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Yaw Stability Control of SBW Electric Vehicle

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Abstract

More and more popular electric vehicles are excellent application platforms for steer-by-wire (SBW) system. Handling stabilities and active safety are greatly improved when SBW is adopted in electric vehicle. The proposed new architecture of SBW electric vehicle is introduced, its dynamics model is built after the principle and structure. The algorithm achieves the yaw motion of electric vehicle by the feedback of both yaw rate and front steering angle. The coordination of these actuators is achieved through the controller of the feedback gains with respect to electric vehicle speed. The gain scheduled steering controller provides the desired yaw rate damping while keeping the yaw-lateral motion decoupled. The simulation test results show the effectiveness of the proposed architecture and steering control system when the SBW electric vehicle is subject to critical driving situations.

Keywords: *Electric vehicle; SBW; Dynamics; Yaw stability*

1. Introduction

A trend in electric vehicles is the application of safety systems to improve electric vehicle handling, stability and comfort. Steer-by-wire (SBW) electric vehicle are expected to improve active safety. Concerning active safety, electric vehicle stability and steering maneuverability are improved by electronic control. Nowadays, many active chassis control technology have been developed and brought into the market, Anti-lock Braking System (ABS) prevents wheel lock-up, Electronic Stability Control (ESC) enhances vehicle lateral stability, and Driving Torque Distribution (DTD) enhance vehicle stability but via driving-torque.

All these control technologies have been proven to provide certain advantages under certain conditions. Shun-Chang Chang [1] investigated a two degree-of-freedom control system in a steer-by-wire vehicle dynamic system. They provided simulation results for the proposed controller in order to validate the claims of improved vehicle stability and steering response. Zhenhai Gao *et al.*, [2] described kinematic models of planetary gear set and steering gear are established, based on the analysis of the transmission mechanism of angle superposition with active front steering system. A. Emre Cetin *et al.*, [3] proposed on-line parameter identification method for a steer-by-wire system that featured algorithms in order to enhance the maneuverability and stability of the vehicle. Ali Tavasoli *et al.*, [4] discussed optimized coordination of brakes and active steering for a 4WS passenger car. Sohel Anwar [5] presented Generalized predictive control of yaw dynamics of a hybrid brake-by-wire equipped vehicle. tests results were presented for single lane change to validate the proposed control algorithms.

The development of the SBW electric vehicle is still the object of intense research activities from both industrial and academic sides. The various electric vehicle dynamics control systems can be classified into longitudinal, lateral and vertical control of translational electric vehicle motions. This work focuses on new architecture and active steering control inside the SBW electric vehicle.

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2. Architecture of the SBW Electric Vehicle

Figure 1 shows architecture of the SBW electric vehicle. It consists of steering resistance torque actuator 1; steering wheel angle sensor 2; front wheel angle sensor 3; steering actuator 4; electronic controller (ECU) 5; and some conventional sensors 6 to monitor vehicle speed, lateral acceleration and yaw rate.

