Development of a Front Suspension System for Entry Into Formula Student

Vehicle Dynamics

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Summary

The early stage of design work has been completed and a feasible, rule compliant system has been created. The system cost of £595.05 is affordable given the team budget of £8,000. The focus on designing to be as adjustable as possible allows for detailed setup during testing to extract the best possible handling characteristics the car can produce as conditions and track layouts change. The choice of a double wishbone setup with a pushrod mounted onto the upper wishbone gives the car good control over the tyre camber and toe, with a recommended static camber of -1.1 ° producing optimal contact patch during worst case cornering for a lower top speed, more corner intensive circuit. The steel and aluminium mechanical design offers a cheaper solution than alternatives such as carbon fibre without a significant cost to performance. Preliminary load analysis has been undertaken and the future work has been mapped out in order to achieve a functional product well before competition, increasing test time.

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Nomenclature

Symbol/Abbreviation/Acronym	Definition
TBRe	Team Bath Racing electric
TBReAI	Team Bath Racing electric: Artificial Intelligence
TBRe22	Team Bath Racing electric team designing for entrance in into competition during 2022
FSUK	Formula Student UK
CoG	Centre of gravity
FS	Formula Student
GBDP	Group Business Design Project
FSG	Formula Student Germany
CoM	Centre of mass
PDS	Product design specification
ARB	Anti-roll bar
h_f	Front roll centre height
h_R	Rear roll centre height
λ_f	Motion ratio front
λ_r	Motion ratio rear
a_F	Height of CoG above front roll centre
a_R	Height of CoG above rear roll centre
K_f	Effective ride stiffness at front wheels
K_r	Effective ride stiffness at rear wheels
ΔW_f	Total Load Transfer across Front
ΔW_r	Total Load Transfer across Rear

1 Needs and Problem Identification

The University of Bath has a history of success at Formula Student across both its petrol and electric vehicle entries into both national and international events. The TBRe22 team aims to continue this history of success and will be entering a vehicle into its first event, Formula Student UK, in 2022. This report will focus on the design process behind the car's front suspension system and how it integrates with other subsystems across the vehicle. *Figure 1* shows how the team is organised.

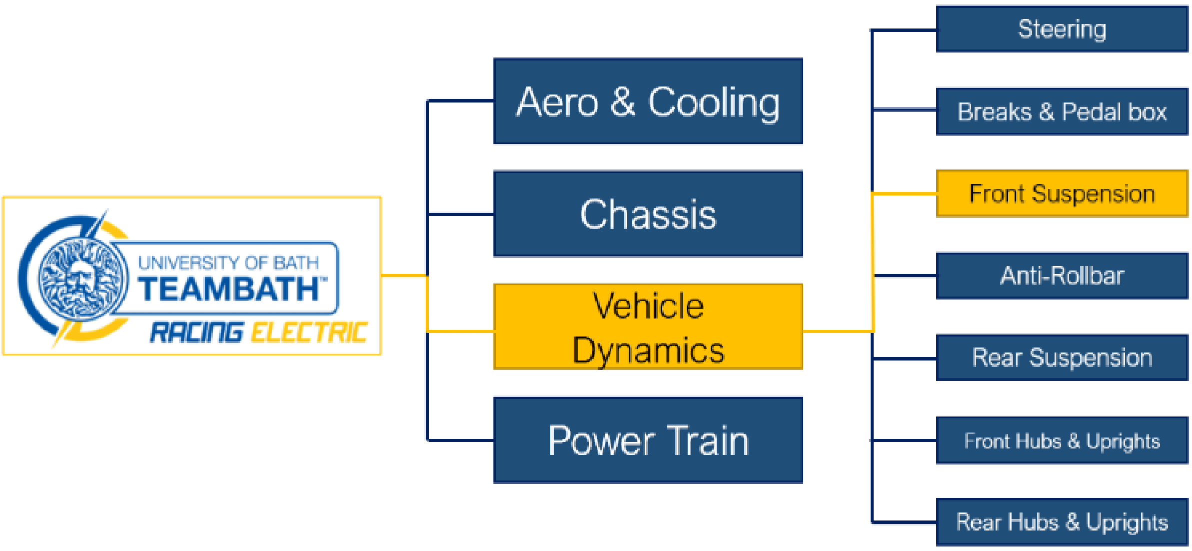


Figure 1: TBRe team breakdown

1.1 Voice of Customer

Understanding the project requirements in detail helps provide the best possible product through better informed design decision making processes. A breakdown of the TBRe22 customer needs is shown below in *Table 1*.

Table 1: Customer needs and priority

Customer	Priority	Needs
Formula Student	High	The Formula Student rules and regulations must be followed precisely when making any decisions throughout the entire process or the car will risk not passing scrutineering, nullifying the entire project. Formula Student body also dictates how each event is awarded points and therefore decides the criteria around which the vehicle should be assessed.
TBRe Driver	Medium	The driver is the one responsible for extracting the potential of the car and therefore should always be kept in mind when making design decisions
TBRe Design Team	Medium	For the car to function well as a complete package, the vehicle dynamics team should strive to accommodate the other sub team's needs in order to produce a well-rounded vehicle that is not over constrained by any single sub team.
Industry Sponsors and the University of Bath	Low	While both customers are pivotal to the car's success, their needs are predominantly met if the higher priority customer's needs are met.

1.2 Team Aims

Having consistent team aims across the design team keeps all decisions being made aligned with each other and reduces the strain on communication within the team. Before the project began, the whole team determined its priorities, based off the customer requirement shown in *Table 1*, and produced a list and weighting of each of the TBRe22's main aims, shown below in *Table 2* .

Table 2: TBRe22 team aims and ranking

Aim	Reasoning	Realisation
Reliability	Designing a reliable car allows for more hours of testing, less chance of failure at competition and a driver to be more confident in the vehicle and therefore push it closer to its limit. This has direct benefits, such as reducing chance of failure at an event, as well as indirect benefits such as longer testing hours allowing for a more accurate simulation of the vehicle and optimisation of a range of sub systems.	A car that works with minor, easily resolved issues over its year competing as well as the year after competing for the TBReAI team.
Handling	A car that handles well is more likely to succeed than cars that focus on other aspects such as straight-line speed due to a large proportion of dynamic event points being earned in cornering situations.	Net positive feedback from a majority of test drivers with competitive lap times in both testing and at competition
Driver Safety	Keeping the driver safe will always be a priority throughout all stages of the design and testing process	All FSUK rules and regulations abided by as well as appropriate safety factors applied for all components
Adjustability	Designing a first-time correct suspension in an environment such as Formula Student is severely unlikely. By designing in adjustability to the relevant key features, the design can be improved dramatically without the need for re-design or re-manufacturing.	A suspension system that allows change to static camber and toe with minimal effort
Flexibility	Flexibility of design provides the necessary leeway to integrate designed systems both internal to the vehicle dynamics team and externally to the other sub teams improving the time to manufacture and range of design possibilities.	Following an agile design process that can respond to changes in requirements from other sub-teams. An example of this can include the construction of variable models to allow rapid adjustment to external changes.

1.3 Technical Challenges

The difficulty of designing a front suspension system is balancing the desired characteristics, designed to provide the best handling, and the interaction of that design with the other vehicle systems. *Table 3* shows the breakdown of the conflicting design directions both internally and externally to the vehicle dynamics team and how they function. The interactions fall into three sections:

- Dependency is a variable that is externally decided and that must be abided by

- Compromise is a variable that should be agreed to match the best interest of both parties

- Precursory is a variable that is decided by the suspension system that other systems must abide by

Table 3: Front suspension dependencies

Internal /external	Subsystem	Decision	Dependency (Precursory /Dependency /Compromise)	Description
E	Chassis	Wishbone mounting location	C	Where the wishbones are anchored onto the body of the vehicle
I	Front Hubs &Uprights	Wishbone mounting location	C	Where the wishbones are anchored onto the uprights
I	Front Hubs &Uprights	Hub size and layout	C	If the wishbones will interfere with the hubs in max steer situations
I	Rear Suspension	Suspension setup	C	Deciding consistent body roll characteristics
I	Anti-Roll bar	Suspension setup	C	Deciding consistent body roll characteristics
E	Powertrain	All wheel drive vs rear wheel drive	C	Allowing for potential front driveshafts or in-hub motor setups
I	Steering	Tie rod locations	P	Providing room for tie rods around wishbone travel
E	Aerodynamics	Packaging constraints	P	Mounting points conflict with undertray and winglet locations
E	Aerodynamics	Wishbone aerodynamic impact	P	Designing aerodynamic impact of wishbones and springs.
E/I	General	Centre of mass height	D	Adjusting suspension setup based on the centre of mass height of the vehicle
E/I	General	Total car mass	D	Adjusting suspension setup based on the mass of the vehicle

The conflicting interests of this project as both designing a full car for competition as well as designing a submission for the GBDP means that some design tasks are considered out of scope for the GBDP hand in due to the volume of work required to produce a final design of the car. *Table 4* shows how the tasks have been broken down into work packages and where they land within the two timescales. While the GBPD hand in is a complete concept, the whole vehicle is likely to undergo design iterations based on the learning outcomes from this first stage and therefore all concepts are likely to change, even if only minorly, in the future.

Table 4: Scope of this report based on tasks completable withing the GBDP timeframe

GBDP/FSUK	Task	Description
GBDP + FSUK	Design Suspension geometry	Deciding pick-up points for the suspension to give the system desirable handling characteristics
GBDP + FSUK	Integration of steering characteristics	Matching the effects on the wheel camber from steering inputs to the effect of the suspension geometry to sum to a viable wheel camber and toe in key scenarios

GBDP + FSUK	Initial desired spring rates	Sizing of spring required based off desired roll rates and motion ratios from the geometry
GBDP + FSUK	Initial mechanical load analysis	Using worst case scenarios to analyse the maximum loads in the wishbones.
FSUK	Anti-pitch/anti-dive geometry	Designing geometric adjustments to reduce the cars pitching and diving during acceleration and breaking
FSUK	Finalised integration with upright and chassis	Final mounting points with the upright based off the final desired geometry
FSUK	Compliance analysis	Adjusting design to reduce compliance of components under load
FSUK	Finalised mechanical load analysis and weight saving optimisation	Sizing of all key components based off total load analysis

1.4 PDS

A product design specification is created to track all design requirements through the development process as well as remain accountable to other teams' requirements. The full PDS is available in Appendix A.

1.5 Design Process overview

The overall design process is shown in *Figure 2*. The task order follows the logical progression of design starting with an understanding of the fundamental physics of the situation following through to design realisation of the system.

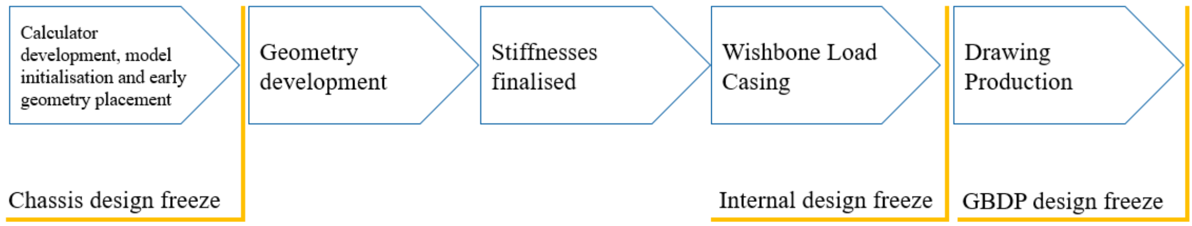


Figure 2: Design flow

A methodology for integrating design was set up early in the project lifecycle in order to reduce chances of miscommunication and improve the chance of a successful implementation without wasting unneeded time. Both parties involved in the decisions are to work in parallel with a minor deadline set based on the type of task. Before the first design review, the two parties create necessary models and calculators and research the surrounding theory to understand their area of the system. Once the deadline occurs, the two parties meet and discuss their desired path forward based off the physics-based analysis they have undertaken and how much leeway they can provide each other. If, after multiple design reviews, the desired outcomes are mutually exclusive then the team agrees which subsystem has more of an impact on working towards the core team aims and side is chosen. A diagrammatic example of how the outboard mounting points for the suspension knuckles was decided is shown in *Figure 3*. The time scale for the early design reviews can be days or weeks but as integration progresses, the timespan shortnes untill detailed design is finalised through micro adjustsments while in a meeting together.

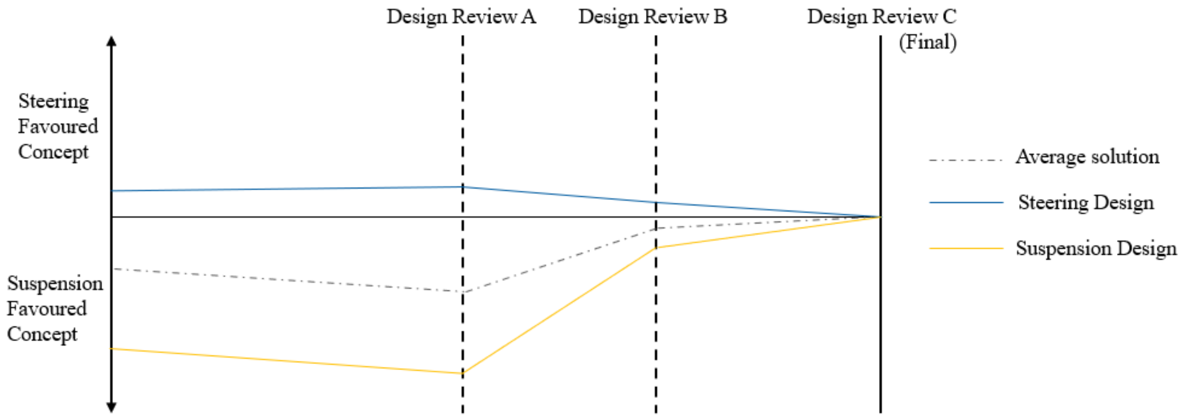


Figure 3: Design integration process

2 Technical Design and Application

Design of the front suspension system is based around the double wishbone setup used by a vast majority of the cars that enter FS competitions across the globe. The system uses two A-arms connected to a lower and upper ball joint on the upright within the wheel. A third member, the steering arm, is then connected to constraint the wheel in the horizontal planes. A final member, the pushrod, is attached to constrain the vertical movement of the wheel, completing the constraint of the wheel in 3D space. The ability to control the position and movement of each of these joint locations allows an engineer to control dynamic changes to camber and toe, explained below in *Figure 4*.

