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Coordinated control of a semi-active suspension system and an electronic stability program on a finite-state basis

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ABSTRACT

Considering the different effective working areas corresponding to respective functions of SASS and ESP subsystems, the coordinated control of SASS and ESP based on the basis of finite-state is proposed to enhance the vehicle comprehensive performance. On the basis of the parallel connection between the SASS and ESP, an upper coordination controller on the finite-state basis is then combined in series with the SASS and ESP systems. The upper coordination controller identifies the vehicle's main driving conditions and make the strategy amendments to the bottom automotive semi-active suspension and electronic stability program. A SASS with PID control is presented along with an ESP with logic threshold control containing a variable slip rate and direct yaw moment control. Simulation results demonstrate that the coordinated control of the SASS and ESP on a finite-state basis can effectively improve the vehicle ride comfort and handling stability under multiple conditions.

Keywords: Semi-active suspension system,Electronic stability system,Finite-state basis,Strategy amendment
I-INCE Classification of Subject Number:40

1.INTRODUCTION

With the rapid development of modern automobile manufacturing and engineering technology, various advanced chassis control systems, for instance, traction control systems (TCS), rear-wheel steering (RWS), direct yawmoment control (DYC) and active front steering (AFS), have been widely used. Vehicle chassis system is an organic whole composed of many subsystems, such as brake, steering and suspension. These subsystems do not exist independently of each other but are a unity of opposites with different dynamic coupling relations and interactions. Subsystems are optimized in accordance with different target functions. When an active control or an intervention is implemented to a subsystem, the order will inevitably influence the performance of other subsystems of the vehicle. Practically, the simple superposition of electronic control subsystems, without taking the interactions into consideration, cannot have an effect on optimizing the overall performance of vehicles.

An integrated vehicle chassis control method has then been developed (Furukawa and Abe, 1997; Nagai et al., 1997; Ono et al., 1994) which deals with the interference and conflicts caused by the different control objectives while takes full advantage of respective function， so as to ensure the coordination between the subsystems and achieve the maximum improvement of the overall performance as much as possible.

Mousavinejad et al. (2017) presented an integrated vehicle dynamics control algorithm based on a coordinated control of AFS and DYC systems as well as two advanced sliding mode control strategies to improve handling and stability of a ground vehicle. Other applications of the coordinated control of AFS and DYC were addressed in Boada et al. (2013)whointegrated rear braking and front steering to obtain the desired yaw rate , Yang et al. (2009) who presented an optimal guaranteed cost coordination controller and Zhang et al. (2016) who took the variation of longitudinal velocity in consideration. An Integrated Chassis Control (ICC) strategy was proposed in Heo et al. (2015) to assists the vehicle in greatly reducing lap time by improving driving speed at the limits of handling through the integration of electronic stability control (ESC), Four Wheel Drive (4WD) and Active Roll Control System (ARS). In Cho et al. (2008), a unified chassis control (UCC) strategy was introduced which provides crucial improvement of vehicle manoeuvrability and lateral stability by involving ESC, AFS and continuous damping control (CDC). Yim et al. (2015) presented a new type of UCC used for the under-steer situation by combining the braking force of ESC with the steering angle of AFS. In Di Cairano et al. (2010), a switched model predictive controller was implemented to coordinate AFS and differential braking for a driver-assist steering system. Song et al. (2015) built a model-based chassis controller for utilizing steering, traction and braking systems of full drive-by-wire vehicles based on a systematic approach. By combining AFS with Rear Torque Vectoring (RTV) actuators, an integrated controller was designed in Bianchi et al. (2010) based on an adaptive feedback technique.

Integrated vehicle chassis control is classified into two different types: (1) Centralized control (He et al., 2006; Hwang et al., 2008) and (2) parallel control (Chen et al., 2006; Chen et al., 2007; Chu, C-B. and Chen, W-W. 2008). Centralized control adjusts the control parameters by analyzing the running conditions of the vehicle. The main characteristic of centralized control lies in that it integrates multiple systems into one controller. However, it has the disadvantage that the entire controller will fail if a parameter or a system fails. In contrast, parallel control connects subsystems in a parallel structure and each sub-controller adjusts parameters automatically according to the different vehicle driving conditions. Nevertheless, application of such a control system also causes a number of disadvantages, some of which are: (1) over-abundant vehicle parameters, (2) repeated use of sensors, and (3) complicated control process.

