Road Friction Estimation from Tire Forces Transferred Through Steering Rod

Chris Velez, McNair Scholar
The Pennsylvania State University

McNair Faculty Research Advisor:
Dr. Sean Brennan
Professor of Mechanical Engineering
Department of Mechanical & Nuclear Engineering
The Pennsylvania State University

Abstract

This research investigates a method of estimating road friction using steering linkage measurements of the dynamic tire aligning moment. The linkage forces act on the steering and suspension geometry of a specially instrumented steer-by-wire test vehicle that has a load cell on the steering arm of each front tire. Changes in forces measured within the geometry of the suspension cause aligning torques on the tire which are nonlinearly related to road friction. Because dynamic vertical tire forces also produce lateral forces on the steering linkage, the estimation of friction effects requires accurate kinematic estimation of the tire loads in ord.168o isolate the tire aligning moment from the total linkage force. Accurate friction detection is the first step in implementing a control system that can detect and minimize the effects of road departure, particularly in low-friction situations.

INTRODUCTION

In the United States of America, the number of fatal vehicle crashes due to unintended roadway departures (URD) is over fifty percent [1]. Due to the frequency with which URD occurs, researchers have sought methods of detecting URD, particularly using vision-based sensors on the vehicle. These vision-based sensors seek to detect the location of the lane markers and thereby help the driver stay on course.

However, lane markers are not always visible or detectable by these sensors. The sensor may fail to recognize lane markers in bad weather conditions, or the sensor will not find markers if the road is unmarked. The limited capability of vision-based sensors presents a problem, as it is necessary to have road departure detection methods that work in all road and weather conditions. This paper investigates the viability of detecting changes in road friction, measurable in vehicle dynamic anomalies, as an alternative means of URD detection.

Much of the prior work done in road friction estimation focuses on situations where the vehicle is traveling on a flat road [2]. This is an assumption that greatly influences the behavior of the vehicle, specifically the loads acting on each individual tire. Situations where weight transfer is important include pitch and roll weight transfers caused by superelevation or severe cornering. The goal of this paper is to show the effects of these weight transfers on normal tire loads, which then affect vehicle dynamics. The first part of the paper covers the method of friction estimation in this work. This is then followed by the calculations to determine accurate tire normal loads due to various vehicle states. Following this is a description of the test vehicle and experimental tests. The latter portion of the paper compares experimental data with simulation results.

METHODOLOGY

A. Friction Estimation Approach

The approach investigated by this paper to estimate road friction calls upon the measuring of tire forces on a test vehicle’s steering rod. The term “steering rod” is used interchangeably with “tie rod” throughout this paper. The steer-by-wire vehicle used in this study is equipped with force transducers on both the left and right front axle tie rods. The measured force will reflect the tire forces transferred through the suspension. The self-aligning moment and forces transferred due to tire loads sum at the tie rod transducer. By performing a subtraction of normal tire load contributions to the sum of forces on the tire rod, the remaining force is expected to be due to the lateral force producing the self-aligning moment. A relationship between the lateral force, the coefficient of friction and the normal force at tire contact patch can then be determined.

Outlined in this paper are methods for accurately modeling tire loads when considering changes in ride height and weight transfer when a vehicle is cornering or on a super elevated road for example. An accurate estimate of tire loads is necessary for robust implementation of the proposed friction estimation. The following sections show the calculations for normal loads due to static weight, pitch weight transfer, and roll weight transfer.

