Control Strategies for Improving Ground Vehicle Stability

State-of-the-art Review

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Abstract— Vehicle stability control system has evolved significantly over the past three decades. Several control strategies have been implemented to enhance the vehicle stability control performance. However, most of these control strategies have not been focused on improving the transient response for vehicle yaw rate and sideslip angle tracking controller in the presence of uncertainties and disturbance. This paper is aimed at reviewing the proposed control algorithms based on their control objectives, different types of actuators required for the vehicle stability control system, advantages and disadvantages. According to this review, we derive that the advanced type of sliding mode control with non-singular fast terminal sliding surface has an ideal capacity to provide a fast transient response of yaw rate and sideslip tracking controllers under the presence of disturbances.

Keywords—control strategy; non-singular fast terminal sliding surface; transient response; vehicle stability control; yaw rate-sideslip tracking controller

I. INTRODUCTION

Since the beginning of 1980s, various active vehicle safety systems have been investigated to improve the dynamics of the vehicle. By understanding of the vehicle state, active safety systems assist a driver to avoid and minimize the effects of a crash. One of the main systems designed for vehicle directional control is vehicle stability control (VSC) system. A vehicle has three directions of motions, namely longitudinal, vertical, and lateral motions. In the vehicle stability control, controlling the lateral dynamics of the vehicle plays a significant role when it comes to stabilizing the vehicle directional motion. A vehicle is directionally stable if, in spite of existing disturbance, the vehicle returns to a steady-state regime within a finite time. On the contrary a vehicle which is directionally unstable is not able to converge into the original path, even after the disturbance is removed [1].

A time interval between the driver's command inserting by steering and / or braking actuators and the vehicle steady-state response is referred to as transient response. Since the vehicle directional motion stability extremely depends on its transient response, a vehicle is expected to have the fastest response in the process of reaching to the steady-state motion.

As a result, an effective VSC strategy is required to enhance the transient response so that the vehicle is able to follow the desired motion characteristics in a fast performance.

In formulation of vehicle dynamics behavior, a discrepancy always exists between the actual vehicle model and its mathematical model used for the controller design. Considering the nonlinearity of tire forces, vehicle motion is represented as a nonlinear system, especially during a severe manoeuvre. Moreover, an external disturbance such as lateral crosswind may have an effect on controlling the vehicle. Therefore, designing control laws that provide the fast transient performance to the closed-loop system in the presence of these disturbance and uncertainties is a challenging task for vehicle directional stability control.

This review study begins with the control objectives in section II. The main types of the VSC system are discussed in section III and followed by reviewing the control strategies and problems in section IV. In section V, a robust control method using fast terminal sliding mode control is discussed and ended with conclusion in section VI.

II. CONTROL OBJECTIVES

In vehicle stability control system, vehicle yaw rate and vehicle body sideslip angle, which is the deviation angle between the vehicle longitudinal velocity and its motion direction, are both used as control objectives for evaluating the lateral (cornering) directional behavior of the vehicle [1]. The reason why these two parameters are critical to control the lateral dynamics of the vehicle is that yaw rate control helps the vehicle to maintain the desired rate and direction of rotation about its vertical axis. However, yaw rate control alone is not sufficient for keeping the vehicle moving along the desired path. For instance, if the tire-road friction coefficient is small or if the vehicle speed is too high, controlling the yaw rate can only maintain the vehicle in the intended orientation, but the vehicle sideslip angle may increase considerably causing the vehicle to deviate significantly from its desired path [1]. Consequently, forcing both the yaw rate and sideslip angle to follow their desired values is essential for controlling the lateral dynamics of the vehicle as shown in Fig. 1.

Fig. 1. Vehicle directional behavior (1) On high friction road (2) On slippery road with both yaw rate and sideslip angle control (3) On slippery road with yaw rate control only [2].

