Technical University of Crete

Department of Production Engineering and Management

MASTER THESIS

Systems Engineering; From project goals to Verification and Validation for a formula student vehicle

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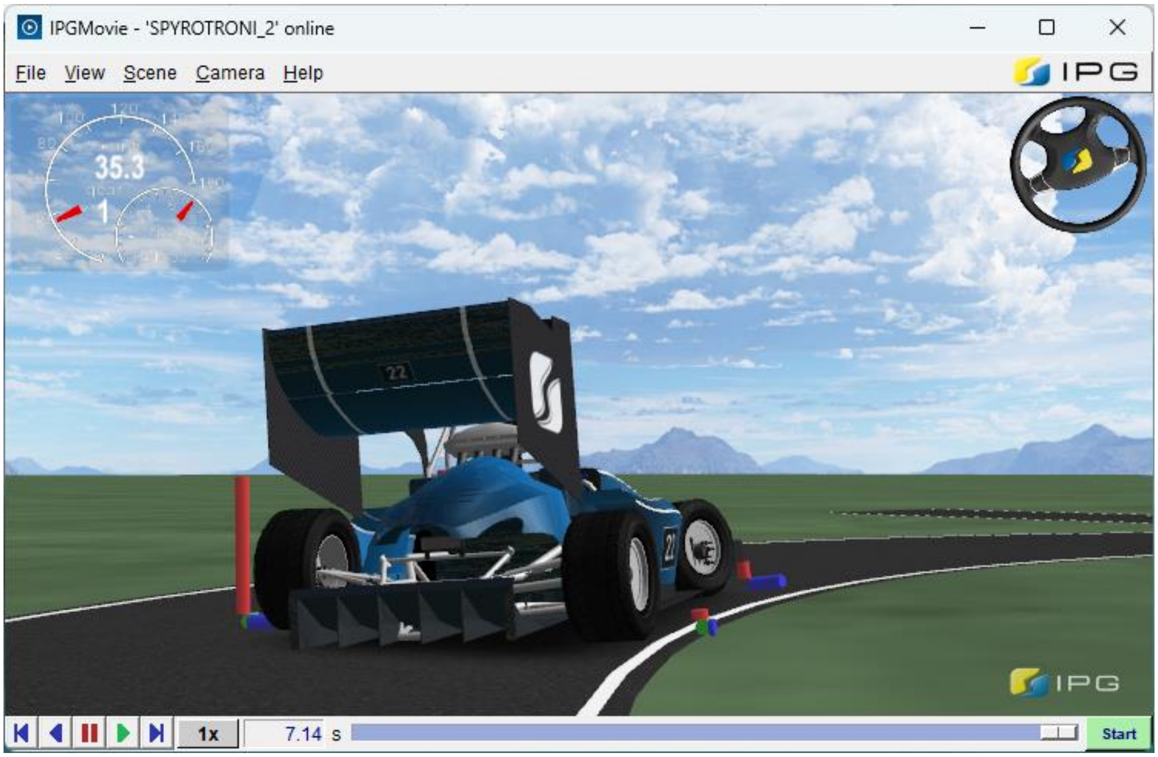
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Abstract

This thesis describes the methodology for developing a Formula Student (FS) vehicle based on Systems Engineering principles. It showcases a condensed version of an automotive Development Process focused on Vehicle Dynamics. It highlights key vehicle Goals, System, and subsystem level Requirements in the form of vehicle attributes and finally the Verification and Validation method. The aim is to guide teams and individuals working on vehicle prototypes, not just limited to Formula Student (FS) projects, where tools and methods are still evolving, and resources like time and budget are tight, which is typical in FS competitions.

The thesis also provides: a) an open source functional decomposition for propulsion, braking and low speed lateral dynamics in Google Sheets [29] where the user using First Principles can see the methodology in practice and b) the parameterization of a vehicle dynamics model.

Nomenclature		
SysEng: Systems Engineering	V&V: Verification and Validation	TUC: Technical University of Crete
CAD: Computer aided design	CAE: Computer aided engineering	Reqs: Requirements
FS: Formula Student	HW: Hardware	SW: Software

I. INTRODUCTION

University students are challenged to design, construct, and race miniature formula-style vehicles in FS, a prestigious international engineering competition. The competition, which is run by the Institution of Mechanical Engineers, combines practical experience with academic understanding, covering topics ranging from high-speed track performance to design and cost analysis. Teams from all over the world push the limits of innovation and technology in the combustion, electric, and autonomous vehicle categories. In addition to being a platform for industry relationships.

I. HYPOTHESIS

Our hypothesis is that utilizing SysEng processes in the design of a Formula Student (FS) vehicle will allow us to meet our primary objectives and goals. The key idea here is to identify and resolve issues early in the conceptual phase, as shown in [Fig. 1] with the SysEng V-model. By breaking down the system into detailed requirements and functions, we aim to ensure that fulfilling all lower-level requirements will lead to the fulfillment of higher-level ones. A crucial part of this approach is creating precise functional requirements and implementing an effective Verification and Validation (V&V) strategy. The project leader, in collaboration with the team, should be responsible for defining the functional requirements and creating verification methods[28].

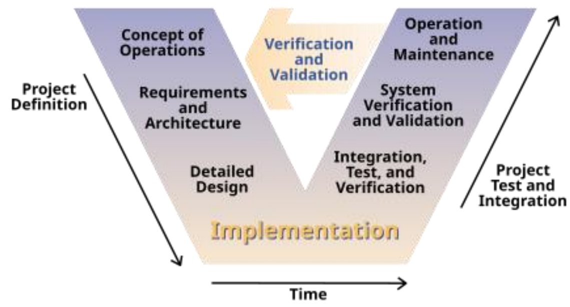


Fig.1 The V-model of the systems engineering process [1]. V&V feedback back to architecture and requirements.

II. STATE OF THE ART

A. Development processes

Automotive companies and Original Equipment Manufacturers (OEMs) use specific product development processes when creating new vehicle platforms worth billions of euros or dollars. Notable examples include Ford's Global Product Development System (GPDS) and Toyota's Production System (TPS) [4], with Toyota being the world's largest manufacturer until 2021 [3]. The GPDS, adopted by Volvo Cars [5], was originally modeled after Mazda's development system during Ford's ownership of both brands. TPS, which emphasizes "incremental kaizen (continuous improvement)," forms the basis of the "Toyota Way" [6], a philosophy focused on long-term value for customers and society, prioritizing customer satisfaction beyond short-term financial gains. Tesla, one of the most innovative companies of the 21st century, appears to have moved away from "decades-old lean manufacturing methods pioneered by Toyota" [7]. Regardless of the specific process used, the ultimate goal is the same: to maximize efficiency, foster company growth, and promote social development as a result.

B. Systems engineering

The common denominator in Development processes is to establish the System level Reqs subject to core system-product goals (i.e., product cost and timing, risk targets) and constraints (i.e. regulations, environment of operation). There exist a handful of commercial Reqs management suites. Some known examples are IBM Engineering Doors, Jama Connect and Siemens Polarion, etc. [8]. The main objective of these SW suites is to have i) central product-company control Reqs management to facilitate collaboration and ii) requirement decomposition and traceability, impact analysis [9] etc. Due to budget constraints in this thesis, the requirement decomposition process was conducted using. Google Sheets instead of specialized requirements management software. Google Sheets provides a cost-effective alternative for managing and tracing requirements in smaller projects according also to valiance[25].

C. *Vehicle Dynamics and Suspension design*

The use of mathematical optimization techniques for vehicle dynamics is well-established in both academia and industry [10][11][12]. Near-optimal designs can be achieved when specific suspension attributes are targeted [13][14][15][16], whether through true mathematical optimization or genetic algorithms . With accurate modeling of vehicle dynamics and suspension, along with verified tools and methods, it is possible to design a vehicle and validate its software for road use without ever physically testing the tires on the road [17][18][19].

III. OBJECTIVES

Summarizing the information provided, the two main objectives of this thesis are:

1. The primary goal is to summarize the development process for designing the vehicle dynamics and suspension of a Formula Student (FS) car by.

- selecting key objectives within the constraints of rules, budget, time, and competition, by analyzing what top-performing FS teams achieve in terms of design and performance.
- aiming to identify the high-level requirements for vehicle dynamics and chassis.

2. Enable virtual development (use CAE and CAD) by performing a series of development and V&V steps from Static, Kinematic and Dynamic events.

IV. METHODOLOGY

Our Objectives can be met with 5 work packages (WP)

1. *WP1 First principles.*

Set vehicle level goals

Set performance goals (i.e. Table I)

- Actuation performance (engine-transmission, steering and brakes selection)
- Dynamics performance (steady state cornering, acceleration, and deceleration)

Set design targets subject to the performance goals

- Dimensions (wheelbase, track width, weight, moment of inertia), CG location
- Select wheels and tires
- Suspension static/kinematic performance static camber/toe, steering ratio, Ackerman configuration, ride rates/roll stiffness
- Powertrain; peak torque-speed and transmission

The goals and targets are broken down into system-level requirements. For example, to achieve a 0 to 100 km/h acceleration target, the powertrain system (engine and transmission) must be appropriately sized, and the vehicle's overall grip (tire performance and load transfer effects) must be sufficient to meet this goal (see Table 1for an illustration).

TABLE 1 Example of vehicle level goals (middle row) and Design Targets

Decomposition of Goal1 to system level Reqs (bottom row) subject to Design Targets. Meeting the traction, weight distribution and propulsion performance will allow to meet the Goal1.
Goal1: *Acceleration: 0 to 100 km/h [s]: 7* DesignTarget1: “The vehicle mass shall be less than 300 kg” DesignTarget2: “The vehicle shall be rear wheel drive”
Reqs decomposition to meet Goal1 subject to DesignTarget1 and DesignTarget2. [Req0] The vehicle longitudinal acceleration from 0 to 100 km/h shall be less than or equal to 7 s. [Req1] The average vehicle acceleration on wide open throttle between 0 to 100 km/h shall be greater than 3.97 m/s^2. [Req1a] The vehicle longitudinal traction from the tires shall allow the vehicle to achieve 1585N. [Req1b] The vehicle’s rear axle tires shall allow peak longitudinal traction equivalent to 7.29 m/s^2. [Req1c] The vehicle center of gravity location shall load the rear axle with at least 62% of the vehicle weight during a static longitudinal acceleration greater than 4.5 m/s^2. [Req1d] The average propulsive force shall be greater than 2500 N.

This decomposition has been taken place at the Google Sheet document that you can find the link at the end of the thesis. Here you can see an example of its functionality.

∉ Step1: we have to set our target

∉ Step2: fill in the constrains of our car and then using the right forms the file calculates the goals. You can see that on the figure down to the appendix.

The top speed target is also broken down into system-level requirements. To achieve a desired maximum speed, the powertrain system (engine and transmission) must deliver sufficient power, while the vehicle's aerodynamic drag and rolling resistance should be minimized. Additionally, the tire specifications and load distribution must be optimized to ensure stability and traction at high speeds. These factors together ensure the vehicle can reach and maintain the specified top speed goal. (see Table 2 for an illustration).

TABLE 2 Example of vehicle level goals (middle row) and Design Targets

Decomposition of Goal2 to system level Reqs (bottom row) subject to Design Targets. Meeting the traction, weight distribution and propulsion performance will allow to meet the Goal1.
Goal2: *Top Speed 180km/h(50m/s)* DesignTarget1: "The vehicle mass shall be less than 300 kg" DesignTarget2: "The vehicle shall be rear wheel drive"
Reqs decomposition to meet Goal1 subject to DesignTarget1 and DesignTarget2. [Req0] The vehicle aerodynamic drag force shall be less or equal to 1601. [Req1] The average vehicle transmission losses should be less or equal to 25%. [Req1a] The vehicle rolling resistance should be less or equal to 0.015 creating a rolling resistance of 54.44 . [Req1b] The vehicle's transmission efficiency should be 2.69. [Req1c] The vehicle's torque distribution in RWD must be 45.9KW

The gradeability target is broken down into system-level requirements. To achieve a specified gradeability, the powertrain system (engine and transmission) must generate enough torque to overcome the incline's resistance. The drivetrain's gearing, along with the vehicle's weight distribution and tire grip, must also be optimized to maintain traction on steep slopes. Additionally, vehicle aerodynamics and load transfer during inclines should be considered to ensure that the vehicle can climb the grade while meeting performance and stability requirements. (see Table 3 for an illustration).

TABLE 3 Example of vehicle level goals (middle row) and Design Targets.

