Development and Validation of a 12-DOF Vehicle Model for Ride and Handling Analysis for Three-Wheeled Vehicle

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***Abstract*—An accurate vehicle model is essential for effectively representing vehicle behaviour, particularly in the study of ride and handling dynamics. This work focuses on developing a comprehensive vehicle model to analyse vehicle behaviour in various driving conditions. A 12-Degrees-Of-Freedom (DOF) vehicle model is derived, incorporating ride, handling, and tire dynamics. Two types of tire models—Linear and Nonlinear (Magic Formula)—are implemented in Simulink, and their performance is evaluated by comparing simulation results with ADAMS outputs. The tire model that best aligns with the ADAMS results is integrated into the 12-DOF vehicle model. All assumptions considered in the model development are detailed. The proposed vehicle model is validated using an instrumented vehicle under different steering inputs. The deviations between simulated and experimental results, particularly in yaw rate, lateral acceleration, roll angle, and individual tire slip angles, are analysed and discussed.**

Keywords - Vehicle dynamics; Multibody simulation; Three wheeler; Constat radius cornering.

I. INTRODUCTION

A three-wheeled vehicle features a single front wheel, similar to a two-wheeler, and two rear wheels, resembling a four-wheeler. This unique configuration combines the advantages of both vehicle types, offering compactness and enhanced maneuverability in congested traffic and narrow roads. However, this design also introduces certain challenges in terms of stability and dynamic performance.

One of the primary concerns with three-wheeled vehicles is their inherently lower rollover stability compared to four-wheeled vehicles due to their asymmetric weight distribution and reduced lateral support [1][2]. The current design employs a trailing arm suspension at the rear, which maintains a fixed roll axis at ground level. Since there is no variation in camber or toe during wheel travel, the roll axis remains significantly lower than the Vehicle's Center of Gravity (CG). This results in a high roll moment, making the vehicle more susceptible to lateral instability and rollover, especially during sharp turns or evasive maneuvers [3].

Furthermore, the absence of an independent suspension system in most three-wheeled vehicles limits their ability to adapt to uneven road surfaces, affecting ride comfort and handling characteristics. The distribution of roll stiffness between the front and rear also plays a crucial role in the vehicle's dynamic behaviour, influencing parameters, such as understeer, oversteer, and load transfer. These factors must be carefully analysed to optimize the stability and safety of three-wheeled vehicles under various driving conditions.

The following sections of this paper are structured as follows: Section II presents the analytical formulation of a 12-Degrees-Of-Freedom (DOF) three-wheeled vehicle model, including the derivation of roll dynamics and an analytical representation of the tire model. Section III introduces a multibody dynamic model, incorporating flexible body dynamics for enhanced fidelity, and discusses the simulation of a step-steer maneuver and Constant Radius Cornering (CRC) simulation, with corresponding results plotted. Section IV details the experimental validation, where a physical prototype instrumented with sensors is used to measure key vehicle dynamics parameters, and a comparative analysis between experimental and simulation results is performed, with correlation graphs presented. Finally, Section V outlines the future work, highlighting planned improvements, further analysis, and potential extensions to refine the proposed models.

II. SCOPE OF THE WORK

Existing research papers primarily emphasize analytical expressions for evaluating vehicle dynamics, often neglecting the critical correlation between physical and virtual simulation results. Analytical models, due to their inherent simplifications, frequently yield results that underestimate experimental findings. In contrast, this study integrates three fundamental approaches: analytical modelling, 3D virtual simulation, and instrumented experimental testing. By incorporating all three dimensions, a stronger correlation between virtual and physical results is achieved, leading to more accurate vehicle dynamics assessments.

The developed virtual Multi-Body Dynamics (MBD) model includes real-time flexible body integration, replicating physical vehicle components with higher fidelity compared to conventional 2D analytical models. This enhancement significantly improves result accuracy, ensuring that the simulation model aligns closely with real-world vehicle behavior.

The high-fidelity MBD model considers key structural flexibilities, including suspension components, such as trailing arms, control arms, steering system elasticity, such as steering column compliance, axle housing deformation, and body-in-white stiffness properties. Virtual simulations account for real-world material properties by iteratively refining stiffness and damping parameters based on actual test data. A scaling factor is applied to material properties to ensure consistency with physical test results, improving the predictive capabilities of the model.

