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Composite Nonlinear Feedback With Disturbance Observer for Active Front Steering

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Abstract

One of the dominant virtue of Steer-By-Wire (SBW) vehicle is its capability to enhance handling performance by installing Active Front Steering (AFS) system without the driver's interferences. Hence, this paper introduced an AFS control strategy using the combination of Composite Nonlinear Feedback (CNF) controller and Disturbance Observer (DOB) to achieve fast yaw rate tracking response which is also robust to the existence of disturbance. The proposed control strategy is simulated in J-curve and Lane change manoevres with the presence of side wind disturbance via Matlab/Simulink sotware. Futhermore, comparison with Proportional Integral Derivative (PID) and Linear Quadratic Regulator (LQR) controllers are also conducted to evaluate the effectiveness of the proposed controller. The results showed that the combined CNF and DOB strategy achieved the fastest yaw rate tracking capability with the least impact of disturbance in the AFS system installed in SBW vehicle.

Keywords: *composite nonlinear feedback, disturbance observer, active front steering, steer-by-wire*

1. Introduction

Steer-By-Wire (SBW) is a steering system which removes mechanical linkages between steering wheel and front wheel systems with electronic components and wires. Due to the removal, an Active Front Steering (AFS) system can be applied independently without the driver's interference because there is no fixed relationship between the steering wheel angle and the front wheels angle [1, 2]. The aim of AFS implementation is to enhance vehicle manouverability and stability [3, 4]. Here, AFS is installed in SBW vehicle for yaw rate tracking control to assist the drivers to keep the vehicle on the desired path.

CNF controller is a combination of linear and nonlinear feedback laws without any switching element. According to the previous research findings, CNF has been proven to have strong capabilites in achieving fast yaw rate tracking performance with minimal overshoot in AFS system [5, 6]. However, Hasan et. al addressed the CNF performance issue when a side wind disturbance is applied to the AFS system [7]. According to the results, the yaw rate tracking response is majorly affected by the disturbance. It means that the CNF cannot stand alone to overcome the existence of disturbance. This issue leads to the decreasing of the vehicle handling performance.

Regarding the issue, an integration of CNF controller with observer such as reduced-order and extended state observers is one of the solution to enhance the robustness of the control system [8, 9, 10]. Thus, this paper proposes the combination of CNF and Disturbance Observer (DOB) in AFS system to improve the transient performance of yaw rate tracking control and at the same time simultaneously reject the element of disturbance. DOB is chosen because it is widely used in SBW system due to its efficiency and design simplicity [11, 12].

This paper is organized as follows. In the next section, the mathematical model of SBW's front wheel system and vehicle model are presented. Then, the following section describes the

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design of the control structure. The fourth section shows the simulation results and discussion. Finally, the conclusion and future works are being concluded and suggested in the last section.

2. System Modeling
2.1. SBW's Front Wheel System

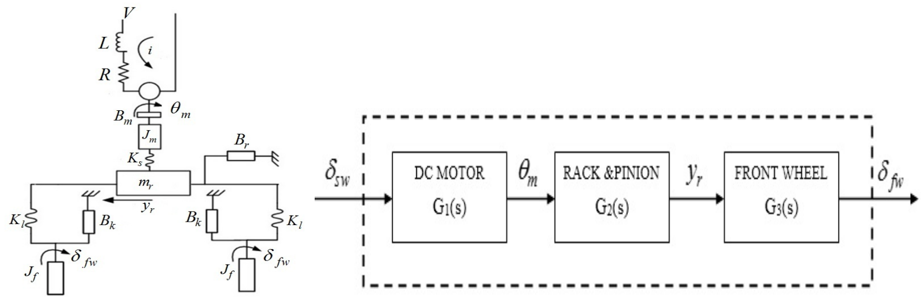


Figure 1. SBW's front wheel system a) Mechanism, b) Block diagram

Table 1. Front wheel system parameter value

Parameter	Definition	Value	Unit	Parameter	Definition	Value	Unit
δ_{sw}	Steering angle	-	rad	B_m	Motor damping coefficient	0.007	Nms/rad
δ_{fw}	Front wheel angle	-	rad	J_f	Front wheel inertia	1.36	kgm^2
θ_m	Motor angle	-	rad	R	Motor electrical resistance	3.1124	Ohm
V	Voltage	-	v	L	Motor electrical inductance	1.6663	$Henry$
i	Motor current	-	A	y_r	Rack displacement	-	m
m_r	Rack Mass	2	kg	B_r	Rack damping coefficient	25	Nms/rad
J_m	Motor inertia	0.0012	kgm^2	B_k	Kingpin damping coefficient	70	Nms/rad
K_s	Lumped torque stiffness	0.257	Nm/rad	r_p	Pinion gear radius	0.1	m
K_b	Motor emf constant	0.2319	Vs/rad	r_l	Offset of king pin axis	0.69	m
K_l	Steering linkage stiffness	26e2	Nm/rad				

