Modification of Vehicle Handling Characteristics via Steer-by-Wire

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Abstract

This paper presents a physically intuitive method for altering a vehicle's handling characteristics through active steering intervention. A full state feedback controller augments the driver's steering command via steer-by-wire to achieve desired handling behavior. Accurate estimates of vehicle states are available from a combination of Global Positioning System (GPS) and Inertial Navigation System (INS) sensor measurements. By canceling the effects of steering system dynamics and tire disturbance forces, the steer-by-wire system is able to track commanded steer angle with minimal error. Experimental results verify that with precise steering control and accurate state information, a vehicle's handling characteristics can be modified to match driver preference or to compensate for changes in operating conditions.

1 Introduction

As a step toward fully integrated vehicle dynamic control systems, active steering capability will be available on select production vehicles within one or two years. The potential benefits of active steering intervention, particularly to improve handling behavior during normal driving, have received considerable attention from both the automotive industry and research institutions. As early as 1969, Kasselmann and Keranen [1] developed an active steering system based on feedback from a yaw rate sensor. More recent work by Ackermann [2] combines active steering with yaw rate feedback to robustly decouple yaw and lateral motions. This method is effective in, for example, canceling out yaw generated when braking on a split friction surface. In [3], Huh and Kim devise an active steering controller that eliminates the difference in steering response between driving on slippery roads and dry roads. The controller is based on feedback of lateral tire force estimates derived from vehicle roll motion. Most recently, Segawa et al. [4] apply lateral acceleration and yaw rate feedback to a steer-by-wire vehicle and demonstrate that active steering control can achieve greater driving stability than differential brake control.

Although feedback of sideslip angle for active steering control has been proposed theoretically [5], the difficulty in estimating vehicle sideslip presents an obstacle to accomplishing this in practice. Stability control systems currently available on production cars typically derive slip angle from sensor integration or a physical vehicle model, but these estimation methods are prone to uncertainty [6]. Because sideslip is extremely important to the driver's perception of handling behavior, quality of the driving experience depends strongly on quality of the feedback signal. While this dependence is less critical for stability control systems—which tend to engage when the vehicle is already undergoing extreme maneuvers—to improve handling behavior during normal driving requires cleaner and more accurate feedback.

A new sideslip estimation scheme combining GPS and INS sensor measurements overcomes many of the drawbacks of previous estimation methods [7]. For this paper, a test vehicle converted to steer-by-wire is used to demonstrate that a vehicle's handling characteristics may be find-tuned through a combination of GPS/INS feedback and precisely controlled active steering. The first part of the paper briefly discusses the estimation scheme along with a physically motivated approach for full state feedback control of an actively steered vehicle. The latter part of the paper describes the design of the steer-by-wire system that provides active steering capability to the test vehicle. Experimental results clearly show the change in handling behavior achieved with full state feedback steering control. In addition to matching handling behavior to driver preference, the system successfully counteracts handling differences caused by shifts in weight distribution.

2 Planar Bicycle Model

A vehicle's handling dynamics in the horizontal plane are represented here by the single track, or bicycle model with states of sideslip angle, β, at the center of gravity (CG) and yaw rate, r.

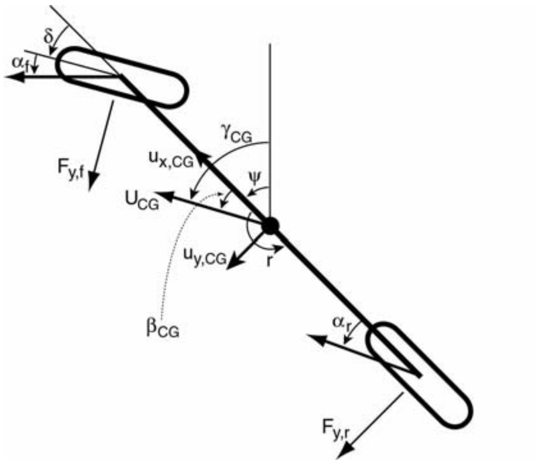

Figure 1: Bicycle model.