Decomposition of Goal 3 to system level Reqs (bottom row) subject to Design Targets. Meeting the traction, weight distribution and propulsion performance will allow to meet the Goal1.
Goal3: *Grade Ability Target 20 grade angles* DesignTarget1: "The vehicle mass shall be less than 300 kg" DesignTarget2: "The vehicle shall be rear wheel drive"
Reqs decomposition to meet Goal 3 subject to DesignTarget1 and DesignTarget2. [Req1] The required gradeability Fg = gravitational forve component along the incline should be 3313N. [Req1a] The vehicle tire traction Ft while ascending the grade should be 1777N [Req1b] The vehicle's force on the rear axle should be 1247.70N.

The brake efficiency target is translated into system-level requirements to ensure optimal braking performance. This involves sizing the braking system (brakes, calipers, and rotors) to provide sufficient stopping power while maintaining stability. The vehicle's weight distribution, tire grip, and load transfer during deceleration are critical factors to achieve efficient braking. Additionally, heat dissipation and brake fade resistance must be considered to maintain consistent performance under prolonged or high-intensity braking conditions. These factors work together to ensure the vehicle meets its braking performance goals effectively. (see Table 4 for an illustration).

TABLE 4 Example of vehicle level goals (middle row) and Design Targets.

Decomposition of Goal 4 to system level Reqs (bottom row) subject to Design Targets. Meeting the traction, weight distribution and propulsion performance will allow to meet the Goal1.
Goal2: Brake distance target for braking distance 17m DesignTarget1: "The vehicle mass shall be less than 300 kg" DesignTarget2: "The vehicle shall be rear wheel drive"
Reqs decomposition to meet Goal1 subject to DesignTarget1 and DesignTarget2. [Req0] The vehicle deceleration should be higher or equal to 11.76 m/s^2 [Req1] The vehicles fundamental brake elements shall provide sufficient brake force at least 4352 N [Req1a] the tire traction shall allow the vehicle to achieve his deceleration rate so should be at least 5220 N [Req1b] The vehicle's weight transfer should be 1033 N [Req1c] The vehicle's aerodynamics shall contribute to braking by providing additional downforce of 34N

Brake proportioning refers to the distribution of braking force between the front and rear wheels of a vehicle. Since the front wheels typically bear more load during braking due to weight transfer, they require a greater share of the braking force compared to the rear wheels. Proper brake proportioning is critical for maintaining vehicle stability, preventing wheel lockup, and ensuring optimal stopping performance. If too much braking force is applied to the rear wheels, the vehicle may become unstable, leading to oversteer or loss of control. Conversely, insufficient braking force at the rear can lead to extended stopping distances. Engineers use proportioning valves or electronic brake force distribution (EBD) systems to fine-tune this balance, ensuring safety and performance in various driving conditions. (see Table 5 for an illustration).

TABLE 5 Example of vehicle level goals (middle row) and Design Targets.

Decomposition of Goal 5 to system level Reqs (bottom row) subject to Design Targets. Meeting the traction, weight distribution and propulsion performance will allow to meet the Goal1.
Goal2: Brake proportioning DesignTarget1: "The vehicle mass shall be less than 300 kg" DesignTarget2: "The vehicle shall be rear wheel drive"
Reqs decomposition to meet Goal1 subject to DesignTarget1 and DesignTarget2. [Req0] The brake proportioning system shall ensure balanced braking force distribution to maintain vehicle stability Fbf/Fbr=Wf/Wr 0.82 [Req1] The front and rear brake forces shall be proportioned to maximize braking efficiency without causing wheel lockup. Fbmax=μ*W where W=the normal force 1958 N [Req1a] The brake proportioning system shall adjust the braking force dynamically based on load and road conditions. Fb,dynamic=a*Fb where a is a factor determined by load and road condition data 2992 N [Req1b] **The braking force on the rear axle shall not exceed the traction limit to prevent rear wheel lockup. Fb,r,limit<=μr*Wr where μr is the rear tire-road friction coefficient and Wr is the rear axle weight** 1360 N

The swept path width is defined by the vehicle's maximum lateral clearance during cornering or maneuvering. Achieving a specific target for swept path width requires balancing the vehicle's overall width, wheelbase, and steering geometry. Minimizing the swept path helps improve maneuverability, especially in tight spaces, while ensuring stability during turns. It's essential to consider suspension movement and tire deflection to maintain control without exceeding the design constraints, especially in competitions or urban environments. (see Table 6 for an illustration).

TABLE 6 Example of vehicle level goals (middle row) and Design Targets.

Decomposition of Goal 6 to system level Reqs (bottom row) subject to Design Targets. Meeting the traction, weight distribution and propulsion performance will allow to meet the Goal1.
Goal2: Swept path width of 2 m DesignTarget1: "The vehicle wheelbase is 1600mm" DesignTarget2: "The vehicle trackwidth 1200mm"
Reqs decomposition to meet Goal1 subject to DesignTarget1 and DesignTarget2. [Req0] The vehicle inner turning radius shall be minimized to 938.04mm [Req1] The vehicles outer turning radius shall be optimized to 2138.04mm [Req1a] The difference between the inner and outer turning radius shall be 1200mm [Req1b] The vehicle's static camber shall be set to 1.15 [Req1c] The vehicle's king pin inclination shall be optimized to reduce the scrub radius to 0.03

The turning circle, or turning radius, refers to the minimum space required for the vehicle to make a full turn. Meeting the turning circle target involves optimizing steering geometry, including the steering rack, wheelbase, and suspension design. A smaller turning circle enhances the vehicle's agility, making it easier to navigate tight spaces and improving handling in low-speed maneuvers. Proper balance between the steering components and vehicle dynamics is key to achieving a precise and efficient turning circle. (see Table 7 for an illustration).

TABLE 7 Example of vehicle level goals (middle row) and Design Targets.

Decomposition of Goal 7 to system level Reqs (bottom row) subject to Design Targets. Meeting the traction, weight distribution and propulsion performance will allow to meet the Goal1.
Goal6: Turning circle target 2.2m DesignTarget1: "The vehicle wheelbase is 1600mm " DesignTarget2: "The vehicle trackwidth 1200mm "
Reqs decomposition to meet Goal 7 subject to DesignTarget1 and DesignTarget2. [Req0] The vehicle's Ackermann inner should be 45.02 degrees [Req1] The vehicle's Ackermann inner should be 29.75 degrees [Req1a] The vehicles steering arm length should be 200mm [Req1b] The vehicle's steering rack length should be 162.86 mm at least

Instrumentation and measurement

We require an instrumentation system to validate the development tools (see a good example in [21]) and assess performance relative to the requirements. This system should, at a minimum, capture the following data and states:

∉ Body

- 3-axis body accelerations and rotational rates
- Longitudinal and lateral speed (body slip angle)

∉ Chassis

- 4x wheel speed sensors
- 4x Strain gauges at wishbones to measure loads per wheel (c.f. [2] for strain gauge measurement and conditioning), suspension travel sensors

∉ Driver controls

- Steering angle, throttle and clutch travel
- Brake travel and pressure

∉ Powertrain

- Torque and gear position request
- Actual gear position, estimated delivered torque and engine RPM

Overall principles for instrumentation should involve signal conditioning to prevent anti-aliasing and the logging needs to be ~ 100 Hz for inertial signals and wheel speeds and ~ 500 Hz for strain gauge measurement. Live connectivity-telemetry is always a plus [28].

Due to budget constraints for this year's vehicle, we were unable to acquire all the necessary sensors for the instrumentation system. As a result, this section remains theoretical and cannot be fully analyzed or verified through actual data collection. The lack of real-time sensor data prevents us from conducting a detailed performance assessment, and any conclusions drawn from this analysis should be considered speculative rather than based on practical testing.

2. *WP2 Systems Engineering and Vehicle Dynamics Simulation*

A. Core attributes for the dynamic events

Table 8 outlines the core dynamic attributes of the vehicle, along with detailed definitions and methods for calculating these metrics, as referenced in [22]. For example, the understeer response is commonly evaluated by having the vehicle increase speed on a fixed radius while adjusting the steering angle to stay on course. By decomposing vehicle-level goals into specific system-level requirements, it becomes easier to analyze and optimize individual vehicle dynamics rather than focusing solely on overall performance metrics, such as lap times during an autocross event. This method improves the traceability of development changes, making it easier to identify the source of performance improvements or issues. This strategy follows a systems engineering approach, like dynamic programming, where a larger problem is broken down into manageable subproblems to facilitate more precise control over the design process. Original Equipment Manufacturers (OEMs) often use this approach in vehicle development, as it provides greater clarity and control when fine-tuning complex systems like vehicle dynamics and handling characteristics.

Table 8 core attributes for dynamics events

Propulsion

- Targets: top speed, acceleration, grade-ability, range
- Constraints: engine torque/power/efficiency/ consumption, transmissions ratio/efficiency

Braking

- Target: braking distance/efficiency
- Brake proportioning

Driver feel

- Pedal Response, Steering system forces
- Vehicle handling

Low dynamics (<2 m/s^2)

- Turning circle, Swept path width, Chassis steering geometry (Ackerman error)

Medium and high dynamics (>2 m/s^2)

- Understeer response, Neutral steering point, Normalized required steering angle, Critical and Characteristic speed, Steady state cornering gains
- Yaw rate gain, curvature gain, lateral acceleration gain, Side slip gain as function of speed
- Yaw and lateral acceleration response

Suspension

- Roll stiffness distribution between axles, Steering and suspension compliances.
- Bump, roll steer, static camber, toe angle
- Kingpin inclination angle (caster and scrub radius) wheel offsets

Peak handling

B. Vehicle Dynamic Simulation Suite (VehDynSim)

To effectively model and analyze vehicle dynamics, it is essential to choose a suitable simulation suite or develop a custom tool tailored to the specific needs of the project. One popular option is IPG CarMaker, which has become a key tool in the Formula Student competition, where it serves as a comprehensive simulation platform. As a "golden" sponsor of the event, IPG CarMaker provides a dedicated Formula Student package, offering teams access to an advanced and complete setup for vehicle dynamics simulation [23].

The primary objective of using a simulation suite like IPG CarMaker is to accurately model the behavior of the vehicle under various conditions. This includes detailed simulation of core aspects such as suspension kinematics, tire forces, powertrain dynamics, and vehicle handling characteristics. By simulating dynamic events such as acceleration, braking, and cornering, these tools enable engineers to evaluate the performance of the vehicle in a virtual environment before physical testing takes place. The right simulation suite not only ensures high accuracy in predictions but also allows for the optimization of key design elements, leading to a more refined and high-performance vehicle by the time it reaches testing and competition stages.VehDynSim is to model:

1. The Vehicle

   i) Body (masses and dimensions), ii) Suspension (suspension hardpoints), iii) Suspension force elements (springs dampers)

2. The dynamics events (c.f. [20] pp. 164)

i) Autocross, ii) SkidPad, iii) Acceleration, iv) Efficiency, v) Endurance

C. Postprocessing interface with VehDynSim

To facilitate the analysis of the Vehicle Dynamics Simulation (VehDynSim) results, we need to develop a robust post-processing software interface. This interface will allow the simulation to run scripts that process the data and adjust configuration parameters, such as different suspension settings. By doing so, it will automate the interpretation of vehicle state information, using the output from VehDynSim to calculate key performance metrics for the core dynamic attributes outlined in table 8.

