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Real-time Tire Parameters Observer for Vehicle Dynamics Stability Control

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Abstract: The performance of the vehicle dynamics stability control system(DSC) is dominated by the accurate estimation of tire forces in real-time. The characteristics of tire forces are determined by tire dynamic states and parameters, which vary in an obviously large scope along with different working conditions. Currently, there have been many methods based on the nonlinear observer to estimate the tire force and dynamic parameters, but they were only used in off-line analysis because of the computation complexity and the dynamics differences of four tires in the steering maneuver conditions were not considered properly. This paper develops a novel algorithm to observe tire parameters in real-time controller for DSC. The algorithm is based on the sensor-fusion technology with the signals of DSC sensors, and the tire parameters are estimated during a set of maneuver courses. The calibrated tire parameters in the control cycle are treated as the elementary states for vehicle dynamics observation, in which the errors between the calculated and the measured vehicle dynamics are used as the correcting factors for the tire parameter observing process. The test process with a given acceleration following a straight line is used to validate the estimation method of the longitudinal stiffness; while the test process with a given steering angle is used to validate the estimated value of the cornering stiffness. The ground test result shows that the proposed algorithm can estimate the tire stiffness accurately with an acceptable computation cost for real-time controller only using DSC sensor signal. The proposed algorithm can be an efficient algorithm for estimating the tire dynamic parameters in vehicle dynamics stability control system, and can be used to improve the robustness of the DSC controller.

Key words: tire, longitudinal stiffness, cornering stiffness, vehicle dynamics stability

1 Introduction

With the development of active control technology of vehicle chassis, accurate adjustment of the range and the distribution of lateral and longitudinal tire forces has become an important way to improve the dynamics stability. The intervention effects of the dynamics control system, such as anti-lock brake system(ABS), traction control system(TCS), and dynamics stability control(DSC), are determined by the tire and road friction[1]. VAN ZENTAN[2] firstly explained the DSC control logic based on the elementary tire force estimation logic, and HATTORI, et al[3], further developed vehicle dynamics management system based on the tire force control with nonlinear optimum distribution. Obviously, the accurate tire force estimation has become a key loop in the dynamics control system mentioned above. With considering the computation complexity and cost of tire force estimation method, the simplified-parameter tire model is used most frequently[4]. The tire parameters, especially longitudinal and cornering stiffness, determine the estimation accuracy of tire forces.

KIN[5], et al, estimated tire forces with tire data map, but the effects, which are influenced by the tire pressure, temperature, materials aging, and the tire trend wearing, cannot be compensated effectively. VAN ZENTAN[2] estimated the tire forces with simplified HSRI tire model. These dynamics states might be obtained from DSC control loop of real time controller[6]. If the tire parameters are observed in vehicle dynamics controller with DSC sensors, the accurate tire force estimation might be accomplished.

Several research groups have proposed various methods to observe the tire parameters for vehicle dynamics control. RAY[7] adopted the extended Kalman-Bucy filtering(EKBF) to obtain the tire parameters. After that, LEE, et al[8], used μ-slip relationship to estimate the friction, even when the tire worked at a large slip rate. PASTERKAMP, et al[9], estimated the tire forces with the neural network method. These methods may be considered in the effect-based $\mu_{\max}$ prediction. The dynamics differences among four wheels, caused by the individual active braking, different vertical loads, or uneven road friction under DSC control, cannot be considered. Meantime, these methods are only used in off-line analysis because of the computation complexity.

RYU[10] estimated the longitudinal stiffness, using the differential global positioning system(DGPS). He proposed a GPS-based real-time identification method of tire cornering stiffness. But in vehicle dynamics control, the DGPS cannot be used for its high cost. With considering

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that vehicle dynamics states estimation is an elementary component in DSC controller, the vehicle and tire dynamics states can be obtained from the control loop of DSC or measured with DSC sensors, such as wheel speed sensors, gyroscope sensor, and steer angle sensor[2]. The tire parameters observer may be integrated into DSC controller to realize a real-time adaptive adjustment of the nominal control model related to the tires.

