Controller Design for an Automobile Steer-By-Wire System

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***Abstract*—The purpose of this study is to propose a control scheme for an automobile Steer-By-Wire (SBW) system. A mathematical model of SBW system is built. A co-simulation platform in software of MATLAB/Simulink and a Hardware-in-the Loop (HIL) system is developed. By using ARX algorithm, the parameters of the SBW system are identified. Then, an Internal Model Controller (IMC) for the SBW system is designed to track desired motion states of controlled vehicle. In typical driving conditions, simulations are carried out to examine the effectiveness of designed controller. Compared with a PID controller, the results demonstrate that designed IMC controller is able to provide better control performance. To reduce the steering efforts of the driver, meanwhile ensure the stability of the vehicle in different driving conditions, a variable gearing ratio control strategy is proposed, and also its characteristics are examined in different cases. Besides, a Slide Mode Controller (SMC) for tracking desired yaw rate is designed to realize the active steering. Co-simulation results show that desired yaw rate can be tracked satisfactorily by using the designed controllers, with enhanced handling and stability performance.**

Keywords—SBW, IMC/SMC, Controller design, ARX algorithm, Active steering

I. Introduction

Recently, increased attention has been paid to autonomous driving due to its potential applications. While Steer-By-Wire (SBW) system of automobile is recognized as an important system for highly autonomous driving [1]. By using motors to control front wheel steering angles directly, SBW system can remove those mechanical connection installed in conventional steering systems. Due to its high efficiency and flexibility in communication and control integration, SBW system provides hardware condition for autonomous steering. Meanwhile, by utilizing useful road feedback and sensing information, SBW system can optimize vehicle response to steering input and improve vehicle stability even in some emergence or limit cases.

Many studies have been carried out and various strategies have been developed for SBW systems, based on different control theories. For example, Yang and Liao [2] proposed a Fuzzy PID controller to enhance the robustness of steering motor control. Wu et al. [3] proposed an estimator and designed an adaptive feedforward torque controller based on steering rack speed error, with considering uncertain parameters. Scicluna et al. removed current sensors and realized closed current loop control of the steering motor by injecting high frequency current [4].

Internal Mode Control (IMC), which was firstly proposed by Horowits and Issac [5], has been widely used in industrial process due to its good robustness. Canale et al. [6-7] proposed IMC controller for the stability control of 4WS vehicles, and compared the IMC with SMC for vehicle yaw control. Men et al. [8] developed IMC controller to solve the nonlinearity problem for front wheel active steering, which showed that the IMC can be applied to most driving maneuvers and road surface conditions. Wu et al. [9] proposed a 2-DOF IMC controller to achieve high tracking performance of yaw rate with certain robustness.

Since SBW system can directly change the gear ratio of steering system and realize feedback control corresponding to state variables with combining vehicle stability controller, Zong et al. [10] proposed a variable gain ratio controller for steering system and realized the variable gear ratio according to prevailing vehicle speed. Yu et al. [11] investigated the effects of yaw rate gain and lateral acceleration gain on the steering system and the design a strategy of the variable angle gear ratio. Shi et al. [12] designed a steering angle gear ratio of SBW system using slide mode control and fuzzy control. Tian et al. [13] set up a HIL test bench for SBW system and carried out experimental study for active steering control.

In the present study, based on a developed test bench and co-simulation platform, an IMC controller is designed for the SBW actuator system, in order to maintain vehicle stability, vehicle tracking and anti-interference capability in different road conditions, while a SMC controller is designed for stability improvement. The paper is organized as below. In Section 2, the mathematical model for the steering actuator (i.e., electrical motor) and reference vehicle model are respectively built, based on which a variable gear ratio strategy is developed. In Section 3, an IMC controller is designed to track desired vehicle state variables based on identified parameters, and a SMC controller is designed to track vehicle desired reference yaw rate. In Section 4, the simulation results in different operation conditions are analyzed and the effectiveness of the designed controller are examined. The conclusions are drawn in Section 5.

