

Estimating Path Clearing Effects during Potential Hydroplaning through the Use of Vehicle CAN Bus Data

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ABSTRACT

During wet weather operating conditions, tire hydroplaning can occur, potentially altering the handling characteristics of a vehicle. The rear tires of the vehicle run in a path previously cleared by the front tires under some operating conditions. Although path clearing has been previously demonstrated both analytically and qualitatively, it is difficult to estimate the changes in the tire/road coefficient of friction resulting from path clearing because of the complexity of the hydroplaning flow regime. In the present work, we utilize wheelspeed information captured from the vehicle CAN bus and photography to examine potential variations in tire/road coefficient of friction that result from path clearing. Results suggest that differences in friction availability may result from such path clearing. Maneuvers performed include steady-state cornering tests, straight-line braking and ISO lane change maneuvers. An understanding of the path clearing phenomenon is central in the estimation of vehicle paths and behavior during hydroplaning conditions.

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INTRODUCTION

Hydroplaning research has involved both theoretical [1, 2, 3] and experimental [4, 5, 6, 7, 8, 9,10] studies. Experimental studies have involved testing of individual tires on drum and flat-track test machines, as well as towed tire and full-scale vehicle testing. Vehicle testing is usually done on a skidpad, but occasionally is performed on an actual roadway surface. Individual tire testing on skidpads and roads has almost universally been conducted with the tire traveling in a straight line or at a constant radius and speed around a circular skidpad. The experimental studies involving individual tire testing on laboratory machinery and in roadway settings did much to improve understanding of the fundamental physics and behavior of a single tire under hydroplaning conditions, but provided little in-sight into the vehicle dynamics that might result from more than a single hydroplaning tire. Studies involving full-scale vehicle testing, on the other hand, provided insight on the large-scale, overall handling and braking effects of hydroplaning without little or no differentiation or insight into the behavior of individual tire(s).

It is well known that if a loss of traction due to hydroplaning occurs on the rear tires of a vehicle traveling in

a straight line and deceleration or roadway superelevation are present, the vehicle may spin [11, 12, 13]. A spin in itself, even if a contact with another vehicle or object is not involved, can be considered a kind of accident.

In [14], an analysis of vehicle loss of control was presented which attempted to correlate the location (front axle vs. rear axle) of worn tires with accident potential. The paper did not address the issue of the running of rear tires in a path or hydrodynamic regime cleared (at least partially) by the front tires, and hence while qualitatively useful, may have been quantitatively pessimistic in some of its conclusions regarding the effects of worn rear tires on vehicle stability.

In many vehicle handling situations, the rear tires are in fact running in a path at least partially cleared by the front tires. This was examined both theoretically and experimentally in [23]. The experimental program conducted involved straight running of the test vehicle and no quantitative measurements of wheelspeed were taken. Detailed analyses were presented for the cases of hill climbing in a straight line and braking in a straight line.

Additional theoretical and experimental results were presented in [31,32]. In [31] calculations of tire overlap and resultant conditions for path clearing were examined. In [32]

experiments were conducted using the CAN bus ABS sensors of a test vehicle, verifying the qualitative aspects of the theory contained in [23, 31]. In [34], steady-state path clearing calculations were undertaken for analysis of masked hydroplaning in circular skidpad situations. Taken as a group, these results reinforce the concept of the path clearing effect generated by the leading tire.

In this work, we examine the possible effects of tire overlap during various maneuvers via use of CAN bus data taken during full-scale vehicle tests, and with photography. A potential alternate approach is through the extraction of vehicle sideslip and electronic stability information, also from the CAN bus, although that method was not used in the present study.

TEST PROTOCOL AND SET-UP

TEST VEHICLE

The vehicle used in the test program was a front-wheel-drive 2012 Ford Taurus Police Interceptor 4-door sedan, VIN 1FAHP2M81FG119317, delivered new to the Wisconsin State Patrol. The vehicle was fitted with a VBox IIIi measurement system which acquired data at 100 Hz. The VBox unit was configured to acquire wheelspeed information from the vehicle CAN bus through the use of a CAN bus map provided by VBox USA. The use of the dual antenna configuration available on the VBox allowed acquiring roll, pitch and body sideslip angles as well. The test vehicle and equipment are shown in [Figures 1 & 2](#).



