

Article

Determination and Evaluation of Three-Wheeled Tilting Vehicle Prototype Dynamic Characteristics Using Pacejka Tire Model

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Abstract

When a new vehicle is being created or developed, many technical parameters that affect dynamic characteristics must be investigated not only on a theoretical level, but also by natural experiments. Especially one of the most important characteristics for a vehicle that can tilt is tire–road contact, which later helps to calculate and simulate different driving conditions in different driving scenarios, applying internal and external forces. This paper presents a unique construction of a three-wheeled tilting vehicle prototype, tire–road contact determination, and evaluation of vehicle behaviour using the Pacejka tire model. To achieve this, the tire and road surface area were investigated. Using the computed method, experimentally determined contact areas were refined and compared with the actual measured. Determined tire–road contact areas were evaluated by applying dynamic external forces for further investigation. Selected a scenario to predict the behavior of a three-wheeled tilting vehicle and the force distribution during tilting, then determined certain vehicle parameters in the static position (load distribution, tire–road contact areas). The inclusion of asymmetric front-left and front-right tire loads under tilt resulted in observable differences in force distribution. The inner front tire unloaded while the outer tire gained load, introducing asymmetry in both lateral and longitudinal forces. This behaviour was not captured in the symmetric model.

Keywords: three-wheeled tilting vehicle; prototype; determination; tire–road contact; Pacejka tire model; evaluation; dynamic characteristics



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1. Introduction

Since electric vehicles produce no exhaust gases, their use is considered an environmentally friendly mobility option. With initiatives such as the EU's Net Zero Industry Act and the US's Inflation Reduction Act [1,2], a transition from internal combustion engine vehicles to electric vehicles may be inevitable. However, electric vehicles are up to 30% heavier than internal combustion engine vehicles due to the weight of their high voltage batteries [3]. Therefore, advanced tire technology for electric vehicles is needed to counteract the effects of increased vehicle weight, meet performance standards, improve efficiency, and reduce noise [4–8].

Tire–road contact behaviour affects road skid resistance and traffic safety, which has attracted widespread attention [9,10]. Statistics show that hundreds of people die from

traffic accidents every day in the United States, many of which are related to the skid resistance of the roads [11]. In Europe, a quarter of traffic accidents are related to slippery roads [12]. Also, road skid resistance can be a function of road surface and aggregate texture properties [13]. Researchers identified that there is a huge difference between the skid resistance of ribbed and smooth tires, and the International Friction Index model is not appropriate for ribbed tires, which indicates that the skid resistance is also closely related to the tires. It is difficult to truly understand the skid resistance problem by simply analyzing either road or tires [14]. Many researchers pointed out that a proper understanding of the skid resistance can only be obtained by analyzing the tire–road contact behaviour [15,16]. Therefore, it is necessary to clarify the tire–road contact behaviour, which is crucial for road skid resistance and traffic safety.

The tire–road contact area alone is insufficient to accurately represent the full performance of the tire under various models to describe in detail the coupling relationships. To address the limitation presented before, the Magic Formula model was developed to describe in detail the coupling relationships among longitudinal force, lateral force, and self-aligning torque in diverse load and slip conditions [17].

The tire–road contact area and contact pressure are the main parameters influencing various aspects of tire performance characteristics. These two factors play a main role in determining a tire's traction, the wear of its tread, and its capability to resist hydroplaning. The study of a comprehensive mathematical model for the motorcycle tire contact area was performed to explore the impacts of varying inflation pressure and vertical load on contact area and contact pressure [18].

Research on tire–road contact characteristics is important for optimizing tire structural design and has great significance to improve the vehicle's handling stability, driving performance limits, and fuel economy [19–21].

The reliability of previously presented research, experimental procedures, and numerical models is usually validated through combined methods, such as contour scanning tests and free-rolling tests. There were obtained conclusions by analyzing the changes in tire contact outer contour, contact footprint, and pressure distribution under different operating conditions and design parameters [22].

A vertical stiffness test was conducted on a smooth-tread motorcycle tire using a tire testing machine. The obtained results showed a linear increase in contact area with rising vertical load and inflation pressure. Furthermore, this effect was more noticeable at higher loads. The maximum contact pressure showed a linear increase at lower loads; meanwhile, it rose at higher loads. The mathematical model for tire–road contact area, as a function of vertical load and inflation pressure for smooth-tread motorcycle tires, was formulated using a linear regression technique with the 5.72% error and a coefficient of determination ($R^2 = 0.91048$) [23].

