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Controller Design for Electric Power Steering System Using T-S Fuzzy Model Approach

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Abstract: Pressure ripples in electric power steering (EPS) systems can be caused by the phase lag between the driver′s steering torque and steer angle, the nonlinear frictions, and the disturbances from road and sensor noise especially during high-frequency maneuvers. This paper investigates the use of the robust fuzzy control method for actively reducing pressure ripples for EPS systems. Remarkable progress on steering maneuverability is achieved. The EPS dynamics is described with an eight-order nonlinear state-space model and approximated by a Takagi-Sugeno (T-S) fuzzy model with time-varying delays and external disturbances. A stabilization approach is then presented for nonlinear time-delay systems through fuzzy state feedback controller in parallel distributed compensation (PDC) structure. The closed-loop stability conditions of EPS system with the fuzzy controller are parameterized in terms of the linear matrix inequality (LMI) problem. Simulations and experiments using the proposed robust fuzzy controller and traditional PID controller have been carried out for EPS systems. Both the simulation and experiment results show that the proposed fuzzy controller can reduce the torque ripples and allow us to have a good steering feeling and stable driving.

Keywords: Electric power steering (EPS), fuzzy control, Takagi-Sugeno (T-S) model, parallel distributed compensation (PDC), time delay, linear matrix inequality (LMI).

1 Introduction

In recent years, the electric power steering (EPS) system has been widely used as the automobile power-steering equipment because of its efficiency, modularity, and tenability. EPS is a control system that electrically amplifies the driver steering torque inputs to the vehicle to improve steering comfort and performance[1].

A typical control strategy of EPS is shown in the block diagram of Fig. 1[2]. Two primary inputs, the driver torque signal T_{sen} detected by a torque sensor on the steering wheel and the vehicle speed signal V, along with other system variables are continuously fed into an electric control module that determines the reference current I_r based on the torque map. The controller computes the control signal u, which minimizes the error between I_r and actual current I_m detected by the motor current sensor so that the amount and the direction of steering assisting torque T_m is controlled. Normally, the classical proportional integral derivative (PID) controller is employed[1].

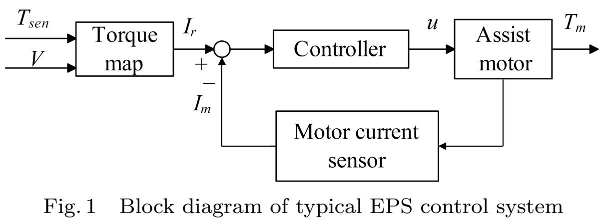

Fig. 1 Block diagram of typical EPS control system

EPS system is a nonlinear multiple-input multiple-output (MIMO) system with multiple objectives, including fast response to the driver′s torque command, good driver feeling, etc. But sometimes annoying pressure ripples in the EPS system will be induced by the phase lag between the driver′s steering torque and steer angle, the nonlinear frictions, and the disturbances from road and sensor noise especially during high frequency and large amplitude maneuvers. To reduce or eliminate torque ripples, previous works normally simplified the EPS model into a linear system, which is subject to significant modeling errors and external disturbances[3−5]. In addition, the typical control method could not eliminate the torque ripples effectively. In this paper, a Takagi-Sugeno (T-S) model for the EPS system with time-varying delays and external disturbances is described. Then, a fuzzy state feedback controller is designed in parallel distributed compensation (PDC) structure. The closed loop stability conditions of EPS system with the robust fuzzy controller are parameterized in terms of linear matrix inequality (LMI) problem.

The remainder of this paper is organized as follows. Section 2 introduces the system model including the steering system and vehicle nonlinear dynamics. Section 3 designs a fuzzy controller based on the T-S fuzzy model. Section 4 analyzes the simulation results. Section 5 validates the feasibility and capability of the controller on the real vehicle by experiments. Conclusions are summarized in Section 6.