In order to construct the observer for tire longitudinal stiffness and cornering stiffness, the longitudinal and lateral dynamics of four tires and the differences led by the load transfer are compensated in the paper. The basic rule is that the longitudinal stiffness is observed through the longitudinal dynamics transfer course according to the change of a certain tire slip rate. The cornering stiffness may be observed through a given steering maneuver with the combined slip excitation. The vehicle and tire dynamics states may be obtained from DSC controller; the observing logic for these related states might be described shortly in the paper. The related lateral vehicle and tire model are described in section 2. The longitudinal stiffness observer is described in section 3. The cornering stiffness is described in section 4. At last, the real vehicle test results are given in section 5.

2 Vehicle and Tire Model

A 7-DOF-4-wheel vehicle dynamic model (Fig. 1), including the longitudinal, lateral and yaw motions, and the rotations of four wheels, may reflect the load transfer effects and the dynamic characteristics of the individual wheel under active brake control. The model can describe the vehicle planar dynamics under steering. These related dynamics states may also be measured or estimated with DSC sensors. Thus, the model is adaptable for tire dynamics observer. The related model parameters are listed in Table 1.

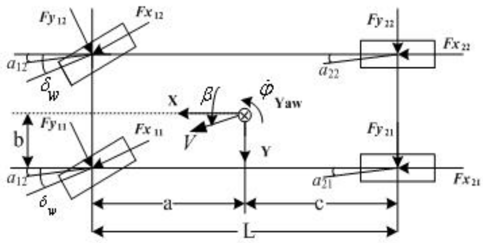

Fig. 1. 7-DOF-4-wheel vehicle dynamic model

Table 1. Related model parameters of the test vehicle

Parameter	Value
Vehicle mass m/kg	1 605
Distance between front and rear axle L/mm	2 790
Inertia moment about vertical axis of the vehicle $J_v/(\mathrm{N\cdot m^2})$	2 867
Distance from gravity center to front axle a/mm	1 110
Distance from gravity center to rear axle c/mm	1 680
Wheel base b/mm	782.5
Wheel inertia moment about rotary axis $J_w/(\mathrm{N\cdot m^2})$	1.084
Radus of wheel R/mm	306.5
Longitudinal stiffness $C_\lambda/(\mathrm{mN^{-1}})$	85
Cornering stiffness $C_\alpha/(\mathrm{kN\cdot rad^{-1}})$	43

The dynamics equation may be expressed as follows[1]:

$$m(\dot{v}_x - v_y\dot{\varphi}) = (F_{x11}+F_{x12})\cos\delta_w - (F_{y11}+F_{y12})\sin\delta_w + F_{x21} + F_{x22}, \tag{1}$$

$$m(\dot{v}_y + v_x\dot{\varphi}) = (F_{x11}+F_{x12})\sin\delta_w + (F_{y11}+F_{y12})\cos\delta_w + F_{y21} + F_{y22}, \tag{2}$$

$$J_v\ddot{\varphi} = (F_{y11}+F_{y12})a\cos\delta_w - (F_{y11}-F_{y12})b\sin\delta_w - (F_{y21}+F_{y22})c - (F_{x11}+F_{x12})a\sin\delta_w - (F_{x11}-F_{x12})b\cos\delta_w - (F_{x21}-F_{x22})b. \tag{3}$$

Equation of the wheel dynamics is

$$F_{x(ij)} = \frac{T_{w(ij)}}{R} - \frac{M_{calhalf}}{R} + \frac{J_{w(ij)}}{R}\cdot\frac{\mathrm{d}w_{whl(ij)}}{\mathrm{d}t}, \tag{4}$$

where ij (i, j=1, 2) represent different wheels. F_x and F_y are longitudinal and lateral tire forces, respectively. v_x and v_y are longitudinal and lateral velocity, respectively. $\dot{\varphi}$ is yaw rate of the vehicle. δ_w is steer angle of the front wheel. T_w is brake torque of the wheel. $M_{calhalf}$ is driven torque, which can be obtained from engine management system. w_{whl} is wheel angular velocity.