II. Modeling

In this section, a steering actuator model is established for road-wheel steer angle tracking controller design and a 2-DOF vehicle model is built for active steering control and for desired yaw rate stability improvement. Moreover, variable gear ratio strategy is designed with constant yaw rate gain and constant lateral acceleration gain correspondingly.

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A. Steering actuator Model

The SBW actuator system receives the front wheel angle signal, with steering motor closed-loop control, to ensure that the actuator system outputs an accurate front wheel angle. The framework of the SBW system investigated in this paper is shown in Fig.1, in which front wheel angle is controlled by steering rack displacement. A test bench is built accordingly, as shown in Fig.2, consisting of a steering motor, a gear rack mechanism and other relative components.

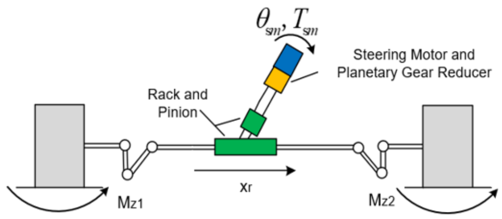


Fig. 1. Physical framework of SBW system

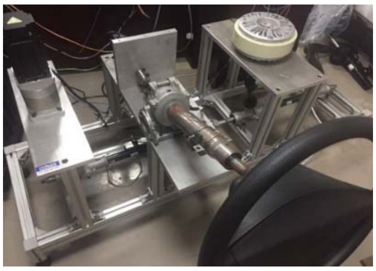


Fig. 2. Test bench of SBW system

A brushless DC motor is adopted as the steering control motor in SBW system. A magnetic powder brake is used for simulating steering load. A simplified model is derived for the motor, represented by its transfer function as below.

$$(M_r s^2 + B_r s + \frac{K_{md} g_{sm}^2}{r_p^2}) \bullet x_r(\mathrm{s}) + F(\mathrm{s}) = \frac{K_{md} g_{sm}}{r_p} \theta_{sm}(\mathrm{s}) \quad (1)$$

$$T_{sm}(\mathrm{s}) = \left(J_{sm} s^2 + B_{sm} s + K_{md}\right) \bullet \theta_{sm}(\mathrm{s}) - \frac{K_{md} g_{sm}}{r_p} x_r(\mathrm{s}) \quad (2)$$

$$U_{sm}(\mathrm{s}) = \frac{R_{sm} + L_{sm} s}{K_2} T_{sm}(\mathrm{s}) + K_{sm} s \theta_{sm}(\mathrm{s}) \quad (3)$$

where M_r is steering rack mass, B_r is steering rack damping coefficient, x_r is steering rack displacement, F is steering load (i.e., resistance force applied on rack), K_{md} is pinion gear torsional stiffness, g_{sm} is steering motor gearing ratio, θ_{sm} is steering motor rotation angle, r_p is pinion radius, T_{sm} is motor torque, J_{sm} is motor moment of inertia, B_{sm} is motor damping coefficient, U_{sm} is motor voltage, R_{sm} is motor resistance, L_{sm} is motor inductance, K_{sm} is back-EMF coefficient, K_2 is electromagnetic torque coefficient of road simulator.

B. Vehicle Model

A 2-DOF linear single-track vehicle model is established for the active steering controller design, with two degrees of freedom for yaw and lateral motions [14], shown in Fig.3, which captures the primary handling characteristics and thus is widely used in linear domain simulation.

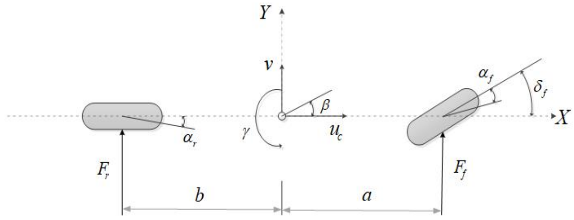


Fig. 3. 2-DOF linear model

The equation of motion of the vehicle model is written in state-space format as,

$$\begin{pmatrix} \dot{v} \\ \dot{r} \end{pmatrix} = \begin{pmatrix} -\dfrac{C_{af} + C_{ar}}{mu_c} & -\dfrac{aC_{af} - bC_{ar}}{mu_c} - u_c \\ -\dfrac{aC_{af} - bC_{ar}}{Iu_c} & -\dfrac{a^2 C_{af} + b^2 C_{ar}}{Iu_c} \end{pmatrix} \begin{pmatrix} v \\ r \end{pmatrix} + \begin{pmatrix} \dfrac{C_{af}}{m} \\ \dfrac{aC_{af}}{I} \end{pmatrix} \delta_f \quad (4)$$

