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Review Article

A Review of Active Yaw Control System for Vehicle Handling and Stability Enhancement

M. K. Aripin,[1] Yahaya Md Sam,[2] Kumeresan A. Danapalasingam,[2] Kemao Peng,[3] N. Hamzah,[4] and M. F. Ismail[5]

[1] *Control, Instrumentation & Automation Department, Faculty of Electrical Engineering, Universiti Teknikal Malaysia Melaka, 76100 Durian Tunggal, Melaka, Malaysia*
[2] *Department of Control & Mechatronics, Faculty of Electrical Engineering, Universiti Teknologi Malaysia, 81310 UTM Johor Bahru, Johor, Malaysia*
[3] *Temasek Laboratories, National University of Singapore 5A Engineering Drive 1, Singapore 117411*
[4] *Faculty of Electrical Engineering, UiTM Pulau Pinang, 13500 Permatang Pauh, Pulau Pinang, Malaysia*
[5] *Industrial Automation Section, Universiti Kuala Lumpur Malaysia France Institute, Section 14, Jalan Teras Jernang, 43650 Bandar Baru Bangi, Selangor, Malaysia*

Correspondence should be addressed to M. K. Aripin; khairiaripin@utem.edu.my

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Yaw stability control system plays a significant role in vehicle lateral dynamics in order to improve the vehicle handling and stability performances. However, not many researches have been focused on the transient performances improvement of vehicle yaw rate and sideslip tracking control. This paper reviews the vital elements for control system design of an active yaw stability control system; the vehicle dynamic models, control objectives, active chassis control, and control strategies with the focus on identifying suitable criteria for improved transient performances. Each element is discussed and compared in terms of their underlying theory, strengths, weaknesses, and applicability. Based on this, we conclude that the sliding mode control with nonlinear sliding surface based on composite nonlinear feedback is a potential control strategy for improving the transient performances of yaw rate and sideslip tracking control.

1. Introduction

In vehicle dynamic control of road-vehicle, controlling the lateral dynamic motion is very important where it will determine the stability of the vehicle. One of the prominent approaches that are reported in the literature for lateral dynamics control is a yaw stability control system. In order to design an effective control system, it is essential to determine an appropriate element of yaw stability control system. In this paper, the elements of yaw stability control system, that is, vehicle dynamic models, control objectives, active chassis control, and its control strategies as depicted in Figure 1, are extensively reviewed.

The linear and nonlinear vehicle models that described the behaviour of lateral dynamic are explained for controller design and evaluation purpose. To achieve the control objectives, it is essential to control the variables of yaw rate and sideslip angle in order to ensure the vehicle stable. It is required that the actual yaw rate and sideslip angle have fast responses and good tracking capability in following the desired responses. During critical driving condition or manoeuvre, inappropriate commands by the driver to control the steering and braking can cause the vehicle to become unstable and lead to an accident. Therefore, an active control for yaw stability control system is essential to assist the driver to keep the vehicle stable on the desired path. By implementing an active chassis control of steering or braking or integration of both systems, the active yaw control system can be realized.

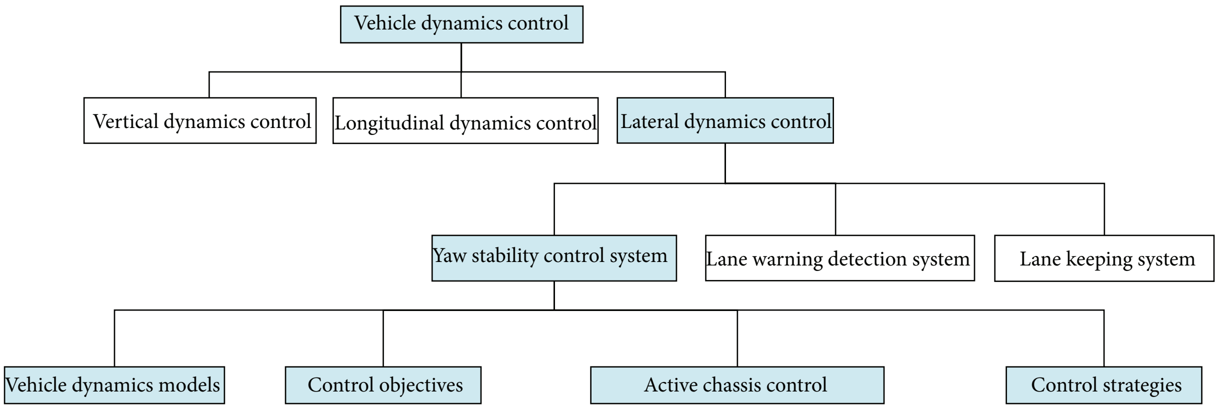

Figure 1: Yaw stability control system for vehicle lateral dynamic.

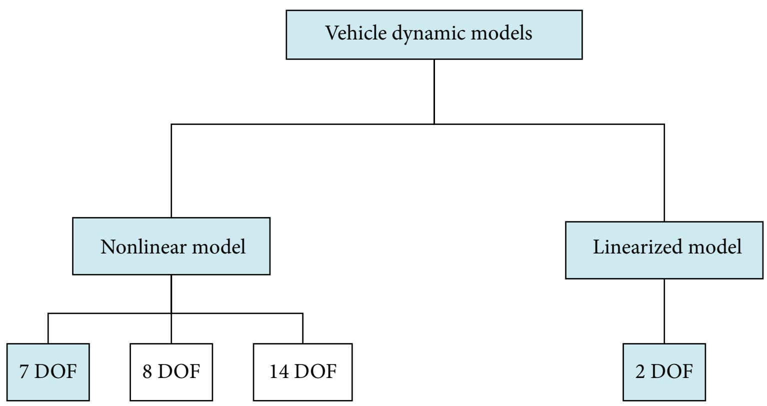

Figure 2: Vehicle dynamic models.

In real driving condition, the lateral dynamics of vehicle are incorporated with uncertainties such as different road surface condition, varying vehicle parameters, and crosswind disturbance. In yaw stability control system, these perturbations could influence the yaw rate and sideslip tracking control performances. From the control system point of view, the transient performances of tracking control are essential. However, from the reviewed control strategies in the literature, the controllers are not designed to cater this matter. Therefore, an appropriate robust control strategy should be proposed to improve the transient performances of the yaw rate and sideslip tracking control in the presence of uncertainties and disturbances. As a finding from the reviews, this paper briefly discussed a possible high performance robust tracking control strategy that can be implemented for yaw stability control system.

