Jurnal Teknologi

Full paper

A Review on Integrated Active Steering and Braking Control for Vehicle Yaw Stability System

M.K. Aripin[a,*], Y.M. Sam[b], A. D. Kumeresan[b], M.F. Ismail[c], Peng Kemao[d]

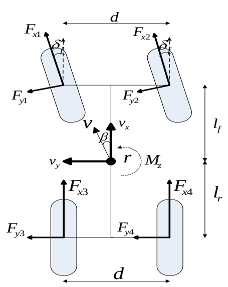
[a]*Control, Instrumentation & Automation Dept., Faculty of Electrical Engineering, UniversitiTeknikal Malaysia Melaka, Melaka*
[b]*Department of Control and Mechatronics, Faculty of Electrical Engineering, Universiti Teknologi Malaysia, 81310 UTM Johor Bahru, Johor Malaysia*
[c]*Industrial Automation Section, Universiti Kuala Lumpur Malaysia France Institute*
[d]*Temasek Laboratory, National University of Singapore*

*Corresponding author:khairiaripin@utem.edu.my,khairi32@live.utm.my

Article history

Received 1 January 2014
Received in revised form:
17 August 2014
Accepted 24 August 2014

Graphical abstract



Abstract

A review study on integrated active steering and braking control for vehicle yaw stability system is conducted and its finding is discussed in this paper. For road-vehicle dynamic, lateral dynamic control is important in order to determine the vehicle stability. The aw stability control system is the prominent approach for vehicle lateral dynamics where the actual yaw rate and sideslip should be tracked by the controller close to the desired response. To improve the performance of yaw stability control during steady state and critical driving conditions, a current approach using active control of integrated steering and braking could be implemented. This review study discusses the vehicle models, control objectives, control problems and propose control strategies for vehicle yaw stability control system. In the view of control system engineering, the transient performances of tracking control are essential. Based on the review, this paper discusses a basic concept of control strategy based on the composite nonlinear feedback (CNF) and sliding mode control (SMC) whichcan be proposed for integrated active steering and braking control in order to improve the transient performances of the yaw rate and sideslip tracking control in the presence of uncertainties and disturbances.

Keywords: Vehicle yaw stability; active steering control, active braking control, lateral dynamics control

■1.0 INTRODUCTION

This paper presented a review study on integrated active steering and braking control for yaw stability control system of road-vehicles. The main objective of this paper is to provide an overview and comprehensive understanding of the main elements in yaw stability control system and its control strategies in improving vehicle handling and stability performances.

In vehicle dynamics studies, vehicle stability is determined by the lateral dynamic motion where yaw stability control system is one of the prominent approaches for lateral dynamics control and it can be controlled by using steering and braking subsystem.

In yaw stability control strategy, it is required that designed controller could control the actual vehicle yaw rate and sideslip angle with fast responses and good tracking capability in following the desired responses. Improper commands to steering and braking subsystems by the driver during critical driving manoeuvres can cause the vehicle to become unstable and could lead to accidents. Therefore, an active control is essential to assist the driver to keep the vehicle stable on the desired path. By keeping the actual yaw rate and sideslip close to the desired responses, the active steering and braking control could improve the handling and stability performances. However, in real environment, the dynamics of the vehicle are incorporated with disturbances and uncertainties due to varying parameters which can influence the yaw stability control performances.

This review study is focused on the main elements of integrated active steering and braking control and its control strategies for particular problems. The reviews begin with the vehicle dynamics models,presented in Section 2. The yaw stability control objectives and problems are reviewed in Section 3 and followed by control strategies of integrated active steering and braking control system in Section 4. Section 5 discussed the basic concept of high performance robust tracking control strategy and lastly the paper ends with the conclusion in Section 6.

■2.0 VEHICLE MODELS

In a control system design requirement, the modeling of a dynamic system is vital as it represents the behaviour and characteristics of the system being controlled. In vehicle dynamics studies, the mathematical modeling of the vehicle dynamic motions is modeled based on physical laws that describe the forces and moments acting on the vehicle body and tires. This section will brief the dynamics of nonlinear 7 degree-of-freedom (DOF) vehicle model, linear bicycle model, and a reference model with the main assumption that the vehicle is moving with constant speed on plane surface where the roll and pitch motionsare neglected.

2.1 Nonlinear 7 DOF Vehicle Model

Vehicle motion with nonlinear tire forces represents a nonlinear system. Nonlinear full vehicle model is usually used to simulate a real vehicle in order to validate the controller that has been designed. It could have different number of DOF where itrepresents the complex vehicle according to their control structure and algorithm. As utilized in [1-5] the 7 DOF dynamic vehicle model would record thelongitudinal,lateral and yaw dynamic motions of the vehicle body while the four tires represented four wheel dynamic motions.

