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

Nan XU, Hong CHEN, Haitao DING, Ping WANG, Lin ZHANG
State Key Laboratory of Automotive Simulation and Control
Department of Control Science and Engineering
Jilin University

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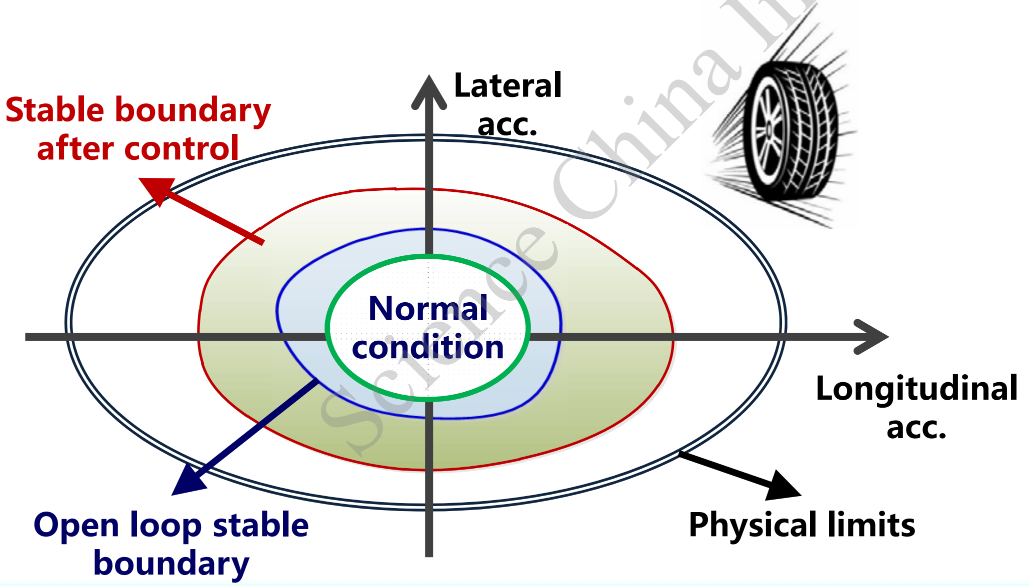
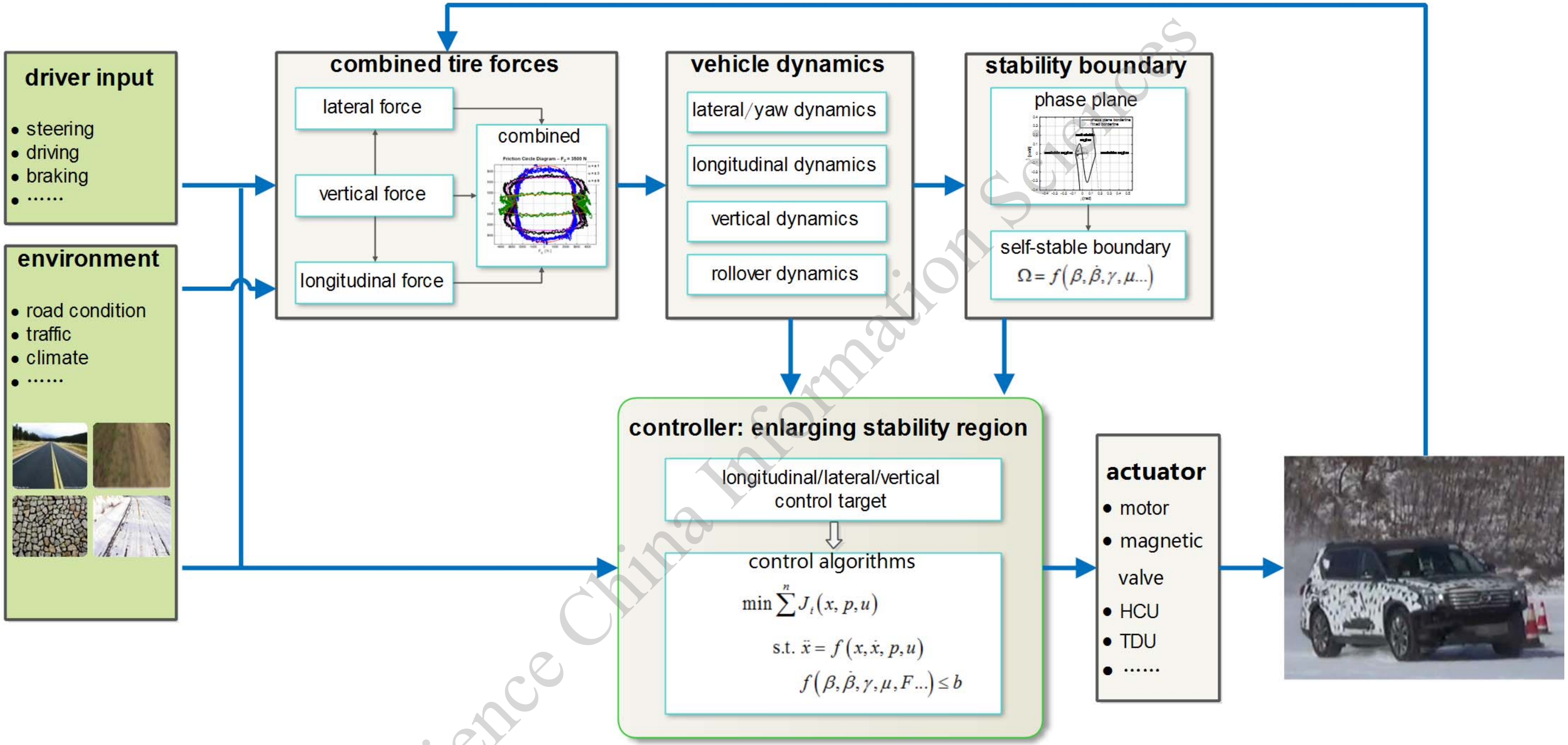
Introduction

- Single vehicle stability systems(ABS, AFS, ESP, etc.) cannot perform well due to the coupling of vehicle dynamics and the highly complex working condition of vehicle.
- Special difficulties for integrated vehicle dynamics control:
  - Vehicle dynamic model: high-dimensional dynamics, combined and nonlinear tire longitudinal/lateral/vertical forces, driver and environmental dependence
  - Accurate describe and identify vehicle stability boundary.
  - Multi-objective coordinated control for enlarging vehicle stability region with different subsystems (ABS, AFS, ESP, etc.)