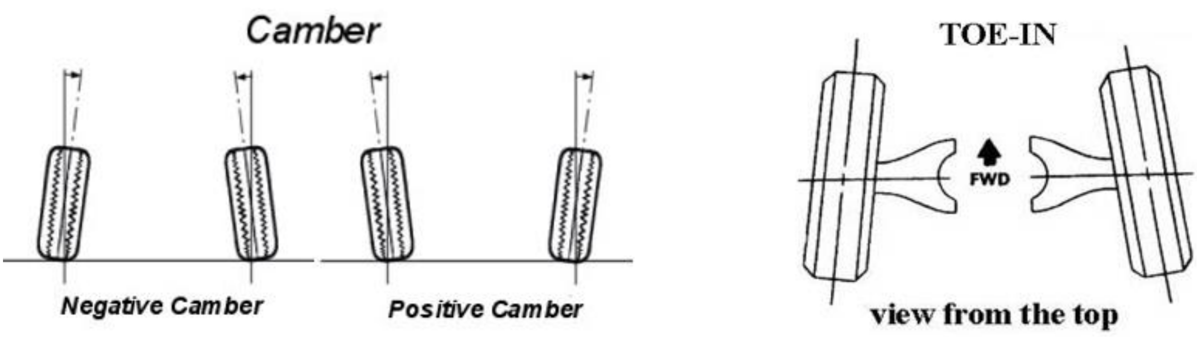


Figure 4: Camber and Toe diagrams [6]

2.1.1 Design setup

Before accelerating into the integration aspects of the design, a range of design tools were set up or initiated to provide a database of tools to respond to any of the upcoming design challenges as well as provide a chance to fully understand the theory that is required to design a suspension system.

2.1.1.1 Optimum Lap

An *OptimumLap* model is used as it provides information key to most aspects of the suspension design including choosing a geometry, sizing the spring stiffness and calculating the loads that each wishbone is expected to experience. While it does have its shortcomings as a simulation, such as the model using a point mass for the vehicle, it can be used for estimation and for sensitivity analysis. The initial values used were based on the TBRe2019 car, however, the tool became far more useful as more information was produced by the other sub-teams i.e., expected motor toques and aerodynamic downforce and therefore increases with usefulness as a project progress. Below is the FSG autocross track used in 2012, the benchmark track decided by the team to measure performance and base calculations off.

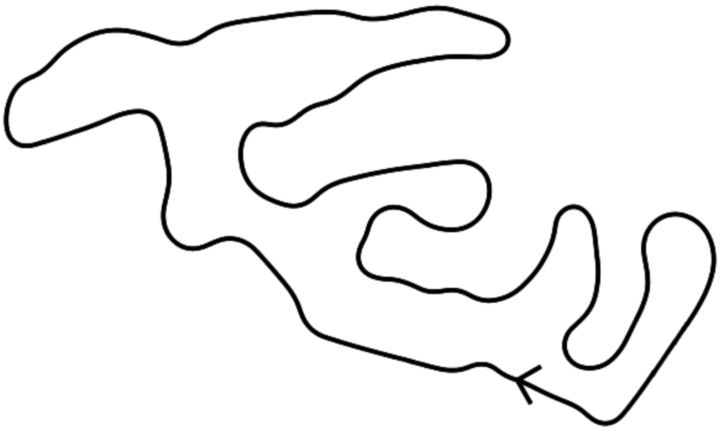


Figure 5: FSG autocross track layout for the 2012 competition

2.1.1.2 Calculators

A variety of calculators were built up in excel to reduce time between iterations in the design phase by reducing the need for hand calculations and reworking large sections of physics if the design is required to change due to an external dependency. The inputs and outputs of the major tools are shown below in tables 1 2 and 3

Table 5: Inputs and outputs of the key calculators used throughout the design process

	Inputs	Outputs
Wheel load calculator	- Whole Car properties (wheelbase, CoM height, vehicle mass etc.) - Suspension properties (Roll centre heights, wheel stiffnesses, motion ratio etc…) -Cornering scenario (Corner radius, Speed)	- Body roll angle - Vertical and Lateral Load per wheel - Spring displacements - Breakdown of load transfer (percentage linkages, springs and lateral CoM shift)
Aerodynamic package driven maximum body roll calculator	-Aerodynamic component dimensions and location	- Allowable body roll in degrees before clash with the ground
Wishbone load calculator	-Aerodynamic downforce -Whole car properties (mass etc.) -Lateral and vertical loads from wheel load calculator	-Expected maximum load in each wishbone in a max cornering and max breaking scenario

2.2 Pushrod pathing

The first major design task that was undertaken was analysing the options for the updated powertrain system. A majority of early design work and research was focused on rear-wheel-drive designs so when the rest of the TBRe22 cohort decided that moving to all wheel drive with four in hub motors was the best path forward, the analytical cycle reset and design started from the ground up. The main conflict that needed solving is the routing of the pushrod to the upright. *Figure 6* shows the final motor and cooling jacket sizing and, while the exact dimensions could not have been known so early in the project, it was necessary to design to use minimal space inside the wheel.

Figure 6: Wheel assembly with motor and gearbox.

Though a wide range of solutions were addressed, the final solution chosen is a simple direct acting pushrod attached to the upper wishbone. The decision-making process for this is shown below in *Table 6*. The weighting used can be found in Appendix B.

Table 6: Pugh matrix used to decide the design of pushrod

Design Criteria	Criteria Weighting	Pullrod with chassis floor mounted rocker	Direct acting pushrod	Pushrod with chassis roof mounted rocker	Pushrod with chassis side mounted rocker
Adjustability	17.86	0	-1	1	0
Cost	3.57	-1	1	-1	0
Integration	25.00	-2	2	-2	0
Design Flexibility	7.14	-1	1	-1	0
Manufacturing Complexity	10.71	-1	2	-1	0
Mass	14.29	-1	1	-1	0
Reliability	21.43	0	1	0	0
	Net Score	-85.7143	100	-67.8571	0
	Rank	4	1	3	2
	Move Forward?	No	Yes	No	No

2.3 Mounting locations and geometry development

Figure 7 shows some key terms used when integrating steering geometry and suspension geometry to produce a coherent system.

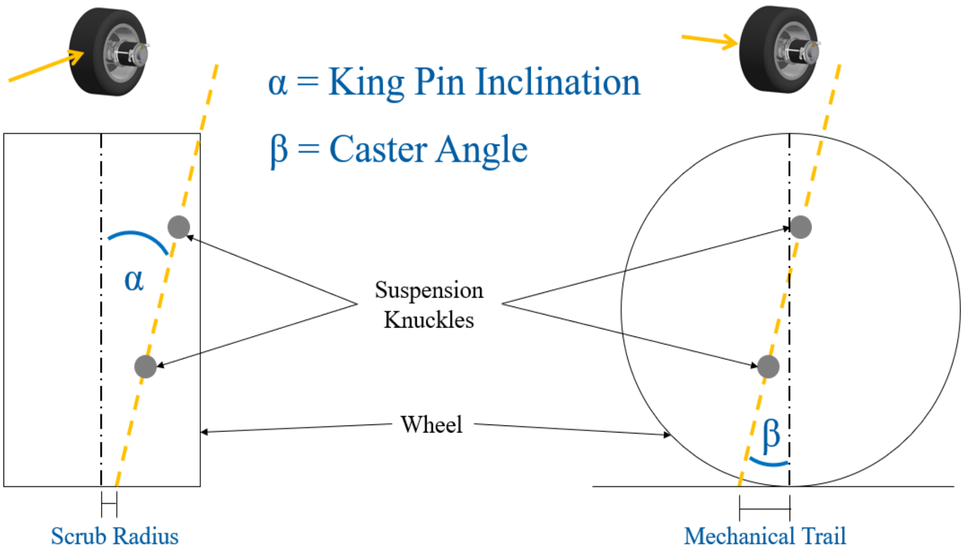


Figure 7: Diagrammatic explanation of king pin inclination, scrub radius, caster angle and mechanical trail

2.3.1 Outboard Mounting

Focusing primarily on outboard mounting locations in order to meet product design specification points 6.7,6.8,6.9 and 6.10 for constraining one side of the design allowed initial progress to be made. Following the design process laid out in *Figure 3*, upright mounting locations could be tested and iterated upon to achieve a functioning system. The design iterated through a process shown below decided upon is show below in *Figure 8*, using the *OptimumKinematics* software as form of digital prototyping to inform design decisions.

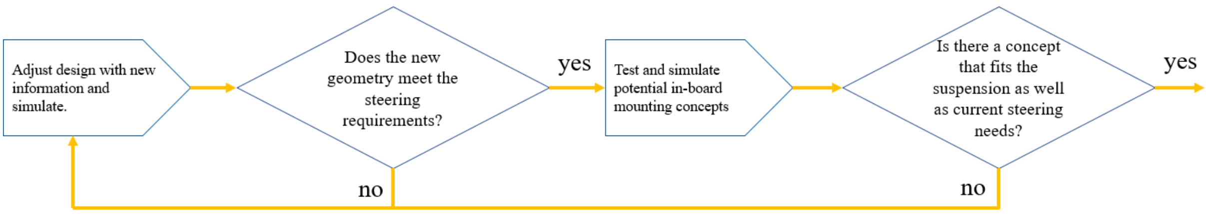


Figure 8: Steering and suspension integration process diagram.

Initial iterations were found to be extremely unfavourable, with scrub radii and mechanical trails of nearly 200% of the maximum decided by both parties due to the lack of space within the wheel rim causing mechanical clashes with components. *Figure 9* shows an early conflict with the lower mounting joint and the break disk that caused a shift of the maximum values provided by steering.

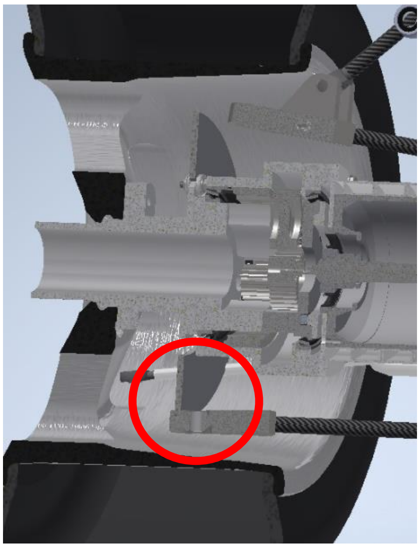


Figure 9: Half section of the front wheel in early development highlighting the clash between the lower wishbone joint and break disk

Once the design had gone through enough iterations, a set of acceptable values for outboard mounting locations were achieved. These values were chosen to provide a balance of desirable steering characteristics as well as a manageable change in camber under steering input, the full characteristic can be seen in *Figure 10.* The final location was also chosen based on likelihood of mechanical clashes with the wheel rim as well as feasibility of in hub mounting solutions to reduce the likelihood of redesign further into the design process.

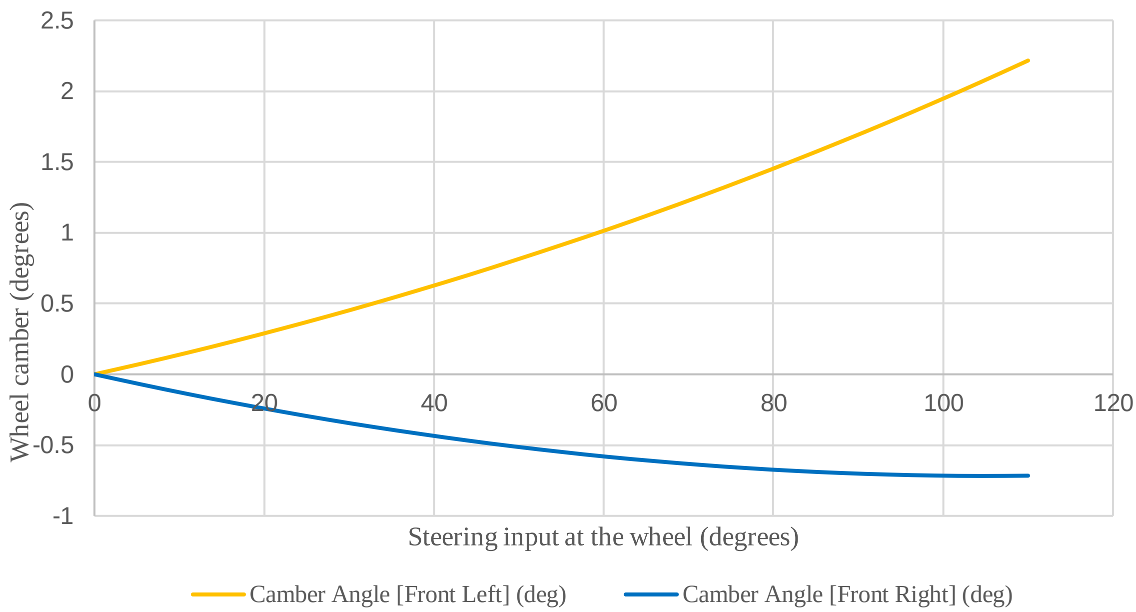


Figure 10: Change in camber due to caster angle through a 110-degree steer input

This knuckle setup shows a relatively flat, predictable characteristic that peaks at -0.7 of camber change at the outside wheel. Since the space inside the rim is so restricted, these values are acceptable as the camber change is well within the range of values that can be manipulated by the suspension geometry camber gain. *Table 7* shows the finalised values achieved and how they compare to the product design specification's targets. While the iterations were varied in both success and time taken, the final solution achieved all the steering requirements while minimising impact on upright design as well as reducing the chances of clashes with the rim during dynamic manoeuvres.