Figure 1. 2012 Ford Taurus Police Interceptor test vehicle

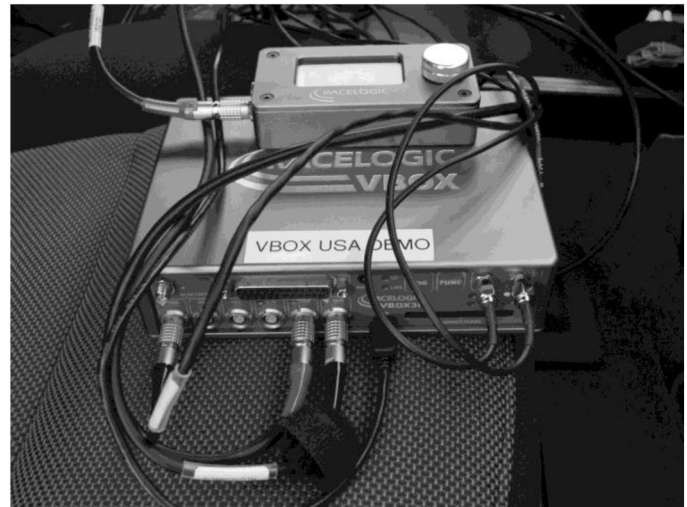


Figure 2. VBox IIIi instrumentation

TEST MATRIX

There were three series of tests performed during the full-scale test program and one flat-plate photography experiment. All tests except for the flat plate experiment were performed at Volk Field, Wisconsin on airport runway surfaces. The surfaces were brushed, striated concrete and were very aggressive. The tests were performed during rain to best identify the effects of path clearing, but rainfall and water depth were obviously not able to be controlled independently.

The test vehicle was equipped with nearly new Goodyear Eagle RSA 245/55-R18 tires at all four wheel locations, with inflation pressures adjusted to 241.3 kPa (35 psi).

The test program consisted of:

- straight-line driving involving acceleration, braking & driving at constant speeds
- circular skidpad driving at non-constant, ever increasing velocity, and
- ISO lane change maneuvers at controlled speeds.

STRAIGHT-LINE TESTS: The purposes of the straight-line runs were (a) to verify that CAN bus wheel revolution values were being successfully captured and recorded, and (b) to examine the possible differences in wheel revolution outputs associated with:

- velocity/centrifugal effects.
- dynamic vertical loading and
- driven vs. coasting wheels,

Wheel revolution CAN bus output values are proportional to, but do not numerically correspond to, actual wheel revolutions/unit time, and must be converted if actual wheelspeeds are desired. (See [Appendix A.2](#) and below.) Hereafter, we simply refer to the CAN bus outputs as wheelspeeds, noting this difference.

A number of similar straight-line braking maneuvers were conducted to different top speeds, ranging from 19-112 kph, and various levels of braking from gentle to severe were used. The actual *instantaneous* rolling radius, and thus wheelspeed, of the tire may vary during a maneuver because of the effects noted above. We examine these effects next.

Radius variation due to velocity/centrifugal effects

Figure 3 displays CAN bus output vs. vehicle speed for a single wheel. During straight running testing, regardless of the speed, it was impossible to distinguish CAN outputs of individual wheels from one another; all four wheels displayed identical values. CAN bus wheelspeed output for all four wheels was also highly linear with velocity. Fitting the data with a linear regression actually produced a correlation coefficient of 1!

From the data given in Figure 3 and the equation of a linear regression of the data:

$$\text{CAN bus value} = 1.5824u - 99.868 \quad (1)$$

Any speed can be used to determine a conversion or scale factor between CAN output and true wheelspeed. Using, e.g., 30 kph = 8.333 m/s and the data shown in Figure 3:

$$\Omega_n = \frac{8.333}{2.284} = 3.649 \frac{\text{rev}}{\text{sec}} \text{ or } 22.923 \frac{\text{rad}}{\text{sec}} \quad (2)$$

