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Integrated Control for Four-Wheel-Independent-Drive EVs' Lateral Stability and Rollover Prevention

Yuye Hou(✉), Lu Xiong, Bo Leng, and Zhuoping Yu

School of Automotive Studies, Tongji University, Shanghai 201804, China
1510814@tongji.edu.cn

Abstract. In order to improve the maneuverability of Four-Wheel-Independent-Drive (4WID) EVs with high center of gravity while guaranteeing rollover stability, a coordinated stability control system is proposed to maximize driving velocity and enhance vehicle stability in cornering. A nonlinear vehicle model is used in the supervisor controller to determine the control target and then model predictive control (MPC) is designed to mitigate the delay effect of vehicle dynamics and also take the combined slip effect into account. Numerical simulations have been conducted to evaluate the proposed stability control system, which show that vehicle maneuverability, lateral stability and rollover mitigation performance can be significantly improved.

Keywords: Coordinated stability control · MPC · Combined slip

1 Introduction

Researches about active chassis control for vehicle stability improvement have made remarkable progress in recent years. With the rapid development of 4WID EVs, torque-vectoring-control (TVC) becomes much more favorable to expand the stability domain of EVs. Most of the TVC systems adopt yaw moment control to adjust the steering characteristic. Nevertheless, there is still need for vehicles with high center of gravity, like SUVs, to achieve rollover prevention. Several studies have investigated the driving control algorithm for improved maneuverability, lateral stability, and rollover prevention. Yi et al. proposed an integrated chassis control to track the target steering response through differential braking which is limited by rollover prevention [1]. Gordon considered a novel approach assigning the braking torque of each wheel to fulfill the desired acceleration [2]. Alberding solved the nonlinear control allocation problem by introducing rollover prevention as a constraint [3]. These algorithms adopted the bicycle model and roll dynamic with single degree of freedom, which ignores the coupling relationship between steering and rollover. When vehicles come across rollover risk, the combined slip effect between the longitudinal force and lateral force becomes more significant.

Time delays exist between the response of the dynamic yaw rate and the roll angle of a vehicle, making it very necessary to take prevention measures before sideslip or

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rollover actually happens. In order to reduce the adverse effect of time delay, MPC has been widely used. Li adopted MPC to calculate the desired tire forces of four wheels under the constraints of the given controllable area [4]. Rajamani predicted the load transfer ratio (LTR) according to the state of vehicle and driver's input, and took steps in time to improve the driving capability of vehicles [5]. A switching MPC controller was proposed to track the desired path while limiting the danger of rollover by braking and rear steering at the same time [6]. However, in order to reduce the calculation burden of the optimization, the model used in MPC is simplified without considering the change of steering angle and lateral force, which might induce accuracy loss for the controller in some critical situations.

Based on those above, the coordinated stability control system in this paper adopts a nonlinear vehicle model to determine the control reference value. Considering the roll dynamic and the tire coupling gives a better prediction about the vehicle dynamic. Finally, a stability controller is proposed to improve the maneuverability, lateral stability and rollover mitigation performance.

2 Vehicle Dynamic Model

A 4-DOF model illustrated in Fig. 1 gives the dynamic characteristic of the vehicle, which includes the longitudinal motion, lateral motion, yaw motion, and roll motion.

$$m(\dot{u} - v\dot{\psi}) = (F_{xfl} + F_{xfr})\cos\delta_f + F_{xrl} + F_{xrr} - (F_{yfl} + F_{yfr})\sin\delta_f - F_{air} \tag{1}$$

$$m(\dot{v} + u\dot{\psi}) = (F_{xfl} + F_{xfr})\sin\delta_f + (F_{yfl} + F_{yfr})\cos\delta_f + F_{yrl} + F_{yrr} \tag{2}$$

$$I_z\ddot{\psi} = [F_{yf}l_f - \Delta F_{xf}\frac{t_f}{2}]\cos\delta_f - F_{yr}l_r - \Delta F_{xr}\frac{t_r}{2} + [(F_{yfl} - F_{yfr})\frac{t_f}{2} + F_{xf}l_f]\sin\delta_f + \Sigma M_z \tag{3}$$

$$I_x\ddot{\phi} = [F_{xf}\sin\delta_f + F_{yf}\cos\delta_f + F_{yr}]\cos\phi \cdot h + m\cdot g\cdot h\cdot \sin\phi - K\cdot\phi - C\cdot\dot{\phi} \tag{4}$$

A five-state space model is formulated as:

$$\dot{x} = f(x) + g\cdot u \tag{5}$$

Where $x = [u, v, \dot{\psi}, \phi, \dot{\phi}];\ u = [F_{xfl}, F_{xfr}, F_{xrl}, F_{xrr}]^T$. Table 1 gives the symbols and parameters in the simulation model.