Another non-linear full vehicle model that wasextensively used for integrated active steering and braking control is the 8 DOF vehicle model as discussed in [6-12]. In this model, the dynamics of the 7 DOF vehicle model is extended with roll motion included. Besides that, the nonlinear vehicle model possesses 27 DOF based on commercial vehicle dynamics software such as CarSim, is also utilized in order to evaluate the proposed controller as implemented in [13-18]. However, in vehicle yaw stability system for vehicle handling and stability improvement, the nonlinear 7 DOF vehicle model that neglected roll and pitch motion is sufficient for simulation and evaluation of the controller design. The typical full vehicle model in cornering manoeuvre is shown in Figure 1 while the vehicle parameters are described in Table 1. On the plane surface (planar motion) where the roll and pitch motions are neglected, the dynamics equations for 3 DOF of longitudinal, lateral and yaw motions of the vehicle body are described based on Newton's 2nd law of rigid body as follows [19];

Longitudinal motion:

$$ma_x = m(\dot{v}_x - rv_y) = (F_{x1} + F_{x2})\cos\delta_f + F_{x3} + F_{x4} - (F_{y1} + F_{y2})\sin\delta_f \tag{1}$$

Lateral motion:

$$ma_y = m(\dot{v}_y + rv_x) = (F_{x1} + F_{x2})\sin\delta_f + (F_{y1} + F_{y2})\cos\delta_f + F_{y3} + F_{y4} \tag{2}$$

Yaw motion:

$$I_z\dot{r} = l_f(F_{y1}\cos\delta_f + F_{y2}\cos\delta_f + F_{x1}\sin\delta_f + F_{x2}\sin\delta_f) - l_r(F_{y3} + F_{y4}) + M_z \tag{3}$$

where

$$M_z = \frac{d}{2}\begin{pmatrix} F_{x1}\cos\delta_f - F_{x2}\cos\delta_f \\ -F_{y1}\sin\delta_f + F_{y2}\sin\delta_f + F_{x3} - F_{x4} \end{pmatrix} \tag{4}$$

In vehicle dynamics 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 [19];

$$I_w\dot{\omega}_i = -R_{wi}F_{xi} + T_{ei} - T_{bi} \tag{5}$$

where $\dot{\omega}_i$ is wheel angular acceleration, R_{wi} is wheel radius, I_{wi} is wheel inertia, T_{bi} is braking torque and T_{ei} is driving torque. The nonlinear lateral and longitudinal tire forces, F_{yi} and F_{xi} can be described using the nonlinear tire model.Pacejka tire model is very prominent and widely used in representing nonlinear tire forces for full vehicle model as well as CarSim vehicle model as utilized in [1, 2, 4-8, 11, 12, 14, 18, 20-24]. The details of Pacejka tire model can be referred to in [25].

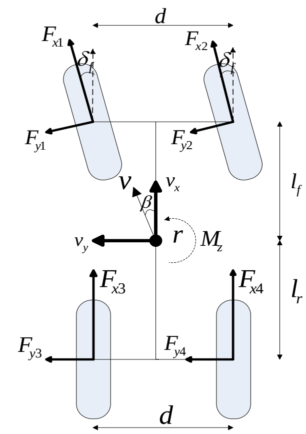


Figure 1 Full vehicle model

Table 1 Vehicle parameters

symbol	description
m	vehicle mass
l_f	distance from front axle to centre of gravity
l_r	distance from rear axle to centre of gravity
d	width track
I_z	moment of inertia
C_f	front tire cornering stiffness
C_r	rear tire cornering stiffness
v	vehicle speed
M_z	yaw moment
δ_f	front wheel steer angle
β	vehicle sideslip
r	yaw rate
v_x	longitudinal velocity
v_y	lateral velocity
F_{xi}	longitudinal tire forces
F_{yi}	lateral tire forces

2.2 Linear Bicycle Model

The classical linear bicycle model as shown in Figure 2 is the linearized model from the nonlinear full vehicle model. It is prominently used to analyse and design the yaw stability controller for vehicle lateral dynamics as mentioned in [5-7, 9-11, 16, 18, 22, 26-28]. Based on assumptions as discussed in the literature, linear bicycle model consists of 2 DOF for lateral and yaw motion only which can be represented in linear state space model as follows [19];

$$\dot{x} = Ax + Bu$$

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \end{bmatrix} u$$

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} \dfrac{-C_f - C_r}{mv} & -1 + \dfrac{C_r l_r - C_f l_f}{mv^2} \\ \dfrac{C_r l_r - C_f l_f}{I_z} & \dfrac{-C_f l_f^2 - C_r l_r^2}{I_z v} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} \dfrac{C_f}{mv} \\ \dfrac{C_f l_f}{I_z} \end{bmatrix} \delta_f \tag{6}$$

where the parameters v, C_f, C_r, m, l_f, l_r and I_z are the same as shown in Table 1. The input of the model is the front wheel steer angle δ_f while the outputs or controlled variables are vehicle sideslip angle β and yaw rate r. Notice that vehicle speed v is always assumed constant which means the vehicle is not involved with accelerating and braking.