The sideslip angle is defined by the difference between vehicle heading, ψ, and the direction of velocity, γ:

$$\beta = \gamma - \psi \tag{1}$$

In Figure 1, δ is the steering angle, u_x and u_y are the longitudinal and lateral components of the CG velocity, F_{yf} and F_{yr} are the lateral tire forces front and rear, respectively, and α_f and α_r are the tire slip angles. Assuming constant longitudinal velocity $u_x=V$, the state equation for the bicycle model can be written as:

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} \frac{-C_f - C_r}{mV} & -1 + \left(\frac{C_r b - C_f a}{mV^2} \right) \\ \frac{C_r b - C_f a}{I} & \frac{-C_f a^2 - C_r b^2}{IV} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} \frac{C_f}{mV} \\ \frac{C_f a}{I} \end{bmatrix} \delta \tag{2}$$

I is the moment of inertia of the vehicle about its yaw axis, m is the vehicle mass, a and b are distance of the front and rear axles from the CG, and C_f and C_r are the total front and rear cornering stiffness that relate lateral tire force to slip angle:

$$\begin{aligned} F_{yf} &= C_f \alpha_f \\ F_{yr} &= C_r \alpha_r \end{aligned} \tag{3}$$

The model is valid for tires operating in the linear region and small slip angles.

3 State Estimation

The ability to obtain accurate information on the vehicle states—yaw rate and sideslip angle—is crucial to implementing an active handling system with full state feedback control. Although yaw rate data is available on many production cars from rate gyroscopes, sideslip cannot be directly measured and must be estimated instead. Two common techniques for estimating this value are to integrate inertial sensors directly and to use a physical vehicle model. Some methods use a combination or switch between these two methods appropriately based on vehicle states [8]. Direct integration methods can accumulate sensor errors and unwanted measurements from road grade and bank angle. In addition, methods based on a physical vehicle model can be sensitive to changes in the vehicle parameters and are only reliable in the linear region.

To overcome these drawbacks, a new method for estimating vehicle sideslip angle using GPS and INS sensor measurements is presented in [7]. In this scheme, GPS measurements from a two-antenna system are combined with INS sensor measurements to eliminate errors due to direct integration. Since both the vehicle heading and the direction of velocity are directly measured from a two-antenna GPS receiver, the sideslip angle can be calculated using Equation (1). INS sensors are integrated with GPS measurements to provide higher update rate estimates of the vehicle states and to handle periods of GPS signal loss. This method is also independent of any parameter uncertainties and changes because it is based on purely kinematic relationships.

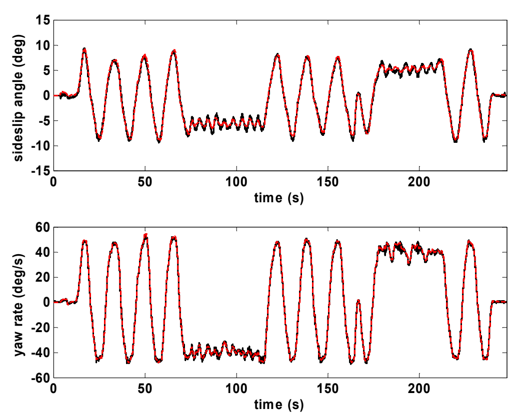

Figure 2: Sideslip and yaw rate estimation.

Experimental results from the GPS/INS integration are plotted in Figure 2 on top of simulation results from the bicycle model for both yaw rate and sideslip angle. The similarity between estimated and simulated yaw rates indicates that the bicycle model used in the comparison is valid and calibrated correctly. The fact that the sideslip measurement is clean and correlates with the model makes it suitable for use as a feedback signal.

