

Estimation of Lateral Force due to Lateral Disturbance for Application to an MDPS-Based Driving Assistant System

2011-01-0977

Published
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doi:[10.4271/2011-01-0977](https://doi.org/10.4271/2011-01-0977)

ABSTRACT

This paper describes a lateral disturbance estimator for an application to a Motor Driven Power Steering (MDPS)-based driving assistant system. A vehicle motion can be disturbed laterally by wind force or load from bank angle acting on the vehicle in the lateral direction. An MDPS-based driving assistant system can be used to reduce steering effort of a human driver in a driving situation with lateral disturbance. In designing the MDPS-based driving assistant system, the lateral wind disturbance should be estimated to determine an assistant torque. An estimator for the vehicle lateral disturbance estimation has been developed. The proposed estimator consists of two parts: a tire self-aligning torque estimator and the lateral disturbance estimator. The lateral disturbance estimator has been designed on the basis of a 2-DOF bicycle model with available sensor signals from the MDPS module. A numerical simulation has been conducted in order to evaluate the proposed estimator. The performance of the proposed estimator considering the vehicle parameter uncertainties has been evaluated for alternative levels of the lateral disturbances.

INTRODUCTION

A lateral disturbance exerting on the vehicle causes unwanted vehicle motion which goes against a driver's intention. The main factors of the lateral disturbance in the actual driving are considered as crosswind and road bank angle. Crosswind is usually blown to the road on the bridge or near the sea shores. The effect of crosswind is negligible with low speed, but with the high vehicle speed or strong crosswind, it can

make the vehicle get out of the lane or overturn. A continuous lateral load from bank angle also makes the unwanted leaning motion in direction of the lateral load. The above factors of the lateral disturbance can reduce the driver's refinement in driving situation, and it can be a threat to the safety of the driver directly above a certain grade [8],[14]. An MDPS system using a motor to assist the human steering torque is expected to be an alternative system to replace the conventional hydraulic steering system. A steering torque, a motor assist torque, steering column velocity, and steering angle can be measured by the MDPS sensors. Since the worldwide mass production of the MDPS is expected, an estimator using the MDPS sensor signal can be easily applicable to automobile equipped with the MDPS. A previous research for estimating the wind disturbance used a vision sensor on high class vehicle which is less common. The lateral displacement and the yaw angle error are chosen as the state vectors and the yaw rate of the vehicle is the additional measures for state estimation method[5].

In this paper, the estimator for the lateral disturbance effect which contains the lateral force and moment has been proposed. The estimator requires the torque signals from an MDPS which uses a motor to assist the steering torque and reduce the steering effort of the driver. It is possible to estimate the lateral disturbance without any additional sensors using the proposed estimator, and for the better performance of the estimation, the appropriate parameters have been chosen through the test data. The vehicle parameter uncertainties have been considered, and those effects have been evaluated to be allowable with the

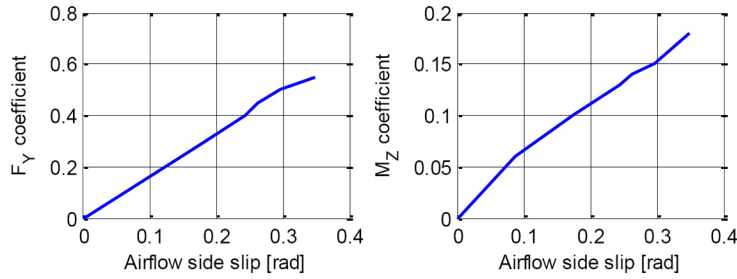


Figure 1. Lateral Force, Yaw Moment Coefficient and Side Shape of Reference Vehicle

appropriate nominal values under the condition of the alternative levels of the lateral disturbances. The characteristics of the MDPS torque sensors have been considered with a sensor delay and resolution, and the effects of the sensor characteristics on the estimator have been applied to the developed simulation model. A reliable mathematical simulation model for applying the lateral disturbance has been developed on the basis of the aerodynamics and the vehicle dynamics. The mathematical vehicle simulation model has been developed using a test data of Hyundai YF Sonata by tuning the parameters of a proposed tire model.