Table 7: Comparison of final steering relevant values to the PDS

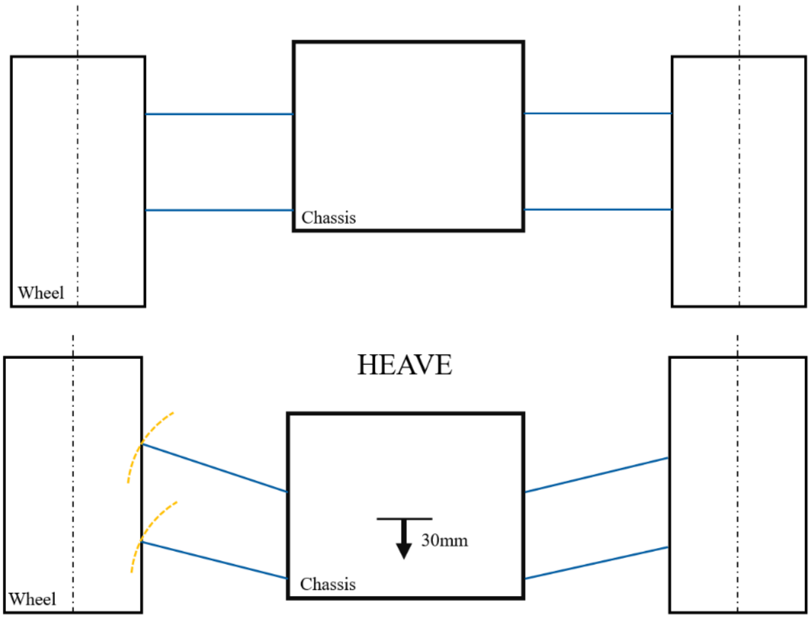
	PDS Value	Achieved Value
Lateral king pin inclination	<15°	12.8°
Caster angle	3-5°	4.3°
Mechanical trail	20+-5mm	19 mm
Scrub radius	<40 mm	23.5 mm

2.3.2 Inboard mounting

With the wheel side locations set, the inboard mounting locations are decided. This is where most of the control of the camber changes in roll and heave takes place due to the constraints within the wheel as previously discussed. There is no true solution to suspension as the optimum set up changes based on weather, track type and many more factors. It is for this reason that the design approach was to find an acceptable start point that is adjustable enough to cover a wide range of situations so that, come testing, the design can be altered to match the needs of the test.

2.3.2.1 Theory

The aim of the suspension system is to keep the contact patch of the tyre working as hard as it can. In general this means that when the car undergoes acceleration or breaking the tyre is roughly flat to the ground, and while cornering that the outside tyre has a camber of -1° (the elastic nature of rubber causes the contact patch to deform leading to improved cornering characteristics at -1° rather than 0° camber). *Figure 12* shows how the camber of the tyres reacts to both heave and roll inputs if the wishbones are set to be the same length and parallel.



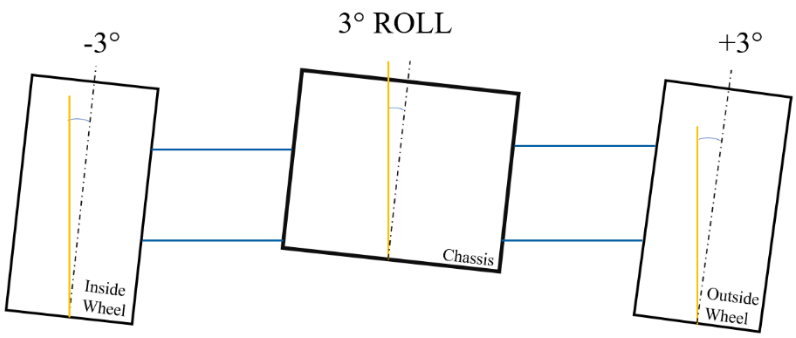


Figure 11: Camber change under roll and heave with parallel, equal length wishbones

Having the same radius and centre point on both the top and bottom wishbones causes the camber not to change under heave. This is beneficial as the contact patch is kept at its maximum under heave which occurs during breaking zones, where the tyre performance is critical. The roll characteristic is undesirable, however, as the outside wheel camber is positive, producing camber thrust in the opposite direction to the corner, worsening handling characteristics dramatically. By adjusting the wishbone to converge and increasing the length of the bottom wishbone compared to the top, the wheel cambers can be changed in heave and roll. This is shown below in *Figure 11*.

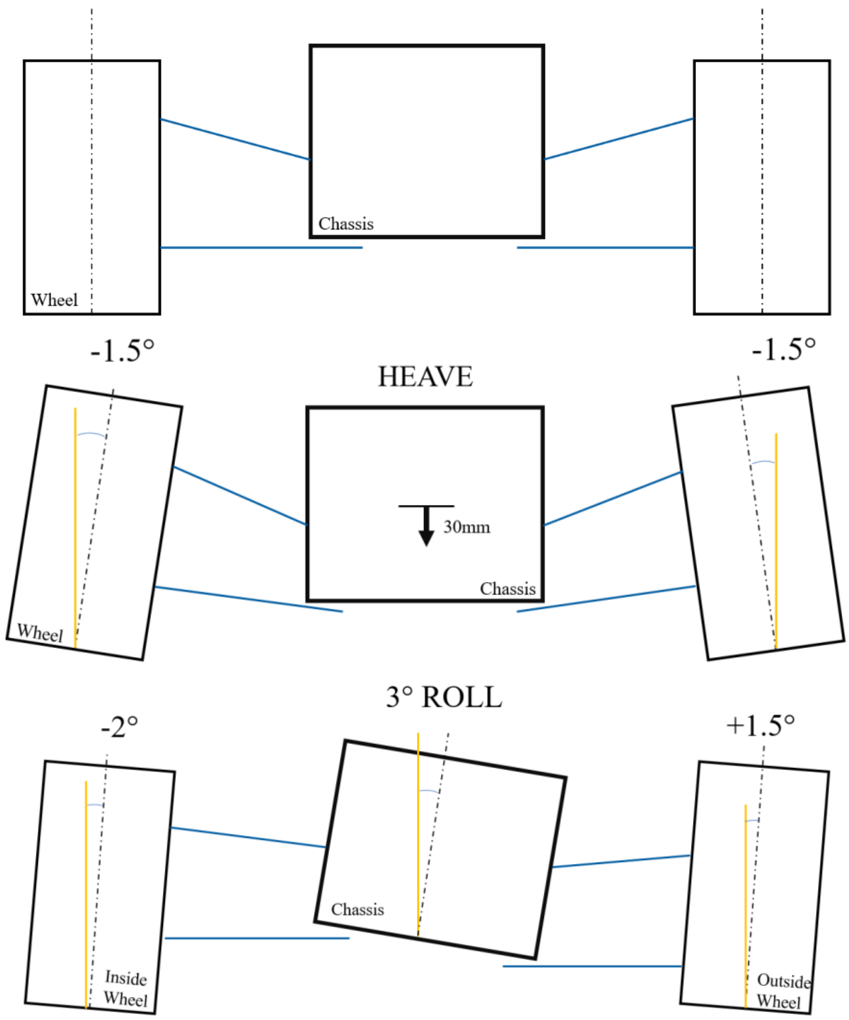


Figure 12: Camber change under roll and heave with converging, unequal length wishbones

While this change has reduced the outside camber to be closer to the desired -1°, there is now a camber change under heave, reducing the contact patch during breaking. This is the fundamental balancing act of suspension geometry design.

2.3.2.2 Design iteration

An initial maximum body roll of target, based off aerodynamic package clashes with the ground (see Appendix C), was used to set the boundaries of simulation in which to analyse the design iterations. A sensitivity study into the effects of the placement of the lower knuckle and upper knuckle independently was used to determine how best to alter design to achieve the desired characteristics. A visualisation of the mounting points tested in the upper wishbone location analysis is shown below in *Figure 13*.

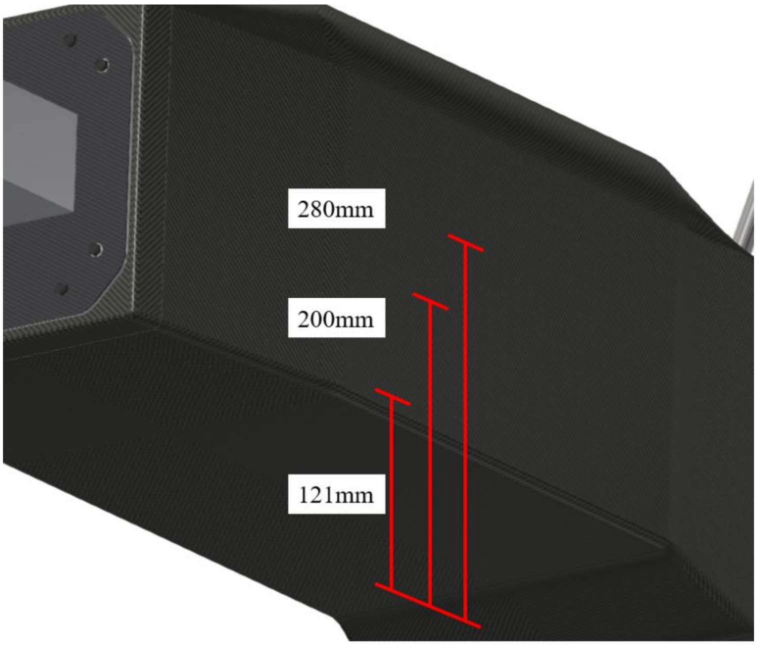


Figure 13: Diagram showing the three heights tested and their location on the chassis (distances labelled from ground)

The ride height requirements alongside chassis and aerodynamic component sizing provided a minimum mounting height from the ground on the chassis wall. This was used alongside a middle point and a highest point (same height as the outboard knuckle) to vary upper wishbone mounting locations and help guide the design process. *Figure 15*, *Figure 14* and showd the respones under roll while *Figure 16* shows the response under heave. The lower knuckle was placed on the base of the chassis in order to achieve a long lower wishbone, reducing roll centre and track variation while keeping the roll centre close to the ground.

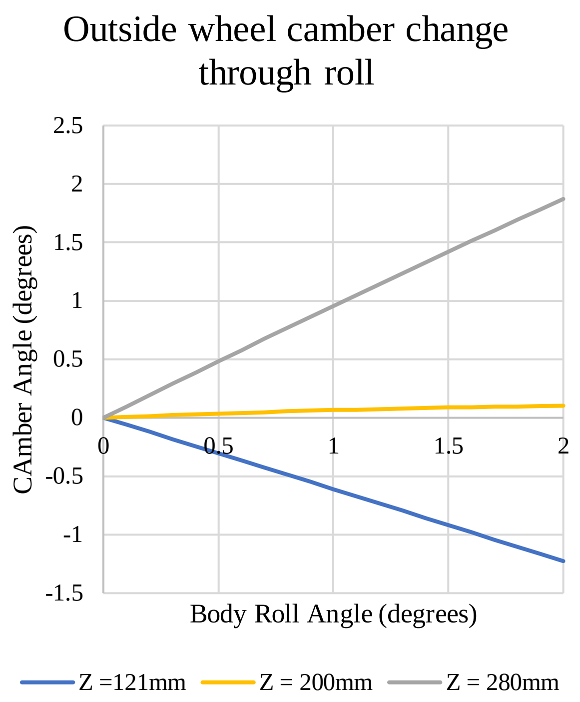


Figure 15: Sensitivity of the outside wheel camber to vertical position of the upper mount location in roll

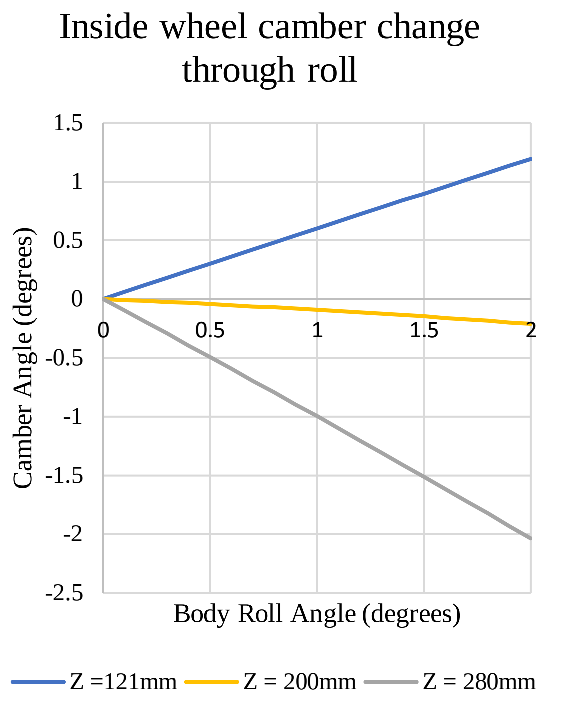


Figure 14: Sensitivity of the inside wheel camber to vertical position of the upper mount in roll

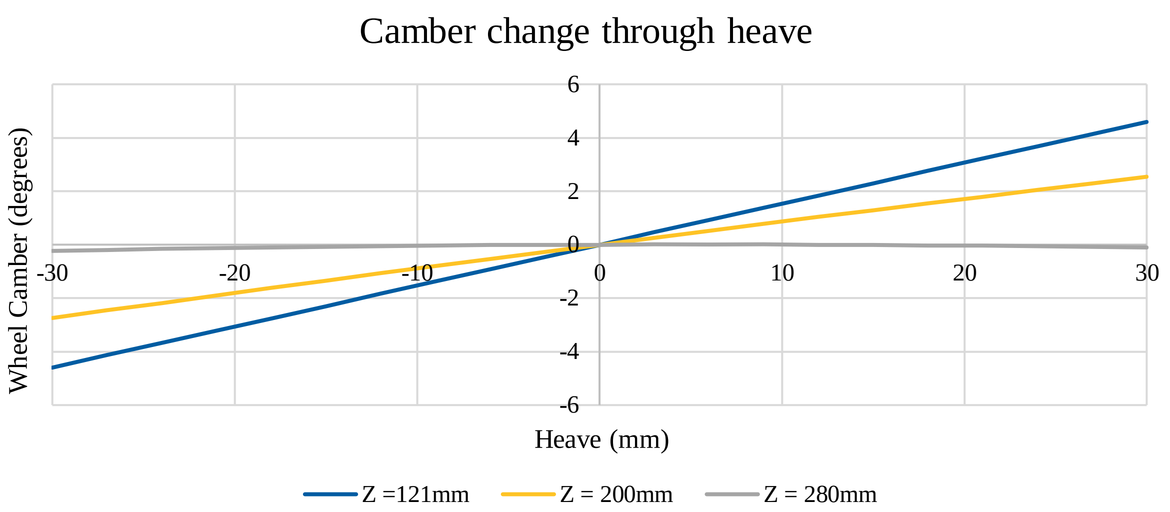


Figure 16: Sensitivity of the wheel cambers to vertical position of the upper mount in heave

The final mounting location for the upper mount used this sensitivity analysis alongside the chrematistics generated by the outboard mounting locations to produce a working solution for the geometry. The final height of 250mm offers a balance of performance under both heave and roll, and example data from simulation below in *Figure 17* shows how the camber changes occur isolated from the changes under steer.

