Statement of Contributions

We the undersigned certify that the content submitted in this report detailing the management and design activities of the 2014-2015 Brunel Racing Formula Student team (vehicle group) is wholly original and solely the work of the contributor (unless otherwise stated whereby credit is given to the author).

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<td>1 DIMENSION</td>
</tr>
<tr>
<td>2D</td>
<td>2 DIMENSIONAL</td>
</tr>
<tr>
<td>3D</td>
<td>3 DIMENSIONAL</td>
</tr>
<tr>
<td>A</td>
<td>AERODYNAMIC DEPARTMENT</td>
</tr>
<tr>
<td>AJ</td>
<td>ARVYDAS JASELSKIS</td>
</tr>
<tr>
<td>AMOLED</td>
<td>ACTIVE-MATRIX ORGANIC LIGHT-EMITTING DIODE</td>
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<tr>
<td>AWG</td>
<td>AMERICAN WIRE GAUGE</td>
</tr>
<tr>
<td>BOM</td>
<td>BILL OF MATERIALS</td>
</tr>
<tr>
<td>BR</td>
<td>BRUNEL RACING</td>
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<td>BR-12</td>
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<tr>
<td>CE</td>
<td>CRAIG ELLIOTT</td>
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<tr>
<td>CEST</td>
<td>CENTRAL EUROPEAN STANDARD TIME</td>
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<tr>
<td>CFD</td>
<td>COMPUTATIONAL FLUID DYNAMICS</td>
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<tr>
<td>CFRP</td>
<td>CARBON FIBRE REINFORCED PLASTIC</td>
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<tr>
<td>CH</td>
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<tr>
<td>CoG</td>
<td>CENTRE OF GRAVITY</td>
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<td>CS</td>
<td>CATHERINE SELLS</td>
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<td>CTSR</td>
<td>CHASSIS TORSIONAL STIFFNESS REQUIREMENT</td>
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<td>DC&amp; E</td>
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<td>DRAG REDUCTION SYSTEM</td>
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<tr>
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<td>FORMULA SOCIETY OF AUTOMOTIVE ENGINEERS</td>
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<tr>
<td>FS-CZ</td>
<td>FORMULA STUDENT CZECH REPUBLIC</td>
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<tr>
<td>FSG</td>
<td>FORMULA STUDENT GERMANY COMPETITIONS</td>
</tr>
<tr>
<td>FS-UK</td>
<td>FORMULA STUDENT - UNITED KINGDOM COMPETITION</td>
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<tr>
<td>FVIC</td>
<td>FRONT VIEW INSTANT CENTRE</td>
</tr>
<tr>
<td>GG</td>
<td>GARETH GWILLIAM</td>
</tr>
<tr>
<td>GM</td>
<td>GABRIELE MOROCUTTI</td>
</tr>
<tr>
<td>GUI</td>
<td>GRAPHICAL USER INTERFACE</td>
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**HIL** HARDWARE-IN-THE-LOOP

**JS** JOHN SHARPE

**LCO** HOOSIER TYRE COMPOUND LCO

**LEDs** LIGHT EMITTING DIODES

**LiFePO₄** LITHIUM IRON PHOSPHATE

**Li-ion** LITHIUM IRON

**NACA** NATIONAL (USA) ADVISORY COMMITTEE FOR AERONAUTICS

**NY** NEW YORK

**PA** POWERTRAIN ANCILLIARIES DEPARTMENT

**PC** PERSONAL COMPUTER

**PCD** PITCH CENTRE DIAMETER

**PDS** PRODUCT DESIGN SPECIFICATION

**PH** POWERTRAIN HARDWARE DEPARTMENT

**PoE** PORTFOLIO OF EVIDENCE

**PPE** PERSONAL PROTECTIVE EQUIPMENT

**PS** POWERTRAIN SIMULATION DEPARTMENT

**psi** PRESSURE - POUNDS PER SQUARE INCH

**PT** POWERTRAIN TESTING DEPARTMENT

**R25B** HOOSIER TYRE COMPOUND R25B

**RPM** REVOLUTIONS PER MINUTE

**SAE** SOCIETY OF AUTOMOTIVE ENGINEERS

**SED** BRUNEL UNIVERSITY SCHOOL OF ENGINEERING AND DESIGN(-2015. NOW CEPDS)

**SES** STRUCTURAL EQUIVALENCY SPREADSHEET

**SR** SLIP RATIO

**TC** TRACTION AND LAUNCH CONTROL DEPARTMENT

**TIRF** CALSPAN TIRE RESEARCH FACILITY

**TM** THOMAS MCLAREN

**TPV** THIRD PARTY VERIFICATION

**TTC** TYRE TEST CONSORTIUM

**US** UNSPRUNG DEPARTMENT

**USA** UNITED STATES OF AMERICA

**UTC** COORDINATED UNIVERSAL TIME

**VS** VEHICLE SIMULATION

**WW2** WORLD WAR 2
### Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tr>
<td>%\text{Braking}_{\text{front}}</td>
<td>PERCENTAGE OF BRAKING PERFORMED BY THE FRONT AXLE</td>
</tr>
<tr>
<td>A</td>
<td>FRONTAL AREA OF THE VEHICLE</td>
</tr>
<tr>
<td>( A_{mc} )</td>
<td>AREA OF MASTER CYLINDERS</td>
</tr>
<tr>
<td>( \alpha_v )</td>
<td>LONGITUDINAL ACCELERATION</td>
</tr>
<tr>
<td>( \alpha_y )</td>
<td>LATERAL ACCELERATION</td>
</tr>
<tr>
<td>( \alpha )</td>
<td>ANGLE OF ATTACK</td>
</tr>
<tr>
<td>( \alpha_0 )</td>
<td>CAMBER ANGLE OF WING</td>
</tr>
<tr>
<td>( \alpha_{\text{Acceleration}} )</td>
<td>ANGLE OF ACCELERATION FROM THE POSITIVE LONGITUDINAL AXIS ON A PLANE PARALLEL WITH THE GROUND AND OFFSET BY THE HEIGHT OF THE CENTRE OF GRAVITY</td>
</tr>
<tr>
<td>( C_D )</td>
<td>COEFFICIENT OF DRAG</td>
</tr>
<tr>
<td>( C_d )</td>
<td>COEFFICIENT OF DRAG</td>
</tr>
<tr>
<td>( C_L )</td>
<td>COEFFICIENT OF LIFT</td>
</tr>
<tr>
<td>( \text{CoG}_Z )</td>
<td>HEIGHT OF THE CENTRE OF GRAVITY</td>
</tr>
<tr>
<td>( D )</td>
<td>SECOND MOMENT OF AREA EQUIVALENT DIAMETER</td>
</tr>
<tr>
<td>( E )</td>
<td>ELASTIC MODULUS</td>
</tr>
<tr>
<td>( F_{\text{bp}} )</td>
<td>BRAKE PEDAL FORCE</td>
</tr>
<tr>
<td>( F_{\text{clamp}} )</td>
<td>BRAKE CLAMPING FORCE</td>
</tr>
<tr>
<td>( F_d )</td>
<td>FORCE DUE TO AERODYNAMIC DRAG</td>
</tr>
<tr>
<td>( F_{\text{FA}} )</td>
<td>NORMAL LOAD ON THE FRONT AXLE</td>
</tr>
<tr>
<td>( F_{fr} )</td>
<td>FRICTIONAL FORCE</td>
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<tr>
<td>( F_p )</td>
<td>PULL-ROD LOAD</td>
</tr>
<tr>
<td>( f_r )</td>
<td>RIDE FREQUENCY (Hz)</td>
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<td>( F_x )</td>
<td>LONGITUDINAL LOAD/FORCE</td>
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<td>( F_z )</td>
<td>NORMAL LOAD</td>
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<tr>
<td>( Fr )</td>
<td>ROLLING RESISTANCE</td>
</tr>
<tr>
<td>( g )</td>
<td>ACCELERATION DUE TO GRAVITY</td>
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<tr>
<td>( G )</td>
<td>SHEAR MODULUS OF A MATERIAL</td>
</tr>
<tr>
<td>( h )</td>
<td>HEIGHT OF THE CENTRE OF GRAVITY</td>
</tr>
<tr>
<td>( H )</td>
<td>DISTANCE BETWEEN THE COG_Z AND THE ROLL CENTRE HEIGHT AT THE LONGITUDINAL POSITION OF THE COG</td>
</tr>
<tr>
<td>( h_{cg} )</td>
<td>HEIGHT OF THE CENTRE OF GRAVITY</td>
</tr>
<tr>
<td>( K_{\text{COMP}} )</td>
<td>TORSIONAL STIFFNESS OF THE VEHICLES COMPLIANCE</td>
</tr>
<tr>
<td>( K_{\text{FRAME}} )</td>
<td>CHASSIS TORSIONAL STIFFNESS ALONG THE SAE X-AXIS</td>
</tr>
<tr>
<td>( K_{\phi_A} )</td>
<td>ROLL STIFFNESS</td>
</tr>
<tr>
<td>( K_{\phi_{\text{des}}} )</td>
<td>ROLL STIFFNESS FOR DESIRED ROLL GRADIENT</td>
</tr>
<tr>
<td>( K_{\phi_{FA}} )</td>
<td>FRONT ARB ROLL STIFFNESS</td>
</tr>
<tr>
<td>( K_{\phi_{RA}} )</td>
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<tr>
<td>( K_{\text{ROLL}} )</td>
<td>ROLL STIFFNESS</td>
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<td>( K_{\text{ROLL FR}} )</td>
<td>FRONT SUSPENSION ROLL STIFFNESS</td>
</tr>
<tr>
<td>( K_{\text{ROLL Re}} )</td>
<td>REAR SUSPENSION ROLL STIFFNESS</td>
</tr>
<tr>
<td>( K_{\text{SPRING}} )</td>
<td>SPRING STIFFNESS</td>
</tr>
<tr>
<td>( K_t )</td>
<td>TYRE STIFFNESS</td>
</tr>
</tbody>
</table>
**K_{TOTAL}**  TOTAL ROLL STIFFNESS

**K_w**  WHEEL RATE/STIFFNESS

**L_{ARB}**  LENGTH OF ARB ARM

**L_{steering arm}**  PERPENDICULAR DISTANCE OF THE OUTBOARD TOE ARM NODE TO THE STEERING AXIS

**L_{TUBE}**  LENGTH OF ARB TUBE

**L10**  BEARING LIFE RATING IN REVOLUTIONS

**L10h**  BEARING LIFE RATING IN HOURS

**M**  MASS (KG)

**M_{sprung}**  SPRUNG MASS OF THE CAR

**μ_{bp}**  BRAKE PAD COEFFICIENT OF FRICTION

**MR**  MOTION RATION OF THE SUSPENSION SPRING

**MR_{FARB}**  MOTION RATION OF FRONT ARB

**MR_{FARB H}**  MOTION RATION OF FRONT ARB IN HEAVE (MM/MM)

**MR_{FARB R}**  MOTION RATION OF FRONT ARB IN ROLL (DEG ROLL/DEG TWIST)

**MR_{RARB}**  MOTION RATION OF REAR ARB

**N_{magic}**  MAGIC NUMBER (LATERAL LOAD TRANSFER DURING CORNERING)

**P**  LOAD

**P**  BEARING LIFE RATING COEFFICIENT

**P_{mc}**  MASTER CYLINDER PRESSURE

**φ_{des}**  DESIRED ROLL GRADIENT OF THE VEHICLE (DEGREES OF VEHICLE ROLL PER G OF LATERAL ACCELERATION)

**r**  RADIUS REQUIRED FOR THE CALCULATED LOAD

**r_{pinion}**  RADIUS OF THE STEERING RACK PINION

**R_x**  FORCE DUE TO ROLLING RESISTANCE

**ρ**  DENSITY OF LOCAL FLUID

**σ_{tensile}**  TENSILE YIELD STRENGTH (MPA)

**SVSA_H**  SIDE VIEW SWING ARM HEIGHT

**SVSA_L**  SIDE VIEW SWING ARM LENGTH

**t**  TRACK WIDTH

**t_f**  FRONT TRACK WIDTH

**T_{pinion}**  TORQUE IN STEERING PINION

**Trail_{Dynamic}**  DYNAMIC TRAIL OF THE TYRE = MECHANICAL TRAIL + PNEUMATIC TRAIL

**V**  VELOCITY OF VEHICLE

**W**  WEIGHT OF THE CAR = CAR MASS MULTIPLIED BY GRAVITATIONAL ACCELERATION

**W_{DF}**  WEIGHT DISTRIBUTION ON THE FRONT AXLE

**w_t**  TRACK WIDTH

**WB**  WHEELBASE

**WT**  WEIGHT TRANSFER
1 Introduction (GG)

1.1 The scope of this report

This report outlines the work and findings of the Vehicle Group Project for BR-16 within the Brunel Racing Formula Student team. This project focuses on the development of the many of the areas typically dealt with by a motorsport vehicle constructor; including all work on the tyres, suspension, vehicle dynamics modelling, chassis, aerodynamics, electronics and all driver interfaces.

1.2 The Task - A Brief outline of Formula Student /FSAE

The overall project aims to produce a vehicle for entry into the Formula Student competition. Formula Student is an engineering design competition which challenges students to build a “weekend racer” [1]. Cars are assessed by experienced engineers in the cost, business and design events and dynamically tested on-track to assess the performance and desirability of the vehicles. Figure 1.2.1 shows a breakdown of the points scoring system for both FS-UK and FSG [2] [3].

![Figure 1.2.1. Pie chart detailing competition points distribution](image)
2 Literature Review

2.1 Project Management (GG, EJ, CE)

2.1.1 Project Planning – CE

In project planning, there are a number of recognised steps to complete a project successfully [4].

Aim of the project: The outcomes of the project need to be fully defined and understood by all those in the team. The benefits of completing the project also need to be made clear to all involved. From this, a list of project deliverables needs to be devised.

Resources: In order to achieve the outputs, resources need to be identified. These can consist of manpower, money and time. The resources required need to be identified at the project planning stage, as some tasks need to be completed by assigning more people to the problem.

Management structure: the decision makers within the project need to be established. When and what type of information that needs to be shared between managers and other members of the team and how this should be communicated i.e. in meetings between the various parts of the team. Also, the transmission and communication of information with those higher up in the organisation is key.

Milestones: A project can be managed more effectively if it is broken down into discrete parts. The inclusion of milestones to mark an important deliverable or submission allows the progress of the project to be tracked. This allows action to be taken if a milestone is approaching, and the level of progress for that particular section is behind schedule.
Dependencies: Now that all of the tasks within the project have been identified, the order of completion needs to be established. From this, a critical path that the project needs to follow in order for completion can be determined.

Risks: There is always risk within a project, and this needs to be accounted for in the planning stage. This can be covered by a risk register document.

Schedule: Once all of the tasks, milestones, resources and dependencies have been established, a document showing how they are set out over time needs to be created. This is often in the form of a Gantt chart, which at the beginning will give a high-level overview of the whole project. As time passes, the plan can be developed to become more accurate and representative of the timescales and resources required to successfully complete the project.

The Brunel Racing management team must also take into account the other university responsibilities throughout the year. Priorities such as assignments and exams take preference at certain times of the year, and this must be reflected in the project plan. The FSUK and FSG submission deadlines can be used as a basis for project milestones.

2.1.2 Budget and Expenditure (EJ)

Expenditure management is a key part of any business and project, and plays a key role in strict budget limited projects such as the Formula Student project. When tracking project expenditure, it is important to have a plan of expected expenditure over the course of the project. The expected expenditure is commonly plotted on a graph with the project budget and the actual spend over time to gauge if the project is over or underspending. [4]
One of the primary requirements of expenditure tracking is keeping the sponsor, in this case the University, happy with the project by producing regular organised and concise reports on how much has been spent on what items, and how much you want/need to spend and when. [5]

2.2  Design Management: Third party verification (TPV) (TM)

2.2.1  Third Party Verification Structure

Third party verification can be defined as “A process where an independent party is asked to confirm whether the client's information is accurate or to validate their intent” [6]. This structure was employed to the design process of the vehicle. After components had been designed by design engineers and technical drawing produced a two person review system was introduced. A group of four managers were named as “drawing checkers” (AJ, GG, TM, and JS). These managers were selected on the basis of previous experience of high-standard technical drawing in industry (from their course placement work or otherwise). The drawings were checked for missing measurements, notes, tolerances and arrangement to improve readability. Issues were highlighted and the drawing recreated. The next iteration of drawing (when possible) was checked by a different member of the aforementioned group. This process was repeated until the drawing was suitable. The final stage of the TPV process was that the team principle had to check the drawing a final time before signing the drawing off for part manufacture.

This initial TPV process was later adjusted to add another step. Each of the drawing checker group was required to check that the designed part or assembly was correctly packaged in respect to the full car CAD assembly to avoid any packaging issues.
2.2.2  TPV Checklist

A core list of checking requirements was constructed from BS 8888:2013 [7] [8]. The main checking criteria were split into three sections below:

2.2.2.1 Part/ Component drawing Checklist

- Ensure the drawing contains all relevant dimensions, and that they are correct
- Ensure the scale of the drawing is correct and all dimensions are correct related to the scale
- Overall dimensions should be clear
- All extension lines are correctly related to the drawing points
- All arrowheads are pointing to the correct witness points
- Linear and geometric tolerances and limits are checked for clearance, any non-global tolerances are specified with callouts
- Ensure strings of dimensions cannot accumulate errors
- Allow for machining component restrictions (input maybe required from a machinist).
- Centrelines (for symmetry) and crosshairs (for circles/holes) are used when necessary.

2.2.2.2 Assembly Checklist

- Ensure all parts can be assembled with associated parts
- Male part’s fillets/chamfers and female part’s fillets/chamfers to be sized to ensure they do not interfere in assembly.
- Ensure parts can be dismantled if required
2.2.2.3 CAD Packaging Checklist

- Ensure completed part/assembly does not intrude on other components access area (the area defined by the body of a components and its required access region for alteration by mechanic).

2.3 Vehicle Testing (CE+TM)

2.3.1 Vehicle Testing – CE

Testing and development of a racing vehicle tends to fall into several categories [9]:

- New vehicle shakedown
- Development
- Reliability testing
- Qualifying and race set-up

The main goal of any test session is to “gather good data on different vehicle configurations” and “to be able to make use of this data ... at a future date” [9]. Good planning and clear direction are required, as simply driving round a circuit will achieve nothing worthwhile [10]. Preparation before a test is important, with everything that is needed at the track loaded into the van and all the vehicle system checked before departure. Time, and possibly the entire day, can be lost at a test when a simple issue was not checked before departure and the vehicle is unable to run. A test plan or program needs to be established before arriving at the track. This should cover the order of running through the day and the specific tasks that need to be completed [10].

Organisation of team personnel at the test is as important as vehicle preparation. Team members should have an assigned role and be aware of what they are required to be doing throughout the day. Communication with the driver is also essential. This should be through
a single person to avoid conflicting information being given and must be given in a clear, precise manner.

2.4 Tyres (CE)

2.4.1 Tyre Dynamics

2.4.1.1 SAE Tyre Axis System

In vehicle dynamics, it is common to use the SAE tyre axis system for analysis, Figure 2.4.1. From the figure the $x'$-axis is the “intersection of the wheel plane and the road plane with a positive direction forward” [11]. The $z'$-axis is “perpendicular to the road plane with a positive direction downward” [11]. The $y'$-axis is “in the road plane, its direction being chosen to make the axis system orthogonal and right-hand” [11]. Lateral force is defined as acting in the $y$-direction, acting from the point on the contact patch where the highest lateral force is generated. Longitudinal force is defined as acting in the $x$-direction, acting from the point on the contact patch where the highest longitudinal force is generated.

![Figure 2.4.1: SAE tyre axis system, defining parameters such as slip angle, slip ratio and lateral & longitudinal force [11]](image_url)
2.4.1.2 Slip Angle, SA

The slip angle is defined by the SAE as the angle between the velocity vector of the vehicle and the tyre heading [12] and is positive for a left hand turn [11]. The cause of the angle is a property of the rubber material used in the construction of the tread that is in contact with the road surface.

2.4.1.3 Slip Ratio, SR

The slip ratio is defined as the difference between the angular velocity of the driven (or braked) wheel and the angular velocity of the free-rolling wheel, which is then divided by the angular velocity of the free-rolling wheel [11]. This is often expressed as a fraction or percentage wheel slip.

2.4.1.4 Instantaneous Cornering Stiffness

The SAE defines the cornering stiffness as “the negative rate of change of lateral force with respect to change in slip angle, usually evaluated at zero slip angle.” [11].

2.4.1.5 Instantaneous Camber Stiffness

Instantaneous camber stiffness describes how much of a change in lateral force will result when the camber angle of the tyre is changed [13] [14]. The peak of the curve and how camber response decreases with slip angle will dictate the camber curves for the vehicle and when choosing caster and king pin inclination angles.

2.4.1.6 Self-aligning torque

The self-aligning torque tends to return the tyre from its current heading to the direction in which the tread is actually rolling [9] by generating a torque around the origin of tyre print axis. Asymmetry in the tyre print, where the elastic distortion increases fore to aft as the
tyre is steered, results in an unequal distribution of lateral force along the length of the print. The total torque is made up of the pneumatic trail, positive castor and the scrub radius.

2.4.1.7 Pneumatic trail

As mentioned previously, the pneumatic trail makes up a part of the self-aligning torque, where the print of the tyre trails behind the steering pivot point. This parameter is an indicator of when the tyre is likely to ‘breakaway’, where the driver perceives a decrease in the amount of available cornering force. This occurs because pneumatic trail approaches zero as the tyre reaches its limit, resulting in a reduction in self-aligning torque.

2.4.2 Tyre Testing & Methodologies

2.4.2.1 Tire Test Consortium (TTC)

The Tire Test Consortium (TTC) was established in 2005 to meet the need for high quality tyre information required by FSAE teams [15]. The tests were conducted at the Calspan Tire Research Facility (TIRF) in Buffalo, NY. Which uses a high speed, high load, six-component flat-belt tyre testing machine that is capable of testing products at a number of normal loads, slip angles, inflation pressures and road surfaces [15].

Several rounds of testing have taken place since the consortium was established, and they have covered a wide range of tyres used in FSAE/FS competitions. For each tyre, two test matrices are carried out to provide data for pure slip cornering and drive/brake combined scenarios. The matrix is made up of slip angle or slip ratio sweeps for different normal loads, inclination angles and inflation pressures. The test matrix is covered in more depth in [15].
2.4.2.2 Tyre Models
Within the tyre model area, there are several types available. A model can be based purely on experimental data, with a curve fitted to the data through various regression techniques. This empirical model describes the Magic Formula tyre model developed by Hans B. Pacejka [16] and is described through tabulated data and mathematical formulae. This method has the advantage of being based on measured data which gives it a good degree of accuracy, but a good number of these tests are required.

2.5 Unsprung (EI)

2.5.1 Wheel Planes and Terms

2.5.1.1 Camber
Camber is the angle of the wheel centreline with respect to the vertical axis of the vehicle [17]. A positive camber angle is denoted by the wheel centre line meeting the vertical axis below the axle and a negative camber angle is denoted by the centreline and vertical axis meeting above the axle [18]. During corner induced vehicle roll, negative camber when the vehicle is static aids cornering by increasing the tyre contact patch on the outside tyre resulting in the possibility of higher cornering speeds [19].

2.5.1.2 Toe
Toe is the angle between the wheel longitudinal centre plane and the longitudinal axis of the vehicle [17]. Under cornering, the toe angle of the steered wheels will change, one wheel will increase in toe angle and the other will decrease. In the static vehicle set-up of the front axle of a rear-wheel drive vehicle, toe-out will help the car turn into a corner, and toe-in will help the car travel in a straight line. Toe-out is defined as the wheel centrelines pointing out and away from the front of the vehicle and toe-in is defined as the wheel centre lines pointing in and towards the front of the vehicle.
2.5.1.3 Steering Axis

The steering axis of a wheel is the connecting line between the upper and lower outboard wishbone points and is axis around which the wheel rotates to gain camber and toe during cornering [20]. The steering axis combines Castor and Kingpin angles to create an outboard inclined axis.

2.5.1.3.1 Castor and Mechanical Trail

Castor is the angle between the steering axis and the vertical axis in the side view. The distance between the intersection point of this axis on the ground plane and the tyre contact patch is the mechanical trail or castor offset of the wheel. Castor is positive when the upper wishbone point is behind the lower wishbone, and mechanical trail is positive when the intersection point is in front of the tyre contact patch [20].

2.5.1.3.2 King Pin and Scrub Radius

King Pin Inclination is the angle between the upper and lower wishbone points in the front view and the distance between its intersection point with the ground plane and the tyre contact patch is the scrub radius. King Pin inclination is positive when the upper wishbone point is inclined towards the vehicle centre and scrub radius is positive when it intersects the ground outside the tyre contact patch outside. [20]

2.5.2 Steering Geometry

In 1816, a carriage builder from Munich, Mr Georg Lankensperger, discovered Axle-steering (Achsschenkellenkung) [21]. Lankenspergers agent, Rudolph Ackerman, filed for the patent in 1818 and this steering principle is generally referenced as the Ackerman principle. This steering geometry applied to drawn carriages and the etching on the plaque explains the principle behind the steering geometry principle, shown in Figure 2.5.1 [18].
The steering geometry of vehicles was almost non-existent until it was shown that vehicles needed to be able to turn the outside and inside wheel of a turn to different angles to gain the most effective cornering. The Lankensperger steering principle applies to low speed steering manoeuvres and discounts the slip angles of the vehicles tyres.

2.5.3 Oversteer and Understeer

Oversteer is defined as the front of the vehicle having more lateral acceleration than the rear, and creating a Yaw Moment about the centre of gravity. Oversteer has also been defined as the vehicle being at a greater angle than the steering wheel. Figure 2.5.2 shows an image showing a simple explanation of oversteer and understeer.
Understeer is the opposite of oversteer and occurs when the front axle has less lateral acceleration than the rear axle. Understeer is also defined as the steering wheel being at a greater angle than the vehicle and the vehicle wanting to continue in a straight line.

2.5.4 Vehicle Roll Centres

Kinematic Roll centres are widely talked about and are directly related to vehicle behaviour in Roll. The Kinematic Roll Centre of an Axle is the Intersection Point of the left and right lines between the respective FVIC and tyre contact patch, as shown in Figure 2.5.3 [22]. During roll, the vehicles CoG rotates about the Roll Centre.
There is another definition of Roll Centre however, the Force-based Roll Centre [22]. This is defined not by the suspension geometry, but by the tyres and is the intersection point between the left and right tyre force vectors in the front view. Figure 2.5.4 [22] shows a schematic with both roll centres defined. To find the Force Roll Centre, the tyres must be investigated and it is their interaction with the ground and with the kinematics of the vehicle that define the Force Roll Centre.
2.5.4.1 Roll Centre Inclination and Roll Axis

The Kinematic Roll Centre inclination is another important factor in vehicle dynamics during cornering. The inclination of roll centres is the angle of the front roll centre taken from the rear roll centre about the horizontal plane. The centre of gravity acts a distance from the Roll axis creates a moment about the roll axis during lateral acceleration and transfers normal loads around the vehicle.

For a single seater race car, the front kinematic roll centre is typically lower than the rear all the time, allowing for the vehicle to transfer weight to the front outside wheel during roll. If the roll axis is negatively inclined, then the vehicle would transfer weight to the rear outside wheel. If a vehicle roll axis varies between positive and negative around a corner, the vehicle will never exhibit neutral steer, but will however exhibit oversteer with positive inclination and understeer with negative.

2.5.5 Roll Stiffness and Roll Gradient

Roll Stiffness is a term, defining a vehicles resistance to roll measured in Nm/degree. The roll stiffness of the vehicle is also related to the vehicles roll gradient (phi), the amount of body roll per lateral acceleration in g. Both of these terms are calculated from the tyre and spring stiffness on each corner for the vehicle, the vehicle track, and the distance between the Centre of gravity height and roll centre inclination axis [18]. The roll stiffness, and roll gradient are both a characteristic of the designed vehicle, and also a design parameter to improve a vehicles handling. By increasing the roll stiffness of a vehicle, the roll gradient decreases.
2.6 Chassis (GG)

2.6.1 Chassis Definition

Senior FSAE design judge Pat Clarke describes the chassis as “a bracket that holds the whole car together” [23]; every major component within the vehicle must attach by some means to the chassis structure.

The chassis however also provides what is often termed the “survival-cell”; the framework around the driver which is intended to protect him/her in the event of a crash.

2.6.2 Chassis Technology

There are many forms of chassis; however the need for high specific-stiffness (discussed in section 2.6.7) means modern race-car construction is limited to 2 main groups; space-frames and monocoque.

Steel space-frame constructions look to take advantage of the stiffness of triangulated truss structures in order to achieve their stiffness.

Monocoque constructions use a “stressed-skin” approach; turning the car into a structure which represents a torsion tube, a tube being the most effective structure for carrying a torsional load.

In modern motorsport, “monocoque construction” typically is thought of as carbon-composite tub constructions; however the monocoque was originally pioneered by Colin Chapman’s Team Lotus using a stressed aluminium sheet. Advances in adhesive technology made honeycomb cored sandwich panels possible in the aerospace industry at the end of WW2; with the technology finding its way into F1 cars of the 70’s and 80’s.
In 1988 John Barnard’s MP4-1 became the first CFRP carbon tub; again motorsport turned to the aerospace industry for the technology with the tub being manufactured by Hercules Aerospace in the USA.

The major problem with moulded composite structures like the MP4-1 is tooling costs. Typically the tooling costs for a component are between 5 and 10 times the cost of the part; something which is difficult to justify for a one-off component.

The cut-and-fold method looks to avoid the tooling costs associated with honeycomb construction. By taking a pre-fabricated flat panel, locally removing parts of the skins and folding the panel into shape the costs of tooling can be avoided, whilst still achieving results which are closer to the moulded tub than a spaceframe on the stiffness spectrum. The construction method is in fact closer in concept to the pre-CFRP era in that it is only possible to use shapes which you can bend or fold.

Recently the cut and fold technique has been used successfully by Tony Pashey in hill-climb vehicles, along with a number of FS teams including Oxford Brookes Racing, Cardiff Racing and Brunel Racing.

2.6.3 Design Criteria

**Basic Laws of Physics for Race-cars**

As with anything on a race-car, performance is the priority; designers are always looking to remove mass. A short-circuit style race-car gets its performance from its acceleration capability. What we term as “acceleration”, “braking” and “cornering” are all accelerations simply with different vectors. For minimal lap-time a race-car should always be accelerating in some direction along the ground, if it isn’t then the driver could have accelerated for
longer, reaching a higher speed before braking; a driver should always be either using the accelerator, brake or steering to accelerate the car at the limits of the acceleration envelope.

Newton’s 2\textsuperscript{nd} law states the force required to accelerate a body is proportional to its mass, there are therefore only 2 ways to increase the acceleration capability of a body; increase the force applied or reduce the mass. Practically the chassis doesn’t generate a force, (although the importance of its load transfer capabilities will be discussed later) therefore to maximise acceleration, and thus the performance of the overall vehicle, mass should be minimised. Given that the chassis represents the 3\textsuperscript{rd} heaviest single assembly after the engine and driver, both of which have minimal opportunities for mass reduction, mass minimisation is hugely important within chassis design.

\textbf{Regulations}

Brunel Racing builds cars to enter into the FSAE series of competitions, therefore the car must be designed to comply with all regulations within the main rulebook [1], as well as the “supplementary regulations” for the 2 events BR typically attends [3], [2].

Around 30% of the main rule book relates to the chassis; mostly concerning minimum requirements for safety purposes. As such much of the chassis design must follow these regulations - looking to find ways to develop the design within the legalities of the regulations. The main areas are:

- Minimum tube thickness requirements
- Required crash structure energy absorption and deceleration
- Minimum driver templates and helmet clearance to roll bars
- Minimum strength requirements for safety critical attachments

2.6.4 Torsional Stiffness

Whilst reducing mass is always desired and various sections of the structure will be mandated by the regulations, there is a further performance metric which is vital to the behaviour of the car – Torsional Stiffness.

Riley & George [24] showed how the chassis of a car can be considered to be a torsion spring in series with a further set of springs at either end of the car representing the roll stiffness at each axel of the suspension; effectively creating a series of 3 torsion springs, where the torsion of the front or rear spring is given by equation 2.6.1.

\[
K_{Roll} = \frac{K_{Spring}}{W_t}
\]

Figure 2.6.1. Considering the chassis as a torsion spring [24]
where $K_{Roll}$ is the suspension roll stiffness across an axle, $K_{Spring}$ is the spring rate at the wheel of the corner spring, and $w_t$ is the track width. This equation assumes that only corner springs are present with no anti-roll bars.

There are also compliance effects within the suspension to take into account of in addition to the torsional stiffness of the chassis. Therefore Equation 2.6.2 demonstrates the overall stiffness of the frame.

$$\frac{1}{K_{Total}} = \frac{1}{K_{Frame}} + \frac{1}{K_{Comp.}} + \frac{1}{K_{Roll Fr.}} + \frac{1}{K_{Roll Rr.}}$$

*Equation 2.6.2: Overall stiffness*

where $K_{Total}$ is the overall stiffness of the chassis platform, $K_{Frame}$ is the torsional stiffness of the frame, $K_{Comp.}$ is torsional spring effect of compliance within the suspension, and $K_{Roll Fr.}$ and $K_{Roll Rr.}$ are the front and rear suspension roll stiffness’s respectively (inclusive of any ARB systems) [25].

Practically, if the vehicle structure is not sufficiently stiff to transmit suspension loads between the axles it will tend to cause the vehicle to understeer [26]. For tuning, the race-engineer wants the suspension to be wholey responsible for the car’s handling with the chassis not affecting the dynamic behaviour of the vehicle; this is however impossible.
To achieve this the chassis would need to be infinitely stiff, something which is not possible. It is however possible to make the chassis stiff enough so that the effect of the chassis’ compliance is barely noticeable during tuning.

Deakin et. al. [26] have shown that a vehicle with suspension roll stiffness typical of a FSAE car requires a chassis torsional stiffness of 1-2000Nm/degrees to allow for predictable front-rear load transfer distribution when measured at the wheels.

However this value is inclusive of compliance effects, therefore the designed chassis stiffness needs to be greater to overcome these compliance effects. From Figure 2.6.2 the Chassis Torsional Stiffness Requirement (CTSR) for a given suspension can be determined, however the required stiffness of the frame must be suitably higher to account for the compliance/installation stiffness of the suspension and other connections.

![Figure 2.6.2. Plot of Front Load transfer and Front Roll Stiffness](image)
2.6.5 FE Modelling Techniques

4-node quadrilateral shell elements have been successfully used in the modelling of honeycomb chassis construction by a number of FSAE teams [27] [29] [30], this simple element formulation is suitable for large scale behaviour as it allows the computational power to instead be utilised to create more refined meshes around complex geometries.

It has been found that 2-node 1D beam elements have been most suitable for modelling a spaceframe [31] [24], and whilst although Timoshenko formulation elements can account for bending [32] they still assume axial loading; meaning that behaviour around joints
cannot be modelled. As stiffness, not yield is typically of concern this is not likely to be a major problem.

2.6.6 Physical Testing Requirements

Monocoque constructions within FSAE are subject to the requirements of the SES. The SES requires a minimum of 2 physical tests, a 3 point bending test and a perimeter shear test.

2.6.6.1 3 Point Bending Test

Rule T3.31.1 [1] states:

"Teams must build a representative test panel with the same design, laminate, and fabrication method as used in the monocoque side impact zone (defined in T3.34) as a flat panel and perform a 3 point bending test on this panel.

They must prove by physical testing that a panel measuring 275mm (10.8”) x 500 mm (19.7”) has at least the same properties as two baseline steel side impact tubes (See T3.4.1 “Baseline Steel Materials”) for buckling modulus, yield strength, ultimate strength and absorbed energy."

The load applicator must have a radius of 50mm [1], the schematic shown in is provided to clarify the test procedure.

Figure 2.6.4. Schematic of mandatory 3 point bending test. [1]
2.6.6.2 Perimeter Shear Test

The second mandatory test is a shear test of the laminate. Rule T3.31.5 states:

*Perimeter shear tests must be completed by measuring the force required to push or pull a 25mm (1”)* diameter flat punch through a flat laminate sample.

*The sample, measuring at least 100mm x 100mm (3.9” x 3.9”), must have core and skin thicknesses identical to those used in the actual monocoque and be manufactured using the same materials and processes.*

*The fixture must support the entire sample, except for a 32mm (1.25”) hole aligned co-axially with the punch. The sample must not be clamped to the fixture. The force-displacement data and photos of the test setup must be included in the SES.*

*The first peak in the load-deflection curve must be used to determine the skin shear strength; this may be less than the minimum force required by T3.33.3/T3.34.3.*

*The maximum force recorded must meet the requirements of T3.33.3/T3.34.3.*

*N: The edge of the punch and hole in the fixture may include an optional fillet up-to a maximum radius of 1mm (0.040”).*

A graphical representation is also included in the guidance notes within the SES form is shown in Figure 2.6.5.
2.6.7 Overall Approach to Chassis Design

Given the information contained herein, it can be said that in order to effectively specify a chassis correctly the suspension roll stiffness should be found, compliance effects estimated and a CTSR value specified. Subsequently the design should look to minimise the mass of the vehicle within the regulations whilst maintaining the CTSR; maximising the stiffness to weight ratio (specific stiffness).

2.7 Aerodynamics (GM)

2.7.1 Lift and drag

There are two main categories of aerodynamic forces on a race vehicle: pressure acting perpendicularly to the vehicle’s surface area contributing to create lift and drag, and shear force which acts parallel to the vehicle’s surface area and create only drag. [33] Negative Lift (or downforce) is a force that acts vertically at the vehicle’s centre of gravity (CoG) literally pushing the vehicle’s body towards the ground. Downforce can be exploited to increase the
performance of a racing car: by transmitting this force to the tyres, additional normal load will act on them resulting in greater lateral accelerations achievable during cornering.

Drag is a force that acts horizontally on the vehicle’s CoG reducing the vehicle’s available power. There are various forms of drag and they all contribute to a reduction of the vehicle’s performance. Some examples are: skin friction, cooling drag, internal flow (e.g. driver’s compartment flow), form drag, and lift induced drag. [33]

Lift and Drag are usually expressed in their non-dimensional form: The coefficients of Lift ($C_L$) and drag ($C_D$). These coefficients are independent of vehicle speed and size and are given by Equation 2.7.2 and Equation 2.7.1:

\[
C_D = \frac{D}{\frac{1}{2} \rho V^2 \text{A}}
\]

*Equation 2.7.2 Equation for calculating Drag Coefficient*

\[
C_L = \frac{L}{\frac{1}{2} \rho V^2 \text{A}}
\]

*Equation 2.7.1 Equation for calculating Lift Coefficient*

It is important to underline that Lift and Drag are related. An effort of increasing Lift will often result in an increased value of drag.

2.7.2 Aerofoils

One way of increasing the downforce generated by a racing vehicle is the adoption of a front and rear wing. An aerofoil is defined as the two dimensional cross section shape of a 3-Dimensional wing. The mechanism that allows an aerofoil to create downforce is very simple (Figure 2.7.1):
As the air travels over the upper surface of the wing (or suction side), the air is accelerated due to the longer distance it has to travel compared to the lower surface (or pressure side). According to Bernoulli’s principle a flow with higher velocity is at a lower pressure compared to a flow with lower velocity (assuming all the rest of the flow parameters remain unchanged). This pressure difference between the aerofoil’s surfaces generates a vertical upward force: the already discussed “Lift”. In automotive applications inverted aerofoils (where the suction side is the lower surface of the wing) are utilised to generate the inverse effect: negative Lift better known as “downforce”.

*Figure 2.7.1 View of the Air Flow over an Aerofoil*
Error! Reference source not found. shows the different geometries characterising a wing. The leading edge and the trailing edge are respectively the furthermost and rearmost part of an aerofoil. The wing chord is the imaginary line connecting the leading edge and the trailing edge. The maximum thickness is usually expressed as a percentage of the chord’s dimension. The angle of attack (\( \alpha \)) of a wing is the angle between the chord line and the incoming airflow. The angle of attack is the easiest parameter to change to modify a wing’s performance because it does not require wing re-design and re-manufacture. [34] For any symmetrical aerofoil, a linear relationship exists between angle of attack (\( \alpha \)) and generated downforce

\[
C_l = 2 \pi \alpha
\]

(for attached flow):  

Equation 2.7.3. Lift and angle of attack relationship (Symmetrical Aerofoil)

This means that by increasing the angle of attack the coefficient of lift linearly increases. [35] If a wing is asymmetric, the suction and pressure side’s profiles differ and have diverse curvatures: the wing is said to have camber. The amount of camber of an aerofoil is strictly
related with the downforce it produces can be seen as an additional angle of attack [35]:

\[ C_l = 2\pi(\alpha + \alpha_{L_0}) \]

The previous equations are assuming that the flow remains constantly attached to an aerofoil’s surfaces. In reality there is always some flow separation or “detachment” present on a body immersed in a fluid flow. For a wing, this phenomenon increases with increasing level of camber and more importantly angle of attack.

Error! Reference source not found. represents a 2D analysis of a NACA 63-615 aerofoil at different angles of attack simulated at a speed of 50m/s. It can be seen that by increasing the angle of attack from 0 to 12 degrees the downforce increases linearly and reaches its maximum value. However, if the aerofoil is “pushed harder” (greater angle of attack) the downforce level starts dropping: the flow does not have enough energy to fight the adverse
pressure gradient on the suction side of the wing and detaches from the aerofoil resulting in a loss of negative lift generated. [34]

### 2.7.3 Flaps

Multiple element wings are very common in automotive aerodynamics. Whilst for the aerospace industry there are very little limitations on the dimension of the wings, higher forces can be achieved by increasing span and chord. In motorsport, however, aerodynamics rules are very strict and the design spaces are limited by other vehicle’s components. By adding a flap to an aerofoil in the correct position, a series of consequences happen: firstly downforce is increased due to the larger projected frontal area of the wing; secondly because of the addition of the flap means an increase of the wing’s camber.

However it is not only by adding camber and planar area that a flap helps increase the performance of a wing. The interaction of the flap and the mainplane (the main and largest aerofoil of a wing) is of primary importance: If the gap between the flap and the mainplane forms a convergent slot, air passing through this gap is accelerated to the suction side of the wing with increased energy and lower pressure. This helps reducing the adverse pressure gradient on the suction side of the wing reducing the likelihood of flow separation on the mainplane. The slot gap also helps separate the wake of the mainplane from the flap so that a new and “fresh” boundary layer forms on the flap and flow separation is less likely to occur on the flap.

There is some design guidance about flap’s dimensions and position, but the best way of testing the influence of a flap on a wing is to perform CFD simulations. Theoretically the flap should have a chord length of about 25-30% of the overall wing chord. The vertical gap
between the flap and the main plane should be around 1% to 2% of the wing chord and the horizontal overlap should be between 1% to 4%. [34]

2.7.4 Ground effect (undertray and front wing)

A body, in this case the wing of a race car, moving through air at a close proximity with the ground, experiences additional effects due to the interaction between the body and the ground itself.

In race car aerodynamics, this phenomenon can be exploited to create additional downforce and improve a vehicle’s performance even further.

One of these effects is the so called “Ground Effect”: the working principles are based on the Venturi effect (Error! Reference source not found.) which states that the flow velocity is directly proportional to the cross-sectional area of the pipe in which it is flowing (according to the principle of continuity) and the Bernoulli’s principle.