The development of the presented models and the use of the obtained results could be one of the steps for the improvement, prediction, and development of more sensitive aquaplaning models [24–26].

Theoretical models and their empirical equations with validation and augmentation by experimental data have a crucial impact on predicting hydroplaning speeds on various tire and road types. The widely accepted prediction model is a combination of Gallaway's equation [27,28] and Huebner's equation [29].

Estimating a tire's contact parameters involves a combination of experimental methods and finite element analysis (FEA) [30–32].

The mechanical behaviour of solid tires was modeled using a hyperelastic constitutive model, and numerical analysis of tire deformation tests was conducted through FEA [33,34].

The tire–road contact behaviour of a passenger car by a pressure distribution measurement system in a laboratory was investigated. The effects of road gradation, vertical force, tire

inclination angle, and tire pressure on the tire–road contact behaviour were analyzed in accordance with the test data. The vertical force on the tire has a significant effect on the contact stress and contact area. Under a vertical force of 4 kN, a typical load borne by a tire of this size on a passenger car, the contact region is approximately circular, and the stress distribution depends on road surface characteristics. The rougher the road, the more stress concentrations. Within the tested range, the tire pressure shows a linear influence on the average contact stress and contact area. The higher the tire pressure, the higher the average stress, the smaller the contact area. As tire pressure increases, the contact region gradually shrinks towards the middle, and the contact force slightly increases in the middle and decreases at the edges [35].

In 2013, a 3D finite element model of soil compaction caused by agricultural tire traffic was developed and validated by González Cueto. The model predicts the effect of inflation pressures, ground pressure, and tire load on the tire–road (soil) contact. The model predicted compaction depth for each tire inflation pressure–load combination and the relationships among inflation pressure, contact stress, load, and soil compaction. It found that vertical stresses transmitted to the soil are independent of water content, while greater compaction in wet soils is due to lower yield stresses. Results agree with experiments and validate the Extended Drucker–Prager model for Rhodic Ferralsol [36].

The contact analysis method is mainly used to evaluate tire deformation during driving, but there are problems, such as the inefficiency of installing many contact sensors inside the tire, or the limited measurement space due to contact sensors installed on the ground. A new analysis method for dynamic tire–road contact evaluation was proposed to overcome the inefficiency and spatial limitations of conventional measurement methods. This method used a 2D image sensor to capture images of the internal deformation upon contact and implemented 3D measurement. Then, the marker with a known size and location was introduced, and the deformation in the Z-axis direction was calculated by precalculating the calibration sheet for the working distance and field of view according to the tire deformation. The deformation of the tire was expressed separately during the rotational movement so that the deformation in the radial and tangential directions could be expressed. To verify the proposed measurement method, the rationality of the measured results was examined using the relationships between dynamic deformation, rolling resistance, and vehicle distance. In conclusion, the proposed analysis method allowed 3D measurement and quantification of dynamic tire deformation [37].

Summarizing the determination of the tire–road contact forces is a necessary first step for new prototype construction, various control systems, improved vehicle safety and performance parameters, but measurement systems are expensive and, due to their specifics, have a significant difference between theoretical models and real experimental data [38]. That is why there should be a compromise between unified theoretical methods and the validation of experimental data.

2. Three-Wheeled Tilting Vehicle Description

Tilting three-wheeled vehicles and their prototypes have been researched and developed since the beginning of the 1950s [39]: Ford Motor Company presented a gyroscopically stabilized two-wheeled tilting vehicle “Gyron”; MIT proposed a tilting vehicle, equipped with an active tilt control; General Motors—“Lean Machine” [40]. Later, different companies developed and presented three-wheeled tilting vehicles as F 300 Life-Jet [41], Clever [42], Carver One [43], and MP3 [44]. Some of them concentrated on similar-sized but four-wheeled vehicles, with other technical solutions, such as those of Triggo [45]. In this decade, one of the most popular examples mentioned is Deliverator [46], created by Arcimoto or Toyota i-ROAD [47].

The development of this type of vehicle has become increasingly relevant as issues such as pollution (especially in cities), energy consumption, vehicle cost, and traffic congestion

demand urgent attention. Addressing these factors now and in the future is essential to transform cities and support sustainable societal growth [48,49].

The main goal of the automotive industry is to create a more efficient, safer, and greener transportation system [50,51].

