DEVELOPMENT OF VEHICLE DYNAMICS TOOLS FOR DESIGNING AND TESTING OF FORMULA SAE CAR TO OPTIMIZE VEHICLE PERFORMANCE AND DRIVABILITY

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This research focuses on developing a vehicle dynamics tool for the design and testing of Formula SAE cars to optimize vehicle performance and drivability. The tool integrates the MRA Moment Method (MMM) with non-dimensional tire modeling using Magic Formula 6.1 to address key challenges in optimizing vehicle performance and drivability. Featuring a user-friendly interface, it facilitates data input and KPI analysis of performance and divability. The methodology includes GUI development, simulation-based design optimization, vehicle setup optimization on the 2025 Chulalongkorn University Formula SAE car. By prioritizing simplicity, adaptability, and innovation, this research aims to significantly enhance development efficiency and competitive performance while providing a robust foundation for future advancements in motorsport vehicle dynamics tool.

KEY WORDS: **vehicle dynamics, suspension system, evaluation technology,** Quasi-Static Simulation, Performance Analysis

1. Introduction

In motorsport, achieving high performance requires operating at the limits of a vehicle's capabilities, but accurately predicting and quantifying vehicle behavior at these limits is challenging. This issue is particularly evident in the Formula SAE competition, where university students worldwide design and build prototype race cars to compete annually. Two major challenges in this context are

1) Designing a vehicle that optimizes both driving performance and drivability.
2) Optimizing vehicle setup during testing to enhance performance and drivability.

The annual nature of Formula SAE competitions means teams typically have only one year to design, manufacture, and test their cars. Without the assistance of advanced vehicle dynamics software, components such as the suspension and steering systems are often developed through trial and error, relying on feedback from the previous year's competition. This approach significantly slows the pace of improvement. Even with isolated calculations or simulations, it remains difficult to predict the integrated performance of the entire vehicle without extensive physical testing, complicating early design decisions. Components like suspension and steering geometries, once finalized, are challenging to modify, underscoring the importance of optimizing these designs as much as possible before manufacturing.

A further complication is the unpredictability of drivability until the car is built. In Formula SAE, where most drivers are amateur university students, drivability is critical to ensure the car remains controllable and stable under various conditions. With the help of vehicle dynamics software, teams can design vehicles not only for optimal performance but also to maintain good drivability. This ensures that the car strikes a balance between speed and stability, accommodating the skill levels of student drivers while maximizing overall competitiveness.

After manufacturing, discrepancies often arise between the designed and actual vehicles due to unavoidable manufacturing errors. This necessitates post-production setup optimization to account for these deviations and adapt the car to real-world track conditions. Without vehicle dynamics tools, teams are limited to evaluating performance through lap times and subjective driver feedback, which makes it difficult to quantify performance and drivability objectively.

Additionally, Formula SAE cars offer extensive setup customizability—parameters like roll bar stiffness, camber angle, and toe angle can all be adjusted. Optimizing these parameters through trial and error is time-consuming and inefficient. The lack of tools to quantify the impact of individual setup changes further complicates the process.

A vehicle dynamics tool capable of quantifying both performance and drivability would significantly improve development efficiency. By enabling pre-testing analysis and providing a systematic way to correlate subjective driver feedback with objective simulation results, such a tool would allow teams to make informed design and setup decisions, saving valuable time during the testing phases.

In the field of vehicle dynamics tools, most software used in the motorsport industry prioritizes performance metrics, such as lap times, with less emphasis on drivability. Tools that account for drivability aspects, such as control and stability, often rely on transient multibody simulations, which provide highly accurate results but come with significant drawbacks. These tools are typically less user-friendly, require numerous hard-to-measure inputs, and make it challenging to study and optimize the impact of individual parameters. Their complexity also limits their practicality during the early stages of vehicle development.

On the other hand, simpler linear models are more accessible and easier to use in early development phases but lack the accuracy required for predicting behavior near the vehicle's limits, primarily due to the non-linear characteristics of tires. To achieve a balance between accuracy and usability, we propose using a non-dimensional tire model in combination with the MRA Moment Method (MMM) to simulate results.

The MRA Moment Method (MMM), first proposed in 1952 by Bill Milliken and later discuss in "Race Car Vehicle Dynamics" [1] which is one of the most authoritative references on vehicle dynamics for racing application, originated from aeronautical stability and control principles and was later adapted for

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automotive applications. Initially used for vehicle testing, the method has since evolved into a simulation tool with the advancement of computing power.

One of the key advantages of the MMM is its ability to provide valuable metrics for both steady-state and non-steady-state cornering conditions, without the computational intensity of a full dynamic simulation. Additionally, the MMM offers a more insightful method for quantifying understeer and oversteer compared to traditional vehicle dynamics theory, which assumes tires operate in the linear range. In real racing scenarios, tires often function near their limits, making linear models less applicable.

Furthermore, while traditional vehicle dynamics are typically based on a half-car or bicycle model, the MMM utilizes a full-car model. This enables the analysis of complex effects such as suspension kinematics or steering geometry, including the influence of Ackermann steering, which cannot be adequately observed with simpler a quarter-car model.

To model the tire for use with the MMM, we will utilize Magic Formula version 6.1 (MF6.1), modeled using tire testing data from the Formula SAE Tire Test Consortium [2] and fitted through the Magic Formula Tyre Tool in MATLAB [3]. A key advantage of MF6.1 is its ability to account for the effects of tire pressure, an improvement over earlier versions of Pacejka's Magic Formula [4].

Given the lack of accurately simulated lap times using this method, users can still analyze key performance indicators (KPIs) relevant to vehicle performance and drivability. These KPIs provide insights into critical aspects such as vehicle maximum grip, balance, control, and stability at various stages of cornering, including turn-in and limit cornering. This approach enables a more quantitative and subjective comparison of vehicle design or setup changes. Additionally, it accounts for the influence of drivability, which plays a significant role in the resulting lap time but is difficult to assess solely through simulated lap times.

In conclusion, the main objective in this research is to develop a vehicle dynamics tool for designing and testing of Formula SAE Car with the use of The MRA Moment Method (MMM) to quantify and optimize vehicle performance and drivability. A key advantage of this method is its ability to provide valuable insights into vehicle drivability while requiring fewer input parameters compared to fully dynamic or multi-body simulations. This reduced input requirement makes it particularly well-suited for early-stage concept design studies and rapid trackside setup evaluations, where efficiency and quick assessments are prioritized over more in-depth analyses.

2. Research Design

The development of this vehicle dynamics tool can be divided into 3 main sections

1) Developing the base Graphical User Interfaces (GUIs)

This phase involves creating an intuitive interface to simplify the process of inputting data, adjusting vehicle parameters, and visualizing results. The GUI will streamline the generation of yaw moment diagrams from The MRA Moment Method (MMM) and the extraction of key performance indicators (KPIs) related to performance and drivability.

