

A Direct Yaw Control Algorithm for On- and Off-Road Yaw Stability

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Virginia TechSaied Taheri
Virginia Polytechnic Inst & State Univ

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ABSTRACT

Models for off-road vehicles, such as farm equipment and military vehicles, require an off-road tire model in order to properly understand their dynamic behavior on off-road driving surfaces. Extensive literature can be found for on-road tire modeling, but not much can be found for off-road tire modeling. This paper presents an off-road tire model that was developed for use in vehicle handling studies. An on-road, dry asphalt tire model was first developed by performing rolling road force and moment testing. Off-road testing was then performed on dirt and gravel driving surfaces to develop scaling factors that explain how the lateral force behavior of the tire will scale from an on-road to an off-road situation. The tire models were used in vehicle simulation software to simulate vehicle behavior on various driving surfaces. The simulated vehicle response was compared to actual maximum speed before sliding vs. turning radius data for the studied vehicle to assess the tire model.

A direct yaw control algorithm was then developed to enhance vehicle yaw stability on both on- and off-road driving surfaces. The algorithm is based on Lyapunov stability theory and utilizes a differential braking strategy to apply a corrective yaw moment to the vehicle to stabilize its yaw rate. The algorithm was tested in a vehicle simulation environment using the developed tire models. Results show that the proposed algorithm improves vehicle yaw stability and handling on both on- and off-road driving surfaces.

INTRODUCTION

Off-road tire testing on larger tires is either too expensive or not feasible. In the past, off-road tire models have been developed for farming equipment [1]. This tire model was used to perform both on- and off-road vehicle simulations [2], [3]. In the current paper, the studied tire required a rig that occupied approximately 40 m² to perform on-road force and moment response tests. As a result, on- and off-road testing was performed on a passenger tire in order to draw conclusions about how the lateral force behavior of a tire will change from one surface to another. It was found in the literature that the change in lateral force behavior of a tire from one surface to another surface is primarily independent of tire size, tire type, vehicle type, and vehicle forward speed [4]. It was also found in [4] that the majority of the change in the lateral force behavior of a tire occurs in the peak value and cornering stiffness. In other words, lateral force peak value and cornering stiffness scaling factors can be determined for a driving surface and will explain how tire lateral force behavior scales from one driving surface to another. The procedure for developing the off-road tire model for the studied tire is as follows.

Force and moment responses for the studied tire were measured due to various slip angles, camber angles, vertical loads, and forward speeds. The tests were performed on a rolling road stainless steel flywheel that closely resembles a dry asphalt driving surface. The collected data was curve fitted to generate a Pacejka Magic Formula tire model [5],[6], [7]. Determination of Pacejka coefficients is well documented in the literature [8], [9], [10]. On- and off-road force and moment testing was then performed on a passenger tire by

using a portable tire test rig. The passenger tire was subjected to various slip angles and vertical loads on dry asphalt, dirt, and gravel driving surfaces. The data for the passenger tire was then analyzed and scaling factors were developed that explain how the peak lateral force and cornering stiffness scale from dry asphalt to dirt and from dry asphalt to gravel. These scaling factors were then applied to the on-road model of the studied tire to generate an off-road lateral force tire model.

A direct yaw control (DYC) algorithm was then developed to improve vehicle yaw stability on both on- and off-road terrain. In the past, algorithms have been presented that use a model regulator [11] and fuzzy logic control [12] to provide vehicle yaw stability through differential braking and active front steering. The algorithm presented in this paper is derived from Lyapunov stability theory and uses vehicle lateral velocity and yaw rate measurements to calculate a corrective yaw moment for stabilizing the vehicle yaw motion. The corrective moment is applied through a differential braking strategy in which one wheel is selected and appropriate brake torque is applied. The algorithm was tested on simulated asphalt and dirt driving surfaces.

ON-ROAD TIRE MODEL

An on-road model for the studied tire was developed by performing rolling road testing. The tire was tested at four forward speeds (8.0, 32.2, 64.4, 104.6 km/h) and seven vertical loads (31.90, 34.03, 40.03, 48.04, 56.05, 64.05, and 68.06 kN). At each combination of forward speed and vertical load a slip angle sweep at zero degrees camber angle and a camber angle sweep at zero degrees slip angle were performed. The slip angle sweep was a sine wave with 20 degrees amplitude and 0.033 Hz frequency, and the camber angle sweep was a sine wave with 16 degrees amplitude and 0.033 Hz frequency. The collected data was filtered with an FFT filter to remove the effects of oscillating forces due to excited modes in the tire during testing. The excited modes occurred mostly at high forward speeds and high slip angles. The filtered data was then averaged with a moving average window to remove the hysteresis effects that arose from tread and carcass compliance and damping. This resulted in steady state data which was used to create the on-road tire model.