![Figure 2.7.4 Representation of Venturi effect](image)

If the volume between a race vehicle and the ground is considered as a pipe, an efficient way of adding additional downforce is to shape the floor (or undertray) as a venturi to increase
the flow velocity, and hence reduce the pressure, beneath it. Since the pressure over the car remains unchanged the net pressure will result in a downward force.

Ground effect becomes an essential consideration whilst designing the front wing of a race vehicle: now the air flow in the suction side it is not only accelerated by the shape of the aerofoil but also by its interaction with the ground. Front wings in ground effect allow much higher values of $C_l$ to be achieved when compared to wings in free stream.

2.7.5 Endplates

The endplates are aerodynamics components that can help increase a wing’s efficiency and downforce. Endplates achieve this by maintaining the pressure difference between the surfaces of a wing, “fighting” the tendency of high pressurised air from the pressure side to spill into the low pressure suction side.

Rather than shape being the dimension of an end plate to be of particular importance, it is the portion extending below the aerofoil (in automotive application) that is of interest. However running end plates that are too big can cause a noticeable increase in drag values for a little increase in downforce. [34]
Due to the limited distance between the ground and a front wing endplate, it is common to fit a “footplate” to a front wing endplate to increase the whole aerofoil’s effectiveness.

By working in conjunction with the endplate, the main function of a footplate is to stop high pressurised air from flowing into the suction side of a wing. Figure 2.7.6 shows the results of a 3D simulation comparing the flow over a wing endplate with and without a footplate:
It can be seen that the footplate physically stops or at least postpones the air from flowing around the endplate towards the low pressure underside of the wing. There is also a secondary effect created by footplates: when the vehicle is moving through air, especially during cornering, vortices are created at the tip of a footplate. (Figure 2.7.7)

Vortices feature a low pressure core that can be exploited to:

1. Create an area of low pressure under the wing
2. Provide extra sealing between the underbody and the external airflow.

In high level motorsport footplates have a “tunnelled” shape to facilitate this vortices creation. (Figure 2.7.7)
2.7.6  FSAE 2015 rules

This year’s competitions feature the biggest rule change concerning aerodynamics devices since the beginning of the Formula Student. During the history of the competition a great freedom was left to aerodynamicists to design and develop their components. In 2015, the design space available for aerodynamic devices was limited drastically with the addition and the modification of many rulebooks’ articles:

The major rule changes can be found below:

**T9.2 Location – Front Mounted Devices**

T9.2.1 In *plan* view, no part of any aerodynamic device, wing, under tray or splitter can be:

a. Further forward than 700 mm (*compared to 762mm in 2014*) (27.6 inches) forward of the fronts of the front tires

b. Wider than the outside of the front tires measured at the height of the hubs.

*T9.2.2 When viewed from the front of the vehicle, the part of the front wheels/tires that are more than 250 mm (9.8 inches) above ground level must be unobstructed by any part of the aerodynamic device, with the exception of any vertical surfaces (end plates) less than 25 mm in thickness. (non-existent in 2014)*

**NOTE**: 9.2.1 and 9.2.2 apply with the wheels in the straight ahead position

**T9.3 Location Rear Mounted Devices**: T9.3.1 In *plan* view, no part of any aerodynamic device, wing, under tray or splitter can be:

a. Further rearward than 250 mm (9.8 inches) rearward of the rear of the rear tires

b. Wider than the inside of the rear tires, measured at the height of the hub centerline.
T9.3.2 In side elevation, no part of the rear wing or aerodynamic device (including end-plates) may be higher than 1.2 meters above the ground when measured without a driver in the vehicle

The design space for 2015 can be visualised below:

2.8 Driver Controls & Elec

2.8.1 Pedal Box

The pedal box is vital in any human controlled vehicle. The pedal box is used for the driver to control the throttle and hence engine load; as well as braking and often clutch application. In many forms of motorsport the clutch is controlled in other ways. It is the
same in Formula Student. This means that the pedal box consists of a single accelerator (throttle) pedal and a single brake pedal.

A well designed pedal box will allow good pedal response, which improves the driver’s reactions and performance. On the other hand, poor pedal response is likely to affect driver’s confidence.

Driver anthropometric and ergonomic data must be considered when designing a suitable pedal box. Driver input forces must be considered in order to ensure the pedal box does not fail and allowing the driver to keep full vehicle control. It is suggested that the maximum driver force applied to a brake pedal is 1330N [36]. However, as per FSAE regulation the brake pedal must withstand 2000N of force [1].

The geometry of the pedal box is also important. Both the geometry of its mounting positions, and the pedal lever ratios. The pedal box must enable a 95th percentile male driver to control the vehicle. This requires the design to be adjusted in order to allow different size drivers to control the vehicle.

The ratio of the throttle cable and master cylinders relative to the ball of the foot from the pedal pivot is vital in determining the “feel” of the pedal [38]. “The pedal position must be correctly matched to the geometry of the driver’s foot and ankle, must remain at a constant height and should be really firm and have minimum travel” [39]. Driver “feel” is vital for reducing lap times and improving performance.
2.8.2 Brake System

The brake system must be designed to stop the vehicle as quickly and safely as possible whilst ensuring the driver can keep control of the vehicle. The brake system has a number of design considerations:

- Master cylinder and bias bar selection, with pedal design
- Brake line specification and line pressure calculation
- Calliper and pad specification
- Brake disc design.

Brake design is a systematic bottom-up design method where considerations start at the pedal and work through each component to the brake disc itself. There are a number of factors that ultimately influence brake pedal design discussed in section 2.8.1. These ultimately effect the design and specification of other brake components down the line.

Traditionally formula student teams use -3 (dash-three) brake lines often without specific design reasoning. However, after discussion with a supplier research into the use of -2 (dash-two) lines should be conducted to see if they were suitable. These lines would reduce the mass of the brake system as they have a smaller diameter and contain less fluid. However, -2 brake lines cannot withstand the same brake pressure as larger diameter fluid lines [38].

Brake discs require thermal and structural analysis and testing in order to ensure material and design suitability. BR-XV had issues with brake disc failure leading to driver safety issues. This led to an internal investigation and research that showed these discs were not in fact the required specified grey cast iron due to supplier fault. Research by Erogenous
showed that the faulty discs buckled at a load of around 40% of verified grey cast iron discs. See Portfolio of Evidence.

2.8.3 Steering Wheel

The steering column is arguably the most important driver control. As well as its obvious function of controlling the direction of the vehicle, a trend has developed in many forms of racing whereby the steering wheel is a hub for electrical controls and communications [41]. As well as there being limited space in a single seat racing cockpit, it allows quick and easy access to the driver to adjust certain controls without removing their hands from the wheel. Common electrical controls to find on the wheel are launch and traction control, engine mapping and gear shifters (behind the wheel).

Many steering wheels are designed to display information for the driver to see such as speed, RPM, shift lights and more. This may provide the driver with the right information to improve performance and overall competition score.

A steering wheel also has to cope with a number of forces applied to it. It is suggested that the steering wheel should be designed to withstand a minimum of 100N steering torque and 660N of lateral force [36].

Smaller steering wheels make it difficult for the driver to make fine adjustments as greater levering forces are required to steer. The wheel must be designed so that it does not extend beyond the top surface of the front roll hoop in any position [37].

The steering wheel must be designed to be comfortable for a range of drivers. Ergonomic data provides a vital starting point. Customised hand grips can be created from materials
such as polymorph and specified for each driver (although this would cause greater expense and loss of time at driver changes).

The steering wheel can be used to display various information for the driver. Due to the expected design of the cockpit having little space for switch pod or dash board, this seems like the best option. Concerns have been raised in previous years regarding difficulty reading a multifunctional display at an angle; for example while cornering [42]. However, research with drivers suggests that drivers do not look at the wheel while cornering as they are focussed on hitting the driving line and corner apexes.

An additional argument for a wheel mounted display is it has the potential to improve the organisation of the wiring loom. It enables the use of an electrical quick release boss, meaning all wheel electronics can be housed inside the steering column without the need for a supply cable on previous vehicles (Figure 2.8.1) [43].
Research on the use of multifunctional display is discussed further in section 2.8.5.

### 2.8.4 Switch Box and Dash

The switch box and dash is a key area of the car. It is used as a driver interface to adjust various vehicle settings and display information such as water and fuel pump status. This is the basis of the design which will applied to BR-16.

The switch box/dash is required to have one particular item: a cockpit mounted master switch. According to FSAE regulations IC4.3.1, this switch must:

- Must be located to provide easy actuation by the driver in an emergency or panic situation
- Must be located within easy reach of the belted-in driver, alongside the steering wheel, and unobstructed.
- Must be a push/pull switch with a minimum diameter of 24mm.
- From the ON, pushing the switch will disable the power to ignition and all fuel pumps.
- From the OFF, pushing the switch will disable the power to ignition and all fuel pumps.
- May act through a relay [37].

As discussed in section 2.8.3 the main display will be a multifunctional screen like display. LEDs are used in the switch box to show activation of relevant switches. In the past this has been wired incorrectly. BR-XV wiring shows that the LEDs are activated by the switch, not by the component feedback. Effectively the LEDs indicate the switch is on, not that the component is on. This must be considered when designing the wiring loom.
2.8.5 Electronics

The electronics of the vehicle are regulated mainly for safety reasons. Article 4 considers the two master switches and their positions in the vehicle [37]. It also specifies the systems these switches must “kill” when activated.

Other electrical regulations specify the power supply/battery chemical composition and protection. Two of the vital regulations in this area are:

IC4.4.4 Battery packs based on Lithium Chemistry: A) must be commercially manufactured items. B) Must have over voltage, under voltage, short circuit and over temperature cell protection. C) Must be separated from the driver by a firewall.

IC4.4.5 All batteries using chemistries other than lead acid must be presented at technical inspection with markings identifying it for comparison to a datasheet or other documentation proving the pack and supporting electronics meet all rules requirements [37].

There are a number of different power cells that can be specified for use. Some power cell chemistries are suited to motorsport more than others.

Lithium iron phosphate (LiFePO₄) cells have a longer life span than other types of lithium chemistry cells. They are considered far safer than many other forms of power cell as they are incombustible in the event of mishandling [37]. LiFePO₄ cells normally have a lower power density (around 330 Wh/l) meaning a more massive cell is normally required [44]. However, they are more stable when under/over charged or over heated. As of 19/11/14 a rules addendum clarified that LiFePO₄ would be accepted without over voltage, under voltage, short circuit and over temperature cell protection [45]. This negates the extra mass the LiFePO₄ cell and makes packaging and wiring simpler.

Lithium-ion (Li-ion) cells are probably the most common cell chemistry. Used in everyday electrical items, these cells can achieve power density figures of 750 Wh/l [46]. These will
also have to meet the requirements of regulation IC4.4.4 adding extra components to the vehicle.

2.9 Innovative Design Solutions (JS)

Innovation and innovative design is a fundamental part of motorsport, with new technologies and vehicle design propelling the sport forward every season [47]. Innovation is one of the major factors that separate the leading teams in racing such as Formula 1, from smaller teams. Large teams invest heavily in research and development looking for new innovative designs that can be implemented to give them the edge over the competitors [48]. The newest and most heavily developed subsystem to come to open wheel racing is aerodynamics with new designs in constant development for maximising aerodynamic efficiency.

2.9.1 Drag Reduction System and Active Aerodynamics

The aerodynamics of an open wheel racing car have a number of key objectives in aiding the performance of the car; production of downforce for improved cornering and braking, limited vehicle drag for straight line performance, control of fluid flow over the vehicle for optimum vehicle system operation [49].

![Figure 2.9.1: Operation of Single Element DRS](image-url)
The trade-off between producing maximum downforce and minimal drag is the greatest obstacle for the design of aerodynamics for a racing car. Downforce is required to increase the load applied to each tyre during cornering to maximise the tyre grip available on the track allowing for cornering at higher velocity, as well as reducing the braking distances of the car [50]. To obtain high downforce for these applications, aerodynamic elements, including front and rear wings with large surface areas are used. However, with increased downforce and the use of components with large surface areas, the car incurs an increase in drag. An undesirable force for racing cars, drag hinders the movement and velocity at which the car can pass through the air, increasing the resistance between the body and the fluid. Drag has the greatest effect on the car in a straight line, with the force increasing as the velocity of the car increases [51]. With advances in technology, the state-of-the-art, in this case Formula 1 has implemented certain aerodynamic design features that can, to an extent, overcome the trade-off between maximum downforce and minimal drag for ultimate vehicle performance.

Initially implemented in 2011, Drag Reduction System (DRS), have been a key aerodynamic feature for Formula 1 cars allowing the movement of a rear wing element to reduce the
drag created by the wing in a straight line yet maintain its effectiveness of creating downforce under braking and during cornering.

Controlled by the driver when within a specific given position on the track, DRS provides an immediately adjustability to the aerodynamic properties of the vehicle. As shown in Figure 2.9.1, DRS enables the design of the component to utilise the wing element for both applications of reduced drag and increased downforce where required rather than a trade-off between them. Further research into DRS operation, application in Formula 1 and system hardware is discussed in the Portfolio of Evidence.

Similarly to DRS, other controllable aerodynamic components and vehicle subsystems can be designed to actively compensate for different track conditions. Although banned in Formula 1, active aerodynamic systems are one of the most effective methods of controlling vehicle dynamics and performance during the operation of the car. Active aerodynamics, most commonly implemented in elite sports cars such as the McLaren P1 or Porsche 918, adjust the positioning of aerodynamic components similar to the motion of a DRS system to accommodate for the driver’s requirement for increased downforce or drag.
However, differing from DRS, active aerodynamics automatically adjusts the car's aerodynamic form without direct input from the driver using vehicle sensors. This can allow for the system to be programmed to adjust certain parameters of the aerodynamic components corresponding to specific measurements recorded by the vehicle sensors. It can further allow for the active aerodynamic system to operate in coherence with other subsystems such as an active suspension. The combination of the two systems, can dramatically improve the vehicle’s accelerating, braking and cornering dynamics with the control over the sprung mass of the vehicle.

A successful active aerodynamics system must react directly to the requirement for improved vehicle dynamics using data gained from the vehicle accelerometers and other sensors at a high enough frequency for currently improvement to the continuously changing dynamics. The resultant activation of moveable aerodynamic components must be then controlled with full understanding of the force and motion effects that will result. In stability...
and inadequate control measures of the system will further reduce the vehicle handling rather than improve, rendering the system a failure. Further research into Active Aerodynamics and the system hardware is discussed in the Portfolio of Evidence.

DRS and active aerodynamics are not common in Formula Student but not abundant. Mainly present in cars produce by teams for the American series, the systems can take a number of iterations to reach their full performance potential, and require heavy development with current vehicle aerodynamic design. A team with a strong active aerodynamics package is University of Texas at Arlington (UTA). Their 2013 car, shown in Figure 2.9.3, uses a fully developed 4th generation system, controlled by 4 on board computers, with independent control over left and right, front and rear wings. The system has successfully aided them to be currently ranked 4th Formula Student Team Globally.

![Figure 2.9.3: University of Texas at Arlington 2013 Formula Student Car](image)

The system design by UTA uses electro-mechanical servo controlled wing elements that can be activated automatically by the on board control unit that react to incoming vehicle dynamic data, or manually by the driver using buttons on the steering wheel.
2.9.2 Active Suspension

Active suspension systems within the automotive and motorsport industry have a variety of designs, specific to design requirements of the specific required operation. Within the consumer automotive market, active suspension features on high end cars, such as salons and sports cars. For cars designed with passenger comfort as a priority over driving performance, active suspension system are designed to reduce road disturbance to the sprung mass of the vehicle. This operation requires data gained from the deviation in road conditions allowing the systems to simultaneously reduce the road disturbance. The output of this operation requires a softer suspension setup for optimum performance, opposite to that of cars designed for vehicle performance as a priority over passenger comfort that require a stiff suspension setup. This operation, most commonly used in sports cars, is designed to improve the vehicle handling, reducing vehicle roll, pitch and yaw. To achieve this direct acquisition of sprung mass data is required.

The design of active suspension in motorsport uses the same operational requirements as the later operation discussed above. With sole output requirements for improved vehicle handling, motorsport applications require the system to store, dissipate and introduce energy to the suspension with minimal weight gain and power consumption. Successfully designed and implemented, a fully active suspension system can improve vehicle cornering, braking and acceleration, ultimately allowing for greater lateral and longitudinal force of the sprung mass and so greater velocity of the vehicle. Further research into Active Suspension and the system hardware is discussed in the Portfolio of Evidence.
2.9.3 Secondary Aerodynamic Components

Although in recent years aerodynamic components have become more popular in Formula Student with more teams developing wing and floor packages, including Brunel Racing, there is still a vast difference in aerodynamic packages between those used in Formula Student and higher open wheel race series, such as Formula 1. Many of these factors are as a result of differences in series rules and regulations, budget restraints and development time and resources. However, there are a number of secondary aerodynamic components, components that can be used in coherence with major aerodynamic components that have not yet been utilised in Formula Student as they have in other series. Components such as turning vanes, splitters, sub system covers, all play a role in the overall performance of an aerodynamic package and can ultimately improve its efficiency.

One major consideration that must be made when comparing Formula Student and high class racing series is the velocity of which the cars operate, and so the allowable aerodynamic forces that can be produced. In majority of cases this makes certain secondary aerodynamic components redundant, with minimal gain over development and manufacturing costs. However, a number of components can still be seen to render a positive performance over cost yield. One component becoming increasing popular in this category within Formula Student are engine covers.

One of the most broadly designed aerodynamic components of a Formula 1 car, the engine cover, varies in design between all teams and is often changed a number of times across the Formula 1 development calendar. The primary purpose of the engine cover is to aid the aerodynamics of the car, moving airstreams over the complex structure of the engine and directing it out the rear of the car. Although the largest and heaviest aerodynamic
component used on a Formula 1 car, the engine cover cannot be disregarded as the overall performance improvement provided by the engine cover counter-acts that of the weight gain. Due to its size the engine cover is often made up of a number of different panels. To reduce the weight of these panels, a very lightweight construction of the carbon composites used for the bodywork of the car is used compared to that of other aerodynamic components on the car. This reduces the durability and strength of the panels and so as a result the panels are frequently replaced [52].

Designing the engine cover in such a way with separate panels rather than a single piece also creates a new issue, fitting the panels together. The panels must be joined in a way that will reduce the disruption produced by gaps between panels. These gaps, known as shut lines can cause turbulence and wake in the airstream as it passes over them. To ensure the panels are accurate and fit closely with little movement between them, spring-loaded fasteners are used, which prevent the panels from coming loose or moving off their mounting points under high vibration. The weight of each fastener must also be considered and so the panels must be designed to use the minimal number of fasteners possible but maintain a secure fitting.

With a high performing light weight design, an engine cover can dramatically reduce the drag produce by the chassis and driver’s helmet, reducing the overall drag produced by the car and increasing the performance of the rear wing. Combined design with other systems of the car, such as cooling and powertrain can enable the design of an engine cover to benefit the performance output of these systems. [53]
Engine cover designs that have previously been used in Formula Student vary greatly with teams implementing full, half or even minimal sections of an engine cover to increase the car’s performance.

Figure 2.9.4 shows an engine cover design used by CULS Prague that sits directly behind the driver’s helmet but does not cover the entire engine. This design reduces the effect of drag created within the cockpit but does not limit the access to the engine. This can prove to be highly beneficial at competition when constant work on the powertrain and drivetrain is often required. The size of the engine cover design also reduces the impact it has on the weight of the car, as well as requiring a small allocation of budget for the minimal amount of material used. [54]
2.10 Vehicle Simulation (CS)

Vehicle simulation is a useful method for testing a vehicle without the need for physical testing. Vehicle simulation can be cheaper to develop along with offering the capability to provide data which may otherwise be difficult to obtain from a vehicle [55].

It is important that a vehicle model is accurate and validated using physical test data in order to ensure that results are reliable. The vehicle model can then be altered according to the vehicle setup and changes can be made in order to optimise the lap time for the vehicle. The program which has been used for this project is IPG CarMaker.

2.10.1 IPG CarMaker

IPG CarMaker is a program which can be used on an individual PC or as part of a HIL test on single or multiple ECUs in industry [56]. CarMaker is used by a wide range of high profile customers including Ferrari and Continental [57]. It uses an open integration system which means that other program files such as those produced in Simulink and Matlab can be imported into the program [56]. It gives the user the opportunity to define the vehicle, road and driver behaviour, allowing a highly accurate simulation to be produced [58]. It uses a GUI to parameterise the vehicle as shown in Figure 2.10.1 below [59].
The Formula CarMaker version as will be used in this project comes with a demo car model called “FS_RaceCar_4.0” which is representative of a common combustion Formula Student vehicle from 2011 and will be used as the basis for this project [59].

For ease of understanding the parameterisation of the vehicle model (described later on in the report), CarMaker uses various axes to define both the vehicle and the road as shown by Figure 2.10.2 below.
• Fr0 – the inertial axis system fixed to the earth [60]. x and y run along the horizontal driving plane and z runs upwards from the ground [60].

• Fr1 – the axis used for moving objects [60]. x represents the driving direction, y is pointed to the left of the driving direction and z runs upwards from the ground [60].

• Fr2 – the carrier axis system used for wheels and carriers [60]. Mnt represents the mounting point of the wheel and is the centre of reference for Fr2 [60].

2.10.2 IPG Kinematics

IPG Kinematics is a program linked to CarMaker which is used to determine the kinematics of the vehicle, the steering system and the suspension compliance [59]. It can be used to generate graphs of this data and this information can then be exported into the CarMaker program [59].
2.10.3 IPG Instruments

This is a feature of CarMaker which graphically displays the steering angle and position values for the brake and accelerator pedals, along with other vehicle measurements [59]. It can also be used to record lap times in CarMaker [59]. Figure 2.10.3 below shows the interface of IPG Instruments.

![IPG Instruments panel](image)

*Figure 2.10.3 - IPG Instruments panel [59]*

2.10.4 IPG Control

IPG Control is the analysis tool used by CarMaker [59]. It can display a range of variables for graphical comparison between vehicle models and the data can be exported into Excel for comparison and validation against hard test data [59]. The graph in Figure 2.10.4 shows two graphs using IPG control to display steering angle in radians and vehicle velocity in m/s.
2.10.5 Model Check

CarMaker has an inbuilt system called “Model Check” which can be used to observe car model specific graphs [59]. It can be used as a validation tool to check certain diagrams against calculated/actual diagrams for the vehicle and it can also be used to check that the model will not produce any errors before running a virtual test [59]. When Model Check is initiated the window shown in Figure 2.10.5 displays:
This can be used to generate diagrams of each section of the car which can be defined by the user, providing a useful tool to compare with real world graphs and immediately show any errors in vehicle parameterisation [61].
3 Critical Analysis

3.1 Management Performance (GM)

In this section the performance of last year’s management is evaluated. The 2014 Brunel Racing team managed to achieve positive result at both competitions.

However serious management lacks showed throughout the year:

1) **Timing**: even though in 2014 the Brunel Racing team opted for a conservative vehicle design the release of component’s technical drawing (and hence their manufacturing) failed often to meet deadlines. This resulted in late car build and lack of track testing which affected the whole performance of the team at competition.

2) **Intra-management personal issues**: the performance of the 2014 management team was affected negatively by the presence of some personal issues between team members. This resulted often in discharge of responsibilities and tension in the team.

3) **Absenteeism**: during the “summer build time” some managers, including the team principal, were not present in the workshop. Most of the work was carried out by level 3’s and alumni which showed great interest and passion for the team and the vehicle.

**Task-allocation**: throughout the year the task allocation from the management to the level 3 students was not ideal. In some occasions there was a lack of hand-power due to poor task planning and in other occasions level 3 students were left without a clear guidance on what to do resulting in a waste of man power.
3.2 Corporate Branding & Public Image (GG)

Within Brunel this project, to a greater extent than any other, has a huge raft of stakeholders; these primarily form three groups.

Presently, the cost of being competitive within Formula Student is vast; to build the level of cars which Brunel Racing wishes we cannot be reliant solely on University funding. Therefore in the last 3 years the team has built-up a significant sponsor/partner base to fund the continued development of our vehicles. To put the size of this into perspective, the value of physical (non-software) contributions is now greater than Brunel’s financial commitment.

Brunel University London has significant commercial interest in our image and competitiveness. The Formula Student project at Brunel is used, in a significant part, to market the Motorsport Engineering course; students who wish to get into the motorsport industry are likely to consider the reputation or successfulness of a university’s FS team when selecting where to study. As a result the team is granted access to facilities within the university and offered financial backing.

Finally we have the team members; most of whom are likely to be looking for employment once their involvement within the team concludes.

For all three of these groups there is a clear need to present a professional public image. All need to ensure that by association their own images are not tarnished, but further enhanced.

But in order to build an image you have to have some level of continuity. Brunel Racing’s approach in this area over the last 5 years has had its good points and bad. Between BR-X
and BR-14 the team enjoyed a consistent livery, which allowed for development of the image. These colours were picked as they were not used by any of our main competitors at the time, and were kept as management teams realised that this continuity was important [31] [62], however these teams did not fully exploit this commercially.

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BR-XV changed colour scheme away from this last season in order to match in with a wider SED/Brunel Motorsport branding template.

3.3 Design Overview (GG)

3.3.1 Analysis of successful competitor vehicles

Brunel Racing competes against teams from all over the world, with victory usually being taken by one of a relatively small group of teams. Within these teams a number of trends have developed over the last 5 years.

Figure 3.3.1. 2014 season cars from (left to right) Monash Motorsport, RennTeam Stuttgart and GFR, arguably the three top teams in the world [130] [131] [129]
Aerodynamic devices have become generally accepted following their successful use by top teams as is evident in Figure 3.3.1.

Top cars, even with aerodynamic devices tend to have a mass of 200kg or lower. Teams have more recently switched to smaller wheel and tyre packages, primarily to reduce unsprung, rotating mass.

Many cars have also decided to switch to inherently more unreliably powertrain options with smaller, more highly tuned engines in a bid to reduce mass. Of the 3 cars shown, only Stuttgart continue to utilise the once common 4-cylinder motorcycle engine; however they also have the best track-record for reliability and typically finish the majority of competitions in one of the top 3 positions.

Although subjective, it has been noticed that teams following the traits examined here tend to have better change of direction and more stable cars with minimal pitch and roll behaviour.

3.3.2 General critique of BR-XV – GG

![Figure 3.3.2. BR-XV competing at FS-CZ](image-url)
As can be seen in Figure 3.3.2, the car Brunel Racing presented last year (at 215kg) did not fall in-line with any of the discussed trends. Whilst following trends does not constitute an engineering justification for a design decision, it is reasonable grounds for further investigation to question and understand why an alternate approach has been taken.

It has been concluded that too many designs have been nonchalantly carried over year-to-year, without proper investigation of their suitability for the intended purpose. Some of the integrity of the data previously presented must be questioned as discrepancies and oversights have been discovered by the authors of this report.

3.4 Competition Performance (GG)

The competition is based on a points system as outlined in Section 1.2, generally these are considered to form 2 groups; dynamics and statics.

3.4.1 Dynamic Performance

Despite the criticisms raised, it is acknowledged that BR-XV performed successfully on track [62]; scoring particularly highly in the acceleration event thanks to a very powerful engine.

The car did however suffer, particularly at FSUK, from handling irregularities as shown in Figure 3.4.1.
BR-XV and indeed BR-14 was noticeably poorer than its competitors and even its predecessors in cornering behaviour. BR has previously been successful in the Skid-Pad event; however both BR-XV and BR-14 scored poorly here. It is suggested that this may be due to the lower chassis torsional stiffness values found on these two cars.

BR-XV did score relatively well in the Sprint/Autocross and Endurance events, however this is attributed to the straight line acceleration of the engine alongside a competitive and experienced driver line-up. The car was visibly handicapped through the corners as shown in Figure 3.4.1

3.4.2 Static Performance

3.4.2.1 Design
BR-XV scored exceptionally poorly in the design event at FSUK, although did better at FS-CZ. The Design Score Sheet from FSUK showed that less than 50% was scored in most areas, with an overall score of 65/150 prior to moderation and 70/150 points being awarded subsequently. The team were criticised heavily for their “execution of concept”, “attention to detail”, quality of manufacture, “no clear targets”, lack of validation, lack of data presented to back up the discussion, the over-use of carry-over components, “spaghetti
“wiring” and a lack of “innovation for 16th car” produced. A copy of this document can be found in the Portfolio of Evidence. [63]

3.4.2.2 Business

BR-XV also scored poorly in the business event. The business event assumes a role-play scenario where the team pitches to potential investors. From feedback on the day from the judges it was apparent that they didn’t know who they were meant to be in our role-play scenario. As this is crucial to how the plan is viewed it obviously had an adverse effect on the scoring.

Prior to FS-CZ the business plan was re-assessed as part of an attempt to improve static performance. At this point it was realised that numbers in the original plan, pertaining to the cost of hiring race-circuits which formed an integral part of the plan, were wildly unrealistic. This combined with further unrealistic initial production volumes made the plan unjustifiable in the judging event.

3.4.2.3 Cost

The cost event is one where BR’s score has been improving for the last few years, 2014 was no exception. The team score particularly well in the real-case scenario however lost points for the lack of technical drawings provided in this sub-task.

None-the-less improvements are still possible within cost report itself, as a number of fasters were found at FS-CZ to be missing from the BOM.

3.5 Vehicle Testing (CE)

3.5.1 Vehicle Testing

In previous years, a large amount of testing has been conducted throughout the year, mostly at Bruntingthorpe Proving Ground. Testing has been used to develop system prior to
their installation on the following car and for driver evaluation. What has been lacking are accurate records of the information gathered at each test, such as vehicle dynamics data and records of set-up, failure reports and test plans. Because of this, mistakes have been repeated, as knowledge and solutions to problems have not been passed from team to team.

3.6 Tyres (CE)

3.6.1 Use of TTC data

For the past seven years, Brunel Racing has been using 13” wheels and tyres. The trend in recent Formula Student competitions has seen a change by many teams to 10” wheels because of the advantages in lower unsprung mass. Since 2008, the team has been a member of the FSAE TTC and has had access to all the past and current tyre data. This resource has never been used to inform the design of the suspension and steering systems of the vehicle, although analysis of one particular tyre has been carried out by the author [64]. Because the tyre is the only point of contact between the vehicle and the road surface, the whole performance of the vehicle is determined by the tyres. By not using the TTC resource to lead the vehicle design, potential performance benefits are being ignored.

3.7 Unsprung (EJ)

3.7.1 Actuation

In both BR-14 and BR-XV, the front suspension was mounted inboard of the space frame chassis and underneath the drivers’ legs. This centralised and lowered the centre of gravity of the front suspension, but made it very difficult to make changes to the front suspension at testing and competition, especially when the front wings and nose were mounted, for which the whole front Aero-package had to be taken off to access the front suspension.
On BR-XV, the rear dampers were mounted in a direct-actuation layout, using carbon fibre extensions. This resulted in stiffer springs being used on the rear due to the compromised motion ratio and made adjustments to the rear ride height harder.

3.7.2 Wheels

BR-14 used Braid 13”x7.0” wheel Rim s, with a 4 x 98 PCD, which had been used by the team for many years. It was found that these wheels were not interacting well with the hubs and the wheel centre experienced severe flexing. Marks were being exchanged between the wheels and hubs and both parts were becoming damaged over time.

The wheels used for BR-XV, 13”x7.0” with a 4x100 PCD, were of better quality and mounted much better to the hub, however, due to a mis-communication in the order form, the wheels required chamfered top-hats for each stud to locate and clamp the wheel correctly. This spacer caused difficulties during the running of the car.

3.7.3 Clevis Shims

All previous versions of Brunel Racing cars have used shims on all outboard suspension geometry mounting positions to adjust outboard wheel properties. Although this gives good adjustability, it also adds complexity to the outboard assembly with increased fixings and the need for more clevises.

3.7.4 Anti-Roll Bars

BR-XV did not have a rear anti-roll Bar and the front Anti-roll bar was a carry-over part from BR-14, which affected the vehicles handling and delayed vehicle set-up optimisation. The team also have no recorded data as to the calculations or stiffness of this carried-over part which complicates the new design of anti-roll bars.
The blade system which was used in BR-14 is well known and is used in many forms of motorsport [19] [5], and is easy to design and adjust, however, if placed between 0 and 90 degrees, the blade is somewhat unpredictable and may not deform as expected/requested, because the load induces bending into the blade.

3.7.5  Steering

Whilst bench-marking BR-14, it was found that the front camber was adjusted by shimming both the lower outboard wishbone point and the toe-arm point, which unintentionally adjusted the steering geometry. This shimming of the toe arm point turned the static parallel steer layout to a static negative steer layout by a few degrees. This would have meant the vehicle could still navigate the corners, but would have changed how the vehicle handled and when it would be able to hit the apex. This is due to the rate of toe change on the inside and outside front tyres and their respective tyre slip angle change rate [5].

During BR-XV, the steering clevis was not shimmed to adjust camber and the drivers thought the car had improved handling [65], however it cannot purely be put down to this contributing factor. Without shimming the toe-arm clevis, the designed steering characteristic was better retained.

3.7.6  Rocker Bearings

The rocker bearings used on BR-14 were not specified correctly and required replacement after 1 race weekend because the ball bearings had fallen out and the bearing races were no longer located where they started. Figure 3.7.1 shows a view of the Front Left rocker and bearing which was damaged after competing. [42]
On BR-XV, the rocker bearings were better selected, but were mounted without a spacer between the mounting tab on the chassis and the external faces of the bearing, meaning that the tab clamped up against the bearing restricting movement when the fixing bolt was tightened. This resulted in the nut having to be loosened to allow the damper to act as it should and release the friction in the rocker bearing.

3.8 Chassis (GG)

3.8.1 Torsional Stiffness Requirements

The last 2 vehicles produced by Brunel Racing have featured spaceframe chassis construction, from driver feedback is has been acknowledged that understeer behaviour has been severely detrimental to the handling of both of these cars to varying extents.

Whilst the problem is apparent particularly on the last 2 vehicles, it has also been noticed on some of our monocoque cars of previous years, with this author having observed this behaviour on track with BR-XI, and further evidenced from video footage of a 4-post rig test at Lola. Whilst the problem over understeer can partially attributed to suspension geometry, set-up and tyre characteristics, Deakin et.al demonstrated that insufficient torsional
stiffness of the chassis leads to an inability to control the front-rear roll stiffness distribution of the chassis; effectively leading to understeer behaviour.

3.8.2 The totality of Design

A common issue of the last several cars has been the lack of a complete design; this has led to a culture of leaving things out with the intention that we will just make something during the build phase which has not been designed. This “welding on a tab” approach produces significant unseen weight gains particularly on the chassis.

3.8.3 FE Modelling

Work by this author on the FE methods employed by the team has found that previous boundary conditions have been unrealistic, often overestimating the torsional rigidity of the vehicles [25]. As BR vehicles have generally been designed to have an adequate torsional stiffness based upon the suspension, this under-approximation in the FE modelling techniques is likely the root cause of the understeer behaviour which has been seen in recent cars.

Previous physical testing and simulations by BR have been constrained at the suspension points on the frame as illustrated in Figure 3.8.1. It was found that these boundary conditions produce, when compared with simulations constrained at the hubs, artificially inflated values by a factor of 2 [25].

![Figure 3.8.1. Illustration of Torsional Stiffness measurement on previous BR vehicles](image-url)
This means BR-XV’s designed 1600Nm/degrees may in reality be as little as 800Nm/degrees, and compliance effects will further reduce this.

A test method which better matches the real world loading should be defined, one axel needs to be constrained in the z axis whilst the other is inclined under load, constraint in the x and y axis should be avoided where possible to avoid over-constraining the model relative to the real world.

Figure 3.8.2 demonstrates a set-up which mostly achieves this [66]. It is suggested that an additional pivot on the rear axle x axis to allow for rear track width change during loading.

3.9 Aerodynamics (GM)

3.9.1 Pitching moment

The main issue found in last year’s design was an excessive pitching moment that the car experienced at high speed. This is mainly attributable to two factors:

1. **Poor front wing design and incorrect aerodynamic balance:**

   Last year’s design target was to generate “as much downforce as possible”. No or little consideration was given to the ratio of downforce generated by the front wing and rear wing, with the rear aerofoils producing almost 65% of the force.
The front wing was also not designed to exploit the ground effect, with the main plane featuring high camber and high angle of attack. This meant that little area of the aerofoil is at the optimum ground clearance.

Furthermore the wing is composed in two different sections: this resulted in a waste of available space (hence reduced frontal projected area) causing a limitation in the maximum lift achievable. (see lift equation).

![Figure 3.9.1 View of the Front Wings of BR-XV in CAD](image)

2. **Location of rear wing:**

The rear wing of BR-XV was designed to generate great amount of downforce (and hence high drag). This component was positioned very high up (to exploit free stream clean air) and very far back compared to the centre of gravity of the vehicle (CoG). This resulted in the lift and drag forces creating very large moments at the CoG. The
combined moments resulted in an overall pitching moment of the vehicle. This tendency of the car to lift the front axle at high speed created understeer and an overall loss of performance.

3.9.2 Wing mount design

BR-XV’s aerodynamic package was design and manufactured well before the 2014’s competition in the United Kingdom, however it was never utilised at the event. This was due to a lack of consideration of last year’s management team towards wing mount design. The attachment methods of the wings were in fact designed after the manufacturing of the aero components: this “last minute job” resulted in unsafe structures that were unable to transmit the loads correctly and failed to restrain the tendency of the aerofoils to move causing a risk for the driver and personnel around the vehicle.

![Figure 3.9.2 Schematic of BR-XV with Force and Moment explanation](image)
3.10 Driver Controls & Electronics (TM)

3.10.1 Brake System and Pedal Box

The brake system on BRXV was functional but not adequate. The brakes worked and efficiently stopped the vehicle (except for disc failure due to poor manufacturing). The system had no reasoning to its design choices.

The pedal box was over-engineered. The pedal box on BRXV had a total mass of 2.646kg [38]. The pedals were adjusted by fixing them to points on rails. By fixing them directly to chassis mounting points and eliminating the rails there is an immediate mass saving of 468g [38]. The pedals are also able to be adjusted by more than the required amount. The pedal box for this year will be designed to accommodate a driver range of 5\textsuperscript{th} percentile female to 95\textsuperscript{th} percentile male.

3.10.2 Steering Wheel and Column

The steering wheel on BRXV was poor in design. There was insufficient space for all wiring to be enclosed meaning wires were exposed to the elements. The traction control switch does not function. The steering column used sub optimal components and therefore compromised steering sensitivity and durability.

The GEMS LDS4 display was not used on BRXV but will be this year. This enables almost any information to be displayed to the driver eliminating RPM LEDs and improving driver performance [67]. The screen will be mounted on the steering wheel.
3.10.3 Data Acquisition

To improve vehicle performance data must be recorded during testing. BRXV had approximately 21 sensors on the vehicle. Due to various proposed and confirmed designs the number of sensors has increased to 42. Full proposed sensor list in section 8.5.2.2.

3.10.4 Driver Environment

Due to restrictions imposed by chassis design, there will be no seat in this year’s car. This will reduce overall mass of the vehicle. Instead foam seating will be used and fitted around the driver(s).

The dash and switches in BRXV are well placed and are easily accessible to the driver, however, are improperly wired. The indicator LED is activated by the switch and not from the relevant component.

3.10.5 Electronics

Design judging on BRXV said “Most features that were planned were not on the car. Spaghetti wiring” [68]. A comprehensive redesign of the wiring loom is required due to additional features. A well designed and constructed loom will make it easy to trace connection problems.

BR-15 wiring diagram was largely incorrect and has been checked by multiple managers. The diagram did not follow the specified wiring pins for the ECU and data logger and mentioned in the Bosch manuals [69] [70]. The diagram must be correct and followed to avoid confusion and damage to components. All safety regulations were followed. If changes need to be made to the loom due to an incorrect diagram the diagram must be updated. This will improve knowledge transfer to the following year group.
3.11 Innovative Design Solutions (JS)

A new sub team to Brunel Racing, Innovative Design Solutions, will explore areas of Brunel Racing’s Formula Student car that have not been considered before. The sub team will focus on innovative design; looking at a problem or development of an area of the car, exploring all possible routes to the design and particularly focusing on designs that have not been considered before by Brunel Racing or competitor Formula Student Teams.

The key development that will be considered by the sub team for BR-16, is the development of an upgrade package for the current in-progress design of the car, focusing on the development of improved vehicle handling. To achieve this, the design of an active aerodynamic and active suspension system will be investigated.

The focus of the previous year’s aerodynamic team for BR-XV was the production of maximum downforce with considerations to drag as an afterthought. As a result a design with large front and rear wing elements was produced. Although achieving the goal of producing high downforce, the design subsequently majorly affected the stability of the car under braking and cornering. Furthermore, with focus purely on designing elements to produce maximum downforce, mounting of all aerodynamic components was also overlooked, resulting in last minute construction which ultimately resulted in failure of the components preventing their use at competition.
To prevent such a reoccurrence, the objectives of the design for the components developed by the sub team must be assessed, ensuring an overall performance upgrade to BR-16’s current design. The components interaction and integration with other sub systems of the car must also be fully investigated to maximise effectiveness of the component without hindrance to other systems.

3.12 Vehicle Simulation (CS)

A review of BR-XV showed that there was no vehicle simulation carried out for the car. Simulation would be a useful tool in the testing of the car as it can be carried out in-house once the model is validated. There is some data available from BR-12, when vehicle simulation using IPG CarMaker was last carried out. The main piece of data used from the BR-12 simulations was a digitized road which mapped the exact layout of Bruntingthorpe kart track. This has enabled the vehicle models to be compared against test data from Bruntingthorpe.
4 Project Objectives & Product Design Specification

4.1 Competition Targets (GM)

- Brunel Racing to become the top team from the United Kingdom in both competitions (FSUK and FSG)
- Achieve a score of 120 points in the Static “Design” event
- Brunel Racing to have a 10th overall finish position at FSUK
- Brunel Racing to have a 20th overall finishing position at FGS
- Represent positively Brunel University by working with professionalism throughout the competitions.
- Marinating sponsors and partners satisfaction with great visibility and good competitions results.

4.2 Role Allocation (GG)

An initial meeting for BR-16 was held in March 2014 with the aim of collecting a group of students ahead of the start of the academic year. Following this, and subsequent meetings, positions were agreed and proposed to the course director Dr. K. Matthys. The final management group was confirmed a week prior to the start of term 1, level 3 positions were proposed and published with interviews taking place in Week 1. Further details of this can be found in portfolio of evidence.

4.3 Sponsorship (GG)

One area of particular success within this project has been the vast increase in external support. For the first time in nearly a decade the level of external support was higher than funding direct from the university.
The primary reason for this success has been an aggressive level of directly contacting companies either at key trade exhibitions or via email communications.

Key to this has been the understanding of what is beneficial for our stakeholders. Most sponsors fit into two groups; potential employers and companies looking to publicise their products.

For this 2\textsuperscript{nd} group, it has been realised that companies with higher quality products at the top-end of any given market are likely to have greater profit margins and therefore the ability to support a project such as ours. Due to their place within a market, these companies typically have stricter quality standards; therefore material which has not been passed for sale, but still suitable for our use, may be of little expense to the company but of great value to BR.

Further details of the nature of the approaches and information about the deals put in place for BR-16 can be found in the portfolio of evidence.
4.4 Vehicle Testing (CE)

In order to become more successful, the team needs to be able to develop systems for the new vehicle throughout the year. The most obvious way of doing this is to use the previous years’ vehicle as a test bed. Test sessions need to be organised to allow this to happen, as well as to develop the new vehicle once it is built prior to competition.