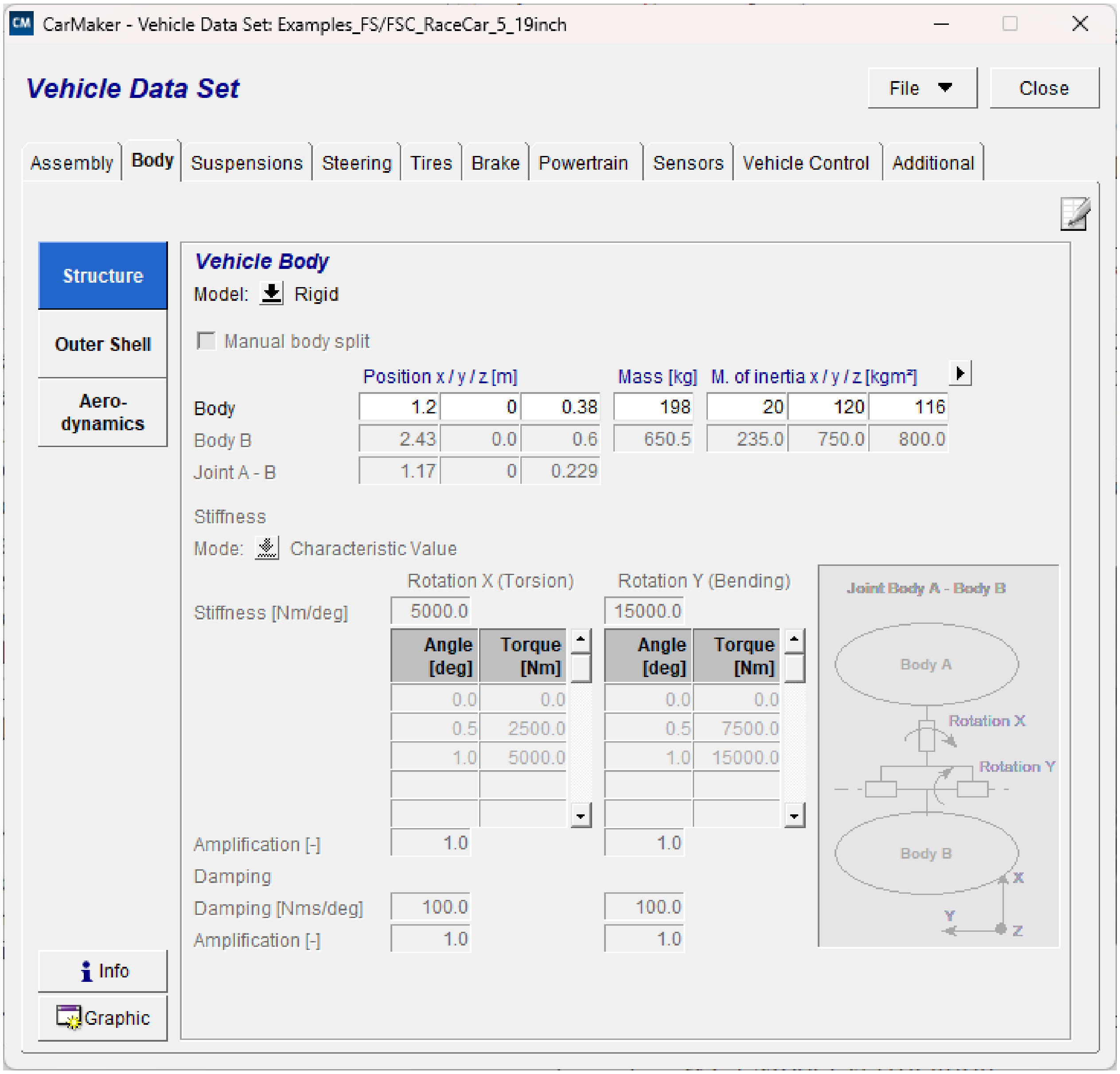


Fig.2 Body inertial properties.

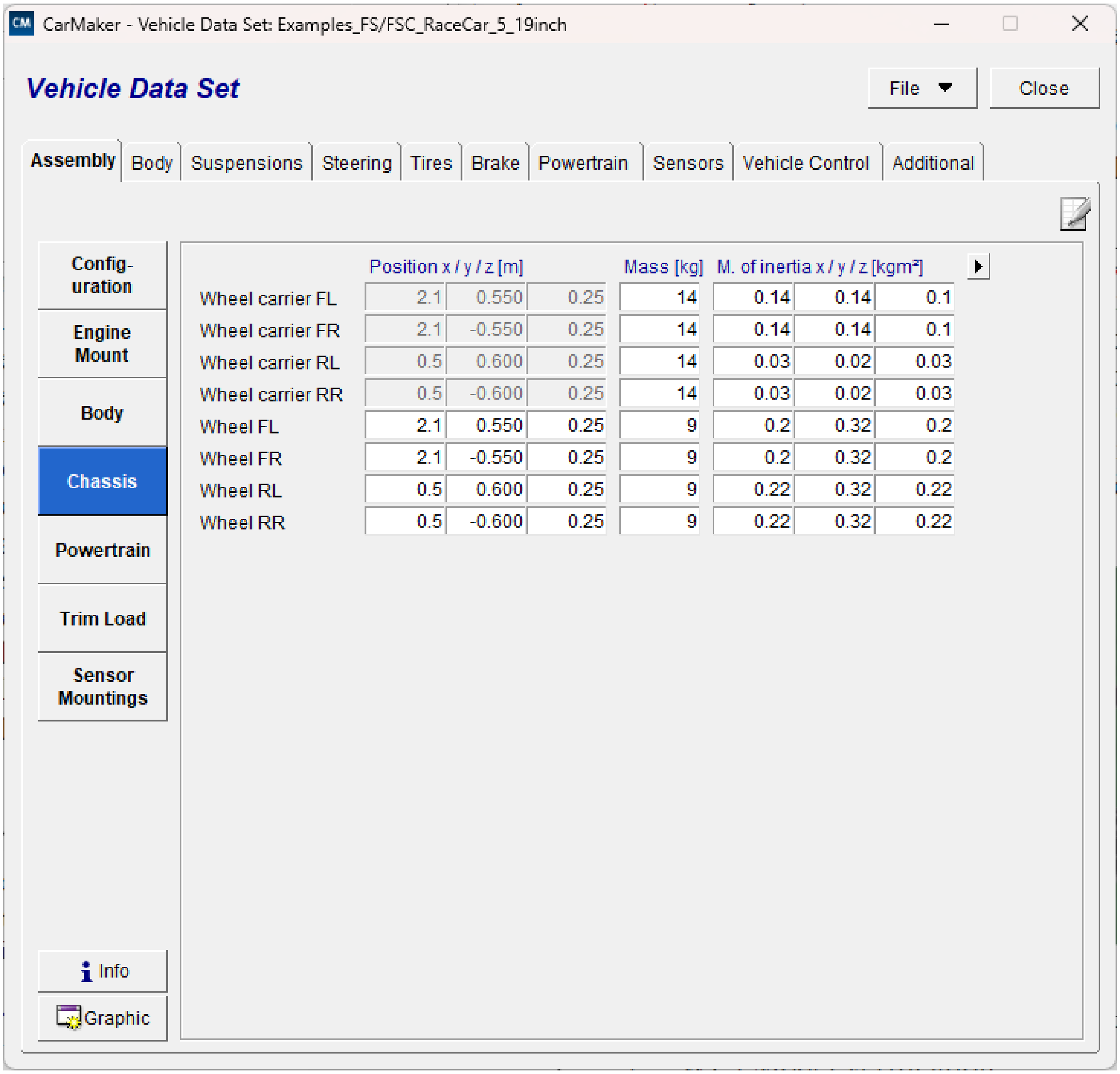


Fig.3 Body geometric properties.

The post-processing will be implemented using the MATLAB environment, a widely-used platform for engineering and data analysis. MATLAB's ability to handle large datasets and its extensive library of built-in functions make it an ideal tool for calculating performance metrics from simulated data. Additionally, MATLAB's scripting and automation capabilities allow for seamless integration with simulation tools, enabling efficient and repeatable post-processing workflows. Using MATLAB, we can script custom algorithms to interpret the outputs of VehDynSim, adapting variables like suspension geometry, tire forces, and powertrain configurations to optimize vehicle performance [34]. This approach ensures that all design changes are assessed quantitatively, providing actionable insights that can guide further refinement of the vehicle's dynamics.

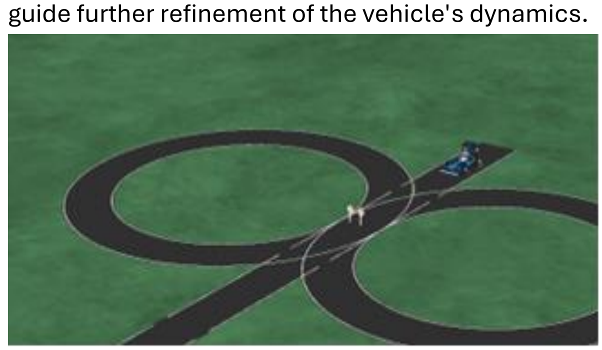


Fig.4 Skidpad dynamic events; instance IPG Formula CarMaker [23].

```
Current_Folder = pwd;
%    https://www.mathworks.com/matlabcentral/answers/376645-how-to-use-a-function-that-is-not-in-the-same-folder-as-your-current-folder
disp('Current_Folder')
TopFolder = fileparts(pwd);
disp(TopFolder)
Top_TopFolder = fileparts(fileparts(pwd));
disp(Top_TopFolder)
addpath(genpath('FunctionsSK' ))
%%
% Open the model for this example
open_system('generic.mdl')
%%
cmguicmd('LoadTestRun "dna/SteerStep__"');
%%
cmguicmd('IFileFlush')
%%
for i=1
cmguicmd('StartSim',0) % pp. programmers guide 1261
%
RunName='SimRun.erg';
LogPath='..\SimOutput\SPYROTRONI_2\';
LogName=[LogPath RunName];
%%
a=cmread(RunName);
% analyzeME(a)
```

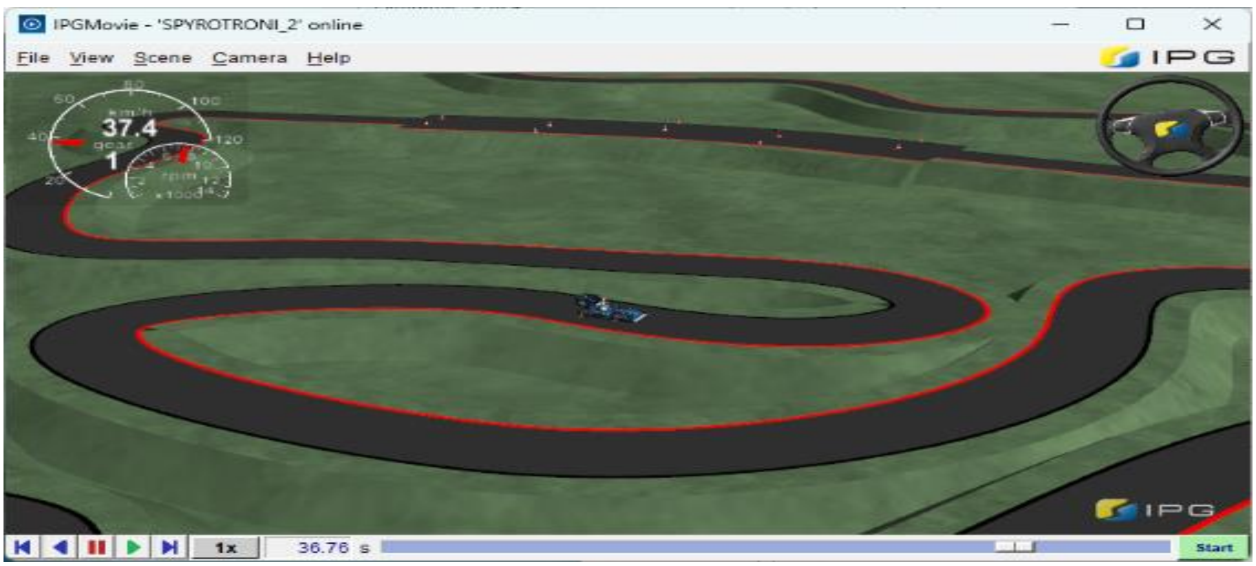


Fig.5 Autocross dynamic events; instance IPG Formula CarMaker [23].

3. *WP3 Model verification*

A. Static verification

A tool such as IPG CarMaker for vehicle dynamic simulation (VehDynSim) provides essential vehicle parameters, including track width, wheelbase, and corner-specific wheel loading. It is crucial that these parameters remain consistent across the CAD design, VehDynSim model, and the physical vehicle to ensure accuracy in both simulation and real-world performance. Therefore, an iterative process is required, refining both the CAD design and the VehDynSim model until the output from the simulation closely aligns with the physical vehicle's specifications, within reasonable engineering judgment.

Measurements on the real vehicle: To measure the vertical displacement of the wheel we attached a laser measurement, while lifting the wheel with a small crane. In every measurement of the wheel, we also measure the displacement of the damper with a steel vernier caliper.

B. Kinematic verification

Table 9 Measurement on real vehicle

Spring (mm)	Wheel (mm)
0	0
2	5
6	11
10	16
14.6	21
21	27
24.6	33
29.8	38

34.5	43
37	47

Motion ration = $\frac{Spring\ Displacement}{Wheel\ Displacemetn}$

Table 10 Motion Ratio

0.4	0.545	0.625	0.695	0.777	0.745	0.784	0.802	0.7873

The average Motion Ratio is 0.67. That means that for 100 mm the wheel moves the spring/damper moves 67 mm

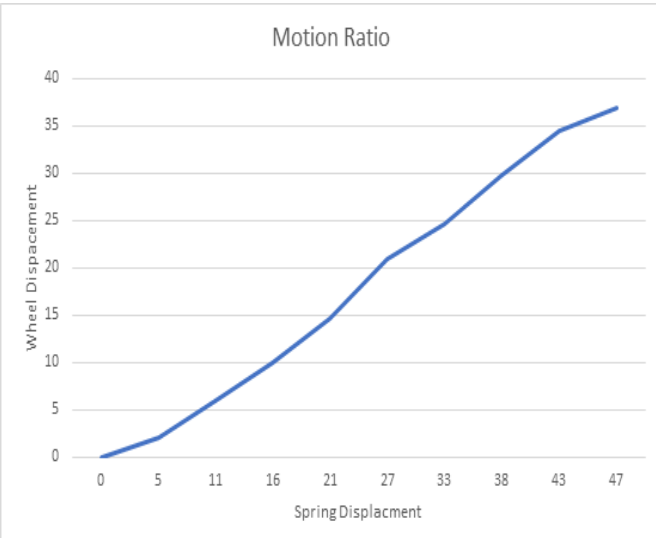


Fig.6 Motion ratio; wheel displacement vs spring displacement.

