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A Driver Model for Dynamic Evaluation of the EPS Assistant Characteristics

Shen Huan, Tan Yunsheng and Mao Jianguo
College of Energy and Power Engineering, Nanjing University of Aeronautics and Astronautics, Nanjing, China

Abstract: Aim to evaluation the dynamic characteristics of Electric Power Steering system (EPS) at low cost design phase, a driver steering control model has been established in this study. The proposed driver model employs both the angle of upcoming road preview deviation and its derivative to calculate the desired yaw rate. Based on lateral dynamics model, the feed-forward control law has been designed, which product an expected steering wheel angle. To establish the dynamic relation between drivers and the steering system, the inertia of the human arm and the torque of the steering wheel are considered, and an interactive feedback controller is obtained. Serpentine line simulation shows that the established control system can effectively complete steering task and provide a closed-loop simulation approach for evaluation the assistant power characteristic of the EPS.

Key words: Driver vehicle system, electric power steering, driver, neuromuscular system, lateral control

INTRODUCTION

Steering system is an important control mechanism that human driver can be used to change or maintain the heading of the vehicle. By manipulating the steering wheel around its longitudinal axis, driving direction of vehicles can be controlled. Generally, steering system has been divided into two categories: The mechanical steering system and the power steering system. The mechanical steering system bases on the physical energy of the driver and its manipulation inconvenient, so, it has been replaced by the power steering system gradually. Electric Power Steering system (EPS) is a class of power steering technology that can provide steering torque by means of assistant motor. To make steering process easier at low speed, and to make driver obtain a clearer road feeling at high speed, EPS provides a feasible assistant torque according to driving speed. The EPS has solved the contradiction between vehicle driving agility and the road feeling effectively, so, it becomes focus of power steering technology (Ken and William, 2000).

Assistant power characteristics and control strategy are critical steps of the EPS design process. Repeated optimization is required and finalize the design must be obtained by combining test data and correcting. This process not only seriously affects the design efficiency, but also hinders the development of system potential. If a measurement of simulation can be provided, the EPS can then be reasonably assessed in low cost designed phase and the design efficiency can be effectively improved as well. In addition, existing EPS evaluation is mainly based on specific evaluation indicators, which can only examine its response characteristics, or through some optimization design algorithms to optimize the conflicting parameters (Parmar and Hung, 2004; Chen *et al*., 2008). In nature, these approaches attribute to open-loop evaluation methods. Driver behavior, however, should also be incorporated into the design process to perform a more comprehensive performance evaluation.

With the development of research on driving behavior modeling, people have proposed models such as the optimal curvature (Macadam, 2003), preview-follower model (Guo *et al*., 2004) and other well-known modeling theories. Some reports have also addressed through parameter identification (Ungoren and Peng, 2004), artificial neural networks (Plochl and Edelmann, 2007) and other intelligent methods to study the driver modeling methods. However, these studies are mainly for understanding and evaluating the characteristics of the vehicle, the driver is generally idealized as an optimal controller to generate steering commands, while the driver steering action are neglected. That is, the steering commands are directly inputted into the vehicle dynamics model. So, it can only be used to evaluate the manipulating stability of the vehicle, but not to the assistant power performance of the EPS.

Corresponding Author: Shen Huan, College of Energy and Power Engineering, Nanjing University of Aeronautics and Astronautics, 210016, Nanjing, China

The purpose of this study is considering the dynamic interaction between the driver and steering wheel, so as to establish a new driver steering control model based on the torque feedback from the steering wheel. This model can expected achieve a good path tracking performance and more important, it can work with EPS integrated steering system when evaluating assistant power characteristics in the fashion of closed-loop simulation is needed.