Table 1. Parameters in the simulation model

Parameter	Symbol/value	Parameter	Symbol/value
Vehicle mass	m/2402 kg	Inertia of yaw	I_z/3864 kg m^2
Inertia of roll	I_y/3738 kg m^2	Wheel-base (l_f/l_r)	1.20 m/1.46 m
Wheel track	(t_f/t_r)/1.56 m	Height from the center of gravity to roll center h	h/0.538 m
Roll stiffness	K/2530 Nm/°	Roll damper	C/180 Nms/°
Longitudinal velocity	u	Lateral velocity	v
Yaw angular velocity	$\dot{\psi}$	Roll angle	ϕ
Longitudinal force	F_x	Lateral force	F_y
Wheel align moment	M_z	Front steering angle	δ

When vehicles come to critical situations, the tires are usually in the non-linear regions. The Magic Formula (MF) tire model with the combined slip theory is used to describe the dynamics of tires. The general form of MF can be expressed as:

$$F_0(x) = D \sin\{C \arctan[B \cdot x - E(B \cdot x - \arctan(B \cdot x))]\} \tag{6}$$

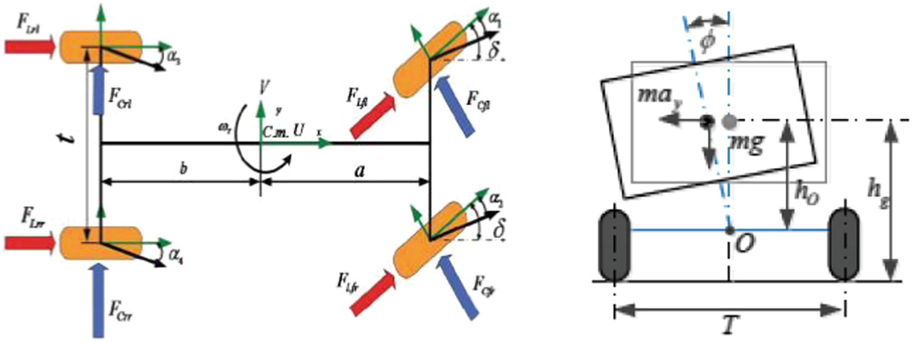

Fig. 1. Vehicle model

And the coupling effect between longitudinal force and lateral force is approximated through the combined slip theory [7]:

$$\begin{aligned} F_x &= f_x[\lambda, \alpha, \mu, F_z, F_{x0}(\lambda), F_{y0}(\alpha)] \\ F_y &= f_y[\lambda, \alpha, \mu, F_z, F_{x0}(\lambda), F_{y0}(\alpha)] \end{aligned} \tag{7}$$

3 Control Structure

The diagram of the controller is shown in Fig. 2, which consists of 3 parts. The first part decides the motion following reference according to the driver's input and the stable region. The normal steering response is obtained according to the predefined understeer characteristic. And the prediction model will calculate the vehicle's state afterwards based on the vehicle model (1–7) to determine the control mode and adjust the desired value.

Then the nonlinear model is linearized. And the limits of the control output are also included in the second part. Tire forces play a significant role in vehicle dynamics. In this paper, combined slip effect and suspension K&C characteristic have been taken into account to improve tire forces and vertical loads estimation accuracy.

Finally, the MPC controller calculates the desired tire forces based on the actuator's ability and the predicted states.