Cad Design

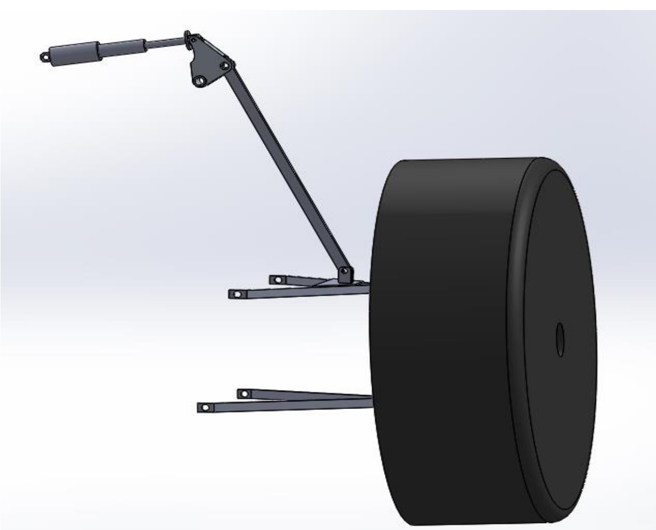


Fig.7 Cad Model of the Front Suspension

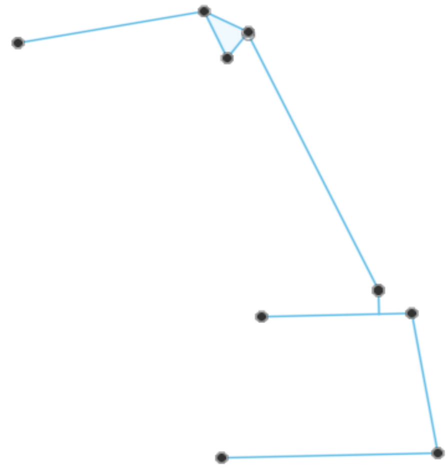


Fig.8 Suspension Geometry of Front Suspension two sensors were added on SolidWorks to measure the spring displacement and the wheel displacement.

Table 11 Measurements on the Cad Model

Spring (mm)	Wheel(mm)	Motion Ratio
3.52	3.98	0.884422
7.28	8.18	0.889976
11.19	12.48	0.896635
14.4	15.97	0.901691
18.19	20.05	0.907232
23.96	26.2	0.914504
30.51	33.12	0.921196

The Motion Ratio for each pair of values was calculated after the measurements.

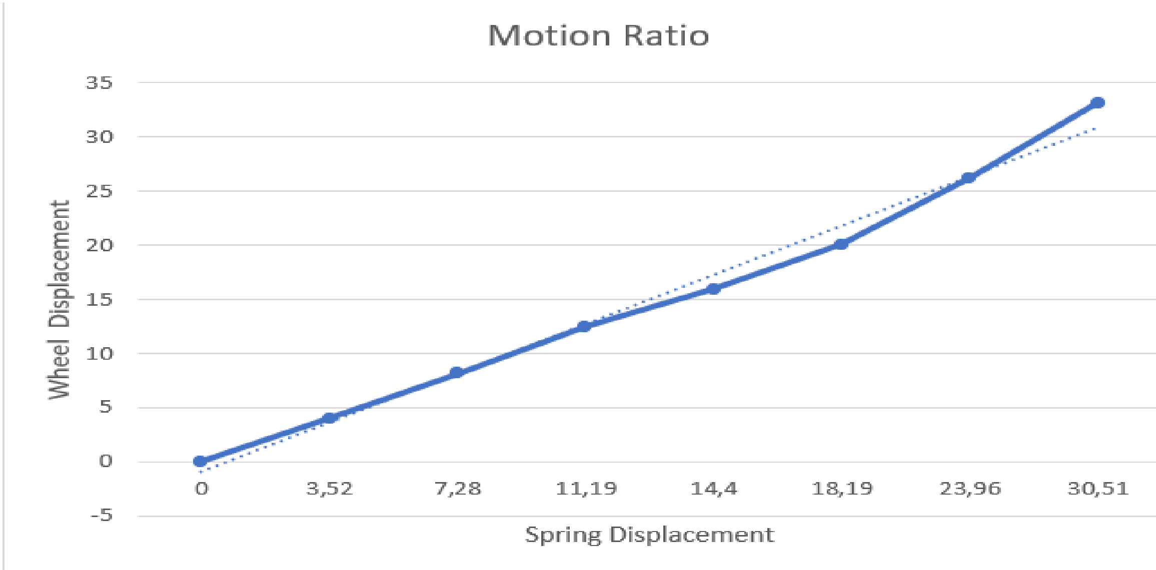


Fig.9 demonstrates a consistent motion ratio throughout the range of wheel travel, indicating uniform suspension response.

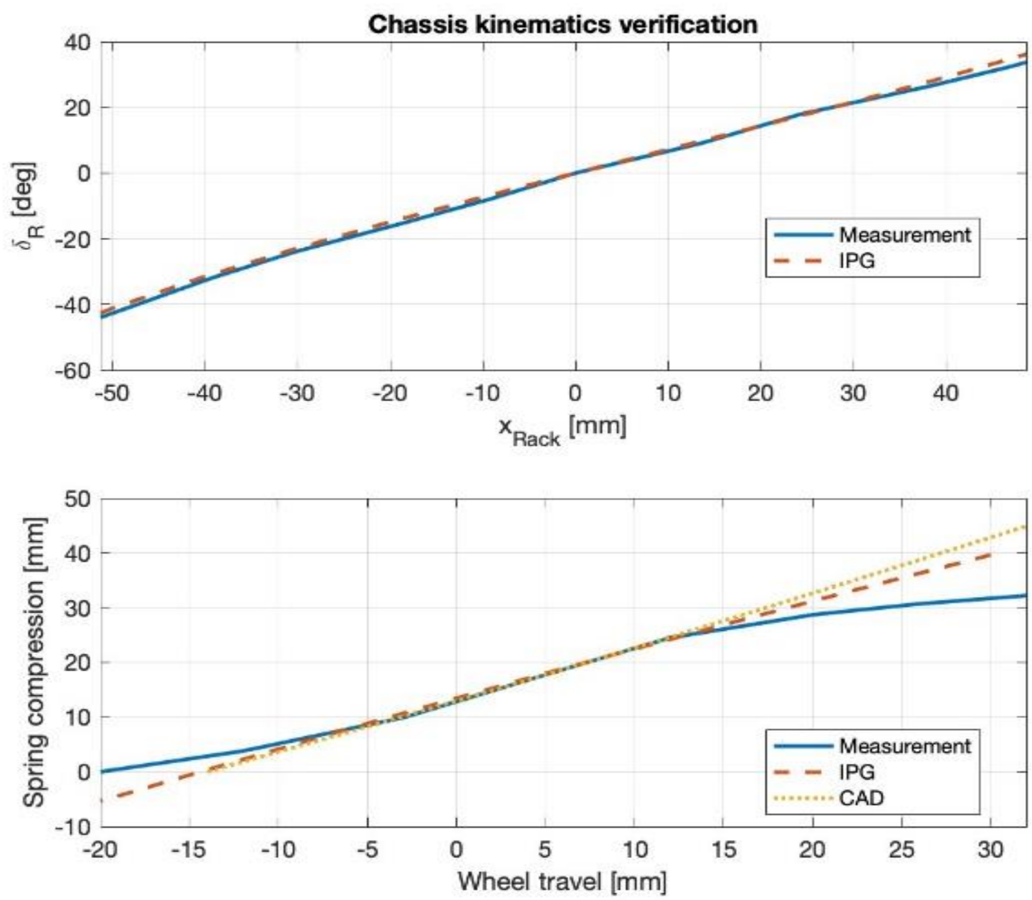


Fig.10 compares physical, CAD, and CAE models for chassis kinematic verification, demonstrating consistency between simulations and real-world behavior, thus validating the design's performance.

The graph above (fig 12) presents a comparison between the actual vehicle, the CAD design, and the VehDynSim model. Ensuring consistency in key parameters—such as track width, wheelbase, and wheel loading at each corner—is crucial for accurate simulation results. By iterating between the CAD design and the simulation model, we strive to achieve close alignment between all three representations. This process ensures that the dynamic performance predicted by the simulation is a reliable reflection of the physical vehicle, as illustrated by the refined alignment seen in the graph.

C. Dynamic verification

Establish a "core" verification catalog:

Include key maneuvers such as sine steer tests (e.g., 0.2 Hz), step steer inputs, low-speed cornering, and combined maneuvers (e.g., braking or accelerating while turning) with a resultant acceleration of less than 6 m/s^2.

Verify and refine the model iteratively: Compare the vehicle's state responses from the core catalog to ensure that the simulated results align with those of the physical vehicle for identical speeds and steering inputs (considering minimal lash and compliance). Focus on key vehicle dynamics data such as longitudinal acceleration (accx), lateral acceleration (accy), yaw rate, and body slip angle, as outlined in the instrumentation and measurement guidelines.

Evaluate vehicle state responses for Formula Student dynamic events: Dynamic events push the vehicle into the nonlinear performance range, with total combined accelerations exceeding 8 m/s^2. Due to this, the correlation between the simulation model and real-world data will likely be less accurate (fig 7).

We should continuously iterate between CAD design adjustments and the simulation model in VehDynSim to meet our dynamic verification criteria (pass/fail benchmarks to be defined).

D. Milestone 1: Completion of Vehicle Dynamics Model Verification

As described, the process of verifying the vehicle dynamics model involves three key stages: Static Verification, Kinematic Verification, and Dynamic Verification. These steps are iterative, meaning that adjustments are made to the model (model refinement) after each round of verification until the results are satisfactory. The process follows a specific sequence, starting with Static Verification, then moving to Kinematic Verification, and concluding with Dynamic Verification. Once all three stages have been successfully completed, we can confidently assert that the model accurately reflects the desired performance criteria.

4. *WP4 Load verification*

A. Simulated dynamic loading for components.

First Iteration of Suspension Dynamic Loading During VehDynSim Simulation

In this phase, the simulation focuses on calculating the following elements:

- The core verification catalog
- The dynamic events
- Durability events, which are carefully chosen for evaluation

This iteration helps ensure that the suspension system's behavior under dynamic conditions is thoroughly assessed and aligns with real-world durability requirements.

Simulation results on dynamic events

Acceleration Event

The acceleration graph depicts the dynamic event where the vehicle reaches its top speed within 7 seconds. The curve initially rises steeply, reflecting rapid acceleration as the vehicle gains momentum from a standstill. As time progresses, the rate of acceleration gradually decreases, indicating the influence of factors such as aerodynamic drag, engine power limits, and rolling resistance. By the 7-second mark, the graph levels off, showing that the vehicle has achieved its maximum speed and no longer accelerates. This plot signifies the equilibrium point where the forces propelling the vehicle forward balance out with the opposing resistive forces. Understanding this acceleration behavior is key for optimizing performance in dynamic events, allowing engineers to fine-tune powertrain settings and vehicle dynamics to maximize efficiency and speed within competitive constraints.