There are not many studies that provide detailed research on three-wheeled vehicles and the determination of their dynamic characteristics.

Electric vehicles have received a lot of attention recently due to being classified as one of the zero-emission solvers; these vehicles additionally have a higher energy efficiency than comparable internal combustion vehicles. It is well known that driving parameters such as rapid acceleration affect fuel economy [52,53], and that changes in driving behaviour can influence it [54].

Numerous studies have been conducted to investigate the acceleration and deceleration of traffic flow consisting of different types of vehicles, as well as separate models for the analysis of individual systems [55–60]. However, in order to optimize traffic flow, the range of vehicle types is being expanded in parallel, which is not only environmentally friendly but also has good dynamic characteristics, without compromising the safety of passengers and pedestrians.

Dynamic characteristics are one of the goals that manufacturers of new vehicles compete for. As a result, there are many different theoretical and experimental methods for determining and evaluating these characteristics [56,57]. These methods are, on the one hand, uniform, but, on the other hand, they are also evolving. In line with this approach, every vehicle designer and manufacturer strives to find the best methods of determination and verification, apart from the technical requirements set out in SAE technical documents. If implemented, three-wheeled vehicles could become a safe alternative to four-wheeled vehicles [58].

Zhang et al. presented an optimal control of a hybrid electric vehicle where the initial task was to determine accurate vehicle acceleration, which can later be adjusted to optimize performance, driver and passenger comfort, and integration into the overall dynamic system for safe driving [59].

Previous studies show that rationally designed energy optimization strategies can optimize the operating points of key power components, thereby increasing energy efficiency when driving vehicles [60–62]. That is why all vehicles should be precisely tested in each phase of the project, especially when we analyze safety and driving properties.

In summary, three-wheeled vehicles, like others, must undergo a long period of refinement. This requires many optimization processes, such as determining the optimal dynamic characteristics of the vehicle. In addition, each step must be verified by experimental testing. Therefore, this article presents a unified theoretical and experimental methodology and the results obtained with its help.

Lithuanian electric vehicle startup AKO introduced a new type of three-wheeled tilting vehicle prototype in 2020 (Figure 1) [63].



Figure 1. AKO trike P150 [63].

This three-wheeled tilting vehicle prototype has two seats and is fully electric. It is a second-generation prototype and features a novel tilting mechanism. This vehicle is

unique due to its simplicity and driving logic, which consists of two independent control channels. The first channel is two front wheel control, like in a usual vehicle, where turning the steering wheel turns the two front wheels. This is sufficient for slow-speed turns. The second channel is a solution used at higher speeds, when the driver can tilt the steering column from side to side, which causes the entire AKO trike to tilt to each side by 30° , like any other two-wheel vehicle [63]. Due to these innovative solutions, this three-wheeled vehicle prototype has various driving scenarios, including motorcycle and classical four-wheel automobile understanding.

In the initial stage, we chose a scenario to predict the behavior of a three-wheeled tilting vehicle and force distribution under tilt; then, we determined certain vehicle parameters in the static position (load distribution, tire–road contact areas).

A previous study [64] presented the dynamic characteristics of this prototype, but, in order to fully evaluate this type of vehicle, parameters such as tire–road interaction, dynamic force behavior, and balance under dynamic conditions need to be investigated in more detail.

2.1. Determination of Area of Tire–Road Contact

The goal of the experiment was to measure tire–road contact areas with the three-wheeled vehicle in a fully vertical position. The tilting three-wheeled prototype used in the experiment is unique for several reasons: (1) it can tilt, so contact patches vary not only with tire geometry, tire pressure, and load but also with the vehicle's tilt angle; and (2) the front and rear axles use different tires, resulting in different contact areas at various tilt angles due to the vehicle's mass distribution.

The experiment began by preparing the three-wheeled tilting prototype with the test tires. Cold inflation pressures were 2.5 bar for the front tires and 2.7 bar for the rear tire. A special ink was applied to each tire tread as a marking agent. Each wheel was lifted in turn, the inked tire was carefully lowered onto a white sheet of paper on a flat surface, and the ink produced a precise imprint of the tire's contact patch. This stamping was repeated three times for each tire, and each imprint served as a raw data point representing a unique contact area.