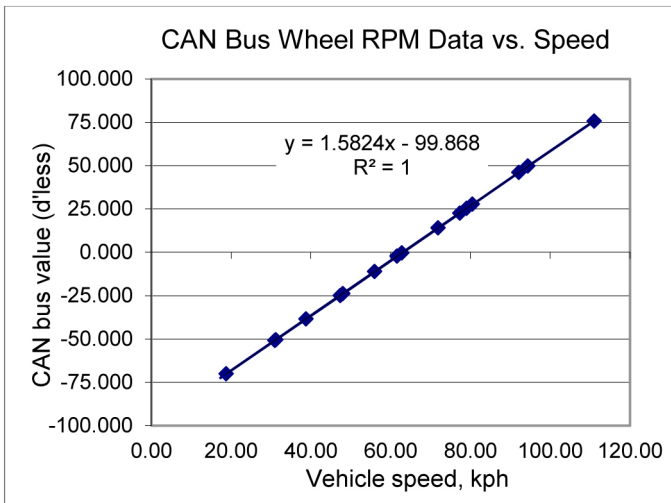


Figure 3. CAN bus calibration from straight-line tests

$$(1.5824 \times 30 - 99.868) = -52.396 = \text{CAN bus value} \\ \equiv 3.649 \frac{\text{rev}}{\text{sec}} \quad (3)$$

$$\frac{3.649}{-52.396} \equiv -0.0696 \frac{\frac{\text{rev}}{\text{sec}}}{\text{CAN value}} \text{ at } 30 \text{ mph} \quad (4)$$

Because of the linearity of the relationship with respect to speed, either calculated rev/sec or actual CAN wheelspeed

output values can be used as a baseline value for the rolling diameter of each wheel during straight running.

Radius variation due to dynamic vertical loading

A rolling radius that changes with dynamic vertical loading will skew CAN bus data, and this possibility must be investigated. With known acceleration, weight transfer from one axle to the other (longitudinal acceleration) or from one wheel to the other across a given (front or rear) axle (lateral acceleration) can be calculated. The static weight distribution and geometry of the test car were as follows:

$$\text{Weight:} \quad W_f = 8,896 \text{ N (2,000 lbf)}$$

$$f/r \text{ dist.} = 58/42$$

$$\text{Dimensions:} \quad \text{wheelbase} = wb = 2,870 \text{ mm (113")}$$

$$\text{front track} = \text{rear track}$$

$$= tw = 1,651 \text{ mm (65")}$$

$$\text{c.g. height} = h = 508 \text{ mm (20")}$$

$$\text{c.g. height/wheelbase ratio} = 0.18$$

$$\text{c.g. height/track width ratio} = 0.31$$

Fractional load transfer (FLT)

In a cornering maneuver, weight transfer from the inside wheels to the outside wheels is given by a Fractional Load Transfer (FLT) calculation for each individual axle:

$$\begin{aligned} \text{FLT (front axle)} &= \frac{W_{if} - W_{of}}{2 \times W_t \times a_y \times h} \\ &= \frac{tw \times g_c}{2 \times 12,276 \times 508 \times a_y} \\ &= \frac{1,651 \times 9.8 \times 1000}{1,651 \times 9.8 \times 1000} \\ &= 0.771a_y \end{aligned} \quad (5)$$

$$\begin{aligned} \text{FLT (rear axle)} &= \frac{W_{ir} - W_{or}}{2 \times 8,896 \times 508 \times a_y} \\ &= \frac{1651 \times 9.8 \times 1000}{1651 \times 9.8 \times 1000} \\ &= 0.559a_y \end{aligned} \quad (6)$$

During a straight-line braking or acceleration maneuver, FLT from rear wheels to front wheels (only one calculation is needed for front/rear weight transfer) is given by:

$$\begin{aligned} \text{FLT}_{f/r} &= \frac{W_f - W_r}{2 \times W_t \times a_x \times h} \\ &= \frac{wb \times g_c}{2 \times 21,172 \times 508 \times a_y} \\ &= \frac{2,870 \times 9.8 \times 1,000}{2,870 \times 9.8 \times 1,000} \\ &= 0.765a_y \end{aligned} \quad (7)$$

In Eqs (5, 6, 7), a_y is measured in g-units.