where C_{af} and C_{ar} are tire cornering stiffness of front and rear wheels, I is yaw moment of inertia, m is mass of vehicle, a is the distance from vehicle c.g. to front axle, b is the distance from vehicle c.g. to rear axle, u_c is constant forward speed, v is lateral speed of vehicle, r is yaw rate and δ_f is front wheel angle.

With SBW system, the gear ratio can be easily adjusted since less physical constraints remain. The variable gear ratio related to yaw speed gain and lateral acceleration gain is designed for ensuring stability and safety. Based on the equation of motion (4), the steady state response of the vehicle can be derived as (5) with the steady-state yaw rate γ_{ss} . Here, steering gear ratio i_1 is maintained constant over the steady-state yaw rate γ_{ss} , which is represented by (6).

$$\gamma_{ss} = \frac{u_c / L}{1 + \frac{m}{L^2}(\frac{a}{C_{ar}} - \frac{b}{C_{af}}) \bullet u_c^2} \delta_f \quad (5)$$

$$i_1 = \frac{u_c / L}{1 + \frac{m}{L^2}(\frac{a}{C_{ar}} - \frac{b}{C_{af}}) \bullet u_c^2} \bullet \frac{1}{C_x} \quad (6)$$

where $C_x = r / \delta_{sw}$ is the gain from steering wheel angle to yaw rate response, and L is vehicle wheelbase.

Based on the evaluation index of steering stability proposed in [15], this paper takes the evaluation indexes respectively accounting for tracking error, motion direction control error, driver's steering effort and lateral acceleration as a comprehensive evaluation with average weighting, shown as below,

$$J = \sqrt{\frac{k_1 J_{e1}^2 + k_2 J_{e2}^2 + k_3 J_r^2 + k_4 J_b^2}{k_1 + k_2 + k_3 + k_4}} \quad (7)$$

where J_{e1} represents lateral trajectory tracking error evaluation, J_{e2} represents motion direction error evaluation, J_r represents driver's steering effort evaluation and J_b represents lateral acceleration evaluation. Assuming that the four evaluation indicators share the same weight, the four weight parameters k_1 , k_2 , k_3 , k_4 are assigned as same value, being equal as 0.25.

Taking the stability evaluation J represented in (7) as fitness function, the yaw rate gain is optimized by Genetic Algorithm to achieve its local minimum, with optimization range constrained in 0~1. Based on the SBW model established above, the optimal yaw rate gain is searched in some cases, e.g. double lane change condition with forward speed of 20km/h, 40km/h, 60km/h, 80km/h and 100km/h respectively. With result analysis, when the yaw rate gain equals to 0.31, the evaluation J at different speeds shows superior stability in searched range and therefore 0.31 is selected as the optimal yaw rate gain C_x.

Due to lateral acceleration and longitudinal speed $a_y = u_c \cdot r$, the gear ratio based on the constant lateral acceleration gain can be obtained as,

$$i_2 = \frac{u_c^2 / L}{1 + \frac{m}{L^2}(\frac{a}{C_{ar}} - \frac{b}{C_{af}}) \bullet u_c^2} \bullet \frac{1}{C_y} \tag{8}$$

where C_y is the gain from steering wheel angle to lateral acceleration, being equal to a_y / δ_{sw}. By using same optimization method as C_x, the lateral acceleration gain C_y is selected to 4.0 in this study.