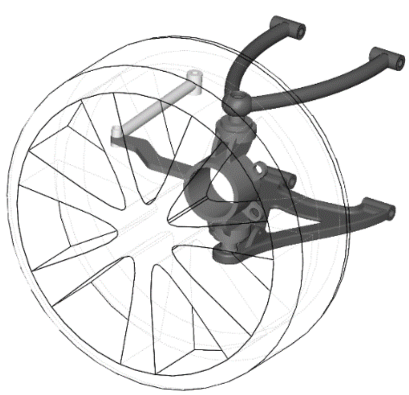

Figure 1: Isometric view of double wishbone

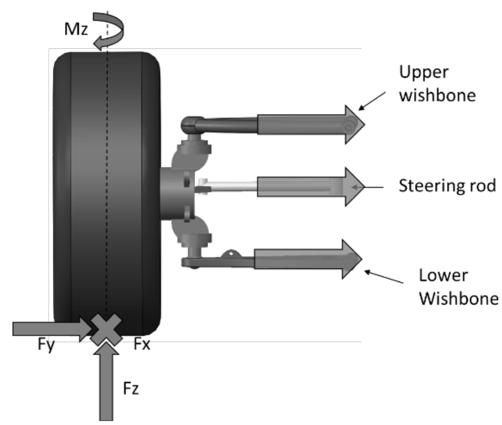

Figure 2: Rearview of tire forces transferring lateral suspension forces into suspension

B. Normal Tire Load Calculations

The normal force at the tire contact patch is derived here for the situation of static equilibrium on a half car model. It is assumed that the left and right sides of the car are identical and symmetrical. Static equilibrium means the sum of forces and torques on the vehicle are equal to zero.

$$\sum M_y = 0 \qquad \sum F_z = 0$$

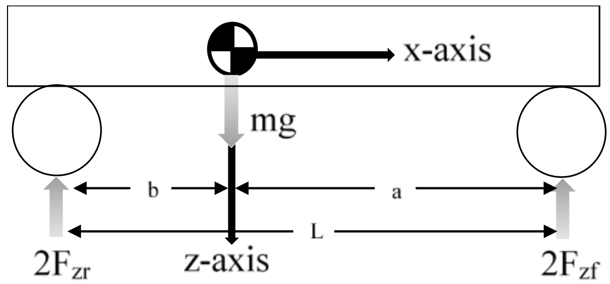

Figure 3: Side view of vehicle model in static equilibrium

The forces acting on the half car model are its weight, located at the center of gravity (CG) location, and the normal force at the front and rear axles. The front axle is a distance “a” from the CG and the rear axle a distance “b” from the CG. The normal forces on the axles depicted below include the normal force for the right and left axles. The distance from the font axel to the rear axle is the wheelbase (L).
The sum of the forces is:

$$2F_{zf} + 2F_{zr} - mg = 0$$

The sum of the moments about the CG is:

$$2F_{zf}\,a + 2F_{zr}\,b = 0$$

Solving this system of equations yields the following results for the normal force acting on each of the front axle tires and rear axle tires:

$$F_{zf}\ mg\,b \qquad\qquad F_{zr}\ mg\,a$$

C. <u>Front to back weight transfer due to longitudinal acceleration (pitch)</u>
This next case includes a longitudinal acceleration acting on the vehicle center of gravity as follows:

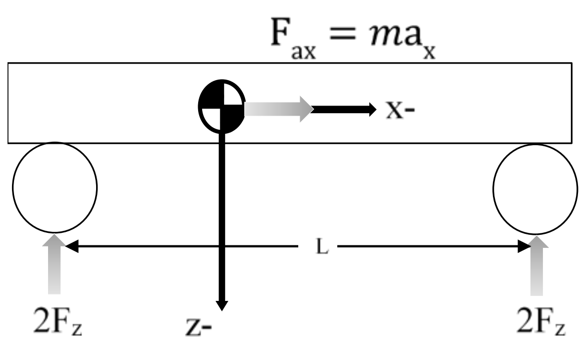

Figure 4: Side view of vehicle model with longitudinal acceleration

The longitudinal force generated at the CG can point to the right or point to the left. This force will produce a moment about the front and rear tire contact points. Ignoring the static weight distribution from the vehicle weight, the sum of the torques can be done about the front contact patch and the rear contact patch to find the extra weight transfer to the rear and front tires respectively.