VAN ZENTAN[2] brought a tire force estimation logic in DSC with Dugoff tire model, which could describe the nonlinear friction performance with a simple relationship of tire dynamic states for real time control. And the lateral tire force might be easily deduced from the longitudinal tire force, based on the relationship described with Dugoff tire model. Therefore, Dugoff tire model is a proper tire model in vehicle dynamics control system[2]. The model is expressed as follows:

$$F_x = \begin{cases} \dfrac{\lambda}{1-\lambda}\cdot C_\lambda, & H<\dfrac{1}{2}, \\ \dfrac{\lambda}{1-\lambda}\cdot C_\lambda\cdot\left(\dfrac{1}{H}-\dfrac{1}{4H^2}\right), & H\geq\dfrac{1}{2}, \end{cases} \tag{5}$$

$$F_y = \begin{cases} \dfrac{1}{1-\lambda}\cdot C_\alpha\cdot\tan\alpha & H<\dfrac{1}{2}, \\ \dfrac{1}{1-\lambda}\cdot C_\alpha\cdot\left(\dfrac{1}{H}-\dfrac{1}{4H^2}\right)\cdot\tan\alpha, & H\geq\dfrac{1}{2}, \end{cases} \tag{6}$$

where H is the combined slip parameter. C_λ and C_α are tire longitudinal stiffness and cornering stiffness, respectively. λ and α are tire slip rate and slip angle, respectively.

3 Tire Longitudinal Stiffness Observer

Dugoff tire model is used in the longitudinal stiffness observer. For the accuracy of the estimated results, the tire forces might be defined in the linear area of tire-road friction curve. CARLSON, et al[11], once proposed a longitudinal stiffness observer, in which the tire radius and the stiffness were estimated at the same time with the supposals that the dynamics of left and the right wheels were the same. But in real-time observer, the wheel radius might be measured easily, and the wheel speed pulses and the noises might differ obviously in different road conditions. So the four wheel dynamics must be included in the observer, when the radius of four wheels and the tire types are treated to be the same.

If the slip rate of a wheel is less than 5%, the longitudinal tire force may be expressed as a linear relationship with slip rate as follows:

$$F_{xij} = C_{\lambda ij}\frac{\lambda_{ij}}{1-\lambda_{ij}} \cong C_{\lambda ij}\lambda_{ij}. \tag{7}$$

Define the slip rate of the driven wheel in driving conditions as follows:

$$\lambda_{ij} = -\frac{v_x - Rw_{fij}}{v_x}. \tag{8}$$

When the vehicle runs straightly, the longitudinal tire stiffness may be observed individually. If the vehicle runs at a low speed, the aerodynamics and rolling resistance may be neglected. Therefore, the longitudinal movement equation of the vehicle on the planar ground may be simplified from Eq. (1):

$$m\dot{v}_x = \sum_{i,j=1}^{2} F_{xij}. \tag{9}$$

Supposing that the longitudinal tire stiffness of the front wheels are the same, only the driven force of the front wheels is considered. The longitudinal equation, based on Eqs. (7)–(9), may be expressed as follows:

$$\hat{\dot{v}}_x = \begin{pmatrix} -\dfrac{1}{m} & \dfrac{\hat{w}_{\mathrm{wh11}}+\hat{w}_{\mathrm{wh12}}}{m\hat{v}_x}\end{pmatrix}\begin{pmatrix}2C_{\lambda_f}\\ RC_{\lambda_f}\end{pmatrix}, \tag{10}$$

where sign ^ means the measured or observed state. The wheel speed sensors may measure the rotary angle of the wheel θ_u. Then $\hat{w}_{\mathrm{wh}(ij)} = \hat{\dot{\theta}}_u$. Thus,

$$\begin{cases} \hat{v}_x = \dfrac{1}{4}R\sum\limits_{i,j=1}^{2}\hat{w}_{\mathrm{wh}(ij)} = \dfrac{1}{4}R\sum\limits_{i,j=1}^{2}\hat{\dot{\theta}}_{u(ij)}, \\ \hat{\dot{v}}_x = \dfrac{1}{4}R\sum\limits_{i,j=1}^{2}\hat{\ddot{\theta}}_{u(ij)}. \end{cases} \tag{11}$$

In DSC system, the controller may capture the rotary angle movement with wheel speed pulse and calculate the wheel speed at the time interval T, which is about 10 ms in general. Then:

$$\begin{cases} \hat{\dot{\theta}}_u^k = \dfrac{\hat{\theta}_u^{k+2}-\hat{\theta}_u^k}{2T}, \\ \hat{\ddot{\theta}}_u^k = \dfrac{\hat{\theta}_u^{k+2}-2\hat{\theta}_u^{k+1}+\hat{\theta}_u^k}{2T^2}, \end{cases} \tag{12}$$