2 Application of the tool during the design phase of an FSAE car

During the design phase, the developed tool will be applied to model and simulate various vehicle configurations, enabling data-driven decisions to optimize performance and drivability early in the design process of 2025 Chulalongkorn University Formula SAE team's car.

3) Application of the tool during the testing phase of an FSAE Car

During the testing phase, the developed tool will be used to analyze key performance indicators (KPIs) to predict and refine vehicle setup to suit track conditions and driver preferences. To evaluate the influence of each KPI on vehicle behavior, particularly drivability, testing will primarily rely on driver feedback and resulting lap times. Testing will be conducted using the 2025 Chulalongkorn University Formula SAE team's vehicle at Pathumthani Speedway, the venue for the TSAE Auto Challenge 2025.

3. Research Methodology

The development of this vehicle dynamics tool is primarily based on the MRA Moment Method (MMM) [1], specifically for generating Milliken Moment Diagrams (MMDs). Once the MMDs are generated, the tool will extract key vehicle performance and drivability metrics, for comparison across different vehicle designs and setups

The Milliken Moment Diagrams (MMDs) are constructed by identifying the lateral acceleration and yaw moment of the vehicle created from a range of vehicle slip angle and steering angle at either constant radius or speed. There are various types of MMDs presented in [1] but we will focus on the C_N-A_Y Diagram which is a plot of the yaw moment coefficient C_N vs. lateral acceleration A_Y at constant speed. For this analysis, only the pure cornering case is considered, where longitudinal acceleration is set to zero (called as Free Rolling C_N-A_Y Diagram in [5]). This assumption simplifies the calculations while maintaining the core dynamics of vehicle handling evaluation. And instead of using the yaw moment coefficient C_N we will be using just the yaw moment (N-A_Y Diagram) for easier interpretation of the plot.

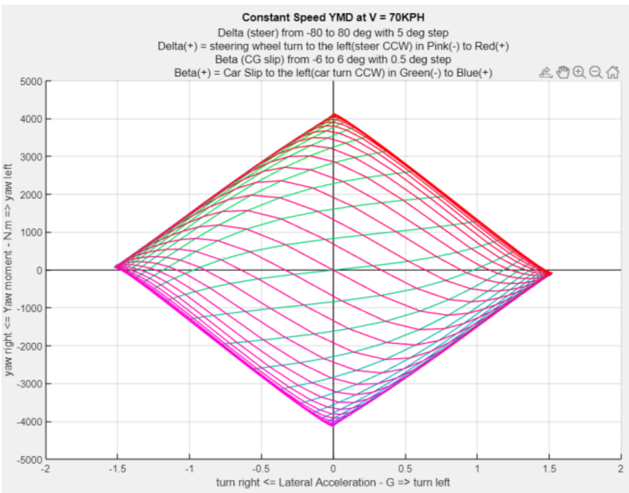

Fig. 1: Milliken Moment Diagram – Free Rolling N-A_Y Diagram at 70KPH

In the N-A_Y Diagram, each colored line represents an iso-line corresponding to either the steering angle (δ) or vehicle slip angle (β). As shown in Figure 1, the transition from pink to red indicates a change in steering angle (δ) from -80° to 80° in 5° increments, while the transition from green to blue represents a change in vehicle slip angle (β) from -6° to 6° in 0.5° increments.

Any point along the X-axis (N = 0) represents a pure steady-state cornering condition, where forces and moments are balanced. This axis is referred to as the steady-state line.

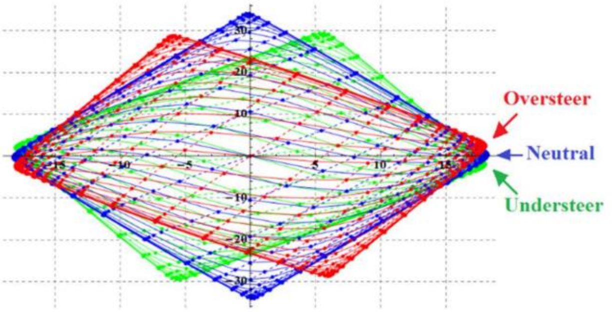

Fig. 2: Diagram of an oversteer, neutral and understeer car. [6]

From the diagram multiple stages of cornering can be studied. The iso-line passing through the origin provides insight into vehicle control and stability during corner entry, based on its slope. The outer edges of the diagram illustrate the vehicle's behavior near the tire grip limit. By analyzing the remaining yaw moment above or below the steady-state line, the vehicle's balance at the limit can be assessed. This helps determine whether the vehicle exhibits a tendency to plow (understeer) or spin (oversteer) as shown in Figure 2.

The working of this tool can be mainly divided into 4 main parts.

3.1) MMD generation algorithm
3.2) Vehicle model
3.3) Tire model
3.4) GUI and KPIs extraction

3.1) MMD Generation Algorithm

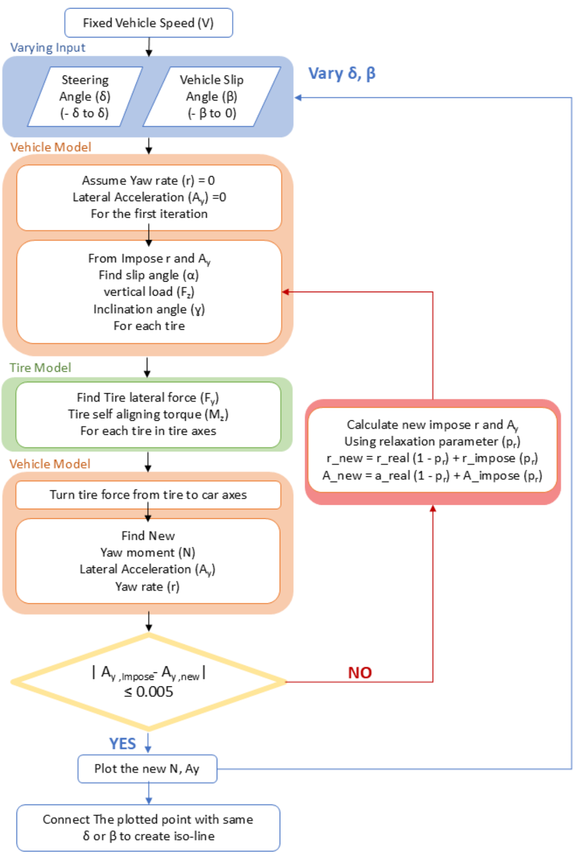

Fig. 3: Flowchart of the MMD Generation Algorithm.