Table 1 presents the contact area measurements for the front axle tires (MITAS 27.0 $\times$ 7.0^{−19}, purchased in Lithuania) and the rear axle tire (Bridgestone Exedra 200/50 ZR 17, purchased in Lithuania), along with the results of digital image analysis. These initial-stage results provide the tire–road contact areas for each wheel when the three-wheeled tilting prototype is in the fully vertical position. For further investigation, these data will be processed and used in the Pacejka tire model.

Table 1. Measured and after digital image analysis of tire–road contact areas.

	Front Left			Front Right			Rear		
	1	2	3	1	2	3	1	2	3
Measured									
Print area, cm ²	23.064	20.451	22.036	15.839	19.426	24.22	56.898	62.142	68.187
After digital image analysis									
Print area, cm ²	28.317	28.034	28.5	29.04	28.544	29.48	73.924	74.351	73.497

2.2. Clarification of the Determined Area of Tire–Road

The research methodology combines practical experiments and digital image analysis to accurately measure the tire's contact patch area under various conditions.

Once the data are collected, each paper is scanned using a monochrome scanner with a resolution of 300 DPI (dots per inch), where it is digitized into a digital image. Since a monochrome scanner scans an image directly and stores it as a binary image of black and white, the bitmap image format is inherently able to classify the contact area and background without any need for thresholding or conversion of the grayscale image.

This digital image is then input into an image processing algorithm implemented using a computer package called Octave. Since a monochrome scanner is able to directly scan an image and represent it as a binary image, where each pixel is represented as either 0 (background) or 1 (contact area), Octave is able to directly count the number of pixels with a value of 1, i.e., measure the contact area.

By using the known resolution of 300 DPI, the script can convert the pixels into a known area. Each pixel represents a known area of $(25.4 \text{ mm}/300)^2$, and, therefore, the size of the contact area can be calculated with precision. This calculated area is stored for each image, and, therefore, a complete set of results is available for all images.

The quality of segmentation is dependent upon the inherent settings of the monochrome scanner because it directly scans a clean binary image based on optical contrast between ink stamps and paper background.

The resulting data set provides a detailed estimate of the tire's contact area over several tests (Table 1).

These data can undergo further statistical analyses, such as average contact area (Table 2), trial differences, and contact area versus any experimental variables (e.g., tilt angle, vehicle load).

Table 2. Tire–road contact areas after digital image analysis.

Wheel	Front left
Avg. print area, cm ²	28.284
Wheel	Front right
Avg. print area, cm ²	29.021
Wheel	Rear
Avg. print area, cm ²	73.924

For a more accurate evaluation of mass distribution, a weighted three-wheeled tilting vehicle prototype (scales DINI ARGEO WWS) was weighed, and this mass distribution is presented in Table 3.

Table 3. AKO trike P150 mass distribution.

	Mass, kg			
	Front Left	Front Right	Rear	Total
1	144.9	138.7	216.4	500
2	145.9	137.2	216.9	500
3	145.1	138.3	216.7	500
Average	145.3	138.1	216.7	500

The mass distribution of the three-wheeled prototype among the wheels was measured in the fully vertical position with the driver aboard, using the same procedure as for determining the tire–road contact areas.

When analysing the results (Table 3), it is noticeable that the front axle distributed 56.7% of the total mass and 43.3% to the rear wheel of the total mass. A left and right mass distribution difference could be argued as an additional component of non-even distribution by location.

Also, the additional parameters of the three-wheeled vehicle prototype were used as mass with a driver of 500 kg, a wheelbase of 2.33 m, and a track width of 0.892 m.

3. Evaluation of the Tire–Road Contact

Typically, a static stability coefficient is used to assess the stability of a vehicle [65]:

$$SSF = T/(2H) \quad (1)$$

where T is the “track width” of the vehicle, and H is the “height of the center of gravity” of the vehicle.

The track width is the distance between the centers of the right and left tires along the axle. In our case, it is:

$$SSF = 0.892/(2 \cdot 0.55) = 0.811 \quad (2)$$

When different scenarios of vehicles (especially tilting, e.g., motorcycles, or other tilting vehicles, such as the AKO trike P150, manufactured by JSC “Rimti Ratai”, Rietavas, Lithuania) in dynamic conditions and vehicle positions are analyzed, a group of parameters occurs that changes over time and has complex parameters. This group of parameters consists of SSF , track width, and the height of the center of gravity, as well as the driving speed, tire–road contact area, gravity center force to this area, centrifugal force, tilt angle of vehicle, vectors position of forces, etc.