At the instant k, the errors of vehicle velocity and longitudinal acceleration of the vehicle may be included in the estimated value. Thus,

$$\begin{cases} \hat{\theta}_u^k = \theta_u^k + \Delta\theta_u^k,\ \hat{v}_x^k = v_x^k + \Delta v_x^k, \\ \hat{a}_x^k = \hat{\dot{v}}_x^k = \dfrac{v_x^{k+2}-v_x^k}{2T} = a_x^k + \Delta a_x^k, \end{cases} \tag{13}$$

From Eqs. (10)–(13), the longitudinal equation may be expressed as follows:

$$a_x - \begin{pmatrix} -\dfrac{1}{m} & \dfrac{\hat{\dot{\theta}}_{u11}+\hat{\dot{\theta}}_{u12}}{mv_x}\end{pmatrix}\begin{pmatrix}2C_\lambda\\ R_fC_\lambda\end{pmatrix} = -\Delta a_x + \begin{pmatrix} 0 & \dfrac{v_x(\Delta\hat{\dot{\theta}}_{u11}+\Delta\hat{\dot{\theta}}_{u12})-(\hat{\dot{\theta}}_{u11}+\hat{\dot{\theta}}_{u12})\Delta v_x}{m(v_x+\Delta v_x)v_x}\end{pmatrix}\begin{pmatrix}2C_\lambda\\ R_fC_\lambda\end{pmatrix}. \tag{14}$$

The above equation is a linear observer for longitudinal stiffness, and the least squares method can be used to calculate C_λ and R_f. The multiplicative item of errors may be expressed as follows:

$$\frac{v_x(\Delta\hat{\dot{\theta}}_{u11}+\Delta\hat{\dot{\theta}}_{u12})-(\hat{\dot{\theta}}_{u11}+\hat{\dot{\theta}}_{u12})\Delta v_x}{m(v_x+\Delta v_x)v_x},$$

which tends to bias the parameter estimations. In order to conquer the errors, v_x may be deduced only by the two rear free wheels. Thus,

$$\begin{cases} \hat{\theta}_u^k = \theta_u^k + \Delta\theta_u^k, \\ \hat{v}_x^k \cong R_r\dfrac{\hat{w}_{21}+\hat{w}_{22}}{2} = R_r\dfrac{(\dot{\theta}_{u21}+\Delta\dot{\theta}_{u21}+\dot{\theta}_{u22}+\Delta\dot{\theta}_{u22})}{2}, \\ \hat{a}_x^k = \hat{\dot{v}}_x \cong R_r\dfrac{\hat{\dot{w}}_{21}+\hat{\dot{w}}_{22}}{2} = R_r\dfrac{(\ddot{\theta}_{u21}+\Delta\ddot{\theta}_{u21}+\ddot{\theta}_{u22}+\Delta\ddot{\theta}_{u22})}{2}. \end{cases} \tag{15}$$

From Eq. (10), the equation may be transferred into:

$$f = m\dot{v}_x - \sum_{i,j=1}^{2}F_{xij} = m\dot{v}_x - \left(C_{\lambda_f}\frac{v_x - R_fw_{11}}{v_x} + C_{\lambda_f}\frac{v_x - R_fw_{12}}{v_x}\right) = 0. \tag{16}$$

Eq. (16) is multiplied by v_x, and substitute v_x and $\dot{v}_x$ with Eq. (15), then we have:

$$
\begin{aligned}
f' = m\frac{R_r^2}{4}(\ddot{\theta}_{u21} + \Delta\ddot{\theta}_{u21} + \ddot{\theta}_{u22} + \Delta\ddot{\theta}_{u22}) \times \\
(\dot{\theta}_{u21} + \Delta\dot{\theta}_{u21} + \dot{\theta}_{u22} + \Delta\dot{\theta}_{u22}) - C_{\lambda_f} \times \\
[R_r(\dot{\theta}_{u21} + \Delta\dot{\theta}_{u21} + \dot{\theta}_{u22} + \Delta\dot{\theta}_{u22}) - \\
R_f(\dot{\theta}_{u11} + \Delta\dot{\theta}_{u11} + \dot{\theta}_{u12} + \Delta\dot{\theta}_{u12})] = 0.
\end{aligned} \tag{17}
$$