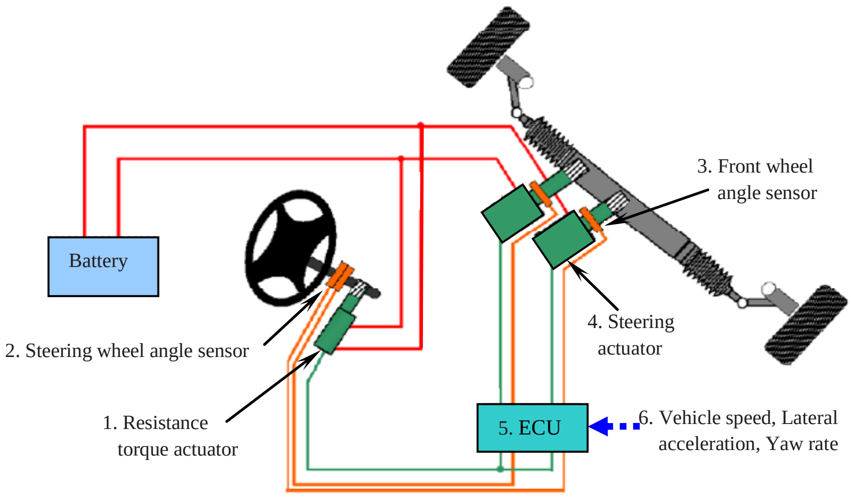

Figure 1. Architecture of the SBW Electric Vehicle

2.1. Resistance Torque Actuator

The resistance torque actuator consists of a steering shaft and an electric motor with a reduction gear for resistance torque against the driver's steering maneuvering. This portion of SBW is light in weight and small in volume, contributing to a comfortable cabin and a flexible cockpit design as compared with conventional power steering.

2.2. Steering Actuator

A typical steering actuator, like the one shown in Figure 1, consists of a ball screw and an electric motor in a concentric arrangement with a steering rod axis between front wheels. Other types of steering actuators consist of rack and pinion mechanism and electric motor with a reduction gear. The ball screw type steering actuator exhibits more accurate and quicker response with less friction, less backlash, and higher stiffness.

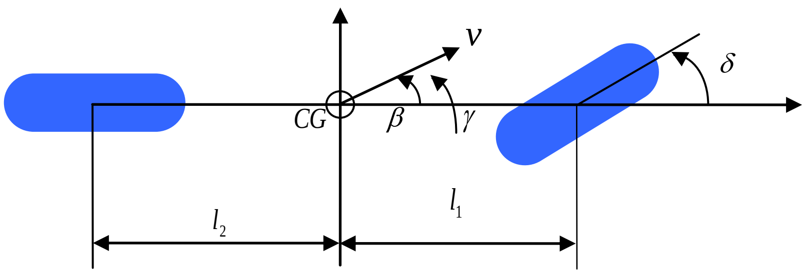

Figure 2. 2-DOF Model of Lateral Electric Vehicle Dynamics

3. Electric Vehicle Dynamics

The dynamics of electric vehicle steering is described by the 2-DOF (Degree of Freedom) classical linear bicycle model. It is obtained by lumping the right and left wheels together in the center of the front and rear axle as shown in Figure 2. The model describes the lateral and yaw motions and neglects the roll motion.

The corresponding linearized dynamic for the electric vehicle can be described with the following equation:

$$\begin{cases} mv\dot{\beta} + \frac{1}{v}\left(mv^2 + k_1 l_1 - k_2 l_2\right)\gamma + \left(k_1 + k_2\right)\beta - k_1\delta = 0 \\ J\dot{\gamma} + \frac{1}{v}\left(k_1 l_1^2 + k_2 l_2^2\right)\gamma - \left(k_2 l_2 - k_1 l_1\right)\beta - k_1 l_1 \delta = 0 \end{cases} \quad (1)$$

where β is sideslip angle between the vehicle center and the velocity at the center of CG, γ is yaw rate with respect to an inertial coordinate system, δ is front steering angle, k_1 (k_2) is cornering stiffness for the front (rear) wheel, l_1 (l_2) is distance from the center of gravity to the front (rear) axis's, $l_1 + l_2$ = wheel base, m is vehicle mass, J is moment of inertia with respect to vertical axis, v is electric vehicle longitudinal velocity which we assume always greater than zero.

Rewriting the (1) into state space format, we have

$$\begin{bmatrix} \dot{\beta} \\ \dot{\gamma} \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \begin{bmatrix} \beta \\ \gamma \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \end{bmatrix} \delta \quad (2)$$

Where

$$a_{11} = -\frac{k_1 + k_2}{mv},\ a_{12} = -1 - \frac{k_1 l_1 - k_2 l_2}{mv^2},\ a_{21} = -\frac{k_1 l_1 - k_2 l_2}{J},\ a_{22} = -\frac{k_1 l_1^2 + k_2 l_2^2}{Jv},\ b_1 = \frac{k_1}{mv},\ b_2 = \frac{k_1 l_1}{J} \quad (3)$$

Consider the following assumptions given in:

(a) The cornering stiffness for the front and rear wheels (k_1 and k_2) are defined as follows:

$$F_1 = k_1 \varepsilon_1,\ F_2 = k_2 \varepsilon_2 \quad (4)$$

where ε_1 & ε_2 are the wheel slip angles and F_1 and F_2 are the cornering forces for the front and rear wheels respectively.