Figure 3 presents the flowchart for generating the Milliken Moment Diagram (MMD). The core concept of the algorithm is as follows: during the first iteration, determining the slip angle (α), vertical load (F_z) and Inclination angle (ɣ) for each tire requires knowledge of the yaw rate (r) and lateral acceleration (A_y). However, calculating r and A_y necessitates prior knowledge of the forces acting on all tires, which in turn depend on α, F_z, ɣ. This creates an implicit problem.

To resolve this, the algorithm initially assumes r = 0 and A_y =0. Using these assumed values, the forces are computed, and the newly obtained A_y is compared against the imposed A_y. Convergence is achieved when the difference between the imposed and calculated A_y falls within a predefined tolerance, set at ≤ 0.005 m/s^2 in this study. If convergence is not met, a new imposed value for r and A_y is computed iteratively.

To enhance convergence speed, a relaxation parameter p_r, as proposed in [5], is employed. This parameter provides a weighted update between the previously imposed and newly computed values, effectively reducing oscillations and improving computational efficiency.

Once the final A_y has converged, the resulting yaw moment (N) and lateral acceleration (A_y) are plotted for a given input steering angle (δ) and vehicle slip angle (β). This process is repeated across the desired range of δ and β values. To further optimize computational efficiency, left-right symmetry is assumed, reducing the number of required calculations by half. The results for β are then mirrored, covering the range from -β_{max} to β=0, getting the all the calculation point needed for generating the MMD as shown in Figure 4

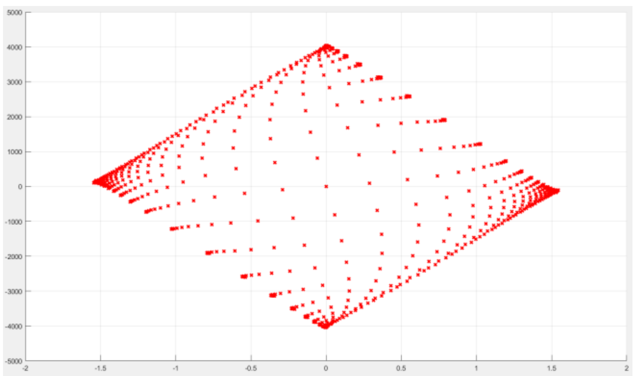

Fig. 4: Plotting of N-A_Y result from varying δ and β

Once all N-A_Y data points have been plotted, the points corresponding to the same input steering angle (δ) or vehicle slip angle (β) are connected to generate iso-δ and iso-β contour lines. To enhance visualization of the Milliken Moment Diagram (MMD), these iso-lines are color-coded based on their respective parameter variations. Specifically, a color gradient from pink to red represents an increasing steering angle (δ) from δ_{min} to δ_{max} , while a transition from green to blue indicates an increasing vehicle slip angle (β) from β_{min} to β_{max} as shown in Figure 5

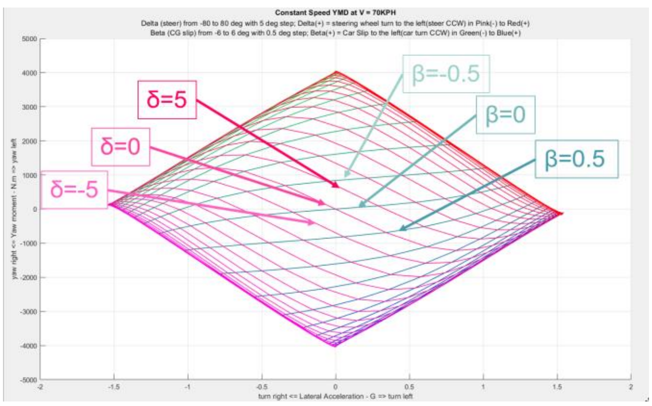

Fig. 5: N-A_Y diagram with color-coded iso contour line.

3.2) Vehicle Model

To simulate vehicle behavior, a mathematical model was developed in MATLAB, primarily based on the method proposed in [6], with additional detailed modeling inspired by [1] and [7]. The primary enhancement over the approach in [6] involves incorporating the effects of the tire inclination angle (ɣ), commonly referred to as the camber angle, which is important for use with MF6.1 tire model. This model accounts for both roll camber and steer camber change, considering the dynamic variation of caster and kingpin inclination angles due to vehicle body roll. This improvement enables the analysis of phenomena such as the influence of caster and kingpin inclination angle changes on vehicle behavior.

Some assumption will be made to help simplifies the model this include

- Neglecting suspension compliance.
- Spring rate value from tire is steady so the total roll rate will be constant.
- Assumed 50% left and right weight distribution to help with computational efficiency.
- Fixed vehicle CG location.
- Fixed roll center location.
- The coefficient of lift of the vehicle is constant.

The 4 main function of this vehicle model is to find

3.2.1) slip angle (α) for each tire
3.2.2) vertical load (F_z) for each tire
3.2.3) Inclination angle (ɣ) for each tire
3.2.4) Yaw moment (N) and lateral acceleration (A_y) of the vehicle from forces generated by the tire

3.2.1) Finding slip angle (α) for each tire

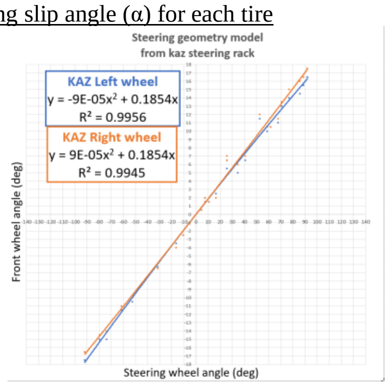

Fig. 6: Steering system modeled as quadratic function.

The slip angle of each tire is a function of three parameters: steering angle (δ), vehicle slip angle (β), yaw rate (r). To model the steering system, following the method proposed in [6], The Ackermann steering geometry can be effectively modeled by approximating the rotation of each wheel as a quadratic function of the steering angle (δ) as shown for example in Figure 6. The relevant equation can be found in [6].

3.2.2) Finding vertical load (Fz) for each tire

Following method proposed in [6], to find vertical load (Fz) acting on each tire consist of three main components: static normal force, lateral weight transfer, and aerodynamic force.

The static normal force represents the force exerted when the car is stationary, primarily due to the vehicle's weight distribution. This force depends on the vehicle's mass and how weight is distributed between the front and rear axles.

Lateral weight transfer occurs as a result of lateral acceleration and can be divided into three sub-components. The first is non-suspended mass transfer, which accounts for the lateral force acting on unsprung masses such as wheels and axles. The second is geometric transfer, which considers how the suspension geometry such as roll center height redistributes the load across the vehicle. The third is elastic transfer, which captures the load shift caused by suspension deflections, including the effects of springs and anti-roll bars.

In addition to these forces, aerodynamic forces contribute to the normal force by generating downforce as the vehicle moves. This aerodynamic effect depends on factors such as air density, frontal area, vehicle speed, and aerodynamic coefficients.

