

Hydroplaning Behavior during Steady- State Cornering Maneuvers

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ABSTRACT

Vehicles running in wet conditions may experience hydroplaning of one or more tires. Hydroplaning can, and often does, change vehicle braking, acceleration and handling characteristics dramatically. Proper analysis of this behavior requires accommodating the clearing of paths for the rear tires that may result from the front tires engaging the water-coated surface first. In this work, a hydroplaning analysis is presented that examines steady-state cornering under potential hydroplaning situations and includes lateral weight transfer, tire load sensitivity and path clearing potential. The sensitivity of vehicle understeer/oversteer characteristics to path clearing and vehicle dimensional characteristics is also examined.

INTRODUCTION

In many accident situations involving inclement weather, hydroplaning is alleged to have initiated the loss of control and subsequent accident. Because a hydroplaning tire can experience a near-total loss of frictional capability, the driver of such a vehicle cannot exert control inputs at the hydroplaning tire/road interface large enough to recover from such a loss of control. From a vehicle dynamics and control point of view, the behavior of a hydroplaning tire is characterized by a drastic reduction in the radius of its friction circle.

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. Such 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. The experimental studies involving individual tire testing on laboratory machinery and on roadway settings did much to improve understanding of the fundamental physics and behavior of a single tire under hydroplaning conditions, but provided little insight into the vehicle dynamics that might result from one or more hydroplaning tires. Studies involving full-scale vehicle testing, on the other hand, provided insight on the handling and braking effects of hydroplaning with 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 is almost certain to 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. While useful as an initial study, 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, was 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 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 wheel spin were taken. Detailed analyses were presented for the cases of hill climbing in a straight line and braking in a

straight line. A brief discussion of cornering was included, but lateral weight transfer and tire load sensitivity were not included. In the present work, we present new results that include these effects on the steady-state cornering capability of a vehicle running in hydroplaning conditions.

THEORETICAL ANALYSIS

To begin the analysis of hydroplaning during steady-state cornering, some fundamental concepts are needed.

TANGENT SPEED

At very low speeds in a circular cornering maneuver, the rear tires track a smaller radius than the front tires because the vehicle is Ackermann steer. Under these conditions, the steering angle required to maintain a desired circular path radius r is purely kinematic, as shown in [Figure 10](#), and is given by:

$$\delta = \tan^{-1}(\ell/r) \cong \frac{\ell}{r} \quad (1)$$

with ℓ =vehicle wheelbase and r =turn radius. As speed is increased and tire slip angles build to develop the lateral forces needed to resist centrifugal force, body attitude β also increases until, at some speed, the front and rear tires track on circles of the same radius. At this speed, the chassis centerline is tangent to the circular path and the vehicle body attitude angle β , measured at the c.g. of the vehicle, is zero. Above the tangent speed, the rear wheels travel on a larger radius than the front wheels and the vehicle assumes a nose-in attitude. The speed at which $\beta=0$ is termed the vehicle tangent speed. Tangent speed is given by [17]:

$$u_t = \sqrt{\frac{-\ell gb C_{rt}}{W_a}} \equiv \sqrt{\frac{-gb C_{rt}}{W_r}} \quad (2)$$

In [Eq \(2\)](#), u_t =tangent speed, g =acceleration of gravity, b =distance from c.g. to rear axle, W_r =weight on vehicle rear axle and C_{rt} =the sum of the tire cornering stiffness coefficients for both rear tires [24]. With front and rear track widths that are approximately equal, a vehicle traveling in a circular path at or near the tangent speed will have its rear wheels running in a path at least partially cleared by its front wheels. We return to the concept of tangent speed later in the paper.