Summing the moments about the front contact patch results in:

$$\sum M_{front} = 0$$
$$F_{zf}(extra) = -ma_x \frac{h}{2 \cdot L}$$
$$2 \cdot F_{zr}(extra) \cdot L - ma_x h = 0$$

Summing the moments about the rear contact patch results in:

$$F_{zr}(extra) = ma_x \frac{h}{2 \cdot L}$$
$$2 \cdot F_{zf}(extra) \cdot L + ma_x h = 0$$

This extra normal force at the front and rear tires will be added to the static normal loads calculated earlier. This assumption results in the following equations:

$$F_{fl} = mg\left(\frac{b}{2L}\right) - \mathrm{m}a_x\left(\frac{h}{2L}\right)$$
$$F_{fr} = mg\left(\frac{b}{2L}\right) - \mathrm{m}a_x\left(\frac{h}{2L}\right)$$
$$F_{rr} = mg\left(\frac{a}{2L}\right) + \mathrm{m}a_x\left(\frac{h}{2L}\right)$$
$$F_{rl} = mg\left(\frac{a}{2L}\right) + \mathrm{m}a_x\left(\frac{h}{2L}\right)$$

D. <u>Left to right weight transfer due to lateral acceleration (roll)</u>

Now that longitudinal load transfer has been taken into consideration, the next step is to include lateral acceleration typical of a vehicle going around a curve. At steady state the sum of the torque about either the inside or outer tire should be zero. The extra weight on the outer tire balances out the centrifugal force pulling the car towards the outside of the turn.

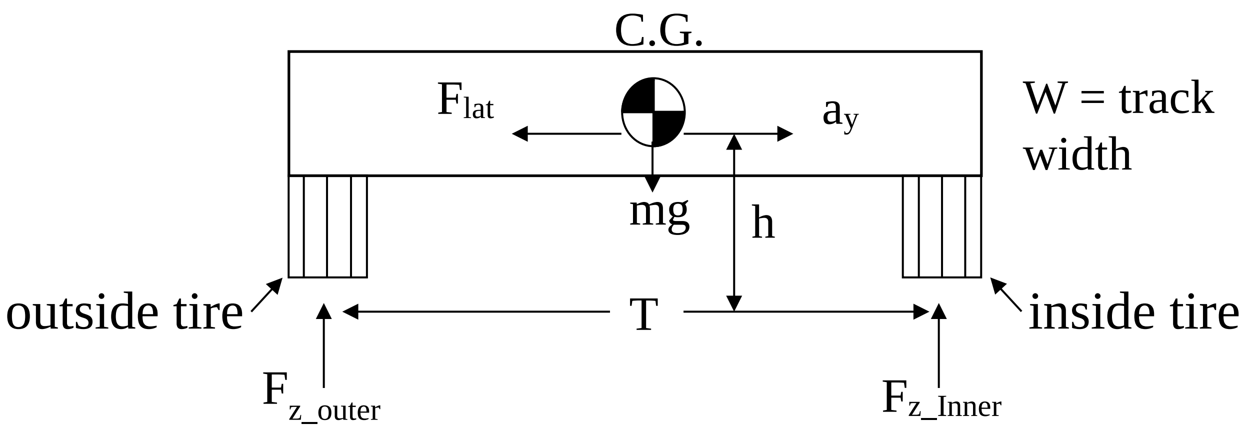

Figure 5: Rear view of vehicle model with lateral acceleration

The sum of the moments about the inside contact point gives:

$$\mathrm{F}_{zo}(\text{extra}) \cdot \mathrm{T} - F_{lat} \cdot \mathrm{h} = 0$$
$$\mathrm{F}_{zo}(\text{extra}) = F_{lat}\frac{\mathrm{h}}{T}$$
$$\mathrm{F}_{lat} = ma_y$$

Similarly, the sum of the moments about the outside contact point:

$$-\mathrm{F}_{zI}(\text{extra}) \cdot \mathrm{T} - \mathrm{m}a_y\mathrm{h} = 0$$

$$\mathrm{F}_{zI}(\text{extra}) = -\mathrm{m}a_y\frac{\mathrm{h}}{T}$$

The following equations are the normal forces on each tire with pitch and roll weight transfer terms included. It is assumed here that all tires have no mass:

$$F_{fl} = mg\left(\frac{b}{2L}\right) - \mathrm{m}a_x\left(\frac{h}{2L}\right) - \mathrm{m}a_y\frac{h}{T}\left(\frac{K_{\phi f}}{K_{\phi f} + K_{\phi r}}\right)$$
$$F_{fr} = mg\left(\frac{b}{2L}\right) - \mathrm{m}a_x\left(\frac{h}{2L}\right) + \mathrm{m}a_y\frac{h}{T}\left(\frac{K_{\phi f}}{K_{\phi f} + K_{\phi r}}\right)$$
$$F_{rr} = mg\left(\frac{a}{2L}\right) + \mathrm{m}a_x\left(\frac{h}{2L}\right) - \mathrm{m}a_y\frac{h}{T}\left(\frac{K_{\phi r}}{K_{\phi f} + K_{\phi r}}\right)$$
$$F_{rl} = mg\left(\frac{a}{2L}\right) + \mathrm{m}a_x\left(\frac{h}{2L}\right) + \mathrm{m}a_y\frac{h}{T}\left(\frac{K_{\phi r}}{K_{\phi f} + K_{\phi r}}\right)$$

THE TEST VEHICLE

The P1 (Fig. 6) was developed by the Dynamics Laboratory and the Product Realization Laboratory at Stanford University. It is a steer-by-wire vehicle which, rather than performing vehicle functions through mechanical linkages, uses instead electro-mechanical systems for steering and throttle inputs (braking inputs are still manual). The P1 vehicle has independent tie motors and independent AC electric drive capability [2]. Onboard P1 there are inertial sensors, steer angle encoders, and a GPS system for obtaining data during test runs.

Figure 6: P1 steer and drive by wire test vehicle

EXPERIMENTAL SETUP

Data collected from P1 during a ramp steer maneuver is used to show the simulation results of accounting for normal tire loads. The experimental run was conducted at The Thomas D. Larson Pennsylvania Transportation Institute’s skid pad. The vehicle velocity, tie rod forces, tie angles along other measurements were taken.

Vehicle velocity reached 10 m/s for the initiation of the ramp steer maneuver (Fig. 7). Both simulation runs are not visible in figure 7 because they used the same longitudinal velocity as inputs. A left-hand turn was performed with the steering angle gradually increased until significant oversteer characteristics were observed (Fig. 8 and 9).