2. DYNAMIC MODEL OF LATERAL LOAD

This section will describe the model representing the lateral load on the vehicle. Two representative lateral load models have been developed in this section: crosswind load and a load generated by the angle of bank. The effect of lateral load is somewhat difficult to confirm in the actual test, so highly reliable mathematical simulation model should be developed.

2-1. CROSSWIND MODEL

Wind forces acting on the vehicle are validated mathematically in previous research through many tests including a wind tunnel test, full size vehicle test and a plat plates test [1],[2]. The forces on the vehicle due to crosswind applied at the center of gravity of the vehicle in longitudinal, lateral and vertical axis. Three difference moments are generated by the forces; roll, pitch and yaw represent each direction. The magnitude of the moment is determined by the magnitude of the force and the distance between the center of gravity and the representative equivalent point of action. Crosswind force and moment are applied on the vehicle differently according to the shapes of the vehicle [7],[11], [12],[13]. General mathematical expression of the force and moment are represented accordingly:

$$F_{CW,i} = \frac{1}{2} \rho C_{f,i} v_r^2 A \quad (1)$$

$$M_{CW,i} = C_{n,i} \frac{\rho}{2} l A v_r^2 \quad (2)$$

The crosswind coefficients on each direction, $C_{f,i}$, $C_{n,i}$ are determined by the reference shape of the vehicle. Since the main interest of this paper is the disturbance in lateral direction, the coefficients according to the side shape of the vehicle need to be described with more precisions[3]. Hyundai YF Sonata is chosen as the reference vehicle.

The airflow side slip is an angle between the velocity of the vehicle and crosswind velocity relative to the vehicle. [Figure 2](#) is a schematic diagram of each velocity and the airflow side slip.

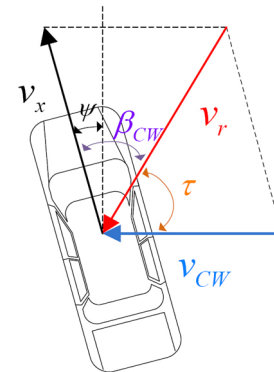


Figure 2. Schematic Diagram of the Airflow Side Slip

The magnitude of the airflow side slip angle is determined by geometrical approach:

$$v_r^2 = v_x^2 + v_{CW}^2 - 2 \cdot v_x \cdot v_{CW} \cdot \cos\left(\frac{\pi}{2} - \psi\right) \quad (3)$$

$$\tau = \sin^{-1} \left(\frac{v_x \sin \left(\frac{\pi}{2} - \psi \right)}{v_r} \right), \quad \beta_{CW} = \frac{\pi}{2} + \psi - \tau \quad (4)$$

The total force acting on the vehicle under the conditions of crosswind is represented as Figure 3. The coefficients are varying with the airflow side slip angle β_{CW} , and β_{CW} is also varying with the vehicle yaw motion by lateral disturbance. So the distance between the center of gravity of the vehicle and the equivalent acting point l_w is varying. The total force and moment under crosswind are represented in Eq.(5),(6).

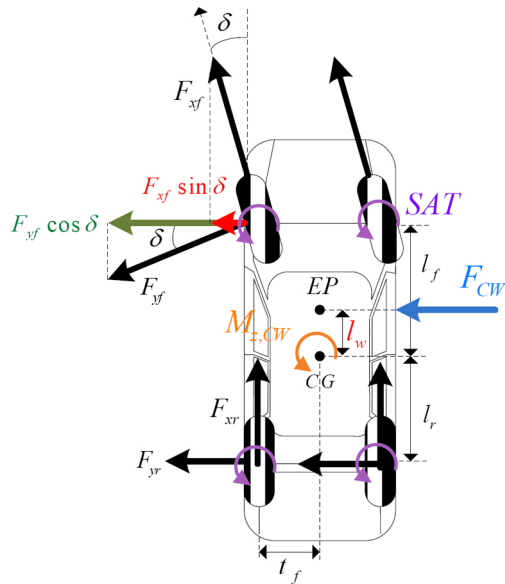


Figure 3. Total Force and Moment Acting on the Vehicle under Crosswind

$$\sum F_y = \sum_{i=1}^2 F_{yfi} \cos \delta + \sum_{i=3}^4 F_{yri} + \sum_{i=1}^2 F_{xfi} \sin \delta + F_{CW} \quad (5)$$