In the real tests, the radius of the tire can almost keep at a constant value with small variation[10]. R_r and R_f can be treated as constant values to decrease the computation complexity. Then, at the instant k, Eq. (17) may be expressed as the following convenient form:

$$
\begin{aligned}
f'^k(\hat{\theta}_{u11}, \hat{\theta}_{u12}, \hat{\theta}_{u21}, \hat{\theta}_{u22}, \Delta\theta_{u11}, \Delta\theta_{u12}, \\
\Delta\theta_{u21}, \Delta\theta_{u22}, C_{\lambda_f}) = 0.
\end{aligned} \tag{18}
$$

The errors of related measured signals from DSC sensors, such as wheel speeds and the estimated longitudinal velocity, might be treated as independent zero mean(IZM) noise[11]. In order to minimize the sum of squared measurement errors, the question may be transferred into finding the correct parameters with the measured IZM noise. Then Eq. (18) can be expressed as follows:

$$
\begin{cases}
\min \|\Delta\theta_{u11}; \Delta\theta_{u12}; \Delta\theta_{u21}; \Delta\theta_{u22}\|, \\
\text{s.t.} \quad f^k(\hat{\theta}_{u11}, \hat{\theta}_{u12}, \hat{\theta}_{u21}, \hat{\theta}_{u22}, \Delta\theta_{u11}, \Delta\theta_{u12}, \\
\qquad \Delta\theta_{u21}, \Delta\theta_{u22}, C_{\lambda_f}) = 0.
\end{cases} \tag{19}
$$

In order to reduce computation complexity in a real-time controller, the observer may be divided into two cascade observers. Firstly, v_x is estimated, and then the longitudinal stiffness is estimated. The initial values are given in a common scope: $C_{\lambda\min} \le C_{\lambda_f} \le C_{\lambda\max}$. Eq. (19) may be simplified as follows:

$$
\begin{cases}
\min \|\Delta\theta_{u11}; \Delta\theta_{u12}\|, \\
\text{s.t.} \quad f^k(\hat{\theta}_{u11}, \hat{\theta}_{u12}, \Delta\theta_{u11}, \Delta\theta_{u12}, C_{\lambda_f}) = 0 \ \& \\
\qquad C_{\lambda\min} \le C_{\lambda_f} \le C_{\lambda\max}.
\end{cases} \tag{20}
$$

The observer can be integrated with DSC control algorithm. If the controller obtains the command to observe the longitudinal stiffness, the driver might be informed to drive the vehicle in a straight line with a moderate acceleration and deceleration maneuver. The controller can judge whether the vehicle is in a straight line or not, based on the measured steering angles. If it is satisfied, the controller may store the signals of the wheel speed, longitudinal velocity, and acceleration for a given time. Then C_λ is calculated. The observer may be illustrated as Fig. 2.

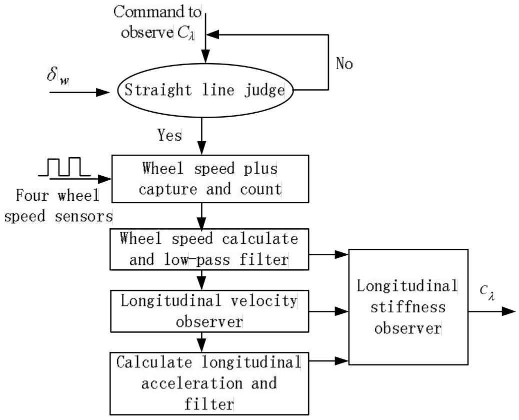

Fig. 2. Sketch of tire longitudinal stiffness observer

4 Tire Cornering Stiffness Observer

The cornering stiffness might be observed in a steering course with free coasting. The driven forces of the front wheels are treated as 0. If the steer angle of the front wheel is small, the dynamics equation of the vehicle may be deduced from Eqs. (2)–(3). Then only cornering stiffness of the front wheels needs to be estimated:

$$
mc(\dot{v}_y + v_x\dot{\varphi}) + J_v\ddot{\varphi} = L(F_{y11} + F_{y12}) + (F_{y12} - F_{y11})b\delta_w. \tag{21}
$$