Three different radii can be defined [46]:

- **Free or unloaded radius r_F** : given by dividing tire circumference by 2π , as outlined in Eqs (A.2.1, A.2.2, A.2.3, A.2.4)
- **Loaded radius r_L** : given by measuring the distance from the wheel axis to the ground when the wheel is loaded (but usually done statically, not on a rolling wheel)
- **Rolling radius r_R** : defined as forward velocity/tire angular velocity

For a wide range of vertical loads, the tire behaves as a linear spring. Tire deflection under the influence of vertical loading can be defined as:

$$\delta_t = \text{loaded radius} - \text{unloaded radius} = R_L - R_F \quad (8)$$

Under normal or vertical load F_z [20], tire deflection near rated load is approximately 18-24% of section height [46]. In terms of tire stiffness K_t :

$$\delta_t = \frac{F_z}{K_t} \quad (9)$$

A typical passenger car vertical tire stiffness K_t is 250 N/mm [46]. An approximate figure for rolling radius based on un-loaded radius, using (18+24)/2=21% deflection, is:

$$r_R \cong r_F - \frac{\delta_t}{3} = \frac{726.7}{2} - \frac{0.21 \times 134.75}{3} = 0.354m \quad (10)$$

A reduction in rolling radius of such a small magnitude produces a percentage change of only:

$$\% \text{ change in rolling radius} = \left\{ \frac{0.363 - 0.354}{0.363} \right\} \times 100 = 2.48\% \quad (11)$$

due to static vertical loading. Clearly, using the free radius r_F or diameter for calculations during straight-ahead maneuvers introduces little error.

Sample Calculation for dynamic loading

Assume a 0.5g lateral acceleration. Then:

$$FLT_f = 0.771 \times 0.5 = 0.386 \quad (12)$$

$$FLT_r = 0.559 \times 0.5 = 0.280 \quad (13)$$

The FLTs at the front and rear are given by:

$$\begin{aligned} FLT_f &= 0.771 \times \frac{12,276}{2} = 4,732 \text{ N} \\ FLT_r &= \frac{8,896}{2} \times 0.280 = 1,245 \text{ N} \end{aligned} \quad (14)$$

and the resulting individual tire loads become:

$$\begin{aligned} W_{if} &= \frac{12,276}{2} - 4,732 = 1,406 \text{ N} \\ W_{of} &= \frac{12,276}{2} + 4,732 = 10,870 \text{ N} \\ W_{ir} &= \frac{8,896}{2} - 1,245 = 3,204 \text{ N} \\ W_{or} &= \frac{8,896}{2} + 1,245 = 5,693 \text{ N} \end{aligned} \quad (15)$$

With a vertical stiffness of 250 N/mm, an increase in vertical loading will produce an additional deflection of magnitude:

$$\delta_{0.5g} = \frac{4,732}{250} = 18.9 \text{ mm} \quad (16)$$

and a corresponding change in rolling radius r_R of:

$$\Delta r_R = \frac{18.9}{3} = 6.3 \text{ mm} \quad (17)$$

At 0.5g, the rolling radius r_R is given by:

$$r_R = 354 - 6.3 = 0.347m \quad (18)$$

And the %-change from the free radius r_F value is:

$$\begin{aligned} \% \text{ - change in rolling radius} &= \left\{ \frac{0.363 - 0.347}{0.363} \right\} \times 100 \\ &= 4.4\% \end{aligned} \quad (19)$$

From the above, it is clear that changes in tire angular velocity/rolling radius due to vertical load changes are small. Thus, changes in wheelspeed measured by CAN bus data during dynamic maneuvers can be attributed to actual %-slip between the individual tire and roadway, leading to the possibility of making inferences regarding tire/road frictional values and hydroplaning masking effects. The lower the acceleration values (fore/aft or lateral), the more closely this approximation applies.

Wheelspeed differences for driven vs. coasting wheels: Wheelspeeds were continuously recorded for all runs to differing top speeds. Wheelspeeds for the front, driven wheels were consistently very slightly lower than those for the rear, coasting wheels, as expected. For example, at 80 kph, the wheel-speed outputs from the CAN bus were:

LF: 27.947 RF: 27.901 Average: 27.924

LR: 27.507 RR: 27.546 Average: 27.526

$$\% \text{ difference} = \left\{ \frac{27.924 - 27.526}{27.924} \right\} \times 100 = 1.42\% \quad (20)$$

Finally, experience shows that rolling radius changes due to lateral tire deflection occurring during high-g cornering maneuvers is $\ll$ the deflections examined above.

We can be confident that differences in wheelspeeds due to driven vs. coasting wheels will not overwhelm any hydroplaning path clearing wheelspeed differences present.