Test vehicle and controller

Synergy Control Framework



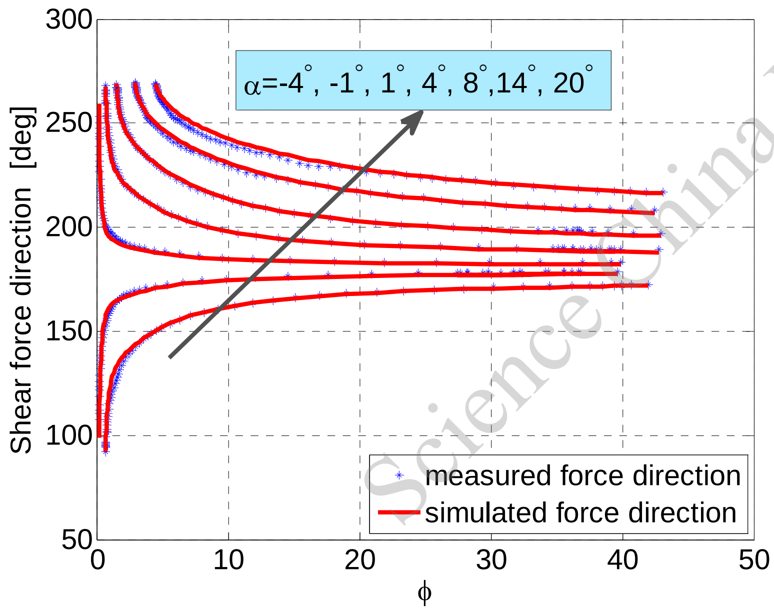
1. Vehicle dynamics model considering combined tire forces
2. Identification of vehicle stability boundary
3. Controller design to enlarge the stability region

Combined Tire Forces and Vehicle Dynamics

- Basic formula of UniTire model

$$\begin{cases} \overline{F} = 1 - \exp\left[-\phi - E\phi^2 - \left(E^2 + \frac{1}{12} \right) \phi^3 \right] \\ F = \overline{F} \cdot \mu F_z \end{cases}$$

$$\text{Direction factor } \lambda = \frac{1 + (\phi/\phi_c)^n \cdot K_y / K_x}{1 + (\phi/\phi_c)^n}$$



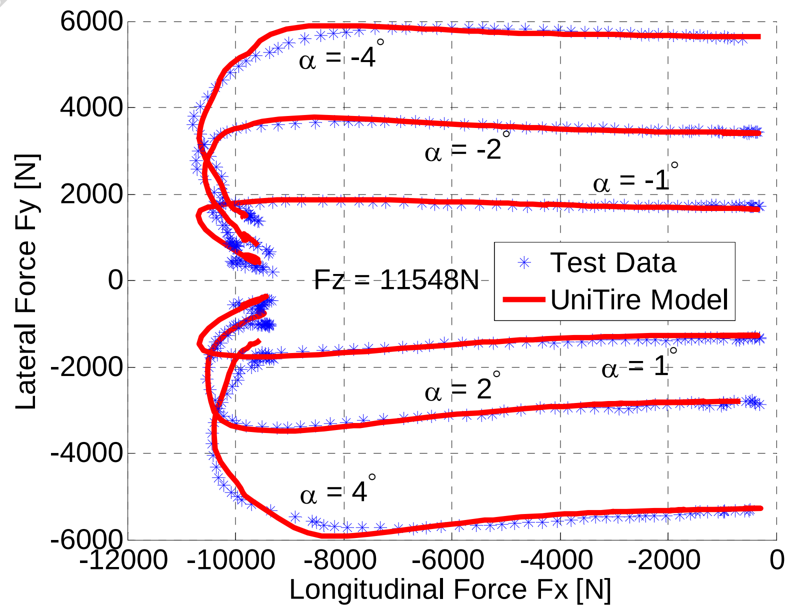
- Nonlinear vehicle dynamics

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

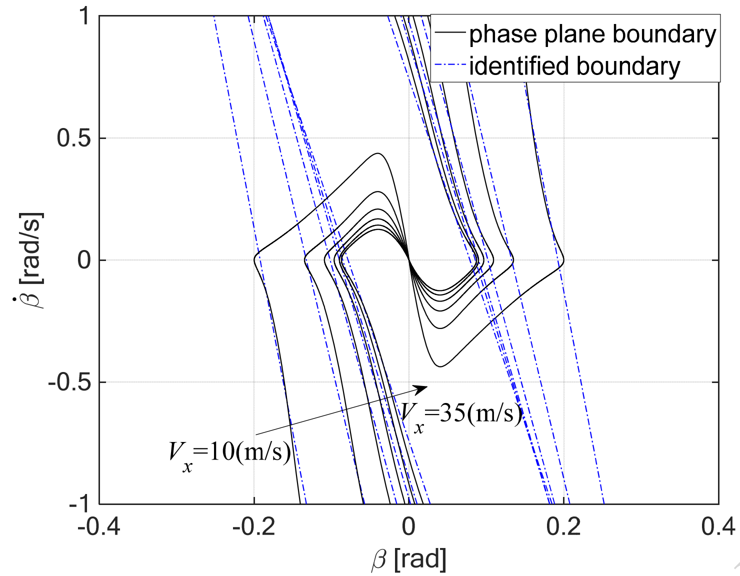
$$\Gamma = [V_x, V_y, V_z, \omega_x, \omega_y, \gamma, \omega_{fl}, \omega_{fr}, \omega_{rl}, \omega_{rr}]$$

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

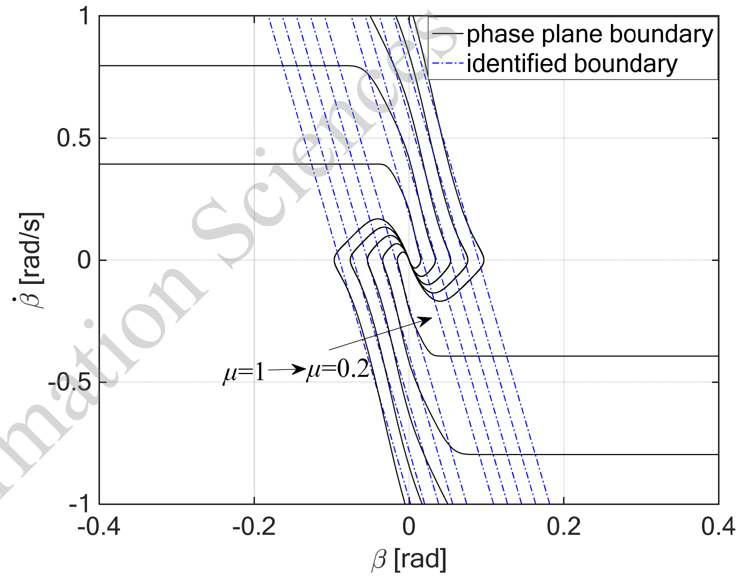
$$y = [V_x, a_x, \omega_x, \gamma, \beta]$$



Identification of stability boundary



δ_f=0(°), μ=1, V_x changes



δ_f=0(°), V_x=25(m/s), μ changes

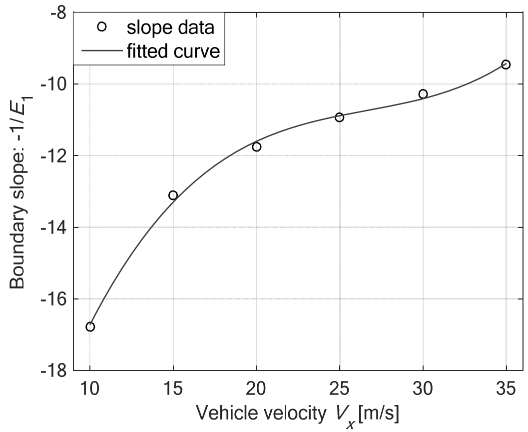
The stability boundary includes 9 parameters (e_1-e_9)