The review begins with vehicle dynamics models in Section 2. The yaw stability control objectives are discussed in Section 3 and followed by active chassis control for in Section 4. Yaw stability control strategies and problems are reviewed in Sections 5 and 6, respectively. In Section 7, a high performance robust tracking controller using sliding mode control and composite nonlinear feedback is discussed. The controller evaluation is discussed in Section 8 and ended with conclusion in Section 9.

2. Vehicle Dynamics Models

In order to examine, analyse, and design the controller for yaw stability control system, vehicle dynamics models are essential where the mathematical modelling of vehicle dynamic motion is obtained based on Newton's 2nd law that describes the forces and moments acting on the vehicle body and tires. In general, there are two categories of vehicle dynamic model, that is, nonlinear vehicle model and linearized vehicle model as depicted in Figure 2. The following subsections will discuss the nonlinear vehicle model for simulation and linearized vehicle model for controller design purpose.

2.1. Vehicle Model for Simulation. The nonlinear vehicle model is regularly used to represent and simulate the actual vehicle for controller evaluation and validation. In recent years, researches in [1–5] have utilized nonlinear vehicle model for vehicle handling and stability improvement studies. Figure 3 shows the typical nonlinear vehicle model in cornering manoeuvre.

The input of this model is front wheel steer angle δ_f while the output variables to be controlled are vehicle sideslip β and yaw rate r. The vehicle parameters are vehicle width track d, distance from front, and rear axle to centre of

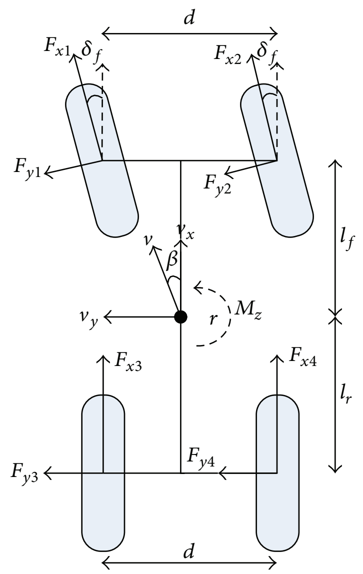

FIGURE 3: Nonlinear vehicle model [6].

gravity (CG) l_f and l_r, respectively. The vehicle forward velocity of centre of gravity (CG) is v, lateral velocity is v_y, and longitudinal velocity is v_x. Other important vehicle parameters are vehicle mass m, moment of inertia I_z, and front/rear tire cornering stiffness C_f/C_r. The wheels are numbered as subscript number with (1) for front-left, (2) for front-right, (3) for rear-left, and (4) for rear-right.

Longitudinal tire force, F_{xi}, depends directly on tire slip ratio, λ_i while lateral tire force, F_{yi}, depends directly on tire sideslip angle, α_i. For smaller slip angle and slip ratio, lateral tire force is described as a linear function of the tire cornering stiffness and tire sideslip angle while longitudinal tire force is described as a linear function of the braking stiffness and the tire slip ratio. For larger slip angle and slip ratio, longitudinal and lateral tire forces exhibit a nonlinear characteristics. Vehicle dynamic motion with nonlinear tire forces represents a nonlinear system. The nonlinear lateral and longitudinal tire forces can be described using prominent Pacejka tire model as implemented in [1, 4, 7] or Dugoff tire model as utilized in [8–10], while studies in [11] used both tire models.

The nonlinear vehicle model could have different number of degree-of-freedom (DOF) where it represents the dynamics motions and complexity of vehicle models. As utilized in [2, 12–14], the 7 DOF vehicle model represents the dynamic motions of vehicle body, that is, longitudinal, lateral, yaw, and four wheels. The dynamic equations for the longitudinal, lateral, and yaw motions of the vehicle body are described as follows.

Longitudinal Motion. One has the following:

$$\begin{aligned} ma_x &= m\left(\dot{v}_x - rv_y\right) \\ &= \left(F_{x1} + F_{x2}\right)\cos\delta_f + F_{x3} + F_{x4} - \left(F_{y1} + F_{y2}\right)\sin\delta_f. \end{aligned} \tag{1}$$

Lateral Motion. One has the following

$$\begin{aligned} ma_y &= m\left(\dot{v}_y + rv_x\right) \\ &= \left(F_{x1} + F_{x2}\right)\sin\delta_f + \left(F_{y1} + F_{y2}\right)\cos\delta_f + F_{y3} + F_{y4}. \end{aligned} \tag{2}$$

Yaw Motion. One has the following:

$$\begin{aligned} I_z\dot{r} = {} & l_f\left(F_{y1}\cos\delta_f + F_{y2}\cos\delta_f + F_{x1}\sin\delta_f + F_{x2}\sin\delta_f\right) \\ & - l_r\left(F_{y3} + F_{y4}\right) + M_z, \end{aligned} \tag{3}$$

where M_z is yaw moment that must be taken into account, that is; $M_z > 0$ if the tires tends to turn at z-axis. In (2), the lateral acceleration a_y can be expressed in terms of vehicle forward speed v, yaw rate r, and sideslip β as follows:

$$a_y = \dot{v}_y + rv_x = v\left(r + \dot{\beta}\right). \tag{4}$$

Therefore, the output variable of sideslip β of two-track model can be obtained as follows:

$$\begin{aligned} \dot{\beta} = {} & \frac{1}{mv}\left[\cos\beta\left(\cos\delta_f\left(F_{x1} + F_{x2}\right) - \sin\delta_f\left(F_{y1} + F_{y2}\right)\right)\right. \\ & \left. - \sin\beta\left(\sin\delta_f\left(F_{x1} + F_{x2}\right) - \sin\delta_f\left(F_{y1} + F_{y2}\right)\right)\right] \\ & - r. \end{aligned} \tag{5}$$

while the output variable of yaw rate r can be determined from (3) and obtained as follows:

$$\begin{aligned} \dot{r} = {} & \frac{1}{I_z}\left[l_f\left(F_{y1}\cos\delta_f + F_{y2}\cos\delta_f + F_{x1}\sin\delta_f + F_{x2}\sin\delta_f\right)\right. \\ & \left. - l_r\left(F_{y3} + F_{y4}\right) + M_z\right]. \end{aligned} \tag{6}$$

In vehicle dynamic studies, each wheel represents 1 DOF. Thus, there are 4 DOF for road-vehicle with 4 wheels. The dynamic motion for each wheel is described as follows:

$$I_{wi}\dot{\omega}_i = -R_{wi}F_{xi} + T_{ei} - T_{bi}, \tag{7}$$

where $\dot{\omega}$ is wheel angular acceleration, n, R_w is wheel radius, I_w is wheel inertia, T_{bi} is braking torque, and T_{ei} is driving torque.