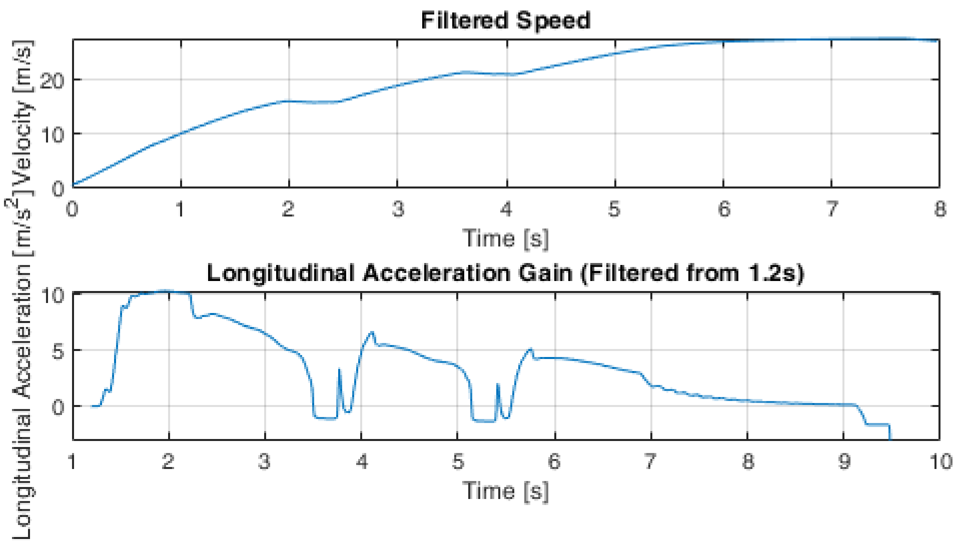
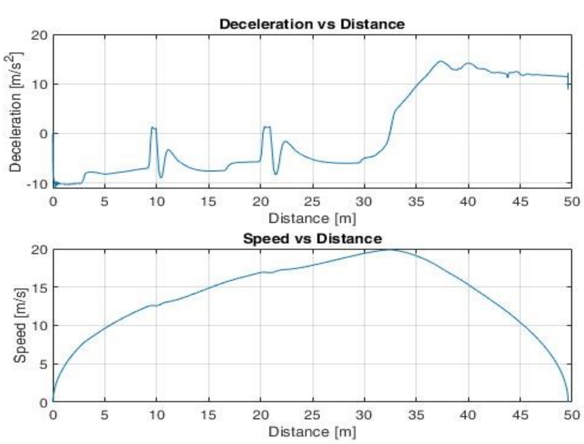


Fig.11 illustrates velocity and longitudinal acceleration over time, highlighting the vehicle's acceleration profile. The graph shows consistent acceleration behavior, validating the performance and stability of the vehicle's longitudinal dynamics.

Braking Event

The braking event graph shows how the vehicle decelerates from high speed to a complete stop. At the beginning of the event, there's a sharp drop in speed as the brakes are applied, causing rapid deceleration. This steep decline reflects the braking system's effectiveness, especially when the tires maintain good grip and the brake force is strong. As the vehicle slows down, the rate of deceleration gradually decreases, which may be influenced by factors like brake heat buildup, reduced tire traction, or modulation of the braking system. By the end, the graph flattens out as the vehicle comes to a stop. Understanding this braking behavior is essential for optimizing brake performance, improving safety, and achieving better stopping distances in dynamic driving conditions.



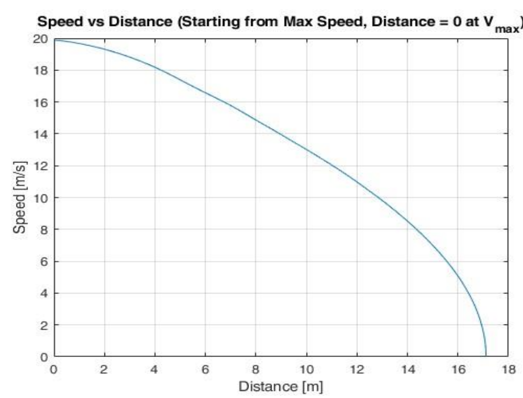
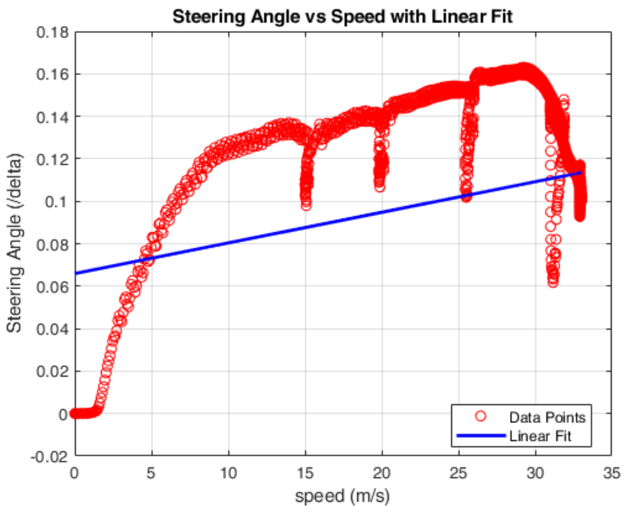


Fig.12 presents the braking simulation results, depicting body speed and longitudinal acceleration.

Steady State Circular event 100m

The steady-state circular event is a vital test for evaluating a vehicle's handling characteristics, particularly regarding understeer and oversteer. During this test, the vehicle is driven in a constant-radius circle at a fixed speed, allowing for analysis of its response to steering inputs. An understeering vehicle requires a greater steering angle to maintain its path, indicating that the front tires lose grip first. In contrast, an oversteering vehicle demands less steering input, suggesting the rear tires are losing traction. By collecting data on lateral acceleration, steering angle, and yaw rate, engineers can fine-tune suspension settings and overall vehicle dynamics to enhance performance and safety.




```
Understeer/Oversteer Gradient (Kus): 0.0014
Vehicle tends to understeer (Kus > 0).
```

Fig.13 shows the relationship between steering wheel angle and speed, used to determine the understeer gradient (Ku). This data reveals the vehicle's handling characteristics, indicating understeer or oversteer tendencies as speed increases.

On the skid pad maneuver

The skid pad maneuver is a critical test used to evaluate a vehicle's lateral grip and stability during cornering. In this controlled setting, the vehicle is driven in a circular path on a flat surface, allowing engineers to assess how well the tires maintain traction while under lateral forces. During the maneuver, drivers aim to reach the maximum lateral acceleration without losing grip, which provides valuable insights into the vehicle's handling characteristics, including its tendency to understeer or oversteer. The results from the skid pad test help in fine-tuning suspension settings and tire performance, ultimately contributing to enhanced safety and performance on the track.

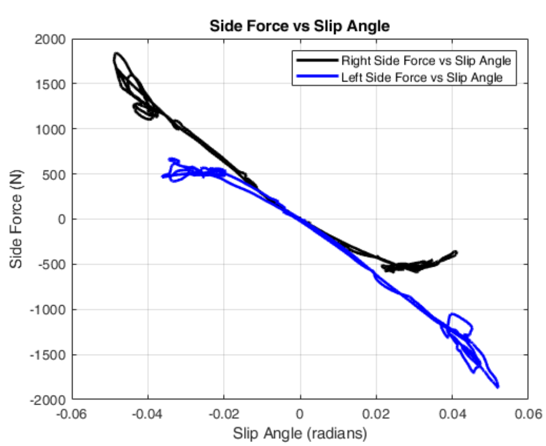


Fig.14 shows the relationship between side force and slip angle, assessing tire slip behavior and lateral grip during cornering.

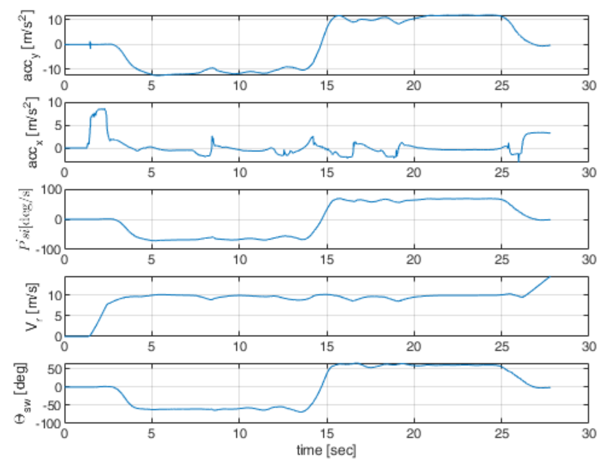


Fig.15 vehicle body states (top panel). Lateral accy and longitudinal accx acceleration, yaw rate dΨ/dt, resultant body speed Vr, and steering wheel angle Θsw.

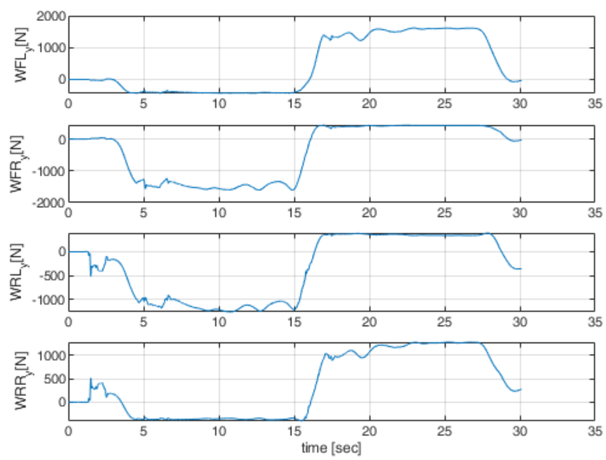
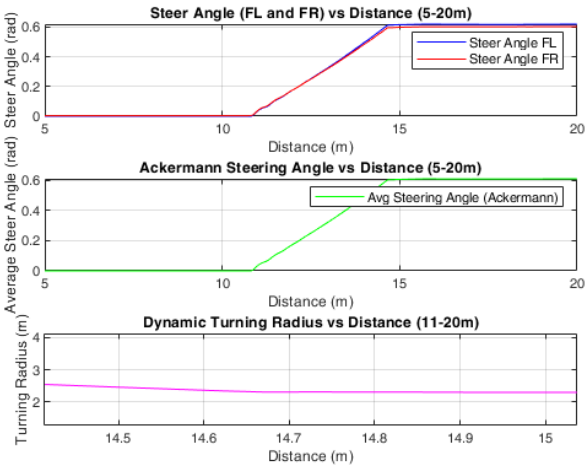


Fig.16 all forces at wheel center (bottom panel) for front left and right and rear left and right wheels.

On a steer step maneuver

The step steer maneuver is a key test to evaluate how a vehicle responds to sudden steering inputs, offering insight into its handling during quick transitions. In this test, the driver makes a rapid and fixed steering movement, simulating scenarios like a sudden lane change or obstacle avoidance. By observing the vehicle's reaction whether it understeers, oversteers, or stabilizes quickly engineers can gather important data about its steering behavior and handling. This test is especially valuable for fine-tuning the steering system, suspension, and stability controls, ultimately improving vehicle control and safety in real-world conditions.




```
Maximum Average Steering Angle: 0.61 rad at index 1451
Turning Radius at Maximum Average Steering Angle: 2.27 m
Turning Radius at Max Steering Angle: 2.27 meters
>>
```

Fig.17 shows steering angles and Ackermann geometry over distance, culminating in the dynamic turning radius in the third plot, highlighting the effect of steering input on turning capability

5. *WP5 Design to Vehicle dynamics.*

At this WP, we should iterate through the vehicle dynamics attributes by adapting suspension and chassis design within reasonable bounds trying to reach our core attribute targets as-well-as improve performance on the dynamic's events.

● For the vehicle dynamics attributes, we should use the appropriate dynamic maneuvers For example, steady-state cornering is one such maneuver where the vehicle is driven in a constant radius circle, which allows us to measure lateral acceleration and examine how the suspension geometry affects the vehicle's balance between understeer and oversteer. Additionally, a slalom test could be performed to assess the vehicle's transient response and agility as it weaves between cones. These maneuvers simulate real driving conditions and provide insight into whether the suspension and steering system are meeting design expectations.

To optimize vehicle performance, it is crucial to focus on the dynamic events that are central to the Formula Student competition. These events are designed to challenge various aspects of the vehicle's dynamics, including handling, acceleration, and braking.

For instance, the skid pad event is a classic test of the car's lateral grip and understeer/oversteer balance as it drives in a figure-eight pattern. Performance in this event helps us fine-tune the stiffness balancing and adjust the dynamic toe to hit our understeer targets. Additionally, the autocross event evaluates both cornering and acceleration through a tight, technical course, demanding quick transitions between braking and turning. Performance in autocross can be improved by adjusting the suspension to handle sharper corners and by refining the dynamic toe links for quicker corner exits.

These competition-specific events are essential for validating and optimizing the vehicle's design, as they provide a real-world gauge of the car's dynamics and help identify areas where further improvements are necessary.

On Fig 20: Some vehicle dynamics attributes vs lateral acceleration (all have the same x-axes) for steady circle of 42 m (R42m) and 100 (R100m) meters simulation. The bottom subplot shows the understeer characteristics of the FS2024 "under" development vehicle. In the plotted form the vehicle is exhibiting first understeering and then oversteering behavior. This is an undesirable in terms of driver controllability. The "hoops" in the plots derive from gear changes.

Fig.19 shows a homogenous (similar response for left and right steering) lateral acceleration and yaw rate subject to the steering input.