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2.VEHICLE MODEL

2.1 Vehicle Dynamics Model

The 7-DOF vehicle dynamics model with semi-active suspension is shown in Figure 1. The axis goes through the suspended mass centroid and that is perpendicular to the ground is defined as the Z-axis. The driving direction of the vehicle is defined as the X-axis. The line goes through the centroid and that is perpendicular to both the X and Z axes is defined as the Y-axis.

Considering the interaction of roll, pitch, yaw, and vertical motion, the motion equations of the vehicle can be derived as follows:

(1) Suspension roll motion equation:

$$\begin{aligned}&(I_x + m_s \cdot H^2)(\ddot{\phi} - \theta \cdot \dot{\omega} - \dot{\theta} \cdot \omega) - (I_y + m_s H^2)(\dot{\theta} \cdot \omega + \phi \cdot \omega^2) - (I_{xz} + m_s \cdot H \cdot L_s) \cdot \omega + \\ &(I_z + m_s \cdot {L_s}^2) \cdot \theta \cdot \dot{\omega} + I_z(\dot{\theta} \cdot \omega + \phi \cdot \omega^2) - m_s(\dot{v} + u \cdot \omega) \cdot (H - L_s \cdot \theta) = \sum MX\end{aligned} \tag{1}$$

where

$$\sum MX = (F_2 + F_3 - F_1 - F_4)d \tag{2}$$

(2) Suspension vertical motion equation:

$$m_s(\ddot{z}_s - L_s \cdot \ddot{\theta}) = \sum F_i \tag{3}$$

where

$$F_i = k_{si}(z_{ui} - z_{si}) + c_i(\dot{z}_{ui} - \dot{z}_{si}) \pm \frac{k_{af}}{2d}(\phi - \frac{z_{ui} - z_{ui}}{2d}) + f_{imag} \tag{4}$$

$$m_{ui}\ddot{z}_{ui} = k_{ti}(z_{gi} - z_{ui}) - F_i \tag{5}$$

When the pitch angle and the roll angle are small, we have approximately

$$z_{si} = z_s \pm a\theta \pm d\varphi \tag{6}$$

(3) Suspension pitch motion equation:

$$\begin{aligned}&(I_y + m_s \cdot H^2)(\ddot{\theta} + \phi \cdot \dot{\omega} + \dot{\phi} \cdot \omega) + (I_x + m_s \cdot H^2)(\dot{\phi} \cdot \omega - \theta \cdot \omega^2) - (I_{xz} + m_s \cdot H \cdot L_s) \cdot \omega^2 + \\ &(I_z + m_s \cdot {L_s}^2) \cdot \theta \cdot \omega^2 - I_z(\dot{\phi} \cdot \omega + \phi \cdot \dot{\omega}) + m_s(\dot{u} - v \cdot \omega)(H - L_s \cdot \theta) - m_s \cdot L_s \cdot \ddot{Z}_s + m_s \cdot {L_s}^2 \cdot \ddot{\theta} = \sum MY\end{aligned} \tag{7}$$

where

$$\sum MY = b(F_3 + F_4) - a(F_1 + F_2) \tag{8}$$

In Equation (1)-(8):

I_x , I_y and I_z stand for the roll, pitch and yaw motion of inertia of the suspended mass,respectively; I_{xz} is the product of inertia of the suspended mass about the longitudinalaxis and the vertical axis crossing the suspended mass centroid; MX and MY are the moments of the suspended mass around the X and Y axes, respectively; m_s denotes the suspended mass of the vehicle; His the vertical height from the suspended mass centroid to the roll axis; L_s is the longitudinal distance from the suspended mass centroid to the vehicle centroid; ω is the angular yaw velocity of the vehicle; Uis the longitudinal velocity of vehicle; V is the lateral velocity of the vehicle; f_{imag} is the controllable damping force of the electromagnetic valve in the semi-active suspension. The physical significance of the other parameters is shown in Figure 1.