In the steady state condition, the desired yaw rate response can be expressed by the following equation:

$$\dot{\psi}_{des} = \frac{V_x}{(l_f + l_r) + K_u V_x^2} \delta_d \quad (1)$$

where the understeer coefficient K_u is defined base on the vehicle parameters as follows:

$$K_u = m \left(\frac{l_r C_r - l_f C_f}{2(l_f + l_r) C_f C_r} \right) \quad (2)$$

In some studies [3]-[7], the desired sideslip angle is simply considered as zero. However, in order to obtain a better approximation, (3) can be used to calculate the desired value of the vehicle sideslip angle. Therefore, improving the stability of the vehicle depends on the capability of the controller to follow the desired values in the fast transient performance.

$$\beta_{des} = \frac{l_r - \frac{l_f m V_x^2}{2C_r(l_f + l_r)}}{(l_f + l_r) + \frac{m V_x^2 (l_r C_r - l_f C_f)}{2C_f C_r (l_f + l_r)}} \delta_d \quad (3)$$

III. TYPE OF VEHICLE STABILITY CONTROL SYSTEM

There are two main types of vehicle stability control system for improving the lateral dynamics of the vehicle. The first type is active steering (AS) system which modifies the driver's steering command and adds a correction angle to front / rear wheels. The second type is direct yaw moment control (DYC), which can be implemented by active differential braking (ADB) or active torque disturbance (ATD) systems. The DYC system generates the required yaw moment to enhance the vehicle lateral dynamics.

A. Active Steering

The active steering control assists the driver in improving the vehicle lateral characteristics through the correction of steering wheel angle as shown in Fig. 2. In general, active steering control can be categorized in three types including active front steering (AFS), active rear steering (ARS), and four-wheel active steering (4WAS). Since the steering angle limitation of rear wheels, ARS system is an effective method to control the vehicle at low speed cornering maneuvers. In order to enhance the vehicle lateral response at low speed and high speed, combination of AFS and ARS called 4WAS can be implemented. The main issues of the AS systems are their inherent nonlinear characteristics and tire saturation with respect to tire slip angles. As a result, in the nonlinear range of tire lateral forces with respect to tire side-slip angles, the aimed control ability is drastically reduced.

B. Direct Yaw Moment Control

The direct yaw moment control improves the vehicle lateral performance by generating the yaw moment produced by the longitudinal force on the individual tires as illustrated in Fig. 3. This longitudinal force can be obtained from the disturbance of driving or braking torque (ATD or ADB) on the right and left sides of the vehicle.

The DYC system is effective in both linear and nonlinear regions of the tire lateral force. However, regarding the ADB, continuous use of the braking force causes the vehicle to slow down significantly, so it has an undesirable effect on the vehicle longitudinal motion. As for ATD, if the coefficient of road friction is small or if the vehicle speed is too high, the active torque distribution system may not afford adequate yaw moment to response the driver needs.

C. Integrated Vehicle Dynamics Control

Recent approaches [8], [9] to the above-mentioned problems of the AS and DYC systems recommended that the maximum benefits could be gained through the integrated use of these two systems (Fig. 4), which can be referred as an integrated vehicle dynamics control (IVDC). From the integrated vehicle dynamics control point of view, a comparative study of AFS, ARS, and 4WAS coordinated with DYC concludes that the AFS is more suitable to be integrated with DYC than the other two active steering systems, ARS and 4WAS, in terms of improving the vehicle stability [10].

Fig. 2. Active steering control system structure

Fig. 3. Direct yaw moment control system structure

Fig. 4. Integrated vehicle dynamics control system structure

The strategy of the IVDC system is that the AFS system can be served to control the yaw rate within the handling region of the tire lateral force and the DYC system is activated when the vehicle exceeds the linear region limit. Therefore, by using steering and braking actuators as control inputs, the lateral and yaw motions of the vehicle can be controlled. As a result, the IVDC system is able to enhance the stability of the vehicle by controlling the main two control objectives, yaw rate and sideslip angle, effectively in severe cornering maneuvers and the steady state driving condition as reported in [5], [11]-[15].

The following section will review the proposed control strategies for the tacking control of yaw rate and sideslip angle based on the types of actuators used in a VSC system.

IV. VEHICLE STABILITY CONTROL STRATEGIES

Vehicle stability implementation by active steering systems can be seen in many research papers, mostly active front steering system.

In [16], feed-forward and feedback H_∞ robust control are implemented to stabilize the vehicle yaw motion where speed and road adhesion variations are considered as uncertainties and disturbance. The control feedbacks the yaw rate sensor and adds a feed-forward action as a function of the driver command to the front wheel.

Control synthesis is implemented on a linear fractional transformations (LFT) model in order to represent the variation of vehicle speed and road adhesion. The feed-forward and feedback gains for H_∞ controller are found through the numerical optimization.

A passivity-based adaptive nonlinear controller for active front steering system is suggested in [17]. The authors claimed that the complexity of adaptive nonlinear control is reduced by using the passivity-based method. The designed controller exhibits high adaption and robustness to various characteristic changes of a vehicle.