Another nonlinear vehicle model used in the previous research is 8 DOF vehicle model that is extensively used in [4, 5, 9–11, 15–18]. For more accurate simulation and validation, the 14 DOF vehicle model is used in [1, 19, 20]. The comparison between the number of DOF of nonlinear vehicle models that have been discussed above can be summarized and compared in Table 1.

Another nonlinear vehicle model used for simulation uses a multi-degree-of-freedom vehicle model based on commercial vehicle dynamics software, that is, CarSim, as

Table 1: Number DOF of nonlinear vehicle models.

Number of DOF	Dynamic motions	Output variable
7 DOF	(i) Longitudinal (ii) Lateral (iii) Vertical (iv) Rotational of 4 wheels	Yaw rate & Sideslip
8 DOF	(i) Longitudinal (ii) Lateral (iii) Vertical (iv) Roll (v) Rotational of 4 wheels	Yaw rate, roll rate, and sideslip
14 DOF	(i) Longitudinal (ii) Lateral (iii) Vertical (iv) Roll (v) Pitch (vi) Bounce (vii) Rotational of 4 wheels (viii) Vertical oscillations of 4 wheels	Yaw rate, roll rate, pitch rate, and sideslip

implemented in [21–26]. By using this software based vehicle model, the dynamic behaviour of vehicle is more precise similar to a real vehicle. However, for yaw rate and sideslip tracking control in yaw stability control system, the 7 DOF nonlinear vehicle model as discussed in the above equations and shown in Table 1 is adequate for simulation and evaluation of the design controller.

2.2. Vehicle Model for Controller Design. In vehicle dynamic studies, the classical bicycle model as shown in Figure 4 is prominently used for yaw stability control analysis and controller design as utilized in [1, 3, 8, 26–30]. This model is linearized from the nonlinear vehicle model based on the following assumptions.

(i) Tires forces operate in the linear region.

(ii) The vehicle moves on plane surface/flat road (planar motion).

(iii) Left and right wheels at the front and rear axle are lumped in single wheel at the centre line of the vehicle.

(iv) Constant vehicle speed i.e. the longitudinal acceleration equal to zero (a_x=0)

(v) Steering angle and sideslip angle are assumed small (≈ 0).

(vi) No braking is applied at all wheels.

(vii) Centre of gravity (CG) is not shifted as vehicle mass is changing.

(viii) 2 front wheels have the same steering angle.

(ix) Desired vehicle sideslip is assumed to be zero in steady state.

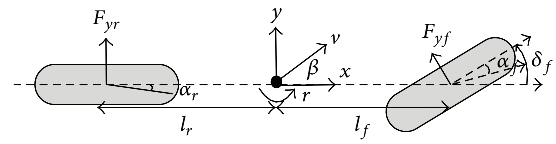

Figure 4: Bicycle model [31].

In the simplest form of planar motion, this model consists of 2 DOF for lateral and yaw motions as describe in the following equations.

Lateral Motion. One has the following:

$$mv\left(\dot{\beta} + r\right) = \left(F_{yf} + F_{yr}\right) - r. \tag{8}$$

Yaw Motion. One has the following:

$$I_z\dot{r} = l_f \cdot F_{yf} - l_r \cdot F_{yr}. \tag{9}$$

In this model, the front and rear lateral tire forces F_{yf} and F_{yr}, respectively, exhibit linear characteristics and described as a linear function of the front and rear cornering stiffness, C_f and C_r, as follows:

$$\begin{aligned} F_{yf} &= C_f\alpha_f, \\ F_{yr} &= C_r\alpha_r, \end{aligned} \tag{10}$$

where the front and rear tire sideslip angle, α_f, and α_r for linear tire forces are given in the following equations:

$$\begin{aligned} \alpha_f &= \delta_f - \beta - \frac{l_f r}{v}, \\ \alpha_r &= -\beta + \frac{l_r r}{v}. \end{aligned} \tag{11}$$

By rearranging and simplifying (8)–(11), the differential equations of sideslip and yaw rate variables can be simplified as a linear state space model as follows:

$$\begin{gathered} \dot{x} = Ax + Bu, \\ \begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \end{bmatrix} u, \\ \begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} \dfrac{-C_f - C_r}{mv} & -1 + \dfrac{C_r l_r - C_f l_f}{mv^2} \\ \dfrac{C_r l_r - C_f l_f}{I_z} & \dfrac{-C_f l_f^2 - C_r l_r^2}{I_z v} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} \\ + \begin{bmatrix} \dfrac{C_f}{mv} \\ \dfrac{C_f l_f}{I_z} \end{bmatrix} \delta_f, \end{gathered} \tag{12}$$

where β and r are state or output variables, C_f and C_r are front and rear tire cornering stiffness, respectively, m is vehicle

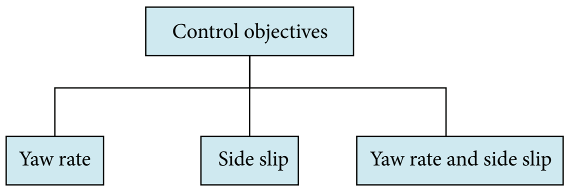

Figure 5: Yaw stability control objectives.

mass, I_z is moment of inertia, l_f and l_r are distance from front and rear axle to centre of gravity, respectively, v is vehicle speed, and front tire steer angle δ_f is the input u to the model. Notice that vehicle speed v is assumed always constant which means the vehicle is not involved with accelerating and braking. Hence, only lateral and yaw motions are analysed.

Besides that, the bicycle model is also regularly used as desired or reference model to generate the desired response of the yaw rate and sideslip angle based on steady state condition or approximated first order response. In designing the control strategy based on vehicle active chassis control, the linear state space model in (13) is essential.

3. Yaw Stability Control Objectives

A vehicle yaw rate r and sideslip angle β are significant variables in vehicle yaw stability control system. As stated in [32], control objectives of yaw stability control system may be classified into three categories, that is, yaw rate control, sideslip control, and combination of yaw rate and sideslip control as illustrated in Figure 5.