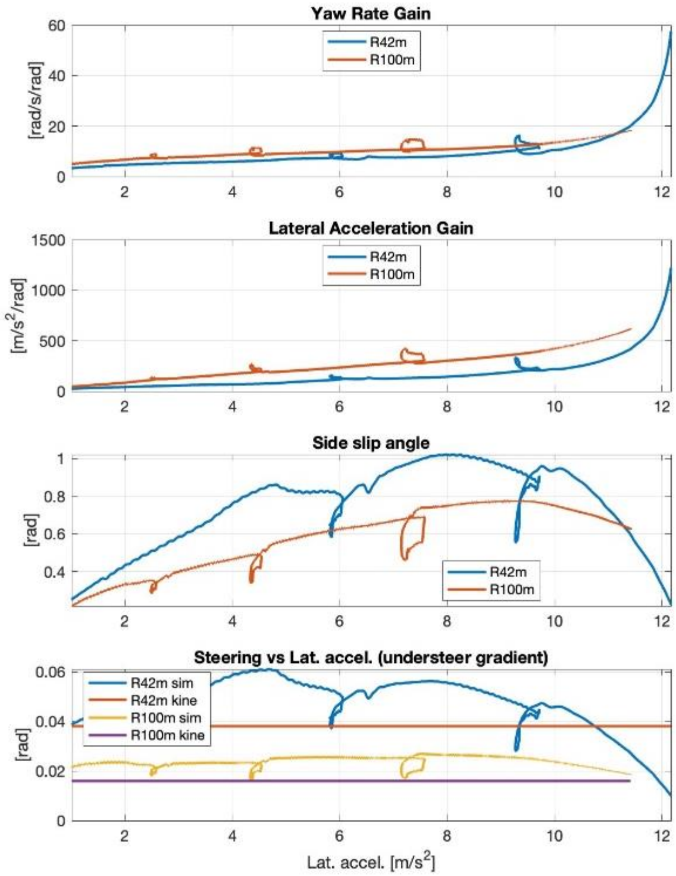


Fig.18 some vehicle dynamics attributes vs lateral acceleration (all have the same x-axes) for steady circle of 42 m (R42m) and 100 (R100m) meters simulation.

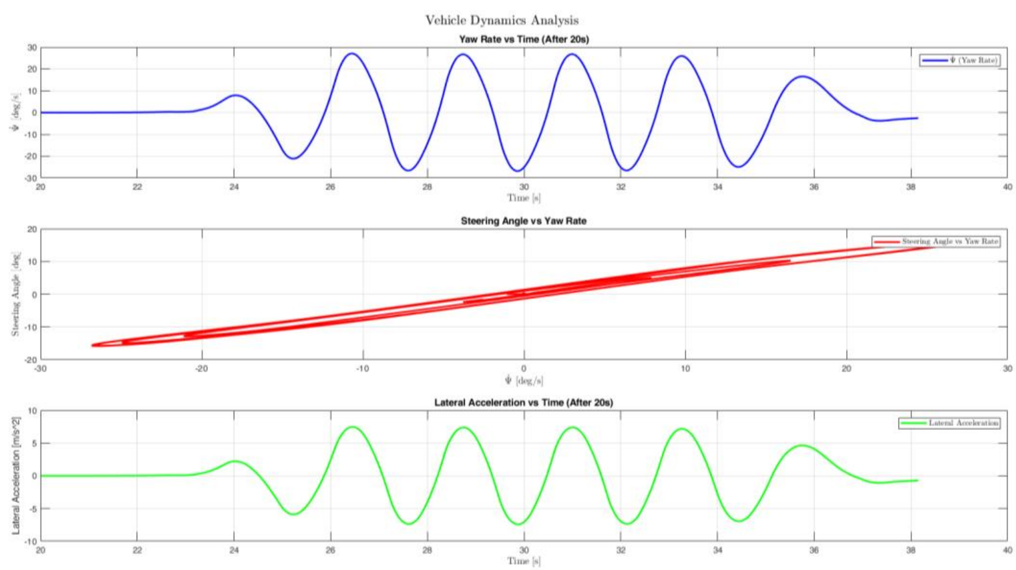


Fig.19 Vehicle Dynamics Analysis from a slalom maneuver from CarMaker simulation.

The following tables present the fixed and adjustable features of the Formula Student vehicle, which are central to its design and performance goals. The fixed specifications, including wheelbase, track width, tire size, and suspension geometry, are determined by competition rules and serve as a foundation for the vehicle's overall dynamic behavior. Key aspects such as front and rear ride rates, steering configuration, and roll center height are set to ensure a balanced and stable performance.

In contrast, the adjustable parameters allow for further optimization of the vehicle's dynamics. These include settings for static and dynamic toe angles, damping, and understeer characteristics. By fine-tuning these adjustable elements, the team can refine the car's handling, cornering abilities, and ride comfort to match specific performance needs. Adjusting these features is essential for tailoring the vehicle to varying track conditions and driver preferences, ultimately maximizing its competitive performance.

Fixed:
Wheelbase/trackwidth: 1.6 m, 1.2 m
Wheels/tires: 13" constrained by competition
Front/rear ride rates: 2Hz
Anti-lift and Anti-squat: 0%
Front/rear static camber: -1 deg
Front/rear dynamic camber: 10 deg/m
Front/rear roll center height: ground
Steering:
Min turning radius: 2 m
Caster angle: 4 deg
Kingpin inclination: 3 deg
Ackerman geometry: Parallel steer

Adjustable:
Front/rear static toe: 0.1 deg (adjustable)
Dynamic toe: none (adjustable via toe link vertical movement)
Damping: critical on jounce (rebound 3x more damping jounce)
Understeer targets: 2 deg (road wheel angle)/9.81 m/s^2 at 6 m/s^2 (adjustable via stiffness balancing and dynamic toe adjustment)

V. Conclusion

This thesis has outlined a comprehensive methodology for the development of a Formula Student (FS) vehicle, employing Systems Engineering (SysEng) principles to ensure a structured and efficient design process. The primary objectives were twofold: first, to summarize the development process for key vehicle dynamics and chassis components, and second, to utilize virtual verification through Computer-Aided Engineering (CAE) and Computer-Aided Design (CAD) tools. By systematically progressing through Static, Kinematic, and Dynamic validation events, the design was iteratively refined and verified to meet performance objectives.

The work packages discussed emphasize the importance of a phased approach in vehicle development, ensuring that each stage of validation is grounded in sound engineering principles and aligned with the overall design targets. The application of SysEng principles enables a holistic view of the vehicle as a system, while still providing focus on specific subsystems for optimization.

Developing such a vehicle requires not only technical expertise but also significant exposure to real-world mechatronic systems, especially in the transition from concept to prototype. The insights shared in this thesis are informed by this experience, and it is anticipated that this work will aid future engineers in adopting a more product-oriented approach when tackling vehicle prototyping. While the methodology is tailored to Formula Student, the principles and processes presented can serve as a valuable reference for broader engineering challenges.

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Appendix

An accurate kinematic model of the suspension is crucial for optimizing vehicle dynamics and ensuring optimal performance on track. The suspension's kinematic properties—such as camber change, toe angle, and roll center movement—play a key role in how the vehicle responds to various forces, especially during cornering, braking, and acceleration. By accurately modeling these parameters, engineers can predict how the suspension will behave under different conditions, leading to better control, improved tire contact, and overall enhanced handling characteristics. Additionally, a precise kinematic model allows for the fine-tuning of hardpoints, which directly impact the suspension geometry and dynamic behavior. This detailed understanding is especially important for Formula Student vehicles, where precision in suspension design can be the difference between a competitive lap time and poor performance. Below, the tables present the hardpoints for both the front and rear suspension, providing the baseline data used in the kinematic modeling of the vehicle's suspension system.

Front Suspension

IPGKinematics Version 3.6.11 Model: Querlenker

Table: Geometry Demo data

Geometry [mm]: X - Y - Z

Axis System

0 = DIN 70000, 1 = Automobil Industrie 1

Point 1 Wheel Center 0.000 -695.790 259.710

Point 2 Center of Tire Contact 0.000 -715.030 0.000

Point 3 Force Application Tire Forces 0.000 -681.030 0.000

Point 4 Body - Chassis Subframe Front -200.000 -400.000 165.000

Point 5 Body - Chassis Subframe Rear 400.000 -400.000 215.000

Point 6 Wheel Carrier - Lower Wishbone 18.530 -639.130 165.230

Point 7 Wheel Carrier - Upper Wishbone 19.990 -636.290 348.694

Point 8 Wheel Carrier - Steering Rod -70.000 -639.290 75.530

Point 9 Subframe - Bushing Front Lower Wishbone 169.090 -261.820 169.940

Point 10 Subframe - Bushing Rear Lower Wishbone -127.530 -253.680 175.740

Point 11 Vehicle - Bushing Front Upper Wishbone 164.090 -298.950 351.717

Point 12 Vehicle - Bushing Rear Upper Wishbone -135.810 -298.950 351.741

Point 13 Chassis Subframe - Stabilizer Bar -250.000 -400.000 150.000

Point 14 Stabilizer Bar - Stabilizer Link 0.000 -600.000 150.000

Point 15 Stabilizer Link - Wheel Suspension 0.000 -600.000 400.000

Point 16 Steering Rod - Steering Rack -25.000 0.000 75.000

Point 17 Chassis Subframe - Steering Gearbox -25.000 0.000 75.000

Point 18 Spring - Body 35.000 -450.000 715.000

Point 19 Spring - Wheel Suspension 30.000 -500.000 465.000

Point 20 Damper - Body 25.000 -400.000 715.000

Point 21 Damper - Lower Wishbone 20.000 -450.000 465.000

Point 22 Axle Drive Shaft - Differential 0.000 -200.000 309.000

Point 23 Axle Drive Shaft - Wheel 0.000 -695.790 259.710

Geometry by using Pull/Push Rods

Point 24 Body - Rocker Arm 13.230 -266.950 680.630

Point 25 Rotation Axis - Rocker Arm -50.000 -266.950 680.630

Point 26 Pull/Push Rod - Wheel Suspension 3.230 -546.500 377.799

Point 27 Pull/Push Rod - Rocker Arm 13.230 -318.800 677.870

Point 28 Spring Element - Body 7.780 -20.400 700.380

Point 29 Spring Element - Rocker Arm 7.780 -262.100 724.370

Geometry by using Pull/Push Rods Stabilizer

Point 30 Joint Stabi Torsion Bar - Body 250.000 0.000 300.000

Point 31 Rod Stabi Torsion Bar - Rocker Arm 50.000 -50.000 500.000

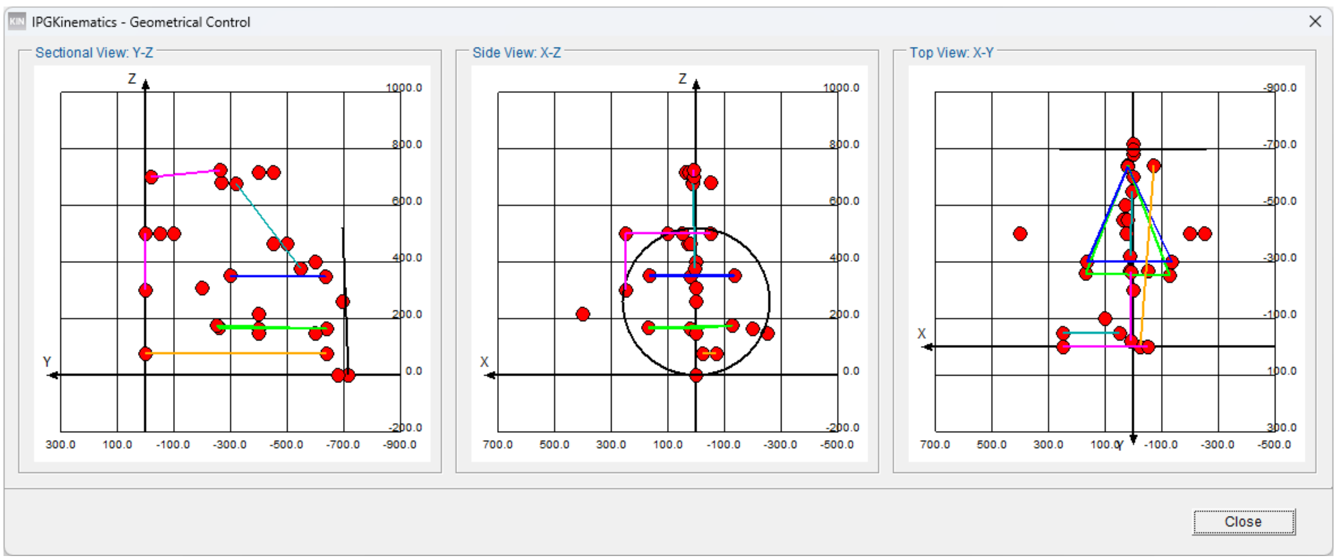
Point 32 Rod Stabi Torsion Bar - Torsion Bar 250.000 -50.000 500.000