(b) The longitudinal mass distribution is equivalent to two masses concentrated at the front and rear axles. Then,

$$J = m l_1 l_2 \quad (5)$$

That is to say that the center of gravity is not shifted by changing the mass m, or we can say that l_1 and l_2 are constant.

Then, the coefficients of the state space mode become

$$a_{11} = -\frac{k_1 + k_2}{mv},\ a_{12} = -1 - \frac{k_1 l_1 - k_2 l_2}{mv^2},\ a_{21} = -\frac{k_1 l_1 - k_2 l_2}{m l_1 l_2},\ a_{22} = -\frac{k_1 l_1^2 + k_2 l_2^2}{m l_1 l_2 v},\ b_1 = \frac{k_1}{mv},\ b_2 = \frac{k_1}{m l_2} \quad (6)$$

4. Steering Control System Analysis

We consider the steering control system that decouples the yaw dynamics and lateral motion of the electric vehicle. These decoupled equations are then used to design a second order controller for yaw stabilization and damping with more tunable parameters to achieve optimal performance which is defined as achieving maximum lateral acceleration without yaw instability and rollover in a steering maneuver. The control objective is to provide the decoupled yaw dynamics equation more freedom to be tuned via the second order controller (ECU) structure. We will first consider the controller that has the following state space form:

$$\begin{cases} \dot{x} = -ex + u \\ \delta = x + fu \end{cases} \tag{7}$$

With

$$u = \gamma_r - \lambda_1 \gamma - \lambda_2 x \tag{8}$$

where e and f are tunable controller parameters. γ_r is the reference yaw rate. λ_1 and λ_2 are controller gains. x is an intermediate variable that relates steering wheel angle δ to control u.

The corresponding closed loop system can be represented by the following state space equations:

$$\begin{bmatrix} \dot{\beta} \\ \dot{\gamma} \\ \dot{x} \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & b_1 \\ a_{21} & a_{22} & b_2 \\ 0 & 0 & -e \end{bmatrix} \begin{bmatrix} \beta \\ \gamma \\ x \end{bmatrix} + \begin{bmatrix} b_1 f \\ b_2 f \\ 1 \end{bmatrix} u \tag{9}$$

By selecting the new state vector as introduced in [7],

$$\begin{bmatrix} a_1 \\ \gamma \\ x \end{bmatrix} = \begin{bmatrix} -k & -\frac{kl_1}{v} & k \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} \beta \\ \gamma \\ x \end{bmatrix} \tag{10}$$

where a_1 is the front axle lateral acceleration, and

$$k = \frac{k_1 l}{m l_2} \tag{11}$$

then we have

$$\begin{bmatrix} \dot{a}_1 \\ \dot{\gamma} \\ \dot{x} \end{bmatrix} = \begin{bmatrix} c_{11} & k & -ke \\ c_{21} & c_{22} & c_{23} \\ 0 & 0 & -e \end{bmatrix} \begin{bmatrix} a_1 \\ \gamma \\ x \end{bmatrix} + \begin{bmatrix} kc_1 \\ b_2 f \\ 1 \end{bmatrix} u \tag{12}$$

where

$$c_{11} = -\frac{k_1 l}{mvl_2}, c_{21} = -\frac{k_1 l_1 - k_2 l_2}{m l_1 l_2}, c_{22} = -\frac{k_2 l}{m l_1 v}, c_{23} = \frac{k_2}{m l_1}, c_1 = 1 - b_1 f \left(1 + \frac{l_1}{l_2}\right) \tag{13}$$