One of the control objectives of yaw stability control system is yaw rate, r. An ability to control the actual yaw rate close to desired response will improve the handling or manoeuvrability of the vehicle. The desired yaw rate which is generated by reference model should be tracked by the controller in order to improve the handling performance as mentioned in [2, 4, 13, 15, 18, 27, 33, 34]. In the steady state condition, the desired yaw rate response r_d can be obtained by using the following equation:

$$r_d = \frac{v}{\left(l_f + l_r\right) + k_{us} v^2} \cdot \delta_f, \tag{13}$$

where stability factor k_{us} is depending on the vehicle parameters and defined as follows:

$$k_{us} = \frac{m\left(l_r C_r - l_f C_f\right)}{\left(l_f + l_r\right) C_f C_r}. \tag{14}$$

Another control objective is the vehicle sideslip angle, β, that is, the deviation angle between the vehicle longitudinal axis and longitudinal axis and its motion direction. The control of sideslip angle close to steady state condition means controlling the lateral stability of the vehicle. For the steady state condition, the desired sideslip is always zero, that is, $\beta_d = 0$ as mentioned in [1, 6, 9, 11, 17, 26, 35]. Therefore, to improve the vehicle handling and stability performances, it is essential to control both yaw rate and sideslip responses. In order to achieve these control objectives, the proposed controller must be able to perform the control task of the yaw rate and sideslip tracking control.

4. Active Chassis Control

Steering and braking subsystems or actuator are part of the vehicle chassis. The active control of yaw stability control system can be realized through active chassis control, that is, direct yaw moment control or active steering control or integrated actives steering and direct yaw moment control as shown in Figure 6. In direct yaw moment control which can be implemented by active braking or active differential torque distribution, the required yaw moment is generated by the designed controller that controls the desired yaw rate and sideslip. In active steering control, the wheel steer angle that commanded by the driver is modified by adding corrective steer angle from the designed controller. This control strategy can be implemented either using active front steering (AFS) or active rear steering (ARS) or four-wheel active steering (4WAS) control. In order to control two variables of the yaw rate and sideslip effectively, two different control mechanisms are required. Thus, related research works on the integration of two vehicle chassis control, that is, integrated active steering and direct yaw moment control, have been extensively conducted recently. The review of direct yaw moment control, active steering control, and integrated active steering and direct yaw moment control are discussed in the following subsections.

4.1. Direct Yaw Moment Control. Direct yaw moment control is one of the prominent methods for yaw stability control where extensive research works using this method have been conducted with different control strategies and algorithms as reported in [1, 3, 5, 8, 9, 15–18, 25, 26, 30, 36]. It is recognized as an effective method to enhance the vehicle lateral stability during critical driving manoeuvre by controlling the slip ratio of individual wheel. As illustrated in Figure 7, the required corrective yaw moment, ΔM_z which is generated by the transverse distribution of braking forces between the vehicle wheels is calculated by the designed controller based on the error between actual and desired vehicle model that have been discussed in Section 2. Another approach of direct yaw moment control is active distribution torque. By using an active differential device as established in [19, 20, 37, 38], the left-right driving torque is distributed by this device to generate the required corrective yaw moment, ΔM_z.

As mentioned in Section 2, direct yaw moment control design is based on the linear state space model. As described in (15), M_z is considered as control input and front steer angle δ_{fd} is assumed as disturbance:

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \end{bmatrix} \delta_{fd} + \begin{bmatrix} b_3 \\ b_4 \end{bmatrix} M_z,$$

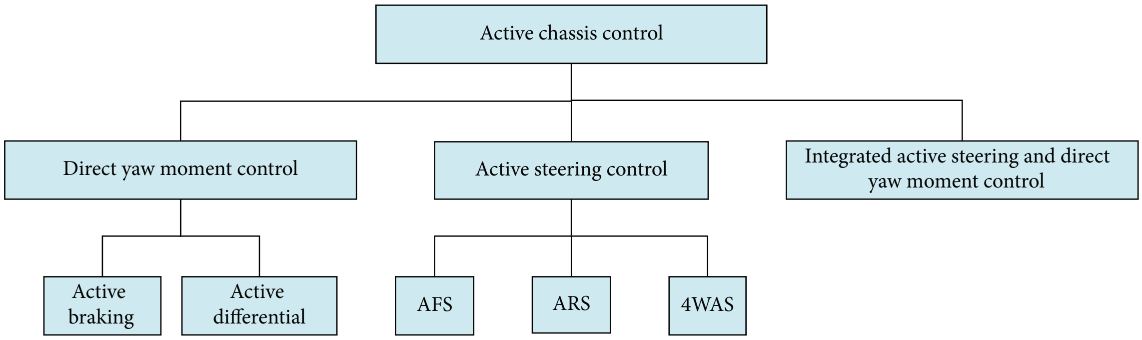

FIGURE 6: Active chassis control.

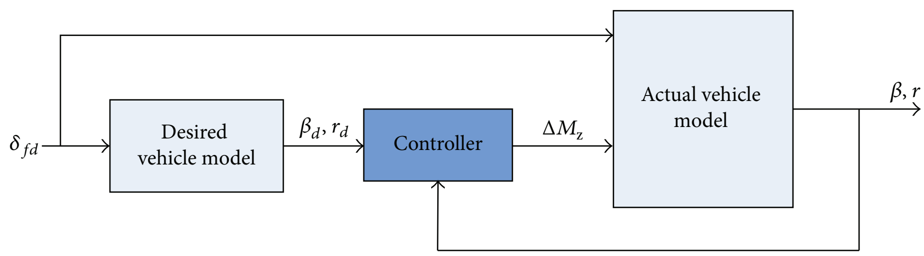

FIGURE 7: Direct yaw moment control [15].

$$\begin{bmatrix}\dot{\beta}\\ \dot{r}\end{bmatrix}=\begin{bmatrix}\dfrac{-C_f-C_r}{mv} & -1+\dfrac{C_rl_r-C_fl_f}{mv^2}\\ \dfrac{C_rl_r-C_fl_f}{I_z} & \dfrac{-C_fl_f^2-C_rl_r^2}{I_zv}\end{bmatrix}\begin{bmatrix}\beta\\ r\end{bmatrix}+\begin{bmatrix}\dfrac{C_f}{mv}\\ \dfrac{C_fl_f}{I_z}\end{bmatrix}\delta_{fd}+\begin{bmatrix}0\\ \dfrac{1}{I_z}\end{bmatrix}M_z. \tag{15}$$

Although direct yaw moment control could enhance the vehicle stability for critical driving conditions, it may be less effective for emergency braking on split road surface. At high vehicle speed steady state cornering, direct yaw moment control could decrease the yaw rate and increase a burden to the driver. To overcome this disadvantage, active steering control is proposed.