4.4.1 Objectives

- Data collection: use data acquisition systems present in BR-XV to gather relevant data on a number of different vehicle systems for trackside and post-test analysis.
- Test Bed: systems in development to be tested on BR-XV before potential use on BR-16 when it has been designed and built.
- Familiarisation: allowing the management team to become familiar with the correct operation of a Formula Student vehicle.
- Driver Training: All drivers, both those that have driven for the team previously and new ones for this year, need to be trained on the handling requirements of a Formula Student vehicle.
- Development: once BR-16 has been constructed, a month’s worth of testing should be carried out to work on reliability, system & procedure familiarisation and performance optimisation.

4.5 Tyres (CE)

The use of 10” wheel/tyre combination is becoming increasingly common amongst the more successful teams competing in the major FSAE/FS events. Whilst a switch should not be made solely based on the fact that everyone else is doing it, the number of teams having success with this set-up cannot be ignored. An investigation should be made so that the
team itself can come to a conclusion as to whether a switch to 10” tyres/wheels is a viable option.

4.5.1 Objectives

- Evaluate the 10” tyres available to Formula Student teams, using the current 13” tyres as a baseline. The comparison will use tyre warm-up characteristics lateral & longitudinal friction coefficients, cornering & slip stiffness, and camber & load sensitivities as metrics.
- After selecting a tyre size and compound, use the knowledge gained to inform the design of the suspension kinematics in terms of camber gain, camber sensitivity and load sensitivity.

4.5.2 PDS

4.5.2.1 Size

Minimise unsprung mass and rolling resistance without sacrificing other performance parameters.

4.5.2.2 Tracking

Tracking a tyres’ usage over the year to provide a log of how and when each tyre has been used.

4.5.2.3 Lifespan

The selected tyre must be able to complete the particular dynamic event without an appreciable drop in performance against the measured metric.
4.6 Unsprung (EJ)

Unsprung project objectives;

- design a legal and usable suspension system for a single seater racing car to compete in the FSAE competition
- Ensure that all designed parts fit with each other and all bought in products fit with designed parts. Extensive use of CAD packaging will be used to achieve this objective.
- All components must be designed to last the expected life-cycle of the vehicle (100hrs)
- Pursue the release of designs early to increase the time that a rolling chassis is in the workshop before competition
- Ensure that all Level 3 Design Engineers meet their goals and contribute to the project.

After the objectives of the project were set, a PDS was created and the design goal for each major component was specified as follows;

- Load cases to be calculated for;
  - 2.4g lateral
  - 2g braking
  - 1g acceleration
  - 0.3 slip ratio, +/- 8° slip angle

- Minimum safety factor of 1.5 – (left to the discretion of the designer and the load case)

- 10.0”x7.0” wheel rim with a 245mm inner rim diameter with a 3-stud mounting pattern
- Optimise Suspension geometry for given packaging constraints to utilise the tyre as much as possible.

4.6.1 Task Allocation

In the Unsprung department, there were 4 Level 3 students, whose design tasks within the department are detailed in Table 4.1, along with the tasks which carry my personal responsibility;

Table 4.1 List of Unsprung Tasks and Task Owners
### 4.7 Chassis (GG)

The chassis objectives primarily relate to packaging the components within as small a space as possible. It is intended that the chassis will attempt to utilise the engine as a structurally integral component. Mass is to be kept to a minimum, however above all else the chassis must have sufficient torsional stiffness to enable predictable handling characteristics.

- Designed Torsional stiffness of $>3500\text{Nm/deg}$ at the wheels to ensure greater than $2250\text{Nm/deg}$ in the real world (inclusive of compliances)
- Overall mass of less than 30kg

Conform to all cockpit templates and minimum tube specifications set out in the 2015 FSAE technical regulations. [1]

### 4.8 Aerodynamics (GM)

#### 4.8.1 Project objectives

- Design and manufacturing of the BR-16 aerodynamics package
- Analysis and review of BR-XV aerodynamic package
- CFD model validation with track testing

#### 4.8.2 Product design specification for Aerodynamics

- Design and development of an aerodynamic package that produces an overall downforce of 670N at 11.1m/s. This shall include:
  
  a. Front Wing capable of generating between 200-300N of downforce.
  
  b. Rear Wing capable of matching the downforce generated at the front for optimum car balance
c. An under tray or “side wings” to add additional downforce.

- Design of lightweight and reliable wing mounting methods.
### 4.9 Driver Controls & Electronics (TM)

Known issues can be used to guide design decisions for BR-16. From this a specification can be created specifically for DC&E components.

#### 4.9.1 Product Design Specification for Driver Controls and Electronics Manager

<table>
<thead>
<tr>
<th>Component</th>
<th>Subcomponent</th>
<th>Description</th>
<th>Reasoning</th>
</tr>
</thead>
<tbody>
<tr>
<td>Driver Controls</td>
<td>Gems LDS4 Display</td>
<td>The Gems display enables the driver to see any information programmed to it.</td>
<td>GEMS makes it easy to display information to the driver (selected gear, tyre temp, revs etc.)</td>
</tr>
<tr>
<td></td>
<td>Seating and positioning</td>
<td>Largely dictated by chassis design, this involves foam padding/seating specific to each driver.</td>
<td>Ensuring the driver is comfortable is key. Most importantly the foam padding/seating must ensure the driver is safe. Driver anthropometrics data must be analysed</td>
</tr>
<tr>
<td></td>
<td>Braking</td>
<td>Brake system specification and force calculations</td>
<td>Calculate dynamic brake forces to allow for brake system design and brake line specification</td>
</tr>
<tr>
<td></td>
<td>Accelerator Pedal</td>
<td>Design an accelerator pedal fit for use with a range of drivers and in conjunction with the brake pedal</td>
<td>FEA analysis on pedal to establish weak points. FEA software use in order to optimize mass of pedal. This area was over-engineered in previous years</td>
</tr>
<tr>
<td></td>
<td>Dash and switches</td>
<td>Locating switches for efficient access and use</td>
<td>Important vehicle functions must be easily accessible for a range of driver sizes.</td>
</tr>
<tr>
<td>Electronics</td>
<td>Sensors</td>
<td>Increase the number of sensors, and their use and accuracy throughout the year</td>
<td>The increase in sensors will enable previously unknown data to be recorded and analysed. This should not only enable set up tweaks, but also advise drivers on. Currently there are 12 sensors (not including powertrain). We intend to add a further 17. This will add mass although measures will be taken to reduce the amount of wire used.</td>
</tr>
<tr>
<td></td>
<td>ECU</td>
<td>Ensure the ECU is easily accessible</td>
<td>The ECU needs to be easily accessible for retrieval and wiring purposes</td>
</tr>
<tr>
<td></td>
<td>Data logger</td>
<td>Ensure wiring of the data logger is correct</td>
<td>Previously, every time the laptop was connected to the Data logger the calibration and map was uploaded from the PC. This wiped the current map that had to then be redone. This will corrected.</td>
</tr>
<tr>
<td></td>
<td>Wire</td>
<td>Ensure no wire is wasted and adding additional mass</td>
<td>With the addition of 17+ sensors, the mass of the electronics is likely to increase. Some wires can be combined to save mass. This can be calculated when the wiring map is created.</td>
</tr>
<tr>
<td></td>
<td>Design</td>
<td>The design of the loom is to be clear and simple. It must be easy to trace an error/broken connection.</td>
<td>The design of the loom should be easy to follow to enable quick detection of a fault. Engine electronics will be connected via a single large Deutsch connector behind the driver for quick disconnection and mass saving.</td>
</tr>
</tbody>
</table>

*Table 4.2 Driver Controls and Electronics PDS*
4.9.2 Objectives and targets for Driver controls and electronics manager

Objectives and targets can be created to measure the successfulness of the designs.

4.9.2.1 Driver Controls: Environment

- Ensure the Gems LDS4 display is in use and a screen for each event is programmed.
  The Gems display enables the driver to see any information programmed to it.
- Fit driver foam seating specific to each driver. The foam padding/seating must ensure the driver is safe. Driver anthropometrics data must be analysed.
- Design an accelerator pedal fit for use with a range of drivers and with the brake pedal which must have a mass of no more than 270g. FEA analysis to be completed on pedal to establish weak points. Achieve a weight reduction of around 50% from BR-XV [38].
- Locating switches for efficient access and use. Important vehicle functions must be easily accessible for a range of driver sizes.
- Pedal box must be adjustable for drivers ranging from 5th percentile female to 95th percentile male [37].

4.9.2.2 Electronics: Data Acquisition

- Ensure all sensors required for data logging are wired correctly and retrieving data.
  There are nine powertrain sensors and at least 25 other vehicle sensors are to be wired into the vehicle. Overall mass will increase, this impact must be minimised.
- The ECU and data logger need to be easily accessible for retrieval and wiring purposes, and easily retrievable.
4.9.2.3  Electronics: Wiring

- Ensure no wire is wasted; adding additional mass. With the addition of many more sensors, grouping power and ground lines as soon as possible can reduce mass of the loom.

- The design of the loom is to be clear and simple. It must be easy to trace an error/broken connection.

4.9.2.4  Key Regulations

- Primary Master Switch. All battery current must flow through this switch. Regulation: IC4.2 [37].

- Cockpit-mounted Master Switch. Must be located for easy actuation for driver. Regulation: IC4.3 [37].

- The Brake-Over-Travel-Switch forms part of the shutdown system and as defined in regulation T7.3 must kill the engine and fuel pumps [37].

- Battery installation must meet FSEA regulations IC4.4 [37].
4.10 Innovative Design Solutions (JS)

The following project objectives were set for the Innovative Design Solutions sub-teams based on the Literature and Critical Review of Brunel Racing’s previous Formula Student cars:

- Design and development of aerodynamic devices as an addition to that of the aerodynamic components also in development by the aerodynamics sub team
  - Target: Provide improved performance and optimisation of aerodynamic package
- Development and integration of an active aerodynamics package
  - Target: Improvement of vehicle handling and downforce distribution through braking and cornering
- Design of DRS for front and rear wings
  - Target: Prevent suspension overloading at high speed and reduce straight line drag to optimise vehicle acceleration and velocity
- Development and integration of an active suspension package
  - Target: Active control of suspension articulation dependent on vehicle acceleration, braking and load distribution to improve handling, reducing vehicle roll, pitch and yaw.
- Creation of full IPG simulation model of active system solutions designed for BR-16
  - Target: Gain simulation data for use in writing active system operating code
- Physical testing of active system on track
  - Target: Validation of simulated model, showing areas of alteration and optimisation
From the project objectives a set of Product Design Specifications were made for each component and system designed:

1. Secondary aerodynamic components must work in cohesion with the developed aerodynamics package, providing an overall performance improvement.

2. Performance improvement gained from secondary aerodynamic components must outweigh the weight deficit incurred by the components.

3. Secondary aerodynamic components must have innovative designs not previously used in Brunel Racing.

4. Active Aerodynamic system must be integrated into developed aerodynamic package, with implementation of minimal additional components reducing system weight and additional cost.

5. Active Aerodynamics system and DRS must use the same hardware yet work coherently.

6. DRS must reduce overall drag of car in a straight line by a minimum of 25% at 15m/s.

7. Activation of DRS must be controlled by the driver via an easy to use UI requiring minimal driver distraction.

8. Active Suspension system must work in coherence with Active Aerodynamics to achieve improved handling objectives.

9. Active Suspension design must have minimal weight deficit at each corner with minimal cost implications.

10. Full active system, Active Aerodynamic and Active Suspension, must reduce vehicle roll and pitch by a minimum of 25% to a maximum of 1 degree about each axis respectively around the Formula Student dynamic event tracks.
11. Full active system must be programmable for each Formula Student dynamic event individually enabling optimal setup for the event.

12. IPG simulation model must incorporate Active Aerodynamic system, Active Suspension system and DRS to gain a realistic simulation of the performance improvement provided by the systems for BR-16.
4.11 Vehicle Simulation (CS)

- Create a vehicle model for BR-XV using IPG CarMaker, using Simulink Models and Matlab where possible to improve accuracy
- Create a vehicle model for BR-16 using IPG CarMaker, using Simulink Models and Matlab where possible to improve accuracy
- Compare vehicle data obtained from physical tests around Bruntingthorpe Proving Ground’s kart track with results obtained from IPG Control to validate vehicle models
- Compare BR-XV model and BR-16 models to determine which performs better around Bruntingthorpe kart track
- Map Silverstone sprint/endurance track to predict BR-16’s performance in the sprint/endurance event at FSUK 2015

Create a vehicle data sheet showing BR-XV and BR-16 technical data which can be put onto a shared drive and updated for future cars
5 Project Management

5.1 Technical Risk Assessment

5.1.1 Risk Register (CE)

To manage the technical risks within the project, a risk register was created. The team sought to identify all of the risks associated with the year, such as those that apply to the design and construction of the final vehicle and others, such as budget issues and academic priorities. The risks in the register were scored from 1-10 on severity and occurrence. The product of these gives an overall score, where a higher score requires more attention. This is a live document and was updated each month to reflect the different risks that arise as the project progresses. Figure 5.1.1 shows an excerpt, with the whole document presented in the PoE. The register shows that some risks deteriorate over time. This is because as designs become further delayed, the risk to other areas of the project will increase. For the risk of injury fields, as the year goes on, the number of people in the workshop will increase, so the potential risk of injury will increase.

| Risk ID | Description                                                                 | Probability (1-10) | Consequence (1-100) | Issue (January) | Issue (February) | Issue (March) | Issue (April) | Issue (May) | Issue (June) | Issue (July) | Issue (August) | Issue (September) | Issue (October) | Issue (November) | Issue (December) | Mitigation                                                                 |
|---------|----------------------------------------------------------------------------|--------------------|---------------------|-----------------|------------------|----------------|---------------|--------------|---------------|---------------|----------------|-------------------|--------------------|------------------|----------------|-------------------|--------------------------------------------------------------------------|
| 1       | Component changes are not compatible on time                               | 4                  | 8                   | 32              | 68               | 50             | 68            | 48           | 48            | 52            | 14              | 14                | 31                |                  |                   | Design status will be tracked by the relevant manager, Design freeze LS managers deadlines will be implemented. |
| 2       | Component changes are not compatible with each other                       | 5                  | 8                   | 40              | 68               | 38             | 68            | 32           | 28            | 14            | 34              | 34                | 31                |                  |                   | The different design teams must communicate so that interacting designs are monitored closely. LS managers. |
| 3       | Computer software not validated in the design office                        | 8                  | 9                   | 34              | 30               | 18             | 12            | 12           | 12            | 12            | 12              | 12                | 12                |                  |                   | Trai  | Maintenance of software will need to be installed. Maintain communication with IT departments to work out what needs to be done. LS managers |
| 4       | Technicians not exercised during the building phase                         | 3                  | 7                   | 21              | 16               | 21             | 21            | 13           | 13            | 23            | 13              | 23                | 33                | 34               |                  | Instruct technicians when out of house access is needed. Test Manager |
| 5       | Risk of injury in the workshop                                             | 5                  | 6                   | 18              | 24               | 14             | 24            | 14           | 14            | 14            | 14              | 14                | 30                | 30               |                  | Correct PPE worn by all members of team working in the workshop. Adequate training provided to individuals before machines will be operated. Those working within the workshop must make sure the area used is cleaned after use. All team members |
5.2 Safety Risk Assessment (CE)

5.2.1 Workshop

With the number of student’s involved with the team this year, there is a likelihood that people are unfamiliar with the correct behaviour required in the workshop. It is therefore the responsibility of the management to educate the team on the correct ways to work safely in a workshop environment. Workshop ‘101’ sessions were organised throughout the year on various pieces of equipment for those who were interested in assisting the team. The correct use of PPE is promoted, as is the culture of ‘If you aren’t sure, ask!’ and a ‘Think before you do’ approach. Table 5.1 below shows the risk evaluation for the workshop. This is assessed using the standard Brunel University London Health and Safety risk evaluation matrix (Figure 5.2.1). Below this are examples of severity and likelihood levels to be used to correctly evaluate the potential risks. This will be used by the team to evaluate potential risks before any activity is undertaken in the workshop.

![Figure 5.2.1: General risk evaluation matrix for Brunel University London](image-url)
5.2.2 Testing – Millbrook

This year, the team had the opportunity of testing at Millbrook Proving Ground on weekends. Figure 5.2.2 shows an excerpt from the risk assessment document, with the full version available in the PoE. The action plan describes the actions that are to be taken by the team to mitigate against the high-scoring risk highlighted earlier in the document. In this case, it is for fire and explosion during refuelling, FS car collision with barrier(s) and FS car frontal crash. Figure 5.2.3 shows an aerial view of the steering pad with the direction of the acceleration runs marked (red line), as well as the 10m zone that must be kept clear at all times.

<table>
<thead>
<tr>
<th>Evaluation after implementation of controls</th>
<th>Severity</th>
<th>Likelihood</th>
<th>Evaluation</th>
<th>Classification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Contact with moving machinery</td>
<td>Major</td>
<td>Possible</td>
<td>12</td>
<td>Medium</td>
</tr>
<tr>
<td>2 Injury by projectile e.g. drilling, grinding</td>
<td>Major</td>
<td>Possible</td>
<td>12</td>
<td>Medium</td>
</tr>
<tr>
<td>3 Hand/wrist injury from spinning work piece</td>
<td>Major</td>
<td>Possible</td>
<td>12</td>
<td>Medium</td>
</tr>
<tr>
<td>4 Burn</td>
<td>Serious</td>
<td>Possible</td>
<td>10</td>
<td>Medium</td>
</tr>
<tr>
<td>5 Slips, trips and falls</td>
<td>Serious</td>
<td>Possible</td>
<td>10</td>
<td>Medium</td>
</tr>
<tr>
<td>6 Electric shock</td>
<td>Major</td>
<td>Unlikely</td>
<td>8</td>
<td>Low</td>
</tr>
<tr>
<td>7 Fire or explosion</td>
<td>Major</td>
<td>Unlikely</td>
<td>8</td>
<td>Low</td>
</tr>
<tr>
<td>8 Contact with sharp materials etc.</td>
<td>Major</td>
<td>Probable</td>
<td>9</td>
<td>Low</td>
</tr>
</tbody>
</table>

Table 5.1: Work risk evaluation

<table>
<thead>
<tr>
<th>Section</th>
<th>Action Plan</th>
</tr>
</thead>
</table>
| 2nd point | 1. No driver in the vehicle or persons not required for refueling are near the vehicle during refueling.  
2. Fire training has been carried out by members of the team and are using appropriate fire extinguishers.  
3. Persons with 1st aid training are in support  
4. Area where refueling is taking place has been cleaned |
| 9th & 11th points | 1. Cones to be used to mark a 10m no-go zone around the pavement to avoid vehicle contact with the metal staircase and the gravel pit surrounding it. |

Figure 5.2.2: Excerpt from the risk assessment for Millbrook Proving Ground, showing the action plans in place to mitigate against the high-scoring risks identified.
Bruntingthorpe Proving Ground has been a test venue for the team for numerous years. The procedure for activities is well defined in terms of operations and track safety. In keeping with good risk assessment practice however, the assessment is reviewed before and after each test to decide if any changes need to be made to the document. The full risk assessment for Bruntingthorpe can be found in the PoE. Figure 5.2.4 shows an aerial view of

Figure 5.2.3: Aerial view of the steering pad at Millbrook. The red line signifies the length and direction in which acceleration runs were conducted. The blue dotted line delineates the 10m exclusion zone around the metal staircase. All members of the team not required to attend to the car when it is stationary must stand behind the yellow line.

Figure 5.2.4: Aerial view of the kart track at Bruntingthorpe Proving Ground. The yellow dots signify locations where marshals in high-visibility jackets equipped with fire extinguishers will be positioned. The blue rectangle shows where the van/pit will be set up.
the kart track at Bruntingthorpe that the team uses. The fire marshal points are identified, as is the location of the van and pit equipment set up.
5.3 Timing Plan (CE)

The nature of the Formula Student project within the academic year means that tight deadlines are imposed. The time that the management team must spend on the project is balanced against other university commitments throughout the year. To give the team direction and organisation, a timing plan with sufficient accuracy must be created. The method to convey this information has been chosen as a Gantt chart, created in Microsoft Project. As early as March 2014, top level decision and deliverables were identified for the vehicle group. This allowed design and manufacturing deadlines to be early on in the year, with the aim being to maximise track testing time in June. Key level 3 deliverables relating to the design of the vehicle were identified and assigned to the most competent students.

5.3.1 Design Phase

From the key deliverables within the design phase of the project, a critical path was identified that outlined the key milestones for completion of the vehicle before the first competition. This allowed the management team to easily track the progress of the project and assign additional resources if necessary to meet upcoming design and/or manufacturing deadlines.

5.3.1.1 Design Freeze

Within the project are several components that need to have the specification of their design frozen, beyond which designs are not normally altered. A design is frozen to ensure the project stays on schedule and because it then allows other components to be designed and manufactured around the frozen part without the fear of the design being altered after these tasks have started. The 3rd party verification procedure outlined in section 2.2 is applied to the part prior to it being frozen.
5.3.1.2 Drawing Release

Following a design becoming frozen, an engineering drawing is released for manufacture. Each drawing has been checked in line with the teams’ 3rd party verification procedure, and in most cases with staff in the machine shop to check adequate tolerances and datum features have been defined.

5.3.2 Build Phase

With the majority of the design phase complete, the build phase of the project is entered. Long lead time items have been identified and a manufacturing plan created to help communicate this information to the management team and level 1, 2 & 3 students assisting during this phase.

5.3.3 Testing schedule

As stated in 4.4.1, one of the main objectives for this year is to use BR-XV as a test bed for developing systems and hardware for the new vehicle. After discussion with the management team, a number of provisional dates for track testing were set throughout the year to allow data to be gathered to drive the design of BR-16. Figure 5.3.1 shows the planned tests throughout the year. In order to minimise the amount of money is spent on testing, certain tests can be combined, as long as they do not interfere with each other and invalidate results.

![Figure 5.3.1: Provisional test schedule for the year.](image-url)
5.4 Budget (EJ)

Managing the budget is an ongoing task throughout the project and has played an important role this year as the University faculty has been restructured and has required weekly budget and expenditure updates. This was also the first year where Brunel Racing was required to submit a budget for all types of expenditure including Marketing, Testing, Equipment, Build, and Competition. With increased budgets to prepare and track, all the budgets were grouped into one document with an expenditure tracking sheet.

![Figure 5.4.1 View of the expenditure tracking sheet](image-url)

Figure 5.4.1 shows a view of the expenditure tracking sheet for each department in the vehicle build and the expected monthly expenditure in each budget. This predicted expenditure relates to the S-Curve shown in Figure 5.4.2, along with the actual expenditure.

By keeping track of all this expenditure against planned and budgeted spend, it was easy to determine if the project was under or over-budget. On the March 15th, the project was £7836 under the budgeted amount, but was £29226 lower than the planned spend. This means that there is £21389 that had not been spent on items that were planned to be purchased by this time, either due to redesign, or unavailability of parts.
5.4.1 Testing Budget

The testing budget covers a finite number of tests that need to be carried out using both BR-XV and BR-16. Expenditure includes hourly rates for hiring the test venue, if applicable, and fuel for both the FS vehicle and the team’s van. Consumables such as brake discs and brake fluid and tyres for testing have also been included. The full budget is shown in Figure 5.4.3.
5.4.2 Cost Report (CS)

5.4.2.1 Rules and Regulations

As per the FSAE regulations, the cost report is a table of costs of parts for the competing vehicle. It includes a BOM which lists all parts of the vehicle and groups them into assemblies [1]. The Cost Report must use the standardised cost tables provided and name the parts and assemblies according to the abbreviation appendix [1].

The Cost Report itself is part of the Cost Event of Formula Student competitions, which consists of:

- Cost Report (presentation and submission) [1]
- Discussion (with cost judges to evaluate overall vehicle cost and the accuracy of estimates) [1]
- “Real Case” (scenario where team will respond to a cost or manufacturing related challenge) [1]

The Cost event is marked on the team’s ability to work to a budget effectively, where vehicles which cost less overall will score higher [1]. It is also marked on accuracy of costing and manufacturing feasibility along with the performance of the team in the real case scenario [1].

The Cost Report is submitted as both an electronic copy and a hard copy for any competitions attended [1]. Subsequently the team must submit a Cost Report for both FSUK and FSG this year.

The deadline for FSUK submission is 29th May 5.00pm UTC [72] and the deadline for FSG submission is 5th June 12.00pm CEST [73].
5.4.2.2 Structure

Work commenced in January 2015 on the Excel document containing the cost report data. The sheet was set up using the format of last year’s Cost Report. This has been compared to the standardised cost tables released for this year and the layout is still applicable. The Cost Report will be filled in using the part numbers listed in the whole vehicle CAD drawing. This is currently being updated so the Cost Report remains blank at present.

![Image of Excel document](image)

*Figure 5.4.4 - An example part sheet from the Cost Report for FSUK submission*

Figure 5.4.4 - An example part sheet from the Cost Report for FSUK submission shows an exemplary sheet from the Cost Report document in its current state. The sheet has been temporarily colour coded for ease of use by the team, with the darkest colour representing an assembly, the medium colour representing a part and the lightest colour representing a diagram (where required). Each subsystem of the vehicle has a different colour. The primary sheet of the Excel document is an instructions sheet which the team can use to understand how the document works. It details which colours correspond to which subsystems and also explains how the assembly sheets work. Currently the assembly sheet is configured to pull cost values from parts sheets and automatically place them into the assembly sheet. The
formulae will need to be slightly modified when part numbers and costs begin to be added to the document in order to ensure that the correct parts sheets are contributing to the correct assembly sheets.

5.4.2.3 Management
The Cost Report documents for both FSUK and FSG competitions are managed by the same Level 5 team member. The reports will be broken down and work with individual team managers will progress in order to add in the corresponding part data one subsystem at a time. This eases the workload on fellow managers but also splits the work a little more evenly as the Cost Report manager will still assist in the development of the document and have a better understanding of what is required. Each manager is in charge of naming the parts for their own subsystems so it is useful to work with them to set up the Cost Report and ensure it is completed correctly.

5.5 Change Management (TM)
Change management is the process of evaluating required changes to a system. If a change is required the following process is adhered to:

- Identify potential change – new function/problem
- Analyse Change – determine feasibility/cost
- Evaluate change – review of request, committee (manager) vote
- Plan Change – analyse impact on other components, time scale
- Implement change – execute and test

Review and close change - documentation completed [74].
5.6 Team Meetings (CE)

The Vehicle Group takes part in three meetings each week:

- Level 5 management meeting, involving all managers from the Vehicle and Powertrain groups
- General team meeting, involving the management team and all level 3 students.
- Vehicle group meeting with the two academics supervising the Vehicle Group management team for the project.

A pre-meeting agenda is distributed to the relevant people before the level 5 management and academic supervisor meetings. Minutes are taken to record the content of each meeting, highlighting the progress made in the past week and the tasks that have been assigned to each team member in the coming week. The PoE contains the meeting minutes taken for both the level 5 management and academic supervisor meetings.

6 Corporate Branding & Public Image

6.1 Our Branding (GG & JS)

6.1.1 Brunel University London Rebranding (Summer 2014)

Prior to the start of the 2015-16 academic year Brunel University London was rebranded (see Figure 6.1.1); leading to the realignment of all sub-brands, including BR.

Figure 6.1.1. Brunel University London logos (left pre-summer 2014, right post-summer 2014) [125]
The new branding scheme primarily focuses on two shades of blue and one shade of red. Happily two of these colours are incidentally similar to the team’s old logo (Figure 6.1.2) which had been use, with minor updates, since the first years of the team until 2013.

![Brunel Racing Logo](image)

*Figure 6.1.2. Pre-2013 Brunel Racing logo*

Although the team has had to once again reinvent its image as a result of these changes, the fortunate choice of colours has allowed the new branding to be linked to our past.

### 6.1.2 New Logo

The new logo makes use of the new Pantone 200 and Pantone 540 colours specified by Brunel’s new branding policy [75].

![Brunel Racing Logo](image)

*Figure 6.1.3. 2015 Brunel Racing Logo*

The previously used font for the “BRUNEL” part of Brunel Racing has been changed to the same font as the “RACING”, the font previously used for this was originally taken from Brunel branding in the late 1990’s and hence given the university rebranding was considered inappropriate for continued use. A few minor changes have also been made to clean up the letters “I” and “G” as shown in Figure 6.1.3.

Often Brunel Racing shortens itself to “BR”, for example cars are typically named BR-“something”. Therefore a shortened version of the logo is shown in Figure 6.1.4.
6.1.3 House Style

Brunel’s brand book outlines fonts and colours to be used [73]; the majority of the team’s branding has attempted to tie directly to this. The team has also made prominent use of the Brunel “bars”; 3 coloured stripes which must run from top to bottom of any material upon which they are used.

6.1.4 General Presentation of the brand

Generally, the team are trying to present a very clean, de-cluttered, minimalist image. This in no small part is based upon examples from within the motorsport industry; teams which are most successful commercially mostly have this approach as shown in Figure 6.1.5.
6.2 Expressing the Brand (JS, GM, GG)

6.2.1 Website (JS)

Most commonly a website is the first iteration between a business and the general public, and this is the case for Brunel Racing. As a result Brunel Racing's website must express the brand identity of Brunel Racing, giving the viewer an immediate insight in the team, racing formulas the team is involved with and the possible partnerships with the team in the case of corporate sponsors. The new website design has been developed in accordance with the new Brunel University house style, yet incorporating engaging design to provide the user with an experience directly related to Brunel Racing. The following figures, Figure 6.2.1, Figure 6.2.2, Figure 6.2.3, and Figure 6.2.4, illustrate the main design structure of the website, the content available and the key features of the website design. The website is not yet published and still in development.

![Figure 6.2.1 View of the Websites Welcome Page](image)
Figure 6.2.2 View of the top of the Formula Student Home Page

Figure 6.2.3 View of the BR-16 sub-systems linking to further information

Figure 6.2.4 View of the Competition result page
6.2.2 Social Media (GM)

6.2.3 Newsletters (GG)

In order to communicate better with our alumni, sponsors and other stakeholders the newsletter was re-introduced in January 2015. Figure 6.2.5 shows an example newsletter which is aligned with the new image.

![Figure 6.2.5. Excerpt from January Newsletter](image)

6.2.4 Clothing (JS)

In building a strong brand identity, apparel, uniforms and outwear are crucial. Clothing is worn by the team at events, competitions and during corporate meetings. Well-designed clothing gives a positive representation of the team and its ethos. Figure 6.2.6, Figure 6.2.7, and Figure 6.2.8 show some of the design iterations.
Figure 6.2.6 View of the Shirt design – Iteration 1  

Figure 6.2.7 View of the Jacket Design – Iteration 1  

Figure 6.2.8 View of the Jacket design - Iteration 2
6.2.5  Pit Display (GG)

6.3  Sponsorship/ Partnerships (GG)

The team has been fortunate to develop a significant increase on the number of partnerships it enjoys with industry. A full list of the partnerships has been provided in the portfolio of evidence.

6.3.1  Approaching Companies

Key to the successful partnerships which have been developed has been the way in which we have attempted to present the team. The majority of companies we are working with are larger corporate organisations who care about the way in which their brand is presented and associated.
7  Vehicle Testing (CE)

The ultimate aim this year for vehicle testing is not to improve the on-track performance of BR-XV, but to use it as a test bed on which relevant data can be gathered to develop systems for use on BR-16 once it has been manufactured. The test program for the year should be constructed to allow data to be gathered to aid in dissertations and to improve the performance of the team at competition. The objectives for the year are highlighted in 4.4.1.

7.1  Methodology

7.1.1  Test venues

The team has traditionally used the kart track at Bruntingthorpe Proving Ground. This has a mix of low and medium speed corners similar to those seen in the coned circuits used at competition events. For this year, the team has the availability of using the steering pad at Millbrook Proving Ground on weekends. This 137m diameter [76] concrete surface is ideal for testing a large number of different vehicle scenarios.

7.2  BR-XV testing

7.2.1  Bruntingthorpe Proving Ground 10/10/14

This first test was mainly to allow the management team to familiarise themselves with the operation of BR-XV and to shakedown the vehicle following competition earlier in the year. Initial procedures for sensor calibration and tyre temperature & pressure measurement could also be determined.

The test highlighted issues with the battery charging and engine & differential mountings. A broken engine mount on the chassis was traced to a weld failure due to lack of penetration.
The differential carriers were seen to bend under drivetrain load, which led to slackening of the chain, which could be re-tensioned at the track, but a more permanent fix would be necessary. The full test plan and report can be found in the Portfolio of Evidence.

7.2.2 Millbrook Proving Ground 06/12/14

This was the first test for the team new venue Millbrook Proving Ground on their steering pad. This test would be to gather initial data for the traction and launch control systems, as well as carrying out constant speed runs to determine the tractive limit of the vehicle in each gear. This would then be used to define the gear ratios for BR-16. The launch control system would be tested on a straight 75m run akin to the Acceleration event at competition. The traction control system would be tested on a coned figure-8 circuit identical in dimensions to that used in the Skid Pad event at competition.

Issues with gear selection curtailed running, but some useful data was gathered. Following the test, work at the Motorsport Centre needed to be carried out to check that with the gear selection issues fixed that launch and traction control systems activated. The full test plan and report can be found in the Portfolio of Evidence.

7.2.3 Millbrook Proving Ground 24/01/15

The second test session at Millbrook was to further test the launch and traction control systems. A fix had been found for the gear selection issues that occurred at the previous test by reprogramming the ECU. Set-up change to the launch control system yielded some positive reviews by the drivers, but there was some inconsistency in the results because of a damp track.
7.2.4 Bruntingthorpe Proving Ground 26/02/15

This second test at Bruntingthorpe would combine two individual tests. The first of these was an aerodynamic test carried out on one of the runways at Bruntingthorpe to correlate CFD simulations with track data and the second test was to develop the traction control system. A more in-depth description of the test can be found in [gaba’s section on the test].

Also tested was an updated traction control system. There was an issue with the wheel speed sensor and engine health that curtailed running late in the day, with the full test plan and report in the PoE.
8  BR-16 Design

8.1  Tyres (CE)

8.1.1  Current tyres used by Brunel Racing

Currently, the team uses Hoosier 20.5x7.0 – 13 R25B dry slicks and Avon 7.2/20.0 wet tyres. In both cases they are mounted to a 7” wide wheel rim.

8.1.2  10” comparison

A comparison will firstly be made between the 10” tyres that have been tested by the TTC; the Hoosier 18.0x6.0 R25B and the Hoosier 18.0x6.0 LCO, both on a 6” and 7” wide rim.

8.1.2.1  Instantaneous Cornering Stiffness

Instantaneous cornering stiffness is a good performance indicator of a tyre. The gradient of the curve is a measure of how quickly the available lateral force decreases with an increase in slip angle.

Figure 8.1.1: Instantaneous cornering stiffness vs. slip angle for the Hoosier 18.0x6.0 LCO 6” rim tyre at 10psi.
Figure 8.1.1 and Figure 8.1.2 show instantaneous cornering stiffness vs. slip angle for the Hoosier 18.0x6.0 LCO 6” width rim at 10 and 12psi respectively. The tyre was also tested at inflation pressures of 8psi and 14psi, but the data had a large amount of noise due to issues with the test rig and in the case of the 8psi runs, excessive movement of the tyre on the rim at higher normal loads. The tyre at 12psi has a larger cornering stiffness, indicating a higher lateral force potential. The differences at lower loads are small, and at higher loads are more noticeable. This lower stiffness value means the rate at which cornering stiffness falls away is less, which indicates a tyre that behaves more predictably.

Figure 8.1.4 and Figure 8.1.3 show the Hoosier 18.0x6.0 LCO 7” width rim at 10 and 12psi respectively. As before, the data set for the 8psi runs contained a large amount of noise. There appears to be a very small difference between the maximum instantaneous cornering
Stiffness at both inflation pressures. At high normal loads (1100 - 1550N), instantaneous cornering stiffness values are nearly equal. This is a possible indication that the tyre has reached its peak lateral force potential in this load range. Above this, no more force may be available. At 10psi, the tyre appears to be less sensitive to changes in inclination angle compared to the 12psi case. The gradient of each curve looks to be similar for both cases. Cornering stiffness is available over a wide range of slip angles.

Figure 8.1.4: Instantaneous cornering stiffness vs. slip angle for the Hoosier 18.0x6.0 LCO 7" rim tyre at 10psi.

Figure 8.1.3: Instantaneous cornering stiffness vs. slip angle for the Hoosier 18.0x6.0 LCO 7" rim tyre at 12psi.
Figure 8.1.5, Figure 8.1.7 and Figure 8.1.6 show the Hoosier 18.0x6.0 R25B 6” rim at 8, 10 and 12psi respectively. Here, the 8psi data contained less noise than in the previous cases. Looking at the 8psi case first, the peak cornering stiffness is at a normal load of around 670N. Anything much above this seems to be beyond the capability of the tyre. For lower normal loads, the curves have a narrow shape, with the rate of decrease in cornering stiffness being greater. In the 10psi case, the tyre again appears to have its peak cornering stiffness at 670N range, similar to the 8psi graph. But at lower loads (200-450N) the cornering stiffness is less than that of the tyre at 8psi. For the 12psi tyre, the peak cornering stiffness occurs at a higher normal load than the previous two inflation pressures. This peak cornering stiffness is higher than either of the two previous cases. At lower loads, the peak cornering stiffness is lower than both the 8psi run and the 10psi one. The range of slip angles that the tyre provides lateral force over is narrower than the 8 and 10psi cases.

Figure 8.1.5: Instantaneous cornering stiffness vs. slip angle for the Hoosier 18.0x6.0 R25B 6” tyre at 8psi.
Figure 8.1.6: Instantaneous cornering stiffness vs. slip angle for the Hoosier 18.0x6.0 R25B 6” rim tyre at 12psi.

Figure 8.1.7: Instantaneous cornering stiffness vs. slip angle for the Hoosier 18.0x6.0 R25B 6” rim tyre at 10psi.
Figure 8.1.8, Figure 8.1.9 and Figure 8.1.10 show the Hoosier 18.0x6.0 R25B tyre on a 7” width rim for 8, 10 and 12psi respectively. On the graph showing the 8psi case, there is little difference between the 670N and 1100N curves. The 1550N curve is lower than both of these, but this may be a result of the head that the tyre is mounted to bottoming out on the test bed. For the 10psi case, the curves have a more rounded profile compared to the tyre at 8psi. The peak cornering stiffness is greater than before, with the curves featuring a steeper gradient. At the medium and high loads, the range of slip angles over which the cornering stiffness is greater than zero is narrower. Of the three cases, the tyre at 12psi inflation pressure has the highest cornering stiffness. For loads less than the 670N curve, the amount of cornering stiffness decreases slightly in comparison to the previous 8 and 10psi cases. Each curve has a similar gradient to the tyre when it was at 10psi, and zero cornering stiffness occurs at a slightly larger slip angle.
Figure 8.1.9: Instantaneous cornering stiffness vs. slip angle for the Hoosier 18.0x6.0 R25B 7” rim tyre at 10psi.

Figure 8.1.10: Instantaneous cornering stiffness vs. slip angle for the Hoosier 18.0x6.0 R25B 7” rim tyre at 12psi.
8.1.2.2 Instantaneous Camber Stiffness

The instantaneous camber stiffness is used to quantify how much of a change in lateral force will result when the camber angle is changed. Figure 8.1.11 and Figure 8.1.12 show instantaneous camber stiffness for the Hoosier 18.0x6.0 LCO tyre mounted in this case on a 6” wide rim at 10 and 12psi respectively. For the five normal loads presented in the 10psi case, the curves are evenly spaced and are all above 0N/deg. There appears to be very little decrease in camber response as slip angle increases, with a large response for all slip angles at all normal loads. This means that camber will have a large effect in near straight lines and under hard cornering. At 12psi, the peak camber stiffness is higher than before, and this is accompanied by more of a sudden drop in camber response on all curves as slip angle increases. The spread of the curves for the range of normal loads is less, and at the lower normal loads, camber response is lower as slip angle increases.

![instantaneous camber stiffness vs. slip angle for the Hoosier 18.0x6.0 LCO 6" rim tyre at 10psi](image)

*Figure 8.1.11: Instantaneous camber stiffness vs. slip angle for the Hoosier 18.0x6.0 LCO 6" rim tyre at 10psi*
Figure 8.1.12: Instantaneous camber stiffness vs. slip angle for the Hoosier 18.0x6.0 LCO 6” rim tyre at 12psi.

Figure 8.1.13: Instantaneous camber stiffness vs. slip angle for the Hoosier 18.0x6.0 LCO 7” rim tyre at 10psi.
Figure 8.1.13 and Figure 8.1.14 show the Hoosier 18.0x6.0 LCO tyre mounted in this case on a 7” wide rim at 10 and 12psi respectively. For the tyre at 10psi, the curves are closely spaced, particularly at the lower end of the normal load range. All of them show a decrease in camber response as slip angle decreases. This drop in response from the peak value is less severe for the lower normal loads. For the tyre at 12psi, there is a much greater loss of camber response at 1100N in comparison to the previous 10psi graph. This falls to a value near zero as slip angle increases above 4deg., indicating that camber will have less of an effect under hard cornering.

Figure 8.1.15, Figure 8.1.16 and Figure 8.1.17 show the Hoosier 18.0x6.0 R25B tyre mounted on a 6” wide rim at 8, 10 and 12psi respectively. For the 8psi graph, the camber response is not greatest at the highest load. At high normal loads, there is less of a decline in camber
response as slip angle increases. Even at lower normal loads, the decrease in camber response was not as severe, though the degree of camber response is quite high. At 10psi, the highest normal load has the highest peak camber response. The peak is quite rounded, indicating that the tyre will be less affected by camber over a wider range of small slip angles. For low normal loads, the peak camber stiffness is lower, but there is a similar level of camber response at higher slip angles. At 12psi, the peak instantaneous camber stiffness is very similar to the 10psi case, with a much more gradual loss of camber response as slip angle increases at medium normal loads.

![Figure 8.1.15: Instantaneous camber stiffness vs. slip angle for the Hoosier 18.0x6.0 R25B 6” rim tyre at 8psi.](image-url)
Figure 8.1.16: Instantaneous camber stiffness vs. slip angle for the Hoosier 18.0x6.0 R25B 6” rim tyre at 10psi.

Figure 8.1.17: Instantaneous camber stiffness vs. slip angle for the Hoosier 18.0x6.0 R25B 6” rim tyre at 12psi.
Figure 8.1.18, Figure 8.1.19 and Figure 8.1.20 show the Hoosier 18.0x6.0 R25B tyre mounted in this case on a 7” wide rim at 8, 10 and 12psi respectively. At 8psi for high normal loads, there is a gradual decrease in camber response, but with a high degree of response at high slip angles. At low loads, there is a more sudden decrease in camber response in comparison. At 10psi, the peak instantaneous camber stiffness is higher relevant to the tyre at 8psi. For the highest normal load, the peak is wider for low values of slip angle. This gives a similar loss of camber response when compared to the 8psi case, but occurs over a smaller range of slip angles. For medium and low normal loads, the peak camber stiffness is lower than the 8psi for a slightly narrower range of slip angles. This means that camber will have a larger effect when moving in a straight line. For the tyre at 12psi, the peak camber stiffness value was very similar to the 10psi graph. For the higher normal loads, 1100N & 1550N, the camber response drops to a lower value. For the lower normal loads, there is almost no response to camber above 4deg. slip angle.