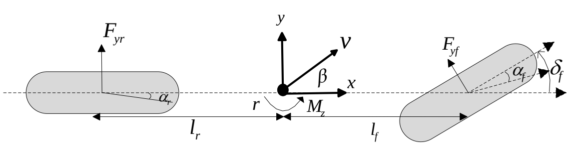


Figure 2 Bicycle model

2.3 Reference Model

The reference model is essential in order to generate the desired response of vehicle side slip β and yaw rate r so that the designed controller is able to track these desired responses. According to [19] a desired model to be followed by an actual vehicle model is based on linear bicycle model in steady state condition. Since the vehicle sideslip is the deviation angle between vehicle longitudinal axis and its motion direction, the desired sideslip β_d for steady state condition is always zero. The desired yaw rate response, r_d in steady state cornering can be approximated as first order system which is obtained in term of steering angle δ_f, vehicle speed v and vehicle parameters as follows [19];

$$r_d = \frac{v}{(l_f + l_r) + k_{us} v^2} . \delta_f \tag{7}$$

where k_{us} is known as understeer parameter and defined as follows [19];

$$k_{us} = \frac{m(l_r C_r - l_f C_f)}{(l_f + l_r) C_f C_r} \tag{8}$$

Hence, the state equations of the desired vehicle model can be expressed as follows [3, 8, 13, 29,30];

$$\dot{x} = A_d x_d + E_d \delta_f$$

$$\begin{bmatrix} \dot{\beta}_d \\ \dot{r}_d \end{bmatrix} = \begin{bmatrix} -\dfrac{1}{t_\beta} & 0 \\ 0 & -\dfrac{1}{t_r} \end{bmatrix} \begin{bmatrix} \beta_d \\ r_d \end{bmatrix} + \begin{bmatrix} \dfrac{k_\beta}{t_\beta} \\ \dfrac{k_r}{t_r} \end{bmatrix} \delta_f \tag{9}$$

where

$$t_\beta = \frac{a_{12} a_{21} - a_{11} a_{22}}{a_{12} (a_{21} e_1 - a_{11} e_2)} \tag{10}$$

$$t_r = \frac{a_{12} a_{21} - a_{11} a_{22}}{a_{22} (a_{21} e_1 - a_{11} e_2)} . k_r \tag{11}$$

$$k_\beta = 0 \tag{12}$$

$$k_r = \frac{v}{l(1 + k_{us} v^2)} \tag{13}$$

and k_{us} is understeer parameter as given in equation (8). Notice also that a_{11}, a_{12}, a_{21} and a_{22} in equation (10) and (11) are as described in equation (6).

■3.0 YAW CONTROLOBJECTIVES& PROBLEMS

In vehicle yaw stability control system, the variables of yaw rate r and sideslip β are very important to determine the stability of the vehicle. As discussed in [31], control objectives of yaw stability control may be classified into three categories i.e. yaw rate control, sideslip control or combined yaw rate & sideslip control. Yaw rate control is more concerned with improving the manoeuverability or handling of vehicle while sideslip control will determine the lateral stability. In order to improve the vehicle handling and stability performances, a combined yaw rate and sideslip tracking control are essential.
In integrated active steering and braking control, steering angle δ_f and yaw moment M_z are considered as two independent inputs for steering and braking actuators[32]. The desired yaw rate and sideslip that were generated by reference model should be able to be tracked by the yaw stability controller. Based on the tracking error, the yaw stability controller will determine an appropriate correction of steering angle δ_c and yaw moment ΔM_z as implemented in [2, 3, 5, 10, 11, 16, 17, 27, 29, 30].