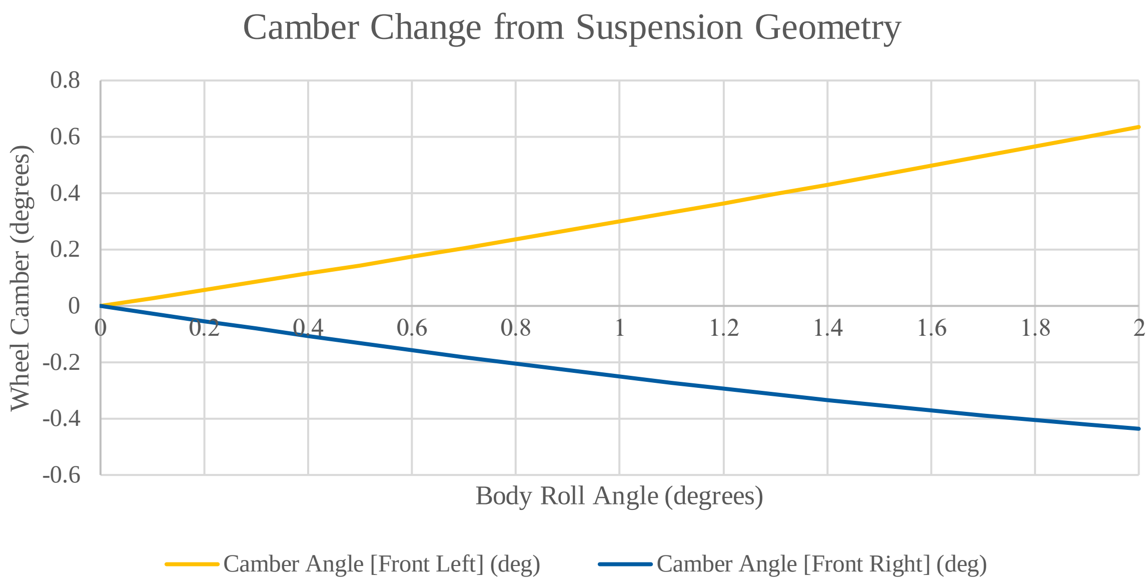


Figure 17: Camber change from suspension geometry

The whole system was then applied to a low speed and high-speed corner i.e. body roll is constant, but steer input is reduced in high speed shown in *Figure 18*. This data can be used to choose a desirable static camber to achieve optimal performance based on the speed of the track, for example a lower speed track with lots of hairpins would use a static camber of roughly -1° to -1.4° whereas the higher speed circuit would use a static camber of -1.2° to -1.6° based on body roll setup. These values are well within the PDS limit (see Appendix A).

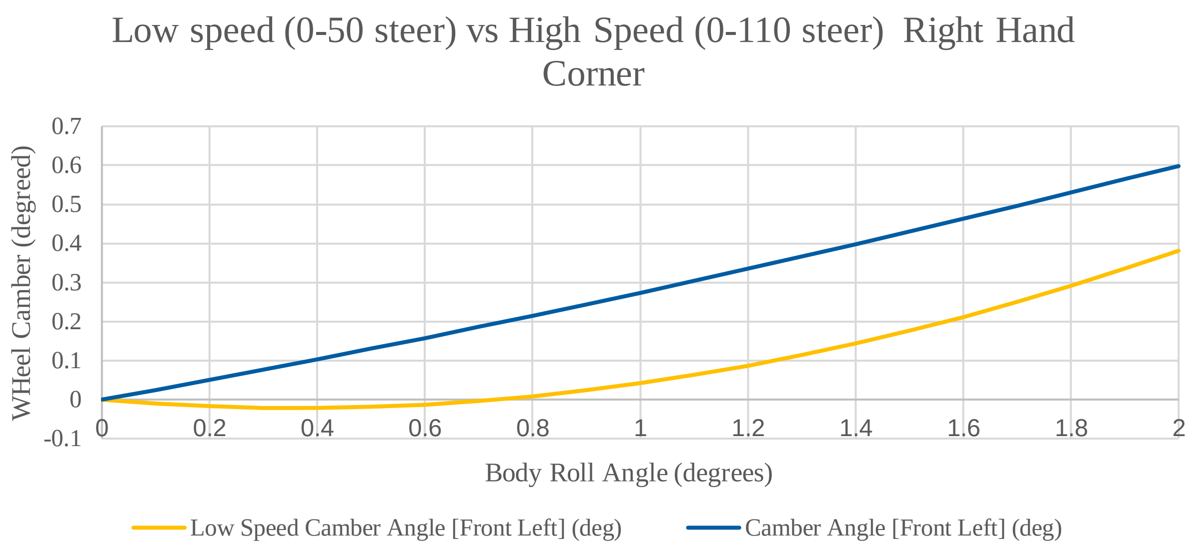


Figure 18: Outside wheel camber change comparison between a low and high-speed corner

2.4 Spring Rates

By this point much of the cars design had progressed enough to allow a far more accurate *OptimumLap* model and for more realistic scenario-based analysis useful for developing the necessary stiffness targets for the wider suspension system. *Figure 20* shows the process diagram for the most valuable tool used for this process, the load calculator sheet, developed by the front and rear suspension leads as well as the anti-roll bar lead. The *OptimumLap* model provided speeds and corner radii accurate to real formula student racetracks which could then be used analyse a variety of cornering scenarios. *Figure 19* displays the iterative process applied to this tool that produced a working solution for the required spring rate.

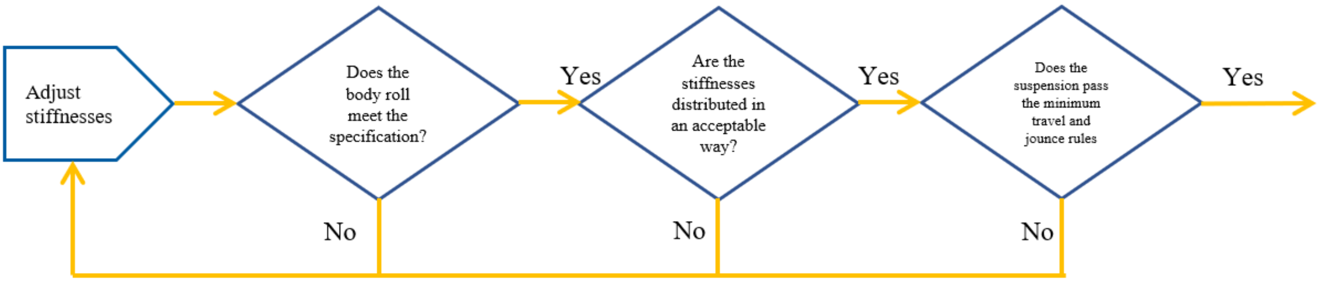


Figure 19: Design process diagram for the calculation of required spring rate

The worst-case scenario chosen as the focal point of design was an 80km/h bend with a radius of 27m. This corner induced a lateral acceleration of 1.83g. This lateral acceleration was chosen for the worst-case scenario as it is the most realistic value based off previous year's competition data as well as accounting for the large overestimate in aerodynamic downforce provided by the aerodynamics sub team while still remaining at the edge of possible scenarios.

Using the input values shown below in *Table 8* as well as the maximum body roll calculated in appendix C, the front, rear and anti-roll bar suspensions were manipulated until the desired roll characteristic and percentage stiffness input from the roll bar was below a maximum of 40% of total car roll stiffness [1].

Table 8: Inputs into wheel load calculator

Variable	Value Used for Calculation	Units	Source
Wheelbase (l)	1.54	m	CAD
Distance from CoG to front axle (c)	0.8624	m	CAD
Distance from CoG to rear axle (d)	0.6776	m	CAD
Vehicle and Driver Mass (m)	300	kg	CAD
Vehicle Forward Velocity (V)	22.00	ms^{-1}	OptimumLap
Turn Radius	27	m	OptimumLap
Track width (t)	1.2	m	CAD
Height of CoG (h)	0.276	m	CAD
Front roll centre height (h_F)	0.012	m	CAD
Rear roll centre height (h_R)	0.057	m	CAD
Average Roll Centre Height	0.0345	m	Calculation
Motion ratio front(λ_f)	0.67		OptimumKinematics
Motion ratio rear (λ_r)	0.96		OptimumKinematics
Coefficient of friction (lateral)	1.6		Tyre test consortium
Speed Datum	22.2	m/s	Aerodynamics sub team
Downforce Datum	1300	N	Aerodynamics sub team

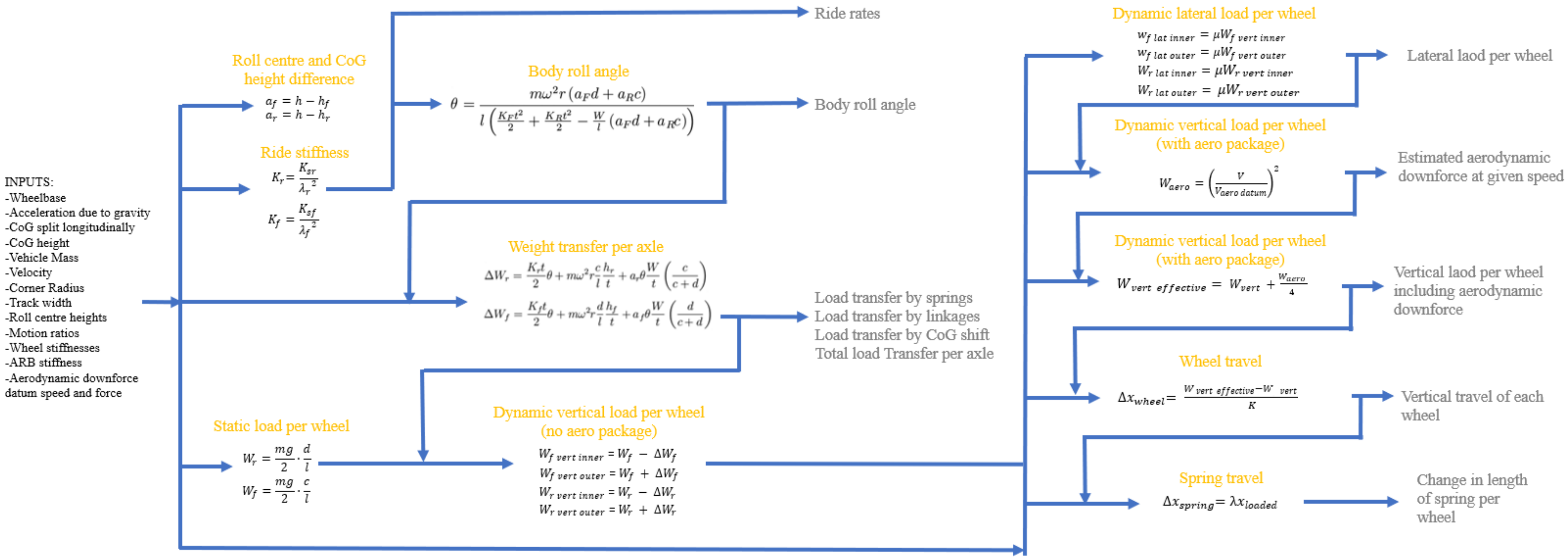


Figure 20: Flow diagram of wheel load calculator created to reduce time between design iterations

After a variety of iterations, the final stiffness decided upon is shown below in *Table 9* alongside the relevant vehicle characteristics due to this choice. Discussions with previous team members offers some confidence in the result based of previous team's values, and a final spring (70 kNm^{-1}) was chosen from the damper supplier to match the calculations [2]. The springs are easily replaced and, with the adjustment of preload, are a very versatile tool when it comes to setting up the car for the specific track.

Table 9: Chosen spring rates and example values of how the vehicle characteristics are affected

	Value	Units
Wheel Stiffness at front wheels	28	kNm^{-1}
Wheel Stiffness at rear wheels	32	kNm^{-1}
Effective ride stiffness at front wheels (K_f)	22.33	kNm^{-1}
Effective ride stiffness at rear from suspension wheels	24.97	kNm^{-1}
ARB contribution to rear roll stiffness	25.84	kNm^{-1}
Effective ride stiffness at rear wheels (K_r)	50.81	kNm^{-1}
Front spring stiffness	70.34	kNm^{-1}
Body Roll Angle (θ)	1.42	deg
Roll Gradient	0.77	deg/g
Roll Stiffness Total	919.08	Nm/deg
Roll Stiffness Front	280.60	Nm/deg
Roll Stiffness Rear (Susp)	313.76	Nm/deg
Roll Stiffness Rear (ARB)	324.72	Nm/deg

2.5 Load Casing

The aim of the early load casing was to gain an understanding of the types of the forces that could be experienced by the system before doing an in-depth analysis. To produce these early calculations, a simplified equivalent model was used, shown in *Figure 21*, with caster and king pin angles set to 0 along with wishbones assumed to be parallel.

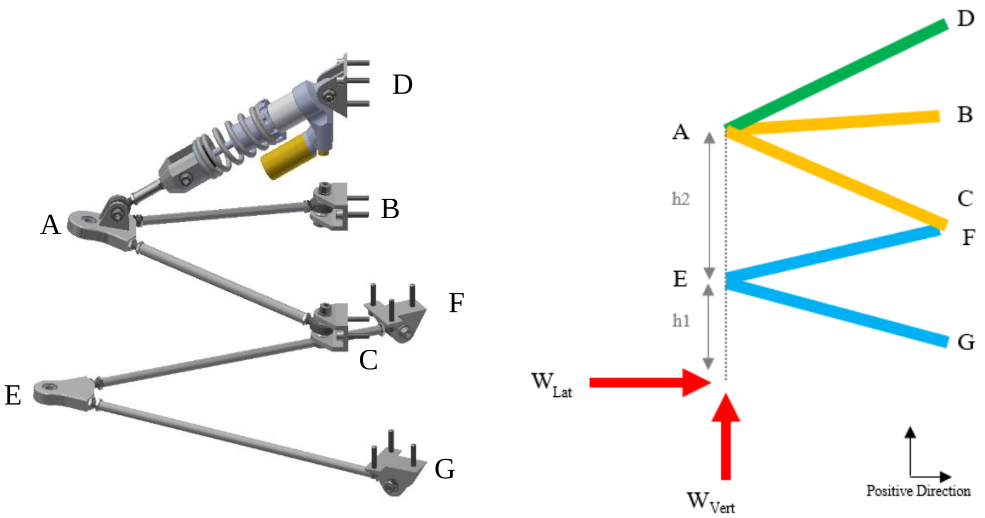


Figure 21: Diagram of system used to analyse the forces in a maximum cornering situation

Table 10 shows the values for all calculations as well as their source. These values are used as a basis for all the loads in the wishbones. For the vertical and horizontal loads, a dynamic factor is applied to account for real world force magnitudes [3].