MATERIALS AND METHODS

Vehicle model: Assuming that tires are working in the linear range, the front-wheel steering angle is regarded as input, lateral speed, lateral displacement, yaw angle and yaw rate are regarded as outputs, the linear model is given by:

$$mv(\dot{\beta}+\omega_r)=c_f\left(\delta-\beta-\frac{\omega_r l_f}{v}\right)+c_r\left(\frac{\omega_r l_r}{v}-\beta\right) \tag{1}$$

$$I_z\dot{\omega}_r=c_f\left(\delta-\beta-\frac{\omega_r l_f}{v}\right)l_f-c_r\left(\frac{\omega_r l_r}{v}-\beta\right)l_r \tag{2}$$

where, m is the vehicle mass and I_z is the yaw moment of inertia around the vertical axis; v is the vehicle velocity; β and ω_r are the sideslip angle and yaw rate of vehicle; δ is the steering angle of front-wheel; c_f and c_r stand for the front and rear axle cornering stiffness; l_f and l_r are the distance from centre of gravity to the front and rear wheels, respectively.

From Eq. 1 and 2, the transfer function from front-wheel angle to yaw rate can be calculated as follows:

$$G_v(s)=\frac{\omega(s)}{\delta(s)}=\frac{l_1 s+l_0}{s^2+k_1 s+k_0} \tag{3}$$

Where:

$$k_0=\frac{l_r c_r-l_f c_f}{I_z}+\frac{c_f c_r L^2}{mv^2 I_z}$$

$$k_1=\frac{l_f^2 c_f+l_r^2 c_r}{vI_z}+\frac{c_f+c_r}{mv}$$

$$l_0=\frac{Lc_f c_r}{mvI_z}$$

$$l_1=\frac{l_f c_f}{I_z}$$

Equation 3 describes the yaw rate response characteristics with the input of the front-wheel steering angle.

EPS model: Typical structure of EPS is shown in Fig. 1, including the steering wheel input, power motor and front-wheel angle output.

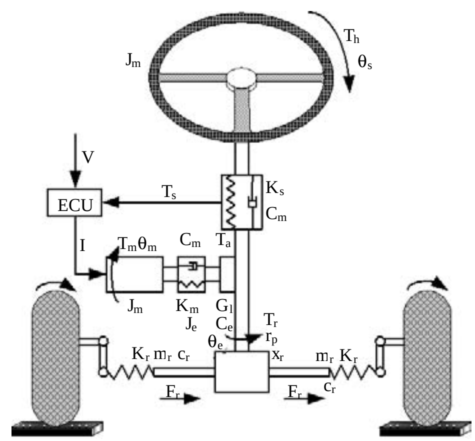

Fig. 1: Typical structure of the electric power steering system

Through the dynamics analysis of the steering wheel and steering column, the equations can be expressed by:

$$\begin{cases} J_s\ddot{\theta}_s+C_s\dot{\theta}_s=T_h-T_s \\ T_s=k_s\left(\theta_s-x_r/r_p\right) \end{cases} \tag{4}$$

where, J_s is the moment of inertia and C_s is the viscous damping coefficient; θ_s is the steering wheel angle; T_h and T_s are the input torque and torque sensor output torque, respectively. The symbol k_s is the stiffness coefficients; x_r and r_p are the displacement of the gear rack and radius of gear, respectively.

The model of the motor can be written as:

$$\begin{cases} J_m\ddot{\theta}_m+C_m\dot{\theta}_m=T_m-T_a \\ T_m=k_t I \\ T_a=k_m\left(\theta_m-\dfrac{G_1 x_r}{r_p}\right) \\ U=L\dot{I}+RI+k_f\dot{\theta}_m \end{cases} \tag{5}$$

where, J_m is the moment of inertia and C_m is the motor viscous damping coefficient; θ_m is the motor rotation; T_m and T_a are the motor electromagnetic torque and output torque, respectively. The symbol k_m denotes the output shaft stiffness coefficient; k_t and k_f are the electromagnetic torque coefficient and back EMF coefficient of the motor, respectively. The symbol U is the motor control voltage; I is the motor current; R is the motor armature resistance; L is the motor inductance (the inductance value is small, generally given ignored); G_1 is the reduction ratio ofretarding mechanism.

Front angle output contains the output shaft and the steering rack two aspects, the first based on the output shaft of the stress analysis, giving rise to the following equation:

$$J_e\ddot{\theta}_e + C_e\dot{\theta}_e = T_s + G_1T_a - T_1 \tag{6}$$

where, J_e denotes the moment of inertia, C_e the damping coefficient, $\theta_e = x_r/r_p$ the steering pinion angle, T_1 the reverse acting output shaft torque.