Considering to achieve the minimum stability evaluation J, the gear ratio optimizaiton is aimed at constant yaw rate gains above. It is significant to adopt i_1, the gear ratio strategy with constant yaw rate gain at low speed and adopt i_2,the gear ratio strategy with constant lateral acceleration gain at high speed to guarantee vehicle stability [16].

III. Controller Design

Based on the established model of SBW actuator and vehicle model, an IMC controller and a SMC controller are designed respectively. The former is for tracking target front wheel angle input by the driver while the latter aims at improving vehicle stability.

A. *IMC Controller for SBW actuator system*

An IMC controller is designed for the steering actuator system to track the target front wheel angle and a typical IMC control scheme is shown in Fig.4, with system output y represented by (9).

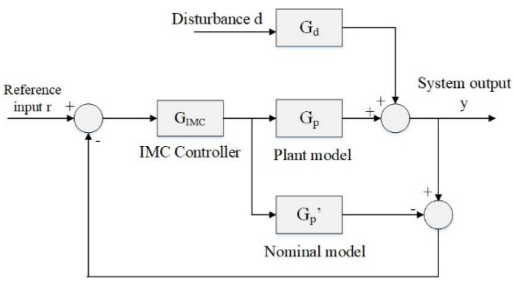


Fig.4. A typical IMC scheme

$$y(s) = \frac{G_{IMC} G_p}{1 + G_{IMC}(G_p - G_p^{'})} r(s) + \frac{(1 - G_{IMC} G_p^{'}) G_d}{1 + G_{IMC}(G_p - G_p^{'})} d(s) \tag{9}$$

In this study, the actuator control scheme is described by Fig.5, with steering resistance on rack as disturbance and the steering output from driver model in CarSim as system reference input. The IMC controller determines steering motor voltage, and the system outputs actual front wheel angle measured by rack displacement. Equations (1)~(3) can be re-written as (10) with the coefficients given as below.

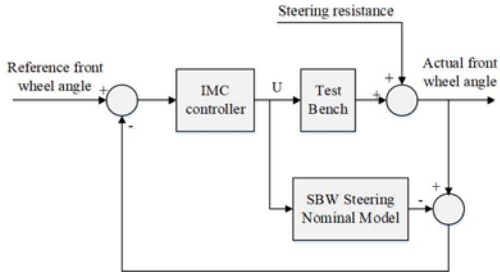


Fig.5. IMC control of the steering actuator system

$$\begin{cases} x_r(s) = \begin{pmatrix} H_1(s) & H_2(s) \end{pmatrix} \begin{pmatrix} F(s) \\ U_{sm}(s) \end{pmatrix} \\ H_1(s) = \dfrac{ABCDF + AEF}{1 - (\text{ABCDFG} + \text{AEFG})} \\ H_2(s) = \dfrac{CDF}{1 - (\text{ABCDFG} + \text{AEFG})} \end{cases} \tag{10}$$

where $\mathrm{A} = \frac{r_p}{K_{md} g_{sm}}$, $\mathrm{B} = -K_{sm}s$, $\mathrm{C} = \frac{1}{R_{sm} + L_{sm}s}$, $D = K_2$,

$E = -\left(J_{sm}s^2 + B_{sm}s + K_{md}\right)$, $\mathrm{F} = -\frac{r_p}{K_{md} g_{sm}}$, $\mathrm{G} = M_r s^2 + B_r s + \frac{K_{md}\, g_{sm}^2}{r_p^2}$.

For the IMC controller, the corresponding voltage is given by

$$U(s) = \frac{e_f}{H_2(s)} \frac{1}{(1+\lambda s)^3} \tag{11}$$

where $1/(1+\lambda^3)$ represents the IMC filter to ensure the denominator order is higher than the numerator order, λ is for hysteresis performance adjustment, e_f represents angle deviation between actual model and nominal model.