$$E_3 \le \beta + E_1\dot{\beta} \le E_2$$

$$where \quad E_1 = 1\Big/\left(e_1V_x^3 + e_2V_x^2 + e_3V_x + e_4\right)$$

$$E_2 = g(V_x)\cdot g(\mu), \quad E_3 = -g(V_x)\cdot g(\mu)$$

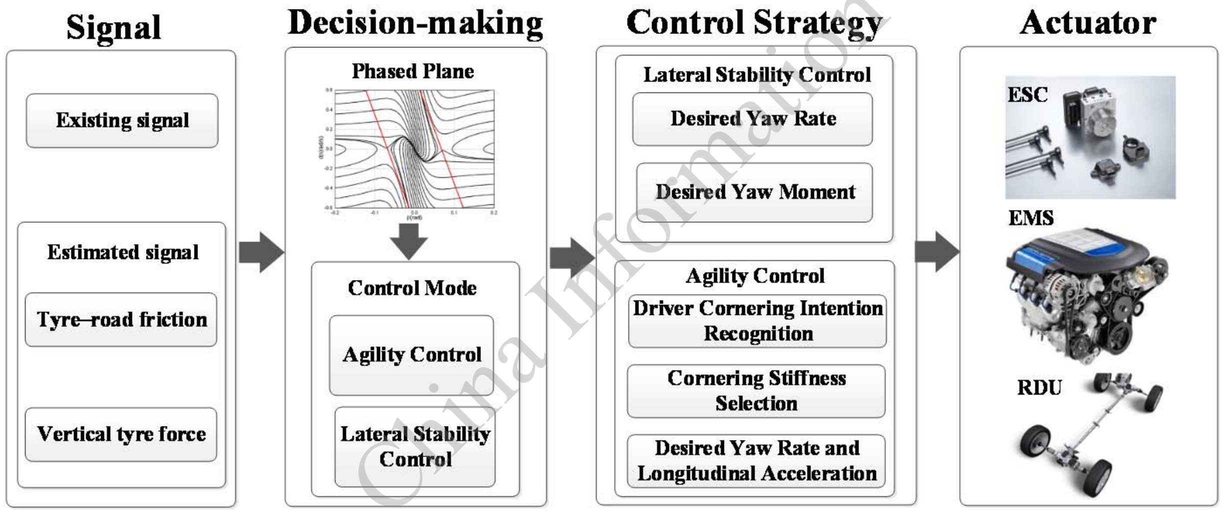
$$and \quad g(\mu) = e_5\mu + e_6, \quad g(V_x) = e_7V_x^{e_8} + e_9$$



Case1-Adaptive Vehicle Dynamic Control (AVDC) for ICV

For internal combustion engine drive vehicle(ICV):

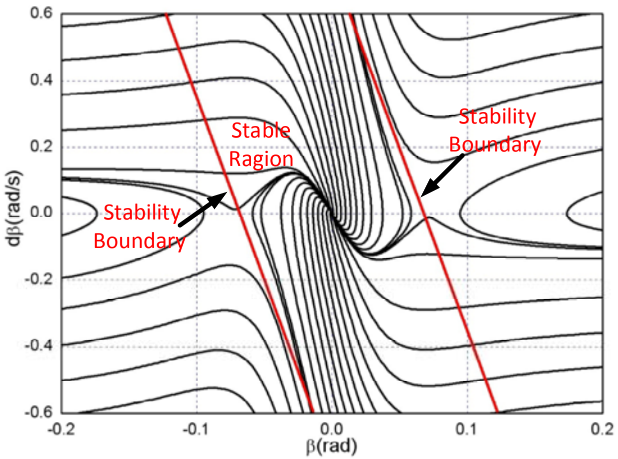
1. Improve vehicle agility and stability.
2. Help drivers navigate through the curve smoothly before ESC intervention.
3. Controls longitudinal motion in accordance with yaw movement.

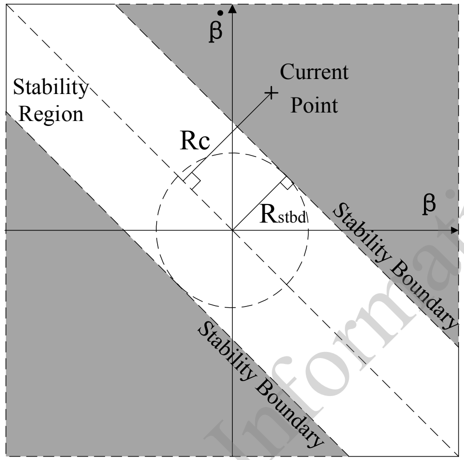


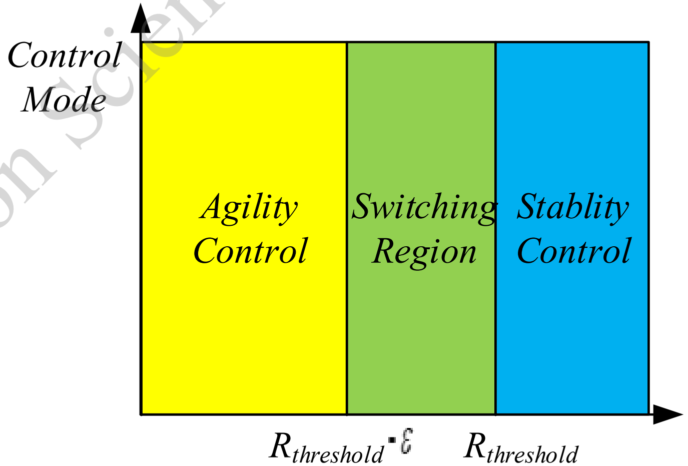
Signal: provide the required signals.
Decision-making: choose a reasonable control mode according to the of vehicle stability.
Control strategy: coordinate ESC, EMS, and RDU with vehicle dynamics and driving intention.
Actuator: perform control commands.

Decision-making

The stability boundary was influenced by the vehicle speed and the road surface friction coefficient . The stability factor R_c is determined according to the distance as shown in figures.







$$\begin{cases} |\beta + E_1\dot{\beta}\ | \le E_2 \\ E_1 = f_1(\mu) \\ E_2 = f_2(\mu, V_x) \end{cases}$$

$$\begin{cases} R_{Stbd} = |E_2|/\sqrt{E_1^2+1} \\ R_c = |\beta_c + E_1\dot{\beta}_c|/\sqrt{E_1^2+1} \end{cases}$$

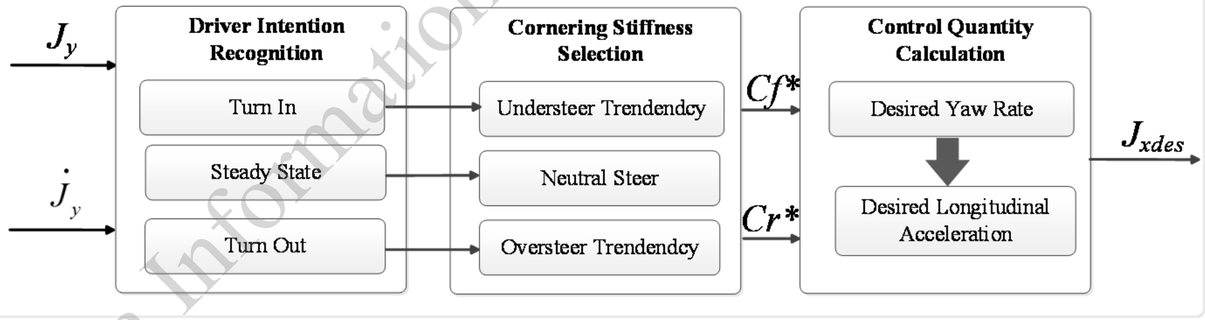
$$\begin{cases} R_c > R_{threshold} \rightarrow stablity\ control\ mode \\ R_c \le R_{threshold} - \varepsilon \rightarrow agility\ control\ mode \\ R_{threshold} - \varepsilon < R_c \le R_c \rightarrow keep\ control\ mode \end{cases}$$

Where *E1*, *E2* are the boundary coefficients, which are related to the vehicle speed *Vx* and the road friction factor μ. $R_{threshold}$ is a threshold value for judging the control mode, in addition, to avoid a discrete change of the agility and stability modes, a switching region was used.

