

A New Control Strategy for Electric Power Steering on Low Friction Roads

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ABSTRACT

In vehicles equipped with conventional Electric Power Steering (EPS) systems, the steering effort felt by the driver can be unreasonably low when driving on slippery roads. This may lead inexperienced drivers to steer more than what is required in a turn and risk losing control of the vehicle. Thus, it is sensible for tire-road friction to be accounted for in the design of future EPS systems. This paper describes the design of an auxiliary EPS controller that manipulates torque delivery of current EPS systems by supplying its motor with a compensation current controlled by a fuzzy logic algorithm that considers tire-road friction among other factors. Moreover, a steering system model, a nonlinear vehicle dynamics model and a Dugoff tire model are developed in MATLAB/Simulink. Physical testing is conducted to validate the virtual model and confirm that steering torque decreases considerably on low friction roads. Finally, simulation results show that the new EPS assist control strategy successfully increases steering effort under slippery conditions without compromising ease of steering, while ensuring that road feel also increases with decreasing in road friction and/or increasing vehicle speed as an additional safety measure.

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INTRODUCTION

Recent developments in Electric Power Steering (EPS) have triggered much attention from automakers and the automotive research community. The energy efficiency, structural simplicity, low environmental impact and excellent handling performance of the EPS have made it an appealing alternative to Hydraulic Power Steering (HPS) systems [1, 2, 3, 4].

When modeling conventional EPS systems, tire characteristics are often made linear, such that the self-aligning torque transmitted to the steering column from the front wheels is assumed to be directly proportional to the steering angle of the front wheels [4, 5]. However, this approximation is not suitable under all circumstances, e.g. when information such as variation in road friction needs to be conveyed to the driver. It is well-known that alignment torque increases with increasing sideslip angle when tire characteristics are within the linear region, but decreases sharply beyond this region. On high friction coefficient roads, this linear region is sufficiently large such that the tire slip angle rarely exceeds it, but such occurrence is more likely on low friction coefficient roads, where the linear region is much narrower [6]. When the linear region is exceeded, as often occurs under slippery conditions,

steering effort decreases considerably, resulting in a loss of road feel. This problem can be addressed using a new EPS control strategy.

Much work has already been done on improving the handling performance of EPS systems under slippery conditions. Reference [6] discussed a method for improving the steering wheel returnability in an EPS system, but it focused on the return control of EPS instead of assist control. Reference [7] considered both EPS assist control and return control in detail and formulated a return control algorithm based on a time-varying sliding mode control strategy, but it did not discuss determination of the current compensation coefficient.

In this paper, a novel EPS system control strategy based on motor current compensation is developed along with a fuzzy logic algorithm that evaluates tire-road friction coefficient and vehicle speed to determine the current compensation coefficient. When road friction decreases, the controller increases steering effort to improve driving safety based on a real-time estimation of the tire-road friction coefficient. Moreover, as road friction gets smaller, steering effort is augmented further to feed back changes in road friction to the driver. The definition of steering effort augmentation factor Δ is

explained later. Vehicle speed also influences the amount of compensation. Though it is possible for an experienced driver to sense a drop in steering effort at low vehicle speeds and react accordingly, this is much more difficult at higher speeds where human reaction is easily overwhelmed, thus necessitating additional software compensation. Not only does the proposed control algorithm enhance driving safety, but it improves road feel as well.

SYSTEM MODELING

Steering System Dynamics

An EPS is a multi-body system and modeling all parts of an EPS can be complicated and unnecessary [5]. Thus a simplified EPS model is created and depicted in Figure 1. Numerous scholars have worked on the modeling of EPS systems, so the modeling process itself will not be explained in detail. The equilibrium equations of the steering system can be defined as follows.