2.2 Half-Vehicle Dynamics Model

If we consider the steering angles of the two front wheels to be identical at all times, then the vehicle can be simplified into a 2-DOF linear model, which is commonly used in the design of vehicle stability controllers to make the application of the control algorithm much more convenient, as shown in Figure 2.

The dynamics equation is

$$(k_f + k_r)\beta + \frac{1}{v_x}(l_f k_f - l_r k_r)r - k_f\delta = m(\dot{v}_y + v_x r) \quad (9)$$

$$(l_f k_f - l_r k_r)\beta + \frac{1}{v_x}(l_f^2 k_f + l_r^2 k_r)r - l_f k_f \delta = I_z r \quad (10)$$

In Equation (9) and (10), β is the side-slip angle of the vehicle; r is the yaw rate; l_f and l_r are the distances from the centre of the vehicle mass to the front and rear axles, respectively; k_f and k_r are the lateral stiffness values of the front and rear wheels, respectively.

2.3 Braking Model

Owing to the complex operating conditions of the chassis system, the influence of the longitudinal and lateral accelerations on the load transfer of the vehicle must be fully considered when a mathematical model of the braking system is to be established. Moreover, the variation in the vertical load also leads to the changes in the lateral force and the longitudinal force on each wheel. Hence, when steering is taken into consideration, the braking model is given by

$$m\dot{u} = -(F_{x1}\cos\delta_f + F_{x2}\cos\delta_f + F_{y1}\sin\delta_f + F_{y2}\sin\delta_f + F_{x3} + F_{x4}) \quad (11)$$

The lateral force and longitudinal force acting on each wheel in Equation (9) can be calculated by the type model described in section 2.4, and the mathematical model of the braking moment can be described as follows:

$$T_{bi} = (4F_b i_b \eta_p B/(\pi D^2{}_m) - p_0)A_{wc}\eta B_{Fi} R \quad (12)$$

The rotational motion equation of each steering wheel can be written as

$$I_{ti}\dot{\omega}_{ti} = F_{ti}R_i - T_{bi} \quad (13)$$

In Equation (11)-(13)：

F_b is the pedal force; i_b is the braking lever ratio; η_p is the efficiency of the manoeuvring mechanism; B is the transmission ratio of the booster; D_m is the diameter of the brake master cylinder; p_0 is the pressure loss; A_{wc} is the wheel area of the brakecylinder; η is the efficiency of the brake cylinder; B_{fi} is the brake efficiency factor; R is the radius of the wheel brake drum; I_{ti} is the moment of inertia of the wheel; F_{ti} is the longitudinal force acting on the ith wheel; R_i is the radius of the wheel.Considering the load transfer as well as the effect of the radial force, the vertical load of the wheel is derived as(Yu F.and Lin Y.,2005):

$$F_{z1} = \frac{1}{L}[bg + \dot{u}h_o + \dot{v}\frac{h_o(b - \frac{h_o}{L})}{R_0}]m/2 + \dot{v}k_{\alpha f}m_s \quad (14)$$

$$F_{z2} = \frac{1}{L}[bg + \dot{u}h_o + \dot{v}\frac{h_o(b - \frac{h_o}{L})}{R_0}]m/2 - \dot{v}k_{\alpha f}m_s \quad (15)$$

$$F_{z3} = \frac{1}{L}[ag - \dot{u}h_o - \dot{v}\frac{h_o(b - \frac{h_o}{L})}{R_0}]m/2 + \dot{v}k_{\alpha r}m_s \quad (16)$$

$$F_{z4} = \frac{1}{L}[ag - \dot{u}h_o - \dot{v}\frac{h_o(b - \frac{h_o}{L})}{R_0}]m/2 - \dot{v}k_{\alpha r}m_s \quad (17)$$

In Equation (14)-(17), R_o is the steering radius; L is the wheelbase; h_o is the distance from the mass centroid to the roll axis of the vehicle; $k_{\alpha f}$ and $k_{\alpha r}$ are the equivalent lateral stiffness values of the front and rear axles, respectively.