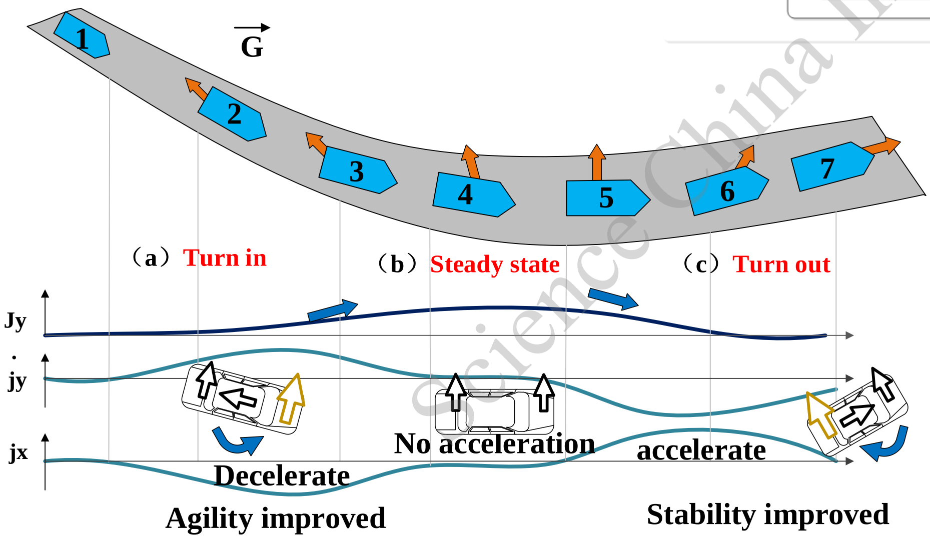
Control Strategy

1. Agility control that works in the tire linear operating region.
2. Agility control scheme can be split into three:
   - driving intention recognition
   - cornering stiffness selection
   - control quantity calculation

Agility Control



Driver intention recognition

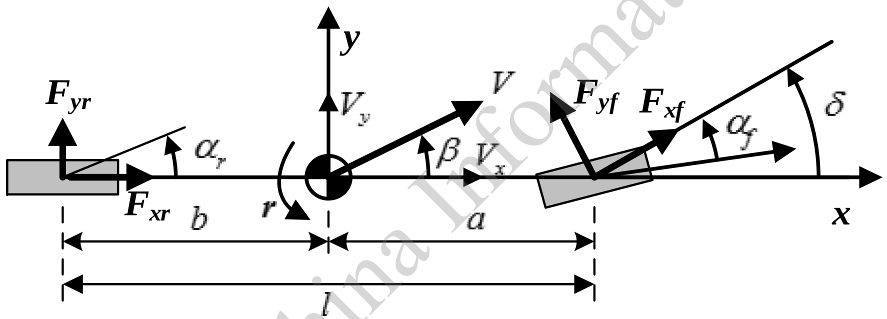


$$\begin{cases} -sign(J_y \dot{J}_y) > 0 \rightarrow entering\ a\ curve \\ -sign(J_y \dot{J}_y) < 0 \rightarrow exiting\ the\ curve \\ sign(J_y \dot{J}_y) = 0 \rightarrow steady-state\ corner\ or \\ driving\ straight \end{cases}$$

Control Strategy

Cornering stiffness selection

1. The cornering stiffness should change with the desired steering characteristics.
2. Increase oversteering tendency when entering the corner.
3. Increase understeering tendency when exiting the corner.
4. The following equation can be used to calculate desired yaw rate for vehicle agility:



$$\frac{w_r}{\delta}(s) = \frac{MV_x aC_f s + lC_f C_r}{MV_x I_z s^2 + [I_z(C_f + C_r) + M(a^2 C_f + b^2 C_r)]s + C_f \frac{C_r}{V_x} l^2(1 + KV_x)}$$

Where, J_x ,J_y is the longitudinal and lateral acceleration, *M* the mass of the vehicle,w_{r_des} the desired yaw rate, I_z the yawing moment of inertia, δ front wheel steering angle, *a*, *b* the distance between the front and rear axles from the vehicle's center of gravity point. Where C_i is cornering stiffness C_i (i = f, r, where f is the front wheel, r the rear wheel)

AVDC Control Strategy

Control quantity calculation

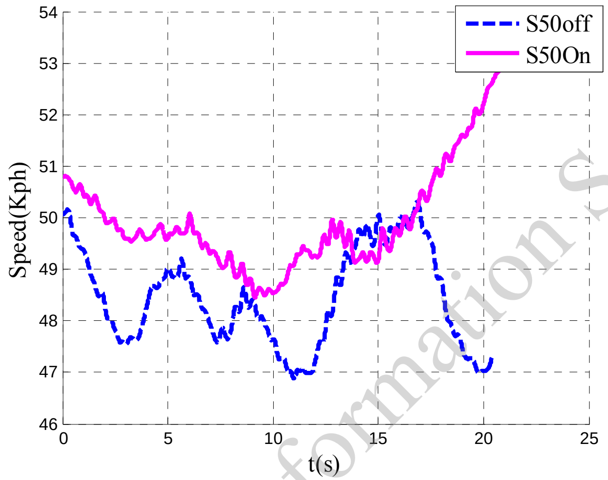
1. Appropriate acceleration or deceleration.
2. Chang the driving torques of the front and rear axles.
3. The relationship between the desired yaw rate, longitudinal acceleration and front and rear drive ratio is derived as follow:

$$\dot{w}_r = f(J_x,\sigma) = \frac{-F_{yr0}b\left(1-\frac{J_x^2M^2(\sigma-1)^2}{W_{zf}^2\mu^2}\right)^{\frac{1}{2}}\left(\frac{J_xh}{ag}+1\right)+F_{yf0}a\left(1-\frac{J_x^2M^2\sigma^2}{W_{zr}^2\mu^2}\right)^{\frac{1}{2}}\left(\frac{J_xh}{bg}-1\right)}{I_z}$$

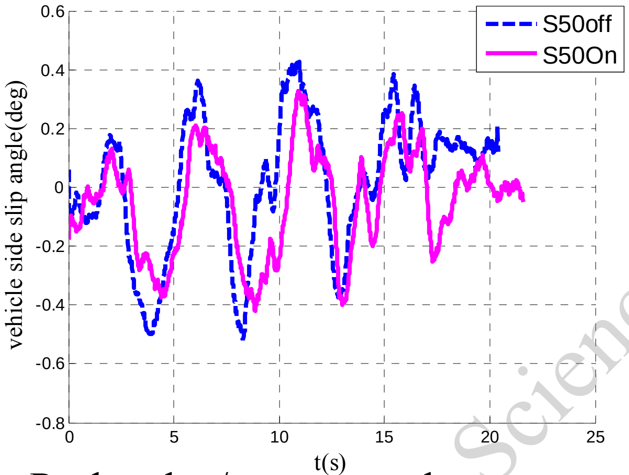
Where, F_{yr0}, F_{yf0} is the cornering force before control. σ is the front and rear axis traction force ratio.
The desired yaw motion of the vehicle can be realized by J_x and σ.
First change the σ to achieve the desired yaw motion, and if necessary, brake or accelerate the vehicle.
For example, when entering a curve, σ is taken as the rear axle driving mode in order to make the J_x small.