4 Full State Feedback Controller

A full state feedback control law for an active steering vehicle is given by

$$\delta = K_r r + K_\beta \beta + K_d \delta_d \tag{4}$$

where δ_d is the driver commanded steer angle and δ is the augmented angle. A physically intuitive way to modify a vehicle's handling characteristics is to define a target front cornering stiffness as

$$\hat{C}_f = C_f (1 + \eta) \tag{5}$$

and the state feedback gains as

$$K_\beta = -\eta \quad K_r = -\frac{a}{V}\eta \quad K_d = (1 + \eta) \tag{6}$$

where η is the desired fractional change in the original front cornering stiffness C_f. Substituting the feedback law (4) into Equation (2) yields a state space equation of the same form as Equation (2) but with the new cornering stiffness $\hat{C}_f$:

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} \frac{-\hat{C}_f - C_r}{mV} & -1 + \left(\frac{C_r b - \hat{C}_f a}{mV^2}\right) \\ \frac{C_r b - \hat{C}_f a}{I} & \frac{-\hat{C}_f a^2 - C_r b^2}{IV} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} \frac{\hat{C}_f}{mV} \\ \frac{\hat{C}_f a}{I} \end{bmatrix} \delta_d \tag{7}$$

Since a vehicle's handling characteristics are heavily influenced by tire cornering stiffness, the effect of this modification is to make the vehicle either more oversteering or understeering depending on the sign of η. Clearly, there are many other ways to apply full state feedback, but the physical motivation behind cornering stiffness adjustment makes clear through the bicycle model exactly how the handling characteristics have been modified. Note that in this formulation, it is not necessary to know the real cornering stiffness of the front tire—only vehicle speed and weight distribution, which are relatively easy to measure—to achieve the desired handling modification.

5 Steer-by-Wire System

A production model 1997 Chevrolet Corvette is modified for full steer-by-wire capability by replacing the steering shaft with a brushless DC servomotor actuator. The stock hydraulic power assist unit and rack and pinion mechanism in the test vehicle are retained as part of the steer-by-wire system, since the incorporation of the power assist unit eliminates the need for extensive modifications to the existing steering system and allows the use of a much smaller actuator. A rotary position sensor measures the lower steering shaft angle, which is equal to the front wheel steer angle scaled by the steering ratio. An identical sensor attached to the upper steering shaft measures the handwheel angle.

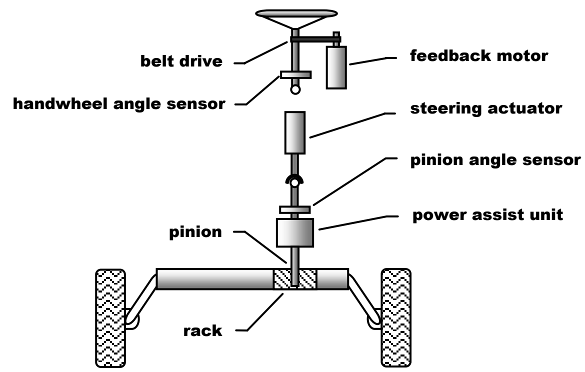

Figure 3: Steer-by-wire schematic.

The servomotor actuator specifications are chosen based on the maximum torque and speed necessary to steer the vehicle under typical driving conditions including moderate emergency maneuvers. On average, steering torque required at the handwheel during normal driving ranges from 0 to 2 Nm, while emergency maneuvers can demand up to 15 Nm of torque [9]. The actuator installed in the test vehicle provides a maximum steering torque of 17.1 Nm with a maximum steer rate of 700 degrees per second.

The differential equation describing the steering system dynamics is as follows:

$$J\ddot{\theta} + b\dot{\theta} + F_c \operatorname{sgn}\dot{\theta} + k_a \tau_a = \tau \tag{8}$$

θ is the pinion angle, J is the total moment of inertia of the system, b is viscous damping, F_c represents coulomb friction, k_a is a scale factor, τ_a is the tire self-aligning moment, and τ is the actuator torque.