3. CONTROLLER DESIGN

The coordinated control block diagram of the SASS and ESP systems on the finite-state basis is depicted in Figure 3. A vehicle finite-operating state identification and coordinated controller is designed at the upper part of the SASS and ESP subsystems. With the information collected and shared from the sensors and the subsystem, the controller identifies the vehicle's main driving conditions, which compose the vehicle's finite-state basis. Then, it makes basic strategic amendments to the SASS and ESP subsystems based on the control objectives under different vehicle driving conditions. The vehicle's main driving conditions, namely, the finite-state basis of the operating vehicle, can be divided into six types according to the driving conditions and movement relationships of the vehicle: uniform driving, emergency acceleration, emergency braking, step steering, steering braking, and steering acceleration. In the bottom subsystem, the SASS system adopts a PID algorithm with parameter adjustment while the ESP braking system uses the logic threshold control with the variable slip rate. The vehicle's body stability system implements the yaw-moment direct hierarchical control, which combines the upper sliding mode with the lower braking torque control.

3.1 SASS Controller Design

In this study, the PID algorithm was proposed to distribute weights of the controllers on the four suspensions. The corresponding control block diagram is shown in Figure 4. The vertical acceleration of the vehicle body above the **i**th suspension $\ddot{Z}_{si}$ was builtas the input of the controller. Assuming that the output of the controller at time k is $u(k)$ **,** the acceleration collected at this time is $\ddot{Z}_{si}(k)$ **,** and the proportional, integral and differential coefficients of the controller on the ith suspension are K_{pi} 、 K_{ii} and K_{di} （i=1~4）,respectively, the output can be formulated with incremental PID algorithm as follows:

$$u(k) = K_{pi}[\ddot{z}_{si}(k) - \ddot{z}_{si}(k-1)] + K_{ii}\ddot{z}_{si}(k) + K_{di}[\ddot{z}_{si}(k) - 2\ddot{z}_{si}(k-1) + \ddot{z}_{si}(k-2)] \tag{18}$$

When the vehicle runs at uniform speed, the controller works under ordinary PID control, wherein control parameters remain constant. When the roll or pitch motion happens during the conditions of steering, acceleration, and braking, the PID coefficients are adjusted as follows. There are two rule tables in the upper coordination distributor about adjusting the scale coefficient of the suspension. One table adjusts the roll ratio factor,with $\dot{\varphi}$ and the rate of change $\ddot{\varphi}$ as the input where the PID control regulatory factors K_{pi} , K_{ii} and K_{di} are the output. (When the vehicle rolls, the 1st and 3rd suspension controllers as well as the 2nd and 4th suspension controllers are adjusted in pairs.) Another table is for pitch coefficient adjustment rules, with $\dot{\theta}$ and its rate of change $\ddot{\theta}$ as the inputs, where the PID control regulation factors K_{pi} , K_{ii} and K_{di} are the output. (When the vehicle pitches, the 1st and 2nd suspension controllers as well as the 3rd and 4th suspension controllers are adjusted in pairs.)

3.2 ESP Controller Design

The braking system of the ESP system adopts logic threshold control with a variable slip rate while the vehicle body stability system uses yaw-moment direct hierarchical control, which combines the upper sliding mode with the lower braking torque control.

3.2.1 Braking System Controller Design

The anti-lock braking system employs a logic threshold control algorithm with a variable slip rate, as shown in Figure 5. First, the vehicle slip rate is calculated by the parameters collected from the vehicle driving state, and then the rate is compared with the target slip rate transmitted by the upper identification coordinator. With the error obtained by the comparison, amendments are made to the brake pressure so that the slip rate can be kept close to the optimal slip ratio and the brake performance of the system can be improved.