By combining these three components, the total normal force on each wheel can be determined. This allows for an accurate representation of the load distribution across the vehicle while also considering the influence of roll rate distribution. The relevant equation can be found in [6].

3.2.3) Finding inclination angle (ɣ) for each tire

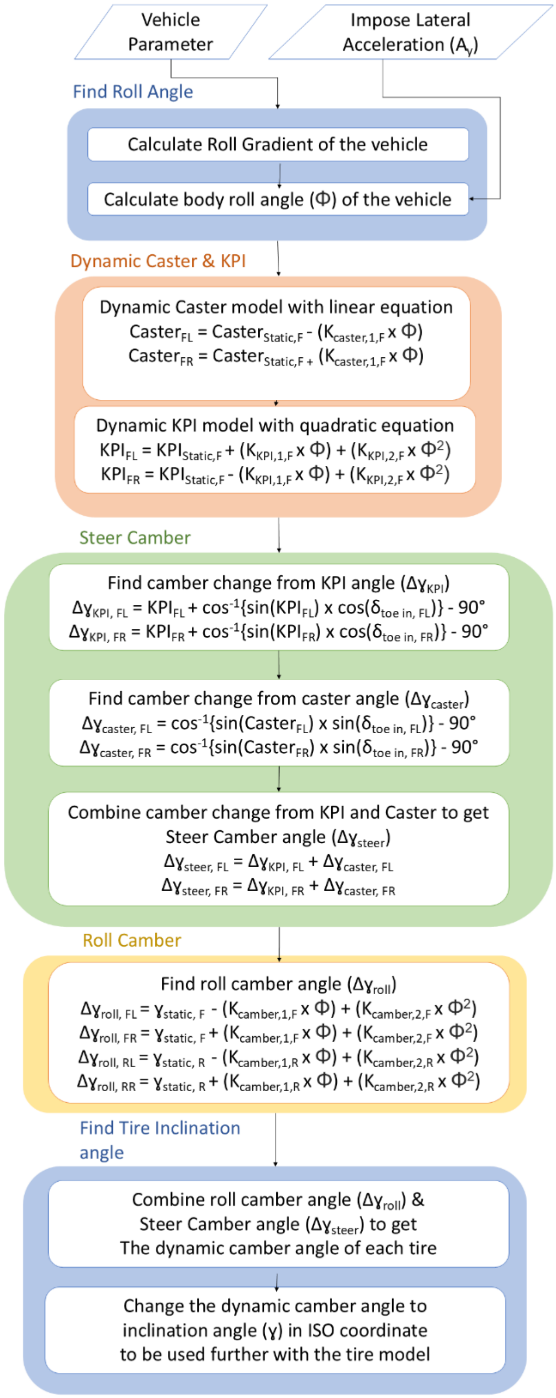

Fig. 7: Flowchart of the inclination angle finding function.

Figure 7 presents the flowchart for determining the inclination angle (ɣ), which is later used in the calculation of tire forces within the tire model. Since the simulation focuses solely on cornering, with no longitudinal acceleration, changes in the inclination angle, or camber angle, result from two components: steer camber and roll camber. The resulting dynamic camber is the combined effect of these two factors.

Firstly, the vehicle body roll angle (Φ) must be determined. According to [1], this can be calculated by multiplying the lateral acceleration (A_y) by the roll gradient ($\frac{\Phi}{Ay}$) which represents the vehicle's roll sensitivity. The roll gradient primarily depends on the locations of the center of gravity and the roll center, as well as the total roll rate or roll stiffness ($K_{\Phi,\ total}$). The detailed calculation is presented in Figure 8 and Equation (3.1) below.

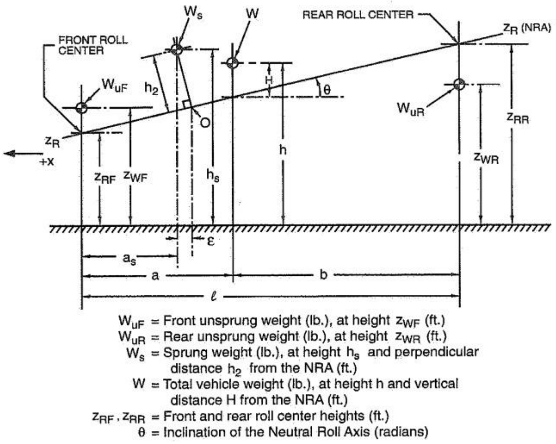

Fig.8 Lateral load transfer geometry [1]

$$\frac{\Phi}{Ay} = \frac{-W_s h_2}{K_{\Phi,\mathrm{total}} - W_s h_2} \tag{3.1}$$

After determining the roll angle (Φ), the next step is to calculate the steer camber change using the equations from [7], presented below as Equations (3.2) and (3.3).

$$\Delta \gamma_{KPI} = KPI + \cos^{-1}\{\sin(KPI)\cos(\delta_{toe\ in})\} - 90° \tag{3.2}$$

$$\Delta \gamma_{caster} = \cos^{-1}\{\sin(Caster)\sin(\delta_{toe\ in,})\} - 90° \tag{3.3}$$

We could see that in order to determine $\Delta\gamma_{KPI}$ and $\Delta\gamma_{caster}$ it is first necessary to obtain the values of the kingpin inclination angle (KPI) and caster angle, both of which vary dynamically with the roll angle (Φ). Therefore, a mathematical model expressing KPI and caster as functions of Φ is required. Through analysis using suspension kinematic software, it was determined that, for this specific suspension configuration, a conventional unequal-length, non-parallel double wishbone suspension, the dynamic caster can be accurately modeled as a linear function of Φ, while the dynamic KPI is best represented as a quadratic function of Φ shown below in Figure 9,10

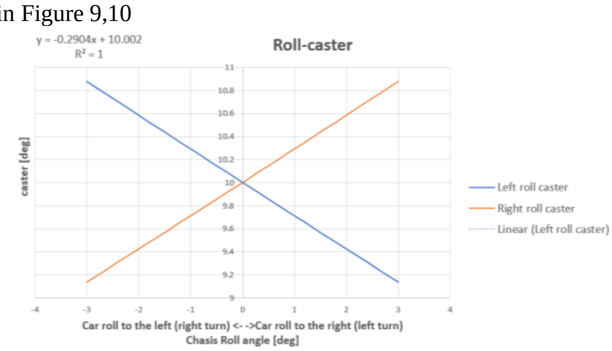

Fig.9 Roll(Φ)-Caster modeled as linear function.