In integrated active steering and braking control for vehicle yaw stability, one of the main problems that influence yaw rate and sideslip tracking control are uncertainties causedby variations of dynamics parameters such as vehicle speed [1, 6, 22], road surface adhesion coefficients [4], tire cornering stiffness [9, 10, 14, 22, 29], vehicle mass and moment of inertia [14]. Besides that, external disturbance such as lateral crosswind [8, 17] may affect the trackingcontrol of the desired yaw rate and sideslip. Therefore, the uncertainties of dynamic parameters and crosswind disturbance that have great influencesto the vehicle handling and stability performances must be overcome. In order toachieve yaw stability controlobjectives in the presenceof these problems, an appropriate control strategy is essential.

■4.0 INTEGRATED ACTIVE STEERING & BRAKING CONTROL STRATEGIES

In vehicle yaw stability system, yaw rate and sideslip tracking control can be achieved via active chassis control i.e. active steering, active braking or integrated active steering and braking control. In this section, the control structure and algorithms of integrated active steering and braking control that was designedis reviewed and discussed.

4.1Control Structure

In recent years, an integration of active front steering (AFS) and braking that are based on direct yaw moment control (DYC) method has become a popular research topic in vehicle dynamics studies especially for lateral and yaw stability control as reported in [1, 5, 9, 10, 20-22, 26, 28-30, 33]. As discussed earlier, two independent inputs to the vehicle i.e. corrective steer angle and yaw moment as illustrated in Figure 3 will give advantages for vehicle handling and stability performances improvement during steady state and critical driving manoeuvres.

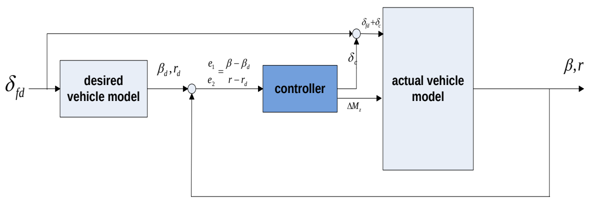


Figure 3 Control structure of integrated AFS-DYC control

To overcome the presence of uncertainties and crosswind disturbances as discussed above, robust algorithms for the controller is a necessity.

4.2Yaw Stability Control Algorithms

In order to implement robust yaw stability controller, a linear state space model of integrated active front steering and direct yaw moment control as described in equation (14) is utilized for the controller analysis. Based on this model, control algorithm is designed to cater to the uncertainties and external disturbances.

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} \dfrac{-C_f - C_r}{mv} & -1 + \dfrac{C_r l_r - C_f l_f}{mv^2} \\ \dfrac{C_r l_r - C_f l_f}{I_z} & \dfrac{-C_f l_f^2 - C_r l_r^2}{I_z v} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} \dfrac{C_f}{mv} & 0 \\ \dfrac{C_f l_f}{I_z} & \dfrac{1}{I_z} \end{bmatrix} \begin{bmatrix} \delta_c \\ \Delta M_z \end{bmatrix} + \begin{bmatrix} \dfrac{C_f}{mv} \\ \dfrac{C_f l_f}{I_z} \end{bmatrix} \delta_{fd} \tag{14}$$

Studies in [1, 4,22] considered the road adhesion coefficient, vehicle speed, cornering and braking stiffness as uncertainties parameters. Gain-scheduled algorithm is implemented in [1] whilestate feedback robust controller is designed based on linear matrix inequality (LMI) in [4, 22] to ensure the robustness of performance for integrated active front steering and active differential braking control. In integrated vehicle dynamics control of [6], phase plane method was used for dynamic stability control while sliding mode control algorithm was utilized for the active front steering in order to cater to the uncertainties. Due to the high nonlinearity and uncertainty of the wheel dynamics, the sliding mode control algorithm was utilised in [7] for active braking and active front steering control. Furthermore, it works with MR suspension system for ride comfort and vehicle stability improvement.

As discussed in [9], cornering stiffness are considered as uncertainty parameter. To ensure the robustness of the designed controller, an adaptive integrated control algorithm based on direct Lyapunov method is designed for integrated active front steering and direct yaw moment control. In [10] an optimal guaranteed cost control (OGCC) technique is utilized for active front steering and direct yaw moment control to cater for the uncertainty of tire cornering stiffness during variations of driving conditions. In [14] an integrated control of active differential and active front steering had utilized PI control scheme based on yaw rate error. Tire cornering stiffness, vehicle mass and moment of inertia are treated as uncertainties parameters.

Integrated control schemes that coordinated the steering, braking and stabiliser may consist of main loop and servo loop [15]. The main loop controller computes the stabilizing forces/moments while servo loop controller consistsof the forces/moments distribution. In this study, sliding mode controller is utilized for stabilising the forces and moments to ensure the robustness of controller. In unified chassis control scheme [16], the tire cornering stiffness is considered as uncertainty parameter and sliding mode control algorithm is used to design robust direct yaw moment control in order to improve lateral stability and vehicle handling. For global chassis control [20]a robust controller is designed based on nonlinear adaptive H_∞ control theory to attenuate an external disturbances and uncertainties of vehicle mass, moment of inertia and centre of gravity position.