Substituting Eq. (8) into Eq. (12), we have

$$\begin{bmatrix} \dot{a}_1 \\ \dot{\gamma} \\ \dot{x} \end{bmatrix} = \begin{bmatrix} c_{11} & k - kc_1\lambda_1 & -ke - kc_1\lambda_2 \\ c_{21} & c_{22} - b_2 c\lambda_1 & c_{22} - b_2 f\lambda_2 \\ 0 & -\lambda_1 & -e - \lambda_2 \end{bmatrix} \begin{bmatrix} a_1 \\ \gamma \\ x \end{bmatrix} + \begin{bmatrix} kc_1 \\ b_2 f \\ 1 \end{bmatrix} \gamma_r = \begin{bmatrix} c_{11} & k(1 - c_1\lambda_1) & -k(a + c_1\lambda_2) \\ c_{21} & c_{22} - b_2 f\lambda_1 & c_{22} - b_2 f\lambda_2 \\ 0 & -\lambda_1 & -e - \lambda_2 \end{bmatrix} \begin{bmatrix} a_1 \\ \gamma \\ x \end{bmatrix} + \begin{bmatrix} kc_1 \\ b_2 f \\ 1 \end{bmatrix} \gamma_r \tag{14}$$

Let λ_1, λ_2, e, and c_1 have the following relations:

$$1 - c_1\lambda_1 = 0, \ e + c_1\lambda_2 = 0 \tag{15}$$

or

$$c_1 = \frac{1}{\lambda_1}, \ e = -\frac{\lambda_2}{\lambda_1} \tag{16}$$

Then (14) becomes:

$$\begin{bmatrix} \dot{a}_1 \\ \dot{\gamma} \\ \dot{x} \end{bmatrix} = \begin{bmatrix} c_{11} & 0 & 0 \\ c_{21} & c_{22} - b_2 f\lambda_1 & c_{22} - b_2 f\lambda_2 \\ 0 & -\lambda_1 & -e - \lambda_2 \end{bmatrix} \begin{bmatrix} a_1 \\ \gamma \\ x \end{bmatrix} + \begin{bmatrix} kc_1 \\ b_2 f \\ 1 \end{bmatrix} \gamma_r \tag{17}$$

State space equation (17) is in the canonical form, and it shows that γ and x are not observable from a_1. Thus the closed loop control system formed by (12) and (13) with c_1 defined as in (16) decouples a_1 from γ and x. Thus we can say that the steering dynamics is split into the two subsystems by the closed loop control. One subsystem is the lateral motion of the front axle represented by

$$\dot{a}_1 = c_{11}a_1 + kc_1\gamma_r \tag{18}$$

The other is the yaw motion represented by

$$\begin{bmatrix} \dot{\gamma} \\ \dot{x} \end{bmatrix} = \begin{bmatrix} c_{22} - b_2 f\lambda_1 & c_{23} - b_2 f\lambda_2 \\ -\lambda_1 & -a - \lambda_2 \end{bmatrix} \begin{bmatrix} \gamma \\ x \end{bmatrix} + \begin{bmatrix} b_2 f \\ 1 \end{bmatrix} \gamma_r + \begin{bmatrix} c_{21} \\ 0 \end{bmatrix} a_1 \tag{19}$$

As stated in the [8], the driver only has to control the lateral motion subsystem, keeping the car, as a mass point at the front axle, on top of the planned path by generating a lateral acceleration via the transfer function,

$$\frac{a_1}{\gamma_r} = \frac{kc_1}{s - c_{11}} = \frac{v - \frac{k_1}{m} f\left(1 + \frac{l_1}{l_2}\right)}{1 + \frac{mvl_2}{k_1 l} s} \tag{20}$$