If the slip angle is less than 5°, the linear lateral tire force may be expressed with HSRI tire model, and the slip angles of two front wheels are the same[2]. Eq. (21) may be simplified in a further step as follows:

$$
\dot{v}_y + v_x\dot{\varphi} + \frac{J_v\ddot{\varphi}}{mc} = \frac{2L}{mc}C_{\alpha_f}\alpha_f. \tag{22}
$$

We can define that:

$$
\alpha_f = \delta_w - \left(\beta + \frac{a\dot{\varphi}}{v_x}\right), \ \dot{v}_y = a_y - v_x\dot{\varphi}, \ \beta = \frac{v_y}{v_x}. \tag{23}
$$

If the integration method is used to calculate the slip angle, the accumulated errors might increase sharply as long as the integration time is large. Therefore, the attenuation coefficient τ ($\tau \ge 1$) is used to compress the error[1]. Then:

$$
\hat{\dot{v}}_y^k = \hat{a}_{y\_f}^k - v_{\text{ref}}^k \cdot \dot{\varphi}_f^k - \frac{\hat{v}_y^k}{\tau}. \tag{24}
$$

The trapezoidal integration is used:

$$
\hat{v}_y^{k+1} = \hat{v}_y^k + \left(\hat{a}_{y\_f}^k - v_{\text{ref}}^k \cdot \dot{\varphi}_f^k - \frac{\hat{v}_y^k}{\tau}\right)T. \tag{25}
$$

Considering the measured errors of steer angle, lateral acceleration, yaw rate, and the wheel speeds, Eq. (22) may be expressed as follows:

$$S=\left[(a_y+\Delta a_y)+\frac{1}{2}R_r(\dot{\theta}_{u21}+\Delta\dot{\theta}_{21}+\dot{\theta}_{u22}+\Delta\dot{\theta}_{22})\times (\dot{\varphi}+\Delta\dot{\varphi})+\frac{J_{\mathrm{v}}}{mc}(\ddot{\varphi}+\Delta\ddot{\varphi})\right]-\frac{2L}{mc}C_{\alpha_f}\times \left\{\delta_{\mathrm{w}}+\Delta\delta_{\mathrm{w}}-\left[\beta+\frac{2a(\dot{\varphi}+\Delta\dot{\varphi})}{R_r(\dot{\theta}_{u21}+\Delta\dot{\theta}_{21}+\dot{\theta}_{u22}+\Delta\dot{\theta}_{22})}\right]\right\}=0. \tag{26}$$

v_x is calculated from wheel speed signals of free rolling wheels. If the vehicle is in a steering course with free coasting, v_x may be treated as an independent parameter for cornering stiffness observing. Therefore, the cornering stiffness observer is a cascaded observer. Firstly, v_y (or slip angle) is observed, and then the cornering stiffness is observed. Eq. (26) may be simplified as follows:

$$S=\left[(a_y+\Delta a_y)+v_x(\dot{\varphi}+\Delta\dot{\varphi})+\frac{J_{\mathrm{v}}}{mc}(\ddot{\varphi}+\Delta\ddot{\varphi})\right]-\frac{2L}{mc}C_{\alpha_f}\left\{\delta_{\mathrm{w}}+\Delta\delta_{\mathrm{w}}-\left[\beta+\frac{a(\dot{\varphi}+\Delta\dot{\varphi})}{v_x}\right]\right\}=0. \tag{27}$$

At the instant k, Eq. (27) may be expressed as convenient form:

$$S^k(\hat{a}_y,\hat{\dot{\varphi}},\hat{\delta},\Delta a_y,\Delta\dot{\varphi},\Delta\delta,C_{\alpha_f})=0. \tag{28}$$

In order to decrease the computation complexity, the approximate initial value is given in around the common scope of the tire stiffness: $C_{\alpha\min}\leq C_{\alpha_f}\leq C_{\alpha\max}$. Then the nonlinear estimation equation is as follows:

$$\begin{cases}\min\ \left\|\Delta a_y,\Delta\dot{\varphi}\right\|,\\ \text{s.t.}\ \ S^k(\hat{a}_y,\hat{\dot{\varphi}},\hat{\delta},\Delta a_y,\Delta\dot{\varphi},\Delta\delta,C_{\alpha_f})=0\ \ \&\\ \qquad C_{\alpha\min}\leq C_{\alpha_f}\leq C_{\alpha\max}.\end{cases} \tag{29}$$

The observer can be illustrated as Fig. 3.

