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A synergy control framework for enlarging vehicle stability region with experimental verification

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Vehicle stability control has made remarkable contributions in improving vehicle safety and performance, especially during critical driving conditions. Many researchers have made tremendous efforts with regard to various active vehicle stability systems, and integrated vehicle dynamics control has attracted wide interest in research [1], including the global chassis control (GCC) [2], vehicle dynamic stability control (VDSC) [3], holistic corner control (HCC) [4], among others. Sometimes, they cannot perform well due to the coupling of vehicle dynamics and the highly complex working condition of the vehicle itself, which highly rely on the nonlinear and interdependent tire longitudinal, lateral and vertical forces. Currently, the vigorous development of the intelligent vehicle, advanced driver-assistance system (ADAS) [5] and new chassis structures (such as 4WD EVs) [6, 7] strongly require the effective utilization of nonlinear tire forces and an extension of the vehicle stability region to develop more effective chassis control strategies.

To solve this problem, a synergy control framework for enlarging the vehicle stability region is proposed. In this framework, a vehicle dynamics model combining the longitudinal, lateral, and vertical tire forces is modeled. Then, the vehicle performance requirements are described as the optimal control objectives. The safety constraints are another key component in this framework. Here, phase plane analysis is conducted based on the nonlinear vehicle model to graphically identify the vehicle stability boundary in real time, which is crucial as constraints for the development of a vehicle stability controller. The structure of overall control is shown in Figure 1.

Combined tire forces and vehicle dynamics. A high precision and concise tire model, the "UniTire model", under combined slip situations is adopted in this research.

The basic formulae for tire forces F of the UniTire model are expressed as

$$\begin{cases} \bar{F} = 1 - \exp[-\phi - E\phi^2 - (E^2 + 1/12)\phi^3], \\ F = \bar{F} \cdot \mu F_z, \end{cases} \tag{1}$$

where $\bar{F}$ and ϕ are the normalized tire force and the tire slip ratio, respectively; F_z is the vertical load, E is the resultant curvature factor, and μ is the dynamic friction coefficient, which considers the significant influence of slip speed. The expression is as follows:

$$\mu = \mu_s + (\mu_0 - \mu_s) \cdot \exp \left(-{\mu_h}^2 \cdot \log^2 \left(\left| \frac{V_s}{V_{sm}} \right| + N \cdot \exp \left(- \left| \frac{V_s}{V_{sm}} \right| \right) \right) \right), \tag{2}$$

where μ_0, μ_s, μ_h, V_{sm}, and N are the friction characteristic parameters, and V_s is the slip speed.

To accurately describe the combined tire forces,

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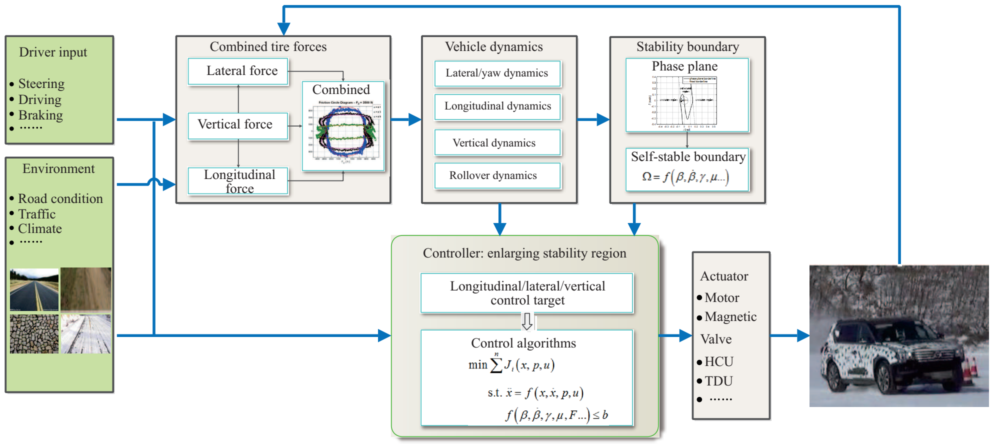

Figure 1 (Color online) Control framework for enlarging vehicle stability region.

a direction factor λ is introduced as

$$\lambda = \frac{1 + (\phi/\phi_c)^n \cdot K_y/K_x}{1 + (\phi/\phi_c)^n}, \quad (3)$$

where ϕ_c and n are model parameters; K_x and K_y are the longitudinal slip stiffness and the cornering stiffness, respectively.

By considering the dynamic friction coefficients and direction factor, the nonlinear and interactional tire longitudinal, lateral, and vertical forces can be precisely expressed.

Tires contribute mostly to the nonlinear characteristics of vehicle dynamics. Adopting the UniTire model mentioned above, the nonlinear vehicle dynamics can be described, including the longitudinal, lateral, vertical forces, yaw, pitch, and roll motion.

$$\dot{\Gamma} = f(\Gamma(t), u). \quad (4)$$

The vehicle state vector Γ, control input vector u, and system output y are defined as

$$\Gamma = [V_x, V_y, V_z, \omega_x, \omega_y, \gamma, \omega_{fl}, \omega_{fr}, \omega_{rl}, \omega_{rr}], \quad (5)$$

$$u = \begin{cases} [T_{fl}, T_{fr}, T_{rl}, T_{rr}], & \text{for EV}, \\ [p_{fl}, p_{fr}, p_{rl}, p_{rr}, T_e, T_c], & \text{for ICV}, \end{cases} \quad (6)$$

$$y = [V_x, a_x, \omega_x, \gamma, \beta], \quad (7)$$

where V_x, V_y, and V_z indicate forward, lateral, and vertical velocities, respectively, of the sprung mass. There is roll angular velocity, ω_x, pitch angular velocity, ω_y, and yaw angular velocity, γ. ω_{ij} is angular velocity of wheel rotation. T_{ij} is the wheel torque; p_{ij} is the pressure of wheel cylinder; T_e is the engine torque; T_c is the friction moment of clutch; a_x is the longitudinal acceleration; β is vehicle sideslip angle.