3.2.2 DYC Controller Design

Direct yaw-moment control has a hierarchical structure and consists of the upper and lower controllers (Liu X-Y. and Chen W-W., 2009), as shown in Figure 6. The DYC upper controller calculates the yaw moment which is used to correct the vehicle's state back to that desired, named the additional yaw moment, and then supplies the result of the calculation to the lower controller. The lower controller calculates the variation in the longitudinal force on the active brake wheel based on the additional yaw moment and then implements the control.

（1）DYC upper controller design

The sliding mode control algorithm is designed according to the 2-DOF vehicle model, and from Equation (9) and (10), we have

$$\dot{x} = Ax + Bu \tag{19}$$

where

$$A=\begin{pmatrix} -(k_f+k_r)/mv_x^2 & (k_f l_f+k_r l_r)/mv_x^2-1 \\ (k_r l_r-k_f l_f)/I_z v_x & -(k_f l_f^2+k_r l_r^2)/I_z v_x \end{pmatrix}, B=\begin{bmatrix}0 & 1\end{bmatrix}^T, x=\begin{bmatrix}\beta & r\end{bmatrix}^T, u=\begin{bmatrix}\Delta M\end{bmatrix},$$

where sliding surface is described by

$$\delta = Sx \tag{20}$$

For brevity, the solution process is not described in detail here, and the results are given directly as follows:

$$S=\begin{bmatrix}-0.0107 & 0.0124\end{bmatrix}.$$

（2）DYC bottomcontroller design

The DYC lower controller mainly transfers the additional yaw moment to the wheel cylinder pressure, and then applies it at the actuator. The calculation of the wheel cylinder pressure proceeds are as follows. First, the additional yaw moment ΔM calculated by the DYC is converted into the change in longitudinal force on one side wheel. Then, we can convert the change in the longitudinal force into the change in wheel cylinder pressure with the wheel movement model. Taking the condition that two right wheels brake simultaneously as an example, the expression for converting the additional yaw moment into longitudinal force (Wang et al., 2009)can be derived as follows:

$$\Delta M = \frac{1}{2}F_{xfr}D_f + \frac{1}{2}F_{xrr}D_r \tag{21}$$

For lack of space, the detailed process is not described here, and the longitudinal force increment is given directly:

$$F_{xfr} = F_{xrr} = F_d = -\frac{1}{R}(J_w \frac{d\omega}{dt} + CP_w) \qquad (22)$$

In Equation (22), J_w is the moment of inertia of the wheel； R is the radius of the wheel; ω is the angular velocity of the wheel; P_w is the target pressure of the wheel cylinders; $C = A_w u_b R_b$ is a coefficient determined by structural parameters such as the brake shoe area A_w, the brake shoe friction coefficient u_b, and the distance from the brake shoe to the wheel centre R_b.

3.3 Upper Identification and Coordination Controller Design

The finite-state basis consists of the following six types of vehicle driving conditions: uniform driving, emergency acceleration, emergency braking, step steering, steering braking, and steering acceleration. The signals obtained and shared by the sensors are the longitudinal velocity u, lateral velocity v, longitudinal acceleration $\dot{u}$, lateral acceleration $\dot{v}$,pitch angular velocity $\dot{\theta}$, roll angular velocity $\dot{\varphi}$ ， steering wheel angle， braking, the rate of change and the opening of the throttle, etc. After being processed, the signals mentioned above are sent to an identification and coordination controller on a finite-state basis. The controller then identifies the vehicle driving conditions accurately through theoretical analysis and calculation. The upper identification and coordination controller makes strategic amendments to the bottom automotive semi-active suspension and electronic stability program based on the identification of the vehicle's main driving conditions. The amendment rules are shown in Table 1, and specific amendment rules are given below：

Table 1 Identification and coordination control rule table on a finite-state basis