![Instantaneous camber stiffness vs. slip angle for the Hoosier 18.0x6.0 R25B 7” rim tyre at 8psi.](image-url)
Figure 8.1.19: Instantaneous camber stiffness vs. slip angle for the Hoosier 18.0x6.0 R25B 7" rim tyre at 10psi.

Figure 8.1.20: Instantaneous camber stiffness vs. slip angle for the Hoosier 18.0x6.0 R25B 7" rim tyre at 12psi.
In the above graphs, the 7” width rim provides a higher peak value of instantaneous camber stiffness across all of the inflation pressures tested. The data for this rim width appears to be more consistent than that for the 6” rim. For this performance metric, the LCO and R25B compounds have very similar levels of performance. The R25B has a smaller variation in stiffness between the inflation pressures tested, and generally has a stiffness value that decreases to zero at higher slip angles, whereas the LCO becomes typically negative in this scenario.

8.1.2.3 Lateral Friction Coefficient

The friction coefficient that a tyre develops can be split into lateral and longitudinal components. It is a function of normal load, lateral/longitudinal force, inclination angle and inflation pressure. Figure 8.1.21 shows how the lateral friction coefficient decreases with increasing normal load for the Hoosier 18.0x6.0 LCO 6” width rim tyre at 10 and 12psi. There is a large difference in friction coefficient at the two inflation pressures tested. The loss of

![Figure 8.1.21: Lateral friction coefficient vs. normal load for the Hoosier 18.0x6.0 LCO 6” rim tyre at 10 and 12psi](image-url)
friction coefficient as normal load increases is greater for the tyre at 10psi.

Figure 8.1.22 shows the Hoosier 18.0x6.0 LCO 7” width rim tyre at 8, 10, 12 and 14psi. As before, there is a large difference in friction coefficients for the range of inflation pressures tested. The rate at which the friction coefficient decreases is similar for the 8, 10 and 14psi cases.

Figure 8.1.23 shows the Hoosier 18.0x6.0 R25B 6” width rim tyre at 8, 10, 12 and 14psi. The graph shows that there is a large difference between 14psi and the other three inflation pressures tested. The rate at which friction coefficient is lost for the four inflation pressures is only marginally different.

![Figure 8.1.22: Lateral friction coefficient vs. normal load for the Hoosier 18.0x6.0 LCO 7” rim tyre at 10 and 12psi.](image-url)
Figure 8.1.23: Lateral friction coefficient vs. normal load for the Hoosier 18.0x6.0 R25B 6” rim tyre at 8, 10, 12 and 14psi.
Figure 8.1.24 shows the Hoosier 18.0x6.0 R25B 7” width rim tyre at 8, 10, 12 and 14psi. The graph shows that there are very similar levels of friction coefficient between the four inflation pressures tested. All of the lines have a comparable rate at which the friction coefficient decreases with an increase in normal load.

The graphs above show that there is a linear relationship between the friction coefficient and normal load. Comparing the two compounds, the LCO has a higher coefficient than the R25B, and generally the rate at which the friction coefficient decreases with normal load is greater as well. However, there is a larger difference between inflation pressures for the LCO tyres than there is for the R25B tyres.

8.1.2.4 Longitudinal Friction Coefficient

Figure 8.1.25 shows longitudinal friction coefficient vs. normal load for the Hoosier 18.0x6.0 LCO 6” width rim tyre at 10 and 12psi. Due to an issue with the sign convention in the
OptimumTire software used to create the tyre models, one of the lines has a negative friction coefficient. The value for the 12psi case has a higher value by around 1. As normal load increases, the tyre at 12psi appears to experience a greater decrease in friction coefficient compared to the 10psi case.

![Figure 8.1.25: Longitudinal friction coefficient vs. normal load for the Hoosier 18.0x6.0 LCO 6" rim tyre at 10 and 12psi.](image)

Figure 8.1.26 describes the Hoosier 18.0x6.0 LCO 7" width rim tyre at 8, 10, 12 and 14psi. The highest value for the 8psi tyre is substantially larger than would be expected on any surface that the vehicle is likely to run on. This anomaly is likely because of the issues discussed earlier when the tyre was run at 8psi. The other inflation pressures tested achieve values within a range of around 1. The rate at which friction coefficient is lost is very similar for the 10 and 14psi cases, but the 12psi line shows a different, more severe decrease in friction coefficient.
Figure 8.1.26: Longitudinal friction coefficient vs. normal load for the Hoosier 18.0x6.0 LCO 7" rim tyre at 8, 10, 12 and 14psi.

Figure 8.1.27 describes the Hoosier 18.0x6.0 R25B 6" width rim tyre at 8, 10, 12 and 14psi.

Again, there is an issue with sign convention. The 10psi line appears to produce the highest friction coefficient of the four inflation pressures, with this and the 14psi case showing a
similar rate of decay in friction coefficient. The tyre inflated to 8psi shows the least amount of friction coefficient loss as the normal load increases.

Figure 8.1.28: Longitudinal friction coefficient for the Hoosier 18.0x6.0 R25B 7” rim tyre at 8, 10, 12 and 14 psi.

Figure 8.1.28 shows the Hoosier 18.0x6.0 R25B 7” width rim at 8, 10, 12 and 14psi. Ignoring the sign convention issues again, the values across all four inflation pressure sensors are very similar. As well as this, the rate of decay is comparable.

As with the lateral friction coefficient, there is again a linear relationship between friction coefficient and normal load. In terms of peak values, the 6” rims appear to slightly outperform the 7” rims. The tyres using the LCO compound show more of a variation in the rate of decay of longitudinal friction coefficient for different inflation pressures. The R25B 7” width rim in particular shows very similar rates of decay, regardless of inflation pressure.
8.1.2.5 Conclusions

Following a comparison of the two different compounds on two different width rims, the R25B compound tyre on a 7” wide rim will be chosen as the potential 10” wheel option. This is for several reasons. Firstly, the R25B compound is a known entity; the team has been using it on their 13” tyres for a number of years. Past team experience is that the LCO has a shorter working life on track, and so would be potentially worse over an endurance event. Instantaneous cornering and camber stiffness for the R25B are generally better than those of the LCO tyres in terms of peak cornering stiffness values and the rate of camber decrease. Even though the LCO has a higher lateral and longitudinal friction coefficient, the R25B compound shows less variation with changes in inflation pressure. Across nearly all of the performance metrics, the 7” wide rim appears to outperform the 6” wide version.

8.1.3 13” vs. 10” comparison

Based on the analysis and conclusions from 8.1.2, the Hoosier 18.0x6.0 R25B tyre on a 7” rim has been selected as a viable option for comparison with the Hoosier 20.5x7.0 R25B 7” tyre currently used.

8.1.3.1 Instantaneous Cornering Stiffness

Figure 8.1.29, Figure 8.1.31 and Figure 8.1.30 show instantaneous cornering stiffness vs. slip angle for the 13” tyre at 8, 12 and 10psi respectively. Comparing the 8psi data here against Figure 8.1.8 for the chosen 10” tyre, the 13” has a higher peak instantaneous cornering stiffness at the three highest normal loads shown. At lower normal loads however, there is less of a difference between the two at this inflation pressure. The rate at which instantaneous cornering stiffness decreases with increasing slip angle is much sharper for
the 13”, and occurs across a much smaller slip angle range. The 10” curves all have a smoother decrease, offering lateral force over a wider slip angle range.

Comparing the two 10psi cases, Figure 8.1.9 and Figure 8.1.30, a similar pattern to that previously described is present. The 13” has a higher peak cornering stiffness at higher normal loads, with less of a difference between the two at lower normal loads.

Figure 8.1.10 and Figure 8.1.31 describe the 12psi cases for the 10” and 13” tyres respectively. As before, similar behaviour to the previous inflation pressures can be observed. Here in particular, the slip angle range for the 13” is very narrow in comparison to the 10” tyre.

*Figure 8.1.29: Instantaneous cornering stiffness vs. slip angle for the Hoosier 20.5x7.0 R25B 7” width rim tyre at 8psi*
Figure 8.1.30: Instantaneous cornering stiffness vs. slip angle for the Hoosier 20.5x7.0 R25B 7” width rim tyre at 10psi

Figure 8.1.31: Instantaneous cornering stiffness vs. slip angle for the Hoosier 20.5x7.0 R25B 7” width rim tyre at 12psi
8.1.3.2 Instantaneous Camber Stiffness

Figure 8.1.18 and Figure 8.1.32 describe the instantaneous camber stiffness vs. slip angle at 8psi for the chosen 10” and 13” tyres respectively. The 13” shows a very sudden decrease in camber response as slip angle increases, dropping to near zero for slip angles greater than 2°. This means that camber will have a greater effect when the vehicle is moving in near straight lines. This is in contrast to the 10” tyre, where there was a much less extreme decrease of camber response as slip angle increased.

![Instantaneous camber stiffness vs slip angle for the Hoosier 20.5x7.0 R25B 7” width rim tyre at 8psi](image)

Figure 8.1.32: Instantaneous camber stiffness vs slip angle for the Hoosier 20.5x7.0 R25B 7” width rim tyre at 8psi

Figure 8.1.19 and Figure 8.1.33 describe the 10psi case for the 10” and 13” tyres respectively. Similar to the 8psi scenario, there is a very sudden decrease in camber response as slip angle increases for the 13” tyre, with it having little effect after 4°. The load variation at small slip angles is similar to the 8psi case for the 13” tyre. This is more uniform than the 10” tyre in the same scenario.
Figure 8.1.33: Instantaneous camber stiffness vs. slip angle for the Hoosier 20.5x7.0 R25B 7" width rim at 10psi

Figure 8.1.34: Instantaneous camber stiffness vs. slip angle for the Hoosier 20.5x7.0 R25B 7" width rim tyre at 12psi
Figure 8.1.20 and Figure 8.1.34 describe the 12psi case for the 10” and 13” tyres respectively. The peak camber stiffness for the 13” tyre is very similar to the 10psi case, which is again higher than that of the 10” tyre. There is again a more sudden decrease in camber response when compared to the 10”. For the normal loads tested, there is almost no camber response for the 13” tyre above 5° of slip angle.

### 8.1.3.3 Instantaneous Slip Stiffness

Instantaneous slip stiffness can show to what degree the longitudinal force potential of a tyre increases or decreases with a change in slip ratio. Figure 8.1.35 and Figure 8.1.36 show instantaneous slip stiffness vs. slip ratio for the 10” and 13” tyres at 10psi respectively. The 10” tyre has a larger peak slip stiffness value at a higher normal load. At the intermediate normal loads, the slip stiffness is at a comparable level for the two tyres. The 13” tyre appears to operate over a wider range of slip ratios than the 10”, and has a slightly more gradual decrease in slip stiffness as slip ratio increases.

Figure 8.1.37 and Figure 8.1.38 show the instantaneous slip stiffness for the 10” and 13” tyres at 12psi respectively. Again, the 10” has a higher peak slip stiffness value, with the 13” having a positive stiffness over a wider range of slip ratios.

Figure 8.1.39 and Figure 8.1.40 show the instantaneous slip stiffness for the 10” and 13” tyres at 14psi respectively. In comparison to the previous two inflation pressures, there is a much smaller difference between the two. For the 10” tyre, the slip ratio range that gives a positive stiffness is wider than in the two previous cases.
Figure 8.1.35: Instantaneous slip stiffness vs. slip ratio for the Hoosier 18.0x6.0 R25B 7" width rim tyre at 10psi

Figure 8.1.36: Instantaneous slip stiffness vs. slip ratio for the Hoosier 20.5x7.0 R25B 7" width rim tyre at 10psi
Figure 8.1.37: Instantaneous slip stiffness vs. slip ratio for the Hoosier 18.0x6.0 R25B 7” width rim tyre at 12psi

Figure 8.1.38: Instantaneous slip stiffness vs. slip ratio for the Hoosier 20.5x7.0 R25B 7” width rim tyre at 12psi
Figure 8.1.39: Instantaneous slip stiffness vs. slip ratio for the Hoosier 18.0x6.0 R25B 7" width rim tyre at 14psi

Figure 8.1.40: Instantaneous slip stiffness vs slip ratio for the Hoosier 20.5x7.0 R25B 7" width rim tyre at 14psi
8.1.3.4 Lateral Friction Coefficient

Figure 8.1.41 shows lateral friction coefficient vs. normal load for the 13” tyre. In comparison to the 10” case (Figure 8.1.24) the difference between the inflation pressures tested is larger. The 13” has a higher peak friction coefficient, but the rate at which this is lost is higher than for the 10” tyre.

![Lateral friction coefficient vs. normal load for the Hoosier 20.5x7.0 R25B 7” width rim tyre](image)

**Figure 8.1.41: Lateral friction coefficient vs. normal load for the Hoosier 20.5x7.0 R25B 7” width rim tyre**

8.1.3.5 Longitudinal Friction Coefficient

Figure 8.1.42 describes longitudinal friction coefficient and how it is affected by normal load for the 13” tyre. The unexpectedly high friction coefficient at 12psi is a result of a similar issues described in 8.1.2.4. The friction coefficient is fairly similar when the 13” and 10” rims are compared, as is the rate at which it is lost as normal load increases.
8.1.3.6 Slip Angle at Peak Lateral Force

Figure 8.1.43 and Figure 8.1.44 represent the slip angle that is achieved at the peak lateral force against normal load at 10psi for the 10” and 13” tyre respectively. For the 13” tyre, peak lateral force occurs at a lower slip angle by roughly 2-3deg. The 10” tyre displays a peak in the curve followed by a decrease as normal load increases. This behaviour is different to the 13” tyre, where the slip angle at which peak lateral force is achieved constantly decreases as normal load increases. There is a constant difference between the three curves in terms of inclination angle for the 10” tyre.
Figure 8.1.43: Slip angle at peak lateral force vs. normal load for the Hoosier 18.0x6.0 R25B 7" width rim tyre at 10psi

Figure 8.1.44: Slip angle at peak lateral force vs. normal load for the Hoosier 20.5x7.0 R25B 7" width rim tyre at 10psi

Figure 8.1.45 and Figure 8.1.46 are at 12psi for the 10” and 13” tyre respectively. Similar behaviour to the previous 10psi case is shown by the 10” tyre, where a peak occurs at a normal load greater than 100N. The slip angle in this case is also higher and there is a slightly larger difference between inclination angles. The slip angle value for the 13” tyre is
comparable to the 10psi case, but there is less of a decrease in slip angle as normal load increase and there is an almost constant difference between inclination angles.

Figure 8.1.45: Slip angle for peak lateral force vs. normal load for the Hoosier 18.0x6.0 R25B 7” width rim tyre at 12psi

Figure 8.1.46: Slip angle for peak lateral force vs. normal load for the Hoosier 20.5x7.0 R25B 7” width rim tyre at 12psi
The shape of the curve is very different to before for the 10” tyre. Slip angle now decreases from 100N as normal load increases, but over a larger range of slip angles than the previous inflation pressures. There is also a larger gap between the inclination angle curves. For the 13” tyre, there are similar results to the 12psi case, but with even less of a decrease in slip angle as normal load increases.

Figure 8.1.47: Slip angle for peak lateral force vs. normal load for the Hoosier 18.0x6.0 R25B 7” width rim tyre at 14psi.

Figure 8.1.48: Slip angle at peak lateral force vs. normal load for the Hoosier 20.5x7.0 R25B 7” width rim tyre at 14psi.
The graphs show that the slip angle for peak lateral force in all cases is less for the 13” tyre. Having less slip angle on the tyre for a given lateral force will result in less tyre drag which will potentially ensure less rolling resistance during cornering. But by having a greater slip angle for a given lateral force, there is the potential that this will put more energy into the tyre, which could reduce the amount of time that it takes the tyre to reach its correct operating temperature.

8.1.3.7 Self-aligning Torque

Figure 8.1.49 and Figure 8.1.50 describe the self-aligning torque at 10psi for the 10” and 13” tyres. At this inflation pressure, the difference between the two tyres in terms of peak positive and negative torque is small. The curve profile of the 10” is much more rounded and appears to decrease in magnitude at a slower rate.
Figure 8.1.50: Self-aligning torque vs. slip angle for the Hoosier 20.5x7.0 R25B 7” width rim tyre at 10psi

Figure 8.1.51 and Figure 8.1.52 describe the self-aligning torque at 12psi for the 10” and 13” tyres. At this inflation pressure, the 10” is less symmetrical about the x-axis in comparison to the 13”. The 10” tyre peaks at a greater slip angle and again decreases at a slower rate. Both tyres have a very low torque at the smaller normal loads.

Figure 8.1.53 and Figure 8.1.54 describe the self-aligning torque at 14psi for the 10” and 13” tyres. The 10” tyre has torque values much lower than the 13”. The 10” tyre is less symmetrical about the x-axis and again decreases at a much slower rate than the 13”.

Figure 8.1.51: Self-aligning torque vs. slip angle for the Hoosier 18.0x6.0 R25B 7" width rim tyre at 12psi

Figure 8.1.52: Self-aligning torque vs. slip angle for the Hoosier 20.5x7.0 R25B 7" width rim tyre at 12psi
Figure 8.1.53: Self-aligning torque vs. slip angle for the Hoosier 18.0x6.0 R25B 7" width rim tyre at 14psi

Figure 8.1.54: Self-aligning torque vs. slip angle for the Hoosier 20.5x7.0 R25B 7" width rim tyre at 14psi
Overall, the 13” tyre generally has a higher self-aligning torque. This means that the tyre will be more stable in a straight line, but will be more difficult to move from this state. The 10” tyre should provide less resistance to steering inputs, making for a more responsive tyre. Comparing to 8.1.3.6 Error! Reference source not found., the 13” has a smaller difference between the slip angle at which peak lateral force is generated and the slip angle for peak torque.

8.1.3.8 Pneumatic trail

Figure 8.1.55 and Figure 8.1.56 show pneumatic trail for the 10” and 13” tyres at 10psi. The curves for the 10” tyre have a much more rounded peak in comparison to the 13”. This implies that for the 13”, the self-aligning torque will be present at small slip angles, which confirms the conclusion in 8.1.3.7 that the tyre will be more resistant to changes in direction at low slip angles. The rounder peaks of the 10” tyre indicate that the self-aligning torque decreases as soon as the tyre is steered away from the straight ahead position, even at small slip angles. This suggests that the 10” would be more responsive in this range, as the self-aligning torque would be less.
Figure 8.1.55: Pneumatic trail vs. slip angle for the Hoosier 18.0x6.0 R25B 7” width rim tyre at 10psi

Figure 8.1.56: Pneumatic trail vs. slip angle for the Hoosier 20.5x7.0 R25B 7” width rim tyre at 10psi
Figure 8.1.57: Pneumatic trail vs. slip angle for the Hoosier 18.0x6.0 R25B 7" width rim tyre at 12psi

Figure 8.1.58: Pneumatic trail vs. slip angle for the Hoosier 20.5x7.0 R25B 7" width rim tyre at 12psi
Figure 8.1.59: Pneumatic trail vs. slip angle for the Hoosier 18.0x6.0 R25B 7" width rim tyre at 14psi

Figure 8.1.60: Pneumatic trail vs. slip angle for the Hoosier 20.5x7.0 R25B 7" width rim tyre at 14psi
Figure 8.1.57 and Figure 8.1.58 show pneumatic trail at 12psi for the 10” and 13” tyres. Similar behaviour is presented to the 10psi case, with the rate at which pneumatic trail decreases being greater for the 13” tyre. Figure 8.1.59 and Figure 8.1.60 show pneumatic trail at 14psi. Again, similar behaviour to the two previous inflation pressure cases is shown.

8.1.4 Conclusions

The 13” tyre has been shown to be considerably stiffer than the 10” in terms of instantaneous cornering stiffness. This implies that along with the higher lateral friction coefficient, the 13” tyre has a higher amount of lateral force potential.

In terms of longitudinal dynamics, the 10” appears to be the superior tyre in terms of peak instantaneous slip stiffness, but operates over a narrow range of slip ratios. The graphs indicate that the 13” tyre achieves its maximum lateral force at a higher slip angle compared to the 10”. Normally, this would be a disadvantage, as tyre drag would increase the wear of the tread and reduce the tyres’ lifespan. This would not be beneficial during the endurance event. But it must be remembered that in the skid pad and sprint events, the tyres are cold at the start. Anything that causes more energy to be put into the tyre would be an advantage in this scenario, as the tyre would reach its normal operating temperature at a faster rate.

The self-aligning torque for the 13” tyre was greater than that of the 10”. This would be beneficial for straight line stability. As nearly all Formula Student circuits do not feature long duration high-speed turns, responsiveness is a more desirable trait in a tyre. The 10” tyre appears to display this in the way that the peak self-aligning torque is lower and the way that the pneumatic trail decreases with increasing slip angle.
Based on the conclusions drawn from the performance metrics stated, the Hoosier 18.0x6.0 R25B on a 7” wide rim has been chosen for the new vehicle. The two tyres are similarly matched in some of the metrics, but the 10” tyre, combined with the reduction in unsprung shows more traits that suit the Formula Student style of dynamic events.
8.2 Unsprung (EJ)

The Unsprung Department covers a large range of components on the vehicle and the rules which stipulate the design of these components for the FSAE 2015 competition are listed below:

- **T2.3** The car must have a wheelbase of at least 1525 mm (60 inches). The wheelbase is measured from the centre of ground contact of the front and rear tires with the wheels pointed straight ahead. [37]

- **T2.4** The smaller track of the vehicle (front or rear) must be no less than 75% of the larger track. [37]

- **T6.1.1** The car must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of at least 50.8 mm (2 inches), 25.4 mm (1 inch) jounce and 25.4 mm (1 inch) rebound, with driver seated. The judges reserve the right to disqualify cars which do not represent a serious attempt at an operational suspension system or which demonstrate handling inappropriate for an autocross circuit. [37]

- **T6.2** Ground clearance must be sufficient to prevent any portion of the car, other than the tires, from touching the ground during track events. Intentional or excessive ground contact of any portion of the car other than the tires will forfeit a run or an entire dynamic event. [37]

- **T6.3.1** The wheels of the car must be 203.2 mm (8.0 inches) or more in diameter.[14]

- **T6.5 Steering T6.5.1** The steering wheel must be mechanically connected to the front wheels, i.e. “steer-by-wire” or electrically actuated steering of the front wheels, is prohibited. [37]
• **T6.5.5** Rear wheel steering, which can be electrically actuated, is permitted but only if mechanical stops limit the range of angular movement of the rear wheels to a maximum of six degrees (6°). This must be demonstrated with a driver in the car and the team must provide the facility for the steering angle range to be verified at Technical Inspection. [37]

In summary, the above mentioned rules conclude the following;

• The car must have a fully functional suspension system, front and rear, and at least 51mm of usable wheel travel, the suspension must be stiff enough as to not let any portion of the car, except tyres, touch the ground,

• the steered wheels must be physically connected to the steering wheel.

• The wheels (rims) must be greater than 8.0” in diameter.

• The cars wheel base must be greater than 1525mm and the smallest track of the car must not be less than 75% of the larger track.

8.2.1 Suspension Geometry Design - Design of kinematics using tyre information – EJ & CE

With increased tyre information this year thanks to CE, the suspension geometry was more effectively designed. To start with, the initial tyre analysis suggested the wheel needed to have a Camber angle of between 0 and -2degrees [CROSSREF! CE] to achieve peak lateral force for a given Tyre Slip Angle.

Changing to a 10” wheel rim was also a big decision as the team had always used 13” wheel rims. This change meant that the outboard points were more restricted than they were previously and more packaging was involved in the initial geometry designs.
8.2.1.1 Camber

When developing a suspension design, camber gain can be achieved in many ways. Through several design iterations, [77], it was decided to gain camber on the rear axle through heave, and on the front axle through roll and steer. This would mean that there would be less camber change on the front axle during heave than on the rear and most of the camber gain on the front would come from the steering and rolling motion, much like some modern cars [78].

![Figure 8.2.1 Plot of Camber variation for the Front and Rear Axles with Heave](image)

Figure 8.2.1 shows a plot of camber variation through heave for the front and rear axle and Figure 8.2.2 shows the camber variation through a lap of our test track.
When looking at the camber variation around the test track, it can be visually extrapolated that the camber, on both the front and rear axle varies by approximately 2 degrees. As mentioned previously, the tyres work best between 0 and -2 degrees of camber, so this 2 degree variation around the test track would suit the tyres well and the vehicle can be set-up to be within this range during a lap.

8.2.1.2 Toe

To make the car more predictable, the geometry was designed to minimise bump steer with a target of less than 0.1 degrees through a 50mm heave sweep. With this in mind, the toe-arms were adjusted to achieve this, without compromising other design factors, especially the steering geometry on the front axle. By designing the geometry to reduce bump-steer, there are several benefits;
• less toe change with ride height change which benefits vehicle set-up – different drivers and payload
• more stable vehicle under braking with less toe change on either side reducing the likelihood of a bump disturbing the motion and the car “darting” to one side
• less tyre slip angle change during bump resulting in less variation in lateral force due to bump

Figure 8.2.3 shows a plot of the final iteration of suspension geometry and the toe change through the heave sweep. The graph shows that the front and rear toe change is <0.05degrees and <0.3degrees respectively. The rear toe change through heave was compromised due to packaging constraints, however was improved from previous iterations.

The front axle toes-out under negative heave (bump) which prepares the car for cornering and toes-in during positive heave (droop), which keeps straight-line stability under
acceleration. On the rear, the wheels toe-in under bump stabilising acceleration and slightly toe-out under droop.

8.2.1.3 Anti-Dive, Squat, and Lift

With the successful testing of an aero-package last year, and the decision to use an aero-package this year, the amount of anti-dive was increased to reduce the pitching motion and reduce the possibility of any damage to the aero-package. After testing BR-14, it was found that the rear suspension would almost fully compress under hard acceleration, shown in Figure 8.2.4, meaning that either the car didn’t have enough Anti-Squat, or the spring stiffness was too low. The springs used in this test were 175lbs/in and the car used 125lbs/in during the race season suggesting that the Anti-Squat designed into the geometry was too low. [77]

Figure 8.2.4 Snapshot of BR-14 testing. Top image shows the rear axle under braking and the lower image shows the rear axle under acceleration
Anti Dive (%) = \frac{M(a_x g)(%Braking_{front})}{m\left(\frac{a_x}{g}\right)\left(\frac{CoG_Z}{WB}\right)}

Equation 8.2.1: Anti-dive

Anti Squat (%) = \frac{SVSA_{Hi}}{SVSA_{Lo}}\left(\frac{CoG_Z}{WB}\right)

Equation 8.2.2: Anti-lift

Anti Lift (%) = \frac{M(a_x g)(1 - %Braking_{front})}{M\left(\frac{a_x}{g}\right)\left(\frac{CoG_Z}{WB}\right)}

Equation 8.2.3: Anti-lift

Table 8.1 Table of Anti-Characteristics for BR-14, BR-XV and BR-16 shows the Anti-Characteristics of the past few BR cars and this year’s design calculated from Equation 8.2.1, Equation 8.2.2 & Equation 8.2.3. By increasing the Anti-Dive to 34.2%, the wishbones will now take 34.2% of the pitching force and the spring will take 65.8%. The reduced force in the spring means that the front ride height will not decrease as much under braking for the same spring stiffness and improve the aerodynamic efficiency of the vehicle. The increased anti-dive reduces the pitching load by 25% and thus reducing the pitch angle by 25% also under braking.

<table>
<thead>
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<th></th>
<th>BR-14</th>
<th>BR-XV</th>
<th>BR-16</th>
</tr>
</thead>
<tbody>
<tr>
<td>Anti-Dive</td>
<td>7.57%</td>
<td>9.52%</td>
<td>34.21%</td>
</tr>
<tr>
<td>Anti-Squat</td>
<td>7.63%</td>
<td>10.36%</td>
<td>26.75%</td>
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<td>Anti-Lift</td>
<td>2.94%</td>
<td>3.99%</td>
<td>5.14%</td>
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</tbody>
</table>

Table 8.1 Table of Anti-Characteristics for BR-14, BR-XV and BR-16
8.2.1.4 Suspension Geometry changeability

To increase the changeability in the suspension some of the inboard mountings were given several positions on the chassis. The front suspension was given 4 extra positions on the front upper wishbone, 2 above and 2 below, and the rear suspension was given 1 extra position on each of the fore points allowing for a total of 20 different geometry combinations to adjust camber gain, roll centre, Anti-Dive and Anti-Squat.

<table>
<thead>
<tr>
<th>Wishbone and Position</th>
<th>Anti-Squat</th>
<th>Anti-Dive</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front Upper +20mm</td>
<td>N/A</td>
<td>32.69%</td>
</tr>
<tr>
<td>Front Upper +10mm</td>
<td>N/A</td>
<td>33.47%</td>
</tr>
<tr>
<td>Front Upper Neutral</td>
<td>N/A</td>
<td>34.21%</td>
</tr>
<tr>
<td>Front Upper -10mm</td>
<td>N/A</td>
<td>34.91%</td>
</tr>
<tr>
<td>Front Upper -20mm</td>
<td>N/A</td>
<td>35.58%</td>
</tr>
<tr>
<td>Rear Upper -20mm</td>
<td>40.93%</td>
<td>N/A</td>
</tr>
<tr>
<td>Rear Neutral</td>
<td>26.75%</td>
<td>N/A</td>
</tr>
<tr>
<td>Rear Lower +20mm</td>
<td>62.83%</td>
<td>N/A</td>
</tr>
<tr>
<td>Rear Upper – 20mm &amp; Lower +20mm</td>
<td>76.62%</td>
<td>N/A</td>
</tr>
</tbody>
</table>

*Table 8.2 Table showing how the different inboard suspension mounting points affect Anti-Characteristics*

Table 8.2 shows the effects of these geometry changes on the percentage of Anti-Squat and Anti-Dive. The front Anti-Dive percentage is relatively unaffected by the inboard changes, however when looking at Figure 8.2.5 and Figure 8.2.6, the front camber gain and roll centre is massively affected as was expected by moving the plane of the upper wishbone and thus moving the front FVIC.
The rear suspension is massively affected by the change in mounting location with respect to the Anti-Squat percentage, but is not effected when looking at camber gain or roll centre in Figure 8.2.5 and Figure 8.2.6 respectively.
8.2.2 Design of steering system using tyre information – EJ & CE

Using the Lankensperger Steering Geometry Principal, the front steering geometry was designed for a forward mounted Steering rack with Pro-Ackerman Geometry. Viewing Figure 8.2.7 [77], this allows for the opportunity to move the outboard point inwards to reduce the amount of Pro-Ackerman steering and potentially achieve Reverse-Ackerman Geometry if so desired.

Figure 8.2.6 Graph showing how the Roll Centre Migration Changes with different inboard Mounting Points. Plus (+) and Minus (-) here represent the positions +20mm and -20mm of their Neutral Position
This year, we are using a new steering rack, the Milter mRack262, which is lighter and quicker than the Titan steering rack previously used [79]. This new steering rack boasts a 210-degree lock-to-lock movement with 24.938mm of rack travel in each direction [79]. This was taken into consideration when designing the steering geometry as the inboard tie-rod coordinate was restricted in movement which meant the outboard tie-rod coordinate had to be closer to the steering axis than when the team used the Titan steering Rack.

8.2.2.1 Steady-State Low Speed
The design for a Pro-Ackerman steering system was derived from a Script calculating the required front Toe-angles of a low-speed, steady-state vehicle traversing a variety of corners in Formula Student [77]. The code determines the angles for a Pro-Ackerman vehicle; however, by manipulating the outputs, you can receive the requirements for a negative and parallel steer vehicle, by swapping and averaging the values respectively.

By developing this code, the basic vehicle geometry can be manipulated to receive the toe angles for a specified corner. By reducing the wheelbase from 1580mm to 1530mm, 5mm
larger than legally stipulated by the rules [37], the required angles reduced by 0.4 degrees at the CoG, and 1.3 degrees toe-out on the front inner wheel. The code also has an integrated corner cutting coefficient which allows the car to navigate a range of corner radii for a given track width & corner.

```matlab
course_width = 9; % defined by FSAE rule (m)
hairpin_d = 9; % minimum corner diameter (m)
hairpin_r = hairpin_d / 2; % minimum corner radius (m)
hairpin_raceline = hairpin_r - racelinefactor*((ft+rt)/2):

rIfr = hairpin_raceline - ft/2; % (m)
rOfr = hairpin_raceline + ft/2; % (m)
rIrr = hairpin_raceline - rt/2; % (m)
rOrr = hairpin_raceline + rt/2; % (m)

% assumed that rcar toe does not change
beta = asind(wb/rIfr); % inside degrees
gamma = asind((CoG_d*WE)/hairpin_raceline); % angle of CoG
alpha = asind(wb/rOfr); % outside degrees
delta = alpha + beta;
zeta = beta/alpha;
```

Figure 8.2.8 View of the Lankensperger Steering Geometry Solver Code

Figure 8.2.8 shows a view of the first few lines of the Lankensperger solver code and shows the equations for all the desired angles including the front inside wheel (beta), front outside wheel (alpha), centre of gravity (gamma) and their relationships to each other (delta and zeta).

8.2.2.2 Steering Geometry and Steering Forces

The Miltera mRack262 Steering Rack has a product specification, which states various characteristics and properties about the steering rack [79]. One important property when designing the steering geometry is the Maximum and Recommended load that the steering rack can withstand. By using Equation 8.2.4, the forces in the steering arm could be calculated, and thus determine if the geometry being analysed was being overloaded.
By analysing the Car after decelerating by 0.3g, and then cornering at increasing lateral acceleration, a plot of Torque in the Rack against lateral acceleration could be plotted, as shown in Figure 8.2.9. The Maximum Load that the Rack can withstand is 91Nm [79], but the data sheet recommends 24Nm [79]. From Figure 8.2.9, it is seen that the vehicle can corner at 2.6g with a steering torque that meets the recommend torque from the manufacturer.

![Figure 8.2.9 Graph showing how Torque on the Steering wheel changes with Lateral acceleration after decelerating 0.3g](image)

### 8.2.2.3 Steering Geometry Adjustment

After designing the steering geometry, the outboard tie-rod point was iterated to represent the shimming of the point inboard to see the effect of this on the percentage Ackerman and Instant Turn Radius. When calculating percentage Ackerman, 100% is equal to Pro-Ackerman geometry, 0% is equal to parallel geometry, and -100% is Reverse-Ackerman.
Figure 8.2.10 shows a plot of the percentage Ackerman change with steering wheel angle input for varying amounts of shims added to the outer steering point from position 1 to 8, relating to designed position and increasing in 1mm shim increments. It is visible that by shimming the outboard point inward the percentage of Ackerman geometry decreases resulting in a more parallel steer car.
Despite this change, Figure 8.2.11 shows that the instant turn radius does not change with shimming the outboard point. This means the car will still navigate the corner but will require different vehicle dynamic properties such as a different roll angle and yaw moment.

8.2.3 Suspension Geometry Design Summary

Error! Reference source not found. shows a summary of the suspension geometry characteristics and design parameters for the BR-16 suspension design.
8.2.4 Hubs and Uprights

With the decision to change to 10” wheel rims for improved tyre and vehicle performance, it was also decided to change to a 3-stud wheel arrangement and M8 wheel studs to reduce components & Unsprung mass.

<table>
<thead>
<tr>
<th>Suspension Geometry/Kinematic Property</th>
<th>Magnitude</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roll Centre height Front</td>
<td>31.19</td>
<td>mm</td>
</tr>
<tr>
<td>Roll Centre height Rear</td>
<td>60.94</td>
<td>mm</td>
</tr>
<tr>
<td>Roll axis inclination</td>
<td>1.11</td>
<td>deg</td>
</tr>
<tr>
<td>Front Castor</td>
<td>2.22</td>
<td>deg</td>
</tr>
<tr>
<td>Rear Castor</td>
<td>0</td>
<td>deg</td>
</tr>
<tr>
<td>Front King Pin Inclination</td>
<td>5.53</td>
<td>deg</td>
</tr>
<tr>
<td>Rear King Pin Inclination</td>
<td>6.07</td>
<td>deg</td>
</tr>
<tr>
<td>Front Mechanical Trail</td>
<td>7.39</td>
<td>mm</td>
</tr>
<tr>
<td>Rear Mechanical Trail</td>
<td>10</td>
<td>mm</td>
</tr>
<tr>
<td>Front Scrub Radius</td>
<td>59.03</td>
<td>mm</td>
</tr>
<tr>
<td>Rear Scrub Radius</td>
<td>40.11</td>
<td>mm</td>
</tr>
<tr>
<td>Anti-Dive</td>
<td>34.21</td>
<td>%</td>
</tr>
<tr>
<td>Anti-Squat</td>
<td>26.75</td>
<td>%</td>
</tr>
<tr>
<td>Percentage of Ackerman Geometry at full lock</td>
<td>85.11</td>
<td>%</td>
</tr>
<tr>
<td>Front corner spring motion ratio (average)</td>
<td>0.97</td>
<td>-</td>
</tr>
<tr>
<td>Rear corner spring motion ratio (average)</td>
<td>1.32</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 8.3: Table Summarising the Kinematic and Geometric Properties of the Suspension design for BR-16
The design of the hubs and uprights was undertaken by Level 3 student Oliver Schönenberger, with supervision and collaboration from myself. Table 8.4 shows the load cases for the Hub analysis.

<table>
<thead>
<tr>
<th>Load-Case</th>
<th>Front</th>
<th>Rear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal (Fz)</td>
<td>8438 N</td>
<td>8438 N</td>
</tr>
<tr>
<td>Braking (Fx)</td>
<td>5554 N</td>
<td>5554 N</td>
</tr>
<tr>
<td>Lateral (Fy)</td>
<td>2884 N</td>
<td>2884 N</td>
</tr>
<tr>
<td>Acceleration (My)</td>
<td>N/A</td>
<td>1515 Nm</td>
</tr>
</tbody>
</table>

Table 8.4 Load Cases for Unsprung Outboard Analysis

The loading criteria were calculated by using the design specification targets and calculating the worst case scenario. The Normal Load was calculated from a vehicle of 250kg decelerating at 2g, cornering at 2.4g, with aero-load, and receiving a 2g bump [80]. This combination of loads is severely unlikely as it essentially calculates full vehicle weight transfer to one wheel at top speed.

The Braking load case is also unlikely as it calculated the load going through one wheel if that one wheel were to decelerate the whole vehicle at 2g [80]. In worst case scenarios, the front axle would be used to decelerate the car completely, so at most, 70% of this load would be taken by one wheel if the front axle slowed the car completely.

The Lateral Load case was calculated from a simple bicycle model with lateral acceleration of 2.4g and 50% weight distribution [80]. This load case out of the 4, including the 2 mentioned before, has the least amount of safety built into the load case, and so it was important that the safety factor on the part with respect to the Lateral Load case took this into consideration.
The acceleration load case was calculated by using the maximum torque of the engine, put through the engines gear ratios and multiplying by 1.5 [81].

8.2.4.1 Hubs
The Hubs are the first part of the car, with the exception of the wheel and tyre, which transmits load into the spring and damper and must be able to withstand all the loads that it will receive without failure.

![Figure 8.2.12 Concept Front Hub](image)

The Hub design was started early with the concept shown in Figure 8.2.12, which had a weight saving over the previous design. This concept incorporated both the wheel mounting face and brake disc flange in one, which saves approximately 14mm of Hub shaft length. The pockets on the main bearing shaft were also a concept of reducing weight of the part, but were not carried forward after talking with the Brunel Racing Machinist [82] and with the need to mount a wheel speed trigger disc in this location. This concept passed the Bump and Braking load cases which were applied to it, but failed under the lateral load case. To
solve this, academic advice was sought as it was initially believed to be a Finite Element modelling related problem however, it was discovered that it was a part problem and a solution needed to be thought of. Table 8.5 shows a summary of the front and rear hub safety factors.

<table>
<thead>
<tr>
<th>Force</th>
<th>Component</th>
<th>Description</th>
<th>Front (Finalised)</th>
<th>Front (BR-XV)</th>
<th>Rear (Finalised)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Max Equiv. Stress (MPa)</td>
<td>Safety Factor</td>
<td>Max Equiv. Stress (MPa)</td>
<td>Safety Factor</td>
</tr>
<tr>
<td>Normal (z)</td>
<td>Full Bump - Upwards</td>
<td>159</td>
<td>2.96</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Normal (z)</td>
<td>Full Bump - Downwards</td>
<td>149</td>
<td>3.14</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Longitudinal (x)</td>
<td>Full Brake – Forward (with drivetrain shock-load for rear)</td>
<td>243</td>
<td>1.93</td>
<td>236</td>
<td>1.99</td>
</tr>
<tr>
<td>Longitudinal (x)</td>
<td>Full Brake – Reverse (with drivetrain shock-load for rear)</td>
<td>261</td>
<td>1.80</td>
<td>259</td>
<td>1.82</td>
</tr>
<tr>
<td>Lateral (y)</td>
<td>Full Corner – Inside Wheel</td>
<td>360</td>
<td>1.31</td>
<td>459</td>
<td>1.023</td>
</tr>
<tr>
<td>Lateral (y)</td>
<td>Full Corner – Outside Wheel</td>
<td>253</td>
<td>1.86</td>
<td>430</td>
<td>1.09</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Property</th>
<th>Front (Finalised)</th>
<th>Front (BR-XV)</th>
<th>Rear (Finalised)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass (kg)</td>
<td>299.06</td>
<td>607.31</td>
<td>570.58</td>
</tr>
<tr>
<td>Maximum ‘y’ Deflection (mm)</td>
<td>0.347</td>
<td>0.0861</td>
<td>0.193</td>
</tr>
</tbody>
</table>

Table 8.5 Table of Front and Rear Hub Safety Factors [80]
8.2.4.2 Uprights

It was decided this year to redesign the uprights, given that the same design has been used for the past 3 years. To redesign the upright, a packaging exercise was first done, knowing the position of the wishbone points, hub, wheel, and brake calliper.

Figure 8.2.13 shows a view of the initial front upright with all the required cutaways and it was this that was used as a design domain for the Topology to be conducted on. In order to create this domain, a cylinder was created to represent the inner volume of the wheel, and cuts were made for the wishbones, clevises, wheel bearings and hub bore and the brake calliper.

When designing this year’s upright, I wanted to reduce the number of clevises on the outboard, so on the front axle, the upper wishbone point is incorporated into the upright,
with the lower wishbone and steering point having clevises. On the rear, the upper wishbone is the only clevis. With this design, the number of outboard clevises was halved. By only having a clevis on the rear upper point, the upright had to be designed for negative Camber, as it would not be possible to shim the upright to have negative camber by only using the upper wishbone clevis. The rear uprights were designed to -4 degrees of camber to account for this, meaning that we could then shim the upright to become more positive.

Using SolidThinking Inspire as Topology software, the wishbone loads were loaded into the model shown in Figure 8.2.14 and the final result is shown in Figure 8.2.14 for the front. After the part had been redrawn from the topology results, they were re-analysed using another piece of software, namely Ansys, to ensure the Topology calculations were correct. The Final uprights have a more “planar” design to them than when compared to previous
years and this is due to the re-design and the manufacturing methods we chose for the part in Topology.

8.2.5 Brake Bobbins

With the redesign of the hubs, the brake bobbins could also be redesigned. By using the maximum deceleration of the vehicle on one of the front wheels from the load case in Table 8.4, where the bobbins have a smaller PCD, it was calculated that the bobbins took 21.16kN.