A model predictive control (MPC) approach for controlling an active front steering vehicle is presented in [18]. The MPC controller is formulated based on successive online linearization (linear time varying – LTV) of the nonlinear vehicle model in the discrete time domain. The optimization problem of the designed MPC controller is solved online using Matlab QP solver. The simulation results prove the effectiveness of the proposed control system as long as the trajectory of the vehicle is known over a finite horizon.

Development of the AFS by quantitative feedback theory (QFT) is proposed in [19] in order to compensate the yaw rate response in presence of uncertain parameters and reject the disturbance. The uncertain parameters considered in the presented study are variation of the vehicle velocity from 72

km/h to 108 km/h and 10% variation of the vehicle mass about the nominal value. A multi DoF nonlinear vehicle model (ADAMS car model) is used to evaluate the performance of the controller. Under various maneuvers and road conditions, the control system has robust performance to improve the vehicle dynamics.

A proportional derivative (PD) controller is implemented for active front steering vehicle in [7]. Strength of this research is that a sideslip observer is utilized in controlling the vehicle yaw rate. The novel linear observer has minimum estimation error compared with conventional linear observer. A gust wind considered as a disturbance for the controller, but there is no consideration of nonlinearity of the vehicle model.

In [20], authors worked on designing a gain-scheduled controller for the AFS system equipped with a steer-by-wire system. The proposed controller achieved the decoupling of the lateral and yaw motions of a vehicle and its yaw damping by the feedbacks of both yaw rate and front steering angle. The feedback gains are adjusted with respect to vehicle speed by using trade-off method between the robust decoupling and yaw rate damping. To evaluate the performance of the controller, the single lane change maneuver is conducted on the vehicle with speeds of 64, 80, 96 km/h. The experimental results indicate that the controller has a significant effect on the vehicle stability in critical situations.

A yaw moment controller based on fractional order PID controller for the AFS system is proposed in [21]. The fractional order PID controller is the expansion result of a conventional integer PID controller based on the fractional calculus. The parameters of the fractional order system are realized by the Oustaloup filter method. The simulations are performed with step signal and single lane change maneuvers on the vehicle moving with speeds of 40 and 120 km/h. The results claim that the controlled vehicle has a better performance in comparison with the uncontrolled vehicle.

In [22], authors of [21] changed the type of controller to fuzzy-logic control (FLC). The feedbacks of yaw rate and sideslip angle are considered as inputs of the controller. The control inputs are fuzzified into five fuzzy sets and outputs variables have seven fuzzy sets. The logic of the fuzzification is based on the Mamdani Fuzzy Interface system and the defuzzification is the center of area rule. The simulation results are obtained through the same maneuvers in [21], but the speeds of the vehicle are 72 km/h and 108 km/h in dry and icy roads, respectively. The simulation outcomes show that the controlled vehicle is able to trace the desired response by using the proposed fuzzy-logic controller.

Controlling of the vehicle stability by fuzzy logic based on the AFS system is also presented in [23]. In this method, the yaw rate error and its derivative are considered as controller inputs. The performance of designed controller is implemented on an 8-DoF nonlinear vehicle model and verified through simulation. The fuzzy rules are similar to [22]. The simulation results, which are based on double lane change maneuver on the vehicle with speed of 108 km/h, prove the capability of the proposed method to control the yaw stability of the vehicle.

Designing a controller for the AFS system based on the conventional sliding mode control (SMC) method is proposed in [24]. A nonlinear 8-DoF model incorporating Magic Formula tire model is used to model the nonlinear controlled plant. The AFS system is designed based on a steer-by-wire system. The steer-by-wire system is modeled using the bond graph method to consider all mechanical and electrical components of the system. Two additional controllers are adopted; one for producing driver's steering feel based on the PID control, and the other one is based on the sliding mode control in order to generate the commanded angle of front wheel through a brushless motor. The proposed control system guarantees high robustness for changing condition of vehicle mass and various severe cornering maneuvers.

Direct yaw moment control method is also widely used by researchers. In [25], authors proposed a DYC system in which the sliding mode method is used to control the vehicle sideslip angle, which is estimated through online real time calculation. They claimed that control of the sideslip angle by DYC system has a distinguished effect on compensation of vehicle lateral dynamics for unstable motion because of nonlinear tire characteristics.