2 EPS model description

The mechanical subsystem model is described by the angular rate and position of the steering column and motor, the linear velocity and displacement of the steering rack. Fig. 2 shows the schematic diagram of a steering mechanism equipped with EPS. It can be formally subdivided into three subsystems: 1) the mechanical steering system consisting of the steering wheel, the steering column, the torsion bar, and the steering rack; 2) the brush-type direct current (DC) motor, which provides the assisting torque; 3) the electronic control unit (ECU) with the related sensors, such as steer-

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ing torque, steering angle sensor, and motor current sensor. The principal mode of operation can be summarized as follows. If the driver turns the steering wheel, the torsion bar is twisted and a steering torque is generated, which in turn moves the steering rack. The change of the vehicle direction depends on the change of the steering rack position x_r, which causes the change of the rack force F_r. In order to assist the driver and provide a good steering feeling, a certain amount of the rack force is compensated by the servo force T_m generated by the assistant motor.

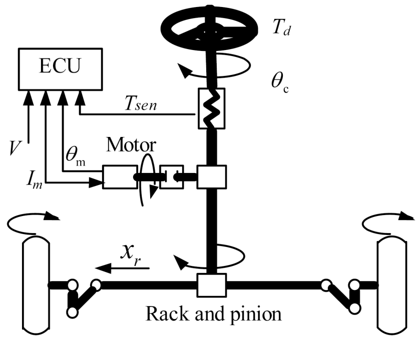

Fig. 2 Dynamics model of EPS system

2.1 Mechanical steering system

The steering wheel is connected to the steering rack via steering torque sensor, steering column, torsion bar, and gearbox. The steering column dynamics are given in [6], and all the symbols in the equations are shown in Table 1.

$$\dot{\theta}_c(t) = \omega_c(t) \tag{1}$$

$$\begin{aligned}\dot{\omega}_c(t) = \frac{1}{J_c}\left(T_d(t) - T_{sen}(t) - B_c\omega_c - f_c(\omega_c(t))\right) = \\ \frac{1}{J_c}\left(T_d(t) - K_c(\theta_c(t) - \frac{x_r(t)}{r_p}) - B_c\omega_c - f_{d,c}(\omega_c(t))\right)\end{aligned} \tag{2}$$

where $\theta_c(t)$ is the steering wheel angular position, $T_d(t)$ denotes the torque on the steering wheel enforced by the driver, $T_{sen}(t)$ is the steering torque measured by the torque sensor, and $f_c(t) = d_{c,c}\text{sign}(\omega_c(t))$ is the coulomb friction. Time delay occurs in the transmission of the torque signal. Thus, $T_{sen}(t)$ is given by

$$T_{sen}(t) = K_c(\theta_c(t-\tau(t)) - \frac{x_r(t-\tau(t))}{r_p}). \tag{3}$$

The mathematical model of the steering rack reads

$$\dot{x}_r(t) = v_r(t) \tag{4}$$

$$\begin{aligned}\dot{v}_r(t) = \frac{1}{M_r}\Bigg(\frac{T_{sen}(t) + G_gT_a(t)}{r_p} - K_tx_r - F_r(t) - B_rv_r - \\ f_r(v_r(t))\Bigg)\end{aligned} \tag{5}$$

where M_r is the mass of the steering rack, $T_a(t)$ is the servo force, $T_{sen}(t)$ is the steering force, and $f_r(v_r(t)) = d_{c,r}\text{sign}(v_r(t))$ is the coulomb friction force. The rack force $F_r(t)$ denotes the sum of all reaction forces of the vehicle, which will be given below.

Table 1 Nomenclature and parameters values

Symbol	Quantity	Value
C_f	Cornering stiffness coefficient	126 000 N/rad
C_r	Cornering stiffness coefficient	126 000 N/rad
G_g	Motor gear ratio	16.5
G_{sc}	Steering system ratio	20
I_z	Moment of vehicle inertia	4240 kg·m^2
J_c	Steering column moment of inertia	11.4 kg·m^2
J_m	Motor moment of inertia	0.0005 kg·m^2
K_a	Motor torque constant	0.05 N·m/A
K_c	Steering column stiffness	114.6 N·m
K_m	Motor torsional stiffness	125 N·m
l_f	Chassis length of front	1.0 m
l_r	Chassis length of rear	1.8 m
M	Vehicle mass	1814 kg
M_r	Rack and wheel assembly mass	32 kg
r_p	Radius of the pinion	0.007 m
T_p	Caster trail	0.033 m