4.2. Active Steering Control. Active steering control is another approach to improving the vehicle yaw stability, especially for steady state driving condition where the lateral tire force is operated in the linear region. Research works of active steering control have been continuously conducted in order to improve the handling and stability performances as reported in [7, 13, 39–42]. In general, active steering control can be divided into three categories, that is, active front steering (AFS) control, active rear steering (ARS) control, and four-wheel active steering (4WAS) control, as shown in Figure 6. As road-vehicle normally has front-wheel steering, AFS control becomes favourite approach among researchers as it can be combined with active braking and/or suspension control. In the AFS control diagram, as shown in Figure 8, the front wheel steers angle is a sum of steer angle commanded by the driver δ_{fd} and a corrective steer angle δ_c generated by the controller. This corrective steer angle is computed based on yaw rate and sideslip tracking errors e_1 and e_2 as implemented in [6, 43–47].

For control design and analysis of AFS control, the linear state state space model as described in (16) is used. Noted that this equation is similar to equation (12) but the front wheel steer angle, $\delta_f = \delta_{fd} + \delta c$:

$$\begin{bmatrix}\dot{\beta}\\ \dot{r}\end{bmatrix}=\begin{bmatrix}\dfrac{-C_f-C_r}{mv} & -1+\dfrac{C_rl_r-C_fl_f}{mv^2}\\ \dfrac{C_rl_r-C_fl_f}{I_z} & \dfrac{-C_fl_f^2-C_rl_r^2}{I_zv}\end{bmatrix}\begin{bmatrix}\beta\\ r\end{bmatrix}+\begin{bmatrix}\dfrac{C_f}{mv}\\ \dfrac{C_fl_f}{I_z}\end{bmatrix}\left(\delta_{fd}+\delta_c\right). \tag{16}$$

On the other hand, ARS control is used to improve the vehicle response for low speed cornering manoeuvres with the input to the control system being the rear steering angle δ_r. In order to enhance the manoeuvrability at low speed and the handling stability at high speed, combination of AFS control and ARS called 4WAS control has been proposed as implemented in [24, 48, 49]. By implementing 4WAS control, the lateral and yaw motion can be controlled simultaneously using two independent control inputs. Noting that front wheel steer angle δ_f and rear wheel steer angle δ_r with the rear axles of rear tire cornering stiffness, C_r and distance from rear axle to centre of gravity l_r are taken into account in the input matric.

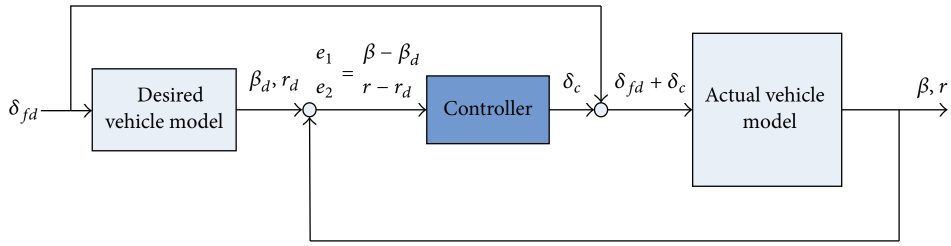

FIGURE 8: Active front steering control [45].

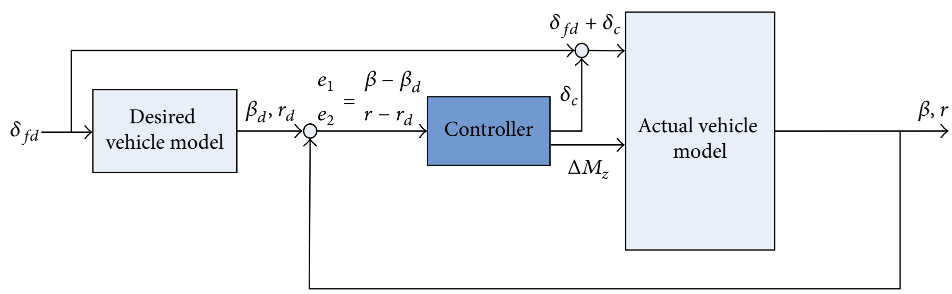

FIGURE 9: Integrated active front steering-direct yaw moment control [53].

4.3. Integrated Active Chassis Control. The integrated active chassis control has become a popular research topic in vehicle dynamics control as discussed in [50]. Vehicle dynamics control can be greatly achieved by integrating the active chassis control of active steering, active braking, and active suspension or active stabiliser as implemented in [12, 23, 51, 52]. Since road-vehicle is usually equipped with front-wheel steering and braking system, an integration and coordination of active front steering and direct yaw moment control are the favourite approaches to achieving the objectives of yaw rate and sideslip control as reported in [2, 10, 11, 27, 28, 53–59]. In this approach, the corrective front wheel steers angle δ_c and corrective yaw moment ΔM_z are considered as two independent control inputs to the vehicle as illustrated in Figure 9.

For controller analysis and design of integrated active front steering-direct yaw moment control, the linear state space model used is describe as follows:

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} \dfrac{-C_f - C_r}{mv} & -1 + \dfrac{C_r l_r - C_f l_f}{mv^2} \\ \dfrac{C_r l_r - C_f l_f}{I_z} & \dfrac{-C_f l_f^2 - C_r l_r^2}{I_z v} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} \dfrac{C_f}{mv} & 0 \\ \dfrac{C_f l_f}{I_z} & \dfrac{1}{I_z} \end{bmatrix} \begin{bmatrix} \delta_c \\ \Delta M_z \end{bmatrix} + \begin{bmatrix} \dfrac{C_f}{mv} \\ \dfrac{C_f l_f}{I_z} \end{bmatrix} \delta_{fd}. \quad (17)$$

The principle of active chassis control of steering and braking for yaw stability control has been discussed. From the above discussion, the differences, advantages, and disadvantages of each active chassis control can be digested as tabulated in Table 2. From this table, it can be observed that, by implementing integrated active front steering-direct yaw moment control, the lateral and yaw motions can be controlled simultaneously using two independent control inputs from two different actuators, that is, steering and braking. Thus, this approach could enhance the vehicle yaw stability where the yaw rate and sideslip can be controlled effectively in emergency manoeuvres and the steady state driving condition.