The Magic Formula (MF) was used to model the force behaviour of the studied tire. A simplified version of the equations presented in [7] are given in equations (1), (2), (3), (4), (5), (6), (7), (8), (9). The lateral force is modelled by the MF equation:

$$F_y = D \sin \left(C \arctan \left(B \alpha_y - E \left(B \alpha_y - \arctan \left(B \alpha_y \right) \right) \right) \right) + S_V \quad (1)$$

where F_y is the lateral force acting at the tire contact patch, B , C , D , E , S_H , S_V , are called main coefficients, and α_y is an

adjusted slip angle. B is called the stiffness factor, C is called the shape factor, D is called the peak value, E is called the curvature factor, S_H is called the horizontal shift, and S_V is called the vertical shift. The main coefficients and adjusted slip angle are defined by:

$$\alpha_y = \alpha + S_H \quad (2)$$

$$C = a_0 \quad (3)$$

$$D = (a_1 F_z^2 + a_2 F_z) (1 - a_{15} \gamma^2) \quad (4)$$

$$E = (a_6 F_z + a_7) (1 - (a_{16} \gamma + a_{17}) \text{sign}(\alpha_y)) \quad (5)$$

$$BCD = a_3 \sin(2 \arctan(F_z / a_4)) (1 - a_5 |\gamma|) \quad (6)$$

$$B = BCD / (CD) \quad (7)$$

$$S_H = a_8 F_z + a_9 + a_{10} \gamma \quad (8)$$

$$S_V = a_{11} F_z + a_{12} + (a_{13} F_z + a_{14}) \gamma, \quad (9)$$

where F_z is the vertical load (-kN) on the tire, γ is the tire camber angle (degrees), α is the tire slip angle (degrees), and $a_0, a_1, \dots, a_{17}$ are called Pacejka coefficients. A Matlab curve-fitting script was written to solve for the Pacejka coefficients for each forward speed. The results of the curve fitting routine are shown in Table 1. These coefficients are used with equations (1), (2), (3), (4), (5), (6), (7), (8), (9) to determine the lateral force for vertical load, slip angle, and camber angle inputs.

Figure 1 shows a plot of the processed (FFT filtered and averaged) lateral force vs. slip angle data at 8 km/h for selected vertical loads. The MF fit using the coefficients in the first column of Table 1 and equations (1), (2), (3), (4), (5), (6), (7), (8), (9) is also shown. Figure 2 shows the collected data for lateral force vs. camber angle overlaid with the MF fits. The plots show that the determined Pacejka coefficients accurately fit the measured tire data.

Table 1. Lateral force Pacejka coefficients for studied tire on dry asphalt.

	Speed (km/h)			
	8.0	32.2	64.4	104.6
a_0	1.048	1.239	1.500	1.200
a_1	-5.498	-6.600	-6.753	-5.899
a_2	-1038.015	-1004.756	-845.097	-899.234
a_3	-4043.512	-4519.586	-5397.504	-5134.564
a_4	-68.628	-73.647	-72.248	-70.403
a_5	-0.022	-0.016	0.030	0.033
a_6	-0.001	-0.018	-0.002	-0.005
a_7	0.347	-0.284	1.140	1.200
a_8	0.009	-0.013	-0.007	-0.004
a_9	2.307	0.815	0.306	0.556
a_{10}	-0.021	-0.008	-0.090	-0.043
a_{11}	-10.913	-80.998	-10.858	-10.918
a_{12}	4184.959	-580.249	-698.940	-207.210
a_{13}	-0.188	0.091	0.007	-0.012
a_{14}	-31.433	-17.417	-11.998	-15.814
a_{15}	0.000	0.000	0.000	0.000
a_{16}	4.075	3.912	-1.964	-16.404
a_{17}	2.361	1.036	0.008	0.019