Similarly in [6], the study in [27] used sliding mode control for active front steering in integrated control with electronic stability program to ensure robustness against uncertainties of vehicle mass, speed and road conditions. In [29], sliding mode control algorithm is also utilized as robust yaw stability and sideslip control for integrated active front steering and direct yaw moment. The designed controller was able to track the desired yaw rate and vehicle side slip with crosswind as an external disturbance and cornering stiffness as system uncertainty.

Based on the above discussion, integrated active front steering and direct yaw moment control or differential braking that use two independent control inputs from two different actuators i.e. steering and braking could improve the yaw rate and sideslip tracking control. From the point of view of tracking control, the transient response performance is very important. However, the control strategies and algorithms discussed above does not improve the transient response performances of the yaw rate and sideslip tracking control in the presence of uncertainties and disturbances. The designed controllers are only sufficient to track thedesired responses in the presence of such problems. Therefore, an appropriate control strategy that could improve the transientresponse performance for robust tracking control of vehicle yaw rate and sideslip shouldbe designed and developed for integrated active steering and braking control.

■5.0 HIGH PERFORMANCE ROBUST TRACKING CONTROL STRATEGY

In previous sections, the control algorithms to ensure a robustyaw stability controller designwere discussed. In this section, a proposed robust tracking control strategy with high performance based on composite non-linear feedback (CNF) control and sliding mode control (SMC) is discussed in brief.

5.1Composite Nonlinear Feedback Control

The composite nonlinear feedback (CNF) control technique that is based on state feedback law could improve the transient performances as designed and implemented in [34] for tracking control of 2nd order linear system and higher order MIMO linear system as implemented in [35]. It was extended and generalized for linear system with actuator nonlinearities in [36], general multivariable system with input saturation in [37], hard disk drive servo system, servo positioning system with disturbance in [38-41] and is continuously developed and applied in last decade.

As in the above studies, the use of CNF control could improve the performance of transient response, especially for a second order system based on variable damping ratios. For most practical situation, it is desired to obtain a fast response 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 keep low damping ratio during transient and varied to high damping ratio as the output response closed to the set point as illustrated in Figure 4.

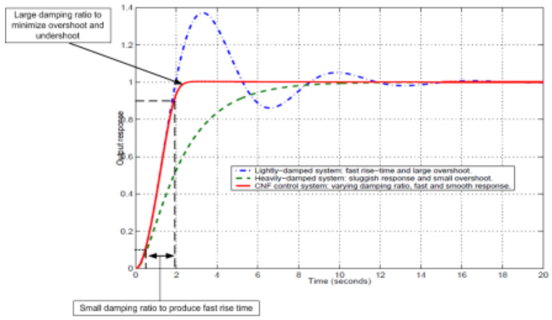


Figure 4 Transient response improvement using a CNF control technique [36]

To realize this concept of variable damping ratio, the CNF control law is designed toinclude the linear and nonlinear feedback laws as described in the following equations;

$$u = u_L + u_N$$
$$u = Fx + Gr + \rho(r, y)B'P(x - x_e) \tag{15}$$

where F is feedback matrix, G is a scalar matrix, B is input matrix.If P>0 is a solution of the Lyapunov equation and $\rho(r, y)$ is a nonlinear function which is chosenas follows;

$$\rho(r, y) = -\beta e^{-\alpha(y-r)^2} \tag{16}$$

$$\rho(r, y) = -\beta e^{-\alpha|y-r|} \tag{17}$$

$$\rho(r, y) = -\frac{\beta}{1-e^{-1}}\left(e^{-(1-(y-y_0)/(r-y_0))^2} - e^{-1}\right) \tag{18}$$

The nonlinear function $\rho(r, y)$ is used to change the damping ratio of closed loop system from low to final high value as the output approaches the reference *r*. The parameters α and β are appropriate positive performances i.e.tuned to yield a desired performances giving fast settling time and small overshoot.

Although the CNF control technique has an advantage in improving transient response, its control structure could not cater to the system with uncertainties and varying disturbances. Therefore, to make the CNF robust without changing its structure, it could be combined with sliding mode control for high performance robust tracking control.