As a conclusion, active chassis control is essential for active yaw stability control system. Therefore, to achieve the yaw stability control objectives, the control strategies for yaw rate and sideslip tracking control are developed based on this active chassis control. The following section will review and discuss the control strategies and algorithms that have been developed in the past.

5. Yaw Stability Control Strategies

From the literature, various control strategies have been explored and utilized based on particular algorithm for active yaw stability control such as classical PID controller in [1], LMI based and static state feedback control in [2, 8, 33], H_∞ control theory in [4, 13, 25], sliding mode control (SMC) in [1, 7, 23, 24, 35, 38, 53], optimal guaranteed cost coordination controller (OGCC) in [10], adaptive based control in [11], mixed-sensitivity minimization control techniques in [16], classical controllers PI in [49, 60], internal model control (IMC) in [37], quantitative feedback theory (QFT) in [45], and μ-synthesis control in [48]. Besides that, a combination or integration of different two control schemes to ensure the robustness of yaw stability control has been explored such as SMC and backstepping method in [3], SMC and Fuzzy Logic Control in [12], and LQR with SMC in [17]. As discussed in [20], the IMC and SMC algorithms are designed

Table 2: Types of active chassis control.

Vehicle actuator	Active chassis control		Advantages	Disadvantages
Brakes	Direct yaw moment control (DYC)	Active braking active differential	(i) Effective for critical driving condition (ii) Good for sideslip/wheelslip control	(i) Less effective for braking on split road surface (ii) Decrease yaw rate during steady state driving condition (iii) Active differential need extra devices
Steering	Active steering control (ASC)	Active front steering (AFS) control	(i) Effective for steady state driving condition (ii) Ease to integrate with braking control (iii) Good for yaw rate control	Less effective during critical driving condition
		Active rear steering (ARS) control	(i) Rear wheel steer angle can be controlled (ii) Good for yaw rate control	Less effective during critical driving condition
		4 wheels active steering (4WAS) control	(i) Two different steer inputs (ii) Goof for yaw rate control	Less effective during critical driving condition
Steering and brake	Integrated AFS-DYC control		(i) Two different inputs from two different actuator (steering and braking) (ii) Good for yaw rate and sideslip control	Effective for critical and steady state driving condition

for yaw stability control and the controllers performances are compared and evaluated.

The control strategies are designed based on active chassis control as discussed in Section 4. In active braking or active differential which operates based on direct yaw moment control (DYC), various robust control strategies have been designed. As reported in [3], yaw stability control that consists of tire force observer and cascade controller that is based on sliding mode and backstepping control method is designed. To solve the external disturbance as discussed in [16], the robustness of mixed-sensitivity yaw stability controller is guaranteed for external crosswind and emergency manoeuvres. To cater the uncertainty from longitudinal tire force, the controller for wheel slip control is designed using SMC algorithm for vehicle stability enhancement [17]. As discussed in [20], the second order sliding mode (SOSM) and enhanced internal mode control (IMC) are designed as feedback controller to ensure the robustness against uncertainties and control saturation issues. Both controllers' performances are compared and analysed for yaw control improvement based on rear active differential device. Besides that, the sliding mode control algorithm is also utilized to determine the required yaw moment in order to minimize the yaw rate error and side-slip angle for vehicle stability improvement [22]. To overcome the uncertainties parameters and guarantee robust yaw stability in [25], the control strategy that consists of disturbance observer to estimate feedforward yaw moment and optimal gain-scheduled H_∞ is designed. In the study of [30], the robust yaw moment controller and velocity-dependent state feedback controller are matrixed by solving finite numbers linear matric inequality (LMI). By using this approach, the designed controller is able to improve the vehicle handling and lateral stability in the presence of uncertainty parameters such as vehicle mass, moment of inertia, cornering stiffness, and variation of road surfaces and also control saturation due to the physical limits of actuator and tire forces.

In active steering control, robust control strategies are designed to overcome the uncertainties and external disturbance problems. In [7], adaptive sliding mode control is utilized to estimate the upper bounds of time-derived hyperplane and uncertainties of lateral forces. As discussed in [13], feedback H_∞ control is implemented for robust stabilization of yaw motion where speed and road adhesion variations are considered as uncertainties and disturbance input. As reported in [49], a proportional active front steering control and proportional-integral active rear steering control are designed for four-wheel steering (4WS) vehicle with the objective to overcome the uncertainties of vehicle mass, moment of inertia, and front and rear cornering stiffness coefficients. To ensure a robust stability against system uncertainties, the automatic path-tracking controller of 4WS vehicle based on sliding mode control algorithm is designed [24]. In this study, the cornering stiffness, path radius fluctuation, and crosswind disturbance are considered as uncertainty parameters and external disturbance. As reported in [42], the model reference adaptive nonlinear controllers is proposed for active steering systems to solve the uncertainties and nonlinearities of tire's lateral forces. Quantitative feedback theory (QFT) technique is implemented for robust active front steering control in order to compensate for the yaw rate response in presence of uncertainties parameters and reject the disturbances [45]. As discussed in [48], robust controller for 4WS vehicle is also designed based on μ-synthesis control algorithm which considers the varying parameters induced by the vehicle during driving conditions as uncertainties while the study in [60] designed the steering control of vision based autonomous vehicle based on the nested PID control to ensure the robustness of the steering controller against the speed variations and uncertainties of vehicle parameters.

In integrated active chassis control, an appropriate control scheme is designed to meet the control objectives. Studies in [2, 27, 33] have designed the control scheme that consists of reference model based on linear parameter-varying (LPV) formulation and static-state feedback controller with the objective to ensure the robust performance for integrated active front steering and active differential braking control. In these studies, tire slip angle, longitudinal slips, and vehicle forward speeds are represented as uncertainty parameters. As reported in [4], integrated robust model matching chassis controller that integrates active rear wheel steering control, longitudinal force compensation, and active yaw moment control is designed using H_∞ controller based on linear matrix inequalities (LMIs) for vehicle handling and lane keeping performance improvement. In integrated active front steering-direct yaw moment control, an optimal guaranteed cost control (OGCC) technique is utilized in [10]. In this study, tire cornering stiffness is treated as uncertainty during variation of driving conditions. As discussed in [11], an adaptive integrated control algorithm based on direct Lyapunov method is designed for integrated active front steering and direct yaw moment control with cornering stiffness is considered as a variation parameter to ensure the robustness of designed controller. As reported in [23], sliding mode controller is utilized for stabilising the forces and moments in integrated control schemes that coordinated the steering, braking, and stabiliser. In this study, the integrated control structure is composed of a main loop controller and servo loop controller that computes and distributes the stabilizing forces/moments, respectively.