TIRE LOAD SENSITIVITY

As the vertical load on a tire increases, the tire/road coefficient of friction under dry conditions decreases. This phenomenon is known as tire load sensitivity. Load sensitivity is typically a weak, monotonically decreasing,

nonlinear phenomenon, and is often linearized, as shown in [Figure 1](#), to the form:

$$\mu = k_o - k_1 F_z \quad (3)$$

In [Eq \(3\)](#), μ =tire/road coefficient of friction, k_o =y-axis intercept, k_1 =slope and F_z =vertical tire load. Consider a vehicle of total weight W . Front and rear axle static or curb loads W_f and W_r are given by:

$$W_f = \frac{Wb}{\ell}, \quad W_r = \frac{Wa}{\ell} \quad (4)$$

With a 50/50 lateral (but not necessarily longitudinal) weight distribution, the individual inner and outer wheel static or curb vertical loads are divided evenly and are given by:

$$W_{fi} \equiv W_{fo} = \frac{Wb}{2\ell}, \quad W_{ri} \equiv W_{ro} = \frac{Wa}{2\ell} \quad (5)$$

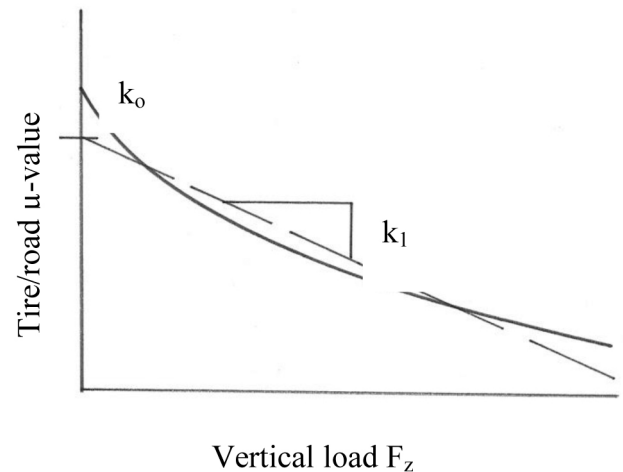


Figure 1. Linearized tire load sensitivity

In [Eq \(5\)](#), the subscripts $()_o$ and $()_i$ indicate outside and inside tires respectively. This notation is carried through the rest of the paper.

LATERAL WEIGHT TRANSFER

Consider a steady-state circular cornering maneuver at lateral acceleration a_y , not necessarily at the vehicle tangent speed u_t . An infinite number of $\{u, r\}$ pairs can produce a given level of lateral acceleration. With knowledge of the height of the center of mass and the front and rear track widths of the vehicle, and ignoring suspension characteristics for the moment, the dynamic inner and outer wheel normal loads can

be readily calculation from the Fractional Load Transfer (FLT):

$$FLT = \frac{2\bar{a}_y h}{t} \quad (6)$$

with $\bar{a}_y$ =lateral acceleration, h =c.g. height and t =track width. For the front and rear axle inner and outer wheel loads at lateral acceleration $\bar{a}_y$:

Front tires:

$$W_{fi} = \frac{Wb}{2\ell} - \frac{2\bar{a}_y h}{t_f}, \quad W_{fo} = \frac{Wb}{2\ell} + \frac{2\bar{a}_y h}{t_f} \quad (7)$$

As before $(\circ)_o$ and $(\circ)_i$ indicate outside and inside tires respectively and $(\circ)_f$ and $(\circ)_r$ indicate front and rear tires; again, similar notation is carried through the rest of the paper. For the rear tires:

$$W_{ri} = \frac{Wa}{2\ell} - \frac{2\bar{a}_y h}{t_r}, \quad W_{ro} = \frac{Wa}{2\ell} + \frac{2\bar{a}_y h}{t_r} \quad (8)$$

With the individual tire vertical loads given by Eqs (7-8) and linearized tire load sensitivity data in the form of Eq (3), the lateral force balance during a steady-state cornering maneuver can be calculated:

$$\begin{aligned} a_y W = & W_{ri}(k_{ori} - k_{lri}W_{ri}) + W_{ro}(k_{oro} - k_{lro}W_{ro}) \\ & + W_{fi}(k_{ofi} - k_{lfi}W_{fi}) + W_{fo}(k_{ofo} - k_{lfo}W_{fo}) \end{aligned} \quad (9)$$

Ignoring suspension effects, the lateral acceleration $\bar{a}_y$ is resisted by each axle in proportion to the fore/aft vehicle weight distribution. In a steady-state corner ($\bar{a}_x=0$), the static fore/aft weight distribution and dynamic fore/aft weight distribution are identical and invariant.