The thickness of the brake disc is 4mm and the bobbins have a diameter of 12mm where they contact the brake disc resulting in an area of 48mm$^2$ each. Using 6082-T6 aluminium for these bobbins, they have a safety factor of 2.11 with 74MPa of stress. By adding a 5mm hole into the centre of the part to remove weight, the bobbins have a stress of 126MPa, meaning a safety factor of 1.23 for the same material. For this application and calculation, a safety factor of 1.2 is satisfactory as it is for 6 of the front bobbins only decelerating the vehicle at 2g.

8.2.6 Wishbones

Using the geometric coordinates of the suspension geometry, and the desired vehicle performance, the wishbone forces were resolved for the maximum acceleration every 10degrees around the Centre of gravity in the Longitudinal-Lateral Plane using Equation 8.2.5, Equation 8.2.6, Equation 8.2.7 & Equation 8.2.8 where 0degrees is in the positive longitudinal direction.

$$F_x = \left(\frac{F_z}{mg}\right)(agM)\cos(\alpha_{acceleration})$$

*Equation 8.2.5: Calculating $F_x$*
\[ F_y = \left( \frac{F_z}{m} \right) (agM) \sin(\alpha_{\text{acceleration}}) \]

*Equation 8.2.6: Calculating \(F_y\)*

\[ F_z = \left( \frac{Mag h \cos(\alpha_{\text{acceleration}})}{WB} \right) + F_{FA} + \left( \frac{aF_{FA} H \sin(\alpha_{\text{acceleration}})}{T_{front}} \right) \]

*Equation 8.2.7: Calculating \(F_z\)*

\[ F_p = \frac{F_z}{\sin(\text{Inclination}_{\text{Pull Rod}})} \]

*Equation 8.2.8: Calculating \(F_p\)*

Figure 8.2.15 shows a plot of the absolute force in each Front wishbone member through the acceleration sweep. It shows that most of the force is received by the lower wishbone
members, meaning these will have to be stronger than the upper wishbone.

By using a table of the T45 steel tubes available from our supplier, the calculated lengths of the wishbone members and the maximum force in each member, the specifications for each member were calculated.

*Figure 8.2.15 Plot of Wishbone member force with angle of the acceleration applied*
Table 8.6 shows a list of the calculated tube with the respective maximum load and safety factor. As the table shows, the minimum safety factor is 1.5 on the Front Lower Aft wishbone member at an acceleration of 2.4g. Table 8.6 also shows that the wishbones will only comprise of 3 tubes;

- 1/2” 22swg (12.7mm x 11.2776mm)
- 5/8” 14swg (15.875mm x 11.811mm)
- 3/4” 18swg (19.05mm x 16.6116mm)

### 8.2.6.1 CFRP Wishbones

After calculating the dimensions of the steel wishbones, the respective CFRP wishbones could be investigated using the same loads. This task was undertaken by Rafal Biesadecki, a Level 3 student in the Unsprung department. Following previous investigations into the implementation of CFRP wishbones, this project went in a slightly different direction. After investigating Carbon tube wishbones bonded to aluminium inserts [83], the idea of CFRP wrapped inserts was investigated and Figure 8.2.16 [84] shows the concept behind this wrapping.
The aluminium is wrapped in carbon fibre so that the resin interacts with both the carbon and the aluminium, bonding them together. The CFRP wraps around the entire external faces of the aluminium insert to increase contact area and reduce the risk of the insert exiting the wishbone. Rohacell is a type of stiff foam that is there to increase compressive strength of the wishbone and reduce their likelihood to collapse under compression [84].

The wishbones were drawn in CAD using the suspension geometry that was designed and the resultant for the Front upper wishbone is shown in Figure 8.2.17 [84]. The bores at each end of the wishbone are designed to fit the staked bearings for this wishbone, and the arms have been designed using hand calculations to work out the dimensions of the oval tube made of CFRP.
Sample pieces were made up and tested on a Load Applicator to verify the design of the wishbones. Two designs were tested:

- Aluminium ring wrapped in CFRP and of ovular dimensions with no Rohacell for a Tension test
- Aluminium ring wrapped in CFRP and of ovular dimensions with Rohacell Foam for a compressive test

Testing these parts both yielded failure results earlier than expected. During the tension test, the CFRP cracked on both the inside and outside of the area wrapping the aluminium ring. The compression test failed at the join between the CFRP, Rohacell and aluminium ring. These failures could be due to many reasons, however manufacture, and quality of the CFRP are likely causes [84].
8.2.7 Rear Wheel Steering

The FSAE rule book allowed rear wheel steering during the 2015 season, and the task of investigating the feasibility of this was performed by Level 3 student Troy Smith. This new rule, T6 5.5 [37], states that the rear wheels can be electronically controlled but must have a maximum of 6 degrees of toe change.

Matlab scripts were developed to calculate the steering geometry of the car with and without rear wheel steer, and if the rear wheels were steering in or out of phase with the front wheels. [85]

The benefit of rear wheel steering, is the reduced steering effort required by the front wheels and the tighter corner radius. If this is taken into consideration for the FSAE endurance event, a driver can go faster, for longer, with the same effort.

![Plot of the required Front Steer angle for 2 and 4 Wheel steering Formula Student Cars](image)
Figure 8.2.18 [85] was drawn from the 2 and 4 wheel steer models and shows the sum of the front toe angles required for navigating a corner of set radius. As the graph shows, there is a difference in curve, and the required front steer angle for a given radius. By comparing 2 corners on this graph, 2m radius and 20m radius, there is a 37% and 55% reduction in front steering angle respectively, meaning a reduced force required by the driver which results in less driver fatigue and increased and quicker vehicle response on corner entry.

8.2.8 Bearings

8.2.8.1 Wheel Bearings

After receiving sponsorship from Schaffler, we also received access to their online Bearing Calculator which allowed for some Wheel Bearing Life Cycle Simulation and verification. By inputting the loading data that was used to design the hubs and uprights, a load case was generated.

After looking through the available and potentially suitable products, the Bearing Calculator was used to analyse and approve the selected bearings. By inputting the basic shaft geometry and the initial bearing selection, it was possible to set-up the calculation. The basis behind the calculation is to use the basic dynamic and static radial load ratings of the bearings (CR, COR) along with the shaft and load properties to calculate the life of the bearings using Equation 8.2.9 and Equation 8.2.10.

\[ L_{10} = \left( \frac{C_r}{P} \right)^p \]  
*Equation 8.2.9 Bearing Life calculation*

\[ L_{10h} = \left( \frac{10^6}{60n} \right) \left( \frac{C_r}{P} \right)^p \]  
*Equation 8.2.10 bearing life calculation in hours*
Equation 8.2.9 shows the Basic Life Rating equation, measured in millions of revolutions, calculates the number of revolutions before a failure occurs. [86]. Of course there are other variables which can influence the life of a bearing which this calculation does not account for.

In comparison, Equation 8.2.10 calculates the Basic Life rating in Life hours, using the average speed that the bearing rotates during its life. This is an estimate of the running time that the bearing can manage without incurring a failure. Both Equation 8.2.9 and Equation 8.2.10 have a variable $p$, where $p$ is equal to 3 for ball bearings, and $10/3$ for roller bearings [86].

The inputs into the calculation for the front wheel bearings were:

- Hub dimensions of 60mm Outer diameter with 48mm inner diameter and of length 50mm. Wheel mounting face of length 20mm and diameter 100mm. Material properties of Aluminium were applied to the shaft.
- Wheel bearings were added to the model at 5mm and 45mm from the inboard side of the Hub. These positions were chosen as the wheel bearing being analysed were 10mm wide and the positions mark the centre of the bearing. The bearings analysed were angular contact bearings and it was ensured that the bearings were opposing each other and the correct orientation, with the thicker side of the inner bearing race facing the external of the shaft. SKF bearing grease was added to the simulation with viscosity of 500mm$^2$/s and 32mm$^2$/s for 40 and 100 degrees Celsius respectively [87]
- The shaft was set to a speed of 840rpm (45mph), the average speed around an endurance course and most test days. The load cases were then applied individually to the shaft.

After finishing the calculation, it was found that the front wheel bearings, which are under more stress than the rear, had a Basic Life Rating of 199 hours in the bump load case and 200 hours in the lateral load case. With the lifetime of the vehicle being 2 competitions and approximately 8 test days lasting at most 10 hours each, this meant that the wheel bearings will not fail, under the loading conditions entered into the calculation. The selected wheel bearings for BR-16 were angular contact bearings with product numbers; Schaeffler 71812-B-TVH, 71816-B-TVH, for the front and rear axles respectively.

After running the analysis for BR-16, I re-ran the calculation with the front hub and wheel bearings (FAG deep groove 61912) used on BR-XV and their Basic Life rating was 99 hours.

8.2.8.2 Wishbone Bearings

Using the Wishbone Load Calculator, the loads in all wishbone members were calculated and plotted in the aforementioned graph, Figure 8.2.15. Using the maximum force at each point from these calculations, and the suspension geometry, the bearings could be selected from both a loading and packaging point of view.

Preferably, all bearings would be staked over rod-ends to reduce the amount of compliance, weight and number of components. To narrow the search further, only bearings with a bore of 3/16” or 1/4” would be used to reduce the overall size of the bearing housing allowing bearings to be mounted as close as possible to the mounting face and reduce excess stress and bending in the mounting clevises. With the above search criteria, Table 8.7 details the selected bearing for each position;
The wishbone bearings are such that the Front Upper wishbone is the only wishbone where all bearings are staked into the wishbone. All other wishbones on the car will require 1 or 2 rod-ends. The decision for rod-ends was led by packaging, however all outboard bearings are staked. The staked bearings will be mounted in steel plates similar to that shown in Figure 8.2.19 and the rod-ends will be mounted similar to that shown in Figure 8.2.20.

The size of these plates for the staked bearings was calculated using the data shown in Figure 8.2.21. As the thickness of the plates was dictated by the bearings themselves, the diameter was the only variable, and is shown in purple for the 1/4” bore bearing (left) and the 3/16” bore bearing (right).

<table>
<thead>
<tr>
<th>Bearing</th>
<th>Company</th>
<th>Application</th>
<th>thread-dg</th>
<th>thread</th>
<th>Ultimate Load (bearing)</th>
<th>Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>ANC-3TG</td>
<td>Aurora</td>
<td>wishbones</td>
<td>N/A</td>
<td>N/A</td>
<td>17681</td>
<td>FUF, FUA, FOU, RUAF, RUO</td>
</tr>
<tr>
<td>ANC-4TG</td>
<td>Aurora</td>
<td>wishbones</td>
<td>N/A</td>
<td>N/A</td>
<td>26867</td>
<td>RLA, FLO, LRO</td>
</tr>
<tr>
<td>ASM-3T</td>
<td>Aurora</td>
<td>wishbones</td>
<td>RH</td>
<td>5/16-24</td>
<td>10497</td>
<td>RUF, RLF</td>
</tr>
<tr>
<td>XAM-3T</td>
<td>Aurora</td>
<td>wishbones</td>
<td>RH</td>
<td>1/4 - 28</td>
<td>23397</td>
<td>RAB, RAB</td>
</tr>
<tr>
<td>XAM-4T</td>
<td>Aurora</td>
<td>wishbones</td>
<td>RH</td>
<td>1/4 - 28</td>
<td>37596</td>
<td>FLN, FLO, VTO, RTO</td>
</tr>
<tr>
<td>XAB-1T</td>
<td>Aurora</td>
<td>wishbones</td>
<td>LH</td>
<td>1/4 - 28</td>
<td>37596</td>
<td>FTI, RTI</td>
</tr>
<tr>
<td>GMM-3M-470</td>
<td>Aurora</td>
<td>pull rods</td>
<td>LH</td>
<td>1/4 - 28</td>
<td>12815</td>
<td>FP, F3</td>
</tr>
<tr>
<td>GMM-3M-470</td>
<td>Aurora</td>
<td>pull rods</td>
<td>RH</td>
<td>1/4 - 28</td>
<td>12815</td>
<td>FP, FARB, F3</td>
</tr>
<tr>
<td>GMM-3M-470</td>
<td>Aurora</td>
<td>ARB</td>
<td>RH</td>
<td>1/4 - 28</td>
<td>12815</td>
<td>FARB</td>
</tr>
<tr>
<td>ASM-4T</td>
<td>Aurora</td>
<td>push rods</td>
<td>RH</td>
<td>1/4 - 28</td>
<td>10000</td>
<td>RP</td>
</tr>
<tr>
<td>ASE-4T</td>
<td>Aurora</td>
<td>push rods</td>
<td>LH</td>
<td>1/4 - 28</td>
<td>10000</td>
<td>RP</td>
</tr>
<tr>
<td>17812-B-TVH</td>
<td>Schaefer</td>
<td>wheel bearing</td>
<td>N/A</td>
<td>N/A</td>
<td>not available</td>
<td>front</td>
</tr>
<tr>
<td>17815-B-TVH</td>
<td>Schaefer</td>
<td>wheel bearing</td>
<td>N/A</td>
<td>N/A</td>
<td>not available</td>
<td>rear</td>
</tr>
<tr>
<td>1602-ZZ</td>
<td>Simply Bearings</td>
<td>Rocker Bearing</td>
<td>N/A</td>
<td>N/A</td>
<td>not available</td>
<td>Front and Rear</td>
</tr>
</tbody>
</table>

Table 8.7: Table listing the specified bearings for the Suspension and each bearings location

Figure 8.2.19 View of the Staked Bearing Housings to be used for the wishbones

Figure 8.2.20 View of the Rod-End housing for the wishbones, complete with "flats" on the wishbone insert for tightening the Jam
8.2.8.3 Rocker Bearings

Knowing that the rocker bearings in BR-14 lasted 1 race weekend [42], and the Bearings in BR-XV lasted a year, it was decided to continue with these bearings, however, some life-cycle calculations were done to check their suitability. Using the equations above in Section 8.2.8.1, Equation 8.2.9 and Equation 8.2.10, the suitability of the proposed bearings was calculated.
By adding the equation in Figure 8.2.22 [88], the corresponding values in Figure 8.2.22, and given the data for the 1602-ZZ deep groove roller bearing (Cr = 635, Cor = 240 [89]), basic life rating calculations resulted in this bearing being suitable for our application this year.

8.2.9 Wheels

The wheels this year were initially intended to be manufactured by Force Racing. Force Racing provide a 3 Piece 10” wheel rim with varying sizes of inner and outer wheel shell sizes, which then bolt together with the wheel centre to complete the wheel assembly. Figure 8.2.23 [90] shows a schematic of a 3 piece wheel and how the 3 pieces interact. There are inner and outer sections which are bolted together with a centre section.

![Assembly view of a 3 Piece wheel](image)

After performing the first packaging checks, the wheels needed to have a 5” inner shell and a 2” outer shell. After contacting Force Racing, it was found that the only 5” shell they provide is meant for the outer shell and thus has a smaller internal diameter compared to the inner shells and would be too small to package the upright and brake calliper. Keizer
Precision were then contacted to receive some dimensions as they also manufacture 3 piece 10” wheel shells and provide more variants, including the shell sizes that we required.

The Keizer wheels were ordered with a 5” inner and 2” outer shell, and a 12 bolt mounting pattern, coming in at 1.8kg per wheel with mounting bolts and excluding the wheel centre. Figure 8.2.24 shows the new Keizer shell on the right, next to last year’s OZ Racing Wheel. The proposed wheel centre takes the wheel weight up to approximately 2.5kg which is 33% lighter than the 13” wheels we have used in the past (Braid 13”= 3.779kg, OZ = 3.6kg).

8.2.10 Actuation Components & Anti-Roll Bars

The actuation sub-section of Unsprung includes all components which link the outboard components to the chassis which control the sprung and unsprung motions and motion ratios. In order for the designed motion ratio to be as intended, all linking components must be able to withstand the load taken by the suspension and not deform or fail.
8.2.10.1 Rear Rockers

Once the suspension geometry had been frozen, the rocker design could be started. To start with, the full heave coordinates were exported from the kinematic simulation and imported into a spreadsheet to resolve and calculate the loading data for each point and the load cases. 2 Load cases were analysed, single wheel heave and axle heave.

The rockers were loaded with remote forces acting on their respective mounting holes and acting in the centre of the 2 rocker plates, with a frictionless support on the bearing housing and a fixed support where the damper mounted. The rocker was then meshed using a body sizing method with 2mm element sizing.

After several iterations, the final rocker and equivalent stress plot is shown in Figure 8.2.25 for the Bump load case. For the material selected for the rocker plates, 6082-T6, maximum Stress here equates to a safety factor of 3.5. The thickness of the plates is 4mm with all holes being 8mm to allow for the damper bearing and spacers for the smaller bearings.
Figure 8.2.26 shows the final rocker design with inserted spacers for the actuation linkages.

8.2.10.2 Front Rockers
The 2 load cases for the front rocker were also calculated using the full heave coordinates of the kinematic simulation and resolved to generate an input for Stress analysis. With the need for spacers between the bearings for the actuation and the rocker plates, all the holes in the plate were made to 8mm before analysing the part for stress.
Figure 8.2.27 shows a plot of the equivalent stress in the part for the bump load case. Like the rear rocker plates, the material is 6082-T6 aluminium and of 4mm plate thickness with a safety factor of 1.86.

8.2.10.3 3rd Spring Rocker

Having a 3rd Spring on the front axle, with pull rod suspension, requires another rocker to translate the tension motion to compression. The distance between the pivot point and the linkage points are similar and short in comparison to the other rockers, which results in high moments. The linkage arms had to be small to ensure the 3rd spring could be packaged under the car, as Figure 8.2.28 shows. With this, there was very little material that could have been taken out of the rocker to save weight.
8.2.10.4 Pull-Rod Load Calculations

Stress calculations were performed on the pull rods to determine what size pull rod is required. As the length of the pull rod is already defined by the suspension geometry which left the diameter and material to be specified for the application.

\[ r = \sqrt{\frac{P}{\sigma_{\text{Tensile}} \pi}} \]

Equation 8.2.11

Equation 8.2.11 [91] was used to calculate the required diameter bar for the maximum Normal Load condition as specified in Table 8.4. The required radius, using a titanium pull-rod, was calculated to be 2.38mm.

The bearings required for the pull-rods are female rod-ends and have an internal thread of 1/4” (6.35mm), see Table 8.7 for FP. This means that the radius of the bar increases to 3.17mm and increases the safety factor to 1.73.
8.2.10.5 Ride frequency and Spring Selection

Selecting spring stiffness for a vehicle will determine how the vehicle handles and responds to road inputs. Spring stiffness’ are commonly selected from calculating the ride frequency of a quarter car model, using Equation 8.2.12.

\[ K_s = 4\pi^2 f_r^2 M_{\text{sprung}} M R^2 \]  

Equation 8.2.12 Spring Stiffness

By using the above equation, Table 8.8 was created; showing the required spring stiffness for each vehicle corner sprung mass and ride frequency. The green cells correspond to Ohlins and Fox springs which we already have and the red cells correspond to the available range of springs from H&R.

<table>
<thead>
<tr>
<th>Ride frequency (Hz)</th>
<th>Sprung Mass (kg)</th>
<th>Spring Stiffness (N/m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.0</td>
<td>30</td>
<td>2.5</td>
</tr>
<tr>
<td>2.5</td>
<td>30</td>
<td>3.0</td>
</tr>
<tr>
<td>3.0</td>
<td>30</td>
<td>3.5</td>
</tr>
<tr>
<td>3.5</td>
<td>30</td>
<td>4.0</td>
</tr>
<tr>
<td>4.0</td>
<td>30</td>
<td>4.5</td>
</tr>
<tr>
<td>4.5</td>
<td>30</td>
<td>5.0</td>
</tr>
</tbody>
</table>

Table 8.8 Table of spring rate for varying Sprung Mass and Ride Frequency

When selecting springs for a car, the front and rear stiffness' should be different so that when the vehicle navigates a speed bump for example, the front axle responds, and then the rear responds slightly after, but they will respond at the same rate and will likely cause a pitching oscillation about the vehicles Centre of Gravity. By having different spring stiffness’ front and rear, this pitching oscillation can be eradicated.
8.2.10.6 Anti-Roll Bar

Anti-roll bars link 2 sides of an axle such that in roll where each side of the axle move in opposite directions of heave, the ARB restricts the difference in motion and increases the cars roll stiffness. The ARB links the two wheels by a tube which must be twisted for the vehicle to experience roll, but has no effect on the vehicle during heave. The ARB can offer up to 10 times the roll resistance of the corner springs and is such a powerful tuning instrument because of this attribute. [92]

\[ K \varphi_{des} = \frac{WH}{\varphi_{des}} \]

*Equation 8.2.13 Desired Roll Gradient*

\[ K \varphi_A = \left( \frac{\pi}{180} \right) \left( K \varphi_{des} K_t \frac{t^2}{2} \right) - \left( \frac{\pi K_w}{180} \left( \frac{t^2}{2} \right) \right) \]

*Equation 8.2.14 Roll Stiffness*

\[ K \varphi_{FA} = \frac{K \varphi_A N_{magic}(MR_{FARB}^2)}{100} \]

*Equation 8.2.15 Front ARB Roll Stiffness*

\[ K \varphi_{RA} = \frac{K \varphi_A (100 - N_{magic})(MR_{FARB}^2)}{100} \]

*Equation 8.2.16 Rear ARB roll Stiffness*

By using Equation 8.2.13, Equation 8.2.14, Equation 8.2.15 and Equation 8.2.16, I was able to create the graph in Figure 8.2.29 which shows the roll stiffness of the front and rear axles required to achieve a specified roll gradient. Trend lines for both axles are also plotted and correlate with the data points after a roll gradient of 0.9.
By using the Equation for calculating the Front Roll Stiffness, Equation 8.2.15, and using Equation 8.2.17, the Moment of Area of the ARB system was established.

\[
d = \sqrt[4]{K_{φ_A} \left( \frac{L_{tube} L_{ARB}^2}{G} \right) \left( \frac{2 \times 180}{2\pi t_f M R_{FARB_H}} \right) \left( \frac{N_{magc} M R_{FARB_R}}{100} \right)}
\]

Equation 8.2.17 ARB Moment of Area

Figure 8.2.30 shows a table of the required moment of area for 2 different lengths of ARB Arms.
The table has 2 variables, roll gradient (columns) and Magic Number (rows). For a particular magic number which will be vehicle dependant, the varying tubes will allow the roll gradient to be adjusted. The colours trending diagonally in the table correspond to 6 different tube dimensions, each of the same length, but with different moments of area.

8.2.10.8 Rear Anti-Roll Bar

With limited packaging on the rear, and the rear of the car likely to be above 50degrees Celsius when running, it is not a place where replacing ARB tubes would be a pleasant job.

\[
L_{ARB} = \sqrt{\frac{1}{MR_{RARB}} \left( \frac{180}{\pi t^2} \right) \left( K_{\varphi A} \left( 100 - N_{magic} \right) \left( MR_{RARB}^2 \right) \right)}
\]

Equation 8.2.18 ARB arm Length

At the rear, it was decided to make an arm with various holes in, relating to different length arms.
Instead of creating a table like Figure 8.2.30 where it calculated a moment of area based on length, roll gradient and magic number, Equation 8.2.17 was rearranged so that the equation would calculate Arm Length (Equation 8.2.18) for a specified tube, magic number and roll gradient. This produced Figure 8.2.31, where the coloured cells represent the chosen arm lengths.

8.2.10.9 Front 3rd/Heave Spring Design

When designing the front 3rd spring, the major difficulty was finding a product that would fit. The search began with looking at mountain bike dampers and one was found. This damper was purchased, benchmarked, and used as basis for the design of the new front 3rd spring.
Figure 8.2.32 shows the front 3rd spring design. The 3rd spring is designed to be at maximum extension at full wheel droop, and compress to an adjustable point depending on how much heave we want the vehicle to achieve. The required travel is adjustable by undoing the nylon nut, and inserting nylon cylinders into the compression chamber, thus reducing the amount of travel permitted.

8.2.11 Damper Dynamometer – Physical Testing

After running the Ohlins TTX-25 Dampers for 2 years, they were taken to a damper dynamometer at Millbrook Proving Ground to check the wear on each damper and to compare the damping coefficients between dampers. After analysing the data from this test, it was found that there was a distinct difference between all 4 dampers on the same setting. This is not ideal and means that the dampers will require refurbishment and re-valving before using them at competition.

The most common combination of damper settings used on the last 2 cars has been 6-3, and Figure 8.2.33 and Figure 8.2.34 show the damper plots at this setting for Bump and Rebound.
respectively. It is shown in Figure 8.2.33 that one damper has a higher coefficient of damping than the other 3 in Bump, but in Figure 8.2.34, a different damper has a lower damping coefficient than the other 3 in Rebound.

![Figure 8.2.34 Plot of Rebound Force against Rebound velocity for all 4 Ohlins dampers on a setting of 6-3 (6low speed clicks, 3 high speed)](image)

Different damping coefficients on the same axle and make predictably improving vehicle handling complicated as the ratio of damping ratio to the settings is different on each damper and so the actual coefficient cannot be determined on track easily.
8.3 Chassis (GG)

Much of the design of the chassis is driven by regulations. Typically a legal structure may be surplus to the design requirements from a performance perspective; therefore a structure which was marginally compliant with the regulations but of the desired overall package was first specified for analysis. Subsequently further work would be undertaken where required to improve the stiffness of the vehicle from the minimum legal baseline design.

8.3.1 Minimum Legal Panel Specification

With the skins being comparatively heavy in relation to the honeycomb core materials, previous authors [31] [93] [94] have discussed how the limiting factor on the panel in terms of improving specific-stiffness is the facing skin’s performance in the mandated perimeter shear test. Using 6082-T6 or 6061-T6 a minimum skin thickness of 0.7 or 0.8mm was required. By utilising an alloy of the same density but with higher shear strength this thickness can be reduced; increasing the performance in the perimeter shear test with less mass.

However the panel must have a specified bending stiffness (EI), this author has previously established [25] that a 32mm overall panel with a 0.4mm 7075-T6 facing skin would provide adequate bending stiffness and perimeter shear strength as per the FSAE regulations. Figure 8.3.1 demonstrates how a thicker laminate can benefit from the specific stiffness increases in much the same was as I-beams function.
8.3.2 Working with the suspension geometry

Once initial suspension geometry was provided work was initially undertaken to ensure that a chassis could be constructed within the regulations. A number of iterative cycles were required from the suspension team to make small modifications to ensure that the geometry points allowed for the creation of a legal chassis.

Subsequently further iterations with much smaller incremental changes were made to aid manufacture or align points with certain nodal positions.

8.3.3 Minimisation of the Rear Frame

This author’s previous work has also demonstrated that by incorporating the engine into the rear frame structurally the stiffness of the vehicle can be dramatically improved [25].
A minimum legal rear frame was specified which, based upon the desired suspension geometry and engine position would satisfy the requirements of the FSAE regulations [37] with minimal mass.

8.3.4 Templates & Driver Cell

The FSAE regulations have a large number of regulations regarding the cockpit; firstly regulation T4.1 states that the template shown in Figure 8.3.2 must, placed horizontally, pass vertically down through the cockpit opening to a height 350mm above the ground.
Regulation T4.2 requires the template shown in Figure 8.3.3 must, placed vertically, pass from the front hoop, horizontally through the car to a position 100mm rearwards of the pedals.

The final major template which must fit within the vehicle is known as Percy (Figure 8.3.4); a representation of a 95th percentile male. Whilst demonstrates the required clearance between the roll hoops and the head of Percy.
8.3.5 Initial Design

Figure 1 shows the initial design of the vehicle incorporating the minimum steel tube properties from the regulations, proposed panel specifications and showing the cockpit templates.
8.3.6 FE Modelling of the chassis.

Following subsequent design iterations to permit satisfactory packaging of the various major components a model was constructed from an IGES export from SolidWorks in Altair HyperWorks.

8.3.6.1 Modelling of the Frame
As discussed in the critical analysis section, boundary conditions are very important to ensuring that representative deformation and stress is achieved under loading in FEA.

2D Shell Elements have been used in order to model the monocoque structure whilst 1D beam elements have been used to model the tubular portions of the frame. These elements allow for a computationally efficient solution to be reached with an acceptable level of accuracy.

The reality of the complexity with modelling a composite structure which involves significant amounts of bonding of multiple components to create a whole structure means that the assembly will always have to be over-engineered for the task in hand. Therefore the limited increased accuracy afforded by more computationally demanding modelling techniques has been deemed unnecessary.

8.3.6.2 Boundary Conditions
Real world conditions were studied to assess the suitability of the boundary conditions; the boundary conditions applied needed to not constrain the vehicle any more than it would in the real world, and be possible to replicate in a physical test.

As can be seen in Figure 8.3.7, selective use of degrees of freedom have been used to allow for joints to pivot in the same manner that the real car would.

Each suspension arm is modelled as a rigid link to allow rotation but not displacement at its ends, whilst the boundary conditions are indicated by the numbers on the 2 front wheels and at the rear. With 1,2,3,4,5,6 indicating constraint in the x,y,z displacement , and x,y,z rotation respectively. The model is therefore minimally constrained and closely representing the condition which is of interest in measuring torsional stiffness. The Jig for testing will also constrain the vehicle in the same manner.
8.3.7 Hard Points

Integral hard points are laminated into the panels for BR-16, this provides maximum load-bearing ability as both skins can be loaded by the fastener when compared to a distance tube.

The hard points have been specified by taking the expected peak load, determining (using a large FoS) the required shear area from the shear strength of the facing skin, then looking to remove additional mass but maintain webs large enough to avoid shearing the material around the fastener out of the insert through its thickness.

8.3.8 Structural Equivalency Spreadsheet

All teams entering an FSAE event must submit a Structural Equivalency Spreadsheet (SES) demonstrating that the vehicle’s structure is equivalent to a baseline steel tubular spaceframe via various metrics.

As already discussed in section 8.3.1 equivalent bending stiffness (EI) along with minimum shear strength must be achieved for the panel. Passing the SES has proven to be one of the biggest problems for BR-16 and led to a significantly compromised design in-order to modify the design to make it legal. Although the intended panel passed the known criteria outlined in section 8.3.1, a number of changes introduced in particularly for the 2015 season but also since BR last constructed a monocoque have rendered the initial design illegal.

8.3.8.1 Limiting factor

Due to the clean-sheet design approach being taken by the BR-16 team it was necessary to design large parts of the vehicle earlier than is normal within BR; in particular the suspension geometry and base chassis design were mostly completed within the first few
weeks of the academic year. Typically the FSAE regulations are published in late July to early August, the 2015 rules however were not released until late September, with the SES only being released in December. Many designs were pursued under the existing regulation framework which is sub-optimal in 2015, it is important to spend some time to highlight these areas to explain why the design has had to be compromised.

Previous authors have all found that the perimeter shear test outlined in section 2.6.6.2 to be the limiting factor for further mass reduction of the panels used in construction of the monocoques. This team-accrued knowledge was the starting point for the minimum legal panel which was proposed in section 8.3.1, and from which the entire design was based.

8.3.8.2 Test Derived UTS and Elastic Modulus Values
Since the last BR monocoque it has been declared that two of the material properties used in the SES to demonstrate equivalency (UTS and E) must be derived values from the 3 point bending test outlined in section 2.6.6.1, a fact which was unknown to the team. There is no reference to this change in any rule books or SES change-logs since the last BR monocoque, except for the explicit statement of this requirement in the 2015 regulations [1] which were released after the design had been frozen and materials ordered. It does however appear that this rule was enforced from 2013 onwards as Dan Jones (chief technical scrutineer and SES reviewer for FSUK) makes reference to the requirement of the test derived values in a book published shortly after BR’s last monocoque was constructed [95]. It seems likely that teams doing otherwise would have been informed when they submitted SES’s in 2013-onward.

Previously the SES featured a material data column called “Carbon” which the test derived data fed into (see Figure 8.3.8). As BR did not use CFRP, data sheet properties were simply
entered into one of the aluminium tabs and this material specified on the various areas, for 2015 this has been changed to “Composite 1” (see Figure 8.3.9) and the material choice is locked to this set of value on the various sheets within the SES where material is selected.

As will be explained in section 8.3.8.4, the test derived UTS value of 71MPa is vastly lower than the 540MPa UTS of the facing skin. Significantly lower UTS values mean that more cross-sectional area is required to have equivalent strength to the baseline steel chassis. As such, a metric which was previously considered a formality has now become the limiting factor on the design.

8.3.8.3 Side Impact Structure Changes

Another problem area for the original design was the redefining of the side impact structure. The 3-point bending test described in section 2.6.6.1 prescribes a given bending
stiffness requirement for the side impact zone. Previously this was allowed to be the full height of the side of the monocoque as is shown in Figure 8.3.10.

As such the optimised mass panel was designed to have sufficient bending stiffness for a panel of 370mm height; Figure 8.3.11 shows the updated definition of this region.

![Figure 8.3.11. 2015 SES Interpretation of the Side Impact Structure.](image)

The problem is further amplified by a design decision to have a step in the monocoque floor to allow for preferable suspension mounting locations as the side impact structure is defined as:

“Side Impact Zone – The area of the side of the car extending from the top of the floor to 350 mm (13.8 inches) above the ground and from the Front Hoop back to the Main Hoop.” [1]
As shown in Figure 8.3.12, the design features a front roll hoop which does not extend to the lowest point on the chassis; it is unclear from the above definition if only the height up to the stepped part of the floor can be used.

As a result of this ambiguity the area shown in Figure 8.3.13 has been used, meaning a panel of only 151mm in height is used in the calculations to prove equivalency. Ultimately this was not a major problem as the panel had to be reinforced significantly to achieve the required strength with the test derived UTS values.
8.3.8.4 Physical Testing

Through physical testing it was found that the panel’s structural performance was not completely as expected. Whilst the panel did satisfy the bending stiffness and perimeter shear strength requirements of the technical regulations the test derived UTS properties proved to be the major problem for the design. A test derived UTS value of around 210MPa would have been required for the panel to be deemed legal, although this was well below the datasheet value of 525MPa [96], it was around 3 times higher than what was ultimately achieved.

Further details of the test program can be found in the portfolio of evidence, but the primary issue concerns the stiffness of the skins in relation to the stiffness of the panel due to its large 2nd moment of area caused by the skin separation distance.

Figure 8.3.14. 3 Point Bending test of monocoque panel after additional reinforcement (1.2mm skin)
Figure 8.3.14 shows how the 3 point bending test fails to bend the panel in its centre but simply causes local core compressive failure and delamination of the skins from the core.

As the test is not yielding the skin, the values of UTS do not represent the actual UTS of the skin, however this test derived value is what must be used. Tests resulted in a UTS of around 71MPa.

Most interestingly was how when testing both a 0.4mm skin and the reinforced 1.2mm skin both panels reached their elastic limit at approximately 2.2mm of deflection, this backs up the hypothesis that the panel is simply too stiff to bend in the expected manner without failing through local core compression. In Figure 8.3.14 it can clearly be seen that only the centre section of the panel appears to be deforming with the end sections being undeformed, this kind of shape would be more in-line with that expected if the ends of the specimen had been fixed.

Figure 8.3.15 shows the perimeter shear test conducted, the value produced by this test was lower than expected and that found in previous work with a 0.4mm skin, however due to the requirement to reinforce the areas to which the test pertains, this poor result was irrelevant.
It is recommended that future work looks at increasing the size of the test sample as it was found bulking of the skin would occur from the edges inwards. This was also found in previous work by this author [25], and by previous submissions for this project [94].

8.3.9 Impact Attenuator

Development of a composite IA structure was the subject of a level 3 project within the chassis group this year, however insufficient progress was made for it to be used on the vehicle in 2015; as a result a more traditional basic foam structure was also designed. Due to changes to the wing mounting regulations, a lower (than typical) deceleration value was required.

The IA was tested at the FIA approved Cranfield Impact Centre on 13th and 20th March, with the 2nd test passing successfully (Figure 8.3.16). The first test however failed due to deformation of the crash rig.
Further details about the design of the IA and rig can be found in the portfolio of evidence.
8.4 Aerodynamics (GM)

8.4.1 Front wing

8.4.1.1 2D study

The design process initially focused on the investigation of various aerofoils for the front wing mainplane: this is the most important aerofoil of the aerodynamics package and it is the one that predictably will produce the most downforce due to its size and by being in ground effect.

Initially, an investigation on the effect of increasing camber and camber position along the chord was carried out using Javafoil on the NACA 4-digit family of aerofoils. All the aerofoils used for these simulations featured an angle of attack (\(\alpha\)) of 1 degree and were placed at a ground clearance of 12% of the chord length (from the lowest point of the aerofoil).

![Cl vs Camber position for aerofoils of different camber (Javafoil)](image)

*Figure 8.4.1: Lift coefficient vs. camber position for aerofoils of different camber.*

Figure 8.4.1 shows how the increase in camber and camber position results in an increase of coefficient of negative lift. The increase in downforce level due to the increased camber was expected because of the theoretical meaning of aerofoil camber which can be seen as
additional angle of attack. [35] The increase in the level of negative lift coefficient due to increased camber position depends only on the fact that with a locus of maximum camber located close to the trailing edge the ground effect is maximised. In fact most, of the suction side of the aerofoil is in close proximity to the ground (the suction side is relatively flat and parallel to the ground), whilst with a locus of maximum camber positioned at, for example 50%, a huge portion of the wing is diverging from the ground.

The comparison between the velocity ratio of the flow around NACA 8412 (maximum camber position at 40% of the chord) and NACA 8812 (maximum camber position at 80% of the chord length) can be seen below.

![Flow Field](image)

*Figure 8.4.2: Contour of velocity ratio for NACA 8412*
It can be seen that the NACA 8812 which presents a “flatter” suction side produces an area of higher velocity much larger compared to the NACA 8412 aerofoil. This larger area of high speed fluid corresponds to a larger area of low pressure area and hence more net downward force.

8.4.1.2 OpenFoam

Even though Javafoil is a reliable software package for aerofoil comparison, the same study was repeated with the Open Source CFD software OpenFoam for validation. With 2D OpenFoam simulations the effects of turbulence (even if in a limited manner compared to 3D simulations) have a direct influence on the outcome to the result. The results are presented in Figure 8.4.4.
Even though the force is presented as positive in the graph, it is worth saying that it represents negative lift (downforce). If a comparison is made between this graph and the one with the results obtained in JavaFoil (Figure 8.4.1), it can be verified that the results are almost identical. Thanks to this preliminary study on the aerofoils’ parameters, it is possible now to restrict the selection of the mainplane characteristics around aerofoils that have similar features: high camber and maximum camber location close to the trailing edge.

A secondary 2D CFD study in OpenFoam was conducted on the NACA 4-digit family of aerofoils to identify the locus of optimal ride height. As previously discussed in section 0, the interaction between the ground and a body moving in its proximity helps in achieving higher downforce values due to the Venturi effect. However some considerations have to be made: if a wing (or an undertray) is too close to the ground the opposite effect is witnessed and values of lift decrease. This is due to the fact that running too little ground clearance forces the air to flow in a gap that is too small. In such area the effects of viscous forces are dominant and they limit the flow velocity. The table below shows values of downforce as a function of maximum camber position and ride height:
It can be seen that downforce levels (expressed in Newton per 100mm of span) are higher for most aerofoils when their lowest point is at a distance from the ground between 8% (H/C of 0.0833) and 12% (H/C of 0.1166) of the chord length. (H is the distance of the lowest point of the aerofoil from the ground and C is the chord length).

Subsequently, online aerofoil databases were investigated and a number of aerofoils matching the already discussed criteria (camber and camber position) were selected and tested in CFD. (see portfolio of evidence)

At the end of the study that featured almost 500 simulations in OpenFoam, a set of 6 aerofoils (that proved to perform the best) were selected and will be used successively for the study on wing flaps. Results are presented below:
8.4.1.3 Flap positioning

The importance of flaps in an automotive wing is discussed in section Error! Reference source not found..

The purpose of this CFD 2D study is to identify the optimal configuration between front wing mainplane and front wing flap. The mainplanes parameters (e.g., angle of attack and ground clearance) remained unchanged. The flaps’ parameters investigated during this series of simulations were chord length, flap angle of attack, and the vertical and horizontal gap between the flap leading and the mainplane trailing edge.

In aerodynamics, there are some guidelines in order to choose these parameters correctly (see section Error! Reference source not found.). However, due to the space limitations dictated by the 2015 rule book it is worth investigating multiple solutions. For example, is it

![Graph showing Downforce vs. Ride Height for 6 different profiles in OpenFoam.](image)

*Figure 8.4.6: Downforce vs. ride height for 6 different profiles in OpenFoam.*
worth having a smaller VGAP (compared to the theoretical ideal VGaP), in order to allow a longer flap chord or greater flap angle of attack?

![Figure 8.4.7: Explanation of Vgap and Hgap.](image)

Error! Reference source not found. is a representation of what Vgap and Hgap physically mean. The red arrows indicate the design space limitations imposed by the rules (x=625mm and y=250mm) (see section Error! Reference source not found.).

Again around 500 coordinate files were created in Excel and converted to solid models in SolidWorks, and simulated in OpenFoam. Simulations showed that for this application running a flap with reduced chord length and higher angle of attack generates higher levels of downforce. The ideal values of HGAP from CFD simulations seems to match theoretical indications given from McBeath [34]. The ideal region of value of VGAP was instead slightly higher than the theory (4-5% compared to 2-3%).
During the course of this design phase, interesting conclusions about the mainplane were also made. Initially the simulations were performed using a NACA 8812 mainplane (which performed the best in the preliminary study); however it did not perform ideally with a flap due to significant flow separation.

The camber of the main plane was reduced to 6% and the locus of maximum camber was changed from eight tenths of the chord to seven tenths (NACA 6712), and downforce values increased due to the improved flow attachment.

8.4.2 3D Design

Once the two-dimensional study was completed a 3D model of the front wing was created using Solidworks. As previously mentioned an ideal ride height location was identified for most of the aerofoils analysed. However for this application such ride height cannot be achieved due to the pitch, squat and roll of the vehicle. Excessive suspension’s movement (minimum suspension travel allowed is 50mm), especially when the pitch and roll
phenomena are combined, could cause the front wings mainplane to be in contact with the ground, possibly resulting in component failure and/or disqualification from an event. For this reason the NACA 6712 profile was modified across its span to maintain an ideal ride height of 60mm at the car centreline and an increased ride height of 80mm on the extremities of the wing span (Figure 8.4.9).

A 3D analysis in CFD showed that this solution is advantageous. With the adoption of the “curved” front wing mainplane, the downforce generated by the front wing was increased by 20N. A comparison of pressure distribution across the two different mainplanes is presented below:
The pressure below the airfoil has decreased by almost 60% in the middle of the span. This proves that the ideal ride height that was previously calculated in two-dimensions still applies in 3D.
Due to the general increase in ride height of the main plane the flap characteristics selected in section Error! Reference source not found. were modified in order to maintain the design into its legal limits (250mm rule). The guidelines of VGAP discovered in 2D were used again to select the new configuration. The Selig 1223 profile (high lift for low Reynolds number) was selected for this application. The final front wing aerofoil configuration can be visualised

![Flow over front wing](image1)

*Figure 8.4.12 Flow over front wing*

Simulations showed that very little flow separation is present making this design solution very effective:

![Pressure distribution and Flow over 2D](image2)

*Figure 8.4.13 Pressure distribution and Flow over 2D*
A uniform low pressure area in the range of -90 to 140Pa is located across the chord length on the suction side of the front. The velocity of the fluid is accelerated from the aerofoil in ground effect from a speed of 11.1m/s (inlet simulation velocity) to a speed of almost 18m/s.

It has to be pointed out that the single flap is run at a reduced angle of attack (25 degrees relative to the mainplane chord line) compared to the optimum value identified in 2D simulations (35 degrees). This is due to the influence of the front wing on the flow over the side wings. This is also the reason why the original design of the front which featured three aerofoils (a main plane and two flaps) was discarded.