To determine and evaluate the three-wheeled vehicle prototype’s dynamic characteristics more accurately, we used the Pacejka model. The Pacejka tire model (also known as the Magic Formula or Pacejka Magic Formula Tire Model) is an empirical mathematical formula developed to describe the behaviour of tire forces depending on the slip angle, slip ratio, load, and other factors. It is widely used in automotive dynamics, especially in vehicle simulations, race car analysis, stability control systems, etc. [17]:

$$Y = D \cdot \sin(C \cdot \arctan(Bx - E(Bx - \arctan(Bx)))) \quad (3)$$

where Y —Output force or moment (lateral force, longitudinal force); x —Input (slip angle α or slip ratio κ); B —Stiffness factor—controls slope at origin; C —Shape factor—controls curve shape; D —Peak factor—maximum value (peak force); E —Curvature factor—controls asymmetry near peak.

Following the main Formula (3), it could be divided for the needed interpretation as follows:

- Longitudinal Force (Braking/Acceleration) [17]:

$$F_x = D_x \sin(C_x \arctan(B_x \kappa - E_x(B_x \kappa - \arctan(B_x \kappa)))) \quad (4)$$

- Lateral Force (Cornering) [17]:

$$F_y = D_y \sin(C_y \arctan(B_y \alpha - E_y(B_y \alpha - \arctan(B_y \alpha)))) \quad (5)$$

The Pacejka tire model was used according to the tire parameters used on the three-wheeled prototype (Table 4). Real Pacejka datasets are heavily protected intellectual

property of the tire manufacturers. This type of tire model data is not typically released by tire manufacturers and must generally be obtained through testing (e.g., flat-trac or drum tire test rig data) and curve fitting. Pacejka tire parameters were chosen according to the same size, similar tire parameters found in previous experimental datasets with real tire experiments, and used in different program libraries referring to vehicle dynamics software, based on the Modelica Multibody, VehicleDynamics, and BikeSim libraries, adapted to the three-wheeled vehicle prototype conception. Additional criteria and steps were used to ensure the chosen dataset was as representative as possible: (1) tire construction and type: matched radial vs. bias and intended application (motorcycle/scooter vs. car); (2) tread pattern and compound: selected datasets with similar tread geometry and usage (street/sport/touring); (3) load and inflation ranges: ensured the experimental data covered similar vertical loads and inflation pressures as our tests or used data that could be interpolated; (4) aspect ratio and rim diameter: matched these geometric parameters to reduce scaling errors.

Table 4. Front and rear tire parameters for Pacejka tire model.

Rear Tire—Bridgestone Exedra 200/50 ZR 17				
Longitudinal Parameters			Lateral Parameters	
1.	Longitudinal stiffness factor	$B_x = 11.5$	Lateral stiffness factor	$B_y = 9.8$
2.	Longitudinal shape factor	$C_x = 1.3$	Lateral shape factor	$C_y = 1.2$
3.	Peak longitudinal force (N)	$D_x = 3000$	Peak lateral force (N)	$D_y = 2500$
4.	Longitudinal curvature factor	$E_x = 0.85$	Lateral curvature factor	$E_y = 0.92$
Front Tires—MITAS 27.0 × 7.0 ^{−19}				
Longitudinal Parameters			Lateral Parameters	
1.	Longitudinal stiffness factor	$B_x = 10.0$	Lateral stiffness factor	$B_y = 8.5$
2.	Longitudinal shape factor	$C_x = 1.1$	Lateral shape factor	$C_y = 1.0$
3.	Peak longitudinal force (N)	$D_x = 2500$	Peak lateral force (N)	$D_y = 1800$
4.	Longitudinal curvature factor	$E_x = 0.75$	Lateral curvature factor	$E_y = 0.8$

The tire characteristics were determined by varying the slip angle α from -20° to $+20^\circ$ in order to capture the linear turn stiffness regime, the transition to saturation, and the behaviour after the peak. Longitudinal slip was set to $\kappa = 0.05$ to reflect light steering/braking slip. Static normal loads were used on a vehicle mass of $m = 500$ kg with distribution between each wheel (Table 3). The measured tire–road contact areas (Table 2) were used to scale the Pacejka D parameter through contact surface normalization. The load sensitivity of D was represented by a simple proportional scaling with F_z to obtain the maximum force values characteristic of the vehicle. The lateral forces F_y were calculated according to Pacejka’s lateral formula as functions of α ; the longitudinal forces F_x were calculated according to Pacejka’s longitudinal formula with $\kappa = 0.05$ and plotted against α (with conservative combined slip modulation).