$$\begin{aligned} \sum M_z = & l_f \left(\sum_{i=1}^2 F_{yfi} \cos \delta + \sum_{i=1}^2 F_{xfi} \sin \delta \right) - t_f \sum_{i=3}^4 F_{yri} - \sum_{i=1}^4 SAT_i \\ & - l_r \sum_{i=3}^4 F_{yri} - t_f \left(\sum_{i=1}^2 F_{xfi} \cos \delta - \sum_{i=1}^2 F_{yfi} \sin \delta \right) + M_{z,CW} \end{aligned} \quad (6)$$

2-1. LATERAL LOAD BY BANK ANGLE

A vehicle driving a road with some angle of bank is under lateral load from a component force of gravity. The load with

bank angle is designed to reduce the driver's steering effort in cornering situation, but in the situation of straight driving, it can be considered as the lateral disturbance to the driver. The total force in the situation of bank angle is represented in Figure 4 and Eq.(7),(8). The yaw dynamics of the vehicle are not disturbed by the bank angle [4]. A former research has been done for estimating bank angle using the yaw rate sensor by Chung[9] and You[10]. In this paper, the load generated by bank angle is to be estimated directly using the additional sensor signal from an MDPS in order to reduce calculating load for real time.

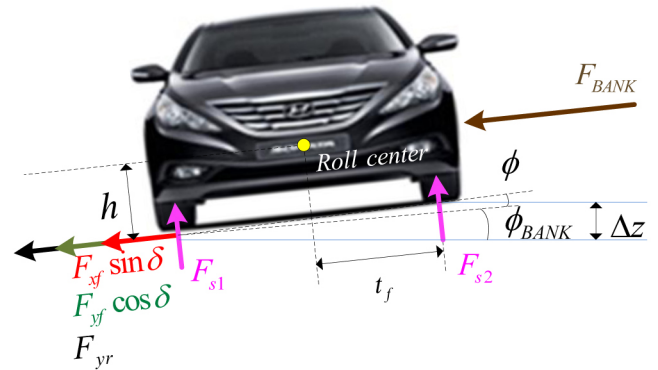


Figure 4. Total Force Acting on the Vehicle under Bank Angle

$$F_{BANK} = mg \frac{\Delta z}{2t_f} \quad (7)$$

$$\sum F_y = \sum_{i=1}^2 F_{yfi} \cos \delta + \sum_{i=3}^4 F_{yri} + \sum_{i=1}^2 F_{xfi} \sin \delta + F_{BANK} \quad (8)$$

3. VEHICLE MODEL

In this section, two different vehicle models have been developed. For applying the lateral load and analyzing the effect of the lateral load, full vehicle model has been developed, and for establishing the lateral load estimator, 2 DOF bicycle model has been used.

3-1. VEHICLE MODEL FOR EVALUATION

In order to apply the lateral load dynamics, a mathematical vehicle simulation model has been developed. The vehicle simulation model is established with the simplified transmission model, suspension model, and the tire model to match with the test data of various drivers' inputs. The Magic tire model is used to represent the vehicle longitudinal, lateral