The purpose of the steer-by-wire controller is to track commanded steer angle with minimal error; the control effort consists of three components:

$$\tau = \tau_{feedback} + \tau_{feedforward} + \tau_{aligning} \tag{9}$$

The proportional derivative (PD) feedback component is given by

$$\tau_{feedback} = K_p(\theta_d - \theta) + K_d(\dot{\theta}_d - \dot{\theta}) \tag{10}$$

where θ_d is the desired steer angle, K_p is the proportional feedback constant, and K_d is the derivative feedback constant. The feedback gains K_p and K_d are selected to give a fast closed loop system response without oscillatory behavior. Because the system is second order, however, PD control alone results in some steady state error when tracking the type of command shown in Figure 4 (steering angle is given at the front wheels). To obtain these measurements, the front wheels are raised off the ground so as to isolate the influence of J, b and F_c from static friction at the tire-ground interface. The addition of feedforward compensation,

$$\tau_{feedforward} = J\ddot{\theta}_d + b\dot{\theta}_d + F_c \operatorname{sgn}(\dot{\theta}_d) \tag{11}$$

to the PD controller cancels any tracking errors associated with the system dynamics and internal friction (Figure 5). J, b and F_c are determined through closed-loop identification of the steering system.

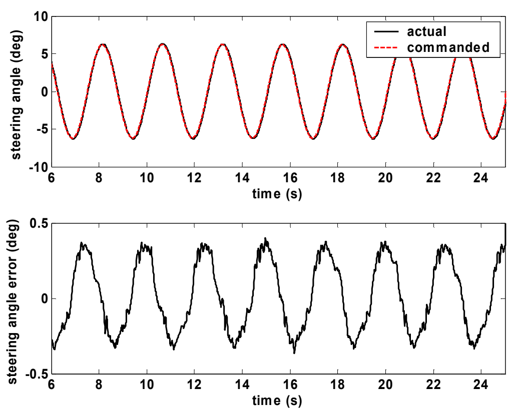

Figure 4: Feedback control only.

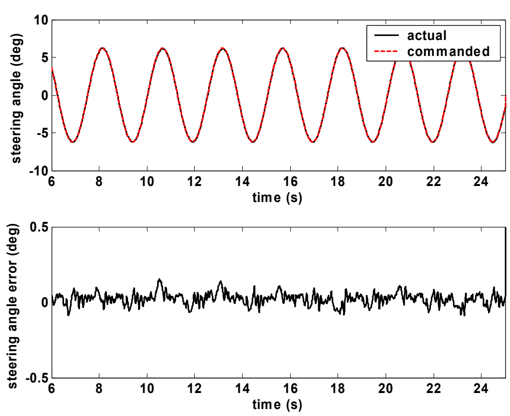

Figure 5: Feedback with feedfoward compensation.

When driving a vehicle over the road, however, an additional disturbance acts on the system causing a steering error (Figure 6) that is directly attributable to tire self-aligning moment. The total aligning moment is given by

$$\tau_a = (t_p + t_m) F_{yf}(\alpha_f) \tag{12}$$

where t_p and t_m are the tire pneumatic and mechanical trails, respectively. Front tire slip angle, α_f, can be calculated from the following relationship involving estimated sideslip and other measurable parameters:

$$\alpha_f = \beta + \frac{ar}{u_x} - \delta \tag{13}$$

Aligning moment may also be directly approximated as an empirical function of tire slip angle [10]. This approximation of aligning moment is added to the feedback and feedforward control as

$$\tau_{aligning} = k_a \hat{\tau}_a(\alpha_f) \tag{14}$$

where k_a is a scale factor to account for torque reduction by the steering gear.

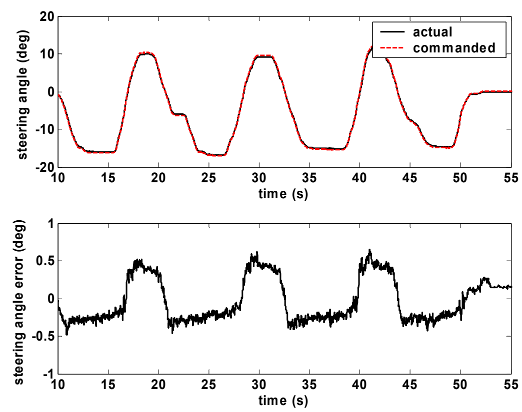

Figure 6: Error due to aligning moment.