Finite-state basis		Identification condition	ESP		SASS
			DYC	ABS	
1	Uniform Driving	$\dot{u} = 0$ and $\dot{v} = 0$	does not work	does not work	PID parameters remain unchanged
2	Emergency Acceleration	$\dot{u} > 0$ and $\dot{v} = 0$	does not work	does not work	adjust PID parameters
3	Emergency Braking	$\dot{u} < 0$ and $\dot{v} = 0$	does not work	works	adjust PID parameters
4	Step Steering	$\lvert \dot{u} \rvert = 0$ and $\lvert \dot{v} \rvert > 0$	works	does not work	adjust PID parameters
5	Steering Braking	$\dot{u} < 0$ and $\lvert \dot{v} \rvert > 0$	works	works	adjust PID parameters
6	Steering Acceleration	$\dot{u} > 0$ and $\lvert \dot{v} \rvert > 0$	works	does not work	adjust PID parameters

（1）Uniform driving

•The ABS and DYC do not work;

•The SASS controller works under ordinary PID control where the parameters remain fixed during the operation.

（2） Emergency acceleration

•The ABS and DYC do not work;

- The vehicle body pitches, and the upper identification and coordination controlleradjusts the weight of the controller in the front and rear suspension by sending out amendment commands to the SASS controller.

Specific adjustment strategies are as follows (the anti-clockwise direction of rotating around the Y axis of the body is taken to be positive):

If $\dot{\theta} > \alpha_1$ and $\ddot{\theta} > \alpha_2$, $K_{p1} = K_{p2} = \sigma_1$, $K_{p3} = K_{p4} = \sigma_2$

In addition, the real-time values of K_{ii} and K_{di} are changed in accordance with Table 2:

Table 2PID parameter adjustment rule table for emergency acceleration

Parameter	$\ddot{\theta}$		
	$\ddot{\theta} < -\alpha_2$	$-\alpha_2 < \ddot{\theta} > \alpha_2$	$\ddot{\theta} > \alpha_2$
K_{i1} K_{i2}	0.98	0.80	1.20
K_{i3} K_{i4}	1.00	0.80	1.10
K_{d1} K_{d2}	3.10	3.00	3.21
K_{d3} K_{d4}	3.11	3.00	3.30

In the rules described above, the values of variables α_i and σ_i are obtained from continuous adjustment during the simulation, and the data under the emergency braking condition in Rule 3 are obtained with the same method.

（3）Emergency braking

- TheABS works based on logic threshold control with a variable slip rate;
- The DYC does not work;
- The vehicle body pitches, and the upper identification and coordination controller adjusts the weight of the controller in the front and rear suspensions by sending out amendment commands to the SASS controller. The specific adjustment methods are similar to the case of emergency acceleration in Rule 2. Thus, the detailed discussion is not provided here.

In addition, the real-time values of K_{ii} and K_{di} are changed in accordance with Table 3:

Table 3PID parameter adjustment rule table for emergency braking

Parameter	$\ddot{\theta}$		
	$\ddot{\theta} < -\alpha_2$	$-\alpha_2 < \ddot{\theta} > \alpha_2$	$\ddot{\theta} > \alpha_2$
K_{i1} K_{i2}	1.00	0.80	1.12
K_{i3} K_{i4}	1.06	0.80	1.00
K_{d1} K_{d2}	3.18	3.00	3.14
K_{d3} K_{d4}	3.16	3.00	3.24

（4）Step steering

- The ABS does not work;
- The upper identification and coordination controller issues amendment commands to the DYC system. In order to make the angular yaw velocity approach the target value, the DYC system applies direct yaw-moment control to the corresponding wheel based on the yaw angular velocity error between the target value and the actual value provided by the controller.
- Owing to the emergence of roll motion in the vehicle body, the upper identification and coordination controller issues amendment commands to the bottom SASS controller based on the degree of roll motion to adjust the weights of the controller on the left and right suspensions.

