

Instantaneous Estimation of Road Friction based on Front Tire SAT using Kalman Filter

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ABSTRACT

This paper proposes a method of estimating road friction and tire slip angle in a cornering maneuver. The method can estimate front tire road friction accurately at low lateral acceleration because it is based on the tire self-aligning torque (SAT) that exhibits high sensitivity to road friction at low slip angles. Road friction and tire slip angle, which are mutually interdependent, are estimated simultaneously using an extended Kalman filter designed around a model describing the relationship between road friction and SAT and a vehicle lateral dynamic model. The front tire SAT is calculated with a mathematical model that describes the torque transmission characteristics from the electric power steering torque to SAT. Therefore, the proposed method is readily applicable to production vehicles. Results of an experimental study show that the change in road friction is instantaneously estimated at low lateral acceleration.

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1. INTRODUCTION

Owing to the strong demand to reduce the number of traffic fatalities, active safety technology has become one of the most intensively researched areas in the automotive industry. Among the technologies developed to meet this demand, Electric Stability Control (ESC) has been implemented on large number of production vehicles in order to stabilize vehicle dynamic behavior in emergency situations such as spinning and drift out [1]. Information on road friction plays an important role in improving ESC performance. For example, road friction provides us with knowledge of the stability limit of vehicle motion, enabling vehicle motion to be controlled predictively by relatively small and smooth actuation commands before exceeding the limit [2]. This reduces the negative feeling of vehicle motion and steering torque when conventional ESC performs strong braking control after the vehicle state approaches significantly close to the limit of stability.

Previous studies of road friction estimation in cornering maneuvers include a method based on tire lateral force, which is calculated from the measured lateral acceleration and yaw rate [3]. However, this method requires accurate knowledge of the vehicle lateral speed, and it is not realistic to implement it on production vehicles that are not equipped with a sensor to measure the lateral speed. The literature [4] proposes an approach for estimating road friction based on tire lateral force. One of the problems with this method is that

lateral force has low sensitivity to road friction at small slip angles, making it essentially difficult to obtain high estimation accuracy when cornering at low lateral acceleration. Another problem is that rapid change in road friction cannot be instantly detected because lateral force is calculated from the measured lateral acceleration and yaw rate, which are delayed by vehicle dynamics. The estimation delay is a major disadvantage for a vehicle dynamic control system that is tuned in real time according to the changes in road friction.

To solve the problems noted above, this paper proposes an estimation method based on self-aligning torque (SAT). Because SAT has greater sensitivity to road friction than lateral force at low slip angles, the proposed method improves estimation accuracy when cornering at low lateral acceleration. Unlike vehicle state variables such as the yaw rate, SAT is directly linked to road friction, thus the proposed method can achieve instantaneous detection of road friction changes. Since the proposed method combines a vehicle dynamic model and road friction dynamics, it is possible to estimate body slip angle simultaneously based on road friction without requiring a sensor to measure the lateral vehicle speed.

This paper is organized as follows. First, mathematical models for the tire force, tie-rod force, vehicle motion and electric power steering (EPS) torque are described in the second section. Then, the design procedure of the extended

Kalman filter (EKF) and an estimation law to enhance robustness are described, and the overall system is summarized in the third section. Finally, estimation performance is examined based on an experiment in the fourth section, followed by some concluding remarks in the final section.

2. MODELING OF RELATIONSHIP BETWEEN ROAD FRICTION AND EPS TORQUE/VEHICLE STATE VARIABLES

To make an estimation algorithm practical enough to be implementable on production vehicles, we introduce an estimator that refers to EPS torque, which depends highly on SAT, instead of referring to a direct measurement of SAT. In order to estimate road friction from EPS torque, we require a model describing the relationship between EPS torque and road friction at the front tires. This model can be further decomposed into a tire torque/force model, a tie-rod force model, and an EPS output torque model as depicted in [figure 1](#). In addition, we also require a vehicle dynamic model for estimating the body slip angle that contributes to an accurate estimation of road friction. The concrete equations of these models are described in this section.