2.2 Assistant motor dynamics

A brush-type DC motor is employed in this paper. The dynamics of the DC motor are described as

$$\dot{\theta}_m(t) = \omega_m(t) \tag{6}$$

$$\begin{aligned}\dot{\omega}_m(t) = \frac{1}{J_m}\left(T_m(t) - T_a(t) - f_{d,m}(\omega_m(t))\right) = \\ \frac{1}{J_m}\Bigg(K_aI_m - K_m\left(\theta_m(t-\tau(t)) - \frac{G_gx_r(t-\tau(t))}{r_p}\right) - \\ B_m\omega_m - f_m(\omega_m(t))\Bigg).\end{aligned} \tag{7}$$

Equation (7) describes the motor column moment of inertia, damping, and torsion coupling with the steering rack. $\theta_m(t)$ is the motor column angular position, $T_m(t)$ is the output torque of the motor, $I_m(t)$ is the armature current determined by the control algorithm, and $T_a(t)$ is the effective assisting torque transmitted to the rack with time-varying delay due to the delay of the transmission of the torque signal T_{sen}. The coulomb friction is $f_m(t) = d_{c,m}\text{sign}(\omega_m(t))$.

2.3 Vehicle dynamics

The rack force $F_r(t)$ mentioned above is the whole reaction force, which can be derived by using the simulation model of a simple single-track model. When a vehicle is running at a constant speed V, the dynamics of lateral vehicle motion are described as follows. The vehicle motion in the horizontal plane is represented herein with states of chassis slip angle $\beta(t)$ at the center of gravity and yaw rate $\gamma(t)$[7].

This model is described by the chassis-based coordinates and the variables are depicted in Fig. 3.

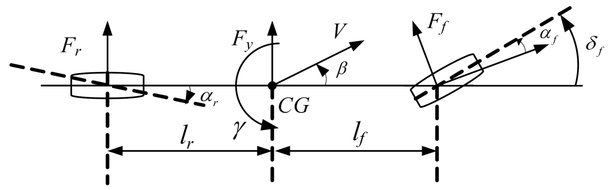

Fig. 3 2D 2-wheel vehicle model (single-track model)

The lateral vehicle motion equations derived from this model are given by

$$\dot{\beta}(t) = \frac{1}{MV(t)}\Big\{ - (C_f + C_r)\,\beta(t) - \Big\{MV(t) - \frac{1}{V(t)}\cdot (l_f C_f - l_r C_r)\Big\}\gamma(t) + C_f \delta_f\Big\} \tag{8}$$

$$\dot{\gamma}(t) = \frac{1}{I_z}\Big\{ - (l_f C_f - l_r C_r)\,\beta(t) - \frac{1}{V(t)}\Big(l_f^2 C_f + l_r^2 C_r\Big)\cdot \gamma(t) + l_f C_f \delta_f\Big\}. \tag{9}$$

Taking small angle approximations, front tire slip angle can be given by

$$\alpha_f(t) = \delta_f(t) - \left(\beta(t) + \frac{l_f}{V(t)}\gamma(t)\right). \tag{10}$$

In the linear region of tire operation, typically, front lateral forces are given by

$$F_y\left(\alpha_f(t)\right) = C_f \alpha_f(t) = C_f\left\{\delta_f(t) - \left(\beta(t) + \frac{l_f}{V(t)}\gamma(t)\right)\right\}. \tag{11}$$

The self-alignment torque $T_{align}(t)$ generated by tires and road surfaces is the dominant torque making the front wheel back to the center while turning. $T_{align}(t)$ is given by

$$T_{align}(t) = T_p \cdot F_y(t). \tag{12}$$

Thus, the reaction force $F_r(t)$ acting on the rack is expressed as

$$F_r(t) = \frac{T_{align}(t)}{r_p} = \frac{T_p \cdot C_f\left\{\delta_f(t) - \left(\beta(t) + \frac{l_f}{V(t)}\gamma(t)\right)\right\}}{r_p} \tag{13}$$

where $\delta_f(t) = \theta_c(t)/G_{sc}$ is the front steer angle, and G_{sc} is the ratio of the steering system.