The primary goal of this work is to establish a complete 3D virtual simulation framework for comprehensive vehicle dynamics testing. This enables early-stage vehicle dynamics target setting, with parameter cascading down to the subsystem level, ultimately accelerating product development cycles. By reducing dependency on physical prototypes, both time and cost constraints associated with mule vehicle manufacturing are significantly minimized. Initial validation of the simulation model against physical test results provides confidence in early-stage vehicle performance assessment.

To enhance the reliability of virtual simulations, key assumptions are incorporated, including equivalent stiffness values for springs, dampers, and bushings, as well as realistic flexibility properties of major structural components. Material properties are iteratively refined using real-world data, ensuring an accurate representation of actual vehicle dynamics.

In this study, Step steer manoeuvring and Constant Radius Cornering (CRC) tests have been utilized for validation. However, additional dynamic test scenarios, such as fishhook, free steer, slalom, and ramp steer tests along with dynamics control systems are planned as future work to further reinforce simulation accuracy and reliability. Expanding the range of test cases will enhance confidence in virtual simulations, reducing dependency on physical testing while ensuring robust vehicle handling performance predictions.

III. Three wheeler vehicle model

The vehicle dynamic model for a three-wheeled vehicle is developed as a nonlinear system with 12-Degrees-Of-Freedom (DOF). This model consists of both sprung and unsprung masses, with the vehicle body having six DOF: translational motions along the x, y, and z axes, and rotational motions (roll, pitch, and yaw) about these axes. Specifically, the roll, pitch, and yaw motions represent the rotations about the x, y, and z axes, respectively. Each wheel is modeled with translational motion in the vertical (z) direction and wheel spins about the y-axis. Additionally, the front wheel is capable of steering about the z-axis, which contributes to the vehicle's overall maneuverability.

In this study, the dynamic model of a typical three-wheeled passenger vehicle is developed, as shown in Figure 1. The model is constructed using Lagrangian mechanics, where the Equations Of Motion (EOM) are derived to describe the complex interactions between the sprung and unsprung masses. The coordinate system for the vehicle follows the Society of Automotive Engineers (SAE) International convention, and the relevant sign conventions are shown in Figure 2. The developed model incorporates both vehicle dynamics and the interaction between the vehicle body and the wheel assemblies, enabling a detailed analysis of the ride and handling characteristics of three-wheeled vehicles.

A. *Equations of Motion*

Governing equations of the Longitudinal, Lateral, and Vertical, Roll, Pitch and Yaw motions can be expressed as [2]:

Equation of motion for longitudinal motion

$$M_t\ddot{x} = Fx_{,f} + Fx_{,rl} + Fx_{,rr}$$

$$M_t\left(\dot{v_x} + \dot{\theta}v_z - v_y\dot{\psi}\right) = Fx_{,f} + Fx_{,rl} + Fx_{,rr} \quad (1)$$

Equation of motion for lateral motion

$$M_t\ddot{y} = Fy_{,f} + Fy_{,rl} + Fy_{,rr}$$

$$M_t\left(\dot{v_y} + v_x\dot{\psi} - \dot{\theta}v_z\right) = Fy_{,r} + Fy_{,rl} + Fy_{,rr} \quad (2)$$

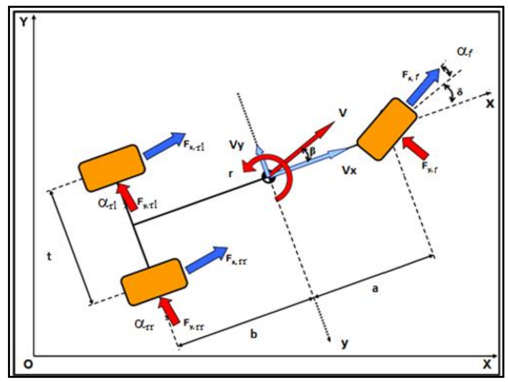

Figure 1. Six DOF Horizontal vehicle model.