The decoupled yaw motion will be attenuated automatically by the control system that has the following characteristic polynomial,

$$p(s) = \begin{vmatrix} s - (c_{22} - b_2 f\lambda_1) & -(c_{23} - b_2 f\lambda_2) \\ \lambda_1 & s + (e + \lambda_2) \end{vmatrix} = s^2 - 2\xi_c\omega_c s - \omega_c^2 \tag{21}$$

where

$$\begin{aligned} \omega_c^2 &= \lambda_1(c_{23} - b_2 f\lambda_2) + (e + \lambda_2)(b_2 f\lambda_1 - c_{22}) = \lambda_1\left(\frac{k_2}{ml_1} + \frac{(\lambda_1 - 1)v}{l}e\right) + e(1 - \lambda_1)\left(\frac{(\lambda_1 - 1)v}{l} + \frac{k_2 l}{mvl_1}\right) \\ &= \lambda_1\frac{k_2}{ml_1} + \frac{(\lambda_1 - 1)v}{l}e + \frac{k_2 l}{mvl_1}e - \frac{k_2 l}{mvl_1}\lambda_1 e = \lambda_1\frac{k_2}{ml_1} + e(\lambda_1 - 1)\left(\frac{v}{l} - \frac{k_2 l}{mvl_1}\right) \end{aligned} \tag{22}$$

$$\xi_c = \frac{(e + \lambda_2) + (b_2 f\lambda_1 - c_{22})}{2\omega} = \frac{(\lambda_1 - 1)\left(\frac{v}{l} - e\right) + \frac{k_2 l}{mvl_1}}{2\sqrt{\lambda_1\frac{k_2}{ml_1} + e(\lambda_1 - 1)\left(\frac{v}{l} - \frac{k_2 l}{mvl_1}\right)}} \tag{23}$$

Here we have used the relationships established in (9), (13), and (15) and,

$$c_1 = 1 - b_1 f\left(1 + \frac{l_1}{l_2}\right) \tag{24}$$

$$f = \frac{(\lambda_1 - 1)ml_2 v}{\lambda_1 k_1 l} \tag{25}$$

5. Stability Analysis and Control Parameter Selection

It can be seen from Eq. (20) that for lateral motion to be stable, the following condition must be satisfied:

$$c_{11} < 0 \rightarrow \frac{k_1 l}{mvl_2} > 0 \tag{26}$$

Since k_1 and other parameters are positive, the above condition is easily satisfied when the vehicle is moving forward ($v > 0$).

For yaw stability, both (22) and (23) should be greater than 0. For $\lambda_1 = 1$, these conditions are easily satisfied for any positive vehicle velocity. If non-unity value for λ_1 is to be selected, then the stability condition becomes as follows:

$$0 < \lambda_1 < 1 \quad if \quad v < \max\left(el,\ l\sqrt{\frac{k_2}{ml_1}}\right), \quad and \quad \lambda_1 > 1 \quad if \quad v > \max\left(el,\ l\sqrt{\frac{k_2}{ml_1}}\right) \tag{27}$$

To ensure yaw stability, λ_1 can either be set to unity or can be scheduled with respect to the vehicle velocity based on the above stability conditions. It is also noticed from Eq. (20) that for better handling, the numerator of the transfer function should be positive, *i.e.*, $kc_1 > 0$ or

$$v - \frac{k_1}{m}c\left(1 + \frac{l_1}{l_2}\right)k_1 > 0 \rightarrow c < \frac{ml_2}{(l_1 + l_2)k_1} \rightarrow c < \frac{ml_2}{lk_1}v \tag{28}$$

For the controller in the forms of (7) and (8) there are two free parameters, two out of e, c, λ_1, and λ_2, that can be adjusted to meet the control system performance requirement. Without loss of generality, we will consider e, and λ_1.

Recall from (20) that the lateral motion transfer function is:

$$\frac{a_1}{\gamma_r} = \frac{kc_1}{s - c_{11}} = \frac{v - \frac{k_1}{m} f\left(1 + \frac{l_1}{lv}\right)}{1 + \frac{mvl_2}{k_1 l}s} \tag{29}$$

To make vehicle lateral motion controllable, we must have

$$kc_1 > 0 \tag{30}$$

From (16), this means that

$$\lambda_1 > 0 \tag{31}$$

For the yaw damping, we have ξ_c from Eq. (23). It can be easily seen that by selecting a properly, and $\lambda_1 > 1$, ξ_c increases when v increases. Hence, by scheduling e and λ_1 with respect to vehicle speed, we can adjust the yaw damping without sacrificing the decoupling while having freedom to tune the closed loop transfer function for lateral motion.