Table 10: Maximum wheel loads and corresponding design loads

	Value	Unit	Source
Vertical Design Load (N)	3998	N	Wheel load calculator
Lateral Design Load (N)	2772	N	Wheel load calculator
h1	0.111	m	CAD model
h2	0.193	m	CAD model
Pushrod angle	31	Degrees	CAD model

Using this model, the free body diagrams shown in *Figure 22* were produced.

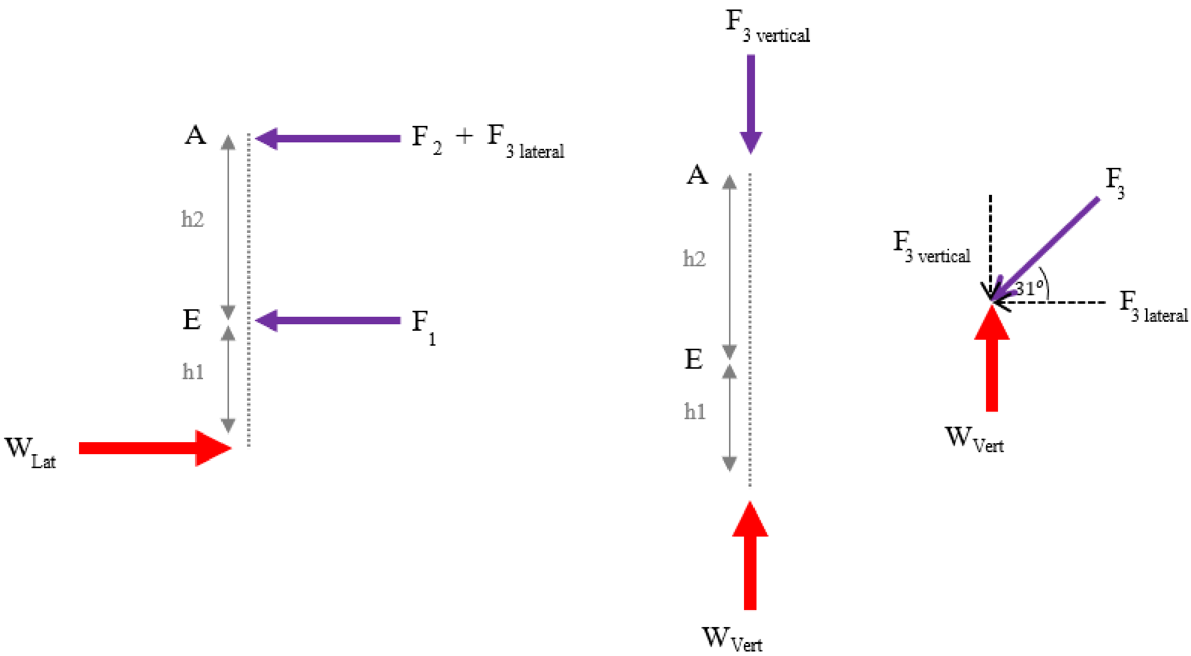


Figure 22: Free body diagrams created from the basic model

Equations 1 through 7 shown below use the force diagrams produced alongside figure x to give a final value of force in each wishbone and whether it's in tension or compression [4].

$$F_{3\,vertical} = w_{vert} = 3998N \tag{2.5.1}$$

$$F_3 = \frac{F_{3\,vertical}}{\sin(31°)} \approx 7760N \tag{2.5.2}$$

$$F_{3\,lateral} = \frac{F_{3\,vertical}}{\tan(31°)} \approx 6650N \tag{2.5.3}$$

Taking moments about E:

$$F_2 = -\left(W_{lat}\frac{h_1}{h_2} + F_{3\,lateral}\right) \tag{2.5.4}$$

$$F_2 = -\left(2772\left(\frac{0.111}{0.193}\right) + 6650\right) \approx -7040N \tag{2.5.5}$$

Taking moments about A:

$$moments\ about\ A\colon\ F_1 = W_{lat}\frac{h_1 + h_2}{h_2} \qquad 2.5.6$$

$$F_1 = 2772\left(\frac{0.111 + 0.193}{0.193}\right) \approx 4470N \qquad 2.5.7$$

These values can then be split down the individual wishbones as shown below in figure x to produce the values shown in Table 11.

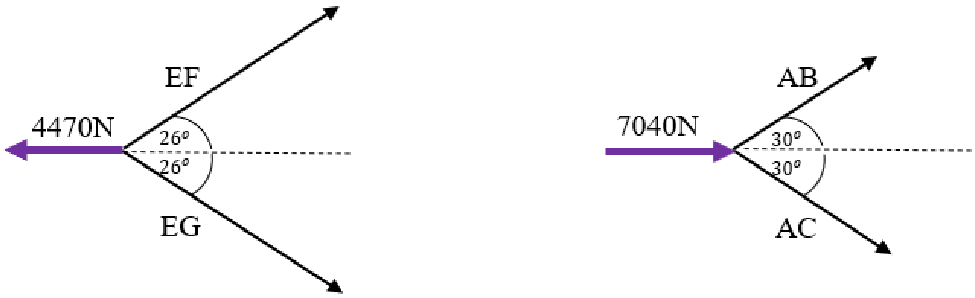


Figure 23: Force diagram for upper and lower wishbones

Table 11: Wishbone loads during maximum cornering event

Member	Force (N)	Tension/Compression
Lower fore arm	2490	Compression
Lower aft arm	2490	Compression
Upper fore arm	4060	Tension
Upper aft arm	4060	Tension
Pushrod	7760	Compression

The loads in the upper wishbone are abnormally high compared to typical suspensions due to the reduced motion ratio of the pushrod. In the future altering designs to incorporate a better motion ratio will reduce the load, thereby reducing the weight of wishbones and reducing the system weight. This also has the effect of improving the damper performance and reducing the spring stiffness required.

2.6 Standards and regulations

This stiffness was then checked against the FSUK rules T 2.3.1 and T 2.3.2 (6.16 in 6.17 in the PDS) as well as the PDS to ensure compliance with the customer needs. *Figure 24* explains the rules constraining minimum jounce and minimum wheel travel.

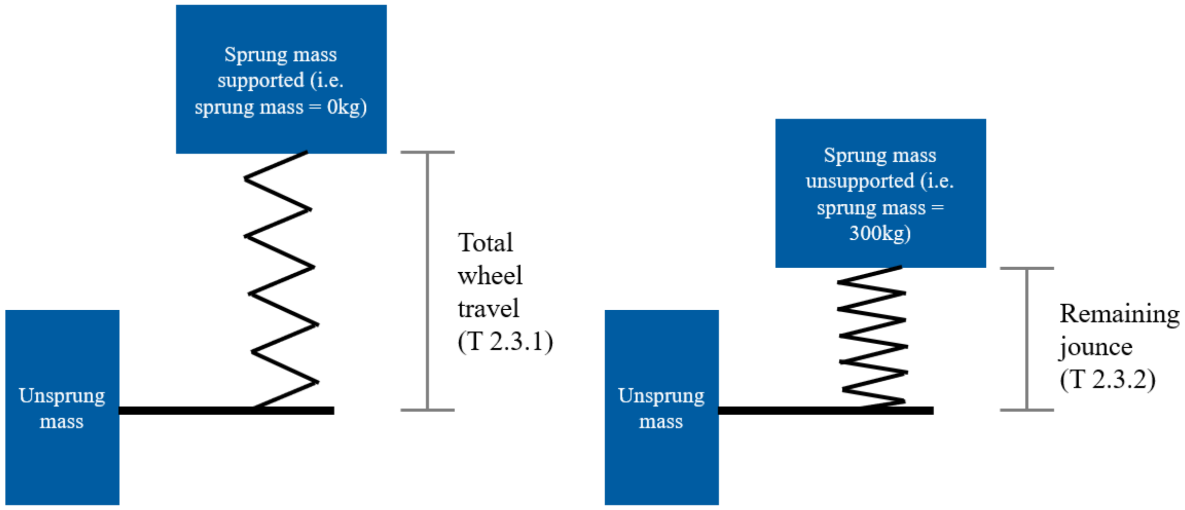


Figure 24: Minimum wheel travel and jounce FSUK rules.

By using the selected spring rates and the car mass, the design can be tested to check if it passes the rules. Using the wheel stiffness per corner ($28kNm^{-1}$) along with the mass of the car per corner at the front (66kg), the laden wheel travel is calculated, shown below in equations 8 and 9.

$$\frac{F}{k} = \Delta x \qquad 2.6.1$$

$$\frac{66 * 9.81}{28{,}000} = 23.1mm \qquad 2.6.2$$

Applying the motion ratio from the working model, this equates to 15.4mm travel at the spring. The spring damper chosen is the TTX 25 MkII FSAE [2]. The variant chosen for application in this project is quoted as having a maximum stroke of 57mm which, when applied via the motion ratio, gives a maximum wheel travel of 90mm. Once the total travel and the laden travel is known, the jounce can be calculated as the difference of the two, 67mm. *Table 12* shows how the systems working values compare with the requirements laid out by the rules and therefore PDS. Ride height is set statically at 35mm and is adjustable via pushrod length in order to minimise risk of not passing scrutineering.

Table 12: Comparison of working values with the relevant rules

	Rule Requirement	Achieved Value	Units
Wheel Travel (T 2.3.1)	>50	90	mm
Jounce (T 2.3.1)	>25	67	mm
Ride Height (T 2.3.3)	>30	35	mm
Wheelbase (T 2.7.1)	>1525	1540	mm

3 Design Detail

Figure 25: Current proposed solution

At this early stage of design, the solution has three main focuses: reducing compliance, designing in adjustability and integrating the components. Keeping compliance to a minimum ensures that all design work is realised to its fullest potential as any deflections in components, such as bending moments in mounting brackets, can drastically affect key parameters such as camber and toe [5]. Adjustability is incredibly important both due to the complex nature of setting up a suspension system, but also because the design is unlikely to be first time right, so adding in a large amount of leeway allows the design to be refined to a much higher performance point than if adjustability was left out of the system.

3.1 External Interfaces

The current suspension system uses system of ball joints attached to mounting brackets to give the wishbones the degrees of freedom they require. A standardised system of an M8 cross bolts used to mount a rod end ball joint and located using bespoke spacers allows consistency across the vehicle, reducing complexity by shrinking the number of parts. *Figure 28* and *Figure 26* shows how these mounts are applied to the pushrod and the upper wishbone and *Figure 27* shows an exploded view of the upper wishbone mount (for parts list and full drawing see Appendix E).

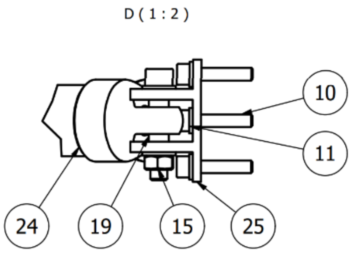


Figure 28: Inboard pushrod mounting assembly

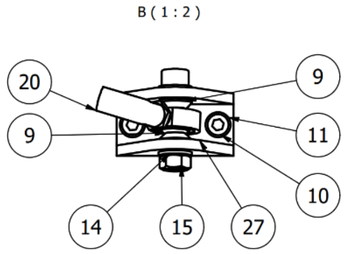


Figure 26: Inboard upper wishbone mounting assembly

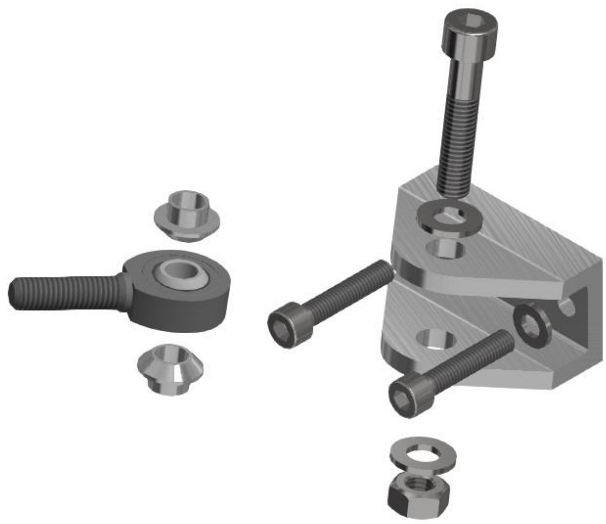


Figure 27: Exploded view of inboard upper mounting bracket

Mounting to the wheel-side structure is far more constrained and a task that was assigned to the lead upright engineer, however, designing a joint that offers the least intereference helps design process for both parties. A ball joint is insterted into the joint via a push fit and held in place with a circlip. This ball joint is then mounted via an M8 bolt to the inhub with bespoke spacers in order to maximise the usage of the available space. This setup is shown in *Figure 29*.

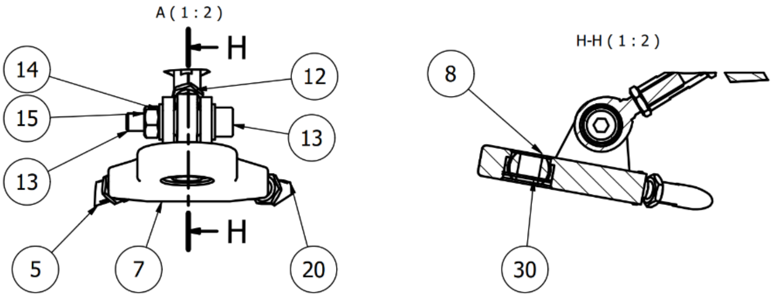


Figure 29: Upper joint assembly design and section view

3.2 Internal interfaces

The most important interface of the system internally is how the pushrod mounts between the pushrod bracket and the upper wishbone joint. The final design uses a small pushrod strut, to allow adjustable ride height, attached to a shoulder mount which is then bolted onto the coilover. The shoulders of the mount align the coilover along the axis of the pushrod so that the spring is compressed evenly and unable to move under bending. *Figure 30* shows and exploded view of the pushrod assembly.