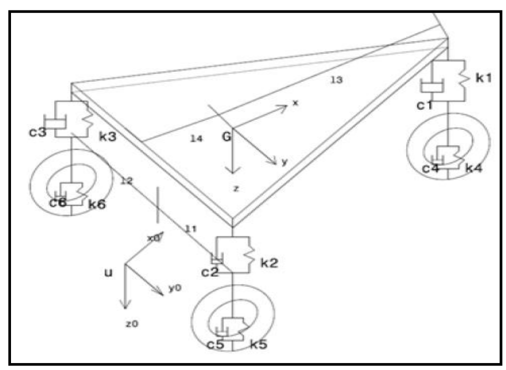

Figure 2. Six DOF Vertical vehicle model.

Equation of motion for sprung mass vertical motion

$$M_s\ddot{z} = Fz_{,f} + Fz_{,rl} + Fz_{,rr}$$

$$M_s\left(\dot{v_z} + v_x\dot{\psi} - \dot{\theta}v_z\right) = Fz_{,r} + Fz_{,rl} + Fz_{,rr} \quad (3)$$

Equation of motion for sprung mass roll motion

$$M_x = I_{sxx}\ddot{\phi} - \left(I_{syy} - I_{szz}\right)\dot{\theta}\dot{\psi} = Fz_{,rl} - Fz_{,rr} \quad (4)$$

Equation of motion for sprung mass pitch motion

$$M_y = I_{syy}\ddot{\theta} - (I_{szz} - I_{sxx})\dot{\theta}\dot{\psi} =$$

$$(Fz_{,rl} + Fz_{,rr})b - Fz_{,f}\,a - ((Fx_{,f} + Fx_{,rl} + Fx_{,rr}) \quad (5)$$

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Equation of motion for sprung mass yaw motion

$$M_z = I_{zz}\ddot{\psi} - (I_{xx} - I_{yy})\dot{\phi}\dot{\theta} = (Fx_{,rl} - Fx_{,rr})t/2 - (Fy_{,rl} + Fy_{,rr})b + (Fy_{,f}\, a) \quad (6)$$

where:

a = Length between the CG and front tire patch
V = Vehicle velocity vector
b = Length between the CG and rear tire patch
V_f = Front tire velocity vector
δ= Steer angle
V_r = Rear tire velocity vector
F_Y= Tire lateral forces
V_x = Vehicle velocity in the x-axis
Ψ= Yaw angle
V_y = Vehicle velocity in the y-axis
I_{zz} Vertical axis moment inertia
a_x, a_y=Longitudinal and Lateral acceleration

The horizontal vehicle model receives lateral and longitudinal forces from the tire model, which are crucial inputs for the vertical dynamics of the vehicle. Based on these forces, a 6 Degrees-Of-Freedom (DOF) vertical vehicle model is developed, as illustrated in Figures 1 and 2. The model incorporates a two-dimensional vertical dynamic system, representing a half-track vehicle model for pitch dynamics and a two-track half-vehicle model for roll dynamics. This approach allows for three DOF associated with the vehicle's mass center (vertical, roll, and pitch dynamics), along with three DOF for each wheel, focusing on the vertical dynamics of the wheels.

The primary dynamics analysed in this study are the yaw and roll motions, which play a crucial role in vehicle stability and handling. The yaw motion is essential for understanding the vehicle's directional control, while roll dynamics influence the lateral stability, particularly in cornering and evasive maneuvers [2]. These factors are critical for evaluating the overall ride and handling characteristics of the vehicle.

B. Roll Model

The roll equations are derived by decomposing the vehicle into sprung and unsprung masses within the y-z plane, as illustrated in Figures 3 and 4. Newton's Second Law, formulated for rigid body dynamics, is applied to analyze the roll motion. In this context, "inside" and "outside" refer to the respective sides of the vehicle relative to the direction of a turn.

The steady state roll model has been derived, by setting the acceleration and velocity dynamic states to zero. This simplification allows the roll angle to be expressed as a linear function of lateral acceleration. The approach assumes that the total roll stiffness remains linear and that the parameter d1 is constant, based on the small-angle linearization technique. Under these conditions, the roll response of the vehicle is primarily governed by lateral load transfer, roll stiffness, and suspension characteristics. This linearized formulation provides insights into the steady-state roll behavior, aiding in the evaluation of vehicle stability and handling performance.