8.4.3 End plates

As mentioned in the literature review a good end plate design is essential for increasing the effectiveness of an automotive wing. Its purpose is to stop the high pressurised air on the top and on the side of an aerofoil flowing into the low pressure area of the suction side and hence affecting negatively the pressure reduction created by the wing. [section 2.8.5]
CFD simulations showed that the footplate included in the final design delays the air spillage between the surfaces of the aerofoil, improves the pressure distribution across the span of the front wing main plane and increases the overall downforce generated by the front wing.

8.4.4 Front wing mount

Obtaining an efficient wing mount design was of primary importance this year. As already discussed in the critical review of BR-XV the aerodynamic package testing was not utilised because of unsafe wing mounting methods. An initial simple design of the wing mount was created using SolidWorks:
The front wing mount consists in an 8mm thick aluminium plate. The bottom edge is designed to follow the shape of the pressure side of the front wing main plane aerofoil. The flat top surface features two holes used for bolting the mount to the front of the monocoque. The choice of the diameter of the holes will be discussed subsequently.

Figure 8.4.16 Front Wing Mount

Figure 8.4.17 Front Wing Mount Location
Figure 8.4.17 shows the front wing mount connected to the mainplane. During manufacturing these two components will be either bonded together or the wing mount will be incorporated into the mainplane.

In order to reduce the weight of the component a topology optimisation was performed with SolidThinking Inspire. The software performs an initial structural analysis of the component and removes material in the areas where the stress is relatively low in order to save weight without affecting the structural performance.

Once the optimal shape was calculated by the software a re-design of the part was performed in SolidWorks and a more accurate analysis was carried out with ANSYS Workbench. A load of 750N (higher than what is expected by half of the front wing) was applied at the expected location of the centre of pressure of the front wing. The Cylindrical holes were considered to be fixed supports. The displacement or the rear top edge of the mount was limited to 0mm in the vertical direction to simulate the counter moment created by the monocoque.
The plot above shows the results of the simulation. A localised minimum safety factor of 1.52 is located where the top surface begins to bend to re-join the bottom surface. The results of the simulation are satisfactory: the mount can withstand the load applied (which has been set higher than the real load) with a minimum safety factor of 1.5.

As mentioned earlier the choice of holes’ diameter depends on the size of the bolts to be used. The characteristic of the fasteners to be used are limited by the impact structure regulations:

\textit{T3.22.3 Teams using a front wing must prove the combined Impact Attenuator Assembly and front wing do not exceed the peak deceleration of rule T3.22.2. Teams can use the following methods to show the design does not exceed 300 kg times 40g or 120 kN:}

\textit{b. Combine the peak force from physical testing of the Impact Attenuator Assembly with the wing mount failure load calculated from fastener shear and/or link buckling}

This rule is to ensure that the front wing assembly fails at a certain force in order to prevent injury in case of a front impact with a pedestrian. The easiest and most efficient way of
ensuring that this rule is met is to make sure that the fasteners connecting the wing mount to the monocoque will fail in shear. This will cause the front wing assembly to detach without causing major damage to the carbon components and prevent serious injury to people.

The calculations performed during the impact attenuator design showed a predicted deceleration of 20g. This means that the front wing assembly cannot cause a deceleration in case of impact of more than 20g.

The test is performed with decelerated mass of 300kg. This means that the maximum available shear strength for all bolts is:

\[ F = ma = 300\text{kg} \times 20 \times 9.81 = 58.860\text{kN} \]

Each of the 4 bolts has to possess maximum shear strength of 14.715kN.

The ASTM, American Society for Testing and Materials, provides just Ultimate tensile strength values and Yield strength values for different graded bolts (8.8, 10.9 and 12.9 in our case) without specifying their shear strength.

However the value of Shear strength can be approximated to be 60% of the UTS.

The Ultimate Tensile strength of a bolt is given by the minimum tensile strength of the ASTM grade multiplied by the stress area of the bolt.

Calculations were performed on different graded and sized bolts and the results are presented below:
As it can be seen from Figure 8.4.20, all bolts up to 10.9 grade have enough tensile strength to withstand the aerodynamic load and weight of the wing and moderate shear strength that will allow the assembly to fail without causing more than 20G of deceleration.

In order to have a higher safety factor to compensate for extra forces applied to the bolt (eg. Bending) a bolt with greater shear area (M6) is preferable. Also by having a greater head an M6 bolts spreads the load more evenly on the wing mount.

### 8.4.5 Side wings

Due to the great limitations imposed by the new FSAE regulations on front and rear mounted aerodynamics devices (see section Error! Reference source not found.) it was decided to exploit the sides of the vehicle to generate extra downforce where little limitations are present. It is also beneficial to create more downforce in such areas due to the little influence that side mounted devices have on the aerodynamics balance of the car: the side wings are located relatively close to the centre of gravity of the car so these devices have very little influence on determining the position of the centre of pressure of the whole vehicle.
Two aerofoils are going to mount on the sides of the vehicle following a “Bi-Plane” configuration:

The top wing mount profile is a Selig 1223: this was selected because of the great $C_l$ of the aerofoil and the good performance at low Reynolds Number. The bottom wing’s profile is a NACA 6712 as it was proved to be the best performing in ground effect during previous studies on the front wing.

Finding the correct position of these aerofoils was of primary importance. Whilst the front and rear wing are interacting with air in “free stream”, the side wings interact with an air flow which has been deflected and “spoilt” by the front wing and the front tyres. After numerous iterations the most efficient positions of the aerofoil was verified in CFD:

*Figure 8.4.21 Side Wings for final configuration*
It can be seen from Figure 8.4.22 that with the final design iteration both top and bottom side wings are “fed” from the front wing with relatively clean air. It has to be noted that there is a great and positive interaction between the top side wing and the rear wing. The fast flowing air leaving the suction side of the top profile converges towards the rear wing mainplane suction side energising its boundary layer and helping to prevent flow separation. The side wings generate almost 30% of the total downforce of the car.

8.4.6 Rear Wing

8.4.6.1 3D design
The rear wing was the last component to be designed. Due to its large dimensions and position, high level of downforce and drag can be produced. This is in some way positive, but on the other hand it could cause problems by unbalancing the aerodynamics of the car towards the rear and by creating the previously discussed pitching moment (and hence understeer) (sees section 3.9.1).
In order to maintain optimum vehicle balance the rear wing aerofoils were selected only after the downforce levels generated by the front and side wings were evaluated. The main plane – flap configuration was evaluated with a series of 2D CFD studies in OpenFoam (see portfolio of evidence). Some 3D full car simulations proved the veracity of the results obtained with 2D simulations. An image of the final configuration can be seen below:

![Rear Wing Profiles](image_url)

*Figure 8.4.23 Rear Wing Profiles*

The Selig S1223 aerofoil was selected once again for both the mainplane and the flap. By possessing a great coefficient of lift, it is possible to achieve great amount of downforce with smaller aerofoils, saving important weight from the vehicle. The mainplane is at 10 degrees angle of attack, whilst the flap is at 25 degrees relatively to the mainplane chord.
A great amount of consideration was given to the position of the rear wing relatively to the car. In order to generate more downforce on the rear axle without creating the “pitching” moment effect, the wing is now much closer to the CoG of the vehicle both vertically and horizontally compared to BR-XV design (see comparison between Figure 8.4.24 and Reference source not found.).

Even if located closer to the body of the car the rear wing still exploit free stream clean air that allows the device to effectively cause a great pressure different from the suction and pressure sides with very minimal flow separation (see Figure 8.4.25 and Figure 8.4.26).
8.4.6.2 Rear wing mounts

By being the largest (and heaviest) aerodynamic assembly on the vehicle the rear wing necessitates of proper supports to transmit the load to the vehicle and to restrain the aerofoils’ movements. Due to the “tight” packaging of the rear end of the vehicle designing this structure was not of easy matter. The original idea was to utilise a “swan neck solution” (Figure 8.4.27):
The “swan neck” carbon mount is attached directly to the rear frame of the vehicle and can be bolted or glued to the pressure side of an aerofoil. By mounting devices to the pressure side rather than the suction side of a wing the overall performance of the wing is less affected. In fact the suction side of a wing is more sensitive due to the possibility of flow separation. The swan neck solution was very tempting due to its strength, low mass, and little influence on the flow over the rear wing. It was however discarded due to the asymmetry of the rear frame and a possible clash with the engine’s air intake.

The only possibility was hence to create a truss structure (Figure 8.4.28):

As it can be seen from Figure 8.4.28 the truss members are going to be connected to the side of the mainplane with cleises. As will be discussed in section 9.3.2 these will be bolted to the ribs present inside the aerofoil. As a safety measure these cleises will also be glued to the carbon skin.
The rearmost members of the wing function is to prevent the rear wing assembly to tilt backwards. These are attached to the rear bulkhead with clevises. (Figure 8.4.29)

The front vertical members of the structures are used to restrain the vertical tilt of the front wing. These members are also expected to transmit most of the load of the wing to the chassis. Steel tabs will be welded onto the engine mount chassis’ nodes. (Figure 8.4.30)
The front vertical members will limit the horizontal (in green) and vertical (in red) movement of the rear wing. (Figure 8.4.31)

Steel tabs will be welded on the chassis where the engine mount support creates a node with the main roll hoop (Figure 8.4.32).
Every member end is connected to clevis’ and tabs with rose joints. This allows adjustability of the height and the angle of attack of the rear wing, which can result in an advantage in aerodynamics performance during dynamic events (e.g. Low drag setup for Acceleration).

8.4.7 Vehicle Testing and Aerodynamics evaluation.

One important purpose of the project was to validate the CFD model utilised at Brunel Racing by comparing the numerical data (CFD) to the experimental data gathered at the track. For this reason a test session at the Bruntingthorpe proving ground was performed on the 26/02/2015.

Data at the track was gathered to assess:

- The downforce generated by BR-XV
- The coefficient of Drag of BR-XV
- The pressure distribution on the rear wing mainplane.

8.4.7.1 Test plan

Pre-requisites: Wind characteristics, vehicle projected frontal area, air density, calibrated damper potentiometers, calibrated differential pressure sensors, Wheel speed sensors, C60 Data logger.

1) Coefficient of Drag evaluation (Part 1-2: (with and without wings) (30 min max): The driver will be performing 8 runs in a straight line (in opposite directions). He will gradually accelerate the car to a speed of 54kph, maintain it for 2 seconds, and come off the throttle whilst activating the clutch. The car will coast down until it reaches a complete standstill. Readings of stopping times will be logged.

2) Coefficient of Lift evaluation and pressure distribution (with wings) (45 min max):
The driver will be performing 8 runs in a straight line at a constant speed 54kph (in opposite directions). Data from damper potentiometers and pressure sensors will be logged for investigating the lift generated by the aerodynamic package and the pressure distribution around the mainplane of the rear wing.

8.4.7.2 Experimental setup

The main 3 instruments for measuring the aerodynamics performance of the vehicle are:

- Differential pressure sensors
- Damper potentiometers
- Wheel speed sensors and data logger.

Differential pressure sensors:

For the occasion eight Honeywell 142PC01D differential pressure sensors were utilised to measure the pressure distribution on the rear wing mainplane. These pressure sensors are piezo-resistive devices, containing a silicon membrane that generates a linear output signal proportional to the pressure with a very short time response.

![Honeywell Diff. Pressure Sensor](image)
The sensor can measure a minimum and maximum pressure difference of respectively 0psi and 1psi with a minimum response time of 1ms. The pressure sensors feature two pressure taps from which two different pressures are measured and outputs their difference. [94]

**Linear Damper potentiometers:**

In order to measure the displacement of the damper and hence the spring by the downforce a set of four linear damper potentiometer were utilised:

“Linear potentiometers are variable resistors with three leads. Two leads connect to the ends of the resistor, so the resistance between them is fixed. The third lead connects to a slider that travels along the resistor varying the resistance between it and the other two connections. The resistance element is excited by either DC or AC voltage.” [98]
**Weather Characteristics**

The weather conditions of the test day were made available from the proving ground. These include wind speed, wind direction and air properties. The various gusts of high speed wind were present throughout the day making the weather not ideal for aerodynamics testing. It also has to be added that the day featured medium/heavy rain. Weather characteristics are presented below:

*Figure 8.4.35 Test Day weather characteristics*
Figure 8.4.36 shows the test area (in red). The vehicle travelled the “red path” in direction 240° and 60°. The team was supposed to test on the main runway, however due to a “last minute” booking, BR-XV had to be tested on nearby and bumpier access road.

8.4.7.3 Data analysis

8.4.7.3.1 Drag evaluation and coefficient of rolling friction evaluation (coast down tests)

The coast down test data was the first one to be analysed so that values of Coefficient of drag and Coefficient of rolling friction can be estimated for both car configurations (with and without aerodynamic package).
A “coast down” is a test performed to mainly assess the resistance of a vehicle to motion because of drag and rolling resistance forces. It is performed by accelerating the vehicle to a certain speed and once this is reached and maintained the vehicle is left naturally decelerating to a complete standstill. In order to remove the effect of powertrain friction the vehicle is allowed to decelerate in neutral or with the clutch activated. By measuring the stopping distance of the car (or the time taken to stop) it is possible to derive values of tyre rolling friction and aerodynamic drag forces. The procedure is explained below:

Initially the data was imported into the Bosch Data Analysis Software WinDarab V7. Windarab allows the user to visualize plots of various sensors reading without any extra input. In addition it is possible to cut the data set at the point of interest and import data files into Microsoft Excel for manipulation.

The data sets containing vehicle speed and time for each of the coast down test run were imported in Excel with a logged rate of 0.1s. In order to obtain values of stopping distance it is necessary to know the instant deceleration of the vehicle at each time step. Unfortunately the car did not feature an accelerometer during the test so values of deceleration have to be calculated.

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<th>Speed (Kph)</th>
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</thead>
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</tr>
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<td>0.1</td>
<td>53.7484</td>
</tr>
<tr>
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<tr>
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</tr>
<tr>
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</tr>
<tr>
<td>10</td>
<td>1</td>
<td>51.2953</td>
</tr>
</tbody>
</table>

*Figure 8.4.37: Data sample*
A data set sample can be seen in Figure 8.4.37. The instant deceleration of the car is calculated between two data points using the following formula: 

\[ a = \frac{dv}{dt} \]

where \( dt \) is always 0.1s and \( dv \) is the velocity difference between two consecutive data points.

Once the values of instant deceleration where calculated for each data point, the results were plotted against time. The plot can be visualised in Figure 8.4.38.

The vehicle deceleration presented in the graph contains a decent amount of noise due to the high logging rate of the wheel speed sensors. A polynomial trend line was fitted to the graph to obtain the RMS curve (and its equation) of deceleration.

It is worth to explain the behaviour of the deceleration curve to facilitate the understanding of future calculations: in the region between 0s and 10s the deceleration happens rather quickly compared to the rest of the curve. This is due to the great contribution of drag.
(which varies exponentially with speed) forces in slowing the vehicle down. Between 20s and 25s the deceleration is almost constant: the speed of the vehicle is now so low that the influence of drag is almost non-existent and the only contribution to deceleration is given by rolling resistance.

The next step is to calculate the force which is causing the deceleration of the vehicle. For this calculation RMS deceleration values are used. The car mass with wings and driver is set to be 300kg. The force acting on the car is calculated at each data point using: 

\[ F = ma \]

An average of the force for the last 3 second of the coast down is calculated in order to quantify the force caused the influence of the rolling resistance (see previous explanation of deceleration curve).

Finally the rolling resistance force was subtracted from the total force acting on the car at each time step in order to obtain the exclusive contribution of drag forces.

By knowing the value of the drag force, the air density, the speed of the vehicle through the fluid, and the frontal area of the car it is possible to derive the Coefficient of drag of BR-XV with Equation 2.7.2:

The projected frontal area of the car was approximated from CAD (see example below):
Now the coefficient of drag for each run can be finally evaluated. Furthermore the coefficient of rolling resistance $f_r$ of the vehicle is given by equation $f_r = \frac{R_x}{W}$ [96] where $R_x$ the rolling resistance force and $W$ is the weight of the vehicle.

The same procedure was repeated for the 6 coast down runs with aerodynamics devices mounted onto the vehicle. A comparison of deceleration curves between “wingless” car and the “aero” car can be made by observing Figure 8.4.38 and Figure 8.4.39.
It can be noticed that the car equipped with the aerodynamic package experiences a relatively higher peak deceleration of $1.4 \frac{m}{s^2}$ when the driver activates the clutch, compared to the $0.8 \frac{m}{s^2}$ achieved by the wingless vehicle.

Tables with results for the coast down tests for both configuration are presented below (Error! Reference source not found. and Error! Reference source not found.). It can be seen immediately that the car without aerodynamics component has a coefficient of drag of almost a 50% compared to the car equipped with wings. As expected the coefficient of rolling resistance remains constant for both test. The coefficient of rolling resistance of slightly higher than the usual range for tyres on tarmac. This is explained by the fact that the values calculated in this test include not only the resistance to motion created by the rolling tyre but also the friction in the bearings (which at the test where at the end of their life
time) and the road imperfections. With these considerations the values calculated are very similar to theoretical values [97].

The coefficients of drag calculated in each run for the “wingless” have an average value of 0.74. The velocity term in the drag equation was adjusted to account for wind speed. This was possible to the uniform wind characteristic during the first test. On the other hand during the coast down test with aerodynamics components wind properties were oscillating rapidly making almost impossible an accurate prediction of the wind speed during each run. (Figure 8.4.35) It is possible with some reverse engineering to iterate the values of Cd by adjusting wind speed values in the excel sheet: however it was preferred to average the
result obtained in opposite direction to remove the influence of wind speed on the outcome of the test. The averaged coefficient of drag for the “aero” car is 1.23.

The simulations in CFD for the car equipped with wings showed that the vehicle at the same speed of 15m/s (54kph) was producing 300N of drag. This means that the vehicle has a coefficient of drag of around 1.25. There is hence a difference of 1.6% between CFD simulations and track testing in the calculations of drag values. This is one important discovery that shows the good accuracy of the CFD model utilised at Brunel Racing.

8.4.7.3.2 Downforce Evaluation

The second data set to be analysed was the one gathered with damper potentiometers. By evaluating the compression/extension of these sensors at a given vehicle speed it is possible to evaluate the force generated by the wings.

A chart of the data gathered during a single run is presented below:
Each line represents the movement of the suspension on each corner of the vehicle during each run.

One important consideration to be mentioned is that the damper potentiometers were not zeroed on flat patches. This procedure was instead conducted at the track: this results in an offset from the data collected during the runs. However, damper potentiometers readings were collected with the car stationary (no downforce) on the test area. This reference reading was collected for both travel direction due to the camber of the test strip.

It can also be seen that the data sets present a great amount of noise due to logging rate and imperfections of the tarmac. The data set for each run was hence averaged over time.

Once the average compression/extension of each damper pot was calculated for each run and the reference offset data was subtracted the real displacement of the suspension for each run can be analysed.

The compression/extension of the springs in each corners due to the influence of the aerodynamic package is presented in Figure 8.4.44.
This data is a valuable tool to verify the statements made in the critical review of BR-XV. Most of the drivers experienced a vast amount of understeer at high speed. This was attributed to the pitching moment generated by the rear wing at such velocities. The damper data shows that at 54kph the front springs are extending instead of compressing. This reduces the normal forces acting on the front axle reducing the maximum lateral acceleration achievable by the tyres.

Due to this phenomenon some restrictions will apply to the calculations to evaluate the coefficient of lift or BR-XV. It is not in fact possible now to calculate the force generated by the front wing. It will also be assumed that the force generated by the rear wing will only act at the rear axle and being the only cause of the compression of the rear suspensions. Data from the front axle damper potentiometers will not be utilised.

In order to evaluate the force acting on the springs it is worth remembering the concept of motion ratio. The force of the wing acts directly perpendicular to the ground. The direct actuation dampers are instead mounted at an angle. This means that for a certain vertical movement of the wheel correspond a different movement in the spring. The motion ratio correlates the movement in the spring and the movement of the mass (in this case the wheel). The rear suspension of BR-XV have a motion ratio of 1.5. [42]

The displacement values for the rear dampers are divided by the motion ratio values of 1.5. This will give the vertical displacement of the springs caused by the downforce produce by the rear wing.
The value of displacement for each spring is averaged for different runs in the same direction. Subsequently the values obtained are averaged with the value of spring displacement for the runs in the opposite direction in order to remove the effects of wind and value of Rear Left and Rear right spring displacement \( x_1 \) and \( x_2 \) are obtained.

Now by using \( F = kx_1 + kx_2 \) [99] the total force acting on the rear axle can be calculated.

Calculations shows that the rear wing produces at total of 380N at a speed of 54kph.

According to the Burton’s simulations [42] the total force generated by the car at 54kph is 537N. Due to the great aerodynamic imbalance and pitching moment it was calculated that the rear wing contributes to generate at least 65% of this force (350N). There is a difference of just 8.57% between CFD data and experimental testing. This shows once again the validity of the CFD model used at Brunel Racing if the many source of errors later discussed in section Error! Reference source not found.Error! Reference source not found...

8.4.7.3.3 Pressure Distribution

The last data set to analyse is the one collected with the differential pressure sensors. A series of 8 differential pressure sensors were utilise to measure the pressure on the rear mainplane. Four sensor took reading at different locations on the pressure side of the aerofoil and the remaining four took measured the pressure on the suction side.
The reference pressure (free stream pressure) of each sensor was measured perpendicularly to the free stream flow on the rear wing endplate. Similarly to the lift force measurements (section 9.7.2), the pressure readings for each run were averaged for both vehicle travel direction to eliminate the effect of wind.
Pressure distribution along the rear wing mainplane were gathered from CFD using Paraview (CFD post processing software) and plotted in excel against the measurements taken at Bruntingthorpe. The comparison can be visualised below:

The difference in pressure distribution along the rear wing mainplane between CFD simulations and actual track measurements is minimal. For both the pressure side and suction side of the aerofoil values of similar magnitude follow the same profile. It has to be specified that the pressure peak on the top surface of the mainplane (yellow line) was probably due to an obstruction in the corresponding pressure sensor. Small stones were in fact found in the corresponding pressure sensor hose and affected the outcome of its reading.

8.4.7.3.4 Conclusion and sources of errors

Track testing verified the reliability of the CFD model used by the team and also verified the performance of BR-XV aerodynamic package. It hence hoped that future teams will keep utilising the current CFD technique and will learn about the importance of track testing for aerodynamics design and vehicle setup and performance. The results of the track test were however influenced by some source of errors:

1. Adverse weather condition (rain and wind)
2. Track camber and bumps
3. Obstruction in pressure sensors
4. Filtering of data and data manipulation

8.5 Driver Controls & Electronics (TM)

This section will cover design work by the Driver control and Electronics Manager as well as relevant level 3 project work which drove design decisions in this department.
8.5.1 Driver Controls

The driver controls are the interface between the driver and the vehicle. These can be split into 4 sections: brake system, pedal box, steering wheel and column and the driver cell. These often have to meet safety regulations as well as “fit” the driver to improve their performance.

8.5.1.1 Brake System Design

The brake system has been defined as the complete system where components are used between the application of the driver’s foot on the brake pedal and the brake disc. A basic representation of this process can be displayed using the flowchart in Figure 8.5.1.

This sequence of processes is vital to the calculation of brake forces. An accurate calculation can lead to better brake system design and driver feedback.

8.5.1.1.1 Brake Calculations

Accurate brake force calculations involve a number of steps using simple calculations. The process is outlined below quoting only a few of the formulas.

1. The lever ratio of the pedal (between application of force and master cylinders) is used to calculate the brake pedal output force $F_{bp}$.
2. The pressure in the master cylinders is calculated by dividing the brake pedal output force by the area of the master cylinder piston:

$$ P_{mc} = \frac{F_{bp}}{A_{mc}} $$

*Equation 8.5.1 Pressure in Master Cylinder [100]*

3. The pressure loss in the brake line is assumed to be zero; therefore the pressure to the callipers is equal to that in the master cylinders (calculated at 3.3MPa)

4. The force applied by the calliper is equal to the pressure multiplied by the area of its pistons. This is then multiplied by two to calculate the clamping force

5. The Frictional force generated by the brake pads is found by the product of the clamping force and the coefficient of friction of the brake pads:

$$ F_{fr} = F_{clamp} \times \mu_{bp} $$

*Equation 8.5.2 Friction force of brake pads [100]*

6. The torque generated by the brake disc is equal to the friction force multiplied by the effective radius of the disc (radius of the pads from the centre of rotation)

7. The tyre torque is assumed to be equal to the wheel torque and the brake disc torque

The above steps were used in a spreadsheet to adjust parameters. The weight transfer

$$ WT = \left( \frac{a_v}{g} \right) \times \left( \frac{h_{cg}}{WB} \right) $$

*Where:*

- **WT** is absolute weight transfer from rear to front (N)
- **$a_v$** is deceleration of the vehicle (ms\(^{-1}\))
- **g** is acceleration due to gravity (ms\(^{-1}\))
- **$h_{cg}$** is the vertical height od CG from ground (m)
- **WB** is the wheelbase (m)

*Equation 8.5.3 Weight transfer under braking [100]*
under braking could then be found.

Braking from 60mph, the weight transfer for our vehicle (250kg) under braking is found to be 705N. This means the vertical weight force under braking for the front and rear axles is 1932N and 520N respectively.

8.5.1.1.2 Relevant Level 3 Work on Brake Design

The brake disc design was researched as part of a level 3 project. Structural and thermal analysis was undertaken by Ermogenous (surname). Research showed that the most suitable material for the brake discs for a vehicle with 10” wheels is 4310 steel.

The research showed that a drilled-hole pattern may remove material and reduce mass but only benefits cooling and braking performance over 250°C. These temperatures are unlikely to be seen in a Formula Student application. The proposed solid discs will be tested on a vehicle along with drilled discs before a final decision is made. This test will also provide information on driver feedback, an important consideration in brake system design.

8.5.1.1.3 Brake Line Specification

Smooth bore PTFE hose is the preferred brake line specification. It has low volumetric expansion at maximum operating pressures [101]. Low expansion reduces pressure loss and provides better brake pedal feedback. PTFE hose can operate at pressures up to 3000PSI (20.7 MPa).

Brake line pressures in section 8.5.1.1.1 have been calculated at 3.3MPa. After further discussion with the supplier (a sponsor) it was agreed to run -2 hose instead of standard -3 hose. All components from the supplier are free of charge so if this venture proves unsuccessful -3 hose will be supplied free of charge and little alteration will be required.
8.5.1.2 Pedal Box Design

The pedal box must be widely adjustable [37]. In order for these requirements to be met a double-hole system will be used. There will be mounting holes in the chassis for adjustment up to 30mm. There will be a separate hole set in the floor of the pedal box itself in order to adjust it to every 10mm.

8.5.1.2.1 Pedal Box Chassis Mounting Points

There are various regulations relating to the positioning of the pedal-box. One such regulation is driver accommodation. The vehicle must be designed to fit a range of drivers.

A1.2.2 The vehicle must accommodate drivers whose stature ranges from 5th percentile female to 95th percentile male and must satisfy the requirements of the Formula SAE Rules [37].

Further clarification on this is given in regulation T3.10.4. A 95th percentile male template needs to fit in a driving position in the vehicle and a number of specified dimensions of this template are given.

T3.10.5 If the requirements of T3.10.4 are not met with the 95th percentile male template, the car will NOT receive a Technical Inspection Sticker and will not be allowed to compete in the dynamic events [37].

There are no specifications on the dimensions of a 5th percentile female driver and so this must be acquired in order to confirm the mounting positions of the pedal box. The
anthropometric data allows the horizontal travel of the pedal box to be calculated.

In order to calculate the horizontal travel whilst ensuring the pedal to hip distance for 95\textsuperscript{th} percentile male is 915mm, upper and lower leg lengths for the extreme body sizes is required.

When considering pedal position for a 95\textsuperscript{th} percentile male, the regulations state that a template must fit within the car. Using this template it was established (and the chassis designed in order) that there is 100mm between the pedal face and the nose of the chassis. Figure 9.5.2 demonstrates the driver geometry. The regulations in this case also specify that length “A” is 915mm.

By calculating the knee angle “α”, the process can be reversed to calculate “A” for a 5\textsuperscript{th} percentile female driver. Lengths “B” and “C” will be acquired using anthropometric data for the upper and lower leg. Angles “β” and “γ” will remain approximately the same but will differ slightly with driver size.

In order to calculate angle “α” for 95th percentile male driver, upper leg “B” is taken to be

\begin{equation}
\text{Equation 8.5.4 Cosine Rule}
\end{equation}
645mm and lower leg “C” as 490mm [102].

Taking the value of $\alpha$ as 111.76° and using upper (520mm) and lower (355mm) [102] leg data for a 5th percentile female, the length “A” can be found.

$$A^2 = B^2 + C^2 - 2BC \cos \alpha$$

$$A = \sqrt{B^2 + C^2 - 2BC \cos \alpha} = \sqrt{520^2 + 355^2 - 2 \times 520 \times 355 \cos 106.6^\circ} = 708.5\text{mm}$$

From this the horizontal pedal travel can be calculated using Pythagoras’ theorem. For a 95th percentile male driver:

$$A^2 = B^2 + C^2$$

$$B = \sqrt{915^2 - 57^2} = 913.2\text{mm}$$

For a 5th percentile female driver:

$$A^2 = B^2 + C^2$$

$$B = \sqrt{709^2 - 57^2} = 706.7\text{mm}$$

Horizontal pedal box travel required:

$$913.22 - 705.70 = 206.5\text{mm}$$

In order to accommodate the required driver size range the pedal box must have a minimum horizontal travel of 206.5mm. To ensure the travel is above this range, and for easier adjustment and manufacture the horizontal travel will be set at 210mm. This is the equivalent of 8 sets of holes spaced 30mm apart. This hole set coupled with the hole pattern on the pedal box itself will enable the pedals to be adjusted at 10mm intervals.
8.5.1.2.2 Accelerator Pedal Regulations

There are no regulations regarding the material or manufacture of the accelerator pedal. There are certain design restrictions, which concern safety and failures. Other regulations must be considered:

IC1.5.6 A positive pedal stop must be incorporated on the throttle pedal to prevent over stressing the throttle cable or actuation system.

IC1.5.7 The throttle pedal cable must be protected from being bent or kinked by the driver’s foot when it is operated by the driver or when the driver enters or exits the vehicle.

IC1.5.8 If the throttle system contains any mechanism that could become jammed, for example a gear mechanism, then this must be covered to prevent ingress of any debris.

IC1.13.2 The foot pedal must return to its original position when not actuated. The foot pedal must have a positive stop preventing the mounted sensors from being damaged or overstressed. Two (2) springs must be used to return the throttle pedal to the off position and each spring must work with the other disconnected [37].

8.5.1.2.3 Accelerator Pedal Design Requirements

The design space for the pedal box and accelerator pedal are completely defined by the chassis design and regulations. This meant there were some restrictions:

- The maximum distance from a 95th percentile male driver template to the front bulk head is 100mm
- The maximum width of the forward most part of the base at floor level is no more 190mm
- Mounting holes are confined to the centreline of the pedal to reduce twisting moment
- The maximum width of the footplate can be 140mm, or else it would cross the centreline of the vehicle.

These restrictions also inflict limitations on the amount of pedal travel. This is also dependent on the pedal height.
Key information relating to driver dimensions and ergonomics provide other design guidelines:

- The foot plate must be a minimum 110mm wide in order to accommodate a 95\textsuperscript{th} percentile male foot [102]
- As most pressure is applied through the ball of the foot, the top of the footplate is at least 210mm from the pedal axle for a 95\textsuperscript{th} percentile male driver [102]
- The bottom of the footplate is a maximum of 160mm from the pedal axle for a 5\textsuperscript{th} percentile female driver [102]
- The exerted force (not stated in regulations) will be taken as 800N. This assumes the pedal is operated by leg motion as well as ankle motion [102].

8.5.1.2.4 Accelerator Pedal Material Specification

Accelerator pedal material as stated in section 8.5.1.2.2 is a free choice. This makes the accelerator pedal an area where considerable mass can be saved. The pedal box from BR-XV had an overall mass of 2.7kg [42]. The accelerator pedal had a mass of approximately 0.55kg [38]. In order to reduce the mass of the pedal it is sensible to investigate other materials.

<table>
<thead>
<tr>
<th>Material</th>
<th>Yield Strength MPa</th>
<th>Ultimate Strength MPa</th>
<th>Young’s Mod. GPa</th>
<th>Shear Mod GPa</th>
<th>Density kg/m(^3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td>435</td>
<td>670</td>
<td>210</td>
<td>80</td>
<td>7850</td>
</tr>
<tr>
<td>Aluminium 6082 T6</td>
<td>340</td>
<td>310</td>
<td>70</td>
<td>26</td>
<td>2700</td>
</tr>
<tr>
<td>Carbon (3ply) Rohacell Sandwich</td>
<td>-</td>
<td>2250</td>
<td>147</td>
<td>-</td>
<td>620</td>
</tr>
</tbody>
</table>

*Table 8.9 Comparative Accelerator Pedal Material Properties*
Three main materials used are Aluminium 6082 T6 grade alloy (used in BR-XV), and 2 others.

It is clear that FE simulations show that CFRP Rohacell Sandwich can withstand sufficient pedal force with a much lower density (and by extension mass) than other available materials. A bonus is that the required CFRP and Rohacell are already needed for other projects. Since the pieces for the accelerator pedal are small they can be made from offcuts of unused material and bonded together. If offcuts do not leave sufficient material then premade CFRP Rohacell sheets can be bought in for around £50-£60. This still leaves sufficient budget for the rest of the assembly.

8.5.1.2.5 Accelerator Pedal Development

Initial designs were similar to previous year’s designs. In this instance it was quickly realised that with the cockpit and chassis design, the driver would be more likely to exert greater force away from the centreline of the pedal. This would create a twisting moment and cause a number of problems with stability. In order to reduce this unwanted moment, the pedal uprights were separated as far as possible. This ensured the greatest amount of force on the pedal was kept between the uprights.

Other adjustments were made to reduce weight. ANSYS Workbench 15.0 was used to conduct FE analysis to guide weight reduction. Two areas were shown to have less strain and thus slots were created.
to reduce weight (label 1 in Figure 8.5.5). Torsion springs would be used to return the pedal to the upright position using the base plate and a parallel bar between uprights (label 2 in Figure 8.5.5). A simple throttle bar was used to pull the throttle cable, which reduced space required behind the pedal (label 3 in Figure 8.5.5). There are 6 mounting holes spaced 10mm apart, meaning the pedals can be adjusted every 10mm for the 210mm travel calculated in 8.5.1.2.1.

These changes led to the design shown in Figure 8.5.5, which is a final design render. This design was only confirmed as the final design after FE analysis discussed in section 8.5.1.2.6. Note that the base plate will change to incorporate both pedals on a single base to add rigidity. This will also enable the possibility of extending the axle so both pedals share the same axle. This would be done for the same reason.

8.5.1.2.6 Accelerator Pedal Analysis

The geometry of the Solidworks 2014 CAD file was imported into ANSYS Workbench 15.0 for FE analysis to be conducted.

The pedal constraints were defined so that the pedal was at maximum throttle; against the pedal stoppers, with 1000N being applied to the footplate. This indicated a safety factor of 1.25 over the 800N specified in section 8.5.1.2.3.
Fixed supports were applied to the lower faces of the pedal uprights, frictionless supports were applied to the bearing/axle holes. A 1000N load was applied to the foot plate over a one second time step see Figure 8.5.6.

This analysis showed how the pedal would react to being compressed when already at maximum displacement and on the pedal stops. The analysis was conducted with the 3-ply carbon skins of the pedal bonded to the Rohacell 51 core.
Figure 8.5.7 shows the equivalent stress of the pedal assembly. As expected the maximum stress occurs at the vertex of the rear face of the upright. This is the area most likely to fail and has a maximum stress of 383 MPa.

Failure can be checked by analysing other results; Figure 8.5.8 shows the calculated safety factor and the total deformation of the pedal under 1kN of force. The maximum deformation occurs at the furthest point from the pedal axle. This can be explained as an accumulation of deformation from the axle up.
The safety factor is calculated from a number of stress parameters. A stress factor of less than 1 means that the assembly would not withstand the current parameters. Since a safety factor of 1.25 has already been applied, \( SF \geq 1 \) is sufficient to pass.

This design passes with an SF of 1.28. If the FEA calculated SF of 1.28 is combined with the originally applied SF of 1.25, an overall SF can be found as 1.6: meaning that the maximum force the accelerator pedal can theoretically withstand before failure is 1280N.

The assembly is able to withstand the required force; more material can in fact be removed but in order to maintain manufacturing time and keep costs down it was finalised here. Only small adjustments may be made if required when combining the two pedals for the pedal box.

8.5.1.2.7 Relevant Level 3 Brake Pedal Design

Brake pedal design was part of a level 3 project. This entailed the design and analysis of a pedal to ensure it could withstand 2000N of force and remain as light as possible. There are
material and manufacture restrictions related to the brake pedal design which state that it must be fabricated from steel or aluminium or machined from steel, aluminium or titanium [37].

Research by Ermogenous shows that aluminium is best suited to this application, provided it is not welded. Analysis in ANSYS 15.0 showed that an aluminium construction with a bolted footplate was suitable can could withstand 2000N of force applied to the footplate with a safety factor of 6.64. The FE analysis is shown in Figure 8.5.9.

This design was ultimately adjusted when combined with the rest of the pedal box. After the master cylinders and bias bar were drawn in Solidworks 2014 the brake pedal attachments to the pedal were adjusted by 2mm in order to ensure everything fitted correctly.

It is possible to alter this design further at a later date to reduce overall mass since there is a high safety factor. This mass is likely to be taken from the footplate as it currently has a thickness of 3mm. Further research is required.
Overall Pedal Box Design

The pedal box had very tight packaging requirements due to restrictions from the chassis. However, these specifications have been met and the mass target of the pedal box has also been achieved. Values are shown in table 9.5.2 for the approximated BR-16 pedal box masses using Solidworks and ANSYS mass and volume information.

Other component mass includes washers, bolts and welding, which has been estimated as this cannot be accurately measured in Solidworks 2014. Figure 8.5.10 is a Solidworks 2014 assembly render of the proposed pedal box design.

<table>
<thead>
<tr>
<th>Part</th>
<th>BR-XV (including rails) (kg)</th>
<th>BR-16 Target (kg)</th>
<th>BR-16 Estimated (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brake Pedal</td>
<td>0.306</td>
<td>0.275</td>
<td>0.390</td>
</tr>
<tr>
<td>Accelerator Pedal</td>
<td>0.152</td>
<td>0.137</td>
<td>0.055</td>
</tr>
<tr>
<td>All Other components</td>
<td>1.720</td>
<td>1.548</td>
<td>≈ 0.820</td>
</tr>
<tr>
<td>Mounting Rails</td>
<td>0.468</td>
<td>0.421</td>
<td>-</td>
</tr>
<tr>
<td>Total</td>
<td>2.646</td>
<td>2.381</td>
<td>1.265</td>
</tr>
</tbody>
</table>

*Table 8.10 Mass properties and estimations of Pedal Box*
Figure 8.5.10 Full Pedal Box Assembly (minus bolts and washers)
8.5.1.3  Steering Column

8.5.1.3.1  Relevant Level 3 Design Work on Column

Torsion and stress calculations were conducted by Ahmad (surname) on magnesium and mild steel columns. An outer diameter of 15mm (taken from the internal diameter of the universal joints) and guideline safety factor of 1.2 was used to find shear stress and inner diameter (shown in Table 8.12). Budget restrictions dictate that 4130 steel will be used.

Ahmad used this information to propose the column geometry shown in Figure 8.5.11.

<table>
<thead>
<tr>
<th>Condition</th>
<th>Magnesium Alloy</th>
<th>4130 Steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer Diameter (m)</td>
<td>0.015</td>
<td>0.015</td>
</tr>
<tr>
<td>Inner Diameter (m)</td>
<td>0.008</td>
<td>0.0102</td>
</tr>
<tr>
<td>Maximum Shear Stress (MPa)</td>
<td>164.77</td>
<td>193.34</td>
</tr>
<tr>
<td>Safety Factor</td>
<td>1.2</td>
<td>1.2</td>
</tr>
</tbody>
</table>

*Table 8.11 Parameters of Magnesium alloy and 4130 steel for steering column*

8.5.1.3.2  Packaging and Column Geometry

The original geometry detailed for the steering column was not possible with the specified components. In order for the proposed geometry to work the universal joints would need to
perform at an angle of 45°. The maximum capability of these joints is 32°.

To solve this issue meant inclining the steering rack at 30° (after discussion with EJ put his name) and using the two universal joints to bend the column a further 30°. This geometry is shown in Figure 8.5.12. Note: the diameters of a Mild steel column is specified in Table 8.12 and the distance from the universal joint axle to the end of the joint is 15mm. This geometry also ensures that the forward most face of the wheel is no more than 250mm from the rear of the front roll hoop. The shaft between the joints is now the shortest as this shaft has the most “wobble” and reducing the length reduces the radial wobble.

8.5.1.4 Driver’s Cell

The driver’s cell is the region of the car where the driver is located. It is important the driver is comfortable and everything the driver needs to actuate fits and is within reach.
8.5.1.4.1 Driver Seating

Due to packaging restrictions in BR-16 there will be no seat. Instead foam will be used to pad the driver cell. This foam will be removable and interchangeable for each driver.

8.5.1.4.2 Switch Pod

The switch pod is one of the main driver interfaces. It houses the switches used to activate certain vehicle components. Some components require certain type of switches. The switch pod will be located above the radiator air duct on the right hand side of the driver’s cell (see Figure 8.5.15). Table 8.13 lists the switch pod components.

<table>
<thead>
<tr>
<th>Component</th>
<th>Function</th>
<th>Requirements</th>
<th>Switch Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ignition</td>
<td>Controls power to the ignition relay and ignition module</td>
<td>LED indicator, 2-way switch</td>
<td><img src="image" alt="Switch Image" /> [120] (£1.79 each)</td>
</tr>
<tr>
<td>Water pump</td>
<td>Turns on the water pump</td>
<td>LED indicator, 2-way switch</td>
<td>Same as Ignition</td>
</tr>
<tr>
<td>Fuel pump</td>
<td>Turns on the fuel pump</td>
<td>LED indicator, 2-way switch</td>
<td>Same as Ignition</td>
</tr>
<tr>
<td>Start button</td>
<td>When activated allows power to starter motor</td>
<td>Momentary push button</td>
<td><img src="image" alt="Switch Image" /> [121] (£4.86 each)</td>
</tr>
</tbody>
</table>

Note: LED indicator switches will be powered from the high-side of the relay not from the switch (see section 2.8.4). It must indicate that the component is powered not that the switch is activated.
The cockpit mounted master switch will not be located on the switch pod to remove the possibility of accidental actuation.

8.5.2 Electronics

8.5.2.1 GEMS Display Programming

The GEMS LDS4 display is a programmable AMOLED (active-matrix organic light-emitting diode) display used in motorsport. The AMOLED technology means the display can be extremely bright. This allows the screen to be viewed in direct sunlight and through a driver’s visor [103]. The display will be mounted on the steering wheel in order for it to be visible to the driver. This display has been available to previous Brunel Racing teams. BR-14 used it successfully; BR-XV planned for it but did not use it. Due to limited cockpit space in BR-16, the integration of this display is a vital means of driver communication.