For the case when $e = 0$, we have:

$$\xi_c = \frac{(e + \lambda_2) + (b_2 f\lambda_1 - c_{22})}{2\omega} = \frac{(\lambda_1 - 1)\left(\frac{v}{l} - e\right) + \frac{k_2 l}{mvl_1}}{2\sqrt{\lambda_1 \frac{k_2}{ml_1} + e(\lambda_1 - 1)\left(\frac{v}{l} - \frac{k_2 l}{mvl_1}\right)}} = \frac{(\lambda_1 - 1)\frac{v}{l} + \frac{k_2 l}{mvl_1}}{2\sqrt{\lambda_1 \frac{k_2}{ml_1}}} \tag{32}$$

Again the damping ratio can be adjusted by gain-scheduling λ_1 with respect to vehicle speed. But the controlled lateral motion dynamics will not be influenced by the tuning of yaw damping.

For the case when λ_1 is selected as 1, we have $f = 0$ from Eq. (25), and

$$\xi_c = \frac{(e + \lambda_2) + (b_2 f\lambda_1 - c_{22})}{2\omega} = \frac{(\lambda_1 - 1)\left(\frac{v}{l} - e\right) + \frac{k_2 l}{mvl_1}}{2\sqrt{\lambda_1 \frac{k_2}{ml_1} + e(\lambda_1 - 1)\left(\frac{v}{l} - \frac{k_2 l}{mvl_1}\right)}} = \frac{(\lambda_1 - 1)\frac{v}{l} + \frac{k_2 l}{mvl_1}}{2\sqrt{\lambda_1 \frac{k_2}{ml_1}}} = \frac{\frac{k_2 l}{mvl_1}}{2\sqrt{\frac{k_2}{ml_1}}} = \frac{l}{2v}\sqrt{\frac{k_2}{ml_1}} \tag{33}$$

It can be easily verified that this is the same robust decoupling case presented in [7]. In this case, the damping ratio is fixed with given vehicle parameters. However, the yaw damping decreases with increasing vehicle speed in this case.

More parameter tuning freedom is obtained by increasing the order of the controller. Here we consider only the second order controller structure. We assume that controller has the following form:

$$\begin{cases} \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -e_2 & -e_1 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 1 \\ g \end{bmatrix} u \\ \delta = x_1 + fu \\ u = \gamma_r - \lambda_1 \gamma - \lambda_2 x_2 \end{cases} \tag{34}$$

In the above equation, x_1 and x_2 are intermediate state variables that relate the control to the steering angle. e_1, e_2, and g are controller parameters. Then, corresponding augmented state space model is given by:

$$\begin{bmatrix} \dot{\beta} \\ \dot{\gamma} \\ \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & b_1 & 0 \\ a_{21} & a_{22} & b_2 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & -e_2 & -e_1 \end{bmatrix} \begin{bmatrix} \beta \\ \gamma \\ x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} b_1 f \\ b_2 f \\ 1 \\ g \end{bmatrix} \begin{bmatrix} b_1 f \\ b_2 f \\ 1 \end{bmatrix} u \tag{35}$$

Similar to the previous section, we introduce the new state vector as follows:

$$\begin{bmatrix} a_1 \\ \gamma \\ x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} -k & \frac{kl_1}{v} & k & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} \beta \\ \gamma \\ x_1 \\ x_2 \end{bmatrix} \tag{36}$$

Then the state space equation takes the form:

$$\begin{bmatrix} \dot{a}_1 \\ \dot{\gamma} \\ \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} c_{11} & k & 0 & k \\ c_{21} & c_{22} & c_{23} & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & -e_2 & -e_1 \end{bmatrix} \begin{bmatrix} a_1 \\ \gamma \\ x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} kc_1 \\ fb_2 \\ 1 \\ g \end{bmatrix} u \tag{37}$$