In addition, the real-time values of K_{ii} and K_{di} are changed in accordance with the following Table 4:

Table 4PID parameter adjustment rule table for step steering

Parameter	$\ddot{\varphi}$		
	$\ddot{\varphi} < -\beta_2$	$-\beta_2 < \ddot{\varphi} < \beta_2$	$\ddot{\varphi} > \beta_2$
K_{i1} K_{i3}	1.06	0.80	1.10
K_{i2} K_{i4}	1.16	0.80	1.04
K_{d1} K_{d3}	3.10	3.00	3.12
K_{d2} K_{d4}	3.18	3.00	3.10

In the rules described above, the values of variables β_i and λ_i are obtained from continuous adjustment during the simulation, and the data under the acceleration condition in Rule 6 are obtained by the same method.

（5）Steering braking

- Based on the change in the initial braking speed, lateral acceleration, and longitudinal acceleration, the upper identification and coordination controller issues amendment commands to the ABS system and adjusts the desired slip ratio value λ_0 of each wheel in a real time;
- The upper identification and coordination controller issues amendment commands to the DYC system. Then, the DYC controller calculates the target angular yaw velocity according to the wheel angle signal and the real-time vehicle speed signal, and implements direct yaw-moment control at the corresponding wheel in order to track the optimal slip ratio of the vehicle;
- The upper identification and coordination controller issues amendment commands to the SASS system. The coupling of pitch motion and roll motion occurs in The vehicle body. Thus, a control factor ε and a priority must be defined. When $\left|\ddot{\theta} > \varepsilon_0\right|$ and $\left|\ddot{\varphi} > \gamma_0\right|$ (the values of ε_0 and γ_0 are obtained from continuous adjustment during the simulation), the coupling of roll motion and pitch motion occurs. Otherwise, the maximum value of $\max\{\ddot{\theta} \quad 2\ddot{\varphi}\}$ is used as a reference for the controller output in the four suspensions. When the coupling of pitch motion and roll motion occurs, the scale factor of the controller in the ith suspension is computed by using the following equations:

$$K_{pi} = \varepsilon K^{\theta}_{pi_{xrr}} + (1-\varepsilon) K^{\varphi}_{pi} \tag{23}$$

where K^{θ}_{pi} is the scale coefficient of the **i**th controller obtained from the pitch rule table; K^{φ}_{pi} is the scale coefficient of the **i**th controller obtained from the roll rule table; ε is deter mined from $\ddot{\theta}$ and $\ddot{\varphi}$ in real time. Due to the relatively small steady-state error and the dy namic-state error, the integral and differential coefficient values of controllers in each s uspension are determined by the following rules in order to reduce computation time:

If $|\ddot{\varphi}| > \beta_2$ and $|\ddot{\theta}| > \alpha_2$ $K_i = 1.0$ $K_d = 3.16$

Else $K_i = 0.8$ $K_d = 3.0$

（6）Steering acceleration

- The ABS does not work;
- The upper identification and coordination controller issues amendment commands to the DYC system. In order to make the angular yaw velocity approach the target value, the DYC system applies direct yaw-moment control to the corresponding wheel based on the yaw angular velocity error between the desired value and the actual value provided by the controller.

- The upper identification and coordination controller issues amendment commands to the SASS system. The vehicle body pitches and the SASS adjusts the weights of the controller in the front and rear suspensions based on the degree of roll motion. The specific adjustment methods are similar to those in the case of step steering in Rule 4, and thus, its detailed discussion is not given here.

The real-time values of K_{ii} and K_{di} are changed in accordance with Table 5:

Table 5 PID parameter adjustment rule table for steering acceleration

Parameter	$\ddot{\varphi}$		
	$\ddot{\varphi} < -\beta_4$	$-\beta_4 < \ddot{\varphi} < \beta_4$	$\ddot{\varphi} > \beta_4$
K_{i1} K_{i3}	0.98	0.80	1.00
K_{i2} K_{i4}	1.10	0.80	1.00
K_{d1} K_{d3}	3.01	3.00	3.06
K_{d2} K_{d4}	3.15	3.00	3.02

4. Simulation Analysis

According to the algorithm mentioned above, the simulation and calculation areperformed in MATLAB and the relevant parameters are listed in Table 6. Six types of simulation conditions are taken into consideration: uniform driving, emergency braking, step steering, steering acceleration at a speed of 60 km/h, emergency acceleration from a stationary state to 60 km/h and steering braking at 60 km/h. For brevity, only a few representative simulation graphs are presented, which are shown in Figure 7 to Figure12, and other simulation results are presented in Table 7.