From the above discussion, these control strategies and algorithms can be summarized and compared in terms of their active chassis control, control objective, advantages, and disadvantages as tabulated in Table 3. In conclusion, an appropriate control strategy must be designed based on particular algorithm. Robust control algorithms such as H_∞, SMC, IMC, OGCC, QFT. are essential to solve the uncertainties and disturbance problems that influenced the yaw stability control performances. It is revealed that the designed controllers in the above discussion are able to track the desired yaw rate and vehicle sideslip response considering external disturbances and system uncertainty.

6. Yaw Stability Control Problems

In the real environments, the dynamics of road-vehicle is highly nonlinear and incorporated with uncertainties. Vehicle motion with nonlinear tire forces represents a nonlinear

Table 3: Yaw stability control algorithms.

Control algorithms	Active chassis control	Control objective	Advantages	Disadvantages
PID controller	DYC	sideslip	Anti-wind-up strategy to avoid high overshoot and large settling time	Uncertainties are not consider
LMI static state feedback	Integrated AFS-active differential	Yaw rate and sideslip	robust for uncertainties	Transient response improvement is not consider
H_∞	Integrated chassis control, active steering	Yaw rate	Robust for uncertainties, reject disturbance	
SMC	DYC, active steering	Yaw rate and sideslip	robust for uncertainties and reject disturbance	
OGCC	Integrated AFS-DYC	Yaw rate and sideslip	Robust for uncertainties	
Adaptive integrated control	Integrated AFS-DYC	Yaw rate and sideslip	Robust for uncertainties	
Mixed-sensitivity minimization control	DYC	Yaw rate	Robust for uncertainty, reject disturbance	
PI controller	4WAS	Yaw rate	Robust for uncertainties	Transient response improvement is not consider
IMC	DYC	Yaw rate	Robust for uncertainty	
QFT	AFS	Yaw rate	Robust for uncertainties, reject disturbance	
μ synthesis control	4WAS	Yaw rate and sideslip	Robust for uncertainties	
SMC-backstepping		Yaw rate and sideslip	Robust for nonlinearities	Uncertainties are not considered
SMC-FLC	Integrated steering, brake, and suspension	Yaw rate, sideslip, and roll angle	Robust for uncertainties and nonlinearities	Transient response improvement is not consider
SMC-LQR	DYC	Yaw rate and sideslip	Robust for uncertainty	

system where the tire dynamic exhibit nonlinear characteristics, especially during critical driving conditions such as a severe cornering manoeuvre. The main problems of yaw rate and sideslip tracking control are uncertainties caused from variations of dynamics parameters as discussed in the previous section such as road surface adhesion coefficients [8, 13, 33, 37, 45], tire cornering stiffness [2, 8, 10–12, 20, 24, 30, 48, 49], vehicle mass [20, 30, 38, 45, 49], vehicle speed [2, 13, 45], and moment of inertia [30, 49]. Besides that, an external disturbance such as lateral crosswind may influence the tracking control of desired yaw rate and sideslip response as reported in [4, 6, 13, 24]. Therefore, appropriate control strategies and algorithms are essential to overcome these problems as discussed in the previous section.

From the view of control system engineering, the transient response performances of tracking control are very important. However, the control strategies and algorithms discussed above are not accommodated for transient response improvement of the yaw rate and sideslip tracking control in presence of uncertainties and disturbances. The designed controllers are only sufficient to track the desired responses in the presence of such problems. Hence, an appropriate control strategy that could improve the transient performance of robust yaw rate and sideslip tracking control should be designed for an active yaw control system which can enhance the vehicle handling and stability performances.

7. High Performance Robust Tracking Controller

In this section, a principle of possible robust tracking control strategy with high performance that can be implemented for yaw rate and sideslip tracking control is discussed. Based on the literature, a sliding mode control with the nonlinear sliding surface can be proposed to improve the transient response of the yaw rate and sideslip tracking control in presence of uncertainties and disturbances.

7.1. Sliding Mode Control (SMC). Sliding mode control (SMC) algorithm that had been developed in the two last decades is recognized as an effective robust controller to cater for the matched and mismatched uncertainties and disturbances for linear and nonlinear system. It is also utilized as an observer for estimation and identification purpose in engineering system. Various applications using SMC are successfully implemented as numerous research studies and reports have been published. In vehicle and automotive studies, SMC is one of the prominent control algorithms that is used as a robust control strategy as implemented in [3, 17, 38, 53, 61–63].

Sliding mode control design consists of two important steps, that is, designing a sliding surface and designing the control law so that the system states are enforced to the sliding surface. The design of sliding surface is very important as it will determine the dynamics of the system being control. In conventional SMC, a linear sliding surface has a disadvantage in improving transient response performance of the system due to constant closed loop damping ratio. Therefore, a nonlinear sliding surface that changes a closed loop system damping ratio to achieve high performance of transient response and at the same time ensure the robustness has been implemented in [64–69]. In these studies, the nonlinear sliding surface is designed based on the composite nonlinear feedback (CNF) algorithm.

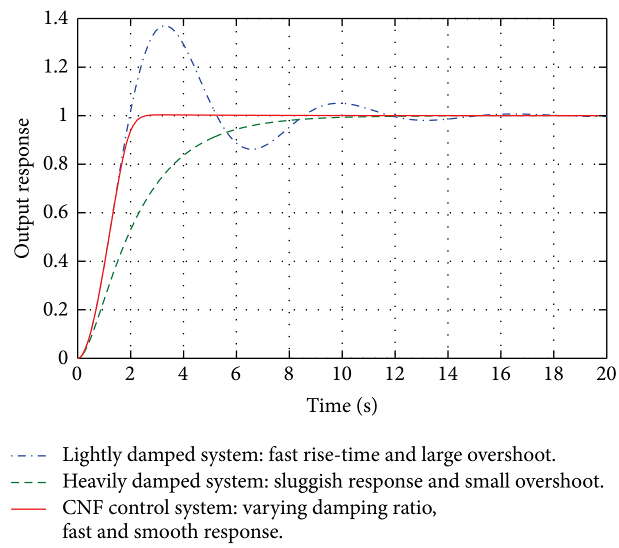

Figure 10: CNF control technique for transient performances improvement [75].