According to the steering rack model, it can be expressed by:

$$m_r\ddot{x}_r + c_r\dot{x}_r + k_rx_r + T_r = T_1/r_p \tag{7}$$

where, m_r is the gear and rack equivalent mass; c_r is the rack damping coefficient; k_r is the equivalent spring constant; T_r is the resistance of the tire that road offer.

From Eq. 6 and 7, the model equation can be obtained as:

$$M_r\ddot{x}_r + C_r\dot{x}_r = T_s/r_p + G_1T_a/r_p - k_rx_r - T_r/r_p \tag{8}$$

where, $M_r = m_r+J_e/r_p^2$ denotes the equivalent mass of rack and pinion gear mechanism; $C_r = c_r+C_e/r_p^2$ the damping coefficient of rack and pinion gear mechanism.

From Eq. 4-8, EPS model can be expressed as the following state-space equation:

$$\begin{aligned} \dot{x} &= Ax + Bu \\ \dot{y} &= Cx + Du \end{aligned} \tag{9}$$

Where:

$$x = \begin{bmatrix} \theta_s & \dot{\theta}_s & \theta_m & \dot{\theta}_m & x_r & \dot{x}_r \end{bmatrix}^T$$

$$y = \begin{bmatrix} \theta_s & x_r \end{bmatrix}^T \quad u = \begin{bmatrix} T_h & I & T_r \end{bmatrix}^T$$

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\dfrac{k_s}{J_s} & -\dfrac{C_s}{J_s} & 0 & 0 & \dfrac{k_s}{J_sr_p} & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & -\dfrac{k_m}{J_m} & -\dfrac{C_m}{J_m} & \dfrac{G_1k_m}{J_mr_p} & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ \dfrac{k_s}{M_rr_p} & 0 & \dfrac{G_1k_m}{M_rr_p} & 0 & -\dfrac{k_rr_p^2+k_s+G_1^2k_m}{M_rr_p^2} & -\dfrac{C_r}{M_r} \end{bmatrix}$$

$$B = \begin{bmatrix} 0 & \dfrac{1}{J_s} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & \dfrac{k_t}{J_m} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & -\dfrac{1}{M_r} \end{bmatrix}^T \quad C = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \end{bmatrix} \quad D = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}$$

Driver model

Direction preview: Visual information is an important cue when the driver controls the vehicle following a certain road. Experienced drivers are capable of binding the road information and the vehicle state information to make an expected steering angle. Meanwhile, due to the inherent physiological limitations of driver, control instruction always has some delays before actually executed, this preview kind of visual information can effectively compensate the driver's delay. As shown in Fig. 2, if one needs lane change, he will shift his attention to the target track firstly. In this case, the preview direction of the driver gaze and the heading of the vehicle form a deviation angle, referred to herein as preview deviation angle θ. Studies have shown that, the driver can gain the desired vehicle yaw rate according to the linear combination of preview deviation angle and its rate of change (Shen *et al.*, 2012). Accordingly, this study defines yaw rate as:

$$\omega_d = \alpha_1\theta + \alpha_2\dot{\theta} \tag{10}$$

where, ω_d denotes the desired yaw rate, θ the deviation angle for the preview, $\alpha_1(\alpha_2)$ the desired yaw angle adjustment parameters.

Feedforward correction: Figure 3 shows the closed-loop system of driver and vehicle. Driver firstly obtains the desired yaw rate ω_d through the preview and then passes it to the feed-forward control model G_d (s) to obtain the desired front-wheel steering angle δ_d. It is noted that one cannot obtain the actually front-wheel steering angle δ^* unless the driver latency delay has been considered. In this study, the $\theta^{-\tau s}$ item denotes the human nervous system response delay, while $1/(t_ds+1)$ the inertia response lag of the arm (shen *et al.*, 2013).