The lateral force curves correspond to the canonical magic formula: an almost linear response at low $|\alpha|$, a clear peak, and a decrease after the peak. The simulated lateral forces for each wheel are reached at the limit position of this test and are as follows: front left wheel peak $F_y = -2356.5$ N at $\alpha = -20.00^\circ$, front right wheel peak $F_y = -2296.6$ N at $\alpha = -20.00^\circ$, and rear wheel peak $F_y = -1358.1$ N at $\alpha = -20.00^\circ$ (Figure 2a). Therefore, based on the static load assumption, the front wheels show significantly higher lateral capacity than the rear wheel. The slight Front left/Front right asymmetry (Front left ≈ -2356.5 N versus front rear ≈ -2296.6 N) corresponds to a slightly smaller Front left contact area (28.284 cm²), which creates higher local contact pressure and, according to the accepted D -scaling heuristic, a slightly higher maximum force. The longitudinal capacities (when

$\kappa = 0.05$) are: front left wheel longitudinal force (F_x) peak 1805.15 N, front right wheel longitudinal force (F_x) peak $F_x = 1759.31$ N, and rear wheel longitudinal force (F_x) peak 1050.67 N (Figure 2b).

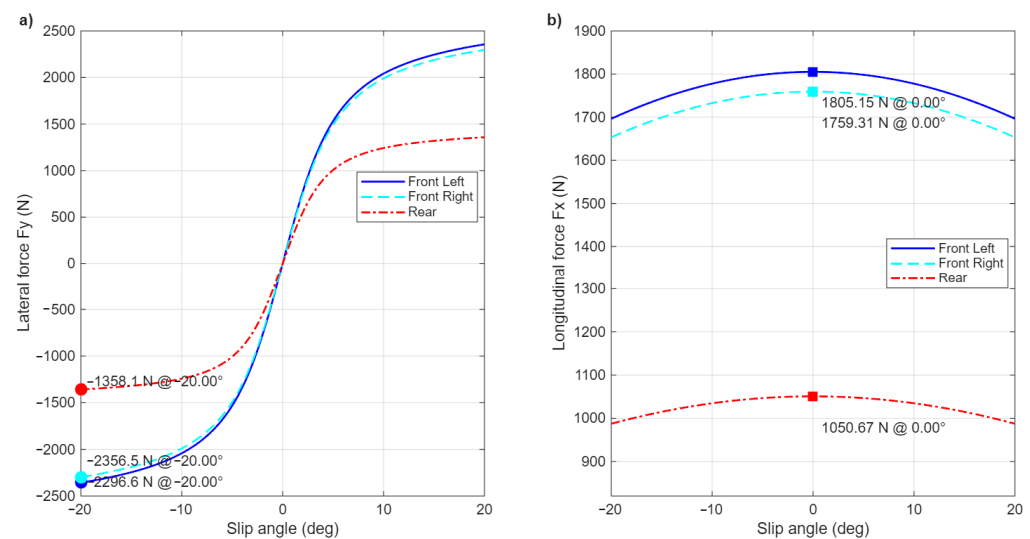


Figure 2. Results of Pacejka tire model: (a) lateral force vs. slip angle; (b) longitudinal force vs. slip angle.

When static loading is selected, and the contact area/load sensitivity scale is applied, the front axle (two wheels together) provides significantly greater turning and longitudinal capacity than a single rear wheel, and due to differences in contact area, a visible asymmetry between the left and right sides is observed. The observed trends—linear low-angle stiffness followed by saturation and reduced force after the peak—correspond to the established tire behaviour. Quantitatively, the above-mentioned maximum F_y and F_x values per wheel should be interpreted as model results related to specific Pacejka base parameters, D scale heuristics, and static load assumptions. In order to predict vehicle handling, these results should be supplemented by (a) calibrated load sensitivity exponent and reference load from tire test data, (b) full implementation of Pacejka's combined slip ($F_x(\kappa, \alpha)$), $F_y(\kappa, \alpha)$), and (c) dynamic load transfer simulation (yaw, roll) to capture temporary axle load redistribution.

$$D_{\text{scaled}} = D_0 \cdot (F_z/F_{z_{\text{ref}}})^n \cdot (A_{\text{average}}/A) \quad (6)$$

where: D_{scaled} —scaled Pacejka D parameter used in the magic formula; D_0 —baseline Pacejka D (D_{x0} or D_{y0}) at reference conditions; F_z —wheel normal load (N); $F_{z_{\text{ref}}}$ —reference normal load (mean per-wheel static load), (N); n —load-sensitivity exponent; A_{average} —mean contact patch area of all wheels (m^2); A —contact patch area of the wheel (m^2).