If p is the %-weight on the front axle of the vehicle, $(1-p)$ is the rear axle %-weight. For a real vehicle with suspension characteristics, obviously p is also affected by antiroll bar and spring rates, suspension design and other factors, i.e., the vehicle roll couple distribution, camber curve characteristics, etc.

Eq (9) gives the maximum lateral acceleration that can be sustained for a given set of vehicle characteristics (h, t_f, t_r, W , fore/aft weight distribution) and tire characteristics (k_o, k_i).

Note that k_o and k_i need not be identical for all four tires, as is common on, e.g., sports cars (Porsche, Corvette, etc.).

Knowledge of the fore/aft weight distribution of the vehicle enables separating out the individual axle resisting forces in Eq (9). With p % of the total vehicle weight on the front axle, expanding Eq (9) for that axle gives:

$$\begin{aligned} a_y W p = & \left\{ \frac{Wb}{2\ell} - \frac{2\bar{a}_y h}{t_f} \right\} \left\{ k_{ofi} - k_{lfi} \left[\frac{Wb}{2\ell} - \frac{2\bar{a}_y h}{t_f} \right] \right\} + \\ & + \left\{ \frac{Wb}{2\ell} + \frac{2\bar{a}_y h}{t_f} \right\} \left\{ k_{ofo} - k_{lfo} \left[\frac{Wb}{2\ell} + \frac{2\bar{a}_y h}{t_f} \right] \right\} \end{aligned} \quad (10)$$

And for the rear axle:

$$\begin{aligned} a_y W (1-p) = & \left\{ \frac{Wa}{2\ell} - \frac{2\bar{a}_y h}{t_r} \right\} \left\{ k_{ori} - k_{lri} \left[\frac{Wa}{2\ell} - \frac{2\bar{a}_y h}{t_r} \right] \right\} + \\ & + \left\{ \frac{Wa}{2\ell} + \frac{2\bar{a}_y h}{t_r} \right\} \left\{ k_{oro} - k_{lro} \left[\frac{Wa}{2\ell} + \frac{2\bar{a}_y h}{t_r} \right] \right\} \end{aligned} \quad (11)$$

Figure 2 displays the results of a typical plot of h/t ratio vs. lateral acceleration for a generic vehicle. With a given value of h/t on the abscissa of Figure 2, the axle that can sustain the highest lateral acceleration $\bar{a}_y$ at that h/t value determines the steady-state understeer/oversteer characteristic of the vehicle. If axle 1 in Figure 2 is the rear axle, the car will understeer; if axle 1 is the front axle, the car will oversteer. If the curves for the two axles of a vehicle fall one atop the other, the car will be neutral steer.

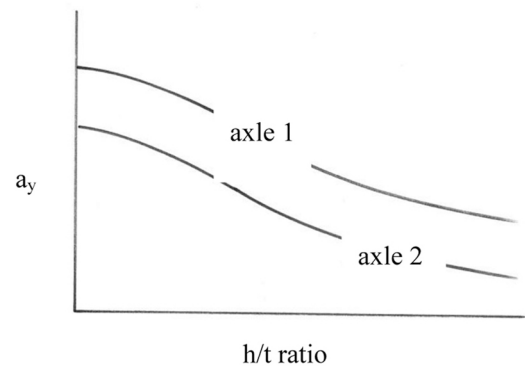


Figure 2. $\bar{a}_y$ vs. h/t ratio for front & rear axles

UNDERSTEER GRADIENT (UG)