5.2Sliding Mode Control with Nonlinear Sliding Surface

Sliding mode control (SMC) algorithm that had been developed in the last two decades is recognized as an effective robust controller to cater to the matched and mismatched uncertainties and disturbances for linear and nonlinear system. It is also utilized as an observer for estimation and identification purposes in engineering system. Various applications using SMC such as in motion control system, servo system, continuous or discrete system, fuzzy and neural network system, electrical motor and drive, power electronics converters, magnetic bearings, advanced robotics, automotive vehicles and underwater vehicles are discussed in [42-44].

Sliding mode control design consists of two important steps i.e. the design of a sliding surface and the design of the control law so that the system states are enforced to the sliding surface. The design of the sliding surface is very important as it will determine the dynamics of the system being controlled. A conventional linear sliding surface of SMC has a disadvantage in improving transient response performance of the system due to constant close loop damping ratio. Thus, a nonlinear sliding surface that changes a closed loop system damping ratio to achieve high performance of the transient response, and at the same time ensure the robustness has been proposed in [45-47]. This concept of varying close loop damping ratio is based on the composite nonlinear feedback (CNF) control as discussed previously. Recently, this type of nonlinear sliding surface is extensively applied for SMC design in current research works [48-50].

Based on tracking error, the nonlinear sliding surface using a nonlinear function of CNF for yaw stability control system can be described as follows

$$s(z,t) := c^T e(t) = \begin{bmatrix} c_1 & I_m \end{bmatrix} \begin{bmatrix} e_1(t) \\ e_2(t) \end{bmatrix} \tag{19}$$

where

$$c_1 := F - \rho(r, y)a_{12}^T P \tag{20}$$

with $e_1(t)$ and $e_2(t)$ representingthe yaw rate and sideslip tracking error respectively. Notice that the nonlinear function $\rho(r, y)$ can be chosen from equations (16 - 18) and I_m is the identity matrix.

Based on the above discussion, the SMC with nonlinear sliding surface based on CNF technique could achieve high performance with the robustness 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 is 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 with integrated active steering and braking,a robust yaw rate and sideslip tracking control.

■6.0 CONCLUSION

The main elements of vehicle yaw stability control system have been reviewed. Vehicle models, control objectives, control problems and control algorithms for integrated active steering and braking control have been discussed and presented. It is observed that the designed controllers is not capable to improve the transient response of the yaw rate and sideslip tracking control, especially in the presence of uncertainties and disturbances where theyare only sufficient to track the desired responses. It is also found thata control strategy using the sliding mode control algorithm with nonlinear sliding surface based on CNF control technique is proposed. It is expected that such control strategy could be implemented for integrated active steering and braking control in order to achieve high performances for robust tracking control of vehicle's yaw rate and sideslip, improvingthe vehicle handling and stability.

Acknowledgement

The authors are grateful to the Ministry of Education of Malaysia, UTeM, UTM, and UniKL for the support to conductthis research.

References

[1] S.C. Baslamisli, I. E. Köse, G. Anlaş.2009.Gain-scheduled integrated active steering and differential control for vehicle handling improvement.*Vehicle System Dynamics*.47(1): 99–119.

[2] R. Karbalaei, M. Ali, S. Hamed Tabatabaei, R. Kazemi, A. Ghaffari. 2008.Integrated control of AFS and DYC in the vehicle yaw stability management system using fuzzy logic control*SAE Technical Papers, SAE World Congress, Michigan, USA,2008-01-1262*.

[3] Y. Wu, D. Song, Z. Hou, X. Yuan. 2007.A fuzzy control method to improve vehicle yaw stability based on integrated yaw moment control and active front steering.*Proceedings of IEEE International Conference on Mechatronics and Automation (ICMA 2007)*. Harbin, China.1508–1512.

[4] S. C.Başlamişli, I. E Köse, G. Anlaş.2006.Design of active steering and intelligent braking systems for road vehicle handling improvement: A robust control approach.*Proceedings of IEEE International Conference on Control Applications (CCA 2006)*. Munich, Germany.909–914.

[5] P. Falcone, H. Eric Tseng, F. Borrelli, J. Asgari, D.Hrovat. 2008.MPC-based yaw and lateral stabilisation via active front steering and braking.*Vehicle System Dynamics*.46(1):611–628.

[6] J. He, D. A. Crolla, M.C Levesley, W. J Manning. 2006.Coordination of active steering, driveline, and braking for integrated vehicle dynamics control," proceedings of the Institution of Mechanical Engineers. *Part D: Journal of Automobile Engineering,* 220(10):1401–1421.

[7] S. B. Lu,Y. N Li, S. B. Choi, L. Zheng, M. S. Seong.2011.Integrated control on MR vehicle suspension system associated with braking and steering control.*Vehicle System Dynamics*.49(1–2):361–380.