CIRCULAR SKIDPAD TESTS

The purpose of the circular skidpad tests was to verify the ability to separate small differences in wheelspeed outputs from the vehicle CAN bus. Two tests were run, one clockwise (CW) and one counterclockwise (CCW). The skidpad radius was 7.6m (25 ft.). Speed was gradually increased until it became impossible to maintain a circular trajectory. At slow speeds, the vehicle wheelspeeds increased linearly with speeds, but as the driver increased speed and it became more and more difficult to maintain accurate path trajectory, wheelspeeds varied considerably. Figure 4 shows CAN bus output wheelspeeds for the CW circular skid-pad run. The noisy trace in Figure 4 is the right-rear wheel CAN bus wheelspeed output. During a CW run, the right-rear wheel will be lightly loaded due to lateral load transfer. As a consequence, wheelspeed is expected to be more variable than the other wheels as lateral acceleration builds and unloads that wheel. These tests were run on dry pavement.

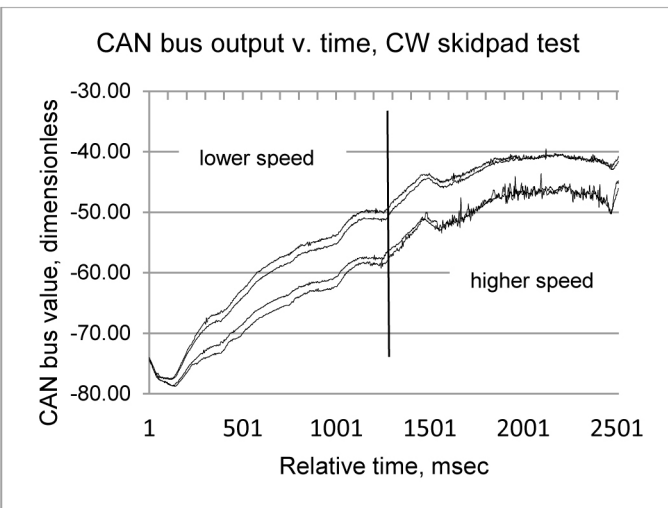


Figure 4. CAN bus outputs for CW circular skidpad test

LANE CHANGE TESTS

The purpose of the lane change tests was to examine changes in tire/road coefficient of friction associated with path clearing during hydroplaning conditions via the use of CAN wheelspeed data. The concept of lane change behavior has been studied extensively. In an early work [38] single lane-change maneuvers were studied using the bicycle model for handling analysis, but no sideslip angle data were presented. A series of differing levels of urgency presented to a driver in a single lane change was developed in [39] and correlation between test data and the urgency of the

maneuver was developed. Using linear control theory for both a driver and vehicle model, analytical work was presented in [40] and compared to test track data; again, no data were presented on sideslip angle. Through the use of Lagrangian dynamics and the calculus of variations, [41,42] developed analytical trajectories for lane change maneuvers, and presented criteria for use in the objective function required for an optimal control approach. In [43], a simplistic model for modeling lane changes was correlated with its potential for accident reconstruction, essentially mimicking that given in [44]. None of the above research directly addresses the issues of tire over-lap and potential hydroplaning.

ISO 3888-1 Test Protocol

Unlike the well-defined and readily understood characteristics of straight-line and circular skidpad testing [20], transient testing presents the vehicle dynamicist with an infinite number of potential test protocols. In order to standardize a protocol for transient maneuvers useful for testing vehicle performance and handling characteristics, the International Standards Organization (ISO) has developed a template for a double lane change maneuver [35]. The maneuver is divided into six sections, as shown in Figure 5:

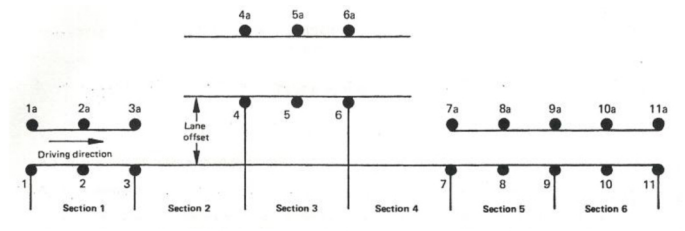


Figure 5. ISO 3888-1 lane change layout [35]

While the maneuver section lengths are independent of the test vehicle dimensions, the widths of sections 1, 3 and 5 depend on the overall width (OAW) of the vehicle; see Tables 1 & 2:

Table 1. ISO 3888-1 lane change protocol [35]

Section 1	15 m
Section 2	30 m
Section 3	25 m
Section 4	25 m
Section 5	15 m
Section 6	15 m