UG is a function of fore/aft weight distribution (a,b) and fore/aft tire distribution (C_f , C_r , in units of lbf/rad) and is given by:

$$UG = 57.3W \left\{ \frac{aC_f - bC_r}{\ell C_f C_r} \right\}, \quad \frac{\text{deg}}{g} \quad (12)$$

The calculation in Eq (12) is obviously valid only in the linear range of tire sideforce development (typically with lateral acceleration $\sim \leq 0.4g$) and ignores drive effects. For an understeer vehicle under these assumptions, $(aC_f - bC_r) > 0$ (recall that C_f and C_r are negative) and thus $UG > 0$ and axle 1 in Figure 2 would be the rear axle of such a vehicle.

Eq (12) was developed for the so-called bicycle model. This model lumps inside and outside tire cornering stiffness coefficients into a single value and thus does not include tire load sensitivity. With load sensitivity, UG is given by:

$$UG = 57.3W \left\{ \frac{a(C_{fo} + C_{fi}) - b(C_{ro} + C_{ri})}{\ell(C_{fo} + C_{fi})(C_{ro} + C_{ri})} \right\} \quad (13)$$

With the above concepts and assumptions, analysis of a sample vehicle in a steady-state corner on a surface that presents the potential for hydroplaning can be developed.

SAMPLE VEHICLE CALCULATIONS

A series of numerical calculations was performed using a 2003-2009 Ford Crown Victoria full-size sedan as an exemplar vehicle. The dimensional and inertial properties of this vehicle are given in the Appendix of the paper, and were taken from the HVE vehicle database for this vehicle [18]. In that database, the standard tire fitted to the exemplar vehicle is a P225/60-R16. A slip angle vs. sideforce curve for this tire is shown in Figure 3 for the case when $F_z = 650$ lbf (2,891 N). The tire characteristics presented in the HVE database are generic in that they represent a typical tire of this size, aspect ratio, etc., but not necessarily any particular manufacturer of the tire (e.g., Goodyear, Bridgestone, Kuhmo, etc.).

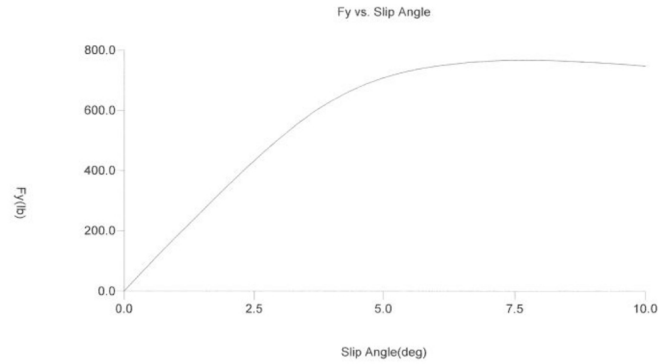


Figure 3. Ford Crown Victoria Tire Data (Source: Ref [18])

Data for this generic tire are available from [18] for three different vertical loads:

Vertical load F_z	C , lbf/deg	C , N/deg	$\mu_{\max}$
650 lbf/2,891 N	-177.4	-789.1	1.15
1,300 lbf/5,782 N	-331.9	-1,476.3	1.10
1,950 lbf/8,674 N	-387.4	-1,723.2	1.05

Following the methodology outlined in Eq (3) for linearization of tire sensitivity and cornering stiffness variation, the following linearized expressions are obtained:

Cornering Stiffness Coefficient:

English units:

$$\begin{aligned} C_i &= -0.1615F_z - 88.9 \\ C_{oi} &= -88.9, C_{li} = -0.1615 \end{aligned} \quad (14)$$

SI units:

$$\begin{aligned} C_i &= -0.0363F_z - 88.9 \\ C_{oi} &= -88.9, C_{li} = -0.0363 \end{aligned} \quad (15)$$

Peak Coefficient of Friction:

$$\mu = -8.0 \times 10^{-5} F_z + 1.2 \quad K_o = -8.0 \times 10^{-5}, k_l = 1.2 \quad (16)$$