Therefore, the nonlinear EPS system model can be described as follows:

$$\dot{x}(t) = A_1 x(t) + A_2 x(t-\tau(t)) + B_1 u(t) + B_2 T_d(t)$$
$$y(t) = Cx(t) + \nu(t) \tag{14}$$

with

$$A_1 = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ -\frac{K_c}{J_c} & a_{22} & 0 & 0 & \frac{K_c}{J_c r_p} & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & a_{44} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ a_{61} & 0 & 0 & 0 & -\frac{K_c + K_t r_p^2}{M_r r_p^2} & a_{66} & \frac{T_p C_f}{M_r r_p} & -\frac{T_p C_f l_f}{V M_r r_p} \\ -\frac{C_f}{G_{sC}} & 0 & 0 & 0 & 0 & 0 & -\frac{C_f + C_r}{MV} & a_{78} \\ \frac{C_f l_f}{I_z G_{sC}} & 0 & 0 & 0 & 0 & 0 & a_{87} & a_{88} \end{bmatrix}$$

$$A_2 = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & -\frac{K_m}{J_m} & 0 & -\frac{G_g K_m}{J_m r_p} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & \frac{G_g K_m}{M_r r_p} & 0 & \frac{G_g^2 K_m}{M_r r_p^2} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix},$$

$$B_1 = \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{K_a}{J_m} \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}, \quad B_2 = \begin{bmatrix} 0 \\ \frac{1}{J_c} \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix},$$

$$C = [\ K_c \quad 0 \quad \frac{-K_c}{r_p} \quad 0 \quad 0 \quad 0 \quad 0 \quad 0\ ]$$

where

$$a_{22} = \frac{-B_c - d_{c,c}\mathrm{sign}(\omega_c)}{J_c}, \quad a_{44} = \frac{-B_m - d_{c,m}\mathrm{sign}(\omega_c)}{J_m},$$

$$a_{61} = \frac{K_C}{M_r r_p} - \frac{T_p C_f}{G_{sc} M_r r_p}, \quad a_{66} = \frac{-B_r - d_{c,r}\mathrm{sign}(\omega_c)}{M_r},$$

$$a_{78} = \frac{(C_f l_f - C_r l_r) - MV^2}{MV^2}, \quad a_{87} = \frac{C_r l_r - C_f l_f}{I_Z},$$

$$a_{88} = -\left(\frac{C_f l_f^2 + C_r l_r^2}{I_z V}\right).$$

The state vector is $x = [\theta_c \quad \omega_c \quad \theta_m \quad \omega_m \quad x_r \quad v_r \quad \beta \quad \gamma]^{\mathrm{T}}$ and the control input is $u = I_m$. The output of the model is the steering torque T_{sen}.

3 Fuzzy controller design

3.1 Development of T-S fuzzy models

The movement directions of the steering column, assistant motor and steering rack change in the same way. Thus, $f_c(t)$, $f_m(t)$, $f_r(t)$, and the delay time $\tau(t)$ mainly depend on the rotation velocity of the steering column. Therefore, a T-S fuzzy model can be obtained from (14) with time-varying delays and external disturbances[8−10]. The premise variable of the T-S fuzzy model is $z(t) = \omega_c(t)$, where $\omega_c \in [\omega_{c,\min}, \omega_{c,\max}]$.

The continuous time-delay affine T-S fuzzy system in this paper complies with the following IF-THEN form:

Fuzzy model rule.

If $z(t)$ is $\omega_{c,\min}$, then

$$\dot{x}(t) = A_{11}x(t) + A_{21}x(t-\tau(t)) + B_1 u(t) + B_2 T_d(t) \tag{15}$$

If $z(t)$ is $\omega_{c,\max}$, then

$$\dot{x}(t) = A_{21}x(t) + A_{22}x(t-\tau(t)) + B_1 u(t) + B_2 T_d(t). \tag{16}$$

The time delay in the state, $\tau(t)$, is assumed to be a bounded time-varying delay. It is also assumed that $0 \leqslant \tau(t) \leqslant \tau$, $0 \leqslant \mathrm{d}\tau(t)/\mathrm{d}t \leqslant \varepsilon \leqslant 1$, where τ and ε are the constant upper bounds depending on the considered plant process to be controlled. The matrix $A_{i,j}$ in (15) and (16) vary with the rotation velocity of the steering column.