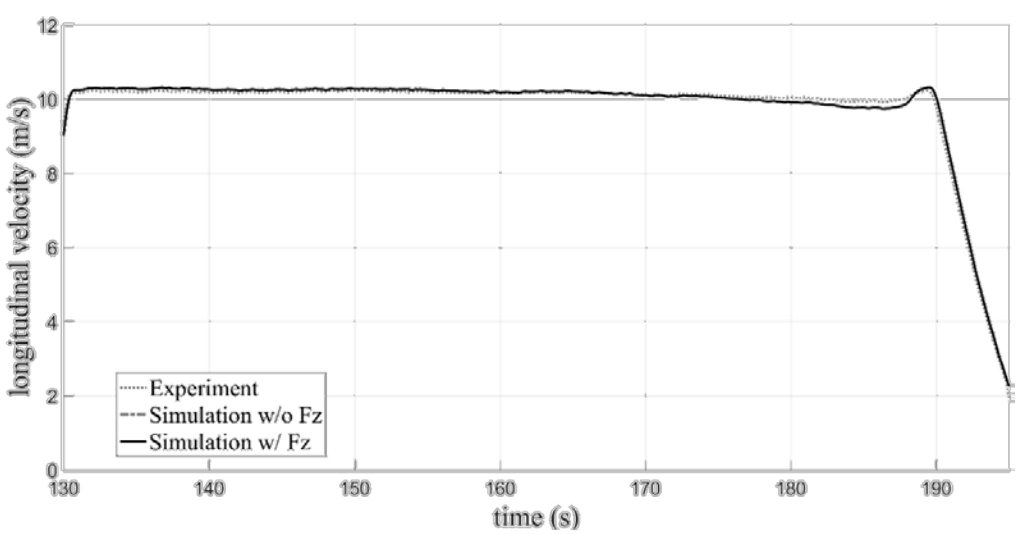

Figure 7: Constant forward velocity during ramp steer test

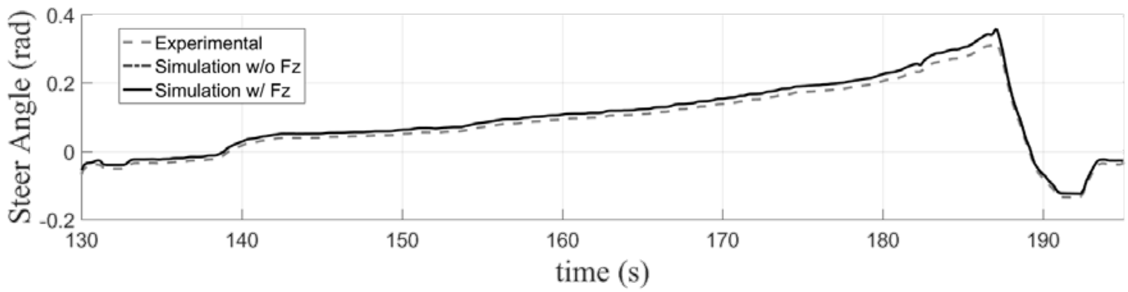

Figure 8: Steering Angle

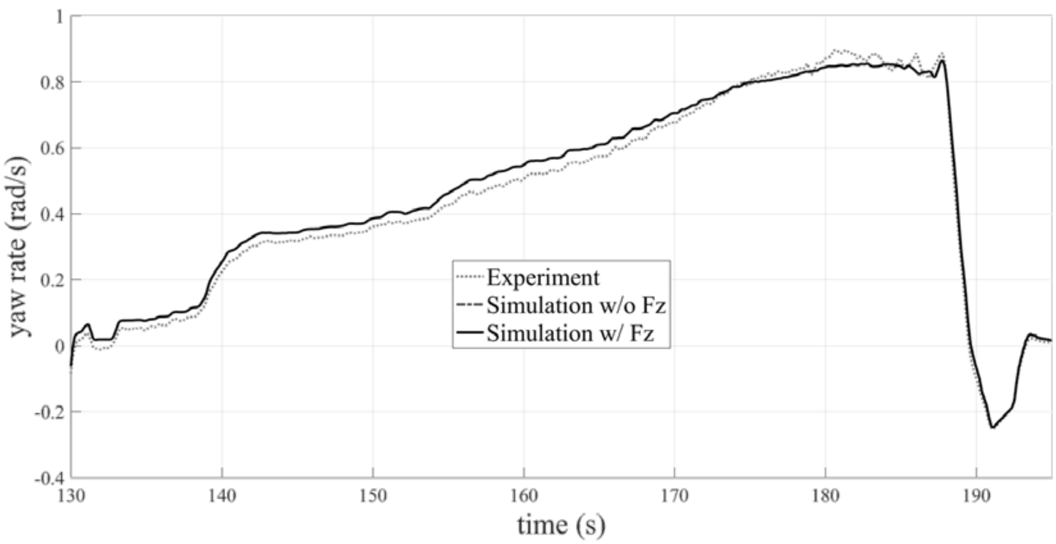

Figure 9: P1 yaw rate during ramp steer maneuver

RESULTS AND CONCLUSIONS

A computer simulation based on blender and MATLAB software uses the inputs of forward velocity and steering angle to produce the plots in figures 10 and 11. The script calculates the force on the steering rod as a result of the forces acting about the tire steering axis. Explaining the plots from top to bottom, this graph shows a much better agreement between simulation and experimental data when we include the normal forces and jacking torques, than if these effects are ignored.

The right steering rod load cell does not show good agreement, which is a result of only slight changes to the outside tire load cell during left-hand cornering. A future research suggestion is to investigate methods for more robust normal load calculations. The next step is isolating the side (lateral) force contribution to the steering rod force by subtracting all other moments producing a force at the steering rod. Finally, a relation can be made between the lateral force and road friction as functions of steering angle.