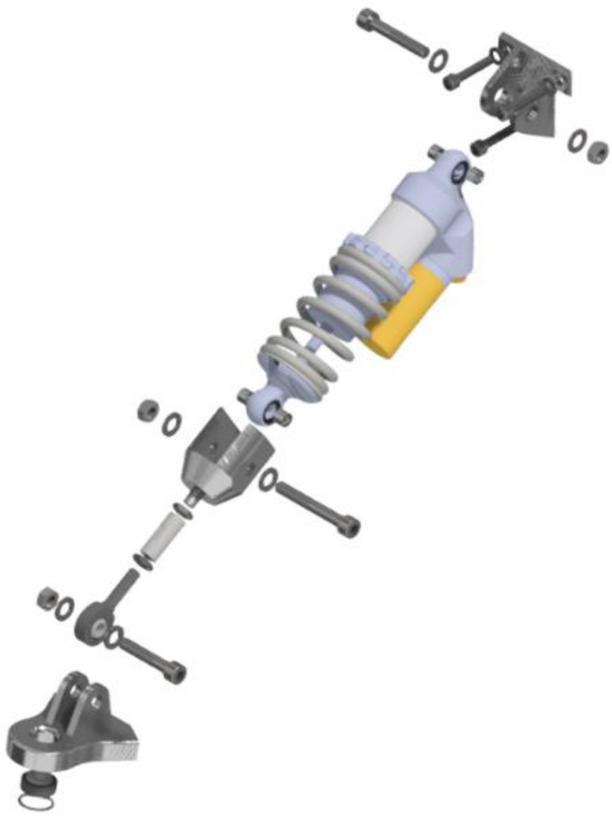


Figure 30: Exploded view of the pushrod assembly and mounting

3.3 Compliance

While completing a full compliance analysis is a future task, designing to minimise its effects is key to reducing workload further into the design process. Reducing compliance is achieved by minimising any bending moments in components in order to keep all forces in either tension or compression. A key example of this shown below in *Figure 31*, an extract from Appendix D. It shows the design of the upper wishbone joint which mounts the pushrod and both upper wishbones. The load path of the pushrod when the car is loaded is shown in blue and acts directly through the ball joint centre point, meaning the part is under compression. There is a bending moment in the two uprights used to mount the rod end ball joint, however, bending moments are minimised through use of double shear arrangements with appropriately sized uprights.

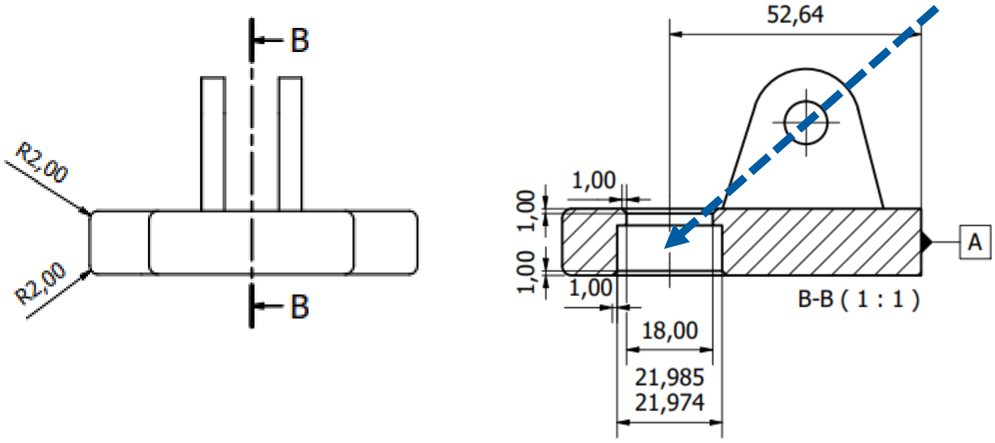


Figure 31:Upper suspension joint showing pushrod load path

3.4 Adjustability

As stated in previous sections, designing in adjustability is key to the performance of the system. The implementation of changes to key values such as ride height and camber for both performance and passing scrutineering is achieved through the change in length of the wishbones and pushrods. By using alternate screw threads on each end of the individual struts, they can be twisted to either lengthen or shorten the relevant geometries and then set via lock nuts. This is shown below in *Figure 32*, an exploded view of the lower aft wishbone and its mounting.

Figure 32: Exploded view of a lower wishbone and mounting

Figure 33 shows the current design integrated into the surrounding systems.

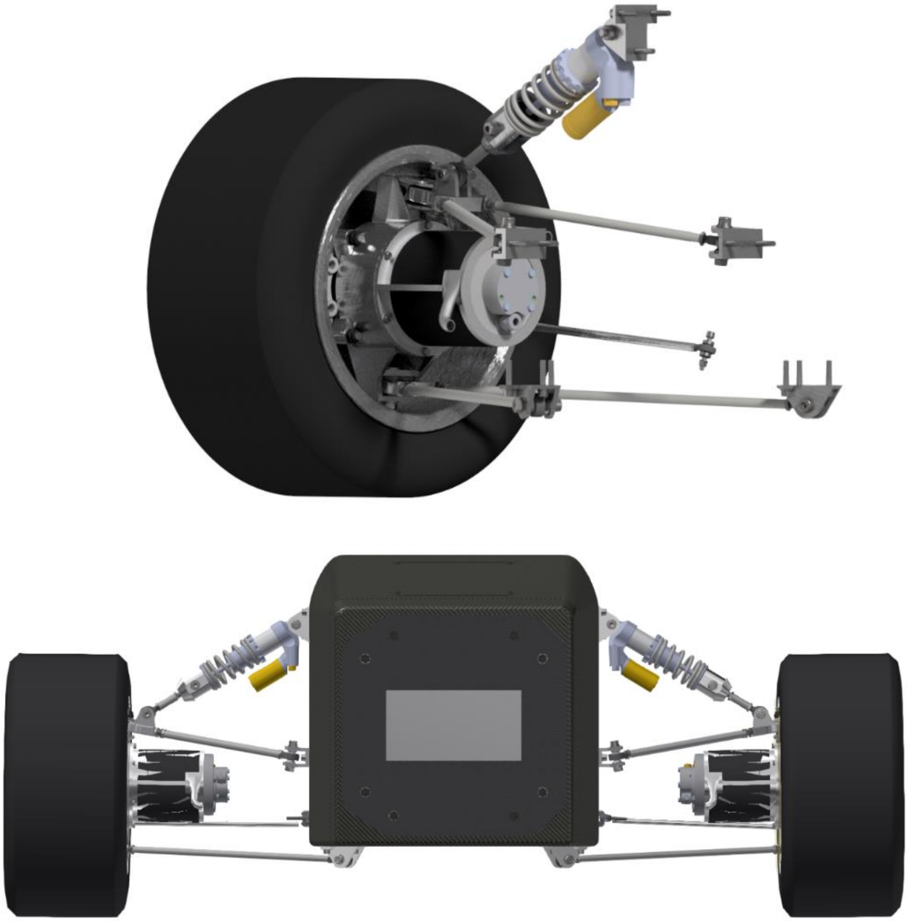


Figure 33: Design integration and wider system

4 Costing and Material Selection

While the accurate costing of a project such as this is vitally important to its success and improves chances of a smooth build phase, the difference between value and cost is important to define. Parts and processes may be extremely expensive, their cost should be measured against the value added to the car rather than compared in a vacuum. An expensive part that makes obvious improvements to a vehicles adjustability, allowing for finer tuning before competition, adds a large amount of value to the system and therefore the higher cost of this item could be acceptable.

The obvious example of this within the suspension design is the choice of Coilover. While there are far cheaper methods of actuating a suspension system, the high-quality damper requires much less range of motion to be effective and therefore the motion ratio of the pushrod has a much wider range of acceptable values. This in turn reduces the need for more complex rocker systems and arguably increases the reliability of the vehicle due to the reduction in moving parts.

4.1 Standardisation and Design for manufacture

Designing to standardise parts across the vehicle played an important role in the development of the front suspension due to the large impact it has on design complexity and costing. It is for this reason that all ball joints, coilovers and bearings are also kept as the same makes and models as the supply chain is kept far simpler.

Using the same steel and aluminium for their respective components as well as choosing a worst-case loading scenario for the wishbones and matching the sizes across the design reduces the manufacturing cost and part variety leading to a more efficient build phase. On top of this, reducing the number of parts required can help to reduce design complexity and add reliability, a key team aim, by reducing the points of potential failure. This is especially important on a student led project such as TBRe where time constraints are harsh and there is little experience amongst the members

4.2 Material and Component Selection

Selecting the most relevant components and materials for this design is reliant on a well thought out cost benefit analysis. The wishbones, in this iteration of the design, are going to be made of steel (AISI 1020 HR107) due to its high tensile strength, capable of withstanding the large compressive and tensile loads being placed on the members. While carbon wishbones were considered, complexities with attaching threading and thereby the rod ends as well as the higher price was too much of a cost for too little of a gain in performance. For the mounting components such as the brackets, joints and spacers, aluminium 7075 was chosen for its lighter weight as its high shear modulus [6]. Both materials are readily available, and the small size of the components means that issues to do with manufacturing speed and material hardness has a reduced impact.

The other key selection detail is the choice of Coilover. The Öhlins damper selected is designed specifically for used in formula student and has high quality damping characteristics with a small enough length to fit in the reduced pushrod length [2]. While cheaper systems are available, investing in a high-quality damper, a component vital to handling characteristics, is well worth the cost.

4.3 Costing

Table 13 show the bill of materials and full cost breakdown created for the suspension system per side including initial estimates of manufacturing processes required. This early estimate is extremely likely to change as these manufacturing processes begin and more reasonable quotes are given, and the inhouse manufacturing capabilities are more well understood. While this cost is a total cost of the system, many manufacturers and suppliers offer agreements on cost reduction in return for sponsorship through brand placement on the car at competition. All cost estimation uses IMechE supplied cost templates for component price estimation or supplier data.

Table 13: System Cost

Description	Volume	Supplier	Material	Cost per kg ($)	Density (g/cm^3)	Billet size (mm^3)	Billet mass (g)	Billet Cost	Volume of final part (mm^3)	Volume of material removed (mm^3)	Material removal cost ($)	Process A	Process A repititions	Process A cost per repitition ($)	Process B	Process B repititions	Process B cost per repitition ($)	Cost per unit ($)	Net cost ($)
Lower fore wishbone	2		Steel AISI 1020 107HR	2.25	7.87	40000.00	314.80	0.71	12599.177	27400.82	3.29	Drill Hole	2	0.35	Hole tap	2	0.35	4.70	9.39
Upper fore wishbone	2		Steel AISI 1020 107HR	2.25	7.87	30000.00	236.10	0.53	20145.48	9854.52	1.18	Drill Hole	2	0.35	Hole tap	2	0.35	2.41	4.83
Lower aft wishbone	2		Steel AISI 1020 107HR	2.25	7.87	40000.00	314.80	0.71	12977.03	27022.97	3.24	Drill Hole	2	0.35	Hole tap	2	0.35	4.65	9.30
Upper aft wishbone	2		Steel AISI 1020 107HR	2.25	7.87	30000.00	236.10	0.53	7876.04	22123.96	2.65	Drill Hole	2	0.35	Hole tap	2	0.35	3.89	7.77
Pushrod strut	2		Steel AISI 1020 107HR	2.25	7.87	3500.00	27.55	0.06	2319.64	1180.36	0.14	Drill Hole	2	0.35				0.55	1.11
Suspension ball joint spacer	16		Aluminium 7075	4.20	2.81	7000.00	19.67	0.08	516.08	6483.92	0.26	Drill Hole	1	0.35				0.69	11.07
Inboard pushrod bracket spacer	4		Aluminium 7075	4.20	2.81	2500.00	7.03	0.03	133.00	2367.00	0.09	Drill Hole	1	0.35				0.47	1.90
Coilover pushrod connection mount	2		Aluminium 7075	4.20	2.81	76000.00	213.56	0.90	30426.88	45573.12	1.82	Drill Hole	2	0.35	Threading	1	0.10	3.17	6.34
Outboard pushrod bracket spacer	4		Aluminium 7075	4.20	2.81	2500.00	7.03	0.03	155.00	2345.00	0.09	Drill Hole	1	0.35				0.47	1.89
Upper wishbone mounting bracket	4		Aluminium 7075	4.20	2.81	82500.00	231.83	0.97	21558.96	60941.04	2.44	Drill Hole	4	0.35				3.76	15.05
Pushrod & lower wishbone mounting bracket	6		Aluminium 7075	4.20	2.81	95000.00	266.95	1.12	16530.32	78469.68	3.14	Drill Hole	5	0.35				4.61	27.66
Upper wishbone joint	2		Aluminium 7075	4.20	2.81	250000.00	702.50	2.95	49657.09	200342.92	8.01	Drill Hole	3	0.35	Hole tap	2	0.35	11.66	23.33
Lower wishbone joint	2		Aluminium 7075	4.20	2.81	60000.00	168.60	0.71	29981.30	30018.70	1.20	Drill Hole	3	0.35	Hole tap	2	0.35	2.61	5.22
M8 Rod end ball joint	2	schaeffler																25.62	51.24
M8 ball joint	4	schaeffler																46.06	184.26
M6x30 Hexagon socket head cap screw	26		Steel Grade 12.9															0.29	7.54
M8 Plain washer	26		Steel Grade 12.9															0.02	0.52
M8 Hexagon thin nuts (chamfered)	28		Steel Grade 12.9															0.02	0.56
M8x40 Hexagon Socket head cap screw	10		Steel Grade 12.9															0.29	2.90
M6 Plain washer	28		Steel Grade 12.9															0.02	0.56
M8 Hexagon nut	10		Steel Grade 12.9															0.02	0.20
M8 threaded stud	8		Steel Grade 12.9															0.29	2.32
M8x55 Hexagon socket head cap screw	2		Steel Grade 12.9															0.29	0.58
M8x45 Hexagon socket head cap screw	2		Steel Grade 12.9															0.29	0.58
Ohlins Formula Student TTX 25 MkII FSAE 200/57mm	2	Öhlins																650.00	1300.00
70 N/mm spring	2	Öhlins																30.00	60.00
Circlips for holes	4																	0.02	0.08
																	Full assembly cost:	$1,736.19	
																		£1,232.70	

Figure 34 shows a breakdown of the costs of each component type and how they compare to each other. It is clear that the springs and dampers are the most expensive component, however, these parts are extremely critical to the success of the systems handling characteristics. It is therefore deemed a worthy expenditure.