Given a pair of $(x(t), u(t))$, the final outputs of the fuzzy system are deduced as

$$\dot{x}(t) = \sum_{i=1}^{2} h_i(z(t))\{A_{1i}x(t) + A_{2i}x(t-\tau(t)) + B_1u(t)+ B_2T_d(t)\}. \tag{17}$$

The fuzzy meaning of the two symbols is $h_1(z(t))$ and $h_2(z(t))$ defined as

$$z(t) = h_1(z(t))(\omega_{c,\min}) + h_2(z(t))(\omega_{c,\max}), \tag{18}$$

$$\frac{1}{z(t)} = h_1(z(t))\frac{1}{\omega_{c,\min}} + h_2(z(t))\frac{1}{\omega_{c,\max}},$$

$$h_1(z(t)) + h_2(z(t)) = 1.$$

Using (14), (18), and the recursive least squares method, the membership function $h_i(z(t))$ is written as

$$\begin{bmatrix} h_1(z(t)) \\ h_2(z(t)) \end{bmatrix} = \left[(\zeta^t \times \zeta)^{-1} \times \zeta^t\right] \times \eta,$$

$$\zeta = \begin{bmatrix} \dfrac{1}{\omega_{c,\min}} & \dfrac{1}{\omega_{c,\max}} \\ \omega_{c,\max} & \omega_{c,\min} \\ 1 & 1 \end{bmatrix},$$

$$\eta = \begin{bmatrix} \dfrac{1}{z(t)} & z(t) & 1 \end{bmatrix}^{\mathrm{T}}, \quad h_i(z(t)) \geqslant 0, \quad i = 1, 2.$$

3.2 Design of state feedback fuzzy controller

For a nonlinear plant represented by $z(t)$, a fuzzy controller is designed to share the same fuzzy sets with the plant. It is developed based on the methodology of parallel distributive compensation (PDC). The T-S fuzzy controller can be described as:

Controller Rule.

If $z(t)$ is $\omega_{c,\min}$, then

$$u(t) = -F_1x(t) \tag{19}$$

If $z(t)$ is $\omega_{c,\max}$, then

$$u(t) = -F_2x(t). \tag{20}$$

The output of the fuzzy controller based on PDC is determined by the summation.

$$u(t) = -\sum_{i=1}^{2} h_i(z(t))F_ix(t). \tag{21}$$

Substituting (21) into (17), we have the corresponding closed-loop systems

$$\dot{x}(t) = \sum_{i=1}^{2}\sum_{j=1}^{2} h_i(z(t))h_j(z(t))\{(A_{1i} - B_1F_j)x(t)+ A_{2i}x(t-\tau(t)) + B_2T_d(t)\} = \sum_{i=1}^{2}\sum_{j=1}^{2} h_i(z(t))h_j(z(t)) \{G_{ij}x(t) + A_{2i}x(t-\tau(t)) + B_2T_d(t)\} \tag{22}$$

where $G_{ij} = A_{1i} - B_1F_j$.

3.3 Robust stability conditions and fuzzy regulators

Assume that all membership functions are continuous, piecewise is continuously differentiable, and the deffuzification method is also continuous. First, an unforced system can be derived as

$$\dot{x}(t) = \sum_{i=1}^{2} h_i(z(t))\{A_{1i}x(t) + A_{2i}x(t-\tau(t))\}. \tag{23}$$

The sufficient conditions for ensuring delay-independent stability of a time delay system derived in [11] can be implemented in this paper.