Modifying the presented model and including the load sensitivity scale (Formula (6)), for a three-wheeled vehicle prototype ($m = 500$ kg, front left wheel $F_z \approx 1425$ N, front right wheel $F_z \approx 1355$ N, rear wheel $F_z \approx 2126$ N), the scenario chosen of $F_{z_{\text{ref}}} \approx 1600$ N and $n = 0.85$. By including these parameters, a more physically realistic, sublinear increase in tire forces under normal loading (e.g., a 10% F_z increase causes an $\approx 8.5\%$ increase in D), measurable left–right differences are introduced through contact area normalization, and force overestimation under high load is avoided.

In practice, this improves the quantitative estimates of F_y and F_x peaks, vehicle handling balance, and sensitivity to load changes while maintaining a simple, calibratable tuning parameter (n).

The simulation using the load-sensitive Pacejka model (Formula (6)) gave results presented in Figure 3, where the lateral force curves show near-linear stiffness at small slip

angles followed by saturation, with per-wheel peak lateral forces of front left wheel peak $F_y = -2038.5$ N, front right wheel peak $F_y = -2091.6$ N, and rear wheel peak $F_y = -1727.8$ N at $\alpha = -20.00^\circ$ (Figure 3a).

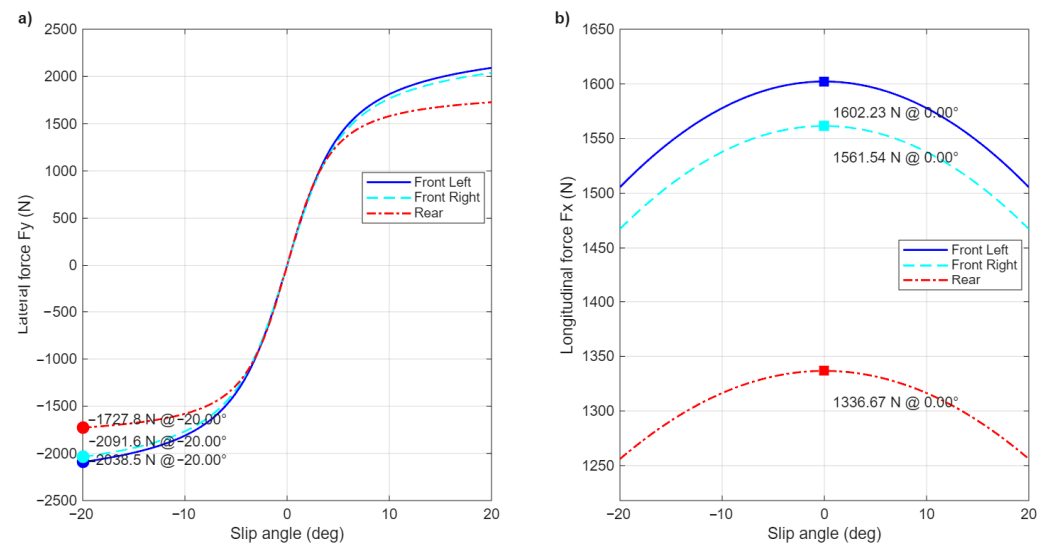


Figure 3. Results of Pacejka tire model with load sensitivity: (a) lateral force vs. slip angle; (b) longitudinal force vs. slip angle.

Longitudinal capacity at $\kappa = 0.05$ (plotted versus α) is front left wheel lateral force peak $F_x = 1602.23$ N, front right wheel peak $F_x = 1561.54$ N, and rear wheel peak $F_x = 1336.67$ N (Figure 3b). These results indicate a front-dominant traction and cornering capability (front axle total to rear wheel), and a modest left–right asymmetry (front left wheel $\approx 2.6\%$ larger than front right wheel) attributable to the slightly smaller front left tire–road contact area and the contact-area/load-sensitivity scaling. The inclusion of load sensitivity (n , $F_{z_{ref}}$) prevents linear over-prediction of forces with F_z , yielding more realistic, sublinear force growth and improved fidelity for handling and stability assessments; however, for quantitative combined braking/turning predictions, the model should be extended to a full combined-slip formulation and calibrated to experimental tire data.