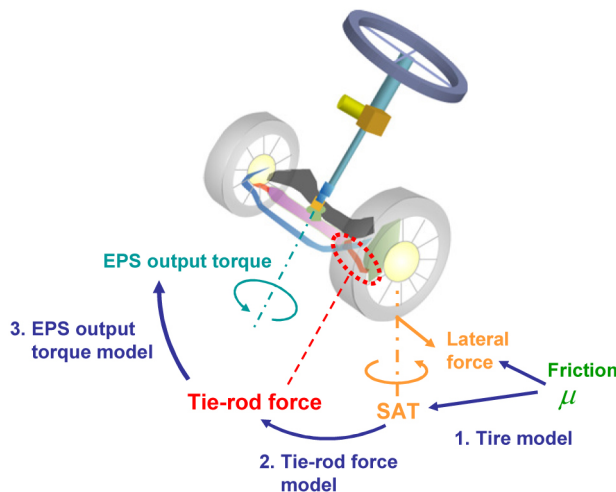


Figure 1. Models describing the relationship between road friction coefficient and EPS output torque

2.1. Tire Force Model

The sensitivity of lateral force to the friction coefficient is modeled by the similarity transformation method [5]. The model of lateral force is expressed as

$$F_y = \frac{\mu}{\mu_0} F_{y0} \left(\frac{\mu_0}{\mu} \alpha_f \right) \quad (1)$$

where F_y denotes the lateral force and μ is the friction coefficient. $F_{y0}(\cdot)$ is a template function of the lateral force

characteristic measured experimentally on a road surface having a friction coefficient of μ_0 . [Figure 2](#) shows how the template lateral force characteristic is transformed according to the change in μ .

For SAT, the similarity transformation method yields

$$T_{sat} = \frac{\mu}{\mu_0} T_{sat0} \left(\frac{\mu_0}{\mu} \alpha_f \right) \quad (2)$$

where T_{sat} denotes SAT, and $T_{sat0}(\cdot)$ is a template function of the SAT characteristic measured experimentally on a road surface whose friction coefficient is μ_0 . [Figure 2](#) shows an example of the lateral force and SAT characteristics. The solid lines correspond to the original template function, and the dashed lines indicate the force characteristics for $\mu = 0.3$ and 0.6 , which are transformed from the template function. A comparison of the lateral force and SAT in terms of sensitivity to μ shows that the latter exhibits greater sensitivity especially at low slip angles in cornering maneuvers at low lateral acceleration. In the example presented in [figure 2](#), while lateral force drops only 30 percent when μ changes from 0.9 to 0.3 , SAT decreases by 80 percent. This suggests that utilizing SAT information extensively improves μ estimation performance in low lateral acceleration cornering maneuvers in comparison with conventional methods that only refer to lateral force.

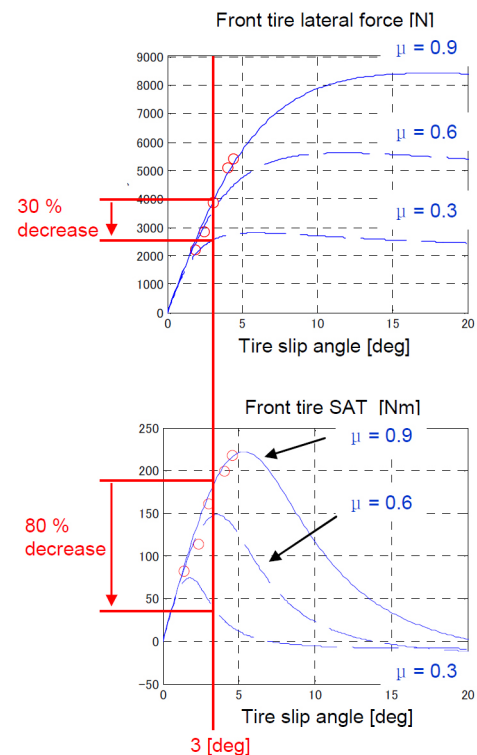


Figure 2. Characteristics of tire lateral force and SAT

2.2. Tie-rod Force Model

In the process where tire force and torque generate EPS torque, tire force and torque are first transformed to axial force in the tie-rod. Thus, a mathematical model describing the relationship between tire force/torque and tie-rod force is explained here.