Figure 1 shows the mechanism and block diagram of front wheel system while Table 1 lists the parameter's details [13]. Basically, this system consisted of steering, motor, rack, pinion and front wheels, as shown in Figure 1(a). The equations for the system can be expressed as,

Motor displacement and current:

$$\ddot{\theta_m} = -\frac{1}{J_m}(B_m\dot{\theta}_m + K_s i) \quad , \quad \dot{i} = \frac{1}{L}(-Ri + K_b\dot{\theta}_m + V) \tag{1}$$

Rack and pinion :

$$\ddot{y}_r = \frac{1}{m_r}\left(\left(-\frac{2K_l}{{r_l}^2}y_r - \frac{K_s}{{r_p}^2}y_r\right) - B_r\dot{y}_r - \frac{K_s}{r_p}\theta_m\right) \tag{2}$$

Front wheel displacement:

$$\ddot{\delta}_{fw} = \frac{1}{J_f}\left(-K_l\left(\delta_{fw} - \frac{y_r}{r_l}\right) - B_k\dot{\delta}_{fw}\right) \tag{3}$$

Based on Figure 1(b), these equations can be represented in the following transfer function forms,

$$G_1(s) = \frac{\theta_m(s)}{\delta_{sw}(s)} = \frac{(-2.274\times10^{-13})s^2 + (2.365\times10^{-11})s}{s^3+506.2s^2+11360s} + \frac{34390}{s^3+506.2s^2+11360s} \quad (4)$$

$$G_2(s) = \frac{y_r(s)}{\theta_m(s)} = \frac{(-3.553\times10^{-15})s + (1.75\times10^4)}{s^2+12.5s+(1.805\times10^5)} \quad (5)$$

$$G_3(s) = \frac{\delta_{fw}(s)}{y_r(s)} = \frac{(1.421\times10^{-14})s + 1015}{s^2+51.47s+1471} \quad (6)$$

2.2. Vehicle Model

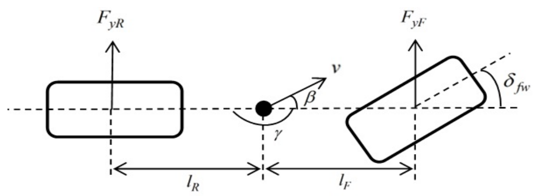


Figure 2. Linear vehicle model

Table 2. Vehicle model parameter

Parameter	Definition	Value	Unit	Parameter	Definition	Value	Unit
l_F	Distance of Front Wheel to COG	1.14	m	F_ω	Crosswind Force	2000	Nm
l_R	Distance of Rear Wheel to COG	1.64	m	F_{yF}	Front wheel lateral force	-	Nm
δ_{fw}	Front wheel angle	-	rad	γ	Yaw rate	-	rad
l_ω	External Force Point	0.5	m	F_{yR}	Rear wheel lateral force	-	Nm
β	Vehicle body slip angle	-	rad	m	Vehicle Mass	1529.98	kg
C_f	Front Wheel Cornering Stiffness	54500	N/rad	I_z	Yaw Inertia	4000	kgm^2
C_R	Rear Wheel Cornering Stiffness	42600	N/rad	v	Velocity	22.22	ms^{-2}

Figure 2 shows the 2-DOF linear vehicle model while Table 2 lists the parameter's details. The vehicle model is used as the vehicle plant to investigate the handling performance. It considers 2-DOF of the vehicle body in lateral and yaw motion. Assuming the velocity to be constant, the linear vehicle model can be presented in the following state space form [14, 15, 16]:

$$\dot{x}_b = A_b x_b + B_b u_b + E_b F_\omega \quad (7)$$

where,

$$A_b = \begin{bmatrix} -\frac{2(C_F+C_R)}{mv} & -1-\frac{2(l_RC_R-l_FC_F)}{mv^2} \\ -\frac{2(l_RC_R-l_FC_F)}{I_z} & -\frac{2(l_F^2C_F+l_R^2C_R)}{I_zv} \end{bmatrix} B_b = \begin{bmatrix} \frac{2C_F}{mv} \\ \frac{2l_FC_F}{I_z} \end{bmatrix}, E_b = \begin{bmatrix} \frac{1}{mv} \\ \frac{l_\omega}{I_z} \end{bmatrix}$$