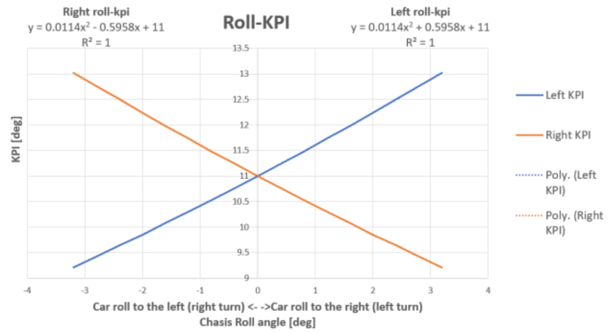

Fig.10 Roll(Φ)-KPI modeled as quadratic function.

Lastly, it is necessary to model the roll-camber characteristics of the suspension. Analysis using suspension kinematic software revealed that the camber change resulting from vehicle roll (Φ) can be accurately represented as a quadratic function of Φ, as illustrated in Figure 11.

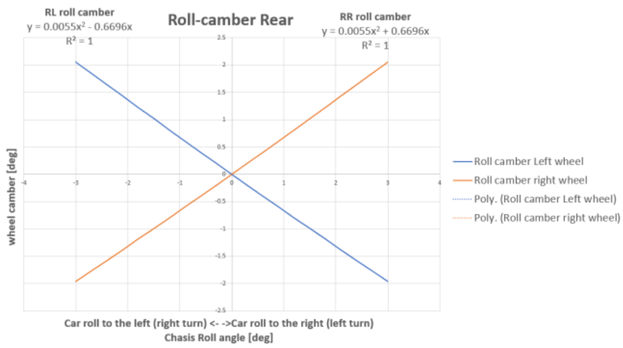

Fig.11 Rear axle Roll(Φ)-Camber modeled as quadratic function.

Then, by combining the changes from both steer camber and roll camber, the camber angle for each wheel can be accurately determined. This camber angle is then adapted to the tire model's ISO coordinate system for use with the MF6.1 tire model in finding tire forces.

<u>3.2.4) Finding yaw moment (N) and lateral acceleration (A_y) of the vehicle from forces generated by the tire</u>

After each tire lateral force (F_y) and self-aligning torque (M_z) has been determine from the tire model, it is essential to turn those forces back from being in each tire coordinate axes to the vehicle coordinate system to determine the resulting yaw moment (N) and lateral acceleration (A_y). The relevant equation can be found in [6].

<u>3.3) Tire Model</u>

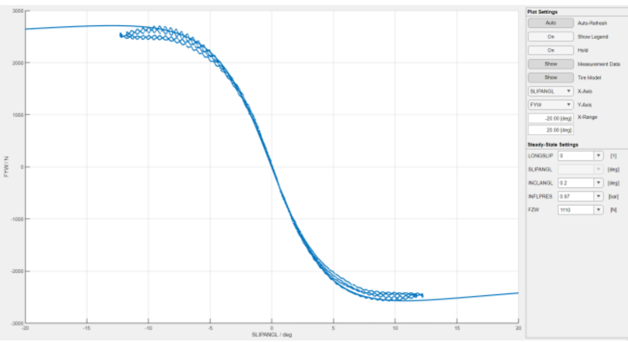

Fig. 12: Tire testing data from FSAE-TTC being curve fitted to a MF6.1 tire model

To model the tire for use in generating the MMD, the Magic Formula version 6.1 (MF6.1) is utilized. This model is based on tire testing data from the Formula SAE Tire Test Consortium [2] and has been fitted using the Magic Formula Tyre Tool in MATLAB [3] as shown in Figure 12. The tire coordinate systems use in this study is in ISO coordinate system.

A key advantage of MF6.1 over earlier versions of Pacejka's Magic Formula [4] is its ability to account for the effects of tire pressure and inclination angle, enhancing the model's accuracy and versatility.

3.4) GUI and KPIs extraction

To enhance usability in both inputting data and analyzing results, a Graphical User Interface (GUI) was developed using MATLAB App Designer. The GUI streamlines the process of entering vehicle and simulation parameters, making the tool more accessible. Additionally, since the Milliken Moment Diagram (MMD) can be challenging to analyze and compare on its own, the GUI extracts key performance indicators (KPIs) related to vehicle performance and drivability, derived from the generated free-rolling N-Ay diagram.

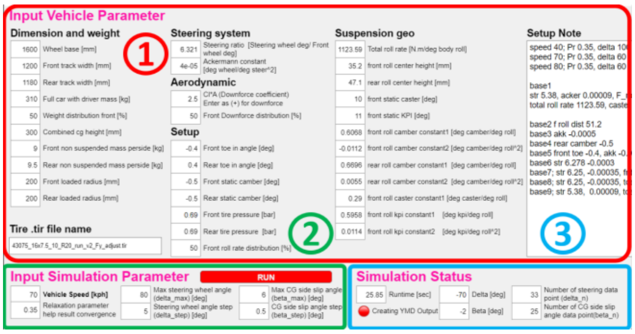

Fig. 13: GUI of the inputting section

Figure 13 presents the data input section of the GUI, which is divided into three main parts:

1. Input Vehicle Parameter – This section allows users to modify the vehicle parameters. It also includes a notes area for recording additional information if needed.
2. Input Simulation Parameter – This section enables users to adjust the simulation settings, including vehicle speed, the range of steering angles (δ), and vehicle slip angles (β) to be simulated. Users can also define the increment steps for δ and β to control the result resolution and adjust the relaxation parameter (p_r) to optimize simulation speed.
3. Simulation Status – This section provides real-time updates on the simulation's progress, indicating the elapsed time and how much of the simulation has been completed.

The result section will be mainly divided into 2 part which is Result KPI and Result Plot.

3.4.1) Result KPI

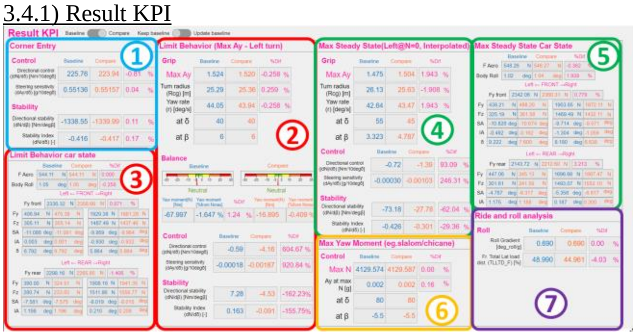

Fig. 14: Result KPI layout

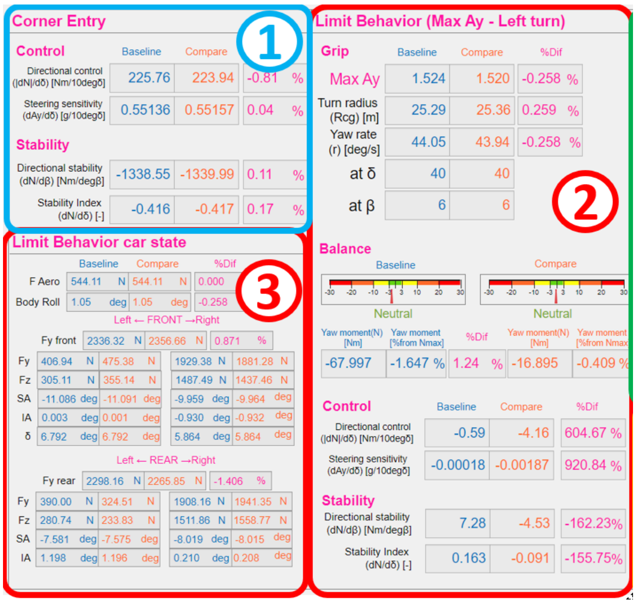

Fig. 15: Result KPI zone 1-3

1) Corner Entry – This area provides KPIs related to control and stability during the initial phase of cornering. These values are derived from the slope of the Milliken Moment Diagram (MMD) at the origin point (N, Ay = 0). The relevant equation can be found in [1].