A DYC system including a feed-forward controller, a state feedback controller, and an estimator for the vehicle body sideslip angle is designed in [26]. The corrective yaw moment is considered as a control output and the front wheel steer angle is regarded as a disturbance. The feedback and feed-forward controller gains are computed by using the linear quadratic regulator (LQR) theory. The simulation results for a J-turn maneuver where the vehicle runs at a constant speed of 60 km/h in a dry road conclude that the LQR provides good tracking capability of the yaw rate to follow its reference value.

In [27], the vehicle stability is improved by using an optimal control law for yaw rate control by DYC system. To model the vehicle dynamics, a 2-DoF vehicle model is used. In order to determine the values of the feedback and feed-forward control gains, a LQR problem is formulated. The authors designed two versions of optimal yaw moment control strategies including the semi-optimal and fully-optimal controllers. Comparing these two types of optimal controllers, they concluded that the semi-optimal law in the linear vehicle dynamics regime is more appropriate due to a simpler structure with more satisfactory performance. However, the influence of tire nonlinearity is not considered in the vehicle dynamics model.

Improvement of the DYC system based on generalized predictive control (GPC) method for a hybrid brake-by-wire equipped vehicle is presented in [28]. The designed controller strategy is the prediction of the future yaw rate of the vehicle and then inserting control action at presented time based on future yaw rate error. Through the slalom maneuver at high speed, the experimental results indicate that the vehicle can be stabilized in both oversteer and understeer conditions on a packed snow road using the predictive controller.

Authors in [4] proposed a cascade structure of control system including a yaw moment major controller and a wheel

slip minor controller for DYC system. The yaw rate controller is designed by using the LQR theory and the sliding mode method is applied for the wheel slip controller. An 8-DoF vehicle dynamics model is presented as a simulation plant and a 2-DoF vehicle model is used as a controller model. The first goal of the DYC system is to make the actual yaw rate to track its desired value. The second purpose is to minimize the vehicle sideslip angle into an acceptable region to prevent vehicle from spinning. The corrective yaw moment is generated by using differential braking through regulating the wheel slip ratio. The simulation is conducted for J-turn and single lane change maneuvers on dry and wet roads, respectively. The stability of the vehicle is significantly improved with the proposed DYC system.

In [29], an optimal yaw rate tracking law is introduced to design the DYC system. A linear yaw rate model limited by tire-road conditions is considered as a desired model to be tracked by the controller. The effectiveness of the designed controller is compared with a sliding mode controller through simulation of various maneuvers. The outcomes demonstrate that an appropriate stability performance through a reduced external yaw moment is achieved by using the optimal yaw rate control. The control algorithm is formulated by linearizing the vehicle model, which is then discretized via a bilinear transformation.

Over the past two decades, integrated vehicle dynamics control has been a significant research topic in the area of vehicle dynamics and control. Comprehensive reviews on this research area may refer to [30] and [31]. Various control techniques have been designed to achieve the aim of functional integration of the vehicle dynamics control systems.

A revised model regulator as the main controller that utilizes coordinated steering and individual wheel braking actuation is presented in [32]. Although the simulation results prove the proposed claims, the model regulator is not effective enough to control the vehicle because of the plant changes (i.e. changes in vehicle speed and mass, road conditions), model uncertainties (i.e. in the parameters of the model) or nonlinearities (i.e. the real system is not actually linear). Therefore, the resulting controller will degrade and the system may become unstable. In [33], the authors also designed a robust model regulator based on mapping frequency-domain criteria for controller parameter space to control the vehicle yaw stability. The controller performance is only useful on linear region of vehicle dynamics. As a result, the proposed system does not work properly when the tire lateral force is saturated.

In [15], an IVDC system including driveline-based, brake-based, and driveline plus brake-based DYC subsystems is implemented to coordinate of AFS and DYC systems. These subsystems are developed separately, and then proposed a switching strategy between the stand-alone systems, according to the driving situations. Consequently, the internal system stability may be in question to the switching process.

In [11], the authors compared the optimal guaranteed cost coordination control (OGCC) method with the optimal coordination (OC) scheme based on LQR theory for integrated

AFS and DYC systems. The tire cornering stiffness is treated as an uncertainty in various driving conditions. The simulation carries out on an 8-DoF nonlinear vehicle model for slalom and single lane change maneuvers. The results illustrate that the OGCC has better stability and tracking performance compared with the OC scheme.