AVDC Experiments Results

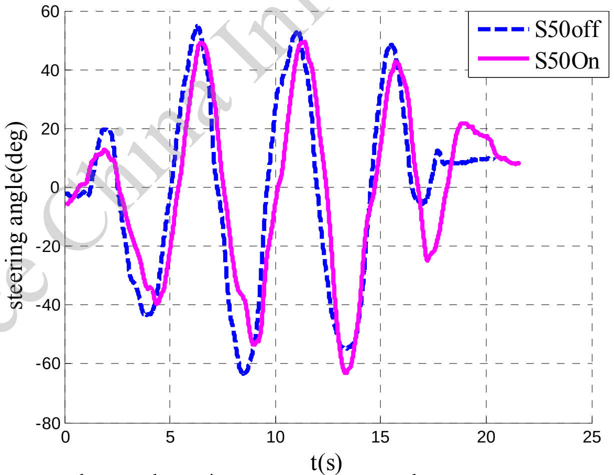
- Slalom on snow and ice



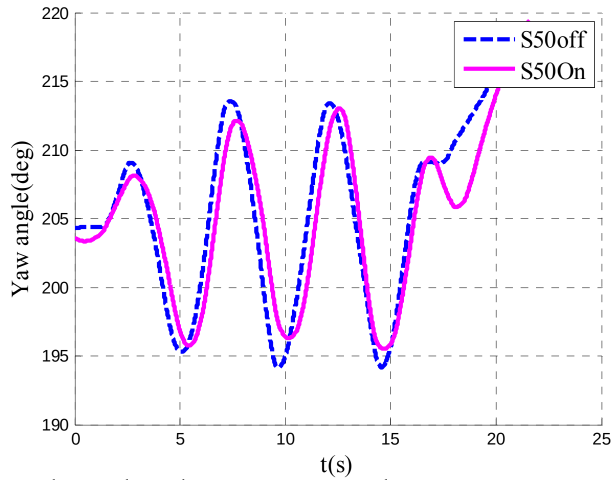
Peak value/ average value
With control: 53/50.58kph
Without control: 50/47.65kph



Peak value/ average value
With control:-0.4/0.085deg
Without control: -0.54/0.15deg



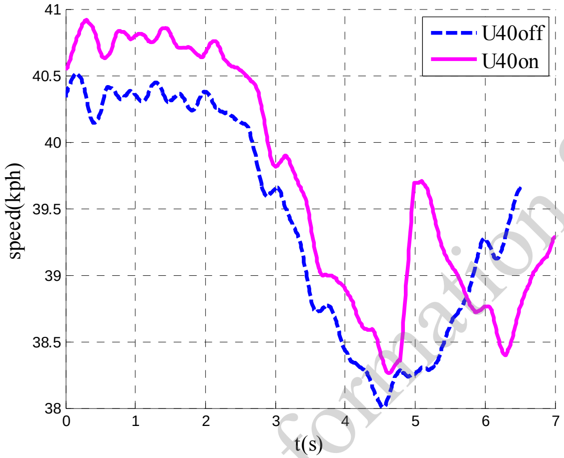
Peak value/ average value
With control:-65deg/-0.13deg
Without control:-66.8deg/-4deg



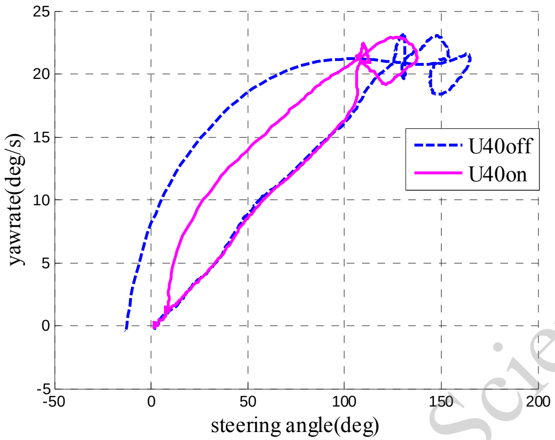
Peak value/ average value
With control:212.6deg/205.3815
Without control: 216.63deg/ 207.2deg

Experiments Results

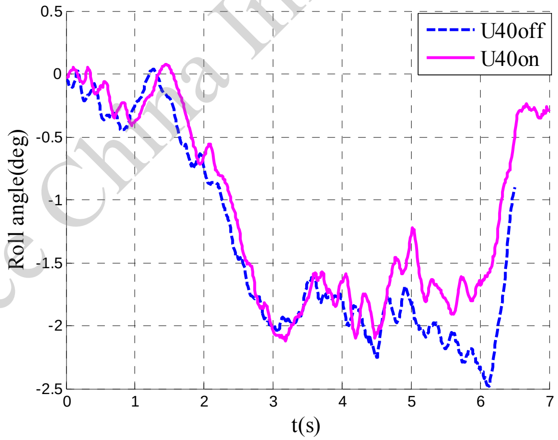
- U turn on snow and ice



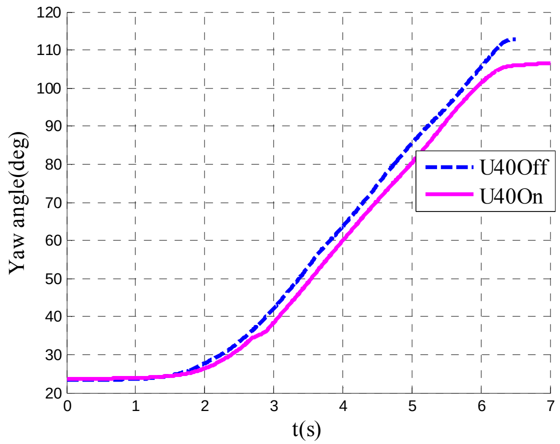
Peak value/ average value:
With control: 40.9/39.8kph
Without control: 40.5/38.3kph



Linearity comparison:
With control: high linearity
Without control: low linearity



Peak value/ average value:
With control:-2.1/-2.5deg
Without control:-1.6deg/-2.2deg



Peak value/ average value:
With control:108/112
Without control: 50.43deg/ 55.9deg