To establish nominal model in the control scheme, all coefficients of A to G described in (10) are identified. In this paper, ARX algorithm, by evaluating input, output signals and errors in discrete form for extremum, is applied for coefficients identification in this paper, described as,

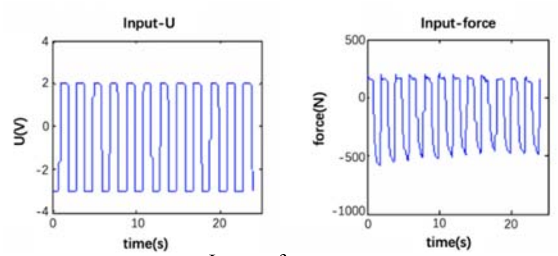
$$A(z)\delta(t) = B_1(z)U(t-d) + B_2(z)F(t-d) \tag{12}$$

And the results are obtained correspondingly as,

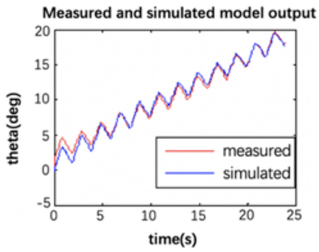
$$\begin{aligned} A(z) &= 1 - 0.9721z^{-1} - 0.1091z^{-2} - 0.00907z^{-3} + 0.1009z^{-4} - 0.01069z^{-5} \\ B_1(z) &= -0.009372 + 0.02649z^{-1} - 0.00885z^{-2} - 0.009274z^{-3} \\ B_2(z) &= -0.000129 - 0.0003104z^{-1} + 0.0004192z^{-2} - 0.0001281z^{-3} \end{aligned} \tag{13}$$

where z^{-1} represents the Lag factor.

By transform of Z to S, the coefficients of the discrete transfer function can be transferred to corresponding unspecified parameters in (10). The verifications have been carried out to examine the effectiveness of identifications. Fig.6(a) presents test examples of both input voltage and resistance force by typical signals, e.g. square wave. The effectiveness has been verified by comparison of measured and simulated data presented in Fig.6(b).



a. Inputs of actuator system



b. Measured and simulated output angle

Fig.6. Parameters verification

B. SMC Controller for vehicle stability

In this section, a SMC controller for tracking ideal yaw rate is designed. The controller takes the deviation of the ideal yaw rate and actual yaw rate as input, and outputs the front wheel angle compensation, which SBW system based on to manipulate steering operation for stability improvement. Combined with 2-DOF model, the derivation of sliding surface for SMC controller design is obtained as (14) and (15) to represent the deviation of the yaw rate.

$$s = \gamma - \gamma_d \tag{14}$$

$$\begin{aligned} \dot{s} &= \gamma - \dot{\gamma}_d = a_{21} \bullet v + a_{22} \bullet \gamma + b_{21} \bullet \delta_f - \dot{\gamma}_d \\ &= a_{21} \bullet v + a_{22} \bullet \gamma + b_{21} \bullet (\delta_f^* + \Delta\delta) - \dot{\gamma}_d \\ &= a_{21} \bullet v + a_{22} \bullet \gamma + b_{21} \bullet \delta_f^* + b_{21} \bullet \Delta\delta - \dot{\gamma}_d \end{aligned} \tag{15}$$

where $a_{21} = -\dfrac{aC_{af} - bC_{ar}}{Iu_c}$, $a_{22} = -\dfrac{a^2C_{af} + b^2C_{ar}}{Iu_c}$, $b_{21} = \dfrac{aC_{af}}{I}$.

In this paper, the specific trajectory of the system convergence to sliding surface is defined with the constant rate reaching law as (16).

$$\dot{s} = -\varepsilon \bullet \mathrm{sgn}(s), \quad \varepsilon > 0 \tag{16}$$

where ε represents the system rate approaching to the sliding surface in constant velocity law.

It is obvious that with such reaching law, the accessibility condition of SMC control $\varepsilon \cdot \dot{\varepsilon} < 0$ can be satisfied. The corresponding control for the system compensation can be described as (17).

$$u_c = \Delta\delta = -\frac{a_{21}}{b_{21}} \bullet v - \frac{a_{22}}{b_{21}} \bullet \gamma - \delta_f^* + \frac{1}{b_{21}} \dot{\gamma}_d - \frac{1}{b_{21}} \varepsilon \bullet \mathrm{sgn}(s) \tag{17}$$

IV. Results and Analysis

To examine the effectiveness of the designed controllers, simulations and experiments are carried out correspondingly, which are presented as below.