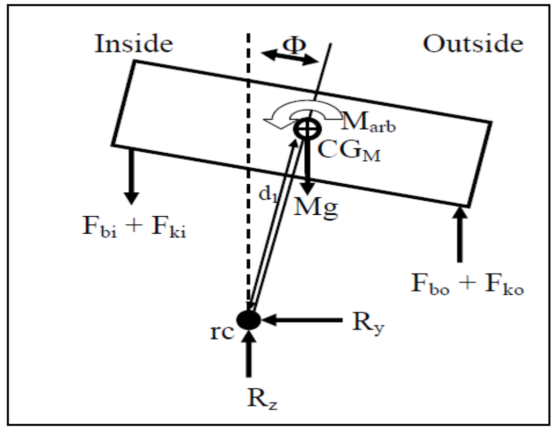

Figure 3. Roll FBD Sprung Mass.

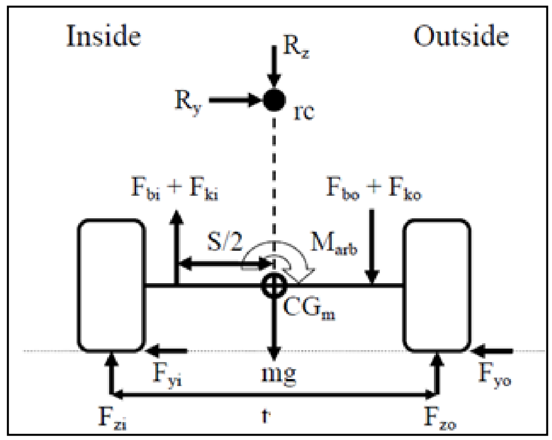

Figure 4. Roll FBD Un-Sprung Mass.

The forces, moment, and lengths in Figures 3 and 4 are defined as:

CG_M = Sprung mass center of gravity
M_{arb} = Anti-roll bar moment
CG_m = Un-sprung mass center of gravity
Φ= Roll angle
d_1 = Length between the rc and CG_M
rc = Roll center
F_b = Damper force (o – outside, i – inside)
R_y = Reaction force in the y-axis
F_k= spring force (o – outside, i – inside)
R_z = Reaction force in the z-axis
F_y= Tire lateral force
S = Length between the springs and dampers
Fz = Tire normal force,
t = Track width

The resulting roll angle equation, given by Equation (7), serves as a fundamental expression for analyzing roll dynamics in steady-state cornering conditions.

$$\Phi_{ss} = \frac{M_s d1}{k_{\Phi t} - M_s d1} \cdot a_y \quad (7)$$

where:

a_y = *Lateral acceleration*
$k_{\Phi t}$ = *Total roll stiffness*
M_s = *Total sprung mass*

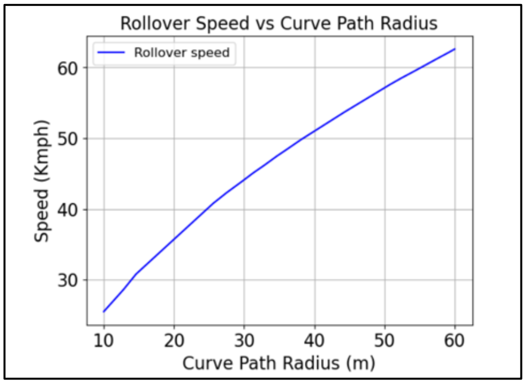

Figure 5. Rollover speed Vs Radius of turn.

C. *Tire Model*

In this study, the Pacejka Magic Formula [6] is used to model tire forces based on existing experimental data for the vehicle, as summarized in Table I. This widely used semi-empirical tire model captures the nonlinear behavior of tires by defining lateral and longitudinal forces as functions of slip angle and slip ratio, respectively [5]. Figure 6 illustrates the differences between the linear and nonlinear Magic Formula tire models.