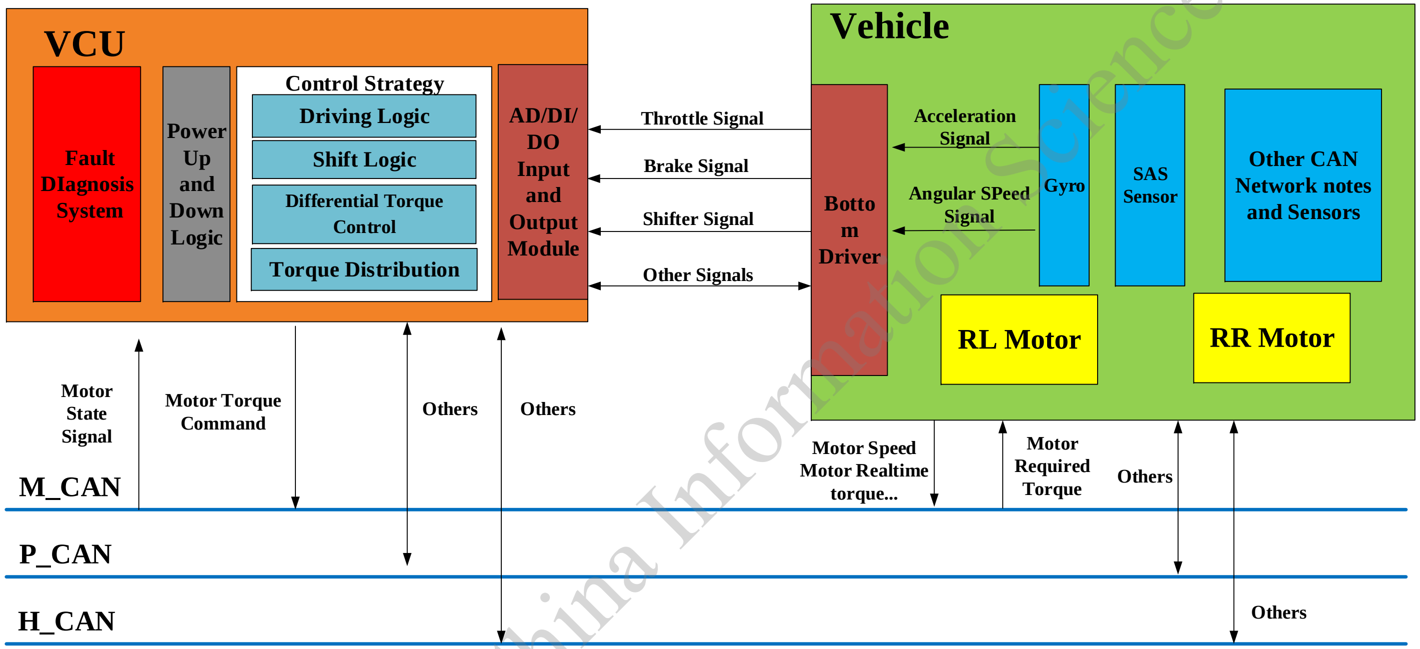
Case2-Differential Torque Control for EV

For electric vehicle with in-wheel motor(EV)

- Improve the passband of the yaw rate response and the natural frequency
- Reduces the response lag

Vehicle Control System Architecture

The vehicle control system contains the vehicle, vehicle control unit and other network notes.



Fault Diagnosis System:

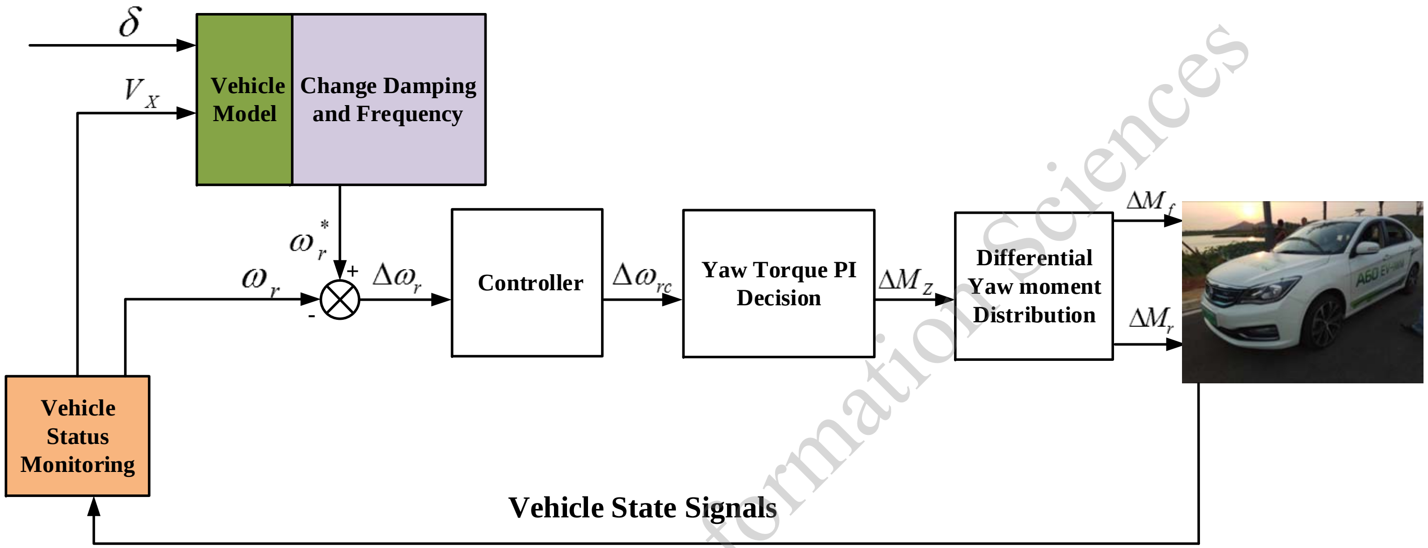
1. health-monitoring, fault detection, fault diagnosis
2. improve reliability, availability, maintenance and life-time

Power up and down: turn on or off the control system

Control Strategy: control motors to complete differential torque

Input and Output Model: receive and send signals

Control Strategy: Differential Torque Strategy



Two-degree-of-freedom vehicle model yaw rate response process according to:

$$\omega_r(s) = G_r \cdot \frac{1+\tau_r s}{\frac{1}{\omega_n^2}s^2 + \frac{2\zeta}{\omega_n}s + 1} \cdot \delta$$

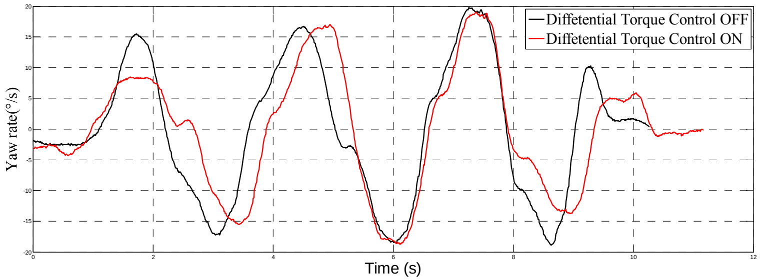
The second-order system dynamics mainly depend on frequency and damping:

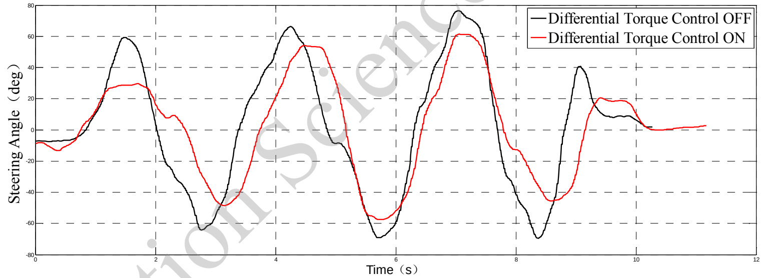
(1) Frequency: the higher the natural frequency, the faster the system reacts, but the worse the stability.