[8] J. Wu, Q. Wang, X.Wei, H. Tang. 2010.Studies on improving vehicle handling and lane keeping performance of closed-loop driver-vehicle system with integrated chassis control.*Mathematics and Computers in Simulation*.80(12):2297–2308.

[9] N. Ding, S. Taheri. 2010.An adaptive integrated algorithm for active front steering and direct yaw moment control based on direct Lyapunov method.*Vehicle System Dynamics*.48(10):1193–1213.

[10] X. Yang, Z.Wang, W. Peng. 2009.Coordinated control of AFS and DYC for vehicle handling and stability based on optimal guaranteed cost theory.*Vehicle System Dynamics*.47(1):57–79.

[11] B.Lee, A. Khajepour, K. Behdinan.2006.Vehicle stability through integrated active steering and differential braking.*SAETechnical Papers, SAE World Congress,Michigan, USA*.2006-01-1022.

[12] M. Selby, W. J.Manning, M. D. Brown, D. A Crolla. 2001.A coordination approach for DYC and active front steering,*SAE Technical Papers, SAE World Congress, Michigan, USA*,2001-01-1275.

[13] S. Moon, W. K.Cho, K. S Yi.2010.Intelligent vehicle safety control strategy in various driving situations.*Vehicle System Dynamics*.48:537–554.

[14] R. Marino, S.Scalzi, 2009.Integrated control of active steering and electronic differentials in four wheel drive vehicles.*SAE International Journal of Passenger Cars - Electronic and Electrical Systems*.2(1):141–149.

[15] D. Li, S. Du, F. Yu. 2008, Integrated vehicle chassis control based on direct yaw moment, active steering and active stabiliser.*Vehicle System Dynamics*.46(1):341–351.

[16] W. Cho, J. Yoon,J. Kim,J.Hur, K. Yi. 2008.An investigation into unified chassis control scheme for optimised vehicle stability and manoeuvrability.*Vehicle System Dynamic*.46(1):87–105.

[17] D. Li, F. Yu.2007.A novel integrated vehicle chassis controller coordinating direct yaw moment control and active steering.*SAE Technical Papers, proceeding of 14th Asia Pacific Automotive Engineering Conference, California, USA*.2007-01-3642

[18] S. H. Tamaddoni, S. Taheri, M. Ahmadian. 2011.Optimal preview game theory approach to vehicle stability controller design.*Vehicle System Dynamics*.49(12):1967–1979.

[19] R. Rajamani, *Vehicle Dynamics and Control: 2nd Ed*, 2011, USA, Springer, Chapter 2.

[20] Z. He, X. Ji. 2012.Nonlinear robust control of integrated vehicle dynamics.*Vehicle System Dynamics*.50(2):247–280.

[21] C. Rengaraj, D. Crolla. 2011.Integrated chassis control to improve vehicle handling dynamics performance.*SAE Technical Papers, SAE 2011 World Congress and Exhibition, Detroit, USA*.2011-01-0958.

[22] S. C. Baslamisli, I. E Kose, G. Anlaş. 2011.Handling stability improvement through robust active front steering and active differential control.*Vehicle System Dynamics*.49(5):657–683.

[23] J. Wang, R. G. Longoria. 2009.Coordinated and reconfigurable vehicle dynamics control.*IEEE Transactions on Control Systems Technolog*.17(3): 723–732.

[24] A. Goodarzi, A.Sabooteh, E. Esmailzadeh. 2008.Automatic path control based on integrated steering and external yaw-moment control.*Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics*.222(2):189–200.

[25] H. B. Pacejka, *Tyre and Vehicle Dynamics*: 2nd Ed, 2006, UK, Butterworth-Heinemann, Chapter 1 & 4.

[26] C. Poussot-Vassal, O. Sename, L.Dugard, S. M. Savaresi. 2011.Vehicle dynamic stability improvements through gain-scheduled steering and braking control.*Vehicle System Dynamics*.49(10): 1597–1621.

[27] J. Guo,L. Chu, H. Liu, M. Shang, Y. Fang. 2010.Integrated control of Active Front Steering and Electronic Stability Program,*IEEE International Conference on Advanced Computer Control(ICACC 2010)*.449–453.

[28] C.Ahn, B.Kim, M. Lee.2012.Modeling and control of an anti-lock brake and steering system for cooperative control on split-mu surfaces.*International Journal of Automotive Technology*.13(4): 571–581.

[29] A. Hu, F. He. 2011.Variable structure control for active front steering and direct yaw moment.*Proceedings of 2nd International Conference on Artificial Intelligence, Management Science and Electronic Commerce (AIMSEC 2011), Zhengzhou, China*.3587–3590.