4. Results and Discussion

The determination and evaluation of the interaction between the tires and the road of the presented three-wheeled tilting vehicle prototype consists of several important stages, starting with the distribution of the vehicle’s weight between the wheels, including the determination of the interaction between the tires and the road, and, finally, using the Pacejka tire model, determining the impact and trends of all measured and investigated parameters on driving results.

The presented simulations compare lateral and longitudinal tire capacities for a three-wheeled prototype vehicle using a Pacejka magic-formula implementation augmented by contact-area normalization and a load-sensitivity scale.

The lateral curves (α swept from -20° to $+20^\circ$) reproduce the expected Magic Formula behavior: near-linear cornering stiffness at small $|\alpha|$, a clear peak, and a post-peak decline.

In conclusion, introducing the load-sensitive scaling substantially alters the per-wheel force predictions: Front left wheel lateral force (F_y) peaks changes from -2356.5 N to -2038.5 N (-318.0 N, -13.5%); front right wheel lateral force (F_y) peak from -2296.6 N to -2091.6 N (-205.0 N, -8.9%); rear wheel lateral force (F_y) from -1358.1 N to -1727.8 N ($+369.7$ N, $+27.2\%$).

The longitudinal peaks at $\kappa = 0.05$ show the same tendency: front left wheel (F_x) peak changes from 1805.15 N to 1602.23 N (−202.92 N, −11.3%); front right wheel (F_x) peak changes from 1759.31 N to 1561.54 N (−197.77 N, −11.2%); rear wheel (F_x) peak changes from 1050.67 N to 1336.67 N (+286.0 N, +27.2%). Overall, the load-sensitive model reduces predicted front wheel forces, increases rear wheel forces, and produces a more balanced axle contribution, reflecting physically plausible sublinear force growth with F_z .

Including load-sensitivity reduces predicted forces for front wheels (~8–14% lower F_y and ~11% lower F_x per front wheel). Raises predicted rear forces substantially (~+27% for both F_y and F_x) because rear wheel normal load is higher in the three-wheel prototype, and D is scaled with F_z . Lowers overall front dominance (front axle vs rear wheel) by ~30–35% in ratio terms—model moves toward more balanced axle contributions. Slight left/right asymmetry remains (Front left > front right), but the magnitude is modest (~2–3% in scaled results) and explained by contact area normalization.

5. Conclusions and Future Research

Determination of tire–road contact area in the vertical static position shows only certain three-wheeled vehicle prototype use conditions (parked vehicle, steady load, ambient temperature), and it is only a starting point. For deeper research, it should be investigated under dynamic conditions. During dynamic conditions, tire behaviour changes because load, temperature, and deformation vary continuously. As the tire heats up, the rubber softens, increasing compliance and improving grip, even if the overall contact area changes only slightly. Dynamic loading from braking, acceleration, and cornering causes the contact patch to distort and become asymmetric, with uneven pressure distribution. Because rubber is viscoelastic, the tire responds differently at high deformation rates, often behaving stiffer than in static tests.

To enhance the accuracy and applicability of the current three-wheeled tilting vehicle prototype tire model, several research extensions can be considered, such as the incorporation of suspension and roll stiffness to simulate transient load redistribution during acceleration, braking, and cornering, rather than relying on quasi-static tilt approximations. Extension of the model to compute true tire slip angles and longitudinal slip from vehicle kinematics (yaw rate, steering input, and wheel velocities) enabled dynamic simulation under realistic driving manoeuvres.

In addition, additional external factors such as temperature and road surface effects should be included. In order to consider the effects of the environment and materials, the dependence of tire parameters (B , C , D , E) on temperature, wear, and road friction coefficient has been described in detail.

The results of this and future research should combine the extended tire model with a full three-wheeled tilting vehicle prototype dynamics model to evaluate rollover risk, understeer/oversteer tendencies, and lateral stability margins for three-wheeled vehicles. Finally, model validation should be preceded by performing physical testing to calibrate and validate the Pacejka parameters under various loads, tilts, and road curvatures, improving model fidelity for predictive simulations.

Implementing these improvements would enable the MATLAB (R2025b) model to predict more realistic vehicle behaviour, support stability and safety assessments, and serve as a robust tool for three-wheeled tilting vehicle prototype design and optimization.

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