Besides, experimental studies of McRuer *et al.* (1977) and McRuer and Weir (1969) show that open-loop transfer characteristic of the driver-vehicle closed-loop system has a slope about -20 dB dec^{-1} near the crossover frequencies. To meet this conclusion, the nerve latency

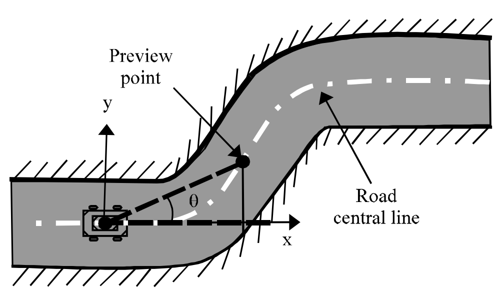

Fig. 2: Diagram of preview deviation angle θ

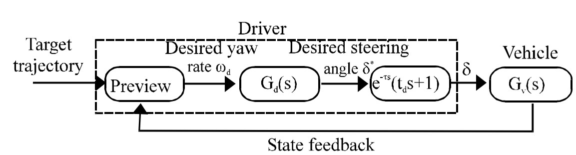

Fig. 3: Structure of driver-vehicle closed loop system

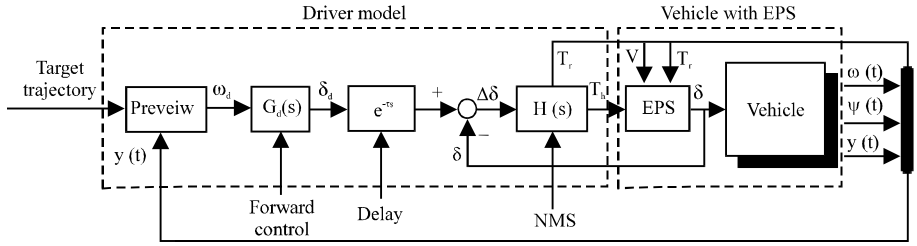

Fig. 4: Scheme of the steering control model

influence should be taken into account, which makes driver-vehicle closed-loop system open loop transfer function holds:

$$G_d(s)\frac{e^{-\tau s}}{t_d s+1}G_v(s)=\frac{\omega_c}{s}e^{-\tau s} \quad (11)$$

Aim to obtain $G_d(s)$, Eq. 11 need to be do some transforms and then developed with Taylor expansion, ignoring terms of fourth and higher order, giving rise to the following equation:

$$G_d(s)=\omega_c\left(f_0+f_1\frac{1}{s}+f_2 s\right) \quad (12)$$

where, ω_c denotes cutoff frequency of the closed loop system and:

$$f_0=\frac{k_1}{l_0}+k_0\left(\frac{t_d}{l_0}-\frac{l_1}{l_0^2}\right)$$

$$f_1=\frac{k_0}{l_0}$$

$$f_2=\frac{1}{l_0}+k_0\left(\frac{l_1^2}{l_0^3}-\frac{l_1 t_d}{l_0^2}\right)+k_1\left(\frac{t_d}{l_0}-\frac{l_1}{l_0^2}\right)$$

Feedback control: Expected steering wheel angle is not directly passed to the front wheels, but applied torque instead to the steering wheel by driver's arm and then it will be force the steering column running gradually up to the expected front-wheel angle. In order to modeling such relation, the transfer delay of the driver neural system is indicated as τ and the inertia delay of the driver arm is defined as t_d. Then, the relationship between the actual angle steering wheel and expected angle of the driver can be described by Eq. 13:

$$\dot{\delta}(t)=-\frac{1}{t_d}\delta(t)+\frac{1}{t_d}\delta_d(t-\tau) \quad (13)$$

where, δ_d and δ are the desired and actual angle of the steering wheel, respectively.