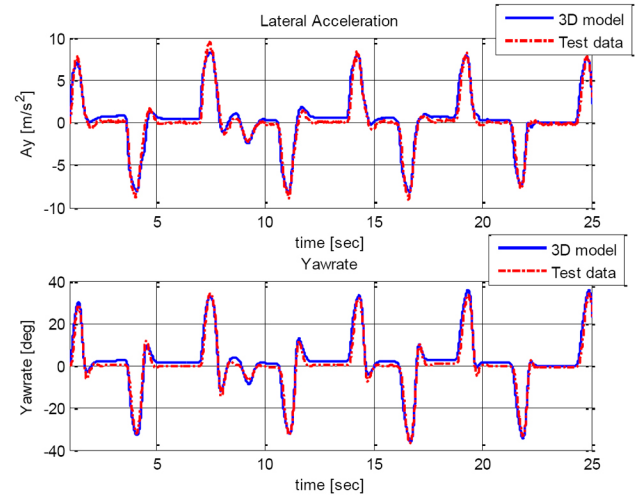
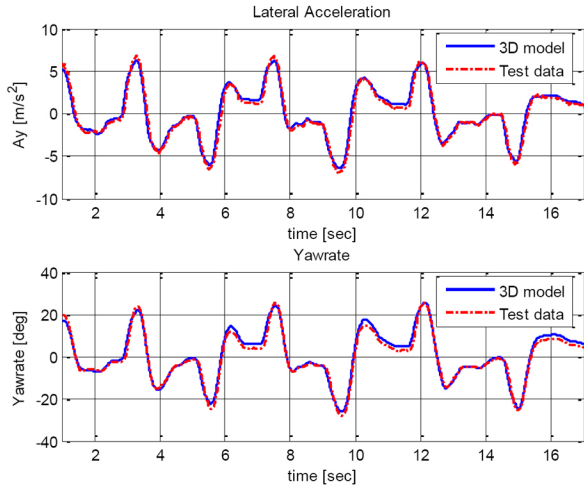


Figure 5. Lateral Behavior Validation with Test Data

and vertical motion. The reference vehicle specification is shown in Table 1.

Table 1. Reference Vehicle Specification

Mass	1466kg (w/o driver)
l	2.795m
l_f	1.071m
l_r	1.724m
Inertia of x-axis	$500 \text{ kg} \cdot \text{m}^2$
Inertia of y-axis	$2392 \text{ kg} \cdot \text{m}^2$
Inertia of z-axis	$2744 \text{ kg} \cdot \text{m}^2$
Steering gear ratio	16
t_f	0.794m

The mathematical vehicle model is validated with the actual test data. The validation results are shown in Figure 5. Each test is conducted in the situation of 60kph with no acceleration, $-150 \sim 150$ deg steering angle (left) and 60kph with no acceleration, $-100 \sim 100$ deg steering angle input (right), respectively.

3-2. 2 DOF BICYCLE MODEL FOR THE ESTIMATOR

2 DOF linear bicycle model is used in the lateral load estimator. The dynamics of the bicycle model under the lateral load is represented in Figure 6. The total force and moment equation of the bicycle model is formed with some assumptions: no slip ratio between the tire and the road, small steering angle, and the same slip angle of the left/right tire. A longitudinal force is not considered in the bicycle model with the first assumption. With this assumptions, the dynamic equation can be summarized as in Eq.(9),(10).

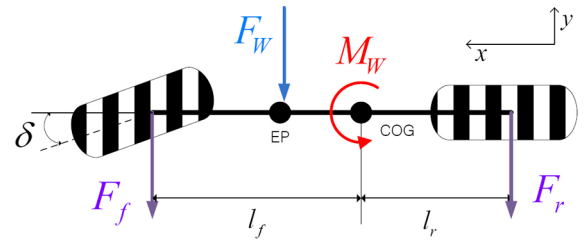


Figure 6. Total Force and Moment of Bicycle Model under Lateral Load

$$\sum F_y = m \cdot v_x (\dot{\beta} + \gamma) = 2F_f + 2F_r + F_w \quad (9)$$

$$\sum M_z = I_z \cdot \dot{\gamma} = 2F_f \cdot l_f - 2F_r \cdot l_r + M_w \quad (10)$$

In order to obtain consistent tendency with full vehicle model under the same driving command, variable parameters should be chosen through the simulation analysis. Tire forces are represented with the slip angles:

$$\begin{cases} \alpha_f = \delta_f - \left(\beta + \frac{l_f \cdot \gamma}{v_x} \right) \\ \alpha_r = -\beta + \frac{l_r \cdot \gamma}{v_x} \end{cases} \quad \begin{cases} F_f = C_f \alpha_f \\ F_r = C_r \alpha_r \end{cases} \quad (11)$$

The cornering stiffnesses C_f , C_r are uncertainties variables which need to be tuned with appropriate values. Two distinguishable levels of external lateral load are applied to the vehicle model with small steering input for straight driving. High level load condition contains 20m/s speed of