7.2. Nonlinear Sliding Surface Based CNF. The concept of varying closed loop damping ratio, which could improve transient response for uncertain system, is based on composite nonlinear feedback (CNF) control technique. This technique that has been established in [70–74] is developed based on state feedback law. In practice, it is desired that the control system to obtain fast response time with small overshoot. But in fact, most of control scheme makes a trade-off between these two transient performance parameters. Hence, the CNF control technique keeps low damping ratio during transient and varied to high damping ratio as the output response closed to the set point as illustrated in Figure 10.

In general, the design of the CNF control technique consists of linear and nonlinear control law as describe as follows:

$$\begin{aligned} u &= [u_{\text{Linear}}] + [u_{\text{Nonlinear}}], \\ u &= [Fx + Gr] + [\rho(r, y) B' P (x - x_e)], \end{aligned} \tag{18}$$

where F is feedback matrix, G is a scalar, B is input matrix, $P > 0$ is a solution of Lyapunov equation, and $\rho(r, y)$ is nonlinear function which is not unique and can be chosen from the following equations:

$$\rho(r,y) = -\beta e^{-\alpha(y-r)^2},$$
$$\rho(r,y) = -\beta e^{-\alpha|y-r|}, \quad (19)$$
$$\rho(r,y) = -\frac{\beta}{1-e^{-1}}\left(e^{-(1-(y-y_0)/(r-y_0))^2} - e^{-1}\right).$$

Based on tracking error, a nonlinear sliding surface adapted from the CNF control law for an active yaw control system can be defined as follows:

$$s := c^T e(t) = \begin{bmatrix} c_1 & I_m \end{bmatrix} \begin{bmatrix} e_1(t) \\ e_2(t) \end{bmatrix}, \quad (20)$$

where

$$c_1 := F - \rho(r,y) B' P, \quad (21)$$

where $e_1(t)$ and $e_2(t)$ could represent the yaw rate and sideslip tracking error, respectively, B is an input matrix of the system, and I_m is the identity matrix. Then, the nonlinear sliding surface stability can be determined using Lyapunov stability analysis and implement in the designed control law of SMC.

Based on the above discussion, the SMC with nonlinear sliding surface based on CNF technique could achieve high performance for uncertain systems. It could improve the transient response performance in the presence of uncertainties and external disturbances. In addition, it is found that this control strategy has not yet been examined for vehicle yaw stability control system and should be further investigated. Therefore, this control technique has initiated a motivation to implement it for robust yaw rate and sideslip tracking control in active yaw control systems. It is expected that this approach could improve the vehicle handling and stability performances.

8. Controller Evaluations

In order to evaluate the performance of designing controller, simulations of emergency braking and driving manoeuvres with the nonlinear vehicle model are usually carried out according to ISO or SAE standards. The pure computer simulations, cosimulation with other software or hardware in the loop simulations (HILS), are the common approaches to conducting the yaw stability test with or without driver model for open loop or closed loop analysis, respectively.

One of the typical emergency braking manoeuvres for vehicle yaw stability test is split-μ braking as reported in [2, 37, 60]. In this test, the step input of brake torque is applied to the vehicle in forward motion with constant speed on split road surface adhesion coefficient, μ, where one side of the wheels is on low μ and the other sides of the wheels are on high μ or vice versa. This test is performed to test the vehicle straight ahead driving stability. Critical driving manoeuvres are also another efficient way to test the yaw and lateral stability performances. A step steer manoeuvre can be implemented to evaluate the steady state and transient behavioural response of the vehicle as conducted in [16, 53, 55, 63]. Similarly, the constant speed J-turn manoeuvre is also conducted for such purpose as reported in [5, 8, 9, 15, 30, 33, 45]. Another type of critical driving manoeuvre is lane change manoeuvre as implemented in [3, 5, 10, 11, 15, 20, 21, 23, 26, 45, 46, 53, 55]. This manoeuvre can be conducted for open loop single lane change or closed loop double lanes change with driver model, lane change on different road conditions, lane change on split-μ road, and lane change with braking effect. With steering angle input is in sinusoidal form, the transient handling behaviour can be evaluated and vehicle yaw and lateral stability can be analysed.

Another test manoeuvres that can be implemented for yaw stability control are steer reversal test for transient performance evaluation [16, 19, 20], constant speed steering pad to evaluate the steady state vehicle performance [19, 20], steering wheel frequency sweep for the bandwidth, and resonance peak analysis [20] and also fishhook manoeuvre as mentioned in [2, 25, 27]. In order to evaluate the yaw stability control system performance in the presence of disturbance, a crosswind disturbance as reported in [4, 6, 20, 24] is considered as external disturbance that can influence the lateral dynamic stability.

During critical driving manoeuvres, the actual response of vehicle's yaw rate and sideslip is obtained and analysed in presence of uncertainties and external disturbances. By performing the test manoeuvres as discussed above, it can be concluded that the ability of the designed controller to track the desired response should be validated. The responses are usually compared to uncontrolled vehicle's responses and other controllers for their steady state and transient response performances.

9. Conclusion

This paper has extensively reviewed the elements of yaw stability control system. In designing yaw stability controller, all these elements, that is, vehicle models, control objectives, active chassis control, and control strategies, play an important role that contributes to the control system performances. For controller design and evaluation, a 2 DOF linear and 7 DOF nonlinear vehicle models are essential. In order to improve the handling and stability performances, the yaw rate and sideslip tracking control are the main objectives that must be achieved by the design controller. To realize an active yaw stability control, an active chassis control of steering, braking or integration of both chassis could be implemented with an appropriate control strategies and algorithms.

In real driving condition, the uncertainties and external disturbance may influenced the yaw rate and sideslip tracking control performances. Hence, the robust control algorithm is necessary. Based on this review, it has been concluded that sliding mode control (SMC) is the best robust controller to address these problems. From the view of control system, transient performances are very important for tracking control. However, an existing SMC configuration does not have capability to improve this transient performance. To address this issue, a nonlinear sliding surface of SMC is designed based on composite nonlinear feedback (CNF) algorithm. This is because the CNF algorithm has been proven in improving transient performances as discussed above. For future works, this control strategy will be implemented for yaw stability control system and the transient performances of yaw rate and sideslip tracking control will be evaluated and compared with classical SMC and other controllers.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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