A. IMC controller simulation

Based on the designed IMC controller and dynamic model, corresponding Simulink computation model is built and CarSim is used for setting up the simulation test conditions. In this subsection, the result in double lane change maneuver is presented for closed-loop simulation tests. A PID controller, with motor voltage U_{sm} as controller output, is used for the same comparison. All parameters of the SBW actuator system used in simulation are presented in Table I.

TABLE I PARAMETERS OF STEERNG MOTOR AND MACHANISM

Parameter	Symbol / Unit	Value
Moment of inertia of motor	$J_{sm}\ /\ kg \cdot m^2$	0.00078
Damping coefficient of motor	$B_{sm}\ /\ kg \cdot m^2 \cdot s^{-1}$	0.00023
Steering motor reduction ratio	g_{sm}	20
Motor resistance	$R_{sm}\ /\ \Omega$	0.51
Motor inductance	$L_{sm}\ /\ H$	0.00033
Back electromotive force coefficient of motor	$K_2\ /\ V \cdot s \cdot rad^{-1}$	0.056
Electromagnetic torque coefficient of motor	$K_{sm}\ /\ N \cdot m^{-1}$	0.056
Pinion gear torsional stiffness	$K_{md}\ /\ M \cdot m \cdot rad^{-1}$	180
Steering pinion radius	r_p / m	0.008
Steering rack mass	$M_r\ /\ kg$	2.31
Damping coefficient of steering rack	$B_r\ /\ N \cdot m \cdot s \cdot rad^{-1}$	642

Taking double lane change maneuver as a typical test for evaluating vehicle stability performance, simulations are carried out in this study. Anti-interference capability of controller is focused on external disturbance from road surface continuously acting on tires while driving. For IMC and PID controllers, the driver's preview time is taken as 0.5*s* and the vehicle speed is set as 30*km/h*. Also, considering the white noise directly added to rack displacement to simulate external disturbance, the reference trajectory obtained by driver model is contrasted with the result after IMC/PID control. The results for both controllers are shown in Fig.7 and Fig.8.