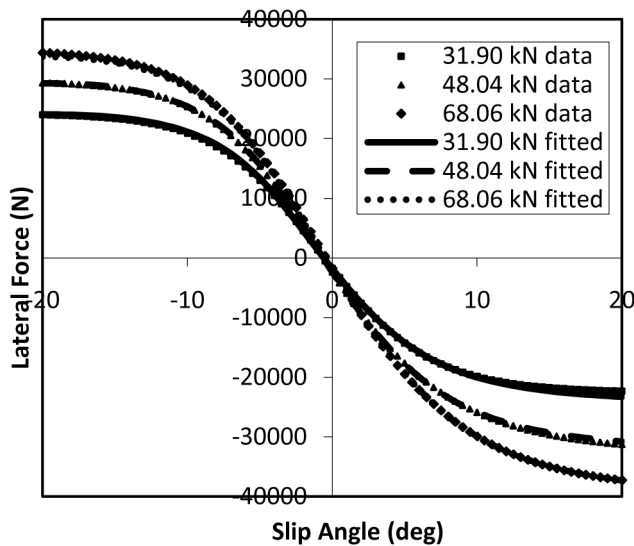


Figure 1. Lateral force vs. slip angle data with MF fit for selected vertical loads at 8 km/h. Test performed on indoor rolling road surface that resembles dry asphalt.

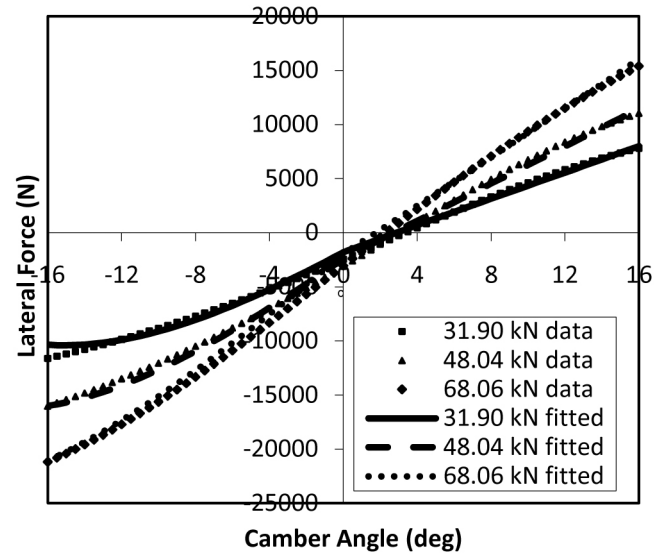


Figure 2. Lateral force vs. camber angle data with MF fit for selected vertical loads at 8 km/h. Test performed on indoor rolling road surface that resembles dry asphalt.

OFF-ROAD TIRE MODEL

An off-road tire model for the studied tire was developed by determining scaling factors for dirt and gravel driving surfaces and then applying them to the on-road tire model. A scaling factor extension to the MF can be found in [7]. The extension places a multiplier (scaling factor) in front of each of the main coefficients (equations (3), (4), (5), (6), (7), (8), (9)) in the MF to account for changes in driving surface. The determination of scaling factors for dry asphalt, wet asphalt, snow, and ice was studied in [4]. However, no studies have been performed to determine scaling factors for the Magic Formula for dirt and gravel driving surfaces. This following presents the procedure for determining these scaling factors.

It was found that the majority of the scaling in lateral force from one surface to another is contained in the peak value and cornering stiffness scaling factors. It was also found that the scaling factors for lateral force are primarily independent of tire type, tire size, vehicle type, and vehicle speed [4]. As a result, the scaling factor extension to the MF used in the development of the off-road tire model is:

$$D = \lambda_D \cdot (a_1 F_z^2 + a_2 F_z) (1 - a_{15} \gamma^2) \quad (10)$$

$$BCD = \lambda_{BCD} \cdot a_3 \sin(2 \arctan(F_z / a_4)) (1 - a_5 |\gamma|) \quad (11)$$

where λ_D is the peak value scaling factor and λ_{BCD} is the cornering stiffness scaling factor. For the off-road tire model, equation (10) replaces equation (4) and equation (11) replaces equation (6) in the MF. To determine values for λ_D and λ_{BCD} ,

a passenger tire was tested on dry asphalt, dirt, and gravel driving surfaces using a portable tire test rig. The rig is capable of applying slip angle sweeps at various vertical loads and forward speeds. All forces and moments acting on the tire are measured by a force and moment measurement hub. The assumption of the scaling factors being independent of tire type, tire size, vehicle type, and vehicle forward speed was used in selecting the passenger tire for this study. This is a convenient assumption, as the studied tire required a room of approximately 40 m² to perform rolling road testing. Therefore, off-road testing of the studied tire using the portable tire test rig was impossible. The passenger tire was tested at five different vertical loads at a forward speed of about 10 km/h. At each vertical load, a 20 degree slip angle sweep with zero degrees camber was applied to the tire and the lateral force response was observed. For each vertical load, peak lateral force and cornering stiffness were extracted, which is shown in Figure 3 and Figure 4. These plots were used with equations (10) and (11) to determine the scaling factors.