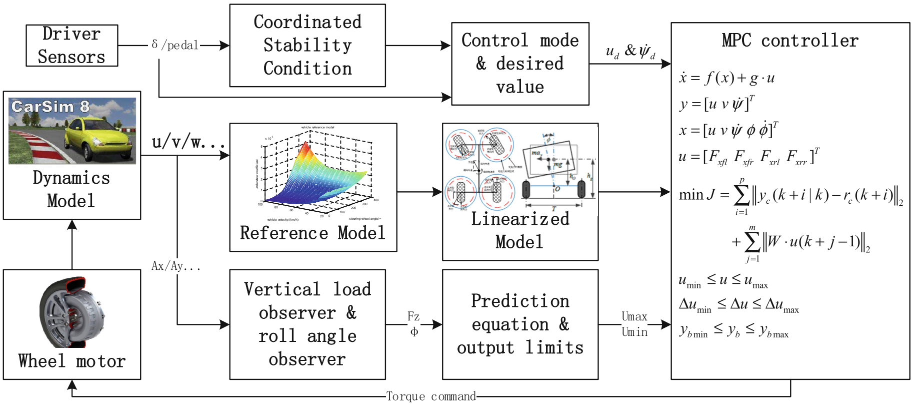

Fig. 2. Control structure schematic diagram

3.1 Control Reference Decision

Normal Drive. To reduce the understeer gradient with respect to the passive vehicle and extend the region of linear cornering response, the steady yaw rate of 2 DOF vehicle model has been widely used as the desired yaw rate. However, it may cause too much additional yaw moment which could be hardly achieved when it comes to large lateral acceleration. Considering the constraint of tire road adhesion coefficient, the quadratic understeer coefficient is adopted here as the reference model.

$$US = \left| \frac{\alpha_f - \alpha_r}{a_y} \right| = \left| \frac{\frac{l\dot{\psi}}{u} - \delta_f}{a_y} \right| = C \cdot a_y \tag{8}$$

$$\dot{\psi}_d = \frac{\min(\left|a_{yref}\right|, \mu g)}{u} \cdot \mathrm{sgn}(\delta_f) = \min\left(\frac{\sqrt{\left(\frac{l}{u^2}\right)^2 + 4C\delta_f} - \frac{l}{u^2}}{2C \cdot u}, \frac{\mu g}{u}\right) \cdot \mathrm{sgn}(\delta_f) \tag{9}$$

And the desired velocity is decided by driver's intention.

$$u_d(k+i|k) = u(k) + \frac{F_{xd} - F_{air} - M_{yRR}/r}{m} \cdot \Delta T \tag{10}$$

Where, F_{xd} is the desired drive force, F_{air} is the air resistance, M_{yRR} is the rolling resistance, C is the desired understeer coefficient and r is effective radius.

Lateral Stability Control. In case of drift or other critical situations, it is necessary to limit the yaw rate to prevent the vehicle from losing control. Rear slip angle can reflect the vehicle's stability directly, so the stead-state value of reference yaw rate is given by:

$$\dot{\psi}_{ref} = (1 - k_\beta)\dot{\psi}_d + k_\beta \dot{\psi}_s \tag{11}$$

$\dot{\psi}_s$ is the stability yaw rate, i.e., a yaw rate value that is compatible with current cornering conditions of the vehicle, corresponding to the measured lateral acceleration [9]. The weighting factor, k_β, is a linear function of the absolute value of the rear side slip angle, and is saturated between 0 and 1:

$$k_\beta = \begin{cases} 0 & |\beta_r| \le \beta_{act} \\ \frac{|\beta_r| - \beta_{act}}{\beta_{\lim it} - \beta_{act}} & \beta_{act} < |\beta_r| < \beta_{\lim} \\ 1 & |\beta_r| \ge \beta_{\lim} \end{cases} \tag{12}$$

$$\dot{\psi}_s = k_{\gamma,s} \frac{\min(|a_y|, 0.85\mu g)}{u} \cdot \mathrm{sgn}(\delta_f) \qquad 0 < k_{\gamma,s} < 1 \tag{13}$$

In lateral stability control mode, yaw rate takes priority over velocity. Thus, the reference velocity should be modified as:

$$u_{ref} = \min(u_d, \left|\frac{a_y}{\dot{\psi}_{ref}}\right|) \qquad |\beta_r| > \beta_{\lim} \tag{14}$$

Rollover Prevention Control. The lateral load transfer ratio (*LTR*) is adopted to represent the rollover risk. Considering the delay between roll motion and *LTR*, the predicted load transfer ratio (LTR_p) is used to calculate the time to rollover and decide the time to intervene. After m times iteration, the roll angle and roll rate at time k to $k + m$ can be obtained. The calculation of $LTR_p(k + m)$ is:

$$LTR_p(k+m) = 2 \cdot (K \cdot \phi(k+m) + C \cdot \dot{\phi}(K+m))/(mg \cdot t) \tag{15}$$

Rollover prevention control mode is activated according to Fig. 3. Once the rollover risk exceeds the preset threshold value, the reference yaw rate and velocity has to be modified accordingly to guarantee the safety.