TABLE I. PACEJKA PARAMETERS

a0 = 0.5;	a9 = 0.0;
a1 = -1300;	a10 = 0.0;
a2 = 2400;	a11 = 0.0;
a3 = -250;	a12 = 0.0;
a4 = -3;	a13 = 0.0;
a5 = -0.0024;	a14 = 0.0;
a6 = -1.6;	a15 = -0.1;
a7 = 1.6;	a16 = 0.0;
a8 = 0.0;	a17 = 0.2

The general form of the Pacejka Magic Formula for longitudinal, lateral, and aligning moment forces is expressed as [5]:

$$y = D \sin\left(C \tan^{-1}\left(Bx - E\left(Bx - \tan^{-1}(Bx)\right)\right)\right) \quad (8)$$

Where:

y - represents the force or moment
x - represents the slip parameter
B - is the stiffness factor, controlling the shape of the curve
C - is the shape factor, determining the curvature
D - is the peak factor, representing the maximum force value
E - is the curvature factor, adjusting the asymmetry of the curve

The linear tire model assumes constant tire cornering stiffness with no saturation, meaning the lateral force increases proportionally with slip angle. In contrast, the Pacejka model accounts for non-linearities by varying the cornering stiffness dynamically.

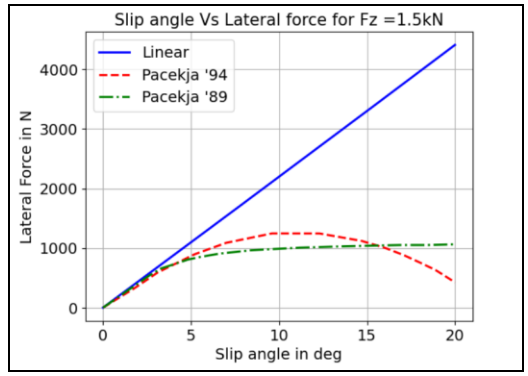

Figure 6. Comparison of Linear vs. Nonlinear Pacejka '89 and '94 Tire Models under a Normal Force of 1.5 kN.

As the slip angle increases, the lateral tire force initially rises, reaching a peak before gradually decreasing due to tire saturation. This behavior accurately represents real-world tire dynamics, particularly during aggressive cornering and limit-handling scenarios. The linear tire model provides a reasonable approximation of the nonlinear Pacejka model for small slip angles [5]. However, as shown in Table II, this approximation becomes increasingly inaccurate as the slip angle grows. The linear model assumes a constant cornering stiffness, leading to an overestimation of lateral force at high slip angles.

TABLE II. LINEAR TIRE MODEL VS. NON-LINEAR PACEJKA TIRE MODEL

Tire Slip Angle, α , [deg]	2.5°	5 °	7 °
Lateral Force difference between Linear and Non-Linear Tire Pacejka ‘89’ Model	4%	33%	65%
Lateral Force difference between Linear and Non-Linear Tire Pacejka ‘94’ Model	14%	25%	40%

In contrast, the nonlinear Pacejka model accounts for tire saturation, capturing the peak lateral force and the subsequent reduction in force beyond this point. This distinction is critical for accurately predicting vehicle behavior in high-speed maneuvers, limit-handling conditions, and dynamic stability analysis.

The peak lateral force generated by a tire is influenced by both slip angle and normal load. However, as the normal load increases, a saturation point is reached beyond which additional load no longer results in a proportional increase in lateral force.

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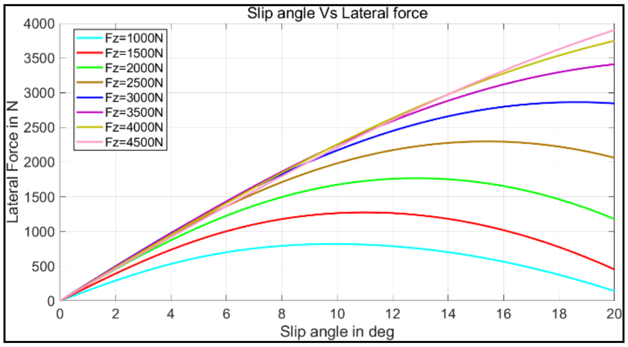

Figure 7. Non-Linear Tire Model with Varying Normal Forces.

This nonlinear relationship, depicted in Figure 7, highlights how both cornering stiffness and peak lateral force vary with changes in normal load. In the Pacejka tire model, the primary inputs are the tire slip angle and normal force, while the output is the resulting lateral force. The model captures the nonlinear behavior of tire forces, showing that while an increase in normal load initially enhances lateral grip, excessive loading can lead to diminishing returns due to tire deformation and structural limitations. Understanding this interaction is crucial for optimizing vehicle dynamics, particularly in suspension tuning and load transfer management.