Table 6 Relevant vehicle parameters

Notation	Symbol	Units	Value
total mass	m	kg	1,360
sprung mass	m	kg	1,200
front unsprung mass	$m_{u1}(m_{u2})$	kg	45
rear unsprung mass	$m_{u3}(m_{u4})$	kg	35
front suspension stiffness	$k_{s1}(k_{s2})$	N/m	19,000
rear suspension stiffness	$k_{s3}(k_{s4})$	N/m	17,000
angular stiffness of stabilizer bar	$k_{af}(k_{ar})$	Nm/rad	6,695
vertical height from sprung mass centroid to roll axis	H	m	0.14
longitudinal distance from sprung mass centroid to total mass centroid	Ls	m	0.16
front suspension damping	$c_{s1}(c_{s2})$	N.s/m	560
rear suspension damping	$c_{s3}(c_{s4})$	N.s/m	560
height of total mass centroid	h	m	0.58
wheelbase	d	m	1.36
distance from front wheel to centroid	a	m	1.1
distance from rear wheel to centroid	b	m	1.24
roll moment of inertia	I_x	kg•m^2	521
pitch moment of inertia	I_y	kg•m^2	1,218
yaw moment of inertia	I_z	kg•m^2	1,207
lower cut-off frequency	f_0	Hz	0.01
road roughness coefficient	G_0	m^3/cycle	5.0*10^{-6}
type stiffness	kt	N.m	13,8000

Table 7 Simulation Result Statistic

Finite-state basis	Control method	vertical	roll	pitch	yaw	brake
		$\ddot{z}_s$(m/s2)	$\ddot{\varphi}$(rad/ s^2)	$\ddot{\theta}$(rad/	ω_z (rad)	S(m)

				s²)			
1	uniform driving	NC	0.9270	0.0153	0.0186	--	--
		IC	0.7590	0.0137	0.0183	--	--
		CC	--	--	--	--	--
2	emergency acceleration	NC	1.1500	--	0.2807	--	--
		IC	1.0310	--	0.2637	--	--
		CC	0.9000	--	0.2306	--	--
3	emergency braking	NC	1.1830	--	0.3060	--	22.8248
		IC	1.1020	--	0.2757	--	21.6871
		CC	0.9890	--	0.2446	--	19.9293
4	step steering	NC	1.1710	0.0267	--	0.0899	--
		IC	1.1060	0.0246	--	0.0850	--
		CC	1.0320	0.0220	--	0.0780	--
5	steering braking	NC	0.7388	0.0147	0.0136	0.2655	25.1100
		IC	0.5651	0.0124	0.0121	0.2198	22.1200
		CC	0.5481	0.0117	0.0107	0.1954	21.1200
6	steering acceleration	NC	0.6408	0.0112	0.0110	0.2310	--
		IC	0.5442	0.0102	0.0100	0.2001	--
		CC	0.5013	0.0094	0.0092	0.1807	--

In summary, the simulation results obtained with the application of coordination control of SASS and ESP on the finite-state basis illustrate that considerable improvements are achieved in the overall vehicle performance. Further, the proposed control system is effective in improving vehicle's vertical, roll, pitch, yaw, and braking performance indices under multiple driving conditions, such as uniform driving, urgent acceleration, emergency braking, step steering, steering braking, and steering acceleration.

5. Conclusions

(1) The controller can effectively improve vehicle ride comfort by inhibiting the body's vertical vibration, pitch, and roll motion under limited driving conditions.
(2) The direct yaw-moment controller can track the vehicle body's target angular yaw velocity accurately and improve vehicle handling stability based on adjustment instructions issued by the upper identification coordinator.
(3) According to the real-time target slip ratio provided by the upper identification coordinator, the braking system controller maintains the wheels at the optimal slip rate and shortens the braking distance, thus improving braking performance.

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