$$LTR_d = \min\left\{\left||2(K\phi_d + C\dot{\phi}_d)/mg \cdot t\right| - k_1|\phi - \phi_s| - k_2|\dot{\phi} - \dot{\phi}_s||, LTR_{\max}\right\} \quad (16)$$

$$a_{yd} = \min\left(\left|\frac{mg \cdot t \cdot LTR_d - 2mg \cdot h \cdot \sin\phi_s}{2m \cdot h \cdot \cos\phi_s}\right|, |a_y|, 0.85\mu g\right) \quad (17)$$

$$u_{dr} = u + \left(\frac{a_{yd}}{|\dot{\psi}|} - u\right)\frac{1}{\tau s + 1} \quad (18)$$

$$\dot{\psi}_{sr} = \frac{a_{yd}}{u}\operatorname{sgn}(\delta) \quad (19)$$

The nonlinear vehicle dynamic model (8) is used to predict the states of vehicle based on the current states and inputs. The inputs refer to the 4 wheel torques and steering wheel angle, which is decided by the drivers and predicted with a quadratic polynomial extrapolation method. Some of the states are obtained directly from sensors such as $\dot{\psi}, a_y, a_x$, while roll angle is converted through the suspension displacement. The other vehicle's state can be estimated [8].

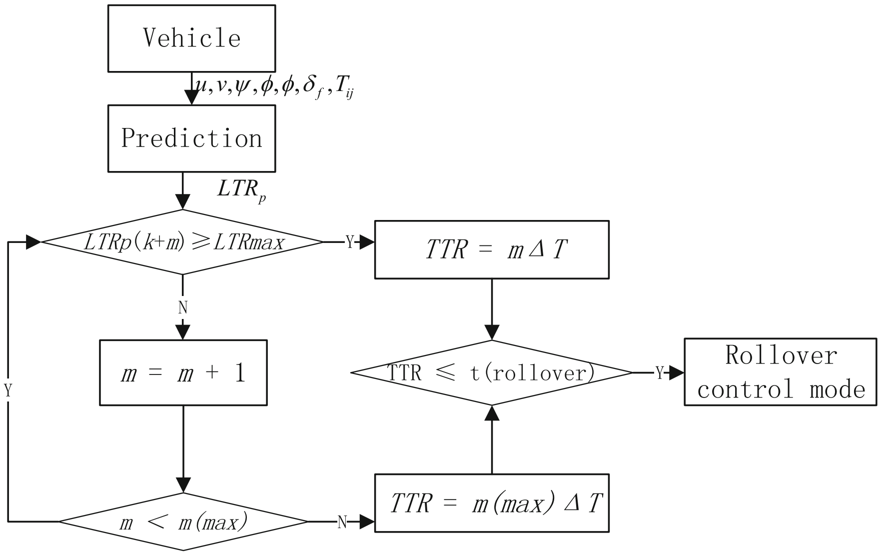

Fig. 3. Rollover control mode decision

3.2 Tire Force Estimation

Thanks to in-wheel motor drive torque feedback, the longitudinal force can be estimated easily.

$$F_x = \frac{T_q - M_{yRR} - I_{yr} \cdot \dot{\omega}}{r} \tag{20}$$

With the combined slip theory, the lateral force estimation precision is improved. And the vertical force can is calculated with the help of suspension sensors.

$$\begin{aligned} K_s S_l + K_\psi (S_l - S_r) + \frac{\partial F_z}{\partial F_x} \cdot F_{xl} + \frac{\partial F_z}{\partial F_y} \cdot F_{yl} = K_t (SL - S_l) \\ K_s S_r + K_\psi (S_r - S_l) + \frac{\partial F_z}{\partial F_x} \cdot F_{xr} - \frac{\partial F_z}{\partial F_y} \cdot F_{yr} = K_t (SR - S_r) \end{aligned} \tag{21}$$

$$F_{zi} = F_{zi0} - K_t (Sij - S_{ij}) \tag{22}$$

3.3 Linearization and Limitation

The discrete and incremental form of model is:

$$x(k+1) = (1 + \frac{\partial f}{\partial x} \cdot \Delta T) x(k) + g \cdot \Delta T \cdot u(k) \tag{23}$$

The controlled output is defined as velocity and yaw rate.