$$x_b = \begin{bmatrix} \beta \\ \gamma \end{bmatrix}, u_b = \delta_{fw}.$$

3. Controller Design

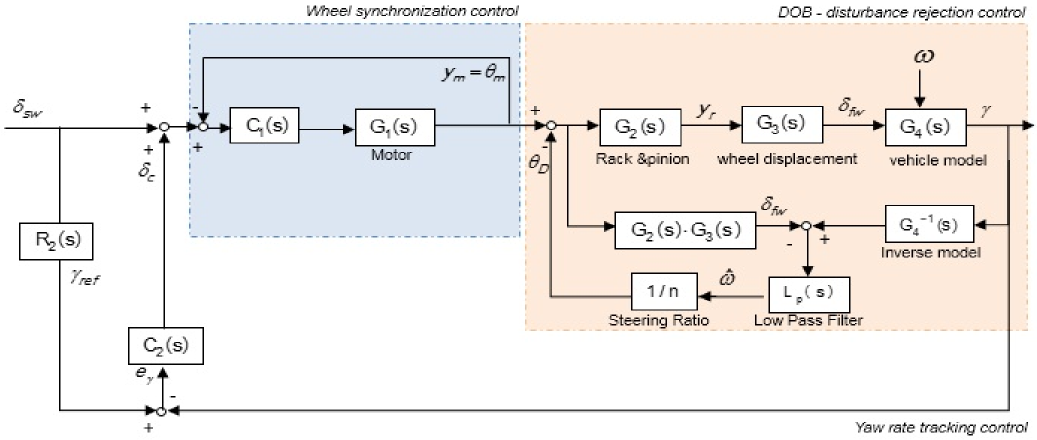


Figure 3. Proposed control structure

Table 3. Parameter of the control structure

Parameter	Definition	Parameter	Definition
δ_c	Corrective steering angle	$G_4(s)$	Transfer function of nominal yaw
γ	Actual yaw rate	$C_2(s)$	Yaw rate tracking controller
γ_{ref}	Desired yaw rate	$C_1(s)$	Wheel synchronization controller
e_γ	Yaw rate error	θ_D	Disturbance compensator control input
$R_2(s)$	Reference model	$G_4^{-1}(s)$	Inverse transfer function of nominal yaw
ω	Side wind disturbance	$\hat{\omega}$	Estimated side wind disturbance

Figure 3 shows the overall view of the proposed control structure, which includes wheel synchronisation and AFS control systems. Meanwhile, Table 3 sets out the parameters used in the proposed control structure. Front wheel system needs a controller to control the motor position in order to steer the front wheels. Therefore, CNF is also implemented as the wheel synchronisation controller to ensure that the front wheel angle is able to synchronise with the steering wheel angle commanded by the driver, as discussed in detail in [17].

3.1. Composite Nonlinear Feedback

According to Figure 3, in yaw rate tracking control strategy, an additional corrected angle δ_c obtained from the CNF controller $C_2(s)$ is added to the steering input δ_{sw} to keep the actual yaw rate response γ closely track the desired yaw rate response γ_{ref} generated by reference model $R_{2(s)}$. The CNF controller is designed based on SBW's front wheel and linear vehicle model systems. Consequently, the equations of the systems is represented in the following state space form where the input $u = \delta_{sw}$ and output $y = \gamma$.

$$\begin{aligned} \dot{x} &= Ax + Bu \\ y &= Cx \end{aligned} \tag{8}$$

where,

$$A = \begin{bmatrix} \frac{-B_m}{J_m} & 0 & \frac{K_s}{J} & 0 & 0 & 0 & 0 & 0 & 0 \\ 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ \frac{-K_b}{L} & 0 & \frac{-R}{L} & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -\frac{K_s}{m_r r_p} & 0 & -\frac{B_r}{m_r} & -\frac{K_l}{m_r r_l^2}+\frac{K_s}{m_r r_p^2} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & -\frac{K_l}{J_f r_l} & -\frac{B_k}{J_f} & -\frac{K_l}{J_f} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & \frac{2C_F}{mv} & -\frac{2(C_F+C_R)}{mv} & -1-\frac{2(l_R C_R - l_F C_F)}{mv^2} \\ 0 & 0 & 0 & 0 & 0 & 0 & \frac{2l_F C_F}{I_z} & -\frac{2(l_R C_R - l_F C_F)}{I_z} & -\frac{2(l_F^2 C_F + l_R^2 C_R)}{I_z v} \end{bmatrix}$$