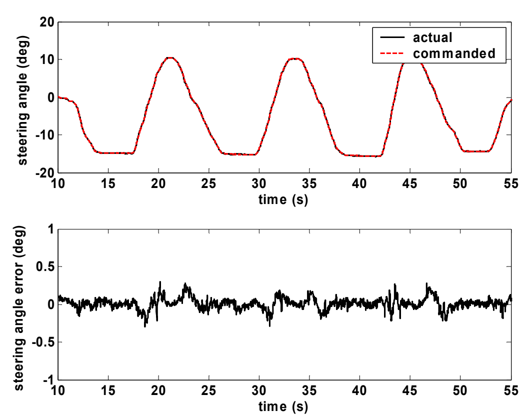

Figure 7: Steering controller with aligning moment compensation.

From a comparison between Figures 6 and 7, the addition of $\tau_{aligning}$ to the actuator effort effectively eliminates most of the steering disturbances that arise when turning at speed.

6 Experimental Results

Figure 8: Steer-by-wire test vehicle.

The steer-by-wire test vehicle is equipped with multiple-antenna GPS configured to provide absolute velocity and heading information. INS sensors measure lateral and longitudinal acceleration, yaw rate, and roll rate. The experimental setup for vehicle state estimation is same as described in [7]. In Figure 9, the measured yaw rate from a sinusoidal steering input while driving at 13.4 m/s (30 mi/hr) compare well to simulation results from the bicycle model.

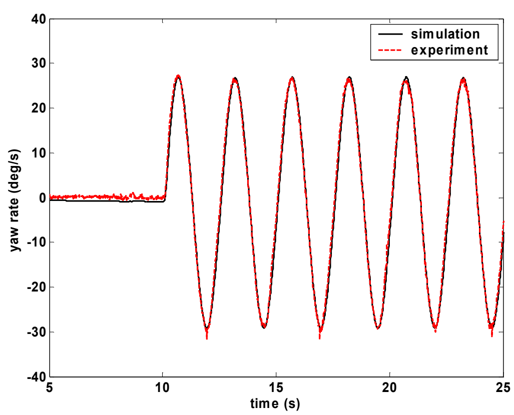

Figure 9: Comparison between bicycle model and experiment with normal cornering stiffness.

Next, handling modification is implemented on the test vehicle. Changes in handling behavior under full state feedback control are evaluated by comparing measured vehicle response to the nominal case shown in Figure 9. In Figure 10, the effective front cornering stiffness is reduced 50% by setting the parameter η to -0.5. The experimental results exhibit lower peak yaw rate and sideslip values than the nominal case. This behavior is expected since reducing the front cornering stiffness causes the vehicle to tend toward understeer. Figure 11 confirms that test results for the reduced case match bicycle model simulation.

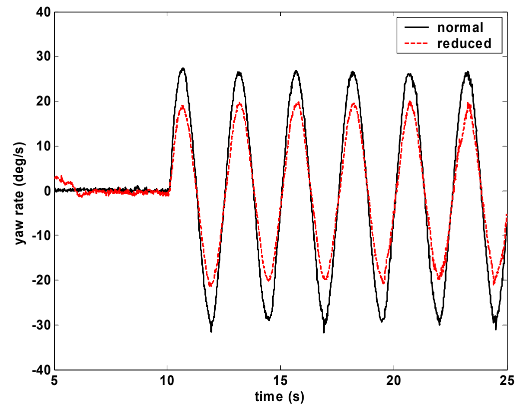

Figure 10: Comparison between normal and effectively reduced front cornering stiffness.