Substituting $u = \gamma_r - \lambda_1 \gamma - \lambda_2 x_2$ into (37), we obtain:

$$\begin{bmatrix} \dot{a}_1 \\ \dot{\gamma} \\ \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} c_{11} & k(1 - c_1\lambda_1) & 0 & k(1 - c_1\lambda_2) \\ c_{21} & c_{22} - b_2 c \lambda_1 & c_{23} & -b_2 f \lambda_2 \\ 0 & -\lambda_1 & 0 & 1 - \lambda_2 \\ 0 & -g\lambda_1 & -e_2 & -(e_1 + g\lambda_2) \end{bmatrix} \begin{bmatrix} a_1 \\ \gamma \\ x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} kc_1 \\ fb_2 \\ 1 \\ g \end{bmatrix} \gamma_r \tag{38}$$

Setting

$$1 - c_1\lambda_1 = 0, \; a - c_1\lambda_2 = 0 \tag{39}$$

we obtain:

$$\begin{bmatrix} \dot{a}_1 \\ \dot{\gamma} \\ \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} c_{11} & 0 & 0 & 0 \\ c_{21} & c_{22} - b_2 c \lambda_1 & c_{23} & -b_2 f \lambda_2 \\ 0 & -\lambda_1 & 0 & 1-\lambda_2 \\ 0 & -g\lambda_1 & -e_2 & -\left(e_1 + b_c \lambda_2\right) \end{bmatrix} \begin{bmatrix} a_1 \\ \gamma \\ x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} kc_1 \\ fb_2 \\ 1 \\ g \end{bmatrix} \gamma_r \tag{40}$$

The lateral motion is now decoupled from the yaw motion with the second order controller. The transfer function for the lateral motion is the same as that with the first order controller, while there is one more parameter, e_2, here that can be used to tune the yaw damping ratio for better performance.

6. Electric Vehicle Tests

The proposed architecture and steering control system were examined in the electric vehicle driving simulator. Figure 3 show the driving simulator, the motion is simulated by a six-axis actuator. Visual in formation is presented to the driver on a spherical screen, and the electric vehicle behavior is calculated by a familiar four-wheel electric vehicle model, with the SBW and ones braking systems (ABS).

Figure 3. Electric Vehicle Driving Simulator

The above control algorithm has been implemented on electric vehicle driving simulator. Since these equations will provide yaw stability control functionality based on a predetermined desired yaw rate, it is necessary to have this data a priori. Also, for a smooth electric vehicle ride and handling it is desired to activate the yaw moment controller based on a threshold value for the yaw rate. We assume that the steering maneuver on a wet surface at 60 kph, the electric vehicle performs a steering maneuver at 60 kph on a slippery wet road. Similar features found in the previous tests can also be observed in this maneuver. Figures 4 to 6 depict the dynamic responses of the uncontrolled and the controlled electric vehicle. Again, the proposed controller shows satisfactory tracking performances for the yaw rate and the desired trajectory, besides the attenuation of the body sideslip angle. Therefore, the electric vehicle handling and stability are improved by the proposed integrated steering control system compared to the corresponding passive electric vehicle, regardless of the road condition.

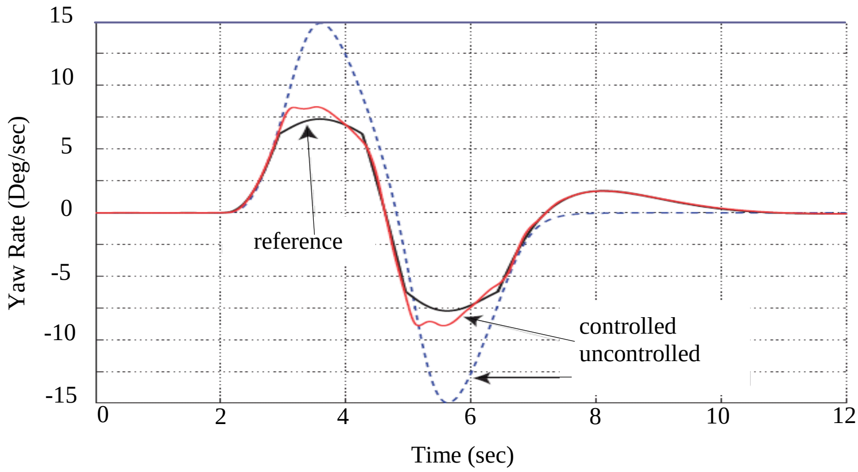

Figure 4. Yaw Rates Comparison of the Controlled and Uncontrolled Electric Vehicle