2) Limit Behavior – This area provides KPIs related to grip, balance, control, and stability near the vehicle's handling limits, corresponding to the right edge of the MMD. Grip and balance are determined from the values of N and Ay at the limit. A negative N indicates a tendency toward understeer, while a positive N suggests oversteer. Control and stability are defined similarly to those during corner entry; however, the slope is calculated at the edge of the MMD instead of the origin.

3) Limit Behavior Car State – This area displays the lateral force generated at each tire, along with the corresponding tire slip angle (SA), vertical load (Fz), and inclination angle (IA) for each tire. This information helps users understand how changes affect these values and provides insights into potential adjustments for optimizing the vehicle setup.

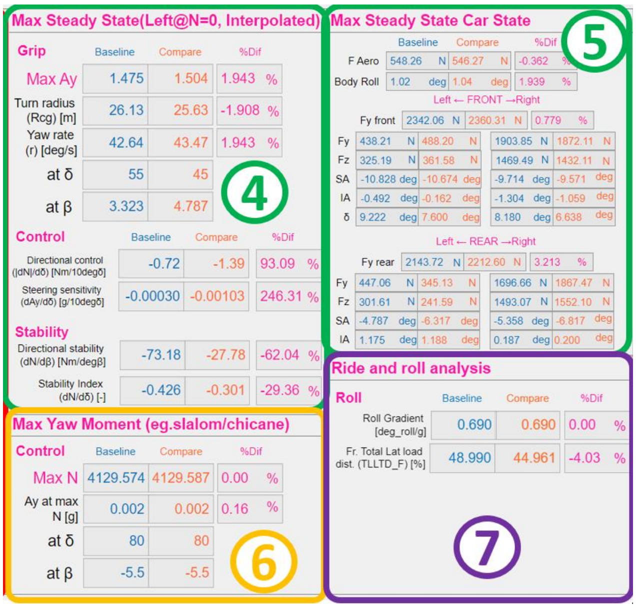

Fig. 16: Result KPI zone 4-7

4) Max Steady State – This area provides KPIs related to grip, control, and stability near the vehicle's handling limits under steady-state cornering conditions, where N = 0. To obtain these values, the point where the iso-line intersects the steady-state line (N = 0) is interpolated. This analysis is particularly useful for evaluating vehicle behavior during mid-cornering in long-duration corners, such as those encountered in a skid pad event in FSAE competitions, which closely resemble steady-state cornering conditions.

5) Max Steady State Car State – Same information as Limit Behavior Car State but for Steady State instead.

6) Max Yaw Moment – This area displays the maximum yaw moment that can be generated, along with the corresponding steering angles (δ) and vehicle slip angles (β) at which it occurs. This information is valuable for assessing the vehicle's controllability in highly dynamic cornering situations, such as slalom events.

7) Ride and Roll Analysis – This area displays the roll gradient and the total lateral load transfer distribution of the vehicle, providing a quick summary of the effects of any changes made.

3.4.2) Result Plot

There are 4 diagrams in the result plot section of the GUI, these following analysis method from [1]

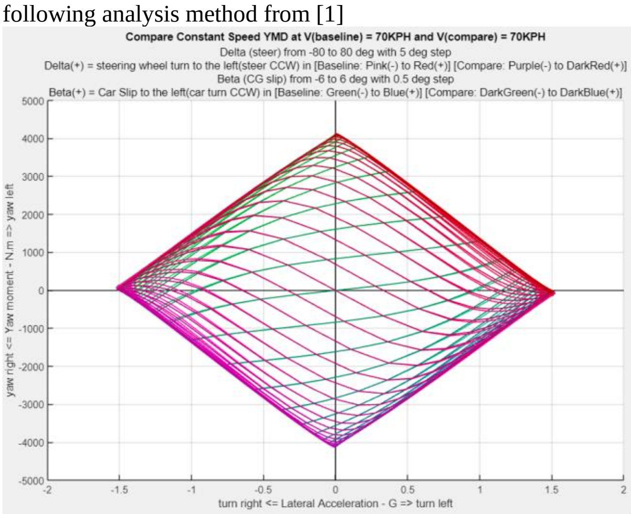

Fig. 17: N-A_Y Diagram in the result plot

1) N-A_Y Diagram – Although KPIs have been extracted from the diagram for convenience of analysis, it is still valuable to examine the raw diagram itself. The GUI allows users to zoom in and move the diagram, enabling a closer analysis of specific areas of interest.

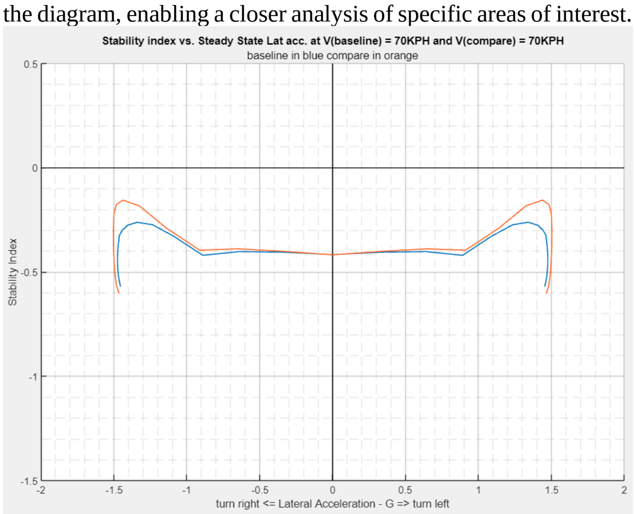

Fig. 18: Stability index in steady state cornering plot

2) Stability index in steady state cornering diagram – This diagram provides valuable insights into the vehicle's stability in response to changes in lateral acceleration during steady-state cornering. A negative stability index indicates that the vehicle is stable, while a value equal to or above zero suggests that the vehicle is unstable. The more negative the stability index, the more stable the vehicle becomes.