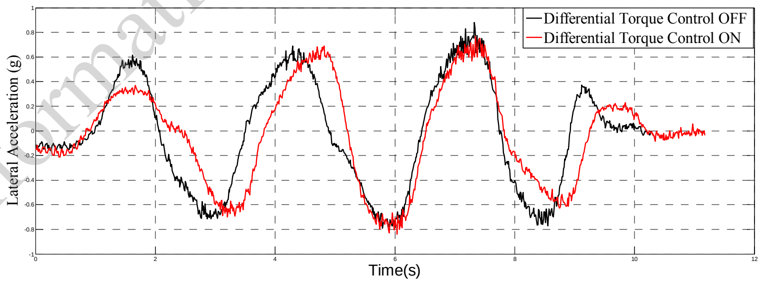
(2) Damping: the greater the damping ratio, the slower the response speed of the system, but the faster the system decays.

Experiments Results

Slalom test:







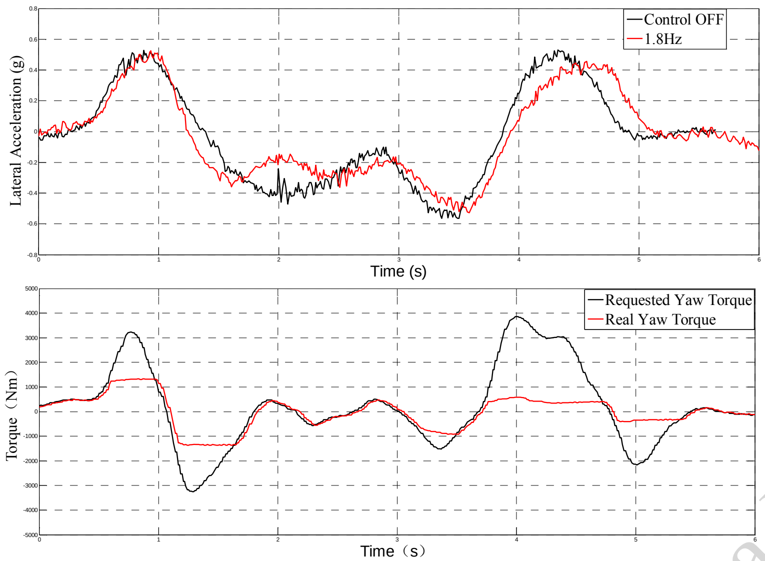
The slalom test data shows that:

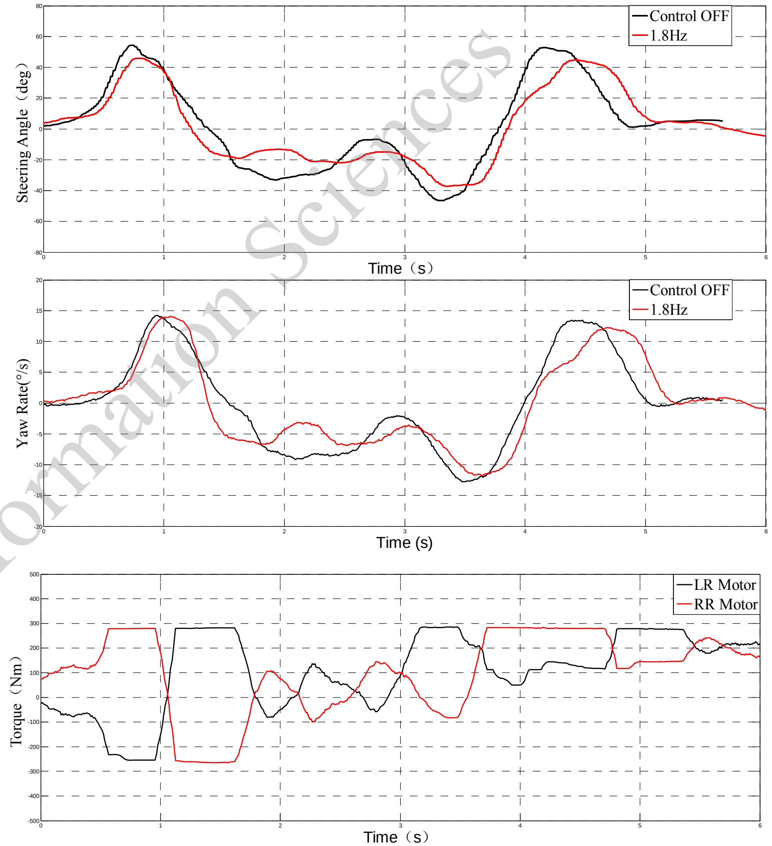
(1) at the same vehicle speed, compared to none differential control, the differential control can significantly reduce the average amplitude of steering wheel, thereby reducing the yaw rate and lateral acceleration of the vehicle when the car is over piled.

(2) the mean steering wheel angle decreases about 19.6%, the average yaw rate decreases about 2.2%, and the average lateral acceleration decreases about 2.4% during the slalom process. The maximum steering wheel angle decreased from 76.5° to 61.5°, a decrease of 19.6%.

Experiments Results

Double Line Change:





The picture shows that:

(1) after the control, the maximum turning angle of the steering wheel by the driver is reduced to some extent.

(2) the actual differential torque cannot be satisfied when the demand torque is large. This is due to the limitation of the current vehicle speed and the differential torque margin of the current throttle.

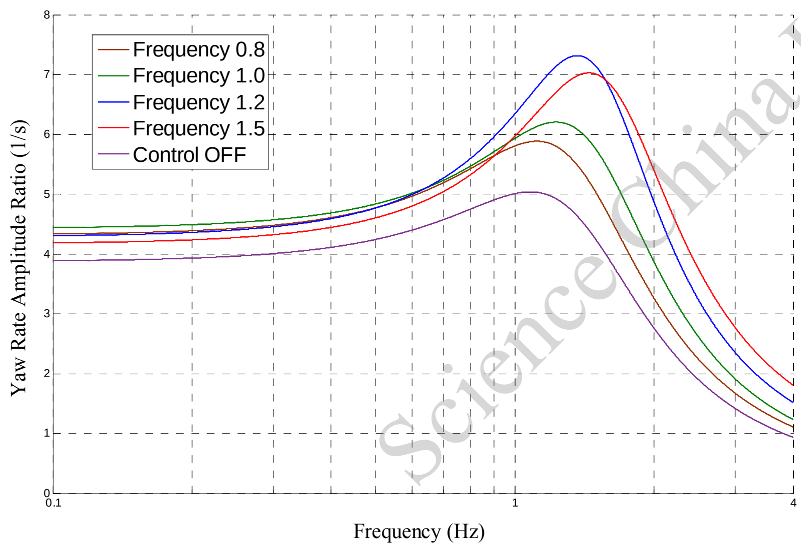
Experiments Results

Double Line Change:

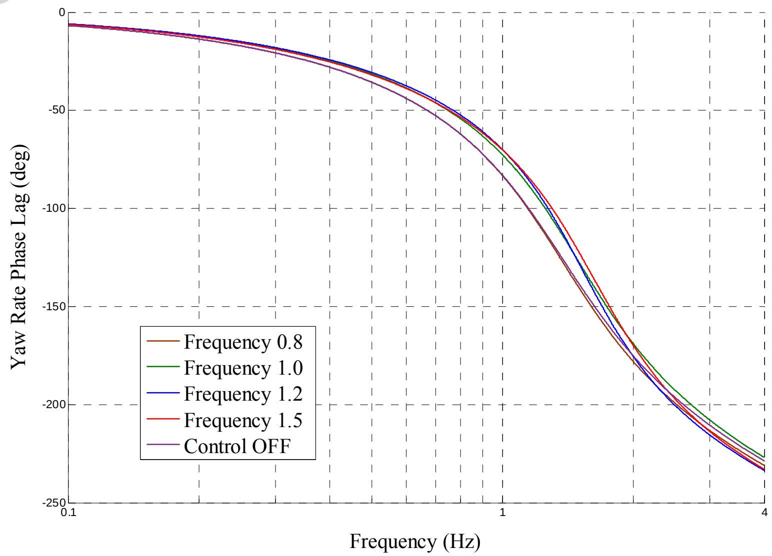
(1) Differential torque control significantly expands the pass band.