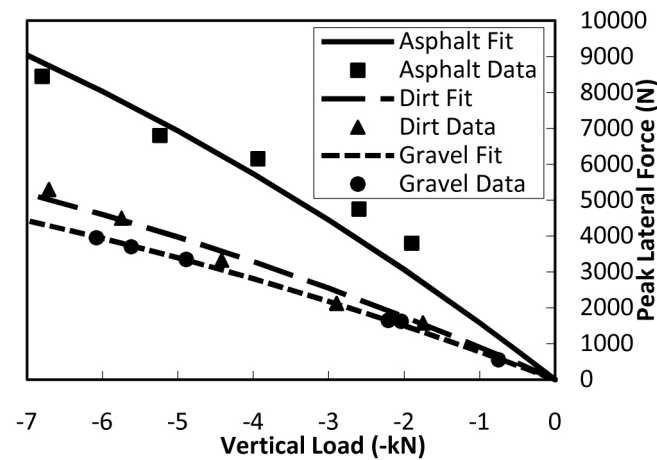


Figure 3. Peak lateral force vs. vertical load for passenger tire on asphalt, dirt, and gravel.

The procedure for determining the scaling factors is as follows. Since a dry asphalt tire model has already been established for the studied tire, it was used as the control surface. As a result, the peak value and cornering stiffness scaling factors were set to unity. The dry asphalt data from Figure 3 was then curve fitted to equation (10) with $\gamma = 0$ to determine values for a_1 and a_2 . The values for a_1 and a_2 were then used with equation (10) to curve fit the dirt and gravel data to get peak value scaling factors. Likewise, the asphalt data from Figure 4 was curve fitted to equation (11) with $\gamma = 0$ to determine values for a_3 and a_4 . The values for a_3 and a_4 were then used with equation (11) to curve fit the dirt and gravel data to get cornering stiffness scaling factors. The scaling factors are presented in Table 2.

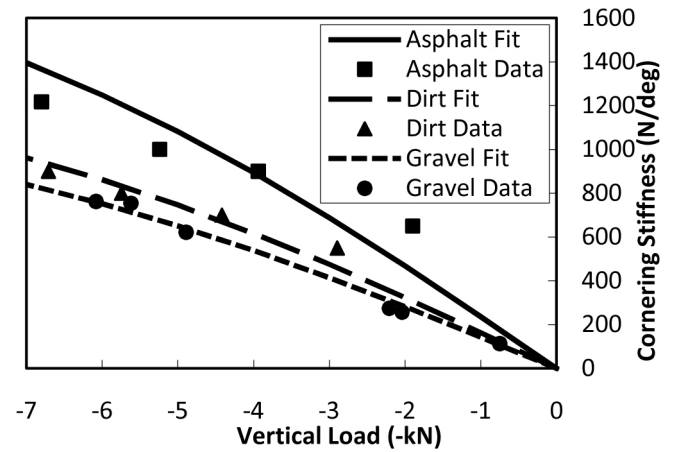


Figure 4. Cornering stiffness vs. vertical load for passenger tire on asphalt, dirt, and gravel.

Table 2. Peak lateral force and cornering stiffness scaling factors

Driving Surface	λ_D	λ_{BCD}
Dry Asphalt	1	1
Dirt	0.573	0.690
Gravel	0.490	0.602

The on- and off-road tire models were then used with an overall vehicle model in vehicle simulation software to simulate asphalt, dirt, and gravel driving surfaces. Since the on- and off-road tire models presented are for a lateral force tire model, the other forces and moments were automatically scaled by the software by inserting the following friction coefficients for the respective surfaces: $\mu_{asphalt} = 0.8$, $\mu_{dirt} = 0.458$, $\mu_{gravel} = 0.392$.

TIRE MODEL ASSESSMENT

Real world sliding data was made available to the authors to compare the simulated vehicle model to. The data gives maximum speed before sliding vs. turning radius for level and banked roads made of asphalt and dirt. The on- and off-road tire models were therefore used with an overall vehicle model in vehicle simulation software to compare the simulated model response to actual vehicle response. The vehicle simulation software contains nonlinear, multiple degree of freedom models for virtually all relevant vehicle components, including tires, suspension, steering, power train, aerodynamic effects, and chassis and body twist and deformation. Exact or near-exact vehicle parameters for vehicle geometry were available for the studied vehicle, which included sprung mass, unsprung mass, vehicle center of gravity (CG) height, distance from front and rear axles to CG, and track width. Many other parameter values were not available to the authors and therefore had to be estimated based on the authors' existing knowledge of similar vehicles and parameters located in the vehicle simulation software for similar