[30] A. Hu, B. Lv.2010.Study on mixed robust control for integrated active front steering and direct yaw moment.*Proceedings of IEEE International Conference on Mechatronics and Automation (ICMA 2010), Xi'an, China*.29–33.

[31] W. J. Manning, D. A. Crolla. 2007.A review of yaw rate and sideslip controllers for passenger vehicles.*Transactions of the Institute of Measurement and Control*.29(2): 117–135.

[32] F. Yu, D. F Li,D. A. Crolla.2008.Integrated vehicle dynamics control-state-of-the art review.*Proceedings of IEEE Vehicle Power and Propulsion Conference (VPPC 2008), Harbin, China*.

[33] J. Tjoønnäs, T. A. Johansen.2010.Stabilization of automotive vehicles using active steering and adaptive brake control allocation.*IEEE Transactions on Control Systems Technology*.18(3): 545–558.

[34] Z.Lin,M.Pachter, S. Ban.1998.Toward improvement of tracking performance - Nonlinear feedback for linear systems.*International Journal of Control*.70(1): 1–11.

[35] M. C. Turner, I.Postlethwaite, and D. J. Walker. 2000.Non-linear tracking control for multivariable constrained input linear systems.*International Journal of Control*.73(12): 1160–1172.

[36] B. M.Chen,T. H.Lee, K. Peng, V. Venkataramanan. 2003.Composite nonlinear feedback control for linear systems with input saturation: Theory and an application.*IEEE Transactions on Automatic Control*.48(3): 427–439.

[37] Y. He, B. M.Chen, C. Wu. 2005.Composite nonlinear control with state and measurement feedback for general multivariable systems with input saturation.*Systems and Control Letters*.54(5): 455–469.

[38] W. Lan, C. K.Thum, B. M. Chen. 2010.A hard-disk-drive servo system design using composite nonlinear-feedback control with optimal nonlinear gain tuning methods.*IEEE Transactions on Industrial Electronics*.57(5): 1735–1745.

[39] G. Cheng, K. Peng. 2007.Robust composite nonlinear feedback control with application to a servo positioning system.*IEEE Transactions on Industrial Electronics*.54(2): 1132–1140.

[40] G.Cheng, W. Jin. 2006.Parameterized design of nonlinear feedback controllers for servo positioning systems.*Journal of Systems Engineering and Electronics*.17(3): 593–599.

[41] K. Peng, B. M.Chen, G. Cheng, T. H. Lee.2005.Modeling and compensation of nonlinearities and friction in a micro hard disk drive servo system with nonlinear feedback control.*IEEE Transactions on Control Systems Technology*.13(5): 708–721.

[42] Wilfrid Perruquetti, Jean Pierre Barbot.2002. *Sliding Mode Control in Engineering*. New York, USA, Marcel Dekker Inc.

[43] Hocine Imine, Leonid Fridman, Hassan Shraim, Mohamed Djemai.2011. *Sliding Mode Based Analysis and Identification of Vehicle Dynamics*.Berlin, Germany, Springer.

[44] Vadim Utkin, Jürgen Guldner, Jingxin Shi. 2008.*Sliding Mode Control in Electro Mechanical Systems*, 2nd Ed.New York, USA, CRC Press.

[45] B.Bandyopadhyay, D.Fulwani, I.Postlethwaite, M. C.Turner.2010.A nonlinear sliding surface to improve performance of a discrete-time input-delay system.*International Journal of Control*, 83(9): 1895–1906.

[46] B.Bandyopadhyay, D. Fulwani. 2009.A robust tracking controller for uncertain MIMO plant using non-linear sliding surface.*Proceedings ofIEEE International Conference on Industrial Technology, ICIT, Churchill, USA*.

[47] Bijnan Bandyopadhyay, Kyung-Soo Kim, Fulwani Deepak.2009. *Sliding Mode Control Using Novel Sliding Surfaces.*Berlin, Germany, Springer.
[48] S. Mondal, C. Mahanta.2012.A fast converging robust controller using adaptive second order sliding mode.*ISA Transactions*.51(6): 713–721.
[49] S. Mobayen, V. J. Majd, M. Sojoodi.2012.An LMI-based composite nonlinear feedback terminal sliding-mode controller design for disturbed MIMO systems.*Mathematics and Computers in Simulation,* 85: 1-10.
[50] S.Mobayen, V. J. Majd, M. Sojoodi. 2012.An LMI-based finite-time tracker design using nonlinear sliding surfaces.*Proceedings of 20thIranian Conference on Electrical Engineering, (ICEE 2012), Tehran, Iran,*810–815.