(2) Compare with none-control model, through differential torque control, the maximum passband is increased by 43.5%, the resonant frequency is increased by 33.9%, and the natural frequency is increased to 1.614Hz.

(3) Compare with none-control model, after the differential torsion control, the 0.1Hz phase angle hysteresis is reduced by at most 12.5%, and the 0.6Hz phase angle hysteresis is reduced by at most 14.5%. During the actual vehicle driving, the driver obviously feels the improvement of the steering agility of the vehicle after the differential torsion control is started.

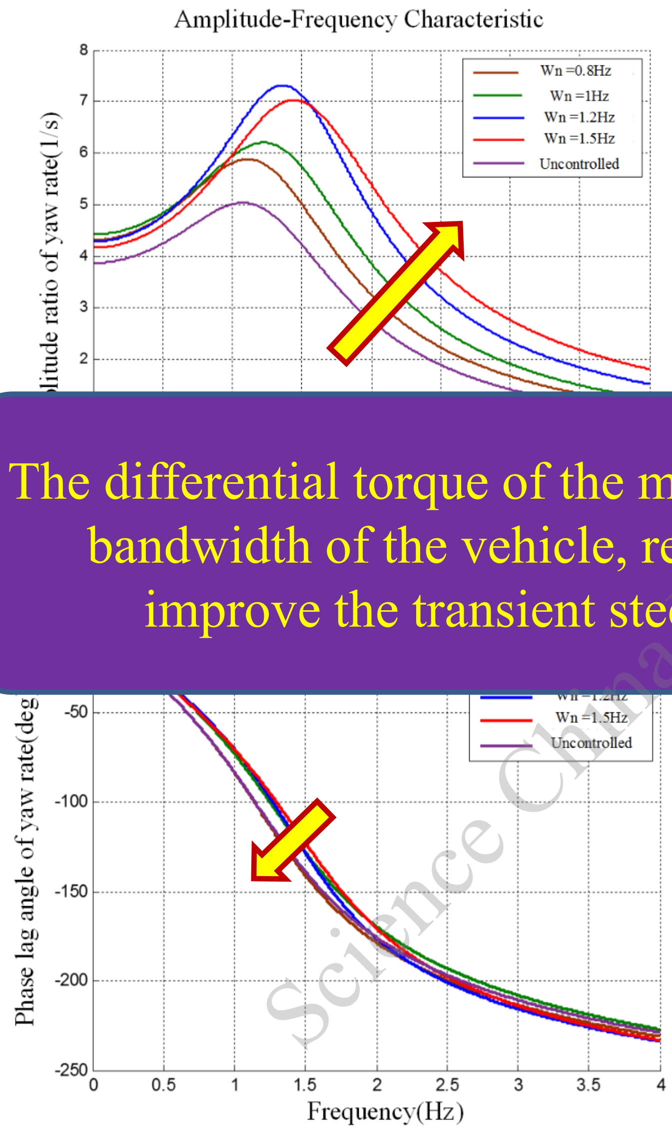


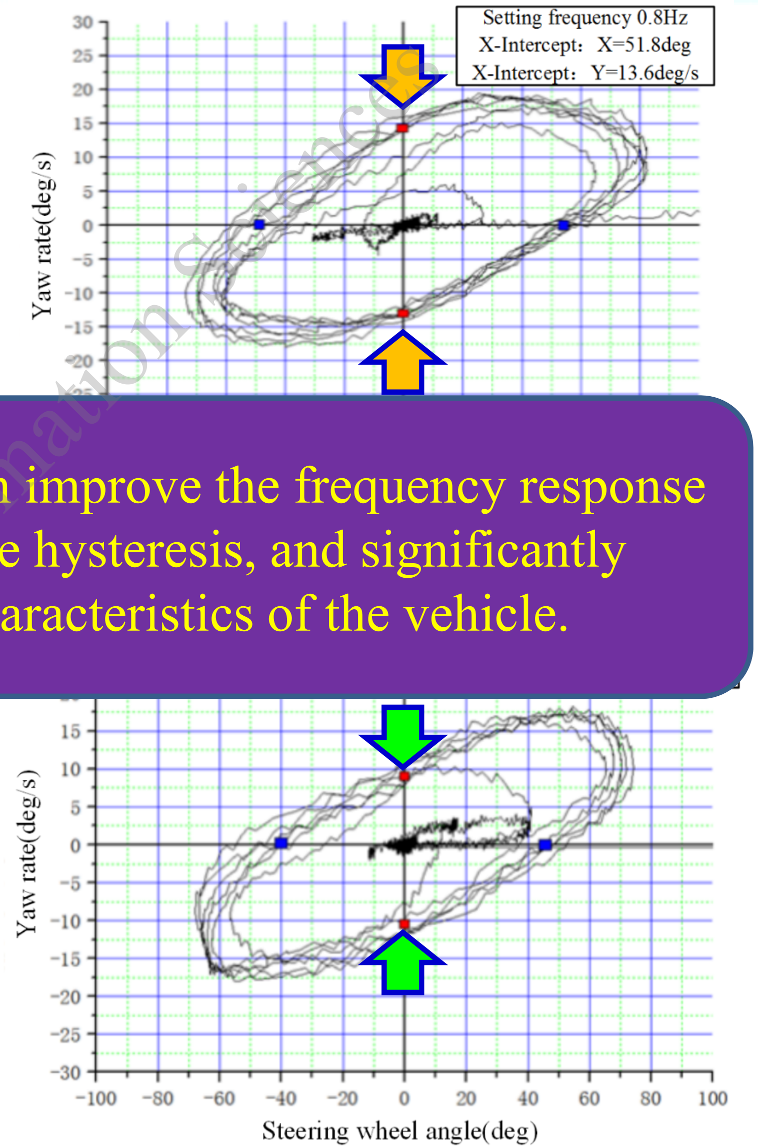
Amplitude frequency characteristic curve



Phase frequency characteristic curve

Experiments Results





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

- A synergy control framework for enlarging vehicle stability is proposed.
- Based on the nonlinear and combined tire forces and vehicle dynamics, the self-stable boundary is identified, which is crucial as constraints for the development of controller.
- Controller for enlarging vehicle stability region is designed, and multiple control objectives can be coordinated and optimized.
- Two kinds of vehicle, i.e., internal combustion engine drive vehicle(ICV) and electric vehicle with in-wheel motor(EV) are utilized to evaluate the effects of control strategies. Experimental results demonstrated how this method can be used effectively in vehicle stability control.

Thanks