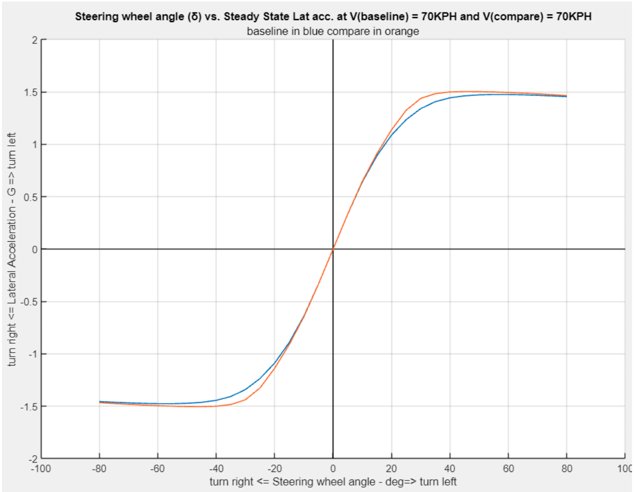

Fig. 19: Steering angles (δ) vs steady state A_y plot

3) Steering angles (δ) vs steady state A_y – This diagram provides insights into the vehicle's steering sensitivity in steady-state cornering situation.

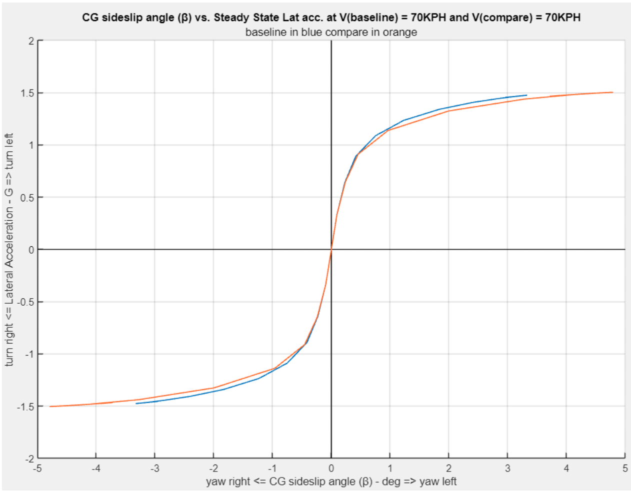

Fig. 20: vehicle slip angles (β) vs steady state A_y plot

4) vehicle slip angles (β) vs steady state A_y – This diagram provides insights into the vehicle's sideslip angle response in steady-state cornering situation.

4. Result and Discussion

After the vehicle dynamic tool GUI have been developed, following the main objective of this research, it has been put to use in the designing and testing process of the development of 2025 Chulalongkorn University Formula SAE team's car. In this section we will discuss about the use of this vehicle dynamic tool and the impact it has on designing and testing of FSAE car.

4.1) Application of the tool during the design phase of an FSAE car

This tool has been used for mainly 2 tasks; design of the steering Ackermann geometry and design of the front and rear anti-roll bars.

4.1.1) Design of the steering Ackermann geometry

The design of the steering Ackermann geometry began by comparing the new configuration with the baseline results from the 2024 Chulalongkorn University Formula SAE team's previous car. To isolate the effects of the Ackermann changes, the front roll rate distribution was adjusted to maintain a similar total lateral load transfer distribution. This adjustment ensured that any observed differences in vehicle behavior across various cornering scenarios were solely attributed to the modified Ackermann geometry, rather than changes in load transfer distribution.

This study revealed that adopting a more anti-Ackermann steering geometry—where the inside wheel turns less than the outside wheel—results in increased grip in both limit behavior and steady-state cornering. However, this comes at the cost of reduced control during corner entry and a lower maximum yaw moment, which negatively impacts maneuverability in slalom sections. Example of the result shown in Figure 21 below.

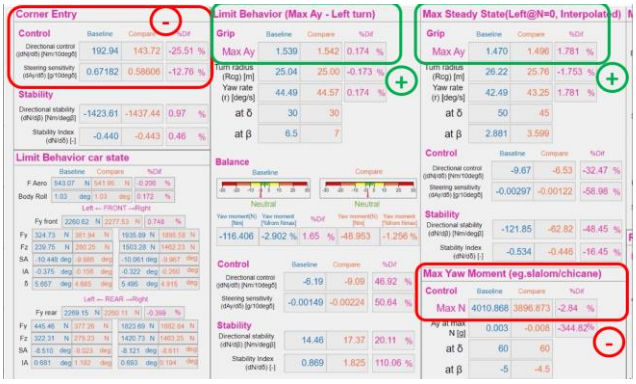

Fig. 21: Result KPI from during designing of Ackermann geometry with baseline being previous year car design

Given that Formula SAE circuits feature numerous tight corners and slalom sections, the trade-off between increased grip and reduced control was carefully evaluated. As a result, the final Ackermann geometry was designed to provide a balance between these factors. Compared to the previous year's car, the new design slightly reduces the inside wheel's steering angle but still maintains a conventional Ackermann configuration, where the inside wheel turns more than the outside wheel. This approach preserves control while leveraging the additional grip to enhance performance in events such as the skid pad test.

4.1.2) Design of the front and rear anti-roll bars.

The design of the anti-roll bars focuses on evaluating the influence of roll rate distribution to establish an optimal adjustment range for both the front and rear anti-roll bars. Within this adjustment range, it is crucial to ensure that the vehicle maintains stability, particularly at higher speeds.

To achieve this, the stability index is analyzed across multiple input speeds, allowing for a thorough assessment of the car's handling characteristics under varying conditions. Careful consideration is given to balancing roll stiffness distribution to avoid excessive understeer or oversteer tendencies, which could compromise vehicle stability.

An example of the stability index diagram comparison is provided in Figure 22, 23, illustrating how different roll rate distributions affect the car's stability at different speed.