In order to achieve the turning process described in Eq. 13, drivers must applied proper torque continuingly to the steering wheel through the arm muscles, until the completion of steering movements. In the steering process, due to the contact of the tire with the road surface, a torque will feedback to the driver, besides, this feedback relation change with the steering states. Therefore, the actually procedure of driver adjusting the steering wheel involved a dynamic force interaction. For the establishment of such a dynamic interaction, a second order mass-spring-damper system can be used to describe it, that means the movement rule of the steering wheel and torque interaction satisfy the following equation:

$$m_s\left(\ddot{\theta}_s-\ddot{\theta}_{sr}\right)+c_s\left(\dot{\theta}_s-\dot{\theta}_{sr}\right)+k_s\left(\theta_s-\theta_{sr}\right)=T_h-T_r \quad (14)$$

where, m_s denotes the expectations mass, c_s (k_s) the damping (stiffness) coefficients in constrained driver interaction with the steering wheel process, θ_{sr} desired steering wheel angle, θ_s actual steering output angle, T_h the torque drivers applied to the steering wheel, T_r the steering wheel feedback torque.

Take the Laplace transform on the Eq. 13 and 14, obtained after proper deformation that:

$$H(s)=\frac{T_h-T_r}{\Delta\delta}=\frac{m_s s^2+c_s s+k_s}{t_d s+1} \quad (15)$$

Equation 15 indicates the feedback control strategy that used by the driver when steering.

Based on the analysis above, the driver-vehicle closed-loop system can be built, as shown in Fig. 4. Through preview the upcoming road, both the direction deviation and its changing rate can be obtain, which driver can be used to make decide the desired yaw rate ω_d and then, get the desired steering wheel angle δ through feed-forward correction, following convert the angle signal into a steering torque signal T_h through the arm system H (s), together with EPS achieve steering, output front-wheel angle to the vehicle, so steering process can be achieved.

Simulation analyses: In order to assess the EPS assistant power characteristic, choose the serpentine

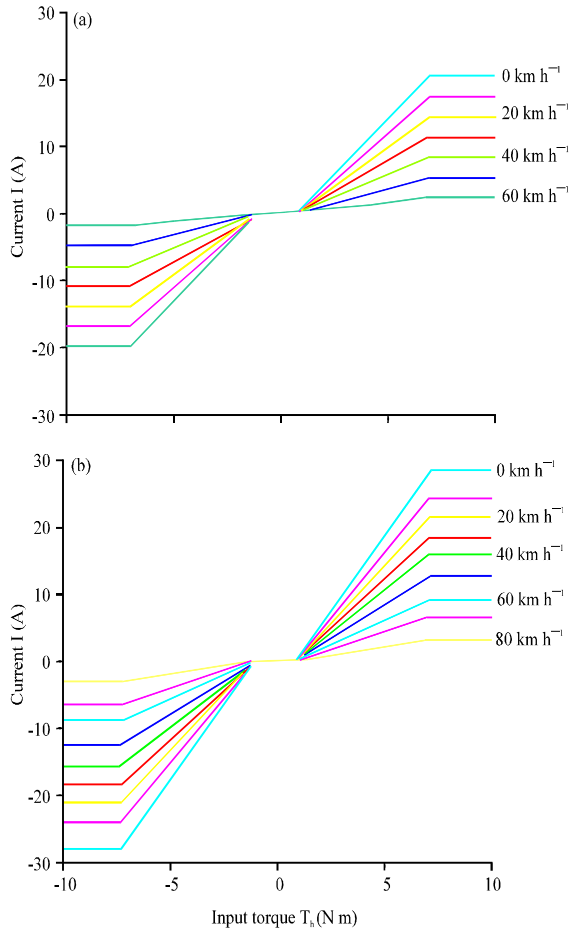

Fig. 5(a-b): Profile of the assistant characteristic curves with the input of current I and torque T_h, (a)Weak assistant power and (b) Strong assistant power

line conditions as driving manipulation condition. Reference GB/T6232.1-94 serpentine curves trial design roads as the input signal. Vehicle simulation parameters are m = 1715 kg, I_z = 2697 kgm^2, L = 2.54 m, l_f = 1.07 m, l_r = 1.47 m, the angular steering transmission ratio G = 20, front and rear equivalent cornering stiffness were c_f = 89733 N rad^{-1}, c_r = 114100 N rad^{-1}. Arm inertial delay is, according to previous studies, t_d = 0.1 and set the speed as v = 50 km h^{-1}.