As discussed in [34], sliding mode control is implemented for stabilizing the forces and moments in integrated control algorithm that coordinated steering, braking, and stabilizer. The control consists of main loop and servo loop controllers. The main loop controller calculates the corrective forces and moments and then the servo loop controller optimally distributes the generated forces and moments to tire control inputs. The simulation results verify that the presented control algorithm can effectively enhance vehicle stability performance.

In the studies [12], [13], and [35], the control algorithm consisting of reference model based on linear parameter-varying (LPV) formulation and static-state feedback controller with the purpose of ensuring the robustness of integrated AFS and DYC system is presented. The tire slip angle, longitudinal slips, and vehicle forward speed are considered as uncertain parameters. It is claimed that if all system states are available for feedback then robust static-state feedback controller can stabilize the vehicle during cornering maneuvers.

In [36], a switched model predictive controller (SMPC) and in [14] and [37], a nonlinear model predictive controller (NMPC) are proposed to coordinate AFS and DYC. The main disadvantage of the model predictive control method is that it is not able to explicitly deal with plant model uncertainties. An additional problem is that online calculation for optimization may cause implementation difficulties.

An integrated robust model-matching controller is designed in [38] using the H_∞ controller based on LMI (linear matrix inequalities). The controller integrates active rear steering, longitudinal force compensation and active yaw moment controls. The effectiveness of the proposed controller on the vehicle handling performance is evaluated under crosswind disturbance. The simulation outcomes confirm that the designed controller achieves acceptable stability performance.

Authors in [39] designed an integrated control strategy involving AFS and DYC based on a gain-scheduled LPV controller which is able to improve stability in a four-wheeled vehicle. The controller is synthesized within the LMI framework, while warranting robust H_∞ performance. The main disadvantage of the gain-scheduled LPV control scheme is that this control method focuses on appropriate formulation rather than actual controller design.

Through the above literature survey, it can be concluded that robust control methods such as SMC, MPC, GPC, NMPC, SMPC, FLC, QFT, H_∞ and OGCC are indispensable to solve the uncertainty and disturbance issues that affect the performance of vehicle stability control system. The proposed studies confirm that various designed controllers can follow the desired yaw rate and vehicle body sideslip angle response

in the presence of external disturbances and system uncertainty.

The prior proposed control strategies can be categorized based on VSC system types, control objectives, advantages and disadvantage as summarized in TABLE I. When it comes to controlling nonlinear systems, the main factor of tracking control is the transient response performance of the controller. However, the above-discussed control strategies are not able to meet the need of fast transient response improvement of the vehicle stability incorporated uncertainties caused by variation of dynamics parameters such as vehicle mass, speed, body moment of inertia and road surface adhesion coefficient and also disturbance such as lateral crosswind. In conclusion, an appropriate control algorithm, which could enhance the transient performance of robust yaw rate and sideslip tracking control, should be designed. One particular approach to robust controller design is the so-called fast terminal sliding mode control.

V. FAST TERMINAL SLIDING MODE CONTROL

The sliding mode control is one type of variable structure control methods. The concept of variable structure control with sliding mode was first proposed in the early 1950s in the Soviet Union [40]-[42]. The most prominent feature of the variable structure control method is its robustness against system parameter variations and external disturbance. The sliding mode control is designed to drive and then restrict the system states to lie within a neighborhood of the switching manifolds which can be prescribed.

Through the sliding, the closed-loop response becomes completely insensitive to both internal parameter uncertainties and external disturbance. However, in conventional sliding mode control (i.e. first order, higher order, integral sliding mode), the system states converge to the equilibrium point asymptotically but not in finite time. In the early 1990s, the terminal sliding mode (TSM) control was developed [43]-[46] that can guarantee the finite reaching time to the equilibrium.

However, in comparison with the conventional sliding mode control, the existing TSM control does not perform effectively when the system state is far away from its equilibrium. Therefore, at the beginning of 21st century, a fast terminal sliding mode (FTSM) controller was proposed [47]-[49]. The FTSM control is able to combine the advantages of the TSM control and the conventional SMC together so that the fast transient convergence both at a distance from and at a close range of the equilibrium can be achieved. According to [50], the prior TSM and FTSM methods both suffer from a singularity problem. As a result, a non-singular fast terminal sliding mode control (NFTSM), which is able to overcome the singularity completely, was proposed. The outcome of the study [50] indicates that the presented NFTSM control method provides some superior such as fast finite-time convergence, strong robustness and singularity avoidance.