Theorem 1. The equilibrium of the continuous-time fuzzy system with time delay described by (23) is asymptotically stable in general if there exist two common matrixes $P > 0$ and $S > 0$ such that

$$A_{1i}^{\mathrm{T}}P + PA_{1i} + PA_{2i}S^{-1}A_{2i}^{\mathrm{T}}P + \frac{1}{1-\varepsilon}S < 0, \quad i = 1, 2. \tag{24}$$

The design of the state feedback fuzzy controller is to determine the local feedback gains F_i such that the following closed-loop system with time delay is asymptotically stable in general.

Theorem 2. There exists a state feedback fuzzy control law (21) such that the equilibrium of the closed-loop fuzzy system with time delay described in (22) is asymptotically stable in general if there exist matrices $X > 0$ and $Q > 0$ satisfying the following LMIs:

$$\begin{bmatrix} XA_{1i}^{\mathrm{T}} + A_{1i}X - B_1Y_i - Y_i^{\mathrm{T}}B_1^{\mathrm{T}} + \frac{1}{1-\varepsilon}Q & A_{2i}X \\ XA_{2i}^{\mathrm{T}} & -Q \end{bmatrix} < 0 \tag{25}$$

$$\begin{bmatrix} \Delta_{ij} + \frac{2}{1-\varepsilon}Q & (A_{2i} + A_{2j})X \\ X(A_{2i} + A_{2j})^{\mathrm{T}} & -2Q \end{bmatrix} \leqslant 0, \quad i < j \tag{26}$$

where $X = P^{-1}$, $Q = XSX$ for all $i = 1, 2$, $\Delta_{ij} = XA_{1i}^{\mathrm{T}} + A_{1i}X + XA_{1j}^{\mathrm{T}} + A_{1j}X - B_iY_j - Y_j^{\mathrm{T}}B_i^{\mathrm{T}}$ for all i and j except the pairs (i, j) such that $h_i(z(t))h_i(z(t)) = 0$, $\forall t$. Then, the state feedback gain can be constructed as

$$F_i = Y_iX^{-1}, \quad i = 1, 2. \tag{27}$$

4 Simulation

To improve the steering feeling and maneuverability when the vehicle is turning with high frequency maneuvers, a robust fuzzy controller is designed based on the T-S fuzzy model with time delay. The simulation is implemented on the above fuzzy system with time delay using Matlab/Simulink. The stability conditions are derived using the Matlab/LMI toolbox to calculate the matrixes of P, S, and Y_i. The parameters used in the simulation are listed in Table 1. The simulations mimic the following conditions. The driver turns the steering wheel continuously in a sinusoidal pattern, which is similar to the steering pattern in practice, at 0.5 Hz, 2 Hz, and 4 Hz, respectively. The speed of the vehicle is set at 13.9 m/s, which is the normal speed in practice as well. To evaluate the performance of the robust fuzzy controller, the PID control method is adopted for comparison.

Figs. 4-6 show the reaction torque T_{sen} to the driver. As shown in Fig. 4, the simulation results of fuzzy controller and PID controller comply with the inputs well, which means that both the fuzzy controller and PID controller have good performances in low frequency maneuvers. Only few pressure ripples occur when using the PID controller, because in low frequency maneuvers time delay hardly occurs in the EPS system. With the increasing maneuver frequency as shown in Figs. 5 and 6, the pressure ripples become bigger and bigger in amplitude when using PID controller, which indicates that the system is unstable and the driver may feel uncomfortable. Resonance may even occur when the pressure ripple couples with the chassis vibration. However, by using the fuzzy controller, pressure ripple can almost be eliminated and T_{sen} complies well with the driver′s input T_d in amplitude and phase even with the increasing frequency of T_d from 2 Hz to 4 Hz. The simulation results indicate that the fuzzy controller has the capability to eliminate pressure ripples under wide frequency maneuvers.