To analyze the assistant effect, two different types of linear power characteristic curve shown in Fig. 5 are designed (Shi *et al*., 2007). The simulation results show in Fig. 6 and 7.

Figure 6 shows the results in the condition of the serpentine line tracking. It can be seen from the figure that with or without assistant in the case, driver-vehicle closed-loop system model built in this study can complete

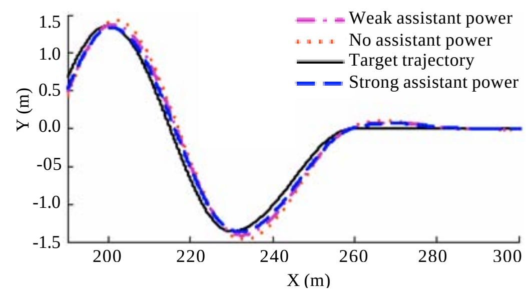

Fig. 6: Comparsion of simulation result of trajectory tracking with different assistant power

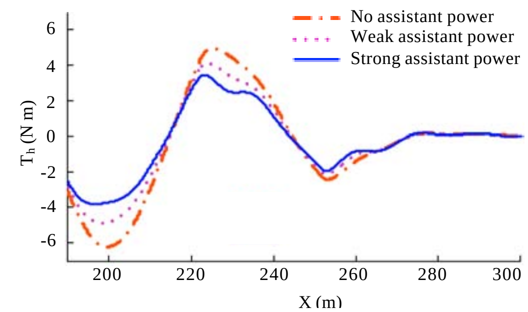

Fig. 7: Comparsion of simulation result of driver's steering control torque with different assistant power

steering tasks quite well, indicating that the driver model has a good path tracking capability. Figure 7 shows the condition when the steering torque is applied to the steering wheel. Intuitively, as the EPS power increases, the torque that the driver's arms need to be applied to the steering wheel decreases. In case of no power assistant, the maximum torque required by driver is 6 N m. In the small power assistant case (Fig. 5a), the maximum torque need to be applied reduces to 4.9 N m; while in the large power assistant case (Fig. 5b) the driver only needs to apply 3.8 N m. It can be concluded that with the assistant of EPS, the driver just need to apply relatively small torque to accomplish the same steering task, indicating that the easiness and convenience performance are improved, which is designed to be coincided with the goal, (Parmar and Hung, 2004; Shi *et al*., 2007). In addition, other observation in Fig. 6 shows that not bigger EPS power is always better. Taking a further look at Fig. 7, when in the absence of steering power assistant, the overshoot occurs and the delay is also present; in the case of a small power assistant, the overshoot is reduced and the delay time is also reduced. This indicates that with the assistant of EPS power, the driver are free from heavy steering movements and can devote more time to focusing on the task of steering control, so that the control performance have been improved. However, in the case of large power assistant, although, a further delay time is reduced, its tracking gain is also reduced and the driver steering torque jitter occurs, which indicates that when EPS power increases, the road feeling of the driver sensed would be reduced. Therefore, bigger EPS power is not always better, one need to balance the performance between steering agility and road feel, so as to optimization can be achieved. Simulation result shows that the model in this study is valuable since there is no attempt has been conducted for evaluate EPS with a driver model into considered.

CONCLUSION

A driver steering control model is established. This model has three components, including direction preview, feed-forward correction and feedback control. Because the driver's visual cues, physiological characteristics and the torque interaction between driver and the steering wheel has been considered comprehensively, so the steering behavior can be described more reasonably. The simulation has been conducted and different EPS assistant power characteristic demonstrated that although the linear assistant power steering system can meet the performance of agility, too much assistant power will cause the loss of information of road feel. To achieve better assistant performance, there still needs further optimization design. For example, exploring non-linear assistant power strategy or developing adaptive control methods. In addition, due to active intervene of the EPS assistant torque; the driver needs a process of adaptation. Therefore, active interventions also need to be carefully processing and combined with driver's steer characteristics to finish integrated design, so as to reflect the modernization design of human-centered.

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