$$y_c = \begin{bmatrix} u & \dot{\psi} \end{bmatrix}^T = C_c \cdot x \tag{24}$$

The optimizing objective is to minimize the reference tracking error and guarantee the stability of vehicle, so the cost function of MPC is defined as:

$$\min J = \sum_{i=1}^{p} \left\| W_y (y_c(k+i|k) - r_c(k+i)) \right\| + \sum_{i=1}^{m} \| W_u U(k+i-1) \| \tag{25}$$

With the prediction equation and the limit conditions:

$$\begin{aligned} &\Delta x(k+j+1) = A \Delta x(k+j) + B \Delta u(k+j) \\ &u_{\min}(k+j) \le u(k+j) \le u_{\max}(k+j) \qquad j = 0, 1, \ldots, m-1 \\ &\Delta u_{\min}(k+j) \le \Delta u(k+j) \le \Delta u_{\max}(k+j) \qquad j = 0, 1, \ldots, m-1 \end{aligned} \tag{26}$$

To solve the quadratic optimal function and reduce computing burden, active set algorithm and CVXGEN code generator is adopted (Fig. 4).

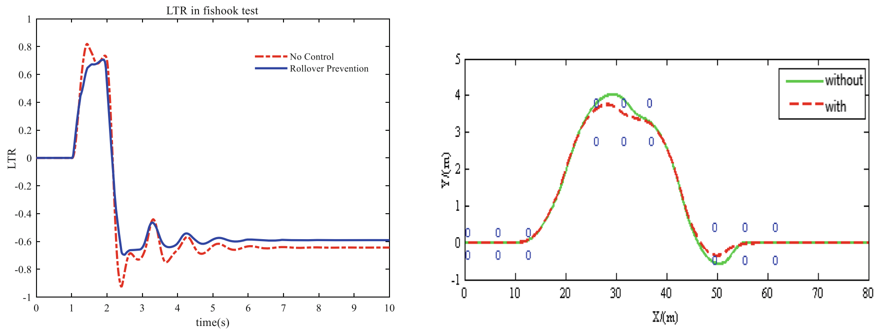

Fig. 4. Simulation results (fishhook test 65 km/h left, Emergency Avoidance test 45 km/h right)

4 Results and Conclusion

Various simulations have been conducted to test the effectiveness of the proposed controller, including a fishhook maneuver and a double lane change maneuver. During the fishhook simulation, the MPC can anticipate the danger of rollover and stabilize the vehicle in time. While in the double lane change simulation, the vehicle with the MPC controller can finish the test under a higher velocity. The results show that the MPC controller can achieve a balance between the lateral stability and rollover prevention.

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References

1. Her, H., Joa, E., Yi, K., et al.: Integrated chassis control for optimized tyre force coordination to enhance the limit handling performance. Proc. Inst. Mech. Eng. Part D J. Automob. Eng. **230**(8), 1011–1026 (2016)
2. Zhang, D., Gordon, T.: A novel control mediation approach to active rollover prevention. In: International Symposium on Advanced Vehicle Control (2016)
3. Alberding, M.B., Tjønnås, J., Johansen, T.A.: Integration of vehicle yaw stabilisation and rollover prevention through nonlinear hierarchical control allocation. Veh. Syst. Dyn. **52**(12), 1607–1621 (2014)
4. Li, L., Lu, Y., Wang, R., et al.: A 3-dimensional dynamics control framework of vehicle lateral stability and rollover prevention via active braking with MPC. IEEE Trans. Industr. Electron. **64**(4), 3389–3401 (2016)
5. Rajamani, R., Piyabongkarn, D.N.: New paradigms for the integration of yaw stability and rollover prevention functions in vehicle stability control. IEEE Trans. Intell. Transp. Syst. **14**(1), 249–261 (2013)
6. Yakub, F., Mori, Y.: Heavy vehicle stability and rollover prevention through switching model predictive control. In: Control Conference, pp. 1–6. IEEE (2015)
7. Pacejka, H.B.: Tire and Vehicle Dynamics, 2nd edn. SAE International, Warrendale (2006)

8. Lin, X., Xiong, L., Xia, X.: Sideslip angle and tire-road friction coefficient estimation simultaneously for autonomous vehicle. In: 25th IAVSD International Symposium on Dynamics of Vehicles on Roads and Tracks, Rockhampton, Australia (2017)
9. Lenzo, B., Sorniotti, A., Gruber, P., et al.: On the experimental analysis of single input single output control of yaw rate and sideslip angle. Int. J. Automot. Technol. **18**(5), 799–811 (2017)