$$B^T = \begin{bmatrix} 0 & 0 & \frac{1}{L} & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}, x^T = \begin{bmatrix} \dot{\theta}_m & \theta_m & i & \dot{y_r} & y_r & \dot{\delta_{fw}} & \delta_{fw} & \beta & \gamma \end{bmatrix}$$

CNF consists of linear u_L and nonlinear u_N feedback laws. The overall of the laws u_o is expressed as [10],

$$u_o = u_L + u_N \tag{9}$$

In linear feedback part, the damping ratio is set to be small so that the system could achieve the reference input closely and have a fast rising time. While in the nonlinear feedback part, the damping ratio was increased to reduce the overshoot resulted from the linear part. The details of linear feedback law is as follows:

$$u_L = Fx + Gr \tag{10}$$

Here, r is the step input command or reference. F is the controller tuning parameter to generate responses with fast rising time. While G is a scalar which is given as:

$$G = \left[C\left(A + BF\right)^{-1} B\right]^{-1} \tag{11}$$

On the other hand, the nonlinear feedback law can be described as,

$$u_N = \rho\left(r, y\right) B' P \left(x - x_e\right) \tag{12}$$

Here, P is the solution of the following Lypunov equation,

$$\left(A + BF\right)' P + P\left(A + BF\right) = -W \tag{13}$$

where W is a positive definite matrix. In this study, W is simplified as I. The equation for x_e is described as,

$$x_e = -\left(A + BF\right)^{-1} BGr \tag{14}$$

Based on Equation 12 , $\rho\left(r, y\right)$ is a nonpositive function locally Lipschitz in x which can be expressed as,

$$\rho\left(r, y\right) = -\varphi e^{-\alpha\alpha_0|h-r|} \tag{15}$$

where, $\alpha > 0$ and $\varphi > 0$ are the tuning parameters to minimise overshoot.

3.2. Disturbance Observer

Based on Figure 3, the function of DOB is to eliminate the side wind disturbance ω effect without affecting the input δ_{sw} from the steering wheel by providing the estimated disturbance $\hat{\omega}$, which is then used to perform the compensation through negative feedback loop. Firstly, the yaw model derived from linear vehicle model is defined as the nominal model $G_4(s)$. From Equation 7, the transfer function of nominal yaw model $G_4(s)$ is given as follows where the input is front wheel angle, δ_{fw} and the output is yaw rate, γ.

$$G_4(s) = \frac{\gamma(s)}{\delta_{fw}(s)} = \frac{(2C_F l_F m v)s + 4C_F C_R(l_F v + l_R)}{m I_z v s^2 + (2(C_F + C_R)I_z + 2(C_F l_F^2 + C_R l_R^2)v)s + a_0} \tag{16}$$

where,

$$a_0 = (4C_F C_R(l_F + l_R)^2 + 2(C_R l_R - C_F l_F))mv^2$$

In DOB design procedure, δ_{fw} is calculated based on $G_2(s)$ and $G_3(s)$. For an easier procedure's explaination, firstly let $G_2(s)$ and $G_3(s)$ assumed as 1. Based on Figure 3 previously, the linear vehicle model can be expressed as,

$$\gamma = G_4(s)(\delta_{fw} + \omega) \tag{17}$$

where ω is the side wind external disturbance. The estimated disturbance $\hat{\omega}$ is given by,

$$\hat{\omega} = G_4(s))^{-1}(G_4(s)(\delta_{fw} + \omega)) - \delta_{fw} \tag{18}$$