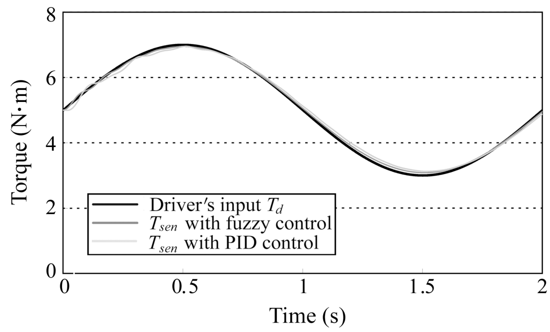

Fig. 4 Simulated response of T_{sen} to driver′s input at 0.5 Hz

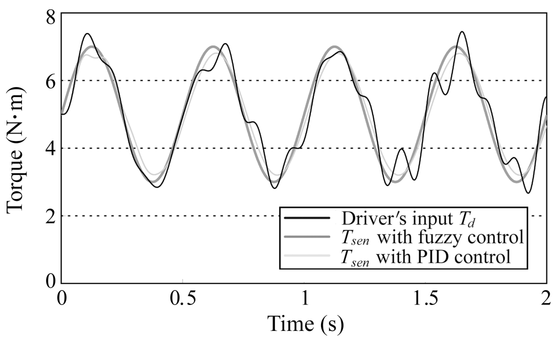

Fig. 5 Simulated response of T_{sen} to driver′s input at 2 Hz

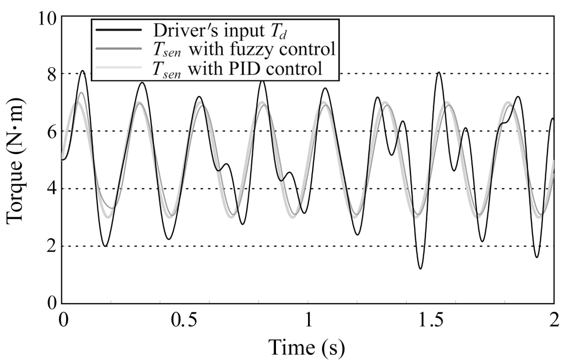

Fig. 6 Simulated response of T_{sen} to driver′s input at 4 Hz

5 Experiment

To further evaluate the performance of the proposed fuzzy controller based on the T-S fuzzy model, various experiments were performed on a real vehicle, as shown in Fig. 7. The vehicle was driven at a velocity of 13.9 m/s, which is the same as that in simulation. Since the driver′s input T_d cannot be measured directly, only the torque sensor signals T_{sen} are compared under fuzzy controller and PID controller.

Fig. 7 Vehicle with EPS system for experiment

The results of the reaction torque measured by torque sensor using the fuzzy controller and PID controller at low frequency (0.5 Hz) and high frequency (4 Hz) are shown in Figs. 8 and 9, respectively. As shown in Fig. 8, no serious torque ripples are generated using fuzzy controller and PID controller under low frequency. Therefore, the driver can get a good steering feeling. However, the pressure ripples became more serious with an increase in maneuver frequency when using the PID controller, which can be felt by the driver, as shown in Fig. 9. On the other hand, the performance of reaction torque is still smooth and most of the torque ripple can be eliminated when using the fuzzy controller indicating that the proposed fuzzy controller can eliminate torque ripples effectively under both low and high frequencies thus verifying the simulation results.

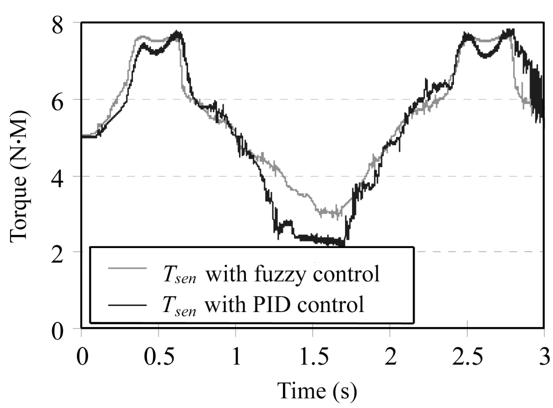

Fig. 8 Testing T_{sen} to driver′s input at low frequency (0.5 Hz)

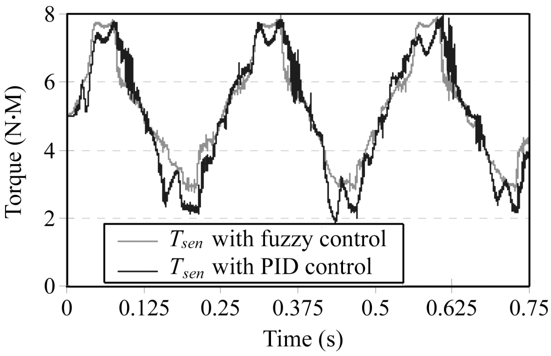

Fig. 9 Testing T_{sen} to driver′s input at high frequency (4 Hz)