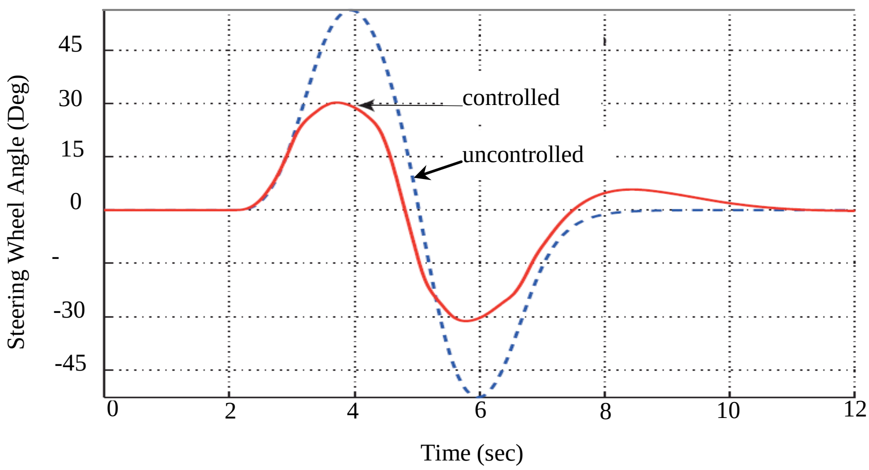

Figure 5. Steering Wheel Angle Comparison of the Controlled and Uncontrolled Electric Vehicle

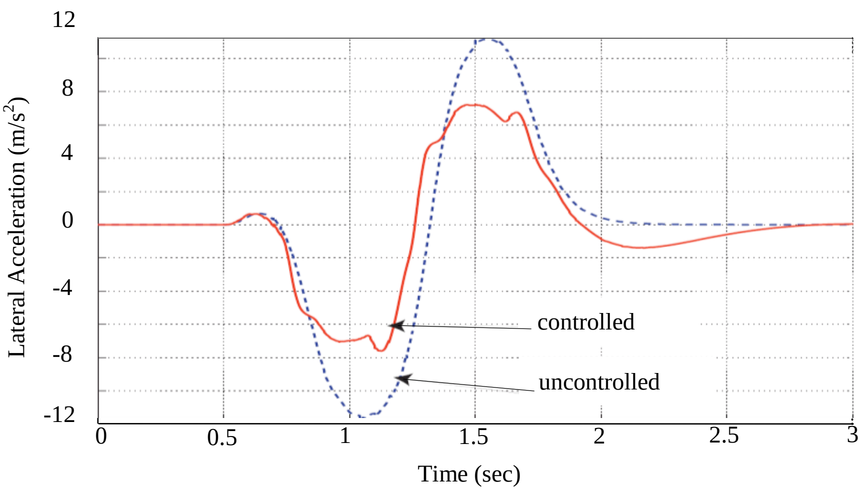

Figure 6. Lateral Accelerations Comparison of the Controlled and Uncontrolled Electric Vehicle

7. Conclusions

A new SBW electric vehicle is presented in view of the steering control system directly affects electric vehicle handling, stability, comfort and active safety. New yaw stability control algorithm via active front wheel steering has been introduced in this paper. The yaw motions of electric vehicle and its yaw damping are achieved by the feedback of both yaw rate and front steering angle with the scheduled gains. The proposed architecture and steering control system have potential to improve electric vehicle safety and reliability inside the SBW system. The time domain simulation tests assess that proposed architecture and steering control system can be effectively stabilized in critical situations condition. This architecture is expected to facilitate the use of the SBW, an indispensable system for electric vehicles of new generation.

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