A. The Fast Terminal Sliding Mode Concept

The TSM concept can be described as, [46]

$$s = \dot{x}_1 + \beta' x_1^{q/p} = 0 \quad (4)$$

TABLE I. VEHICLE STABILITY CONTROL STRATEGIES

Control Strategy	VSC System Type	Control Objective	Advantages	Disadvantages
PD	AFS	Yaw rate	Easy software implementation	Uncertainties are not considered
Fractional PID	AFS	Yaw rate	Low percentage overshoot and small settling time	Online optimization of fractional order
MPC	AFS	Yaw rate	Robust for uncertainties	Online optimization
GPC	DYC	Yaw rate	Robust for uncertainties	Online optimization
SMPC	IVDC	Sideslip, Yaw rate	Robust for uncertainties	Online optimization
NMPC	IVDC	Sideslip, Yaw rate	Robust for uncertainties	Online optimization
SMC	AFS, IVDC	Sideslip, Yaw rate	Robust for uncertainties and reject disturbance	Fast transient response improvement is not considered
H_∞	AFS	Yaw rate	Robust for uncertainties and reject disturbance	Difficult software implementation
OC	DYC	Yaw rate	Robust for uncertainties	Online numerical parameter calculation
OGCC	IVDC	Sideslip, Yaw rate	Robust for uncertainties	Online numerical parameter calculation
LMI static-state feedback + H_∞	IVDC	Sideslip, Yaw rate	Robust for uncertainties	Availability of all system states and online optimization
FLC	AFS	Yaw rate	Robust for uncertainties	Transient response improvement is not considered
QFT	AFS	Yaw rate	Robust for uncertainties and reject disturbance	Transient response improvement is not considered
Passivity- based adaptive nonlinear	AFS	Yaw rate	Robust for uncertainties	Numerical parameter estimation
SMC+LQR	DYC	Yaw rate	Robust for uncertainties	Transient response improvement is not considered

where $x_1 \in \mathbb{R}^1$ is a scalar variable, $\beta' > 0$ and p, q ($p > q$) are odd positive integers. It can be easily verified that, the time to reach the equilibrium, $x_1 = 0$, is

$$t_s = \frac{p}{\beta'(p-q)} |x_1(0)|^{\frac{(p-q)}{p}} \quad (5)$$

The nonlinear term $x_1^{q/p}$ improves the convergence toward equilibrium. When the system states are close to equilibrium, the convergence rate is faster resulting in finite time convergence. However, when the system states are far away from the equilibrium, the TSM cannot dominate the linear counterpart since the term $x_1^{q/p}$ tends to reduce the magnitude of the convergence rate at a distance from equilibrium. To solve this problem, the fast terminal sliding surface is introduced as

$$s = \dot{x}_1 + \alpha'' x_1 + \beta'' x_1^{q/p} = 0 \quad (6)$$

where $\alpha'', \beta'' > 0$. Therefore, it can be concluded that

$$\dot{x}_1 = -\alpha'' x_1 - \beta'' x_1^{q/p} \quad (7)$$

The physical interpretation of above equation is: when x_1 is far from the equilibrium (i.e. $x_1 = 0$), the approximate dynamic of (7) becomes $\dot{x}_1 = -\alpha'' x_1$ whose fast convergence is well understood [49]. When x_1 is close to equilibrium, the

approximate dynamic becomes $\dot{x}_1 = -\beta'' x_1^{q/p}$ which is a terminal attractor. The reaching time to the equilibrium can be derived as

$$t_s = \frac{p}{\alpha''(p-q)} \ln \frac{\alpha'' x_1(0)^{(p-q)/p} + \beta''}{\beta''} \quad (8)$$

According to [49], the fast convergence performance of FTSM compared with the conventional sliding mode can be verified through the following example. Consider $\alpha'' = 1$, $\beta'' = 1$ and initial condition $x_1(0) = 1$. First, assuming $p = 3$ and $q = 1$, the reaching time to equilibrium is $t_s = 1.03972077083992$. In comparison with the situation where $p = q = 1$, the simulation suggests that at approximation $t_s = 1.039699999999990$, for the case of $p = 3$ and $q = 1$, $x(t_s) = 0.00000009178540$ and for the case of $p = 1$ and $q = 1$, $x(t_s) = 0.12500519281775$.