6 Conclusions

In this paper, the novelty of the proposed controller is illustrated by applying the robust fuzzy control method based on a T-S fuzzy model to the EPS system to eliminate the pressure ripples induced by the phase lag between driver′s steering torque and steering angle, the nonlinear frictions, and the disturbances from road and sensor noise especially during high frequency maneuvers. Simulations and experiments have been implemented with a robust fuzzy controller and PID controller under conditions of sinusoidal input with low and high frequency. Both the simulation and experimental results showed that the fuzzy controller and PID controller can achieve smooth response of reaction torque at low frequency. At high frequency, the torque ripples become more serious when using PID controller, while they still can be eliminated by using fuzzy controller. It indicates that the robust fuzzy controller can eliminate most torque ripples at both low and high frequencies, and therefore can allow us to achieve a better steering feeling.

References

[1] H. P. Schoener, P. Hille. Automotive Power Electronics – New Challenges for Power Electronics. In *Proceedings of the 31st IEEE Annual Power Electronics Specialists Conference*, IEEE Press, Galway, Ireland, vol. 1, pp. 6–11, 2000.

[2] J. H. Kim, J. B. Song. Control Logic for an Electric Power Steering System Using Assist Motor. *Mechatronics*, vol. 12, no. 3, pp. 447–459, 2002.

[3] R. C. Chabaan, L. Y. Wang. Control of Electrical Power Assist Systems: H_∞ Design, Torque Estimation and Structural Stability. *JSAE Review*, vol. 22, no. 4, pp. 435–444, 2001.

[4] M. Kurishige, T. Kifuku. Static Steering-control System for Electric-power Ssteering, Technical Reports, Mitsubishi Electric Advance, 2001.

[5] M. Kurishige, R, Nishiyama, N. Inoue, S. Wada, T. Kifuku. A New EPS Control Strategy to Improve Steering Wheel Returnability, SAE Technical Paper Series, No. 2000-01-0815, pp. 1–6, 2000.

[6] M. Parmar, J. Y. Hung. A Sensorless Optimal Control System for an Automotive Electric Power Assist Steering System. *IEEE Transactions on Industrial Electronics*, vol. 51, no. 2, pp. 290–298, 2004.

[7] J. P. Switkes, E. J. Rossetter, I. A. Coe, J. C. Gerdes. Handwheel Force Feedback for Lanekeeping Assistance: Combined Dynamics and Stability. *Journal of Dynamic Systems, Measurement, and Control*, vol. 128, no. 3, pp. 532–542, 2006.

[8] L. X. Wang. Design and Analysis of Fuzzy Identifiers of Nonlinear Dynamics Systems. *IEEE Transactions on Automation Control*, vol. 40, no. 1, pp. 11–23, 1995.

[9] W. E. Messoussi, O. Pagès, A. E. Hajjaji. Four-wheel Steering Vehicle Control Using Takagi-Sugeno Fuzzy Models. In *Proceedings of IEEE Fuzzy Systems Conference*, IEEE Press, pp. 1–6, 2007.

[10] A. E. Hajjaji, M. Chadli, M. Oudghiri, O. Pagès. Observer-based Robust Fuzzy Control for Vehicle Lateral Dynamics. In *Proceedings of the American Control Conference*, IEEE Press, Minneapolis, Minnesota, USA, pp. 4664–4669, 2006.

[11] Y. Y. Cao, P. M. Frank. Analysis and Synthesis of Nonlinear Time-delay Systems via Fuzzy Control Approach. *IEEE Transactions on Fuzzy Systems*, vol. 8, no. 2, pp. 200–211, 2000.

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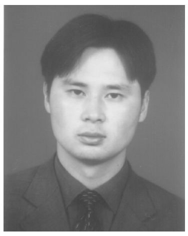

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