After a group discussion with three drivers, it was agreed that a different screen layout for each event seemed like the best programming option. For different events the drivers require different information to be displayed. A list of features was agreed for each event.

<table>
<thead>
<tr>
<th>Acceleration</th>
<th>Skid Pad</th>
<th>Auto-cross (sprint)</th>
<th>Endurance</th>
</tr>
</thead>
<tbody>
<tr>
<td>• RPM</td>
<td>• Lateral ratio</td>
<td>• RPM</td>
<td>• RPM</td>
</tr>
<tr>
<td>• Gear</td>
<td>• Gear</td>
<td>• Gear</td>
<td>• Gear</td>
</tr>
<tr>
<td>• Traction Control Status</td>
<td>• G/slip</td>
<td>• Traction Control Status</td>
<td>• Traction Control Status</td>
</tr>
<tr>
<td>• Launch Control Status</td>
<td>• Gear</td>
<td>• Launch Control Status</td>
<td>• Control Status</td>
</tr>
<tr>
<td>• Battery Voltage</td>
<td></td>
<td>• Battery Voltage</td>
<td>• Battery Voltage</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>• Lap Delta</td>
</tr>
</tbody>
</table>

*Table 8.13 GEMS display driver requirements by event*

Drivers in many forms of motorsport are not interested in vehicle speed. This is confirmed in Table 8.13. With this information acquired some initial designs were drawn up.
The GEMS has an editor software used to program the display. This software has a number of preset items such as the tachometer. The user can drag and drop, resize and position items on the GUI using pixel coordinates or the computer mouse. Features and inputs can be edited and displayed on screen. Figure 8.5.13 shows the editing box for the tachometer (tacho) for rpm. The maximum is set to 14500 rpm with each bar representing 500 rpm. A lower and upper tacho limit is set at 6000 and 12000 rpm respectively. The tacho fill colours are set to match branding guidelines in section 0. The bar fills white up to 6000 rpm, blue to 12000 rpm and red above 12000rpm. The input data is selected and then the information can be displayed.

It is also possible to display numerical values and variable strings. This means after the input is correctly selected the gear can be displayed. The battery voltage level can be directly read and displayed, using similar programming as in Figure 8.5.13. Most information will be
transmitted to the screen using the CAN network. The CAN network can carry almost any information from the ECU to the screen to be displayed [67].

Figure 8.5.14 shows the proposed GEMS display design for the acceleration event. The left side is labelled and the screen of the stationary vehicle. The right side shows the screen with:

- The vehicle in second gear
- The engine at 13400 rpm
- Traction control set to setting 4
- Launch control set to setting 1
- Live battery voltage

This design has been approved by the drivers and can be used on BR-16.

8.5.2.2 Data Acquisition and Sensor Increase

BR-16 will use a Bosch ECU and data logger. The MS4 ECU has been used for within Brunel Racing for many years. The ECU logs various sensors as well as controlling numerous vehicle parameters. The Data logger is a Bosch C60 and has many analogue inputs which can be assigned to sensors of choice. The complete list of sensors is shown in Table 9.5.6
<table>
<thead>
<tr>
<th>Sensor</th>
<th>Quantity</th>
<th>MS4 ECU or C60 logger</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oil temp.</td>
<td>1</td>
<td>ECU</td>
</tr>
<tr>
<td>Oil press.</td>
<td>1</td>
<td>ECU</td>
</tr>
<tr>
<td>Fuel temp.</td>
<td>1</td>
<td>ECU</td>
</tr>
<tr>
<td>Fuel press.</td>
<td>1</td>
<td>ECU</td>
</tr>
<tr>
<td>Lambda</td>
<td>1</td>
<td>ECU</td>
</tr>
<tr>
<td>Throttle pos.</td>
<td>1</td>
<td>ECU</td>
</tr>
<tr>
<td>MAP</td>
<td>1</td>
<td>ECU</td>
</tr>
<tr>
<td>Inlet Air temp.</td>
<td>1</td>
<td>ECU</td>
</tr>
<tr>
<td>Coolant temp.</td>
<td>1</td>
<td>ECU</td>
</tr>
<tr>
<td>Gear</td>
<td>1</td>
<td>ECU</td>
</tr>
<tr>
<td>Damper potentiometer</td>
<td>4</td>
<td>C60 Logger</td>
</tr>
<tr>
<td>Accelerometer</td>
<td>3</td>
<td>C60 Logger</td>
</tr>
<tr>
<td>Brake position</td>
<td>1</td>
<td>C60 Logger</td>
</tr>
<tr>
<td>Brake press.</td>
<td>2</td>
<td>C60 Logger</td>
</tr>
<tr>
<td>Brake temp</td>
<td>4</td>
<td>C60 Logger</td>
</tr>
<tr>
<td>Wheel speed</td>
<td>4</td>
<td>ECU</td>
</tr>
<tr>
<td>Steering position</td>
<td>1</td>
<td>C60 Logger</td>
</tr>
<tr>
<td>Tyre temp.</td>
<td>4</td>
<td>C60 Logger</td>
</tr>
<tr>
<td>Air pressure</td>
<td>6</td>
<td>C60 Logger</td>
</tr>
<tr>
<td>Current</td>
<td>1</td>
<td>C60 Logger</td>
</tr>
<tr>
<td>GPS</td>
<td>1</td>
<td>C60 Logger</td>
</tr>
<tr>
<td>Lap beacon</td>
<td>1</td>
<td>C60 Logger</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>42</strong></td>
<td></td>
</tr>
</tbody>
</table>

*Table 8.14 List of sensors*
8.5.2.3  Wiring Loom Design

Designing a wiring loom is a lengthy process. There are a number of steps that need to be completed in order to create a simple, neat loom:

- Find which components need to go on the vehicle
- Find what each component requires in terms of inputs/outputs
- Cross reference all electronic component needs with ECU pins and data logger pins
- Verify all component positions
- Route all wiring avoiding major harmful components (e.g. exhaust)
- Calculate all wire lengths
- Update wiring diagram throughout the process

8.5.2.3.1  Wiring Loom Design Considerations

There are a few areas that must be considered when designing a wiring loom. The number of sensors, battery positioning, relay positioning and wire gauges are all commonly and correctly considered.

One area traditionally gets overlooked at Brunel Racing. The lifetime of the vehicle can be declared as the period of time for which it is used. This is often incorrectly stated as length of competition. The true lifetime of the vehicle is for approximately 8 months after completion to allow for testing. With this in mind, the loom is to be designed in order to make it as easy as possible to splice/add new wires, without compromising the design of the loom itself. An example of this is that an unused connector can be added to the loom with the intention for it to be used for sensors next year. This will add little mass, but if well designed, could be an invaluable time saver next year. Other areas of the loom which are likely to require a splice for added components can be extended slightly in order to make
splicing easier. This philosophy will be used when routing the loom as much as possible (wary of the restrictions of the chassis).

8.5.2.3.2 Wiring Loom Routing

Before a wire route map can be constructed the major electrical components must be mapped onto the vehicle. Figure 8.5.15 shows the approximate position of major electrical components and the power routing.

All power from the battery (1) goes through the master switch on the right-hand-side of the vehicle (5). From here the power goes to the relays (3) and the fuse box (2).

Power cables are routed along the side of the vehicle as far as possible and mainly along the right-hand side. This reduces the total wire length as this is where the relays, fuses and master switch are located.
The electronics for the steering wheel and GEMS display are routed through the steering column using an electronic steering boss and exit via a hole in the column before the first universal joint (refer to Figure 8.5.12). The 5V power, CAN lines 1 and 2 (white and green) and the ECU data lines (light blue) reach the steering wheel via the steering column. Traction and launch control settings are adjustable on the wheel and relay to the ECU via the ECU data lines.

The loom will have two halves. The front half is located in the monocoque (exceptions being outboard sensors). The rear half will be located on the rear frame. A “quick-disconnect” system will be used to connect both sides of the loom near the master switch.

Engine electronics, such as the ignition module (IM4) and engine data acquisition sensors are all part of Data loom 1. Data loom 1 is situated on the rear frame and uses the quick-disconnect system through the monocoque to the ECU.

Data loom 2 consists of more engine electronics as well as the wheel speed sensors, which have specific pin allocations in the ECU.

Data loom 3 is mainly vehicle dynamics sensors. These sensors return the data to the C60 data-logger for post processing.

The brake over-travel switch must kill the engine and the fuel pumps. It therefore cuts the power to the fuel pump relay and the engine (represented by the orange dot on Figure 8.5.15).

When routing the wiring there are further considerations. In order to protect wiring on the rear frame from over-heating it should be routed on the opposite side of the frame to the heat source [104].
Another consideration is the wire gauge used for each component based on required current supply. Most sensors only require milliamps of current therefore predominately use 28 AWG (American wire gauge) wire to save mass on data logging looms. Table 8.16 shows current capacity for a range of AWG.

<table>
<thead>
<tr>
<th>AWG Gauge</th>
<th>Conductor diameter (mm)</th>
<th>Ohms per Km</th>
<th>Maximum Current (A)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>3.2639</td>
<td>2.060496</td>
<td>73</td>
</tr>
<tr>
<td>9</td>
<td>2.90576</td>
<td>2.598088</td>
<td>64</td>
</tr>
<tr>
<td>10</td>
<td>2.58826</td>
<td>3.276392</td>
<td>55</td>
</tr>
<tr>
<td>11</td>
<td>2.30378</td>
<td>4.1328</td>
<td>47</td>
</tr>
<tr>
<td>12</td>
<td>2.05232</td>
<td>5.20864</td>
<td>41</td>
</tr>
<tr>
<td>13</td>
<td>1.8288</td>
<td>6.56984</td>
<td>35</td>
</tr>
<tr>
<td>14</td>
<td>1.62814</td>
<td>8.282</td>
<td>32</td>
</tr>
<tr>
<td>15</td>
<td>1.45034</td>
<td>10.44352</td>
<td>28</td>
</tr>
<tr>
<td>16</td>
<td>1.29032</td>
<td>13.17248</td>
<td>22</td>
</tr>
<tr>
<td>17</td>
<td>1.15062</td>
<td>16.60992</td>
<td>19</td>
</tr>
<tr>
<td>18</td>
<td>1.02362</td>
<td>20.9428</td>
<td>16</td>
</tr>
<tr>
<td>19</td>
<td>0.91186</td>
<td>26.40728</td>
<td>14</td>
</tr>
<tr>
<td>20</td>
<td>0.8128</td>
<td>33.292</td>
<td>11</td>
</tr>
<tr>
<td>21</td>
<td>0.7239</td>
<td>41.984</td>
<td>9</td>
</tr>
<tr>
<td>22</td>
<td>0.64516</td>
<td>52.9392</td>
<td>7</td>
</tr>
<tr>
<td>23</td>
<td>0.57404</td>
<td>66.7808</td>
<td>4.7</td>
</tr>
<tr>
<td>24</td>
<td>0.51054</td>
<td>84.1976</td>
<td>3.5</td>
</tr>
<tr>
<td>25</td>
<td>0.45466</td>
<td>106.1736</td>
<td>2.7</td>
</tr>
<tr>
<td>26</td>
<td>0.40386</td>
<td>133.8568</td>
<td>2.2</td>
</tr>
<tr>
<td>27</td>
<td>0.36068</td>
<td>168.8216</td>
<td>1.7</td>
</tr>
<tr>
<td>28</td>
<td>0.32004</td>
<td>212.872</td>
<td>1.4</td>
</tr>
<tr>
<td>29</td>
<td>0.28702</td>
<td>268.4024</td>
<td>1.2</td>
</tr>
<tr>
<td>30</td>
<td>0.254</td>
<td>338.496</td>
<td>0.86</td>
</tr>
</tbody>
</table>

Table 8.15 American Wire Gauge (AWG) Current Capacity [122]

Other electrical components such as the 12V power cable from the battery to the master switch and the relay boxes, will require 12 AWG. This will ensure there are no overheating problems or restricted current to key electrical components.
8.5.2.4 Power Cell

After using the research conducted in section 2.8.5 the power cell selected for use in BR-16 is a lithium iron phosphate (LiFePO₄) cell. It has twice the capacity of the BR-XV specified cell which should allow use of the racing alternator; although testing will be required.
8.6 Innovative Design Solutions (JS)

8.6.1 Drag Reduction System

The current design for the aerodynamic package of BR-16 includes front, rear and side wings, all of which have 2 elements. The design of these aerodynamic components is discussed in 0. To achieve minimal straight line drag of the car, the Drag Reduction System (DRS) will be design and developed for all three of the wings. This will produce the minimal drag coefficient of the car in a straight line and so increase the vehicles top speed.

A number of initial design considerations were made prior to the development of the DRS:

1. The main plain of each wing would remain static and not have rotational availability.
2. The second wing elements will provide the rotational availability for reduced drag.
3. The system must be easily controlled by the driver whilst driving the car.

8.6.1.1 2-D Computational Fluid Dynamics

Initially 2-D Computational Fluid Dynamics (CFD) was used to calculate the fluid flow over the wing elements as this allowed for reduce computational time and enabled an increased number of design iterations. The results from 2-D can also be seen to have a scaled relationship to 3-D and so the results gained will be translate into 3-D further reducing the number of 3-D simulations required. The hand-calculated 2-D to 3-D results must then be validated by a 3-D CFD simulation as discussed in 8.6.1.1.3.

8.6.1.1.1 CFD Setup

CFD simulation was carried using Ansys Fluent 2-D and 3-D modes dependant on the type of simulation. The simulation setup is briefly outlined below:
8.6.1.1.2 Rotation Angle and Pivot Point Location

The first stage in the design and development of the DRS was to calculate the angle of the second element at which each wing produced the least drag. This will be the angle at which the electro-mechanical servos will rotate the wing elements upon activation of the system by the driver. The 2-D wing profiles were analysed independently at a fluid velocity of 15 m/s and rotational increments of 5 degrees. The second elements were initially rotated about half the chord length of each element, a pivot point established as a neutral location. The results of each wing are shown below in Table 8.16. The table shows the reduction in drag of each wing between its original rest position and the determined minimal drag position.
<table>
<thead>
<tr>
<th>Second Element DRS</th>
<th>Rest Position</th>
<th>DRS Position</th>
<th>Drag Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Front Wing</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Angle (deg)</td>
<td>0</td>
<td>25</td>
<td>0.86</td>
</tr>
<tr>
<td>Main Element Drag (N)</td>
<td>8.3</td>
<td>7.44</td>
<td>10.38</td>
</tr>
<tr>
<td>Second Element Drag (N)</td>
<td>13.94</td>
<td>3.56</td>
<td></td>
</tr>
<tr>
<td>Total Drag (N)</td>
<td>22.24</td>
<td>11</td>
<td>11.24</td>
</tr>
<tr>
<td><strong>Side Wing</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Angle (deg)</td>
<td>0</td>
<td>10</td>
<td>8.85</td>
</tr>
<tr>
<td>Main Element Drag (N)</td>
<td>25.22</td>
<td>16.37</td>
<td></td>
</tr>
<tr>
<td>Second Element Drag (N)</td>
<td>15.78</td>
<td>11.01</td>
<td>4.77</td>
</tr>
<tr>
<td>Total Drag (N)</td>
<td>41</td>
<td>27.38</td>
<td>13.62</td>
</tr>
<tr>
<td><strong>Rear Wing</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Angle (deg)</td>
<td>0</td>
<td>25</td>
<td>62.15</td>
</tr>
<tr>
<td>Main Element Drag (N)</td>
<td>68.31</td>
<td>6.16</td>
<td></td>
</tr>
<tr>
<td>Second Element Drag (N)</td>
<td>50.01</td>
<td>4.56</td>
<td>45.45</td>
</tr>
<tr>
<td>Total Drag (N)</td>
<td>118.32</td>
<td>10.72</td>
<td>107.6</td>
</tr>
</tbody>
</table>

Table 8.16: 2-D Drag Reduction achieved by rotating second elements to DRS position

The final rotational angle was chosen for each element as the position at which overall drag of the 2 elements within the wing was lowest. The full table and CFD analysis produced when obtaining the optimal angle for DRS operation can be found in the Portfolio of Evidence.

The pivot point, about which the second elements of each wing will rotate, was determined to be the point which resulted in the lowest value of total drag when the element was rotated to its minimal drag position determined above. Post processing of the CFD simulation also looked at the moment of the second element of each wing about the pivot point of the element. The moment directly correlates to the torque required by the electro-mechanical servos to rotate the element, and so a pivot point with an excessively high moment will demand a higher power output or even failure of the servo. This was taken into consideration with the drag reduction for each pivot point.
Three simulations of pivot point location were run along the chord length of the element; quarter, half and three quarters chord. The analysis was limited to these three locations to prevent excessive manufacture time and complication. The simulation was again run at 15 m/s. Table 8.17 shows the moment of the second elements and drag of each wing with the optimal position highlighted.

<table>
<thead>
<tr>
<th>Second Element Pivot Point</th>
<th>Position</th>
<th>1/4 Chord</th>
<th>1/2 Chord</th>
<th>3/4 Chord</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Front Wing</strong></td>
<td>Open</td>
<td>7.78</td>
<td>7.44</td>
<td>7.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.51</td>
<td>3.56</td>
<td>3.31</td>
</tr>
<tr>
<td></td>
<td></td>
<td>11.29</td>
<td>11</td>
<td>10.61</td>
</tr>
<tr>
<td></td>
<td></td>
<td>-2.21</td>
<td>-2.48</td>
<td>3.07</td>
</tr>
<tr>
<td></td>
<td>Closed</td>
<td>8.3</td>
<td>8.3</td>
<td>8.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>13.94</td>
<td>13.94</td>
<td>13.94</td>
</tr>
<tr>
<td></td>
<td></td>
<td>22.24</td>
<td>22.24</td>
<td>22.24</td>
</tr>
<tr>
<td></td>
<td></td>
<td>12.6</td>
<td>13.86</td>
<td>15.06</td>
</tr>
<tr>
<td><strong>Side Wing</strong></td>
<td>Open</td>
<td>15.53</td>
<td>16.37</td>
<td>16.28</td>
</tr>
<tr>
<td></td>
<td></td>
<td>11.46</td>
<td>11.01</td>
<td>10.86</td>
</tr>
<tr>
<td></td>
<td></td>
<td>27</td>
<td>27.38</td>
<td>27.14</td>
</tr>
<tr>
<td></td>
<td></td>
<td>90.28</td>
<td>103.87</td>
<td>82.31</td>
</tr>
<tr>
<td></td>
<td>Closed</td>
<td>25.22</td>
<td>25.22</td>
<td>25.22</td>
</tr>
<tr>
<td></td>
<td></td>
<td>15.78</td>
<td>15.78</td>
<td>15.78</td>
</tr>
<tr>
<td></td>
<td></td>
<td>41</td>
<td>41</td>
<td>41</td>
</tr>
<tr>
<td></td>
<td></td>
<td>-142.65</td>
<td>-116</td>
<td>-96.36</td>
</tr>
<tr>
<td><strong>Rear Wing</strong></td>
<td>Open</td>
<td>6.08</td>
<td>6.16</td>
<td>6.46</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4.24</td>
<td>4.56</td>
<td>4.4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>10.32</td>
<td>10.72</td>
<td>10.85</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.02</td>
<td>4.42</td>
<td>-1.99</td>
</tr>
<tr>
<td></td>
<td>Closed</td>
<td>68.31</td>
<td>68.31</td>
<td>68.31</td>
</tr>
<tr>
<td></td>
<td></td>
<td>50.01</td>
<td>50.01</td>
<td>50.01</td>
</tr>
<tr>
<td></td>
<td></td>
<td>118.32</td>
<td>118.32</td>
<td>118.32</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.72</td>
<td>6.28</td>
<td>9.36</td>
</tr>
</tbody>
</table>

Table 8.17: Drag Reduction and Second Element Moment experienced at different Pivot Point locations

The moment of the second element about the pivot was calculated for the element in both the open and closed positions. The moment was calculated by finding the line of pressure of the element, the line along which the force magnitude of lift and drag acts upon the
element, as illustrated in Figure 8.6.1. The force magnitude and the angle at which its acts in relation to the horizontal plane was found using the following equations:

\[ F = \sqrt{L^2 + D^2} \quad \theta = \tan^{-1}\left(\frac{L}{D}\right) \]

*Equation 8.6.1: Force magnitude and angle*

Where \( L \) is the lift force and \( D \) is the drag force of the second element.

The momentum of the element in the two positions was then found in post analysis of the 2-D simulations of the element in Fluent. A datum point was required to find the moment about and so for ease of comparison between the different Fluent solution files this was set as the leading edge of the main plain of each wing. The next stage was to relate this moment to the moment about the pivot point. The following equation was used to relate the two moments, with the components of the equation broken down below:
The full calculation for finding the pivot point can be found in the Portfolio of Evidence.

The results showed in each position instance for the wings there was a trade of between total drag and moment about the pivot point. It can also be seen that this trade of was between the ¼ chord and ¾ chord positions for each of the wings. Based on the calculation findings the pivot location for each wing was set as ¾ chord length. This decision was based on several design specifications:

- The pivot momentum has minimal difference between positions and is within servo operating conditions in each instance and so the decision will be based upon minimal drag induced.
- A greater drag reduction is incurred using ¼ chord length for two of the wings and so to save on design compliance and time, all three wings with use this position.

\[ Pivot \text{ Moment} = F(x_p \cos(\theta - \alpha) + y_p \sin(\theta - \alpha) + x_m \cos \theta + y_m \sin \theta - \frac{M_m}{F}) \]

*Equation 8.6.2: Pivot moment*

\[ F(x_p \cos(\theta - \alpha)) : x - \text{momentum component about pivot} \]

*Equation 8.6.3: x-component about pivot*

\[ F(y_p \cos(\theta - \alpha)) : y - \text{momentum component about pivot} \]

*Equation 8.6.4: y-component about pivot*

\[ F \left( x_m \cos \theta + y_m \sin \theta - \frac{M_m}{F} \right) : \text{momentum magnitude in relation to main plain} \]

*Equation 8.6.5: Momentum magnitude*
8.6.1.3 2-D to 3-D Conversion

The 2-D results gained are converted to 3-D to give a better representation of the performance effects gained by the car using the DRS. Drag and lift forces experience a number of subsequent component factors on a solid 3-D wing form over that of a 2-D wing profile. The effects of wing endplates, designed to reduce wing tip vortices and so reduce drag, must be included within the conversion to determine realistic drag values of the wings. This is also true for the types of drag produced by the 3-D wing form. 2-D simulation of the wing profile will exclusively take into account viscous drag of the fluid flow; however, the 3-D wing form will also experience induced drag second to the viscous drag. The final fluid flow characteristic that must be accommodated for is the ground effect for the wings positioned close enough to the floor for the effect to occur. For the case of the wings in this conversion, ground effect will only occur for the front and side wings. The following equations are used to encounter these considerations in the conversion as discussed by Joseph Katz [105].

- **2-D Drag and Lift Coefficients:**

  \[
  2D \, C_l = \frac{2l}{\rho v^2 c} ; \quad 2D \, C_d = \frac{2d}{\rho v^2 c}
  \]

  *Equation 8.6.6: 2-D drag and lift coefficients*

  where \( c \) is the chord length of the wing.

- **Aspect Ratio of a wing without endplates:**

  \[
  AR = \frac{b}{c}
  \]

  *Equation 8.6.7: Aspect ratio*
where \( b \) is the wing span.

- **Effective Aspect Ratio of a wing to include effects of endplates:**

\[
AR_e = AR \times \left(1 + 1.9\frac{h}{b}\right)
\]

*Equation 8.6.8: Effective aspect ratio*

where \( h \) is the height of the endplate.

- **Induced drag experienced by a wing, inclusive of its elements.** ‘\( e \)’ is the efficiency factor of the wing’s lift distribution, this is taken as 1 for an elliptical wing:

\[
Induced Drag, IC_d = \frac{C_l^2}{\pi eAR_e}
\]

*Equation 8.6.9: Induced drag*

- **3-D Drag and Lift Coefficients for wings not experiencing ground effect:**

\[
3D \ C_l = 2D \ C_l \ ; \quad 3D \ C_d = 2D \ C_d + IC_d
\]

*Equation 8.6.10: 3-D drag and lift coefficients*

- **3-D Drag and Lift Coefficients for wings not experiencing ground effect.** The constant for ground effect factor is taken from the for ground effect of a dual element elliptical wing as illustrated by Katz.

\[
3D \ C_l = GC_l3D \ ; \quad C_d = \frac{(2D \ C_d + IC_d)}{G}
\]

*Equation 8.6.11: 3-D drag and lift coefficients for wings not in ground effect*

Table 8.18 below shows the results of the 3-D conversion and a comparison between the 3-D drag of the wings when the elements are in their closed position and when the DRS is
active. This highlights the drag reduction achievable by the system when minimal drag is
desired.

<table>
<thead>
<tr>
<th>2D Drag</th>
<th>2D Lift</th>
<th>2D Cd</th>
<th>2D Cl</th>
<th>3D Cd</th>
<th>3D Cl</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front Wing</td>
<td>Open</td>
<td>10.610</td>
<td>196.460</td>
<td>0.123</td>
<td>2.277</td>
</tr>
<tr>
<td></td>
<td>Closed</td>
<td>22.240</td>
<td>415.110</td>
<td>0.258</td>
<td>4.812</td>
</tr>
<tr>
<td>Side Wing</td>
<td>Open</td>
<td>27.140</td>
<td>439.270</td>
<td>0.266</td>
<td>4.302</td>
</tr>
<tr>
<td></td>
<td>Closed</td>
<td>41.000</td>
<td>543.710</td>
<td>0.401</td>
<td>5.324</td>
</tr>
<tr>
<td>Rear Wing</td>
<td>Open</td>
<td>10.850</td>
<td>212.450</td>
<td>0.093</td>
<td>1.818</td>
</tr>
<tr>
<td></td>
<td>Closed</td>
<td>118.320</td>
<td>333.540</td>
<td>1.012</td>
<td>2.854</td>
</tr>
</tbody>
</table>

Table 8.18: DRS 2-D to 3-D hand calculated conversion

To validate the 3-D hand calculated conversion 3-D simulation of the full car aero package
was carried out for both instances of the second wings’ element position, open and closed.
Both the hand calculated conversion and the 3-D simulation were performed for a half car
model of the car. This enabled reduced computational time. Table 8.19 below shows a
comparison between the sets of results.

<table>
<thead>
<tr>
<th>Converted</th>
<th>Simulated</th>
</tr>
</thead>
<tbody>
<tr>
<td>3D Cd</td>
<td>3D Cl</td>
</tr>
<tr>
<td>Front Wing</td>
<td>Open</td>
</tr>
<tr>
<td></td>
<td>Closed</td>
</tr>
<tr>
<td>Side Wing</td>
<td>Open</td>
</tr>
<tr>
<td></td>
<td>Closed</td>
</tr>
<tr>
<td>Rear Wing</td>
<td>Open</td>
</tr>
<tr>
<td></td>
<td>Closed</td>
</tr>
</tbody>
</table>

Table 8.19: Comparison between 3-D converted and simulated data
Analysis of the results show a substantial difference between the 3-D hand calculated drag and lift coefficients to the simulated coefficients. This difference can be associated to errors and tolerances within the 3-D simulations; however, the result show a correlation between the simulated results and the hand calculations and so indicates a scaling factor will need to be applied. This analysis also emphasises the importance of physical testing and the need for physical system calibration.

8.6.1.1.4 DRS Control

As specified in the preliminary design specifications the DRS will be controlled by the driver via an input on the steering wheel. The system must account for interference and coherence with other systems using the same hardware, preventing failure of the DRS and other systems. This coherence is discussed in 8.6.2.2. Using the above design and development of the DRS, activation of the system via the steering input will result in the electro-mechanical servos simultaneously rotating the secondary elements of the three wings to their set ‘DRS Open’ position. Release of the driver input will result in the reversed rotation of the servos,
returning the elements to their ‘DRS Closed’ position. To operate this system a Matlab Simulink case was created to control the flow of data from the input signal to the resultant output signal. The case is shown below in Figure 8.6.2.

Activation, control and driver user input (UI) of the Simulink case is discussed in 9.5.1.3. Figure 8.6.2, illustrates the activation procedure of the system as a result of driver input using an ‘Enabled Subsystem’. Once enabled the subsystem proceeds to send data to a conversion block which translates the signal to a voltage output that corresponds to a specific servo rotational output required for the DRS. Figure 8.6.2 also highlights a secondary system input. This secondary input is a safeguard design to activate the DRS upon excessive maximum suspension travel required in the instance of excessive load acting through the suspension resulting in bottoming out of the car.

### 8.6.2 Active Suspension and Active Aerodynamics

Further to the developed DRS, the Active Aerodynamics system utilises the same hardware and theoretical aerodynamic principles to manipulate the forces created within the fluid flow for improved active vehicle dynamics. An extension of the DRS operation, the Active Aerodynamics system expands the control of the aerodynamic behaviour of the car, creating a greater database of force attributes available with the rotation of the wing elements. Twinned with the development of the Active Suspension system, a fully automatic vehicle dynamic control model can be created. The Active Suspension system, using the same sensor input as that of the Active Aerodynamic system, acts as a secondary system to the current suspension design of the car developed in employing hardware to exert external force to the suspension to improve the sprung mass damping and control.
8.6.2.1.1 Vehicle Model

Prior to the development of the full active system the key design specifications and assumption must be considered:

1. The active system is designed for an open wheel track racing car and so its suspension is primarily designed for improved vehicle performance, reducing negative vehicle dynamics over disturbance experienced by the road surface, which for a race track are considered negligible.

2. Optimal performance of the car will be experienced when the car experience minimal roll, pitch and yaw, enabling increased cornering speed, vehicle handling and aerodynamic consistency with constant ground effect factors.

3. The suspensions design of the vehicle is considered to have components positioned in a parallel, linear manner at the corners of the vehicle.

![Active System vehicle model](image-url)
The full design specifications for the system are discussed in 0.

Using the above specifications and assumptions the dynamic force model of the car can be created. The assumption that the car experiences negligible road disturbances and the systems is required to primarily accommodate sprung mass dynamics enables exemption of the unsprung mass and related forces from the model. This will result in a reduce number of variables for the system to compute and so ultimately increase its robust control. The physical system input locations are also considered, with the location of the accelerometers and sensors relation to the suspension units included within the vehicle model to ensure the most accurate representation of the physical car as possible.

Figure 8.6.3 illustrates the vehicle model with all vehicle dynamics, displacement and forces related to the global coordinate system. It also shows the axis about which the moments of area for roll, pitch and yaw act. The centre of gravity (CoG) of the vehicle is related to the front axle and front suspension units by length ‘a’, to the rear axle and rear suspensions units by length ‘b’ and to the ground plane by length ‘h’. The car track is denoted by ‘w’, also taken as the distance between the left and right suspension units. The vehicle model also highlights the aerodynamic force locations as well as the forced actuators added to each suspension unit used for output control of the active suspension. The hardware for the full system is discussed in 9.5.1.

8.6.2.1.1.1 Inputs

The system comprises of 15 inputs, 6 of which are exogenous and directly recorded by the vehicle sensors, whilst the other 9 are controllable inputs that act as a result of the system command:
Exogenous Inputs:

- Lateral and Longitudinal acceleration of the vehicle’s sprung mass acting at the CoG of the vehicle.
- Vehicle sprung mass displacement at each corner, parallel to suspension unit displacement.

Controllable Inputs:

- Suspension unit linear actuator forces acting at each corner
- Aerodynamic forces acting at 5 locations of the secondary wing elements.

8.6.2.1.1.2 Outputs

The state-space equations used for the computational control of the system have the ability to output an extensive set of outputs, however, for a robust and accurate control of the full system 9 specific outputs are chosen for system weighting and dynamic adaption.

- Deflection of vehicle suspension springs within each suspension unit.
- Deflection of anti-roll bars within each suspension unit.
- Vertical velocity of the vehicle sprung mass acting at the CoG.
- Pitch and Roll Angles of the vehicle sprung mass at the CoG.
- Lateral and Longitudinal acceleration of the vehicle sprung mass acting about the CoG.

8.6.2.1.1.3 Component Linearisation

To allow for modelling of the full vehicle dynamic system with constant connectivity between components, in particularly in the suspension unit, a degree of component linearisation must be assumed. In reality many of the components, including the suspension
springs, dampers and forced actuators operate with slight deviations from linear behaviour outside a certain operating band. Due to the operating conditions of the vehicle and the components in use, designed for high performance racing processes, the conditions under which the components are used falls within the linear operating band [106] and so the following assumptions are accepted:

- All suspensions springs, dampers and force actuators are linear.
- All linear spring have equilibrium displacements.

8.6.2.1.4 State System Equations

The linearisation of the system as previously discussed allows for state-space representation of the system model and so the following fundamental first order differential equation can be used, derived from the vehicle model illustrated in 8.6.2.1.1. The equations follow the form below:

$$\dot{X} = AX + BU$$

*Equation 8.6.12: Steady-state equations*

In conjunction with a matrix based control system, the system model will use the 9 state variable equations to calculate the required external force input values of the controllable input components.

Rate of anti-roll bar deflections:

$$\dot{x}_{RBR} = w \left( \frac{W_R}{J_R} \right)$$

$$\dot{x}_{RBF} = w \left( \frac{W_R}{J_R} \right)$$

*Equation 8.6.13: Rate of anti-roll bar deflections*
Rate of suspension spring deflections:

\[ \dot{x}_{SRR} = V_{RR} - \frac{z_{CG}}{m_s} - b \left( \frac{W_p}{J_p} \right) + \frac{w}{2} \left( \frac{W_R}{J_R} \right) \]

\[ \dot{x}_{SRL} = V_{RL} - \frac{z_{CG}}{m_s} - b \left( \frac{W_p}{J_p} \right) - \frac{w}{2} \left( \frac{W_R}{J_R} \right) \]

\[ \dot{x}_{SFR} = V_{FR} - \frac{z_{CG}}{m_s} + a \left( \frac{W_p}{J_p} \right) + \frac{w}{2} \left( \frac{W_R}{J_R} \right) \]

\[ \dot{x}_{SFL} = V_{FL} - \frac{z_{CG}}{m_s} + a \left( \frac{W_p}{J_p} \right) - \frac{w}{2} \left( \frac{W_R}{J_R} \right) \]

Equation 8.6.14: Rate of suspension spring deflection

Vertical Moment at CoG:

\[ \dot{z}_{CG} = -A F_{RS} - F_{RR} - (x_{SRR} \times K_{SRR}) + B_{SRR} \left[ -\frac{z_{CG}}{m_s} + V_{RR} - b \left( \frac{W_p}{J_p} \right) + \frac{w}{2} \left( \frac{W_R}{J_R} \right) \right] - A F_{LS} \]

\[ - F_{RL} - (x_{SRL} \times K_{SRL}) + B_{SRL} \left[ -\frac{z_{CG}}{m_s} + V_{RL} - b \left( \frac{W_p}{J_p} \right) - \frac{w}{2} \left( \frac{W_R}{J_R} \right) \right] - A F_{FR} \]

\[ - F_{FR} - (x_{SFR} \times K_{SFR}) + B_{SFR} \left[ -\frac{z_{CG}}{m_s} + V_{FR} + a \left( \frac{W_p}{J_p} \right) + \frac{w}{2} \left( \frac{W_R}{J_R} \right) \right] - A F_{FL} \]

\[ - F_{FL} - (x_{SFL} \times K_{SFL}) + B_{SFL} \left[ -\frac{z_{CG}}{m_s} + V_{FL} + a \left( \frac{W_p}{J_p} \right) - \frac{w}{2} \left( \frac{W_R}{J_R} \right) \right] - A F_{R} \]

Equation 8.6.15: Vertical moment at CoG
Pitch Angular Momentum at CoG:

\[ \dot{W}_p = h(F_{PITCH}) \]

\[ + b \left[ -F_{RR} - (x_{SRR} \times K_{SRR}) + (x_{RBR} \times K_{RR}) \right] \]

\[ + B_{SRR} \left[ -\frac{z_{CG}}{m_s} + V_{RR} - b \left( \frac{W_p}{J_p} \right) + \frac{w}{2} \left( \frac{W_R}{J_R} \right) \right] \]

\[ + b \left[ -F_{FL} - (x_{SRL} \times K_{SRL}) - (x_{RBR} \times K_{RR}) \right] \]

\[ + B_{SRL} \left[ -\frac{z_{CG}}{m_s} + V_{RL} - b \left( \frac{W_p}{J_p} \right) - \frac{w}{2} \left( \frac{W_R}{J_R} \right) \right] \]

\[ + a \left[ F_{FR} + (x_{SFR} \times K_{SFR}) - (x_{RBF} \times K_{RF}) \right] \]

\[ - B_{SFL} \left[ -\frac{z_{CG}}{m_s} + V_{FR} + a \left( \frac{W_p}{J_p} \right) + \frac{w}{2} \left( \frac{W_R}{J_R} \right) \right] \]

\[ + a \left[ F_{FL} + (x_{SFL} \times K_{SFL}) + (x_{RBF} \times K_{RF}) \right] \]

\[ - B_{SFR} \left[ -\frac{z_{CG}}{m_s} + V_{FR} - a \left( \frac{W_p}{J_p} \right) - \frac{w}{2} \left( \frac{W_R}{W_R} \right) \right] + (AF_{FR} \times C_{FR}) + (AF_{FL} \times C_{FL}) \]

\[ + (AF_{RS} \times C_{RS}) + (AF_{LS} \times C_{LS}) - (AF_R \times C_R) \]

*Equation 8.6.16: Pitch angular momentum at CoG*

Roll Angular Momentum at CoG:
\[ W_R = h(F_{ROLL}) \]
\[ + \frac{w}{2} \left[ +F_{RR} + (x_{SRR} \times K_{SRR}) - (x_{RBR} \times K_{RR}) \right] \]
\[ - B_{SRR} \left[ - \frac{z_{CG}}{m_s} + V_{RR} - b \left( \frac{W_p}{J_p} \right) + \frac{w}{2} \left( \frac{W_R}{J_R} \right) \right] \]
\[ + \frac{w}{2} \left[ -F_{FL} - (x_{SRL} \times K_{SRL}) - (x_{RBR} \times K_{RR}) \right] \]
\[ + B_{SRL} \left[ - \frac{z_{CG}}{m_s} + V_{RL} - b \left( \frac{W_p}{J_p} \right) - \frac{w}{2} \left( \frac{W_R}{J_R} \right) \right] \]
\[ + \frac{w}{2} \left[ F_{FR} + (x_{SFR} \times K_{SFR}) - (x_{RBF} \times K_{RF}) \right] \]
\[ - B_{SFL} \left[ - \frac{z_{CG}}{m_s} + V_{FR} + a \left( \frac{W_p}{J_p} \right) + \frac{w}{2} \left( \frac{W_R}{J_R} \right) \right] \]
\[ + \frac{w}{2} \left[ -F_{FL} - (x_{SFL} \times K_{SFL}) - (x_{RBF} \times K_{RF}) \right] \]
\[ + B_{SFL} \left[ - \frac{z_{CG}}{m_s} + V_{FL} + a \left( \frac{W_p}{J_p} \right) - \frac{w}{2} \left( \frac{W_R}{W_R} \right) \right] + (AF_{FR} \times D_{FR}) \]
\[ - (AF_{FL} \times D_{FL}) + (AF_{RS} \times D_{RS}) - (AF_{LS} \times D_{LS}) \]

*Equation 8.6.17: Roll angular momentum at CoG*

The Matlab written control system, as discussed in 8.6.2.1.3, requires the state-space equations in matrix form for use within the Simulink case file and so below the Matlab code written to convert the 9 equations is shown:
%% Matrix A: 11 States; Xsfr, Xsfl, Xsrr, Xsrl, Xrbf, Xrbr, Zcg, Wp, Wr, Tp, Tr

% Xsfl = Front Left Spring Deflection
% Xsfr = Front Rear Spring Deflection
% Xsrl = Rear Left Spring Deflection
% Xsrr = Rear Right Spring Deflection
% Xrbf = Front Anti-Roll Bar
% Xrbr = Rear Anti-Roll Bar
% Zcg = CG Vertical Displacement
% Wp = Pitch Angular Momentum
% Wr = Roll Angular Momentum
% Tp = Pitch
% Tr = Roll

A(1,:) = [0,0,0,0,0,0,-1/Ms,((1/Jp)*a),((1/Jr)*(w/2)),0,0];
A(2,:) = [0,0,0,0,0,0,-1/Ms,((1/Jp)*a),((1/Jr)*(w/2)),0,0];
A(3,:) = [0,0,0,0,0,-1/Ms,((1/Jp)*b),((1/Jr)*w),0,0];
A(4,:) = [0,0,0,0,0,-1/Ms,((1/Jp)*b),((1/Jr)*w),0,0];
A(5,:) = [0,0,0,0,0,0,0,0,((1/Jr)*w),0,0];
A(6,:) = [0,0,0,0,0,0,0,0,((1/Jr)*w),0,0];
A(7,:) = [(1*Ksfr),(1*Ksfl),(1*Ksrr),(1*Ksrl),0,0,((1/Ms)*Bsfr)-((1/Ms)*Bsfl)-((1/Ms)*Bsrr)-((1/Ms)*Bsrl),((1/Jp)*a*Bsfr)+((1/Jp)*a*Bsfl)-((1/Jp)*b*Bsrr)-((1/Jp)*b*Bsrl),((1/Jr)*(w/2)*Bsfl)+((1/Jr)*(w/2)*Bsrr)-((1/Jr)*(w/2)*Bsrl),0,0];
A(8,:) = [-(1*Ksfr*a),-(1*Ksfl*a),((1/Ksfr*b)+(1/Ksfl*b)),((1/Ksrr*b)+(1/Ksrl*b)),((1/Ksfr)*(w/2)),((1/Ksfl)*(w/2)),((1/Ksrr)*(w/2)),((1/Ksrl)*(w/2)),(1*Krf*a)+(1*Krf*b),-(1*Krf*a)+(1*Krf*b)];
A(9,:) = [-(1*Ksfr*(w/2)),-(1*Ksfl*(w/2)),-(1*Ksrr*(w/2)),-(1*Ksrl*(w/2)),-(1*Ksfr*(w/2)),-(1*Ksfl*(w/2)),-(1*Ksrr*(w/2)),-(1*Ksrl*(w/2)),((1/Ksfr)*(w/2)),((1/Ksfl)*(w/2)),((1/Ksrr)*(w/2)),((1/Ksrl)*(w/2)),(1*Krf*a)+(1*Krf*b),-(1*Krf*a)+(1*Krf*b)];
A(10,:) = [0,0,0,0,0,0,0,0,0,0];
A(11,:) = [0,0,0,0,0,0,0,0,0,0];

%% Matrix B: Inputs; Fpitch, Froll, Vfr, Vfl, Vrr, Vrl, Ffr, Ffl, Frr, Frl, AFfr, AFfl, AFrs, AFls, AFr

% Epitch = Pitch Force acting on cg
% Eroll = Roll Force acting on cg
% Vfr = Front Right Velocity Input
% Vfl = Front Left Velocity Input
% Vrr = Rear Right Velocity Input
% Vrl = Rear Left Velocity Input
% Ffr = Front Right Controlled Actuator Input
% Ffl = Front Left Controlled Actuator Input
% Frr = Rear Right Controlled Actuator Input
% Frl = Rear Left Controlled Actuator Input
%AFr = Front Right Aero Input
%AFl = Front Left Aero Input
%Afr = Right Side Aero Input
%Afl = Left Side Aero Input
%Afr = Rear Aero Input

B(1,:) = [0,0,0,0,0,0,0,0,0,0,0,0,0,0,0];
B(2,:) = [0,0,0,0,0,1,0,0,0,0,0,0,0,0,0];
B(3,:) = [0,0,0,1,0,0,0,0,0,0,0,0,0,0,0];
B(4,:) = [0,0,0,0,1,0,0,0,0,0,0,0,0,0,0];
B(5,:) = [0,0,0,0,0,0,0,0,0,0,0,0,0,0,0];
B(6,:) = [0,0,0,0,0,0,0,0,0,0,0,0,0,0,0];
B(7,:) = [0,0,(1*Bsfr),(1*Bsfl),(1*Bsrr),(1*Bsrl),-1,-1,-1,-1,-1,-1,-1,-1];
B(8,:) = [(1*h),0,0,0,0,0,0,0,0,0,0,0,0,0,0];
B(9,:) = [0,(1*h),0,0,0,0,0,0,0,0,0,0,0,0,0];
B(10,:) = [0,0,0,0,0,0,0,0,0,0,0,0,0,0,0];
B(11,:) = [0,0,0,0,0,0,0,0,0,0,0,0,0,0,0];

%% Matrix C & D: Output Matrices
C=eye(11);
D=zeros(11,15);

sys=ss(A,B,C,D);

Figure 8.6.4: State-space representation in matrix form for Matlab use

Figure 8.6.4 also shows the input matrix, B, and the output matrices C and D. The four matrices complete the state-space system.