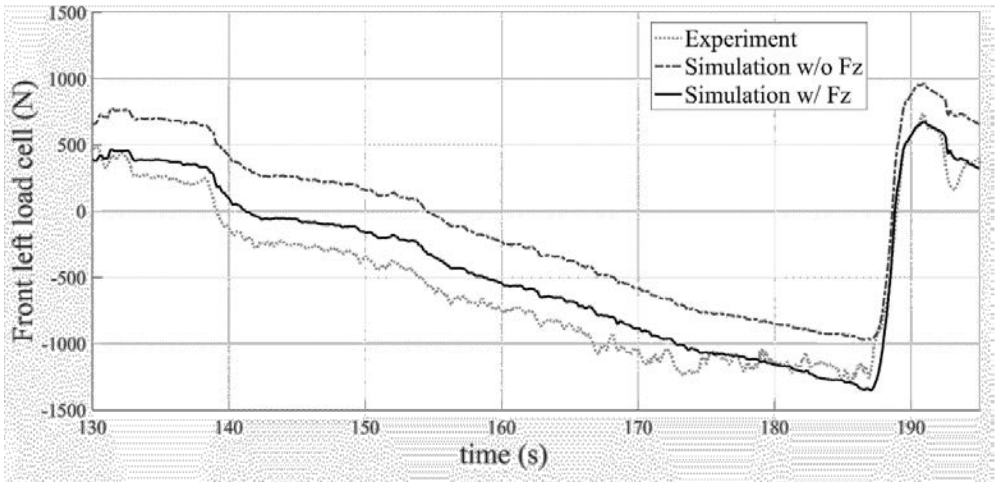

Figure 10: Left Steering rod load cell measurements

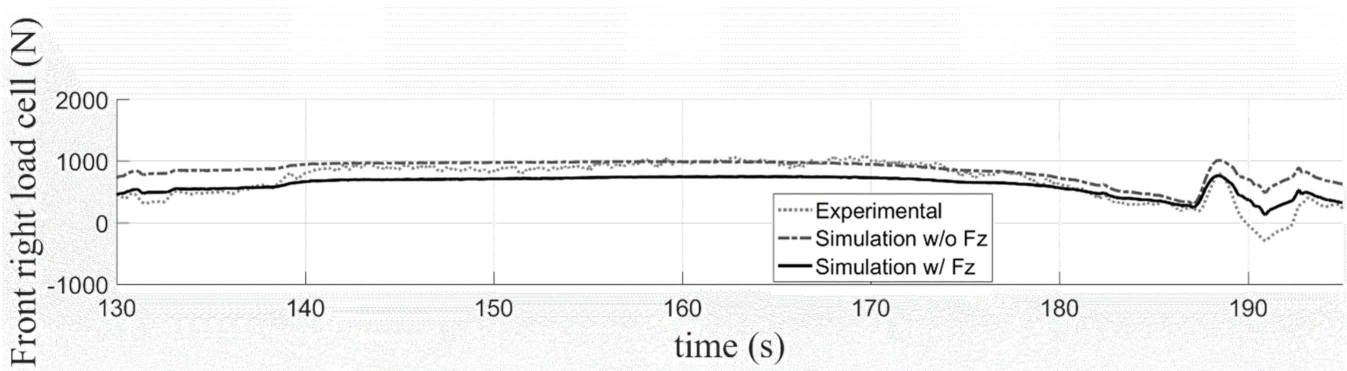

Figure 11: Right steering rod load cell measurements

References

[1] Federal Highway Administration, "Roadway Departure Safety," [Online]. Available: 2016. [Accessed July 2016].

[2] Yung-Hsiang Judy Hsu, "A Method to Estimate the Friction Coefficient and Tire Slip Angle Using Steering Torque," in *ASME International Mechanical Engineering Congress and Exposition*, Chicago, 2006.

[3] Andrew Gray, "Integrated threat assessment and control design for roadway departure avoidance," in *2012 15th International IEEE Conference on Intelligent Transportation Systems*, Anchorage, 2012.

[4] A. Alleyne, "A Comparison of Alternative Intervention Strategies for Unintended Roadway Departure (URD) Control," *Vehicle System Dynamics,* vol. 27, no. 3, 1997.