IV. Virtual Vehicle Model and simulation

A three-wheeled MBD flexible model, as shown in Figure 8, has been developed using ADAMS CAR. The system is represented as two primary subsystems: Rear frame: Includes the rider, engine, chassis, body, seat, and rear wheels. Front frame: Comprises the front fork, handlebar, and front wheel.

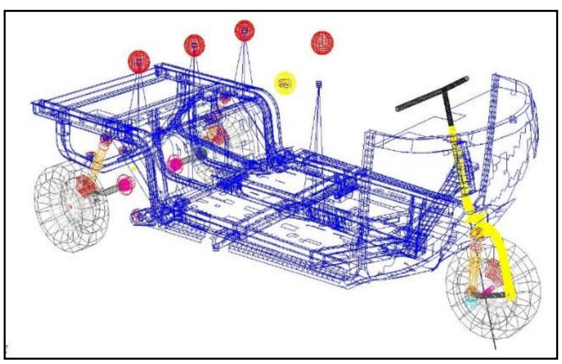

Figure 8. Three-Wheeler MBD Model.

The rear and front frames are connected at the steering axis via a revolute joint, allowing relative rotation between the two sections. During motion, the tires are free to sideslip, generating lateral forces that depend on sideslip and camber angles. These lateral forces, from a dynamic perspective, act as restoring forces similar to those produced by springs, influencing vehicle stability and handling behavior. To enhance model accuracy, key parameters, such as mass properties, inertia, hard point locations, suspension characteristics (spring/damper properties, jounce, and rebound characteristics), and tire properties are updated based on experimental data. A complete vehicle model is constructed using modular templates with user-defined input data.

For tire modeling, both the Pacejka '89 and '94 handling models are developed, with the Pacejka '94 model being implemented in simulations due to its improved accuracy in capturing tire behavior under dynamic conditions [5]. The use of this detailed MBD model allows for a comprehensive analysis of three-wheeled vehicle dynamics, particularly in evaluating stability, ride quality, and handling performance.

A. MBD Simulation

A full-vehicle MBD simulation has been conducted to analyze the handling characteristics of a three-wheeled vehicle. The simulations were performed using ADAMS/Car, which provides a driving machine module capable of executing various handling maneuvers [3]. These maneuvers are broadly classified into: Open-loop steering maneuvers: Driver-independent inputs, useful for evaluating fundamental vehicle dynamics. Closed-loop maneuvers: Driver-in-the-loop simulations, considering control feedback mechanisms. In this research, the following handling analyses were performed, and their results were evaluated:

Step Steer Maneuver – A sudden steering input is applied to examine the transient response of the vehicle, focusing on yaw rate, lateral acceleration, and roll stability.

Constant Radius Cornering (CRC) – The vehicle is driven in a steady-state circular path to assess lateral grip, understeer/oversteer characteristics, and roll behavior at different speeds. This is shown in Figure 9.

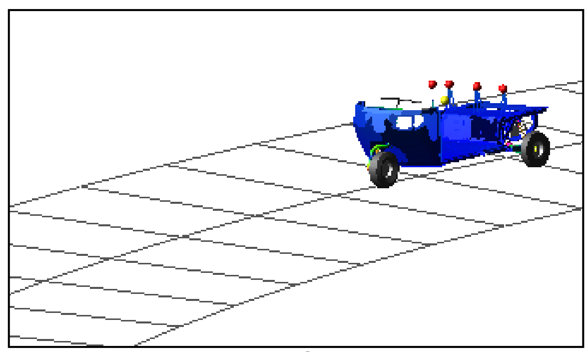

Figure 9. CRC – MBD Simulation in ADAMS.

These simulations provide critical insights into the stability, responsiveness, and overall handling performance of the three-wheeled vehicle under dynamic conditions.

a) Step Steer maneuver

A step steer analysis yields time-domain transient-response metrics. During a step steer analysis, ADAMS/Car increases the steering input from an initial value to a final value over a specified time. The most important quantities measured are shown in Figure 10.