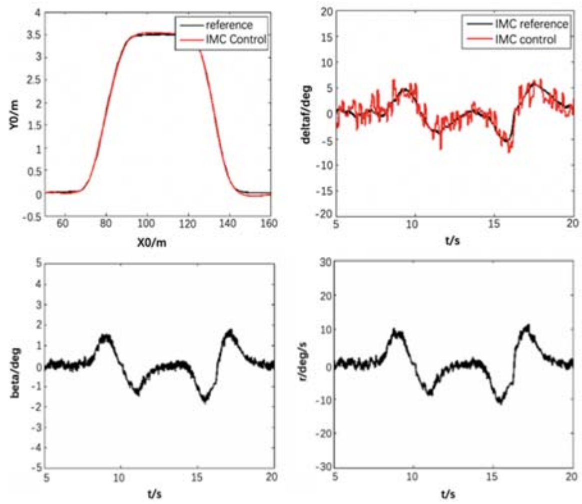


Fig.7. Vehicle states by IMC control

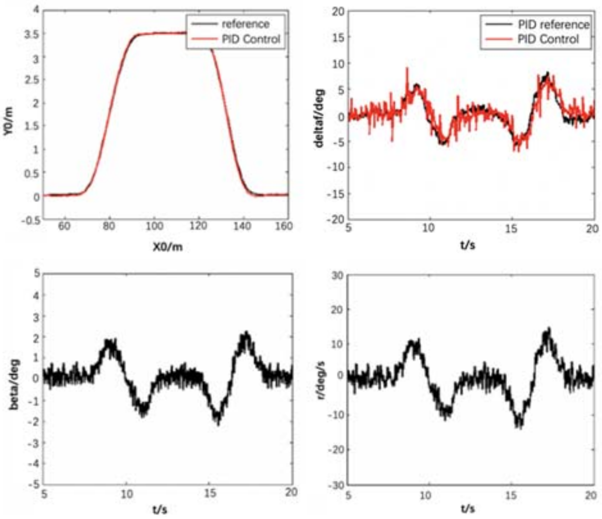


Fig.8. Vehicle states by PID control

The results show that the IMC can provide more robustness to resist disturbances and maintain system stability. While, although trajectory tracking performance by using PID controller is also satisfactory, vehicle response, e.g. yaw rate and side slip angle, fluctuate more seriously. Since the reference front wheel angle can be used to measure driver's effort to eliminate the impact caused by disturbance, the designed IMC shows better performance than PID control, which fluctuates more seriously, thus leading to more effort for steering corrections.

In order to verify the robustness of IMC controller to parameters perturbation in SBW system, the actuator parameters, i.e., J_{sm} , K_{md} , r_p , are allowed to variation within $\pm 10\%$ of nominal values shown in Table 1. Then, the resultant responses are obtained with three different sets of parameters. Here, the preview time of driver is $0.5s$ and vehicle speed is set as $30km/h$. The results are compared in Fig.9, showing that even the system parameters change, the outputs of the system changes little. This means the designed IMC controller is not sensitive of the changes of system parameters.

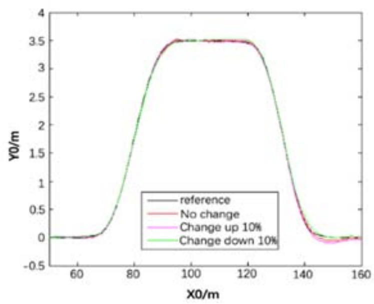
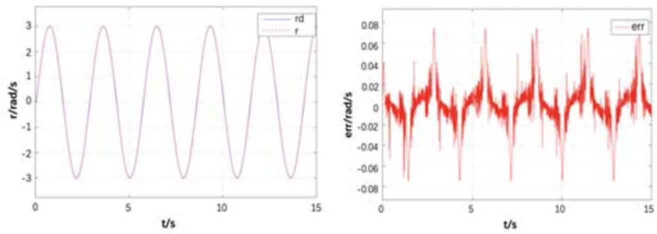


Fig.9. Tracking performance with different sets of parameters

B. Simulation results of SMC controller

In order to examine the designed SMC controller, the open-loop tests with sinusoidal input signals from the driver, shown as (18). The vehicle parameters are set based on SAIC ROEWE and simulation results are presented in Fig.10.

$$\delta_f^* = \pi \bullet \sin(0.7\pi t) \tag{18}$$



a. Yaw rate response b. Error of yaw rate

Fig.10 Simulation result with sinusoidal inputs from steering wheels

The results demonstrate that the designed SMC controller can track the ideal yaw rate satisfactorily, but the fluctuation is severe. The reason can be that system crosses the sliding surface frequently to change the compensation, resulting in high frequency variation of the error. Hence, in order to minimize the system fluctuation, the saturation function sat(s) defined as (19) is chosen instead of the switching function [17] to solve this problem. The approach law can be rewritten as (20). The compensation of control system using saturation function is shown in Fig.11 and the simulation results are shown in Fig.12.

$$\mathrm{sat}(s)=\begin{cases} s/\delta_1, & if\ |s| < \delta_1 \\ \mathrm{sign}(s), & if\ |s| \geq \delta_1 \end{cases} \tag{19}$$

$$u_c = \Delta\delta = -\frac{a_{21}}{b_{21}} \bullet v_y - \frac{a_{22}}{b_{21}} \bullet \gamma - \delta_f^* + \frac{1}{b_{21}}\dot{\gamma}_d - \frac{1}{b_{21}}\varepsilon \bullet \mathrm{sat}(s) \tag{20}$$