Assuming that kingpin torque consists of SAT and tire lateral force as shown in [figure 3](#), the equation of the model is written as

$$M_{kp} = T_{sat} + t_{cas} F_{yf} \quad (3)$$

where M_{kp} is the kingpin torque, T_{sat} is SAT, F_{yf} is the front tire lateral force, and t_{cas} is the caster trail. Note those components are per-axle forces that are obtained as the sum of the forces acting on the right and left tires. The contribution of jacking torque can be ignored in the model since the jacking torque caused by the vertical load on the right and left wheels is mutually cancelled. The kingpin torque is transformed to the tie-rod force by considering the length of the knuckle arm that acts as a moment arm to the tie-rod force. The equation of the model is written as

$$F_t = \frac{M_{kp}}{\eta_0 L_k} \quad (4)$$

where F_t is the tie-rod force, L_k is the knuckle arm length, and η_0 is the link efficiency at a zero steering angle.

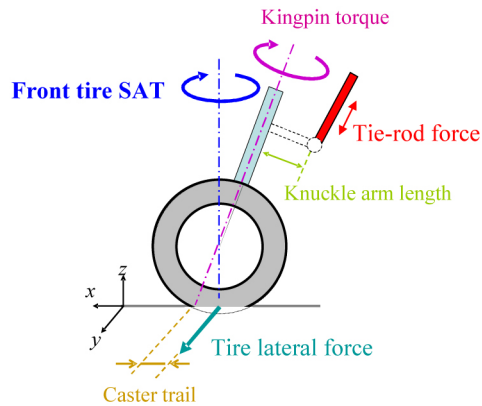


Figure 3. Transformation of tire SAT and lateral force to tie-rod force

2.3. EPS Output Torque Model

This section explains the model describing the relationship between the EPS output torque and tie-rod force. It is difficult to install a sensor on production vehicles for directly measuring the tie-rod force because of manufacturing cost constraints. Using the model described here, the estimation method proposed in the next section can be implemented on a vehicle equipped with EPS, which is a

common feature on many current production vehicles. The EPS output torque can be calculated from the assist motor current and steering torque according to the following relationship.

$$T_{eps} = K_i i_m + T_{str} \quad (5)$$

where T_{eps} is the EPS output torque, i_m is the assist motor current, K_i is the torque coefficient of the assist motor, and T_{str} is the steering torque. The assist motor current can be obtained from the EPS control unit, and the steering torque is the torque applied by the driver and can be measured by the torsion-bar torque sensor that is installed on EPS-equipped vehicles.

The EPS output torque is then translated to the tie-rod force by the following equation.

$$F_t = \frac{N_{str}}{\eta_0 L_k} T_{eps} - F_{f0} \operatorname{sgn}(\dot{\delta}_f) \quad (6)$$

where N_{str} is the steering gear ratio, L_k is the knuckle arm length, η_0 is the link efficiency of the knuckle arm, F_{f0} is the

magnitude of the steering friction, and $\dot{\delta}_f$ is the steering angle rotational speed. The coefficient of the first term converts torque around the steering axis to the tie-rod force. The second term compensates the effect of dry friction. The magnitude of the steering friction F_{f0} can be identified by experimentally measuring the tie-rod force and the EPS output torque while turning the steering wheel. Since the left-hand side of [Equation \(9\)](#) corresponds to the measured tie-rod force and the first term on the right-hand side can be computed from the EPS output torque, the magnitude of F_{f0} is found as the difference between the two terms.

2.4. Vehicle Model

Lateral motion of the vehicle is modeled using a bicycle model with a nonlinear tire lateral force model.

$$\dot{\beta} = \frac{1}{mV} \left\{ F_{yf} (\beta + l_f r / V - \delta_f, \mu) + F_{yr} (\beta - l_r r / V, \mu) \right\} - r \quad (7)$$

$$\dot{r} = \frac{1}{I_z} \left\{ l_f F_{yf} (\beta + l_f r / V - \delta_f, \mu) - l_r F_{yr} (\beta - l_r r / V, \mu) \right\} \quad (8)$$

where β is the body slip angle, r is the yaw rate, V is the vehicle speed, l_f is the distance from the vehicle's center of gravity (CoG) to the front axle, l_r is the distance from the CoG to the rear axle, m is the mass, I_z is the yaw inertia of the