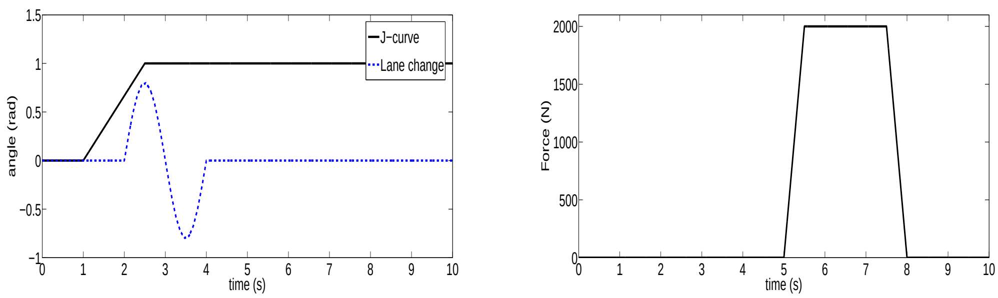
The selection of Low pass filter $L_p(s)$ determined the disturbance rejection performance at low frequencies in the closed-loop system. The $L_p(s)$ order is chosen to be at least equal to the $G_4(s)$ order for causality of $L_p(s)/G_4(s)$ [18]. The equation of $\hat{\omega}$ multiplied by $L_p(s)$ is as follows,

$$\hat{\omega} = L_p(s)[G_4(s))^{-1}(G_4(s)(\delta_{fw} + \omega)) - \delta_{fw}] \tag{19}$$

where $L_p(s)$ is designed as second order transfer function.

4. Simulation and Result

4.1. Simulation Set up



(a) (b)

Figure 4. Manoeuvres a) J-curve and Lane change b) Side wind disturbance

The performance of the proposed control structure is evaluated in simulation using Matlab/Simulink software. As shown in Figure 4(a), the simulation is conducted in two types of manoeuvres which are J-curve and Lane change. Then, side wind disturbance as in Figure 4(b) is purposely added to the manoeuvres to investigate the robustness of the proposed controller. Moreover, comparison between the output responses of CNF controller and DOB (CNF-DOB), Proportional Integral Derivative controller and DOB (PID-DOB), and Linear Quadratic Regulator controller and DOB (LQR-DOB) based systems are carried out to compare and analyzed the robustness and effectiveness of the proposed control structure. During simulation, the vehicle is assumed running in constant speed $80km/h$ at normal road condition. The yaw rate reference model is derived as [15, 16, 19],

$$\gamma_{ref} = \frac{nv(1+T_s s)}{L(1+K_s v^2)(T_{fw}s+1)}\delta_{sw} \tag{20}$$

Here, K_s is the stability factor, T_s is the desired time constant, n is the steering ratio, T_{fw} is the delay time and δ_{sw} is the steering wheel angle.

4.2. Simulation Results and Discussion

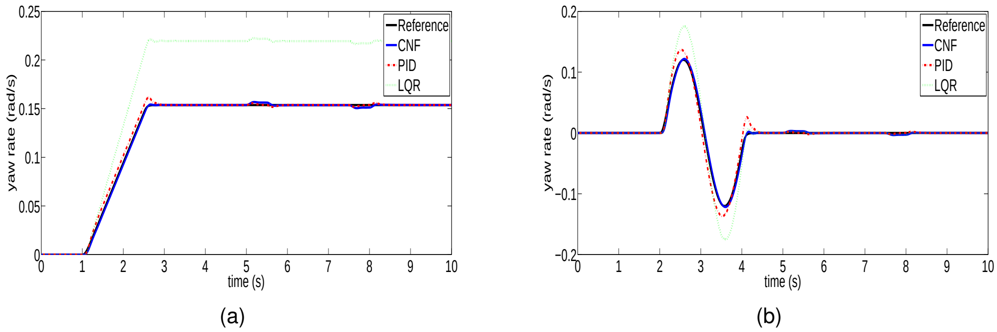


Figure 5. Simulation results a) Yaw rate response during J-curve, b) Yaw rate response during Lane change

Figure 5 depicts the output responses of yaw rate tracking performances during J-curve and Lane change manoeuvres. Based on the figure, the fastest rising and settling time with the smallest margin of steady state error of yaw rate tracking response is accomplished by CNF-DOB control method, followed by PID-DOB and LQR-DOB. Moreover, the influence of the external side wind disturbance in CNF-DOB based system is reduced by 87%. The results show that the combination of CNF and DOB is able to overcome the disturbance issue compared to the findings in [7], where the tracking performance is majorly affected by the existence of disturbance.

5. Conclusion

In this paper, the combination of CNF and DOB is successfully proposed as AFS controller in SBW vehicle. Based on the simulation results, it can be concluded that the CNF-DOB based system is able to enhance the SBW vehicle handling performance by producing the fastest yaw rate tracking responses with least disturbance effects compared to PID-DOB and LQR-DOB. As for the future work, experiment should be conducted to validate the proposed control method in a real time condition.

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