiactive suspension control system has been designed. Figure 4 presents the general structure of this combined control system.

6. Simulation and Analysis

To verify the effectiveness of the proposed control algorithm, this study carries out offline vehicle dynamics simulations with a commercial simulation software TESIS veDYNA. The simulations contain the step steering condition test and the split-$\mu$ road turning condition test.

6.1. The Step Steering Condition. Simulation has been carried out to evaluate the performance of the designed target yaw moment control system with step steering maneuvers. The simulation conditions are as follows: choose B-class concrete road; set the initial vehicle velocity to 20 m/s; set the peak adhesion coefficient of both sides of tracks to 0.8; the steering angle of front wheels is increased up to 6 degrees after 2 s and keeps unchanged till the end of simulation. Figure 5 refers to the results of simulation. Without the target yaw moment control system, the vehicle has a great deviation between the yaw rate and the target value which approaches 3.27 deg/s.
The side-slip angle has a great overshoot which is up to 0.68 degree. In addition, the displacement curve indicates that the under steering leads to a great offset from the practical travelling route.

If the vehicle adopts the target yaw moment control system, when the vehicle enters the under steering state, the interwheel torque distribution device quickly engages the right clutch to generate a friction torque, which is transferred to the rear-right wheel. The friction torque varies around 350 N-m. The yaw rate responds rapidly and finally stabilizes to an ideal value of about 18 deg/s. Compared with the case without any control, the overshoot of side-slip angle is decreased and the side-slip angle can track the ideal value well. The steering state is changed from under steering to normal steering because of the increasing yaw rate.

Figure 6 shows the contrast of rear driving wheels longitudinal forces with the combined control against the case without the combined control. The driving torque cannot be transferred between both sides of driving wheels without the combined control. During turning, the slip ratio of the rear-right wheel exceeds the rear-left one, so the driving force of the rear-left side is greater than the rear-right side. After applying the combined control, the vehicle steering state is changed to under steering, so the driving torque is actively distributed on the rear axle to increase the driving force of the rear-right wheel and decrease the rear-left one dramatically. This action generates an additional yaw moment to restrain the under steering.

6.2. The Split-\(\mu\) Road Turning Condition. In order to verify the performance of the combined control system and validate the effectiveness of improvement in the steering and ride performance, this paper integrates the lateral and vertical dynamic response of vehicle, and the simulation has been carried out on the split-\(\mu\) road in turning condition. The simulation conditions are as follows: the input of the front wheel steering angle is a semiperiod sinusoidal signal with period and amplitude are 4 s and 6 degrees, respectively; set the initial vehicle velocity to 20 m/s; choose B-class concrete road; set the peak adhesion coefficient of the left and right side track to 0.9 and 0.5, respectively; the radius of the road center line is 25 m; the simulation time is 8 s.

As shown in Figure 7, without the combined control, the yaw rate and side-slip angle can initially follow the ideal value when the steering angle gradually increases during the initial 2~3 s. However, when the front wheels align, the yaw rate completely exceeds the ideal value because of the unbalance of rear wheels driving force. The absolute value of the practical side-slip angle also reaches about 6.5 degrees which far exceeds the corresponding ideal value. Great shift happens to the travelling route, and the vehicle comes to instability. When the combined control is applied, an additional yaw moment is generated to adjust the state of vehicle travelling. This additional yaw moment ensures the yaw rate and side-slip angle trace the ideal value very well. As shown in Figure 7, the side-slip angle is up to the maximum value of 4.7° at 3.7 s, which is decreased by 27.7% compared with the condition without the combined control. This control system makes the vehicle ride along the ideal trajectory and efficiently restrains the trend of over steering. The results verify that the combined control system can improve the steering performance and the lateral stability, when the vehicle is turning on the split-\(\mu\) road with high initial velocity.

Figure 8 shows that, without the combined control, the driving force that acted on the left wheel is smaller than the right one because the peak adhesion coefficient of left side track is smaller than the right side. When the combined control is applied, it distributes the driving torque between the rear-left and rear-right wheels according to the steering.
characteristic at the moment, increases the driving torque of rear-left wheel, and decreases the other one. This restrains the trend of over steering effectively and improves the lateral stability of vehicle in turning condition.

Figure 9 shows that the vehicle velocity is not decreased when the combined control is applied. On the contrary, the vehicle can ride along the ideal travelling route by restraining over steering. This indirectly guarantees the vehicle dynamic property. After 8 s simulating, the vehicle velocity reaches 26.23 m/s which is increased by 2.6% compared to the condition without any control.

Figure 10 shows that the semiactive suspension control system can suppress the body vibration acceleration at the peak value in turning condition, especially at the moment that the vibration amplitude is higher, and the control effect is more obvious.

The suspension dynamic deflection is an important indicator that influences the ride performance of vehicles. Figures 11 and 12 show that, compared with passive suspension, the suspension dynamic deflection is well suppressed and the tire dynamic deformation corresponding to semiactive suspension is improved. The fuzzy controllers successfully optimize the tire dynamic deformation at the peak value, but the improvement is limited when the tire dynamic deformation is minor.

Table 2 shows that, compared to the passive suspension, the body vertical acceleration is decreased by 39.25%, and the vehicle ride comfort is improved. In addition, the
dynamic deflection of rear-left and rear-right suspension is decreased by 25.52% and 31.42%, respectively. The tire dynamic deformation of rear-left and rear-right suspension are decreased by 21.82% and 20.97%, respectively. Results indicate that the semiactive suspension controller designed by this author effectively decreases the suspension dynamic deflection and the tire dynamic deformation, improves the ride performance, reduces the probability of colliding stopper, and avoids damaging the suspension.

7. Conclusions

This paper analyzed the torque distribution characteristic of the interwheel torque distribution device, designed a target yaw moment control system using the sliding mode variable structure control, proposed the torque distribution control strategy based on the target yaw moment control, identified the controller structure of the semi-active suspension system, and designed the control system based on double fuzzy controllers.

A simulation model based on the combined control of wheel torque distribution and semiactive suspension was set up with TESIS veDYN. The control systems were simulated in the step steering condition and the split-\( \mu \) road turning condition. The results indicate that the proposed torque distribution control strategy can distribute the driving torque according to the vehicle state, the performance of vehicle steering and stability was improved, the proposed semiactive suspension control system can reduce the body vibration acceleration and the suspension dynamic deflection, and the
Advances in Mechanical Engineering

Figure 11: Simulation curves of suspension dynamic deflection.

Figure 12: Simulation curves of tire dynamic deformation.

Table 2: Comparison of simulation results.

| Performance index                        | Passive suspension | Semiactive suspension based on fuzzy controller | Performance improvement (%) |
|------------------------------------------|--------------------|-----------------------------------------------|----------------------------|
| Body vertical acceleration (m/s²)        | 0.0205             | 0.0197                                        | 39.25                      |
| Rear-left suspension dynamic deflection (m) | 0.0482             | 0.0359                                        | 25.52                      |
| Rear-right suspension dynamic deflection (m) | 0.0541             | 0.0371                                        | 31.42                      |
| Rear-left tire dynamic deformation (m)   | 0.0055             | 0.0043                                        | 21.82                      |
| Rear-right tire dynamic deformation (m)  | 0.0062             | 0.0049                                        | 20.97                      |
ride performance and ride comfort were improved. The combined control system with the two subsystems guaranteed the dynamic property of the 4WD vehicle and improved its comprehensive performance.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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