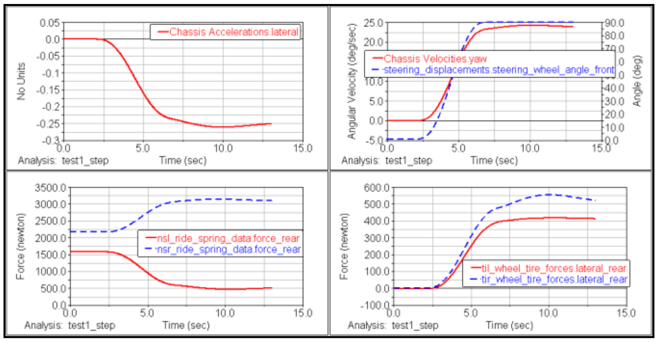

Figure 10. Vehicle lateral acceleration, yaw rate, steering angle input, rear (left/right) spring forces, tire lateral forces.

b) *Constant radius cornering*

For constant-radius cornering analysis, the Driving Machine drives full vehicle down a straight road, turns onto a skidpad, and then gradually increases velocity to build up lateral acceleration. One common use for a constant radius cornering analysis is to determine the understeer characteristics of the full vehicle [2].

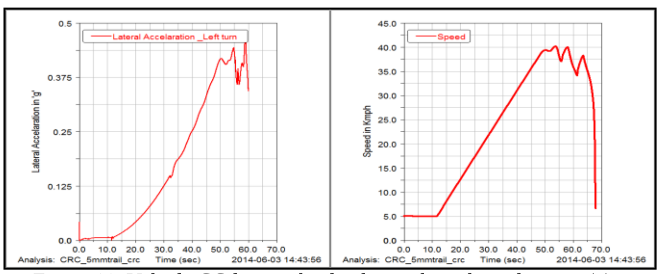

Figure 11. Vehicle CG longitudinal velocity, lateral acceleration (g).

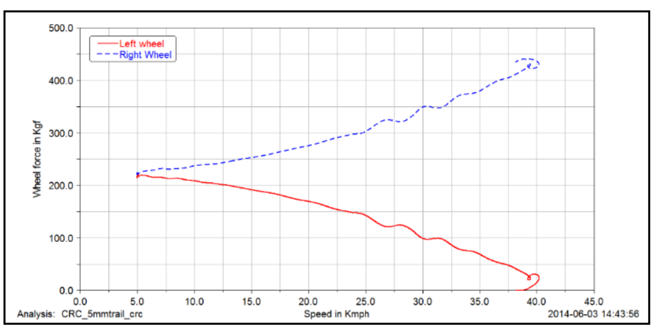

Figure 12. Rear Tire (left and right) normal forces.

It is also useful to find out vehicle velocity at which roll-over instability starts. From the Figure 10 and Figure 11 given below, it was observed that the vehicle stars to roll when its velocity reaches around 38 kmph for 30m radius.

V. Experimental Validation of Simulation Results

An experimental study was conducted to validate the vehicle dynamics simulation results by performing real-world tests on a controlled test track shown in Figure 12. The test vehicle was driven in a steady-state circular maneuver on a track with a 30 m radius. The objective was to assess lateral acceleration, roll angle, and yaw velocity under varying speed conditions and determine the threshold at which wheel lift-off occurs [8].

The test was conducted across a range of speeds, beginning from the lowest feasible velocity and gradually increasing to the maximum possible speed before instability. The speed increment strategy was designed to ensure a systematic variation in lateral acceleration:

- Up to 28 km/h, the speed was increased in steps that corresponded to an approximate lateral acceleration increment of 0.05 g.
- Beyond 28 km/h, the speed was increased in fixed increments of 2 km/h until wheel lift-off was observed.

The RT1003 Inertial Measurement Unit (IMU) and Steering Sensor is a high-precision system designed for vehicle dynamics analysis, capturing yaw, pitch, and roll rates using advanced gyroscopes, along with linear accelerations via high-accuracy accelerometers shown in Fig.13. It provides real-time roll and slip angle estimation, essential for stability control and performance evaluation. Integrated with a high-resolution steering angle sensor and torque measurement capability, it enables detailed analysis of steering response and driver input [9].

Figure 12. Constant Radius Cornering at Test rack.

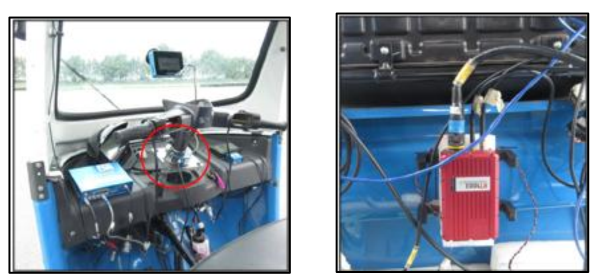

Figure 13. RT1003 IMU and Steering sensor for Vehicle dynamics Parameter Measurement.