It can be concluded that the convergence rate of the FTSM is far better than its linear counterpart because the convergence of the conventional sliding mode remains a constant while the convergence rate of the FTSM grows exponentially.

B. Singularity Problem

Consider the following second-order nonlinear system [48]

$$\begin{aligned}\dot{x}_1 &= x_2 \\ \dot{x}_2 &= f(X) + g(X) + b(X)u\end{aligned}\quad (9)$$

where $X = [x_1 \ x_2]^T \in \mathbb{R}^2$ are states vector, $u \in \mathbb{R}^1$ is the control input, $f(X)$ and $b(X) \neq 0$ are nonlinear functions $\mathbb{R}^2$, $g(X)$ is disturbance and uncertainty, which is $|g(X)| \leq l_g$. If the FTSM switching surface is defined as (6), then the control law for the system of (9) is expressed as

$$u(t) = -\frac{1}{b(X)} \left(f(X) + \alpha'' x_2 + \beta'' \frac{q}{p} x_1^{\frac{q}{p}-1} x_2 + \dots \right. \\ \left. k_1'' \text{signs} + k_2'' s \right) \quad (10)$$

in which $k_1'' > 0$, $k_2'' > 0$.

In FTSM control (10), it is obvious that the term containing $x_1^{\frac{q}{p}-1} x_2$ may cause singularity to occur if $x_2 \neq 0$ when $x_1 = 0$. This singularity problem may occur in the reaching phase when there is insufficient control to ensure that $x_2 \neq 0$ while $x_1 = 0$. In order to overcome the singularity problem in the FTSM control, authors in [50] proposed a new switching surface called non-singular fast terminal sliding surface as follows,

$$s = x_1 + \alpha''' \text{sign}^{\gamma_1} x_1 + \beta''' \text{sign}^{\gamma_2} x_2 = 0 \quad (11)$$

where $\alpha''' > 0$, $\beta''' > 0$, $1 < \gamma_2 < 2$, $\gamma_1 > \gamma_2$ and $\text{sign}^{\gamma_i} x_i = |x_i|^{\gamma_i} \text{sign}(x_i)$. To make the system in (9) reach the equilibrium fast along the switching surface (11), the control input, u , is chosen as

$$u = -\frac{1}{b(X)} \left[\frac{1}{\beta''' \gamma_2} \text{sign}^{2-\gamma_2} x_2 \cdot (1 + \alpha''' \gamma_1 |x_1|^{\gamma_1-1}) + \dots \right. \\ \left. f(X) + k_1''' \text{signs} + k_2''' s \right] \quad (12)$$

in which $k_1''' > 0$, $k_2''' > 0$.

From the system in (9), if the switching surface is chosen as (11), and controlled by (12), then the system trajectory will converge fast to zero within finite time. Moreover, no occurring of singularity is ensured during the whole process.

VI. CONCLUSION AND FUTURE WORK

The aim of this study is reviewing the proposed control strategies for the vehicle stability control system. Since the vehicle has two main actuators called steering and braking subsystems, there are two types of vehicle stability control systems and each of these actuators or integration of them can be implemented to control the stability of the vehicle.

As the yaw rate and sideslip tracking control performance are affected by uncertainties and external disturbance in actual vehicle model, the controller should be robust enough to improve the vehicle lateral performance during severe cornering maneuvers. According to this review, it can be concluded that sliding mode control approach is the best

robust control strategy to overcome the effect of uncertainties and disturbance. The response of tracking controllers is really dependent on transient performances. However, in the existing conventional SMC algorithm, the convergence rate does not occur in the finite time. To solve this problem, a non-singular fast terminal sliding surface is proposed. This advanced type of SMC guarantees the fast transient response so that the system state will converge to the equilibrium in finite time. For future work, non-singular fast terminal sliding mode control strategy will be implemented for integrated vehicle dynamics control system and the theoretical analysis of transient response of the yaw rate and sideslip angle tracking controller in the presence of vehicle uncertainties and external disturbance will be evaluated and compared with conventional SMC.

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NOMENCLATURE

C_f, C_r	: Front and rear cornering stiffness of tire
l_f, l_r	: Distance from the CG to the front and rear axle
m	: Mass of the vehicle
V_x	: Longitudinal vehicle velocity
δ_d	: Driver front wheel angle
δ_c	: AS corrective wheel angle
M_{zc}	: Corrective yaw moment
β	: Body side-slip angle
ψ	: Yaw angle
γ	: Yaw rate