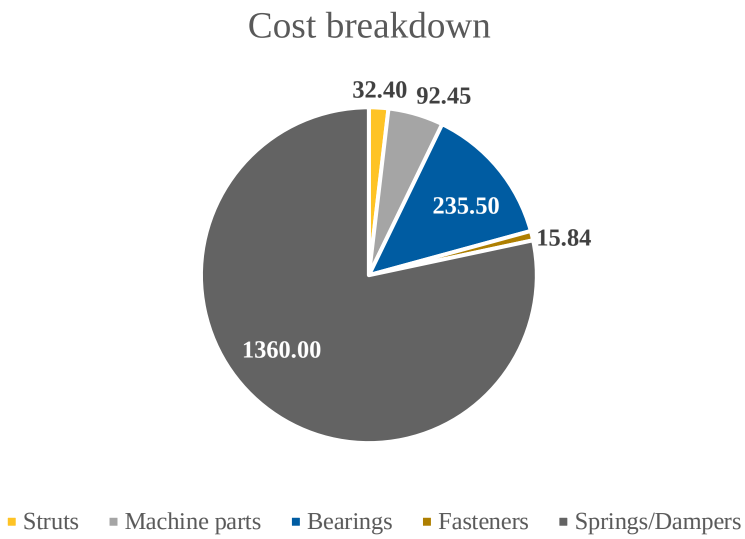


Figure 34: Cost breakdown by component type

5 Critical Review and Further Work

The early stages of this project required rapid growth in both technical knowledge and skills, leading to some preliminary decisions that, given the chance to repeat the process, should be approached differently. Integration with the other sub teams was completed inefficiently, with hours wasted on discussing potential mounting locations without data and analysis to back up any claims, instead relying on best practice and recommended values from textbooks and online sources. While this was remedied, often times the most important decisions are those that are made early in progress and therefore getting it correct from the start could dramatically improve the entire vehicle design and remove any relatively unfounded decisions within the suspension system, the vehicle dynamics system and the TBRe22 team as a whole.

5.1 Further work

The future tasks can be split into two sections: design updates and testing. These will take place over the course of a year and will culminate in the car competing at FSUK at Silverstone. *Table 14* shows the key areas of design that need progressing before the car can be constructed.

Table 14: Future design tasks

Further geometry analysis	While the geometry can be considered final, adjustment to camber gain in both roll and heave is a simple change to make based on the positioning of the wishbone mounts if new information requires a change in design philosophy. Some exampled of these changes are a dramatic shift in body roll requirements or chassis requirement changes. Further development time is needed into the pushrod arrangement as there are some underexplored options for motion ratio management that will require integration discussions with the chassis sub team. While sensitivity studies showed that small roll centre changes have little effect on the overall handling characteristics, further research is needed to validate

	this. Anti-geometries also require analysis and understanding to define whether or not they are needed to counter the pitching of the car.
Spring rate adjustment and damping coefficient selection	There is likely to a large change in the values used as inputs for all simulations and models in the coming months. This will lead to a redesign of worst-case cornering situations as aerodynamic downforce and motor output torques change and therefore the required spring rates are likely to change. This will in turn lead to development of new damping coefficients. The setup of the springs and dampers is adjustable up until the events in order to extract the maximum potential from the vehicle dynamic system. Analysis of pitching under breaking is needed to understand potential issues with the front wing colliding with the ground in heavy breaking scenarios.
Detailed load calculation	A full statics model will be created in order to fully analyse the loads in the wishbones to allow for far more accurate wishbone load cases based on the developments in the various model accuracies.
Component sizing	With more realistic load values, detailed FEA can take place on all the components to test for compliance and load baring capabilities. This can be used to optimise component weights to reduce the mass of the suspension and improve the vehicles yaw moment of inertia.

5.2 Testing

A sign of a good suspension system is the effect it has on the tyres. One way that testing is planned to take place once the car has been constructed is through analysis of the tyres after test runs and track days. Assessing where the tyre is worn, the kind of wear it has experienced and how fatigued the rubber is can be a vital sign of the kind of performance the tyre was outputting and how it can be adjusted. If the tyre is waring considerably more on the inside it is a sign of too much negative camber and indicates that static camber, or the camber gain, needs to change (example shown in *Figure 35*). This method of testing can be used even if the tyre is not being pushed to its maximum through the use of temperature probes. By checking the temperature gradient across the tyre as the setup is adjusted, a dataset can be built up that can be used to inform optimally vehicle setup.

Figure 35: Example of camber wear on a race tyre [7]

A method of testing that can be implemented before the vehicle is completed is via a rig. By attaching the wishbones to a temporary mount in positions correct relative to where they will end up on the chassis, the wheel can be moved through its travel path to correlate the simulations to the final design, improving the usefulness of the simulations and models themselves by providing real world data.

6 Conclusions

The suspension provides adjustable characteristics that have been centred around simulation and modelling of relevant tracks and vehicle characteristics, with the rules being abided by throughout the process and the necessary mechanisms and leeway in place to avoid an arduous scrutineering process and avoid risking the vehicle failing to enter any events. The recommended static camber of roughly -1.1 ° offers large amounts of flexibility based on the course and conditions while combining with the steering characteristics to give a coherent and effective front suspension system. While the motion ratio is not as close to some initial targets, it is within a working range that is acceptable for the springs that will be purchased as well as the chosen damping characteristics. Preliminary load analysis has set a magnitude of forces than can be expected during worst case scenarios providing vital information used to size components further into the design process. The mechanical design has yet to be optimised but a working solution is in place that, if required, could be constructed and used as a viable suspension system at competition as of this report. The future of the design process is clearly mapped out and the costs and processes associated with manufacturing assessed, with an optimistic final system cost of £1,232.70. This section of the design process will provide a strong foundation for father design work and eventually success at both national and international competition.

7 Glossary

Term	Definition
Compliance	The bending of components causing unwanted changes in key characteristics of the suspension system such as camber
Heave	Vertical movement of the tyre relative to the chassis
Jounce	The remaining spring travel after the vehicle is on the ground with driver seated
Sprung mass	The mass of the chassis body and all the components it contains
Unsprung mass	The mass of the wheel and all the components it contains
Wishbone	A pair of struts connecting the chassis to the upright
Coilover	A damper encased in a spring
KPI	King pin inclination, the angle between the ground and an axis drawn between the suspension mounts in the lateral direction
Caster	The angle between the ground and an axis drawn between the suspension mounts in the longitudinal direction
Pushrod	The strut that carries the vertical load from the tyre to the chassis, usually containing a spring damper element
Contact patch	The area of the tyre which is in contact with the track.
Roll gradient	The degrees of body roll induced per g of lateral acceleration

8 References

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Appendix

Appendix A - PDS

No.	MoSCoW	Requirement	Success Criteria	Source	Assesment	Reference
		Overall Vehicle				
1.1	M	CoM height to pass the tilt test	<332mm	FSUK20 rulebook	CAD	FSR20 - IN7
1.2	S	Minimise CoM height	<300mm	TBRe Team Decision	CAD	
1.3	S	Longitudinal position of centre of mass	No further than 44:56	Analysis of vehicle dynamic systems	CAD	
1.4	S	Lateral position of centre of mass	50:50	TBRe Team Decision	CAD	
1.5	M	Wheelbase	>1525mm	FSUK20 rulebook	CAD	FSR20 - T 2.7.1
1.6	S	Gross mass (wet)	<250kg	TBRe Team Decision	CAD	
1.7	M	The system must be designed to reliably perform for the duration of the product life cycle	No component failures before 1500km	Team Aim & Calculation	FEA, fatigue analysis, testing	
		Economics				
2.1	W	Deliver vehicle dynamics system below budget	< £8000	TBRe Team Decision	BOM	
		Assembly & Manufacturing				
3.1	M	Design for manufacture	Components can be made with available resource		CAD & Technical Drawings	
3.2	M	Designed for assembly; all parts are designed with suitable tolerances	Components can be assembled with available resource		CAD & Technical Drawings	
3.3	C	Minimise number of parts to reduce probability of component failure and assembly complexity			CAD design review	
3.4	S	Deliver a fully-assembled vehicle on time to allow time for testing and tuning	TBC	TBRe Team Decision	Project Plan	
		Enviromental & Lifecycle Considerations				
4.1	W	Use standarised bought-in parts to minimise component waste where possible		TBRe Team Decision	BOM	
4.2	M	Components must not fail during planned product lifetime (50 battery charge cycles)	No component failures before 1500km	TBRe Team Decision	FEA, Testing and Fatigue Analysis	
4.3	W	Designs should reduce the amount of material used where possible		TBRe Team Decision	CAD	
		Rules and Regulation				
5.1	M	Designs must be compliant with all relevant FSUK and FSUK Supplementary rules	Compliance with all regulations		CAD review against rulebook	FSUK and FSUK Supplementary rulebook
5.2	C	Design should Compliant with all European FS rules (FSG, FSN, FSS, FSATA, FSCZ, FSA, FSEAST, FSR, FSSwitzeland).	Compliance with all regulations		CAD review against rulebook	FS Europe rulebooks

No.	MoSCoW	Requirement	Success Criteria	Source	Assesment	Reference
Front Suspension						
6.1	M	Front track width	1200mm	Agreement With Rear Suspension + Analysis	CAD	
6.2	S	Static Roll centre height	<45mm >0mm	Agreement With Rear Suspension + Analysis	CAD	
6.5	S	Maximum body roll	<1.5 degrees	Agreement With Rear Suspension + Analysis	OptimumKinematics	
6.6	S	Maximum body roll gradient	<1°/g	TBRe Team Decision + Analysis	OptimumKinematics	
6.7	S	King pin inclination (lateral)	<15°	Agreement With Steering	OptimumKinematics	
6.8	S	Scrub radius	< 40 mm	Agreement With Steering	OptimumKinematics	
6.9	S	Caster angle	<5° >3°	Agreement With Steering	OptimumKinematics	
6.10	S	Mechanical trail	20±5 mm	Agreement With Steering	OptimumKinematics	
6.11	S	Static camber	adjustable: 0 -> -3°	TBRe Team Aim + Best Practice	CAD	
6.12	C	Static toe	adjustable: 0 -> 1°	TBRe Team Aim + Best Practice	CAD	
6.14	C	Roll centre movement	Minimal	Race Car Design	OptimumKinematics	(see bibliography)
6.15	S	Target outise wheel camber during cornering	-1°	Agreement With Rear Suspension + Analysis	CAD	
6.16	M	Minimum wheel travel	>50mm	FSUK20 rulebook	CAD + real world testing	FSR2020 - T 2.3.1
6.17	M	Minmum travel in jounce	>25mm	FSUK20 rulebook	CAD + real world testing	FSR2020 - T 2.3.1
6.18	M	Minimum ride height (static, with driver)	>30mm	FSUK20 rulebook	CAD + real world testing	FSR2020 - T 2.3.2
6.19	M	Suspension mount visibility	mounting should be visible at technical inspection, either by direct view or removing covers.	FSUK20 rulebook	CAD + real world testing	FSR2020 - T 2.3.3

Appendix B – Weighting matrix for pushrod decision making

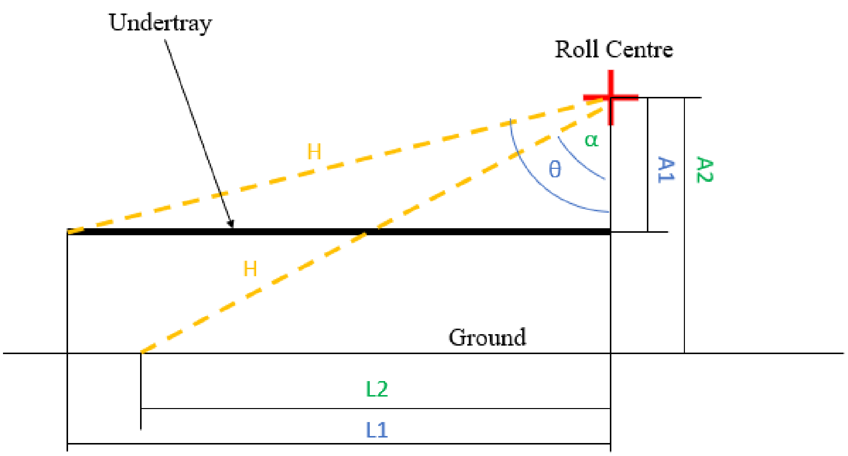
	Adjustability	Cost	Integration	Design Flexibility	Manufacturing Complexity	Mass	Reliability
Adjustability		0	1	0	0	0	1
Cost	1		1	1	1	1	1
Integration	0	0		0	0	0	0
Design Flexibility	1	0	1		1	1	1
Manufacturing Complexity	1	0	1	0		1	1
Mass	1	0	1	0	0		1
Reliability	0	0	1	0	0	0	
Total (+1 for base weight)	5	1	7	2	3	4	6
Weighting	17.86	3.57	25.00	7.14	10.71	14.29	21.43