Identification of stability boundary. The phase plane of sideslip angle, yaw rate, or tire slip angles, among others, could be used to reflect the influence of the initial vehicle states on its subsequent trajectory for understanding vehicle behavior. The phase plane is divided into self-stable and unstable regions. Vehicles are always stable in the self-stable region, but for vehicles that have entered the unstable region, advanced chassis control should be applied to stabilize the vehicle. Therefore, it is crucial to identify and predict the vehicle stability boundary in real time.

For different maneuver coefficients and vehicle parameters, the feature of the vehicle trajectories can be reflected by the types of singular points (i.e., node, saddle, and spiral) representing the equilibrium solutions in the phase plane. While the parameters of the tire and vehicle system are predetermined, the phase plane trajectories mainly depend on vehicle velocity V_x, road friction coefficient μ, steering angle input δ_f, and the coupling relationships among them. The boundary of the self-stable region Ω can be described as follows:

$$\Omega = f(\beta, \dot{\beta}, \gamma, \mu, V_x, \delta_f). \quad (8)$$

Controller for enlarging stability region. The main control requirement is to ensure good handling and stability on cornering, and the vehicle speed should not slow down obviously compared with the driver's demand. Besides, for vehicles having a high-gravity center, such as a sports utility vehicle, rollovers might still occur. Thus, longitudinal speed control, lateral/yaw stability, and rollover prevention should be incorporated into the synergy control targets.

(1) The longitudinal motion control contains speed tracking control and active acceleration/deceleration control. The speed tracking control is designed to calculate the desired traction force of a vehicle to follow the desired vehicle speed V_{xd}. The active acceleration/deceleration control that follow the desired vehicle a_{xd} is used for im-

proving the vehicle cornering agility by integrating TDU and ESC.

(2) The lateral/yaw motion control use direct yaw moment for improving the handling and stability of the vehicle. The structure of the control system is a model-following controller that makes the vehicle track the desired dynamic model through the state feedback of both the yaw rate γ and the sideslip angle β.

(3) The rollover prevention control can be achieved by active differential braking or semi-active suspension to adjust the roll rate of vehicle ω_x. In practice, the three-dimensional coupled characteristics will complicate the vehicle dynamic control.

Considering all the longitudinal, lateral, and rollover coupling control targets, the criterion for minimizing the deviation between the outputs and the given references is usually chosen in a quadratic form. Therefore, we define the control objective as follows:

$$\min J = \|y_{\rm d}(k) - y(k)\|^2, \tag{9}$$

where $y_d = [V_{xd}, \alpha_{xd}, \omega_{xd}, \gamma_d, \beta_d]$ is the desired control reference.

Owing to actuator saturation, such as motor torque saturation the hard constraints of the control input $U(k)$ and the output $Y(k)$ related to the identified stability boundary have to be considered. They can be written as follows:

$$U(k) \in U_{\rm bound}, \tag{10}$$

$$Y(k) \in Y_{\rm bound}. \tag{11}$$

In summary, the synergy control method needs to solve the problem of multi-objective optimization with constraints. A variety of control methods such as LQR, MPC, among others, can be used for different control demands. From an engineering viewpoint, the control strategy should be easily implementable in the main control unit of the vehicle. The two test vehicles used in this study adopted the feedforward-feedback control strategy. It can allow for real-time operation with minimal load on the processing hardware.

Experiment. The control algorithm effectiveness was evaluated on two types of vehicle: one is an internal combustion engine drive vehicle (ICV) and the other is an electric vehicle with two in-wheel motors (EV).

For the ICV, a test track was designed on a low-friction road surface such as slalom, U-turn test on the snow-covered road. In the course of cornering, vehicle agility and stability can be properly controlled using the effects of the vertical load transfer and drive torque distribution exerted on the cornering forces.

For the EV with two in-wheel motors, with the vehicle control system architecture, the differential torque of the motor can improve the frequency response bandwidth of the vehicle, reduce the hysteresis, and significantly improve the transient steering characteristics of the vehicle.

Conclusion. In this article, we propose a synergy control framework for enlarging vehicle stability, which can make full use of the nonlinear and combined tire forces by identifying the self-stable boundary. By considering the coupling vehicle dynamics and the constraints of stability boundary, a controller for enlarging vehicle stability region is designed, and multiple control objectives are coordinated and optimized. Experimental results demonstrated that this method can be effectively used for vehicle stability control.

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Supporting information Videos and other supplemental documents. The supporting information is available online at info.scichina.com and link.springer.com. The supporting materials are published as submitted, without typesetting or editing. The responsibility for scientific accuracy and content remains entirely with the authors.

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