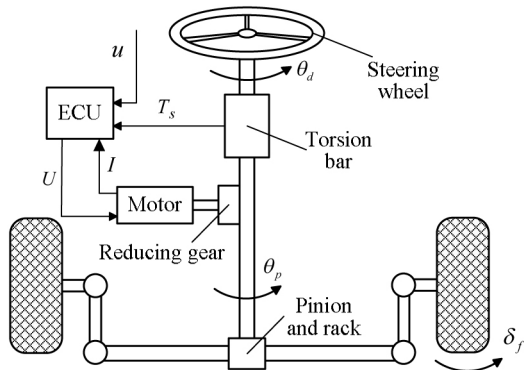


Figure 1. Steering mechanism with EPS

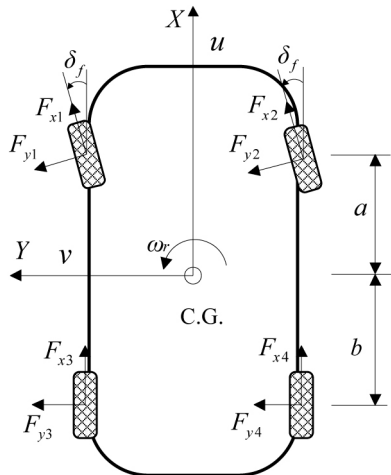


Figure 2. Vehicle model coordinates

$$J_s \ddot{\theta}_d + B_s \dot{\theta}_d + T_s = T_d$$

(1)

$$J_e \ddot{\theta}_p + B_e \dot{\theta}_p + T_r = T_s + T_a$$

(2)

$$\text{Where } T_s = K_s(\theta_d - \theta_p), J_e = J_p + m_r r_p^2, B_e = B_p + B_r r_p^2, \theta_p = G_e \delta_f$$

The actuator utilized is a DC motor, the equilibrium equations of which can be written as follows.

$$U = RI + L\dot{I} + K_b \dot{\theta}_m \quad (3)$$

$$T_a = G_m K_z I \quad (4)$$

$$\theta_m = G_m \theta_p \quad (5)$$

The definition and value of each parameter can be found in the appendix.

Vehicle Dynamics

The vehicle model presented in this paper is a simplified model in which vehicle velocity along X axis is assumed to be constant and cross wind, roll motion and pitch motion are neglected. The vehicle model is depicted in Figure 2 with the chassis-based coordinate system fixed to the center of gravity of the vehicle.

Through force analysis, three equations can be derived as follows.

$$\begin{cases} F_x = m_v(\dot{u} - \omega_r v) \\ F_y = m_v(\dot{v} + \omega_r u) \\ M_z = I_z \dot{\omega}_r \end{cases} \quad (6)$$

Where F_x and F_y are the generalized external forces acting along the X and Y axes, respectively, M_z is the generalized moment around the Z axis, m_v is the vehicle mass, u and v are the vehicle velocity along the X and Y axes, respectively, ω_r is the yaw rate, and I_z is the moment of inertia around the Z axis perpendicular to the page.

It is through the tires that forces generated from roads act on the vehicle, so force analysis of the tires is very important. The following equation can be derived through tires force analysis.

$$\begin{cases} F_x = (F_{x1} + F_{x2})\cos\delta_f - (F_{y1} + F_{y2})\sin\delta_f + F_{x3} + F_{x4} \\ F_y = (F_{x1} + F_{x2})\sin\delta_f + (F_{y1} + F_{y2})\cos\delta_f + F_{y3} + F_{y4} \\ M_z = a(F_{x1} + F_{x2})\sin\delta_f + a(F_{y1} + F_{y2})\cos\delta_f - b(F_{y3} + F_{y4}) \end{cases} \quad (7)$$

Where a is the distance of C.G. from the front axle, b is the distance of C.G. from the rear axle, the subscripts (1, 2, 3, 4) represent the front left, front right, rear left and rear right wheels, respectively.

The rotational speed ω_w for non-driving wheels is calculated in this section for the purpose of estimating road-tire friction coefficient below.

$$\omega_w = u / r \quad (8)$$

Where r denotes the effective radius of the wheel.