Point 33 Spring Stabi Torsion Bar - Body -50.000 0.000 500.000

Point 34 Spring Stabi Torsion Bar - Torsion Bar 250.000 0.000 500.000

Point 35 Marker Damper Rocker Arm 100.000 -100.000 500.000

Point 36 Body -> Turning Circle Dia.: X-Y-Radius -500.000 -775.000 250.000



Rear Suspension

IPGKinematics Version 3.6.11 Model: Querlenker

Table: Geometry Demo data

Geometry [mm]: X - Y - Z

Axis System

0 = DIN 70000, 1 = Automobil Industrie 1

Point 1 Wheel Center 0.000 -612.740 242.050

Point 2 Center of Tire Contact 0.000 -612.740 0.000

Point 3 Force Application Tire Forces 0.000 -612.740 0.000

Point 4 Body - Chassis Subframe Front -200.000 -400.000 165.000

Point 5 Body - Chassis Subframe Rear 400.000 -400.000 215.000

Point 6 Wheel Carrier - Lower Wishbone 0.000 -539.110 140.230

Point 7 Wheel Carrier - Upper Wishbone 0.000 -524.140 344.660

Point 8 Wheel Carrier - Steering Rod -70.000 -639.290 75.530

Point 9 Subframe - Bushing Front Lower Wishbone 181.140 -292.700 142.330

Point 10 Subframe - Bushing Rear Lower Wishbone -161.420 -292.700 142.330

Point 11 Vehicle - Bushing Front Upper Wishbone 181.140 -292.700 322.330

Point 12 Vehicle - Bushing Rear Upper Wishbone -161.420 -292.700 322.330

Point 13 Chassis Subframe - Stabilizer Bar -250.000 -400.000 150.000

Point 14 Stabilizer Bar - Stabilizer Link 0.000 -600.000 150.000

Point 15 Stabilizer Link - Wheel Suspension 0.000 -600.000 400.000

Point 16 Steering Rod - Steering Rack -25.000 0.000 75.000

Point 17 Chassis Subframe - Steering Gearbox -25.000 0.000 75.000

Point 18 Spring - Body 35.000 -450.000 80.000

Point 19 Spring - Wheel Suspension 30.000 -50.000 465.000

Point 20 Damper - Body 25.000 -400.000 80.000

Point 21 Damper - Lower Wishbone 20.000 -50.000 465.000

Point 22 Axle Drive Shaft - Differential 0.000 -200.000 309.000

Point 23 Axle Drive Shaft - Wheel 0.000 -750.000 309.000

Geometry by using Pull/Push Rods

Point 24 Body - Rocker Arm 9.060 -257.580 358.340

Point 25 Rotation Axis - Rocker Arm 169.060 -200.000 350.000

Point 26 Pull/Push Rod - Wheel Suspension 69.500 -522.830 186.850

Point 27 Pull/Push Rod - Rocker Arm 69.500 -298.270 358.580

Point 28 Spring Element - Body 69.500 -236.460 152.980

Point 29 Spring Element - Rocker Arm 69.500 -198.780 358.970

Geometry by using Pull/Push Rods Stabilizer

Point 30 Joint Stabi Torsion Bar - Body 250.000 0.000 300.000

Point 31 Rod Stabi Torsion Bar - Rocker Arm 50.000 -50.000 500.000

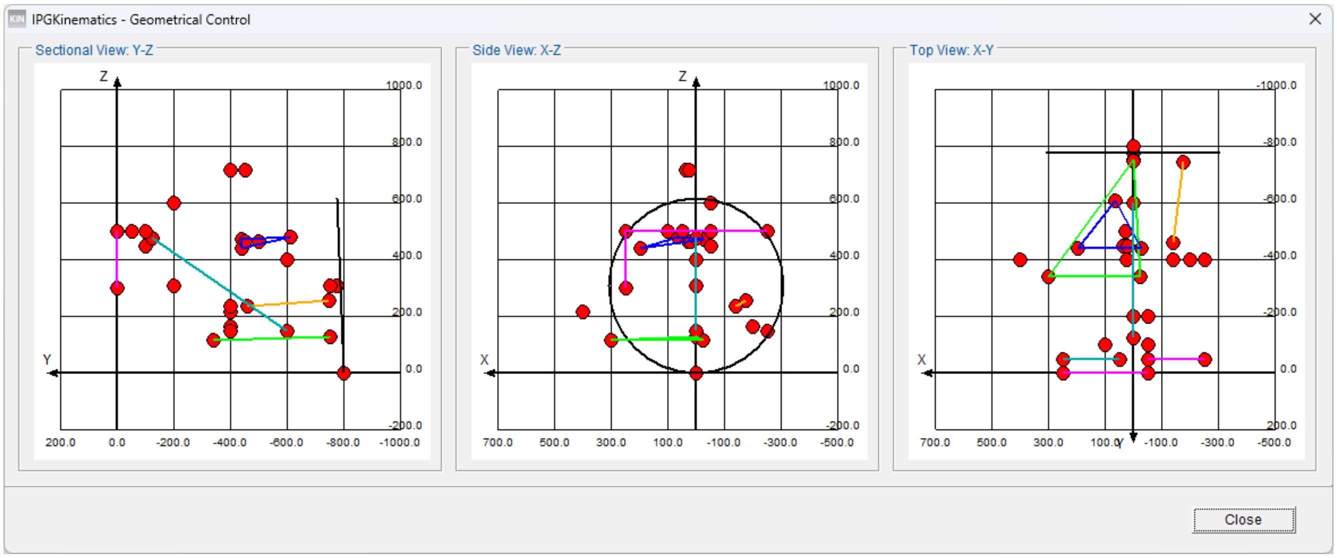
Point 32 Rod Stabi Torsion Bar - Torsion Bar 250.000 -50.000 500.000

Point 33 Spring Stabi Torsion Bar - Body -50.000 0.000 500.000

Point 34 Spring Stabi Torsion Bar - Torsion Bar 250.000 0.000 500.000

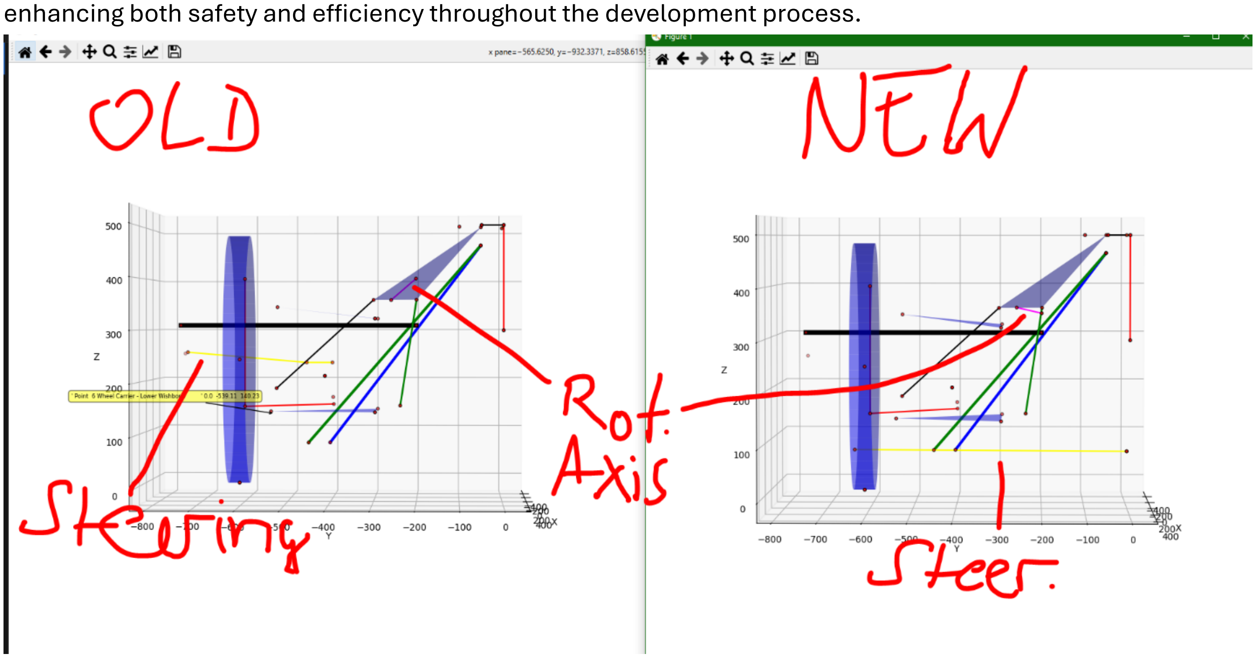
Point 35 Marker Damper Rocker Arm 100.000 -100.000 500.000

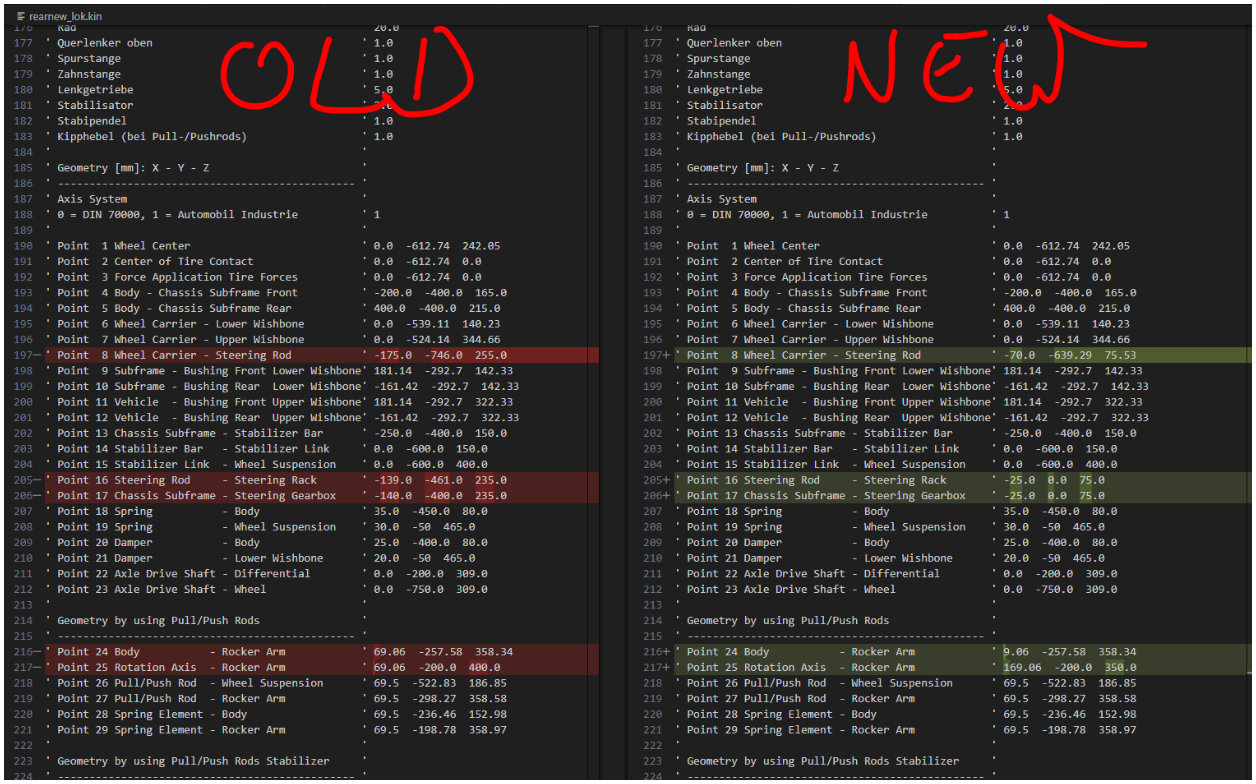
Point 36 Body -> Turning Circle Dia.: X-Y-Radius -500.000 -775.000 250.000