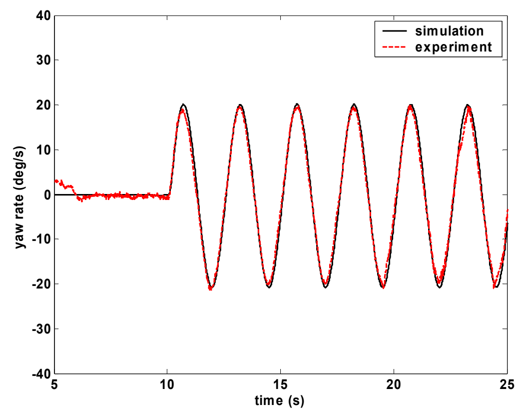

Figure 11: Comparison between bicycle model and experiment with reduced cornering stiffness.

Experimental data show a corresponding but opposite change in handling behavior when the effective front cornering stiffness is increased such that the vehicle tends toward oversteer.

For the final series of tests, 182 kg (400 lbs) of weight are added to the rear of the vehicle so that 57% of the total vehicle weight lies over the rear axle with 43% over the front axle. The unloaded vehicle has a weight distribution balanced equally front to rear. As seen in Figure 12, the loaded vehicle exhibits slightly more oversteering behavior than the unloaded vehicle. However, with active handling modification, a 20% reduction in front cornering stiffness returns the controlled vehicle to the near neutral handling behavior of the unloaded vehicle (Figure 13). While the difference in handling behavior may seem small when viewed on a graph, the improvement is readily apparent to both driver and passenger.

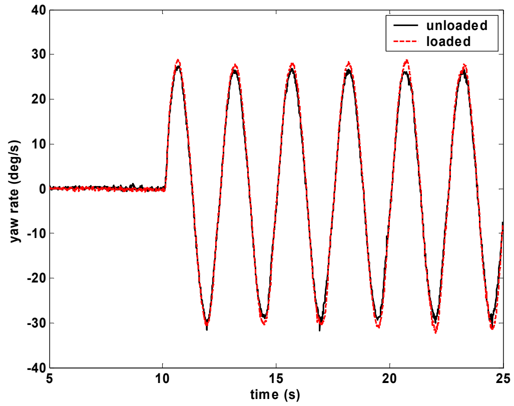

Figure 12: Comparison between unloaded and loaded vehicle.

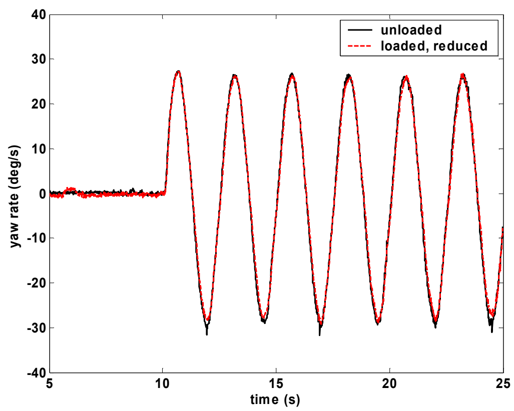

Figure 13: Comparison between unloaded vehicle and loaded vehicle with handling modification.

7 Conclusion

This work represents one of the first applications of GPS-based state estimation to dynamic control of a vehicle with active steering. A full state feedback controller has been developed to alter a vehicle's handling characteristics by augmenting the driver's steering input. The controller is experimentally validated on a steer-by-wire vehicle equipped with GPS and INS sensors. Experimental results confirm that it is possible to effectively change the cornering stiffness of the front tires by full state feedback modification of the driver's steering command. Thus, a vehicle's handling characteristics may be tuned to driver preference or adjusted for variations in operating conditions such as load distribution. Future work will investigate the possible extent of vehicle handling modification by active steering and any fundamental limitations imposed by the feedback or control structure.

Acknowledgements

The authors wish to acknowledge Michael Grimaldi, Robert Wiltse and Pamela Kneeland at General Motors Corporation for their donation of the test vehicle and the GM Foundation for the grant enabling its conversion to steer-by-wire. Thanks also to Dr. Skip Fletcher, T.J. Forsyth, Geary Tiffany and Dave Brown at the NASA Ames Research Center for providing the use of Moffett Federal Airfield for vehicle test purposes.

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