Tire Model

For the purposes of this research, a simple linear tire model is insufficient for reflecting the influence of friction coefficient and front axle load. Therefore, we adopted a Dugoff tire model [8] to illustrate the nonlinear characteristic of the tire. The Dugoff tire model allows us to calculate longitudinal and lateral forces of each tire. Its mathematical equations are as follows.

$$\begin{cases} F_{xi} = C_{si} s_i f(\lambda_i) / (1 + s_i) \\ F_{yi} = C_{ai} \tan \alpha_i f(\lambda_i) / (1 + s_i) \end{cases} \quad (9)$$

Where the subscript $i = 1, 2, 3, 4$, C_{si} and C_{ai} represent the longitudinal and lateral tire stiffness, respectively. The variable $f(\lambda_i)$ can be calculated by the following equations.

$$\lambda_i = \frac{\mu F_{zi} (1 + s_i)}{2 \sqrt{(C_{si} s_i)^2 + (C_{ai} \tan \alpha_i)^2}} \quad (10)$$

$$f(\lambda_i) = \begin{cases} (2 - \lambda_i) \lambda_i & \lambda_i < 1 \\ 1 & \lambda_i \geq 1 \end{cases} \quad (11)$$

Where, μ is the tire-road friction coefficient, F_{zi} denotes the normal load of each road wheel calculated using equation (12), S_i is the longitudinal slip rate and α_i is the slip angle calculated using equation (13).

$$\begin{cases} F_{z1} = \frac{mgb}{2(a+b)} - \frac{m_v(\dot{u} - \omega_r v)h}{2(a+b)} - \frac{m_v(\dot{v} + \omega_r u)hb}{T_f(a+b)} \\ F_{z2} = \frac{mgb}{2(a+b)} - \frac{m_v(\dot{u} - \omega_r v)h}{2(a+b)} + \frac{m_v(\dot{v} + \omega_r u)hb}{T_f(a+b)} \\ F_{z3} = \frac{mga}{2(a+b)} + \frac{m_v(\dot{u} - \omega_r v)h}{2(a+b)} - \frac{m_v(\dot{v} + \omega_r u)ha}{T_r(a+b)} \\ F_{z4} = \frac{mga}{2(a+b)} + \frac{m_v(\dot{u} - \omega_r v)h}{2(a+b)} + \frac{m_v(\dot{v} + \omega_r u)ha}{T_r(a+b)} \end{cases} \quad (12)$$

$$\begin{cases} \alpha_1 = \delta_f - \arctan\left(\frac{a\omega_r + v}{u - 0.5T_f\omega_r}\right) \\ \alpha_2 = \delta_f - \arctan\left(\frac{a\omega_r + v}{u + 0.5T_f\omega_r}\right) \\ \alpha_3 = \arctan\left(\frac{b\omega_r - v}{u - 0.5T_r\omega_r}\right) \\ \alpha_4 = \arctan\left(\frac{b\omega_r - v}{u + 0.5T_r\omega_r}\right) \end{cases} \quad (13)$$

Where T_f and T_r are respectively the front track and rear track, h is the height of C.G. from the ground, g is the acceleration of gravity.

Calculation of Steering Resistance Torque

During a steering maneuver, the resistance torque comprises three components, namely self-alignment torque M_{f1} , gravity alignment torque M_{f2} and friction torque M_{f3} . Tire deformation at the front part of the contact patch is greater than at the rear, giving rise to uneven distribution of cornering force which generates self-alignment torque. M_{f1} can be defined as:

$$M_{f1} = (F_{y1} + F_{y2})(t_m + t_p) \quad (14)$$

Where t_m and t_p are the mechanical trail and pneumatic trail, respectively.

The gravity alignment torque, M_{f2} , results from the geometry of the steering system. It can be expressed as follows [9].

$$M_{f2} = (F_{y1} + F_{y2})(c + r \tan \varphi) \cos \varphi \sin \varphi \sin \delta_{fm} \quad (15)$$

Where c is the kingpin offset distance, φ is the kingpin inclination angle, and δ_{fm} is the average steering angle of the two front wheels. In this paper $\delta_{fm} = \delta_f$.

The friction torque, M_{f3} , results from friction within the tire-road interface and the steering system and can be measured by experiment [7]. The total resistance torque at the steering column can be calculated as follows:

$$T_r = (M_{f1} + M_{f2} + M_{f3}) / G_e \quad (16)$$

Note that the equations presented here describe decentering steering torques only, as opposed to recentering torques, which require a different set of equations.