Also, here you can find an error and the solution on the rear suspension

Detecting errors in the kinematic model of the suspension during the research and development phase is essential for ensuring the accuracy of the overall design. Early identification of inaccuracies allows for timely adjustments, optimizing vehicle performance and handling. This proactive approach mitigates the risk of costly design flaws, enhancing both safety and efficiency throughout the development process.






```
' Rad                                           ' 20.0
' Querlenker oben                               ' 1.0
' Spurstange                                    ' 1.0
' Zahnstange                                    ' 1.0
' Lenkgetriebe                                  ' 5.0
' Stabilisator                                  ' [illegible]
' Stabipendel                                   ' 1.0
' Kipphebel (bei Pull-/Pushrods)                ' 1.0
'                                               '
' Geometry [mm]: X - Y - Z                      '
' --------------------------------------------- '
' Axis System                                   '
' 0 = DIN 70000, 1 = Automobil Industrie        ' 1
'                                               '
' Point  1 Wheel Center                         ' 0.0  -612.74  242.05
' Point  2 Center of Tire Contact               ' 0.0  -612.74  0.0
' Point  3 Force Application Tire Forces        ' 0.0  -612.74  0.0
' Point  4 Body - Chassis Subframe Front        ' -200.0  -400.0  165.0
' Point  5 Body - Chassis Subframe Rear         ' 400.0  -400.0  215.0
' Point  6 Wheel Carrier - Lower Wishbone       ' 0.0  -539.11  140.23
' Point  7 Wheel Carrier - Upper Wishbone       ' 0.0  -524.14  344.66
' Point  8 Wheel Carrier - Steering Rod         ' -175.0  -746.0  255.0
' Point  9 Subframe - Bushing Front Lower Wishbone' 181.14  -292.7  142.33
' Point 10 Subframe - Bushing Rear  Lower Wishbone' -161.42  -292.7  142.33
' Point 11 Vehicle  - Bushing Front Upper Wishbone' 181.14  -292.7  322.33
' Point 12 Vehicle  - Bushing Rear  Upper Wishbone' -161.42  -292.7  322.33
' Point 13 Chassis Subframe - Stabilizer Bar    ' -250.0  -400.0  150.0
' Point 14 Stabilizer Bar   - Stabilizer Link   ' 0.0  -600.0  150.0
' Point 15 Stabilizer Link  - Wheel Suspension  ' 0.0  -600.0  400.0
' Point 16 Steering Rod     - Steering Rack     ' -139.0  -461.0  235.0
' Point 17 Chassis Subframe - Steering Gearbox  ' -140.0  -400.0  235.0
' Point 18 Spring           - Body              ' 35.0  -450.0  80.0
' Point 19 Spring           - Wheel Suspension  ' 30.0  -50  465.0
' Point 20 Damper           - Body              ' 25.0  -400.0  80.0
' Point 21 Damper           - Lower Wishbone    ' 20.0  -50  465.0
' Point 22 Axle Drive Shaft - Differential      ' 0.0  -200.0  309.0
' Point 23 Axle Drive Shaft - Wheel             ' 0.0  -750.0  309.0
'                                               '
' Geometry by using Pull/Push Rods              '
' --------------------------------------------- '
' Point 24 Body           - Rocker Arm          ' 69.06  -257.58  358.34
' Point 25 Rotation Axis  - Rocker Arm          ' 69.06  -200.0  400.0
' Point 26 Pull/Push Rod  - Wheel Suspension    ' 69.5  -522.83  186.85
' Point 27 Pull/Push Rod  - Rocker Arm          ' 69.5  -298.27  358.58
' Point 28 Spring Element - Body                ' 69.5  -236.46  152.98
' Point 29 Spring Element - Rocker Arm          ' 69.5  -198.78  358.97
'                                               '
' Geometry by using Pull/Push Rods Stabilizer   '
' --------------------------------------------- '
```

```
' Rad                                           ' 20.0
' Querlenker oben                               ' 1.0
' Spurstange                                    ' 1.0
' Zahnstange                                    ' 1.0
' Lenkgetriebe                                  ' 5.0
' Stabilisator                                  ' [illegible]
' Stabipendel                                   ' 1.0
' Kipphebel (bei Pull-/Pushrods)                ' 1.0
'                                               '
' Geometry [mm]: X - Y - Z                      '
' --------------------------------------------- '
' Axis System                                   '
' 0 = DIN 70000, 1 = Automobil Industrie        ' 1
'                                               '
' Point  1 Wheel Center                         ' 0.0  -612.74  242.05
' Point  2 Center of Tire Contact               ' 0.0  -612.74  0.0
' Point  3 Force Application Tire Forces        ' 0.0  -612.74  0.0
' Point  4 Body - Chassis Subframe Front        ' -200.0  -400.0  165.0
' Point  5 Body - Chassis Subframe Rear         ' 400.0  -400.0  215.0
' Point  6 Wheel Carrier - Lower Wishbone       ' 0.0  -539.11  140.23
' Point  7 Wheel Carrier - Upper Wishbone       ' 0.0  -524.14  344.66
' Point  8 Wheel Carrier - Steering Rod         ' -70.0  -639.29  75.53
' Point  9 Subframe - Bushing Front Lower Wishbone' 181.14  -292.7  142.33
' Point 10 Subframe - Bushing Rear  Lower Wishbone' -161.42  -292.7  142.33
' Point 11 Vehicle  - Bushing Front Upper Wishbone' 181.14  -292.7  322.33
' Point 12 Vehicle  - Bushing Rear  Upper Wishbone' -161.42  -292.7  322.33
' Point 13 Chassis Subframe - Stabilizer Bar    ' -250.0  -400.0  150.0
' Point 14 Stabilizer Bar   - Stabilizer Link   ' 0.0  -600.0  150.0
' Point 15 Stabilizer Link  - Wheel Suspension  ' 0.0  -600.0  400.0
' Point 16 Steering Rod     - Steering Rack     ' -25.0  0.0  75.0
' Point 17 Chassis Subframe - Steering Gearbox  ' -25.0  0.0  75.0
' Point 18 Spring           - Body              ' 35.0  -450.0  80.0
' Point 19 Spring           - Wheel Suspension  ' 30.0  -50  465.0
' Point 20 Damper           - Body              ' 25.0  -400.0  80.0
' Point 21 Damper           - Lower Wishbone    ' 20.0  -50  465.0
' Point 22 Axle Drive Shaft - Differential      ' 0.0  -200.0  309.0
' Point 23 Axle Drive Shaft - Wheel             ' 0.0  -750.0  309.0
'                                               '
' Geometry by using Pull/Push Rods              '
' --------------------------------------------- '
' Point 24 Body           - Rocker Arm          ' 9.06  -257.58  358.34
' Point 25 Rotation Axis  - Rocker Arm          ' 169.06  -200.0  350.0
' Point 26 Pull/Push Rod  - Wheel Suspension    ' 69.5  -522.83  186.85
' Point 27 Pull/Push Rod  - Rocker Arm          ' 69.5  -298.27  358.58
' Point 28 Spring Element - Body                ' 69.5  -236.46  152.98
' Point 29 Spring Element - Rocker Arm          ' 69.5  -198.78  358.97
'                                               '
' Geometry by using Pull/Push Rods Stabilizer   '
' --------------------------------------------- '
```

Below is an image of the first sheet from the attribute decomposition Google Sheet file. This sheet is designed to calculate the values of various attributes once a target is set and the constraints and other vehicle parameters are entered. It serves as a tool to help evaluate whether the vehicle's design meets the specified performance goals based on the inputs provided.

Name [Unit]	Symbol	Value	Name [Unit]	Symbol	Value	Name [Unit]	Symbol	Value
DRAG COEFFICIENT(clean number)	Cd	0,35	TIME [sec]		7,00	~~TORQUE AT WHEEL [watt]~~	~~Tw~~	~~351.342,80~~
FRONTAL AREA (m^2)	A	0,3	Vfinal [m/s]	Vf	27,78	VEHICLE MASS [kg]	m	370,00
AIR DENSITY [kg/m^3]	p	1,225	Vstart [m/s]	Vs	0,00	ACCELERATION DUE TO GRAVITY [kgm/s²]	g	9,81
						GRADE ANGLE [rad]	θ	20,00
ROLLING RESISTANCE COEFFICIENT (kg/t)	Cr	0,015	ACCELERATION AVERAGE [m/s^2]	A	3,97	WHEEL RADIUS [mm]	rw	285,00
			ACCELERATION DUE TO GRAVITY [kgm/s²]	g	9,81	GEAR RATIO	u	2,69
MASS OF THE CAR [kg]	m	370	TRACTION FORCE (N)	FT	1.585,71			
ACCELERATION DUE TO GRAVITY [kgm/s²]	g	9,81	MASS OF THE CAR [kg]	m	370,00	sin(θ)		0,91
ENGINE POWER (Watt)	Pengine	54000	ACCELERATION FOR TRACTION FORCE [m/s^2]	aFT	4,29	NORMAL FORCE [N]	Fn	1.481,22
						cos(θ)		0,41
TRANSMISSION EFFICIENCY(ηtrans)	η trans	0,85	TOTAL WEIGHT [N]		3.629,70	TIRE RATE	μ	1,20
SPEED TARGET [m/s]		27,72	WEIGHT ON REAR AXLE [N]		2.250,41	DIST FROM FRONT AXLE TO COG [mm]	lf	550,00
			WEIGHT DISTRIBUTION [N]		0,62	WHEELBASE [mm]	L	1.600,00
			PEAK TRACTION FORCE [N]		2.700,50	T engine [watt]		54.000,00
P total (to overcome both aerodynamic drag and rolling resistance)[watt]	Ptotal	2.879,09	μ(high performance tires on dry pavement) [dimensionless quantity]	μ	1,20	ηtrans		0,85
Power at the wheels after counting transmission efficiency[watt]	Pw(wheels)	45.900,00	a peak	a	7,30			
Cd		11,34						
A		0,30						
Cr		3.387,16	1,08					
Pengine		3.387,16						
GEAR RATIO	u	2,688						
			propulsion					
top speed target			acceleration target			grade ability target		
Aerodynamics the drag coefficient (Cd) shall be less or egual to 0.6 the frontal area shall be minimized to a value of 1 m^2 the total aerodynamic drag force Fd at 180km/hshall be calculated and minimized	AERODYNAMIC DRAG FORCE =	1.601,40	avg acceleration on wide open throttle		3,97	~~torque at wheel level~~ ~~to calculate the required torque at the wheel level to overcome the gravitational force on a 30% grade~~ ~~Tw=((m*g*sin(θ))*pw)/n~~ ~~Tw= torque at wheel~~ ~~m=vehicle mass~~ ~~g = gravity acceleration~~ ~~θ=grade angle(16.7degrees)~~ ~~rw=wheel radius~~ ~~n=gear ratio~~		~~351.342,80~~
Transmission loses the transmission efficiency(ηtrans)shall be greater than or equal to 0.85 the total power loss in the transmission system shall be minimized	ηtrans =	0,85	long traction from tires shall allow vehicle to achieve acceleration greater than		1.585,71	vehicle mass the veh mass must be considered in the torque calculation to ensure it can achieve the required gradeability Fg=m*g*sin(Θ) Fg= gravitational force component along the incline m= veh mass g=acceleration due to gravity θ=grade angle		3.313,72
Tire losses the rolling resistance coefficient(Cr)shall be greater than or equal to 0.0015 the total rolling resistance force (Fr)at 180 km/hshall be minimized	rolling resistance =	54,45	vehicles rear axle tires shall allow peak longitudinal traction		7,30	tire traction to ensure the tire traction is sufficient to prevent slip while ascending the grade Ft=μ*Fn Ft= traction force μ=coefficient of friction between tire and road surface Fn= normal force (m*g*cos(θ))		1.777,46
transmission ratio the transmission ratio shall be designed to optimize the balance between acceleration and top speed the final drive ratio shall be selected to enable the vehicle to reach 180km/h at the engines optimal RPM		2,69	The vehicle center of gravity location shall load the rear axle with at least % of the vehicle weight during a static longitudinal acceleration greater than		0,62	mass distribution ensuring a significant portion of the veh weight is on the driving wheels to maximize traction Fnr=(m*g*lf)/l Fnr=normal force on the rear axle m=veh mass g=accel due to gravity lf=dist from front axle to cog l=wheelbase length		1.247,71
						The torque distribution in an RWD vehicle must be optimized: Trwd=Tengine*ηtrans		45.900,00

Attribute decomposition Google Sheet

Here is the link to the Excel document containing the complete attribute decomposition, which outlines all the key performance attributes, and their relationships based on the set targets and constraints.

https://docs.google.com/spreadsheets/d/1Qajx8lQkw1T76uyZSf7erOBVNLVBK4KMchkMrdC-2n8/edit?usp=sharing

Matlab codes

Below, you will find the MATLAB scripts we used for post-processing, which played a key role in analyzing and refining the collected data. These codes were essential for extracting valuable insights from the raw information and helped optimize various aspects of the vehicle's performance.

https://drive.google.com/drive/folders/1wRaNbGwNgPkzJc1h-_CSCmwx6OAHkXll?usp=drive_link

Car model

Below, you will find the file containing the analytical model of the car that I developed using IPG CarMaker, which includes detailed parameterization of the steering system, engine, suspension, masses, and other critical components.

https://drive.google.com/file/d/1ByEMJVz0qQOj-ruOynQ8PAlIL8xeDFki/view?usp=drive_link