From the experimental results, it was observed that the onset of wheel lift-off occurred at approximately 40.38 km/h, indicating the point of critical lateral acceleration at which the vehicle's roll stability limit was exceeded. The findings are graphically represented in Figure 14, illustrating the relationship between speed, lateral acceleration, and roll angle leading to the instability condition.

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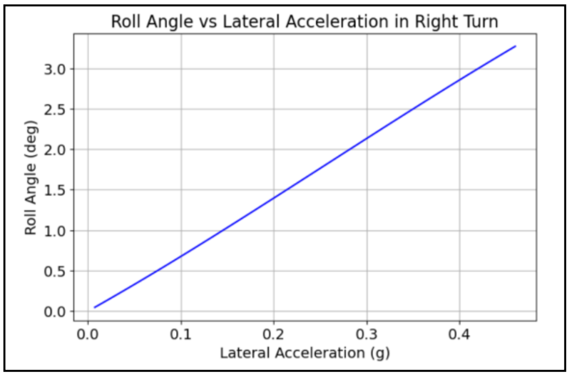
The test results provide valuable insight into the real-world validation of vehicle stability limits and rollover tendencies, offering a critical comparison with the simulated predictions. These findings contribute to improving vehicle safety analysis and the refinement of suspension and stability control systems.



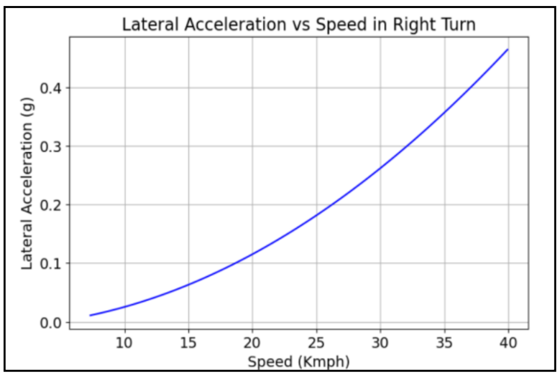


Figure 14. Lateral accelerations, Speed and Roll angle.

VI. CONCLUSION

This study presents the development and validation of a 12-Degrees-Of-Freedom (DOF) vehicle model to analyze ride and handling dynamics under various driving conditions. The integration of both linear and nonlinear (Magic Formula) tire models allowed for comparative analysis against Automated Dynamic Analysis of Mechanical Systems (ADAMS) simulation results, with the nonlinear model demonstrating superior alignment.

The validation process included MBD simulations and experimental tests, focusing on key handling aspects, such as yaw rate, lateral acceleration, and roll stability. The results showed that the vehicle exhibited roll instability at approximately 38 km/h in simulations, closely aligning with the experimentally observed wheel lift-off at 40.38 km/h.

The findings highlight the importance of incorporating high-fidelity tire models and validating vehicle dynamics through real-world testing to ensure accuracy. This research provides valuable insights into vehicle stability and rollover tendencies, which are crucial for improving suspension and stability control systems.

VII. FUTURE WORK

Future research can be extended to a wider range of road conditions, including asphalt, pave, and Belgian blocks, under varying friction (μ) levels to better understand their impact on vehicle dynamics. Additionally, investigations into diverse vehicle configurations, such as three-wheeled and unconventional architectures, will enable a more comprehensive assessment of dynamic behavior under real-world operating conditions. This will facilitate the refinement of suspension kinematics, steering response, and stability control strategies, ultimately enhancing vehicle safety, ride comfort, and performance.

Furthermore, the current model will be improved by incorporating a higher Degrees-Of-Freedom (DOF) and integrating structural compliances to better capture real-world dynamic behavior. Advanced analyses will be performed by incorporating roll stability detection and control. Wheel Force Transducer (WFT) data acquired from Road Load Data Acquisition (RLDA) to quantify component-level excitations. These excitations directly influence key vehicle dynamics parameters, including ride quality, handling characteristics, and structural durability, enabling a more precise evaluation of system-level interactions and optimization of vehicle performance.

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