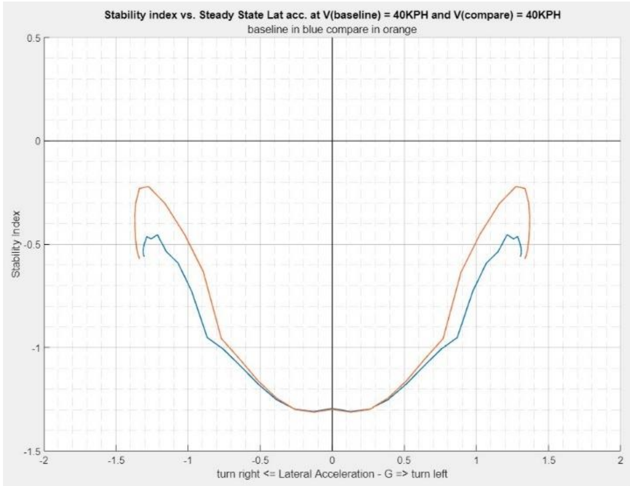

Fig. 22: Stability index comparing roll rate distribution from 55% in blue to 40% in orange at 40KPH

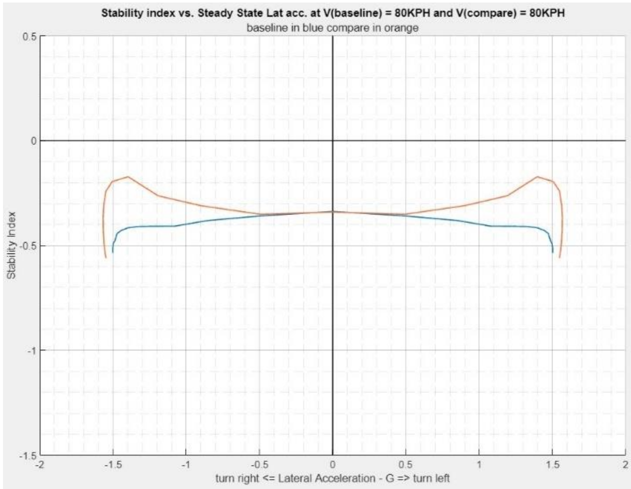

Fig. 23: Stability index comparing roll rate distribution from 55% in blue to 40% in orange at 80KPH

Through multiple design iterations and analysis, an optimal adjustment range for the anti-roll bars was determined. This process involved assessing the influence of roll rate distribution on vehicle balance and handling characteristics, using reference data from the previous year's car to establish a baseline for comparison.

4.2) Application of the tool during the testing phase of an FSAE car

The developed vehicle dynamics tool has also been used in optimizing the setup of the 2025 Chulalongkorn University Formula SAE team's car during testing. Given the limited track time available at Pathumthani Speedway this year, maximizing efficiency in testing and setup adjustments was crucial.

By utilizing this tool, the process of optimizing the vehicle setup became more systematic and quantifiable. Instead of relying solely on driver feedback and trial-and-error adjustments, the tool allowed for data-driven tuning, enabling informed decisions on setup changes such as roll bar adjustment, static toe angle, and camber angle.

This structured approach significantly reduced the number of on-track iterations required, helping the team make the most of the available test sessions. By simulating and predicting vehicle behavior before making physical adjustments, the tool ensured that each test run provided maximum insight into performance improvements.

The testing was conducted primarily on two track configurations: a Formula SAE competition skid pad and a mock FSAE autocross track, as shown in Figures 24 below. These test environments were selected to evaluate both steady-state and transient vehicle dynamics, ensuring a comprehensive assessment of the car's handling characteristics.

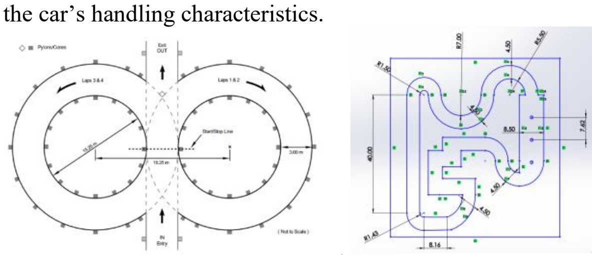

Fig. 24: Formula SAE competition's skid pad layout [8] and a mock FSAE autocross track used during testing

Through optimizing the simulated key performance indicators (KPIs) and correlating them with driver feedback, we were able to determine an optimal vehicle setup. This setup ensures that the driver feels confident in car control and stability while still maintaining high grip at the limit.

4.3) Result of 2025 Chulalongkorn University Formula SAE team's car

By integrating this tool into both the design and testing phases of the 2025 Chulalongkorn University Formula SAE team's car, its benefits were clearly demonstrated in the final competition results.

This year, the Chulalongkorn University Formula SAE team achieved first place in the skid pad event, recording a final time of 22.89 seconds—a significant improvement compared to last year's 24.24 seconds, despite having less practice time. Additionally, the team also secured first place in the autocross event, further showcasing the effectiveness of this tool in optimizing vehicle performance.

Unfortunately, an unexpected engine problem during the endurance event prevented the team from securing the overall first-place position. However, the results clearly highlight that utilizing this tool can significantly enhance competition performance by enabling data-driven design improvements and more efficient testing processes

5. Future Work

While the tool has demonstrated significant benefits, there are still several areas for improvement to further enhance its usability and accuracy.

1. Improved Input System – Streamlining the input of vehicle parameters to enhance accessibility. For example, instead of directly entering the total roll rate, it will be automatically calculated based on the inputted roll bar setup and selected spring stiffness. This approach simplifies data entry for users with limited vehicle dynamics knowledge.
2. Automatically Adjusting Simulation Setup – Simplifying the process by eliminating the need for manual adjustments of simulation parameters, such as the range of steering angles (δ), vehicle slip angles (β), and the relaxation parameter (p_r). This automation enhances user-friendliness and streamlines the setup process.
3. Improved Result Interpretation – Making the outputs more intuitive and easier to relate to lap time performance would help streamline the decision-making process during setup optimization. Developing clearer visualizations and comparative metrics could enhance usability for the user.
4. Expanded Testing and Correlation Analysis – Conducting more extensive testing would allow for a deeper understanding of how changes in KPIs directly affect track performance. Establishing a stronger correlation between simulated metrics and real-world lap times would improve the tool's predictive capabilities.
5. Refinement of Tire Scaling Methods – Improving the tire model scaling would enhance the simulation's accuracy, especially in replicating tire behavior at the real track surface.
6. Inclusion of Longitudinal Acceleration – Incorporating longitudinal acceleration into the simulation enables a more comprehensive analysis of vehicle performance and drivability during acceleration and braking phases, allowing for deeper insights into dynamic behavior under varying conditions.

By addressing these areas, the tool can become even more effective in vehicle setup optimization and performance prediction.

6. Conclusion

This research successfully developed a comprehensive vehicle dynamics tool tailored for the design and testing of Formula SAE cars, focusing on optimizing both vehicle performance and drivability. By integrating the MRA Moment Method (MMM) with the Magic Formula 6.1 tire model and a comprehensive vehicle model, the tool provides a balance between simulation accuracy and usability, enabling in-depth analysis without the complexity of full dynamic simulations. The user interface also help streamlines data input and KPIs analysis, facilitating efficient design iterations and setup optimizations. The tool demonstrated its effectiveness during the development and testing of the 2025 Chulalongkorn University Formula SAE car, contributing to improved results in key competition events. While future enhancements, such as improved result interpretation systems, automated simulation adjustments, and refined tire modeling, will further strengthen its capabilities, this tool already shown that it could fulfill the main objective of being able quantify and optimize vehicle performance and drivability for both during the design and testing of Formula SAE cars.

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