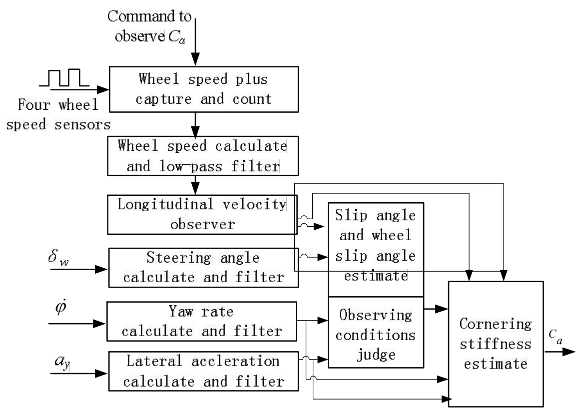

Fig. 3. Sketch of tire cornering stiffness observer

The observer may be integrated with DSC control algorithm. If the controller obtains the command to observe the cornering stiffness, the driver needs to drive the vehicle with a moderate steering angle; the longitudinal velocity must fit to the steering angle for the control of the lateral acceleration. Then the tire can work in the linear area in the lateral direction. The controller can judge whether the vehicle is in an adaptable way or not, based on the measured steering angles, yaw rate, and lateral acceleration. If the condition is satisfied, the controller may store these related values for a given time. Then the observer begins to calculate C_α.

5 On-line Test Force

The observers are integrated into DSC controller, and a subroutine is used to calibrate the tire stiffness, when the driver handles the vehicle with a given maneuver.

5.1 Longitudinal stiffness validation

Firstly, the tire longitudinal stiffness is observed in a test maneuver process. The vehicle is accelerated in a straight line and a_x is in the scope of 0–3 m/s^2. The typical data set is illustrated in Fig. 4. There are 2 acceleration-deceleration cycles in the test. The DSC sensors measure the four wheel rotary angle. Then longitudinal velocity and acceleration may be deduced with the finite differences method. Then the longitudinal stiffness is estimated by Eq. (20).

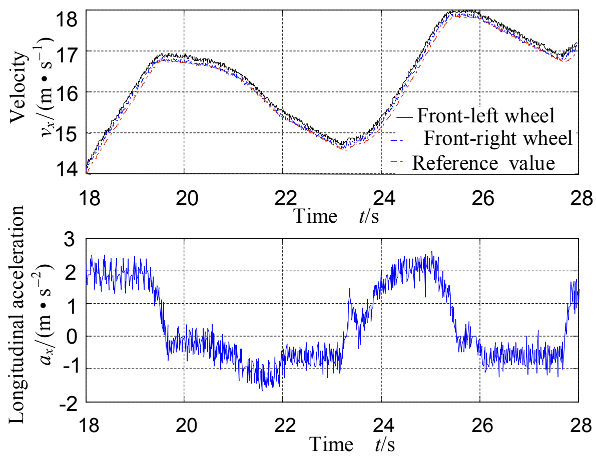

Fig. 4. Typical data set for C_{λ_f} observing

As illustrated in Fig. 5, the longitudinal stiffness is estimated based on linear and nonlinear observers respectively. The repeated tests with different initial values are conducted; the estimated values are listed in Table 2. The nonlinear observer is more accurate, and the nitration number is smaller than the linear one. The type of tire is Michelin MXV8-205/55R16-91V. The vertical load is about 4 120 N. The parameters of the vehicle and tire are provided by Brilliance Auto Co. and Michelin.

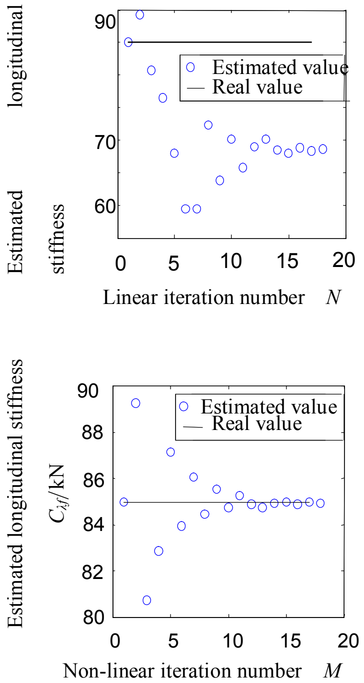

Fig. 5. Estimated longitudinal stiffness by linear or non-linear method

Table 2. Estimated value of longitudinal stiffness

No.	Longitudinal stiffness C_{λ_f} /kN				Tire pressure p/MPa
	Initial value	Linear method	Non-linear method	Real value	
1	85	82	82		
2	65	52	94	85	0.2
3	55	65	89		
Max error e/%		38	9.5		

5.2 Cornering stiffness validation

In order to observe the cornering stiffness, the test is set as follows.