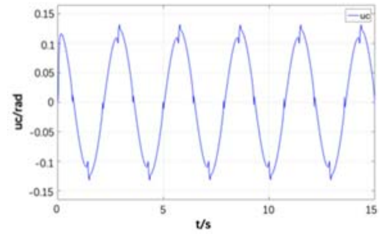
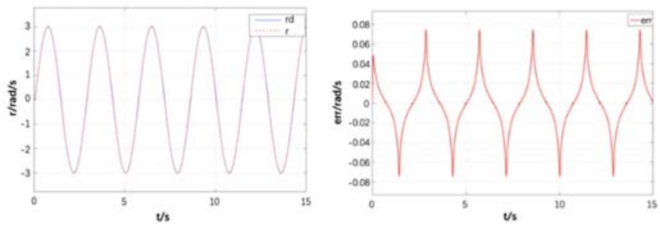


Fig.11 Compensation with saturation function



Yaw rate response b. Error of yaw rate

Fig.12 Simulation result of sinusoidal input with saturation function

From Fig.11 and 12, it can be shown that the ideal yaw rate of the vehicle can be tracked, satisfactorily with good control precision by using the saturation function for SMC. Compared Figs.10(b) and 12(b), the fluctuation of the system on the sliding surface is significantly reduced.

For further verification of designed controllers, co-simulations, including IMC and SMC controller, are further carried out, illustrated by Fig.13.

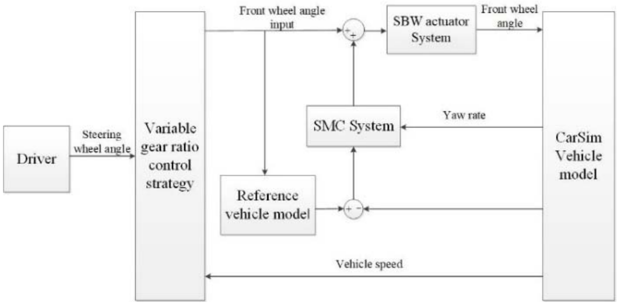
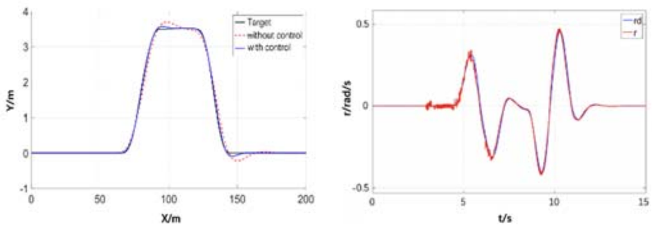


Fig.13. Co-simulation control diagram

By using variable ratio control strategy, which is more effective in SBW car, the gear ratio is adjusted according to the feedback of current speed from CarSim and sequentially converted into a front wheel angle input. Simultaneously, the front wheel angle compensation value is output by the SMC controller based on yaw rate error. The SBW actuator system track the signal of the corrected input front wheel angle with IMC controller. Still, double lane change is taken as example here and simulations under different speeds have been completed. The representative simulation results at the speed of 48*km*/*h* are shown in Fig.14.



a. Trajectory tracking result

b. Yaw rate tracking result

Fig.14. Simulation results of DLC trajectory

It can be seen from the Fig.14 that the designed controllers can track the desired vehicle yaw rate satisfactorily and reduce the lateral offset in double lane change maneuver, with improved vehicle stability performance.

V. Conclusion

In this paper, an IMC controller is designed for the SBW system with identified parameters by bench-testing in order to achieve desired tracking performance for front steering wheel angles. Corresponding simulation cases, including open-loop and closed-loop, are carried out. By comparing the control effects between the IMC and a PID controllers, the results show that the designed IMC in this study is more effective in trajectory tracking, anti-interference and reducing driver steering workload. In order to improve vehicle steering stability when cornering, considering both cases of steering maneuver in low-speeds and stability in high-speeds, a variable steer gear ratio control scheme is proposed, and a SMC controller is designed to track vehicle ideal yaw rate accordingly. The simulation results show that the designed SMC can track the desired vehicle yaw rate satisfactorily, with enhanced vehicle steering stability hereby.

Acknowledgment

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