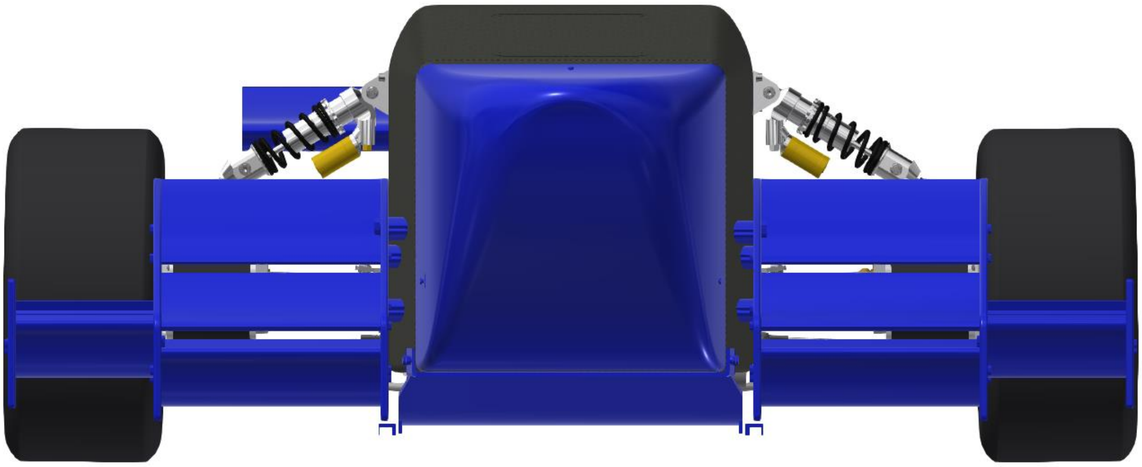
Appendix C – Maximum allowable body roll calculations

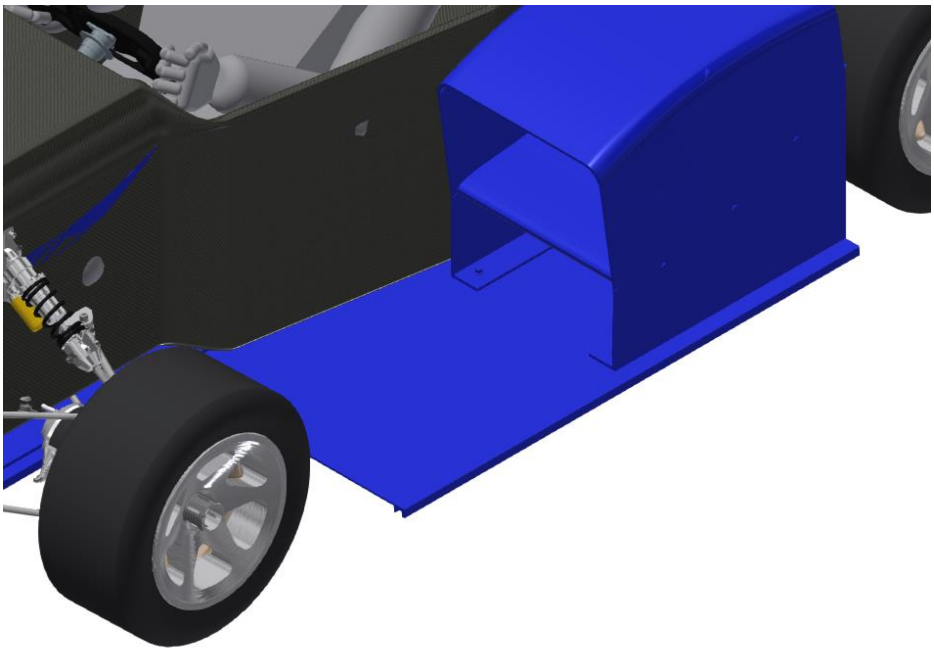
Example of how the maximum allowable body roll can be set by the undertray dimensions:

	Value	unit
L1	703	mm
A1	2	mm
A2-A1	30	mm

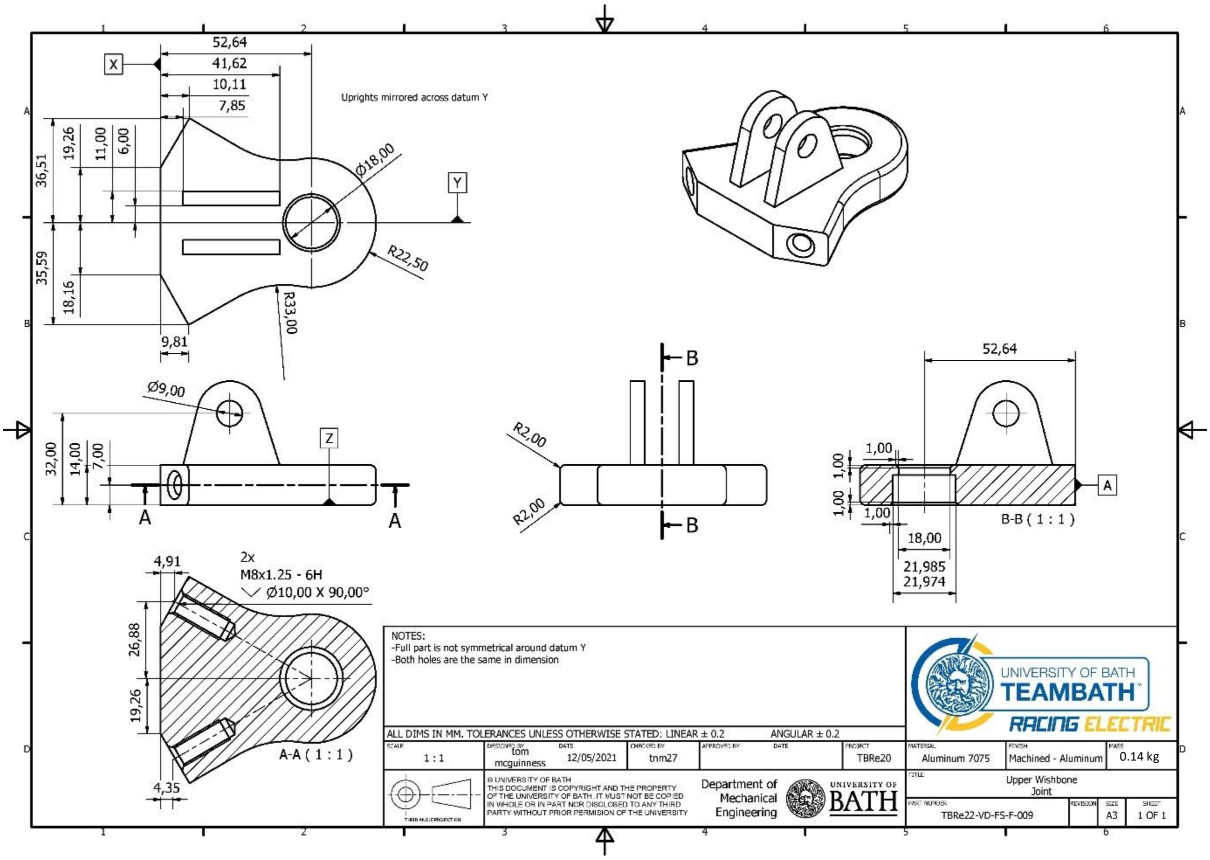
	Value	unit	Value	unit
theta	1.5679514	rad	89.836997	deg
H	703.00284	mm		
alpha	1.5252616	rad	87.391051	deg
roll range	0.0426898	rad	2.445946	deg



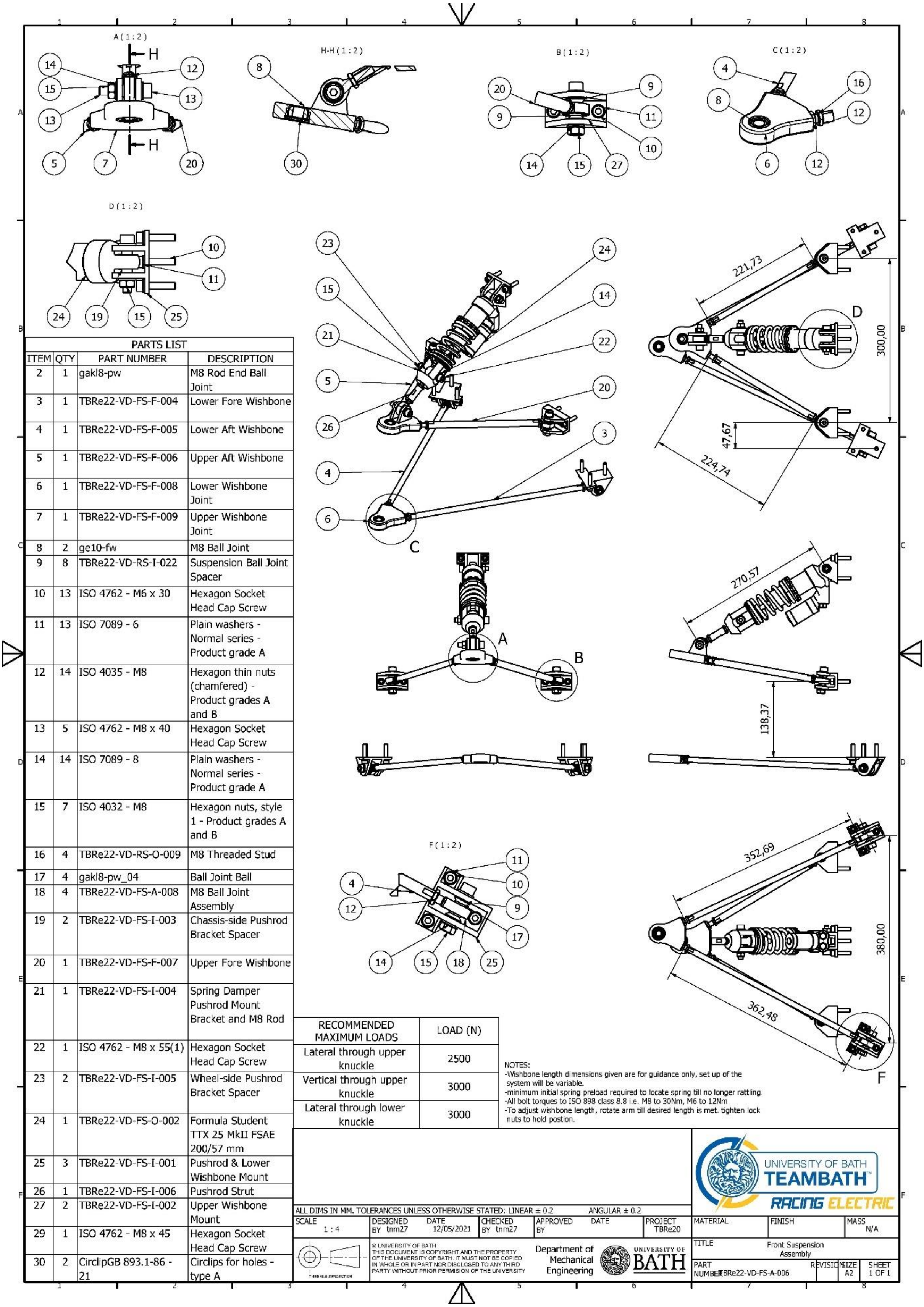




Appendix D – Part drawing: Upper Wishbone Joint



Appendix E – Assembly drawing



PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
2	1	gakl8-pw	M8 Rod End Ball Joint
3	1	TBRe22-VD-FS-F-004	Lower Fore Wishbone
4	1	TBRe22-VD-FS-F-005	Lower Aft Wishbone
5	1	TBRe22-VD-FS-F-006	Upper Aft Wishbone
6	1	TBRe22-VD-FS-F-008	Lower Wishbone Joint
7	1	TBRe22-VD-FS-F-009	Upper Wishbone Joint
8	2	ge10-fw	M8 Ball Joint
9	8	TBRe22-VD-RS-I-022	Suspension Ball Joint Spacer
10	13	ISO 4762 - M6 x 30	Hexagon Socket Head Cap Screw
11	13	ISO 7089 - 6	Plain washers - Normal series - Product grade A
12	14	ISO 4035 - M8	Hexagon thin nuts (chamfered) - Product grades A and B
13	5	ISO 4762 - M8 x 40	Hexagon Socket Head Cap Screw
14	14	ISO 7089 - 8	Plain washers - Normal series - Product grade A
15	7	ISO 4032 - M8	Hexagon nuts, style 1 - Product grades A and B
16	4	TBRe22-VD-RS-O-009	M8 Threaded Stud
17	4	gakl8-pw_04	Ball Joint Ball
18	4	TBRe22-VD-FS-A-008	M8 Ball Joint Assembly
19	2	TBRe22-VD-FS-I-003	Chassis-side Pushrod Bracket Spacer
20	1	TBRe22-VD-FS-F-007	Upper Fore Wishbone
21	1	TBRe22-VD-FS-I-004	Spring Damper Pushrod Mount Bracket and M8 Rod
22	1	ISO 4762 - M8 x 55(1)	Hexagon Socket Head Cap Screw
23	2	TBRe22-VD-FS-I-005	Wheel-side Pushrod Bracket Spacer
24	1	TBRe22-VD-FS-O-002	Formula Student TTX 25 MkII FSAE 200/57 mm
25	3	TBRe22-VD-FS-I-001	Pushrod & Lower Wishbone Mount
26	1	TBRe22-VD-FS-I-006	Pushrod Strut
27	2	TBRe22-VD-FS-I-002	Upper Wishbone Mount
29	1	ISO 4762 - M8 x 45	Hexagon Socket Head Cap Screw
30	2	CirclipGB 893.1-86 - 21	Circlips for holes - type A

RECOMMENDED MAXIMUM LOADS	LOAD (N)
Lateral through upper knuckle	2500
Vertical through upper knuckle	3000
Lateral through lower knuckle	3000

NOTES:
-Wishbone length dimensions given are for guidance only, set up of the system will be variable.
-minimum initial spring preload required to locate spring till no longer rattling.
-All bolt torques to ISO 898 class 8.8 i.e. M8 to 30Nm, M6 to 12Nm
-To adjust wishbone length, rotate arm till desired length is met. tighten lock nuts to hold postion.

ALL DIMS IN MM. TOLERANCES UNLESS OTHERWISE STATED: LINEAR ± 0.2 ANGULAR ± 0.2

SCALE	DESIGNED BY	DATE	CHECKED BY	APPROVED BY	DATE	PROJECT	MATERIAL	FINISH	MASS
1 : 4	tnm27	12/05/2021	tnm27			TBRe20			N/A

THIRD ANGLE PROJECTION

Department of Mechanical Engineering

UNIVERSITY OF BATH

TITLE Front Suspension Assembly

PART NUMBER	REVISION	SIZE	SHEET
TBRe22-VD-FS-A-006		A2	1 OF 1