(1) The steering angle inputs are fixed and the vehicle runs on a circle. The radius is about 16 m.

(2) The steering angle, yaw rate, lateral acceleration, and wheel speed are measured by DSC sensors.

(3) The slip angle of the vehicle is estimated with Eq. (23) and Eq. (25) [2,4]. The estimation logic is also integrated into DSC controller.

(4)The observer collects the typical data set and calculates the cornering stiffness.

The test data set are illustrated in Fig. 6. And the estimated cornering stiffness is illustrated in Fig. 7.

The estimated values from the three tests of the nonlinear observer are listed in Table 3.

Table 3. Estimated value of cornering stiffness (δ_w=0.16 rad)

No.	Cornering stiffness C_{α_f} /(kN • rad^{-1})			Tire pressure p/MPa
	Initial value	Estimated value	Real value	
1	42	38		
2	32	49	43	0.2
3	22	41		
Max error e/%		14		

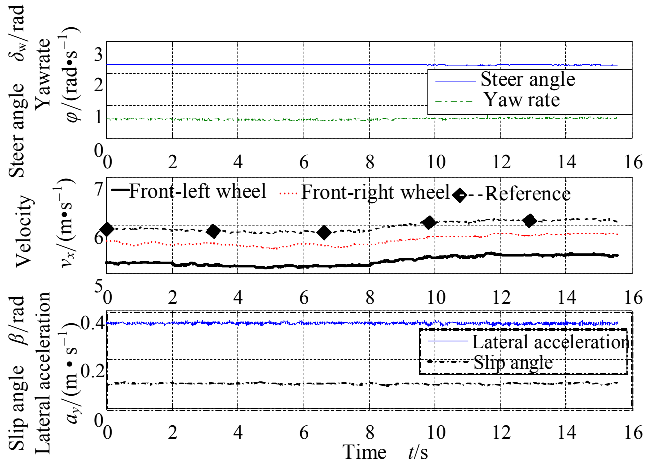

Fig. 6. Test data set for cornering stiffness observing

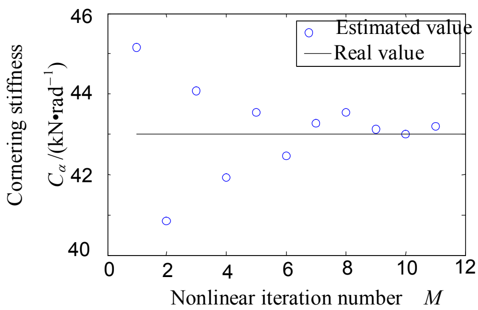

Fig. 7. Estimated cornering stiffness

Because the vehicle is stable and DSC controller is not activated in the test, the computation ability is enough to calculate the tire stiffness. The main chip of the controller is Infineon XC2000 and the control cycle is 40 ms. The computation time of the observer is about 9 ms. The estimation process of the longitudinal stiffness might be finished in 10 s and the cornering stiffness estimation process might be finished in 15 s. Therefore, the whole tire parameters estimation process might be easily realized in the real time controller of DSC.

6 Conclusions

(1) With the measured signals of DSC sensors and some related vehicle dynamics states obtained directly from DSC controller, the tire stiffness parameters can be estimated during a given calibration maneuver course with the proposed observer.

(2) The computation complexities of the observers for the tire longitudinal and cornering stiffness were validated with the real vehicle tests. The results show that the estimated algorithm can be used in real-time controller.

(3) The calibration maneuver course is simple and the controller can easily activate the estimated algorithm, when the vehicle runs in some ordinary driving conditions.

(4) The accuracy of the estimated parameters of the tire is independent on the vehicle and tire model. The method might be integrated into the control algorithm of DSC to improve the robustness performance.

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