8.6.2.1.2 Control System Development

8.6.2.1.2.1 LQR Control Strategy
The controller selected for the active system is a Linear Quadratic Regulator (LQR). This controller is specifically designed for time-dependant systems with multiple inputs and outputs, where optimal control of the system allows for direct formulation of the performance objectives [107]. Although LQR control strategy doesn’t have the ability to adapt to road profiles and has low robust control with a high number of states [108], for this system, where road disturbance is negligible and there are minimal states, an LQR will prove...
the mist affective for improved performance. A further two features of LQR control strategy make it the best solution for this system:

- The control strategy functions best with linear model states
- The control strategy works on the foundation of optimising or minimising an objective function, whilst minimising the energy input required.

These features enable the setup of the system to minimise specific outputs, in this case, the displacement and momentums of roll, pitch and heave as previously discussed. It also ensures minimal input energy and force from the controllable inputs is required, in turn reducing the weight penalty of the system.

8.6.2.1.2.1.2 Stability

The stability of a system is considered the ability of the system to return to equilibrium when one component is displaced [107]. For the case of an active system such as this, the displacement is represented by the deflection of one suspension unit at the corner of the vehicle and the stability of the system is measured as the system's ability to return the chassis to an equilibrated state. To determine if the system is stable a plot of system Eigenvalue poles is plotted. The system is considered stable if there are no positive poles.
Figure 8.6.5 shows the Eigen-value plot of the system’s stability. As illustrated, all poles are negative and so the system can be classified as stable.

8.6.2.1.2.1.3 Controllability

The controllability of the system determines the number of system states over which it has control. Full optimal control, the controllability of the system must have control over all states. If less than all states are controlled then the effectiveness of the system is limited. The controllability of the system is found by computing a controllability matrix and control rank. The Matlab function, Rank ctrb() is used to find the controllability rank and so the number of controllable states of the system. For this system it was found that all 11 states were fully controllable.
LQR Controller Design

LQR control is based upon the use of a gain matrix, \( K \), to supply the controllable inputs with the require data to optimise the performance of the vehicle model. The gain is derived from the state-space equation in 8.6.2.1.1.4 and the optimal control law for the gain matrix is shown below.

\[
U = -K \times X
\]

*Equation 8.6.18: Optimal control law for the gain matrix*

A Quadratic Cost Function is used to generate the gain matrix data, where \( K \) is used to minimise the cost function, \( J \). Matrices \( Q \) and \( R \), known as the weighting matrix and cost control matrix respectively, are used as penalisation matrices to determine the priority and weighting of the system states and inputs.

\[
J = \int_{0}^{T} (x^T R x + u^T Qu) dt
\]

*Equation 8.6.19: Quadratic cost function*

The values of the horizontal components in matrices \( Q \) and \( R \) are based on all vehicle parameters, in particularly the suspension unit components’ coefficients, and so with the current values of BR-16 entered the penalisation and weighting of different states and inputs can be determined using trial and error to produce an optimal response to simulation track data. The final weighting and cost control matrices based upon BR-16 are shown below:
Initially all the diagonal values of the weighting matrix, \( Q \) were set to 1 with a number of iterations were simulated until the weighting of the priority states was found. As the system is designed for an open wheel racing car, roll and pitch angles and momentums were considered the highest priority states to weight, with the heave and corner displacements lesser. The difference between the states is dependent on the vehicle parameters and how when passive it handles whilst cornering, braking and accelerating. The off diagonal terms represent combinations of states and inputs within the state-space equations and so are set to 0.
The first 6 terms of the cost control matrix, R, represent the exogenous inputs of the system. As the controller has no control over these inputs they are penalised by 6 orders of magnitude to prevent the system attempting to use them for performance optimisation. The remaining 9 diagonal terms represent the controllable inputs; forced actuators and aerodynamic forces. The actuator forces are all set to 1 so that the system uses the maximum available input from each. The aerodynamic forces are each set with a variable. This is variable is primarily set to 1. However, rather than a constant, these inputs are set as variables so that the penalisation of the aerodynamic forces can be altered for coherence with other systems. During operation of the DRS, no positive or negative aerodynamic forces are available and so these inputs are penalised to prevent the system controller relying on the aerodynamics for aid in the improved performance of the vehicle dynamics. As DRS is primarily operational when the vehicle is traveling in a straight line, the system will not experience cornering forces and so the requirement for external force input is considered less and achievable by the 4 suspension actuators. System integration is discussed further in 8.6.2.2.

8.6.2.1.3 Active System Software

The full active system was modelled in Matlab Simulink, with subsequent control files written for Simulink model activation and dependency. The LQR controlled active suspension and active aerodynamic Simulink model is shown in Figure 8.6.6. The model is based around the state-space function block, with system inputs to the left of this function block and outputs to the right. The LQR control is situated within the output-input cycle of the controllable inputs. The gain matrix, as previously discussed, is applied to the
controllable inputs within the LQR function block, which then supplies the subsequent model iteration with the required inputs for performance optimisation.

The primary exogenous inputs are displayed as external inputs within the Simulink model, which are then paired to the respective input terminals receiving data from the vehicle accelerometers and suspension damper pots. Data from the controllable inputs, post adaption by the LQR gain matrix, is output from the Simulink model and sent to an external model that outputs a voltage signal to the respective forced actuators and aerodynamic

*Figure 8.6.6: LQR Controlled Active System Simulink Model*
hardware to enable the physical input of the required force corrections at each location. All input and output data is output to the Matlab workspace for system tracking and on track telemetry during testing.

For system activation and function block dependency, a Matlab control file was written. This file initiates the system Simulink model and supplies the different internal function blocks, including the state-space function block, with the appropriate model data such as the state-space equations and vehicle parameters. This control file is shown in Figure 8.6.7.

```matlab
%Load Parameters
run 'Parameters.m'

%Load Test Data
run 'testdata.m'

%State-Space Equations
run 'StateSpace.m'

%open model
model = 'LQRModel';
open_system(model);

%LQR Controller Gain Design

%State Weighting
Q=eye(length(A));
Q(1,1) = 1;
Q(2,2) = 1;
Q(3,3) = 1;
Q(4,4) = 1;
Q(5,5) = 100;
Q(6,6) = 100;
Q(7,7) = 1000;
Q(8,8) = 1000;
Q(9,9) = 1000;
Q(10,10) = 10000000000;
Q(11,11) = 30000000000;

%Input Weighting
R=eye(length(B));
R(1,1) = 100000;
R(2,2) = 100000;
R(3,3) = 100000;
R(4,4) = 100000;
R(5,5) = 100000;
R(6,6) = 100000;
R(7,7) = 1;
R(8,8) = 1;
R(9,9) = 1;
```
\begin{verbatim}
R(10,10) = 1;
R(11,11) = 1;
R(12,12) = 1;
R(13,13) = 1;
R(14,14) = 1;
R(15,15) = 1;

% Determine QLR Gain
[K,S,e] = lqr(A,B,Q,R)

% run simulation
sim(model)
\end{verbatim}

\textit{Figure 8.6.7: LQR Controlled Active System Matlab Control File}

\subsection*{8.6.2.1.3.1.1 Model Testing}

The system model was tested using the current design iteration parameters of BR-16 and data of different competition track events, recorded using IPG simulations as discussed in 8.7. The state penalisation and weighting was set to the matrices previously shown and the optimal performance gain was simulated. The Matlab control file was edited to produce graphical information of the system over data of the system ran without LQR gain and so in a passive state. The simulation was run using data of the skid pad event. This event is discussed in 1.2, and was chosen for the model testing as it test the extremes of vehicle sprung mass role and so the reduction in role with the implementation of the system. The results are displayed below:
Figure 8.6.8 shows the system successfully reduces the roll angle experienced by the vehicle up to 83% as it is driven at the limit of lateral acceleration on the skid pad event. This result shows that with the reduction in roll angle experienced, the vehicle will be able to take the 360° turns of the skid pad event at a higher speed and so with higher lateral acceleration over the passive vehicle without the full active system. Although the vehicle experiences minimal pitch change during the ski pad event as the vehicle is constantly acceleration, Figure 8.6.9 shows the pitch experienced is also greatly reduced by the LQR control strategy up to 50%.

Figure 8.6.8: LQR against Passive system Vehicle Roll Angle
To understand how the system uses the physical components integrated to achieve these results, graphs of force actuator usage and aerodynamic force usage are produced, displayed in Figure 8.6.11 and Figure 8.6.10. The graphs show, as expected, the opposing force output of the component located on opposing sides of the car. They also show the maximum force requirement of the components during the skid pad event and this data will be used to select the appropriate off-the-shelf (OTS) components, as discussed in 9.5.1.

Figure 8.6.9: LQR against Passive system Vehicle Pitch Angle
Figure 8.6.10: LQR system Force exerted by Aerodynamic Components during Skid Pad Simulation
8.6.2.1.3.1.2 Data Output to Hardware

The controllable input data for the required force exerted into the system at each forced actuator and aerodynamic force location is output in the form of force values (N). This data must be converted into appropriate signal data that can be registered by the system hardware for particular force production. Simulink models are created for the two controllable input systems, with signal conversion and quantifiable outputs for the hardware components.

![Figure 8.6.12: Forced Actuator Data conversion model](image1)

![Figure 8.6.13: Front Wing Aerodynamic Components Data conversion model](image2)

The forced actuator control model uses a simple function, converting force input data to the corresponding voltage data that is used to control the pneumatic solenoid valves, resulting
in the require pressure within the actuator cylinder and so the output force required. The system hardware is discussed in greater depth in 9.5.1.

The aerodynamic force control model requires greater provision than the forced actuator model, with a greater number of factors affecting the force produced by the wing elements. The model uses a number of look-up graphs to find the necessary angle of the secondary wing element that produces the required force as the current velocity of the vehicle. Calculating the data to produce these look-up graphs is discussed in 8.6.2.1.3.2. Once the necessary element angle is obtained, the output data follows a similar path to that used in the forced actuator model, using a conversion function to attain servo operating voltage for the particular force input. However, the design of the wing element control requires a further step to ensure reliable operation. As a result of constant use of the servos to control the angle of the wing, tracking of the servo position must incorporated into the model to calculate the required operational rotation of the servo at each time step from its current position, as data received from the look-up graphs is output as an angle from the servo rest position. A Simulink memory function block is used to achieve this, inputting the previous time step’s position output as a zero position for the current time step for the model to calculate the required rotation of the servo.

8.6.2.1.3.2 Active Aerodynamics Force Calibration

In order to utilise the active aerodynamics in the full active system, the force produced by the active wing elements for the different vehicle velocities at the available rotational orientations must be calculated. This will allow the system to output the appropriate signal to achieve the required force from that location. Due to the number of instances that can be
achieved with the multiple positions of each wing operating independently of one another, 2-D rather than 3-D simulations will be run for each wing at incremental velocities and element orientations to reduce computational time. The results gained from 2-D will follow the same conversion as discussed in 8.6.1.3 to gain 3-D data plots. The front wing 2-D and converted 3-D data is shown in Table 8.20 and graphically in Figure 8.6.14, Figure 8.6.15, Figure 8.6.16 & Figure 8.6.17.
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Table 8.20: Front Wing Active Aerodynamics 2-D simulated results and 3-D conversion
Major Group Project: Formula Student Vehicle Team

**Figure 8.6.14:** Front Wing 2-D Drag Coefficients at multiple element angles and vehicle velocities

**Figure 8.6.15:** Front Wing 2-D Downforce Coefficient at multiple element angles and vehicle velocities
Figure 8.6.16: Front Wing 3-D converted Drag Coefficients

Figure 8.6.17: Front Wing 3-D converted Downforce Coefficients
As previously discussed in 8.6.1.1.3 the 3-D hand calculated results proved to be out of scale to the 3-D simulated results and so although the above 3-D data gives a representation of the look-ups graphs used by the system to calculate the angle of the wing element required for the request force input, the system will need to be physically calibrated. Testing will be undertaken with the car run in a straight line and all 1000+ instances of wing element orientation across the three wings will be calibrated.

8.6.2.2 DRS and Active System Integration

With multiple systems using the same hardware and acting simultaneously, integration of the system is highly important to ensure coherence between them, preventing clashing signals and system failure. It is also essential to consider integration design for system setup, activation and user input / feedback.

Integration of the DRS and active system will incorporate the use of a Graphical User Interface (GUI), system modes, activation procedures and system operating conditions with fail safes. The complete system will include different system modes, allowing the control of particular operations and so specific performance goals, critical for the different competition dynamic events. The modes will also allow for testing procedures of the car, controlled easily with the steering wheel UI and so allowing for immediate system diagnosis in any circumstance. The system modes will be selected and activated through the GUI. The GUI, shown in Figure 8.6.18, is created within Matlab and enables speed and ease of use of the system. The GUI will also be translated into a format readable by the GEMS display, which will be the focal component for driver UI within the car. This is discussed further in 9.5.1.3.
The GUI also incorporates driver feedback, with mode selection, DRS activation, and graphical display of forced actuator usage and wing element orientation. These features will also be transmitted to the team engineers via the on board telemetry system.

To ensure coherent system control, combined Simulink models for the different system modes were created. These models include the integrative design activations and fail safes between the operations of the system commands. The complete model and subsequent system mode models are shown below:
Figure 8.6.19: Full Active System Simulink Model
Figure 8.6.20: Suspension Test Simulink Model

Figure 8.6.21: Aerodynamics Test Simulink Model

Figure 8.6.22: DRS Simulink Model
8.6.3 Secondary Aerodynamic Components

8.6.3.1 Engine Cover Development

As an aerodynamic component that has not been utilised within an aerodynamic package by Brunel Racing before, no physical, internal comparable data is available for the development of an engine cover. This means design is dependant purely on CFD analysis, and physical testing will be implemented only when the design can prove to be have a significance great enough to justify the cost and time implications of the design and its manufacture. However, the development of an engine cover within this report will continue from that of designs developed for BR-XV by John Sharpe. Design concepts and the accompanying results produced within the report for BR-XV will be discussed with developments made where applicable for performance optimisation of BR-16.

8.6.3.1.1 Conceptual BR-XV Engine Cover Review

In the development of the aerodynamic package of BR-XV, analysis of the full car model highlighted a major area of drag production behind the driver’s helmet and rear roll hoop. It showed the production of large vortices and high turbulence in the wake of the helmet resulting in decreased rear wing performance has turbulent flow passing over the leading edge of the main plain. The analysis further illustrated separation of the flow as segments of the flow stream were refracted into the car engine bay. The full car CFD analysis of BR-XV is shown in Figure 8.6.23 and Figure 8.6.24.
The conclusions drawn from the analysis enabled the initial design directions of the engine cover. Two initial concepts were produced, a half-height and a full-height engine cover. The full car was then re-simulated with these initial concepts to determine the highest performing design route. The simulations can be seen below in Figure 8.6.25 and Figure 8.6.26.
The results showed that the full height engine cover was more effective in both aspects of drag reduction and flow filtering for improved performance of the rear wing. Subsequently the full height design was taken forward and developed. It was established that one area that could be vastly improved with development was the directing and filtering of the fluid flow in front of the rear wing leading edge.

Developments of the design let to the final concept that utilised a direction vane or fin, incorporation with the full height cover to successfully increase the performance of the rear

*Figure 8.6.25: Velocity Streamline Plot of BR-XV Half-height Engine Cover*  

*Figure 8.6.26: Velocity Streamlines Plot of BR-XV Full-height Engine Cover*
wing, whilst dramatically reducing the drag production at the rear of the car. The streamline results of the final design are shown in Figure 8.6.27.

The extent of improvement made with development of the full height engine cover and incorporation of the fin design is shown in Table 8.16. It can be seen that the second iteration and use of a fin dramatically decreased the drag production as well as increasing the downforce created by the rear wing.

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Table 8.21: Aerodynamic Advantage gained by BR-XV Engine Cover designs

The conclusion drawn from the analysis of each design iteration and design decisions that proved most effective in this report will be used in the continued development of the engine cover design for the use in the aerodynamic package of BR-16.
8.6.3.1.2 BR-16 Current Aerodynamic Package Analysis

To understand where the current aerodynamic package for BR-16 requires improvement to reach optimal performance, and how the existing design of the engine cover can be applied to BR-16 to aid this improvement, a full car simulation of the car was analysed. Figure 8.6.28 and Figure 8.6.29 shows the full car simulation.

Figure 8.6.28: Velocity Streamline Plot of BR-16 Engine bay (side view)

Figure 8.6.29: Velocity Streamline Plot of BR-16 Engine bay (top view)
Analysis of the simulation identified similar characteristics as experienced in BR-XV, highlighting areas of induced drag behind the driver’s helmet, resultant turbulent flow upstream of the rear wing and separation of the flow in the engine bay. However, with the design of the rear wing for BR-16, the distance between the rear wing and the driver’s helmet is greatly reduced with the leading edge of the rear wing positioned very close to the rear roll hoop. The rear wing is also situated in a higher location that the design of BR-XV, resulting in the turbulent flow downstream of the driver’s helmet passing under the bottom surface of the main plane and so majorly reducing the rear wing’s performance. These design changes must be considered in the development of the engine cover for BR-16 so that not only is drag reduced in the engine bay region but performance of the rear wing is optimised.

Initial design sketches were produced of the outer engine cover shape within the current CAD model of BR-16, incorporating the final design iteration developed for BR-XV and ensuring compliance with all vehicle components with the engine bay. From the compliance sketches initial CAD models of the engine cover were produced. With the position relation between the rear wing and the rear roll hoop it was found that a full height engine cover would be too large and would not package within the available area and so the maximum height achievable was considered a compromise. Two variations of the direction fin design feature were modelled to determine how the available area could be used most effectively. Simulation results of the two design iterations are shown below.
Figure 8.6.30: BR-16 Engine Cover Design 1 - Thin Directional Fin

Figure 8.6.31: Velocity Streamline Plot of BR-16 Engine Cover Design 1
Analysis of the simulations showed that the first iteration using a thinner direction fin design resulted in highly accelerated flow over the profile of the engine cover and to the rear of the rear axle. This accelerated flow dramatically increased the drag at the rear of the car, rendering the design a failure. This analysis is backed up by the drag and downforce data displayed in Table 8.22. It can be noted that both the drag and downforce were increased, however the lift-drag ratio was still decreased to that of the car without the engine cover.

Figure 8.6.32: BR-16 Engine Cover Design 2 - Thicker Directional Fin

Figure 8.6.33: Velocity Streamline Plot of BR-16 Engine Cover Design 2
The second design iteration with the wider direction fin design proved more successful with greater positive effects on the flow passing past the driver’s helmet. Figure 8.6.33 shows that drag was dramatically decreased in the critical area, simultaneous to successful filtering of the fluid upstream of the rear wing. Although a reduction in downforce produced by the rear wing is witnessed, the overall lift-to-drag ratio of the car is improved. Table 8.22 confirms these findings highlighting an increase in the lift-drag ratio over the car without the engine cover. This design was taken forward and over a number of iterations optimised.

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Table 8.22: Aerodynamic Advantage gained by BR-16 Engine Cover designs

The final design iteration, achieving the optimal overall performance of decrease drag and allowing maximum rear wing performance is shown in Figure 8.6.34, Figure 8.6.36 and Figure 8.6.35. The design was developed further to allow for cooling and fluid flow enclosed within the inner surface area of the engine cover. The developments included internal turning vanes and external ventilation. The final simulation results are shown in Figure 8.6.37 and Figure 8.6.38, and force data shown against the original car simulation in Table 8.23.
Figure 8.6.34: BR-16 Final Engine Cover Design

Figure 8.6.35: BR-16 Engine Cover Directional Fin Detail

Figure 8.6.36: BR-16 Engine Cover Ventilation Detail
### Table 8.23: Aerodynamic Advantage gained by BR-16 Final Engine Cover Design

<table>
<thead>
<tr>
<th></th>
<th>BR-16 Final Engine Cover Design</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Control</td>
<td>Final Design</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Simulated Results</td>
<td>Improvement</td>
<td></td>
</tr>
<tr>
<td>Drag (N)</td>
<td>335.62</td>
<td>150.05</td>
<td>185.57</td>
</tr>
<tr>
<td>Downforce (N)</td>
<td>896</td>
<td>490.2</td>
<td>-405.8</td>
</tr>
<tr>
<td>Lift-to-drag ratio, r</td>
<td>2.67</td>
<td>3.28</td>
<td>0.61</td>
</tr>
</tbody>
</table>

*Figure 8.6.37: Velocity Streamline Plot of BR-16 Final Engine Cover Design (side view)*

*Figure 8.6.38: Velocity Streamline Plot of BR-16 Final Engine Cover Design (top view)*
The extent of the improvements to the overall performance of the vehicle can be seen by the increase in lift-drag ratio. This result is considered a significant gain and the conclusion is drawn that this significance is justifiable for the cost and time investment into physical testing of the designs.

Post conversations with Brunel Racing team members in other vehicle departments; the engine cover design was split into multiple sections that would allow quick access to vehicle components enclosed by the engine cover. The sections were design in such a way that the higher priority components that required more frequent access were the most accessible. It was determined that the side sections of the engine cover would need to be removed most often and so were designed for quick detachment. However, at this stage in the design of BR-16, a number of powertrain components are still in development; including intake and exhaust systems and so prior to adjusting the final engine cover design for manufacture, the design must be adapted for these systems upon their completion.

Further stages in the development of the engine cover, including testing for interior to exterior heat transfer and physical testing is discussed in 11.9.

8.6.4 Future Design Methods

The continued expansion and performance improvement of Brunel Racing can be directly correlated to an investment in greater design capabilities and innovative approaches to the design of the team’s cars. As such there is a requirement for reduced design periods for certain vehicle departments to allow for time investment in the development of innovative design concepts that will ultimately give Brunel Racing the edge of its competition. To reduce these design periods, new approaches to the design process and the utilisation of
new design tools must be undertaken. A number of tools that have been developed for future use are discussed below.

8.6.4.1 Fluent Adjoint Solver

An automatic design iteration tool, Fluent Adjoint Solver can be used to automatically derive the optimal solution of aerodynamic components for desired CFD characteristics. The tool uses iterations of fluid dynamic simulation to assess where component design can be altered to improve the design’s aerodynamic profile. Per iteration the tool adapts the original input mesh with the specified parameter limits until the optimal geometry is obtained. This tool will prove particularly useful in the design of wing element profiles, their orientation and relation to one another. The process is currently very long winded with design iterations based on theory then assessed individually. However with the utilisation of Fluent Adjoint Solver, this process can be dramatically improved.

Figure 8.6.39 and Figure 8.6.40 demonstrates the utilisation of the tool on the current front wing profiles of BR-16. The process was run over a 5 iteration loop and the successful end results can be seen by the reduction of 24.3% overall drag and improvement of 23.13% overall lift, as shown in Table 8.24.
## Front Wing Profile Optimisation

<table>
<thead>
<tr>
<th></th>
<th>Before Optimisation</th>
<th>After Optimisation</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Main Element Drag (N)</strong></td>
<td>8.301</td>
<td>4.852</td>
</tr>
<tr>
<td><strong>Second Element Drag (N)</strong></td>
<td>13.942</td>
<td>18.808</td>
</tr>
<tr>
<td><strong>Total Drag (N)</strong></td>
<td>22.243</td>
<td>23.659</td>
</tr>
<tr>
<td><strong>Main Element Downforce (N)</strong></td>
<td>388.34</td>
<td>507.19</td>
</tr>
<tr>
<td><strong>Second Element Downforce (N)</strong></td>
<td>26.77</td>
<td>32.81</td>
</tr>
<tr>
<td><strong>Total Downforce (N)</strong></td>
<td>415.11</td>
<td>540</td>
</tr>
</tbody>
</table>

**Figure 8.6.40: Front Wing Profile after Fluent Adjoint Solver Optimisation**

**Table 8.24: Front Wing Profile Optimisation Aerodynamic Forces**

Fluent Adjoint Solver has also proved useful in the final design iterations of the full aerodynamics package, with the process run in 3D and all aerodynamic components selected for optimal performance of lift-drag ratio. Figure 8.6.41 and Figure 8.6.42 shows the design modification suggested by the tool after 2 iterations, highlighting the weakness within the design once the individually designed components have been integrated to form the full package.
To monitor the optimisation of the model and the scale factor that the optimisation toll was running at, the vehicle wheels were also selected for optimisation. As theory states, the wheels are one of the highest inducers of drag in an open wheel racing car, and although not modifiable, give a good control for design improvement.
8.6.4.2 Ansys Scheme Files

An additional tool not currently incorporated into the design process of components developed by Brunel Racing, Scheme Files are a useful method of reducing time invested in the simulation stage of design. Allowing for external setup and control of processes including FEA and CFD run in Ansys Workbench, Scheme Files remove the laborious and time expensive simulation setup procedure currently in use. The tool can also be used to setup multiple simulations simultaneously, a highly desirable trait in the design of components where a large number of design concepts require simulation. An example Scheme File is shown in Figure 8.6.43. This file is designed for the setup of a CFD simulation for a wing profile with the activation of Fluent Adjoint Solver to calculate the optimal design iteration. In time a database of Scheme files for all simulation processes required to design a Formula Student car will enable Brunel Racing to invest time in the development of innovative designs, further improving the overall performance of the car over its competition.

Figure 8.6.43: Fluent Scheme File for multiple optimisation iterations using Fluent Adjoint Solver
8.7 Vehicle Simulation (CS)

CarMaker relies on the user defining the geometry and inertia values of the vehicle in order to run a simulation. Below are the tables of parameters which have been used to define the vehicle. These were based on the example model file “FS_Racecar_4.0” which CarMaker supplied the team with. It is representative of a common Formula Student car from the 2011 season. Two models were developed, one for BR-XV and one for BR-16.

8.7.1 Vehicle Parameterisation – BR-XV

8.7.1.1 Vehicle Body

This is defined as all the sprung masses of the vehicle minus the engine and any external loads (including the load that will be induced by the addition of a driver) [60]. The body was defined as rigid, based on the assumption that the chassis has minimum flexure. The geometry and body inertias were defined for BR-XV as shown in Table 9.7.1 below:

<table>
<thead>
<tr>
<th></th>
<th>x (m)</th>
<th>y (m)</th>
<th>z (m)</th>
<th>Mass (kg)</th>
<th>$I_{xx}$ (kgm$^2$)</th>
<th>$I_{yy}$ (kgm$^2$)</th>
<th>$I_{zz}$ (kgm$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle body</td>
<td>1.2584</td>
<td>0</td>
<td>0.3024</td>
<td>210</td>
<td>11.98</td>
<td>53.17</td>
<td>51.12</td>
</tr>
</tbody>
</table>

Calculated overall vehicle mass (kg) 244.22

*Table 8.25 - Parameters of Vehicle body for BR-XV*

The $x$, $y$ and $z$ values in Table 9.7.1 are coordinates of the centre of gravity of the vehicle according to the axis system mentioned in 2.10.1. Here it should be noted that the values $I_{xx}$, $I_{yy}$ and $I_{zz}$ are the inertia values which the vehicle model FS_Racecar_4.0 uses. Hand calculations of an entire vehicle’s inertia would be seemingly impossible to determine accurately and experimental values of whole vehicle inertia require the use of an inertia
bed. Due to time and budget constraints it has not been possible to obtain these values for either BR-XV or BR-16.

IPG CarMaker uses both differential and algebraic equations to calculate the motion of the vehicle as a whole. Figure 9.7.1 below shows the forces on the vehicle which IPG CarMaker takes into account.

![Diagram of forces on a vehicle](image)

*Figure 8.7.1 - Forces used to calculate vehicle motion in CarMaker*

Subsequently the inertia values used for BR-XV and BR-16, which are for a car that weighs 332.9 kg overall, are predicted to slow the vehicle models down.
### 8.7.1.2 Bodies

Here the wheel carriers (including all the unsprung masses of the vehicle which remain stationary such as the wheel carrier itself and the wishbone mounts) and wheels (including all rotating masses of the wheel such as the wheel itself and the brake disc) are defined [60].

Their positions, masses and inertias are defined as shown in Table 9.7.2 below.

<table>
<thead>
<tr>
<th></th>
<th>( x ) (m)</th>
<th>( y ) (m)</th>
<th>( z ) (m)</th>
<th>Mass (kg)</th>
<th>( I_{xx} ) (kgm(^2))</th>
<th>( I_{yy} ) (kgm(^2))</th>
<th>( I_{zz} ) (kgm(^2))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel carrier FL</td>
<td>2.08</td>
<td>0.6</td>
<td>0.254</td>
<td>1.67</td>
<td>0.14</td>
<td>0.14</td>
<td>0.1</td>
</tr>
<tr>
<td>Wheel carrier FR</td>
<td>2.08</td>
<td>-0.6</td>
<td>0.254</td>
<td>1.67</td>
<td>0.14</td>
<td>0.14</td>
<td>0.1</td>
</tr>
<tr>
<td>Wheel carrier RL</td>
<td>0.5</td>
<td>0.585</td>
<td>0.254</td>
<td>1.44</td>
<td>0.03</td>
<td>0.02</td>
<td>0.03</td>
</tr>
<tr>
<td>Wheel carrier RR</td>
<td>0.5</td>
<td>-0.585</td>
<td>0.254</td>
<td>1.44</td>
<td>0.03</td>
<td>0.02</td>
<td>0.03</td>
</tr>
<tr>
<td>Wheel FL</td>
<td>2.08</td>
<td>0.6</td>
<td>0.254</td>
<td>7</td>
<td>0.2</td>
<td>0.32</td>
<td>0.2</td>
</tr>
<tr>
<td>Wheel FR</td>
<td>2.08</td>
<td>-0.6</td>
<td>0.254</td>
<td>7</td>
<td>0.2</td>
<td>0.32</td>
<td>0.2</td>
</tr>
<tr>
<td>Wheel RL</td>
<td>0.5</td>
<td>0.585</td>
<td>0.254</td>
<td>7</td>
<td>0.22</td>
<td>0.32</td>
<td>0.22</td>
</tr>
<tr>
<td>Wheel RR</td>
<td>0.5</td>
<td>-0.585</td>
<td>0.254</td>
<td>7</td>
<td>0.22</td>
<td>0.32</td>
<td>0.22</td>
</tr>
</tbody>
</table>

*Table 8.26 – Parameters of Bodies for BR-XV*

As with whole body inertia, the values for \( I_{xx} \), \( I_{yy} \) and \( I_{zz} \) were taken to be the set value of the FS_RaceCar_4.0 model. The masses of the wheel carriers were predicted to be the same as FS_RaceCar_4.0 for both BR-XV and BR-16, although their positions differ slightly to those of the original Formula Student vehicle model. This will result in a slight conflict of inertia values, although the difference will not be major.
The wheels for BR-XV weigh 1.71kg less than the original vehicle model, so these inertias are likely to be greater than the real wheel inertias and will contribute to the vehicle speed around track, as with body inertias mentioned in section 8.7.1.1.

### 8.7.1.3 Suspension

Here all suspension elements and geometries are defined. CarMaker has the following sections which must be defined in order for it to correctly calculate the vehicle kinematics:

- Spring
- Secondary spring
- Damper
- Buffer (Bump stop)
- Stabilizer (ARB)
- Kinematics geometry
- Compliance
- Wheel bearing friction
- External forces

The front and rear springs for BR-XV were set as characteristic values, with the front spring having a stiffness of 30625 N/m and the rear spring with a stiffness of 87500 N/m. Both springs were taken to have a free length of 0.2m. BR-XV did not contain a secondary spring system so this was set as unspecified. Dampers were defined using a look up table for the front and rear when under compression and rebound as shown in Tables 9.7.3 below. This data was retrieved from the Ohlins website for the TTX25 dampers bought a few years ago [109]. It should be noted however that the valves in these dampers will be worn after the years of use and the values below may not be the same for the dampers in their current
state. There is currently a plan underway to get the dampers refurbished so that they are reaching the values below for competition, but due to time and testing constraints this has not yet been possible. Both the front and rear dampers were defined using the same push and pull damping values.

<table>
<thead>
<tr>
<th>Damping (Ns/m)</th>
<th>Velocity (m/s)</th>
<th>Force (N)</th>
<th>Damping (Ns/m)</th>
<th>Velocity (m/s)</th>
<th>Force (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Push</td>
<td>0</td>
<td>0</td>
<td>Pull</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>0.025</td>
<td>83.35653</td>
<td></td>
<td>0.025</td>
<td>137.2931</td>
<td></td>
</tr>
<tr>
<td>0.05</td>
<td>153.9644</td>
<td></td>
<td>0.05</td>
<td>166.7131</td>
<td></td>
</tr>
<tr>
<td>0.075</td>
<td>183.3844</td>
<td></td>
<td>0.075</td>
<td>196.133</td>
<td></td>
</tr>
<tr>
<td>0.1</td>
<td>196.133</td>
<td></td>
<td>0.1</td>
<td>211.8236</td>
<td></td>
</tr>
<tr>
<td>0.125</td>
<td>212.8043</td>
<td></td>
<td>0.125</td>
<td>225.553</td>
<td></td>
</tr>
<tr>
<td>0.15</td>
<td>221.6303</td>
<td></td>
<td>0.15</td>
<td>245.1663</td>
<td></td>
</tr>
<tr>
<td>0.175</td>
<td>234.3789</td>
<td></td>
<td>0.175</td>
<td>259.8762</td>
<td></td>
</tr>
<tr>
<td>0.2</td>
<td>243.2049</td>
<td></td>
<td>0.2</td>
<td>270.6635</td>
<td></td>
</tr>
<tr>
<td>0.225</td>
<td>249.0889</td>
<td></td>
<td>0.225</td>
<td>284.3929</td>
<td></td>
</tr>
<tr>
<td>0.25</td>
<td>254.9729</td>
<td></td>
<td>0.25</td>
<td>304.0062</td>
<td></td>
</tr>
</tbody>
</table>

*Table 8.27 - Damping for BR-XV front and rear dampers*

Buffers were also defined using look up tables. The buffers for the front and rear dampers had the same push/pull characteristics, shown in Table 9.7.4 below:

<table>
<thead>
<tr>
<th>Compress (m)</th>
<th>Force (N)</th>
<th>Compress (m)</th>
<th>Force (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Push</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>0.01</td>
<td>0</td>
<td>0.005</td>
<td>0</td>
</tr>
<tr>
<td>0.02</td>
<td>0</td>
<td>0.01</td>
<td>0</td>
</tr>
<tr>
<td>0.03</td>
<td>0</td>
<td>0.04</td>
<td>0</td>
</tr>
<tr>
<td>0.04</td>
<td>0</td>
<td>0.05</td>
<td>81400</td>
</tr>
</tbody>
</table>
The length of vertical wheel travel from which the buffer is activated was taken to be 0.04m in the compression direction and -0.04m in the rebound direction.

The ARB for the front was set to a stiffness of 10713.51 N/m and the rear was set to 0 N/m as there was no rear ARB used in BR-XV.

Kinematics were defined using an SKC file. This is an output file from IPG Kinematics which takes into account all of the vehicle kinematics when static and under compression, such as camber, caster and springs and bushings. For the kinematics data of BR-XV it should be noted that the example files from CarMaker, FS_RaceCar_4.0_front.skc and FS_RaceCar_4.0_rear.skc were used. This means that the kinematics data will be slightly different in the vehicle simulation. Kinematics and their simulation for the car are currently under investigation and have been mentioned in the Initial Designs section of the Vehicle Simulation Portfolio.

The Compliance, wheel bearing and external forces were not specified for the simulation of BR-XV.

8.7.1.4 Steering
Steering was defined by the use of steering angle with no power steering for the BR-XV model. The rack travel to steering pinion angle was defined as 104 rad/m.
8.7.1.5 Tyres

For the BR-XV model the example tyre file, FS_195_50R13, was used. This is for a 13 inch wheel and is the closest example to the tyres used currently on the vehicle. Work has been ongoing into using Pacejka tyre models, although it may be more of an option to look at in future years. For current validation purposes the vehicle still has 13 inch wheels as opposed to the 10 inch ones proposed for competition at FSUK and FSG.

8.7.1.6 Brakes

For this model an external file was used. This was the example file provided by IPG which includes ESP simulation and one master cylinder. Pressure distributions can also be parameterised which determines the ratio between the force at the brake pedal to the brake force at each wheel. The block diagram in Figure 9.7.2 below shows a generic set up for a braking system which uses ESP. This model was used as opposed to the pressure distribution model as this data is difficult to obtain from data provided by sensors currently on the vehicle.
8.7.1.7 Powertrain

CarMaker provides the user with an opportunity to define powertrain parameters including torque curves. For BR-XV the inertia of the engine was set as the example value of 0.01 kgm$^2$. As with whole vehicle inertia this is extremely difficult to calculate due to the constant movement of engine components and the number of components within the engine. The idle speed for the car is 1500 rpm and the starter motor torque is 15 Nm. The speed at which the starter disengages is 430 rpm and the torque at which the ignition cuts was left at the example setting of -80 Nm. The engine orientation was set to transverse. The torque curve for BR-XV is shown by Figure 9.7.3 below:
The powertrain model also has the option for fuel consumption data for the car. This was deactivated for this project as the fuel consumption of the car is not a top priority for racing purposes. Fuel tank data was parameterised, with volume set to 50L and initial filling percentage set at 80%. The fuel density was 0.75 kg/l [110] and the heat value of the fuel 11.3 kWh/kg [111] – the approximate values for 97 RON petrol. It should be noted that in reality the fuel tank is much smaller than these values, although CarMaker at present doesn’t have the capabilities for a fuel tank smaller than this. CarMaker was originally designed for passenger vehicles which have much larger fuel tanks and these values were the lowest settings for CarMaker to use. It will affect the performance of the vehicle model somewhat but this data is mainly used for fuel consumption calculations, which as mentioned have been disabled for these simulations.

The clutch was left at the example values provided by CarMaker for the FS_Racecar_4.0 model. This is mainly because CarMaker only has the capability to allow the user to define a clutch that would be used in a passenger car. IPG do offer a separate program called IPG MotorcycleMaker which has the ability to define a wet clutch as used in the Yamaha R6
engine used in Brunel Racing’s Formula Student vehicles, although the two programs are not interchangeable and such a generous sponsorship package is not available for MotorcycleMaker [112]. Investigation is ongoing into a Simulink model which can be used with CarMaker to improve the accuracy of certain parameters like the clutch, although inter-program compatibility has presented issues with this feature and has not yet been resolved.

Gear ratios were set to those shown in Table 9.7.5 and the synchronisation time set to 100ms.

<table>
<thead>
<tr>
<th>Gear</th>
<th>Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.5</td>
</tr>
<tr>
<td>2</td>
<td>2.133</td>
</tr>
<tr>
<td>3</td>
<td>1.889</td>
</tr>
<tr>
<td>4</td>
<td>1.737</td>
</tr>
</tbody>
</table>

*Table 8.29 – Gear ratios for BR-XV*

The overall drive ratio was set to 3.45. The differential was set to rear viscoelastic as this model has the capability to transfer torque when there is a speed difference between both wheels, so it is suitable for modelling an LSD as used in BR-XV [60].

8.7.1.8 Aerodynamics

For these simulations the aerodynamics function was turned off.

8.7.2 Vehicle Parameterisation – BR-16

8.7.2.1 Vehicle Body

<table>
<thead>
<tr>
<th></th>
<th>x (m)</th>
<th>y (m)</th>
<th>z (m)</th>
<th>Mass (kg)</th>
<th>I_{xx} (kgm^2)</th>
<th>I_{yy} (kgm^2)</th>
<th>I_{zz} (kgm^2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle</td>
<td>1.265</td>
<td>0</td>
<td>0.3</td>
<td>190</td>
<td>11.98</td>
<td>53.17</td>
<td>51.12</td>
</tr>
<tr>
<td>body</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Calculated overall vehicle mass (kg) | 222.22

Table 8.30 – Parameters of Vehicle body for BR-16

As mentioned in section 8.7.1.1, inertia values for BR-16 are at the values set in the FS_RaceCar_4.0 model. The effect on this vehicle model of the different inertia values will be more drastic as the car is lighter than BR-XV, so it is expected to be a little slower than the BR-XV vehicle model.

8.7.2.2 Bodies

<table>
<thead>
<tr>
<th></th>
<th>x (m)</th>
<th>y (m)</th>
<th>z(m)</th>
<th>Mass (kg)</th>
<th>$I_{xx}$ (kgm²)</th>
<th>$I_{yy}$ (kgm²)</th>
<th>$I_{zz}$ (kgm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel carrier FL</td>
<td>2.03</td>
<td>0.6</td>
<td>0.229</td>
<td>1.67</td>
<td>0.14</td>
<td>0.14</td>
<td>0.1</td>
</tr>
<tr>
<td>Wheel carrier FR</td>
<td>2.03</td>
<td>-0.6</td>
<td>0.229</td>
<td>1.67</td>
<td>0.14</td>
<td>0.14</td>
<td>0.1</td>
</tr>
<tr>
<td>Wheel carrier RL</td>
<td>0.5</td>
<td>0.6</td>
<td>0.229</td>
<td>1.44</td>
<td>0.03</td>
<td>0.02</td>
<td>0.03</td>
</tr>
<tr>
<td>Wheel carrier RR</td>
<td>0.5</td>
<td>-0.6</td>
<td>0.229</td>
<td>1.44</td>
<td>0.03</td>
<td>0.02</td>
<td>0.03</td>
</tr>
<tr>
<td>Wheel FL</td>
<td>2.03</td>
<td>0.6</td>
<td>0.229</td>
<td>6.5</td>
<td>0.2</td>
<td>0.32</td>
<td>0.2</td>
</tr>
<tr>
<td>Wheel FR</td>
<td>2.03</td>
<td>-0.6</td>
<td>0.229</td>
<td>6.5</td>
<td>0.2</td>
<td>0.32</td>
<td>0.2</td>
</tr>
<tr>
<td>Wheel RL</td>
<td>0.5</td>
<td>0.6</td>
<td>0.229</td>
<td>6.5</td>
<td>0.22</td>
<td>0.32</td>
<td>0.22</td>
</tr>
<tr>
<td>Wheel RR</td>
<td>0.5</td>
<td>-0.6</td>
<td>0.229</td>
<td>6.5</td>
<td>0.22</td>
<td>0.32</td>
<td>0.22</td>
</tr>
</tbody>
</table>

Table 8.31 – Parameters of bodies for BR-16

As mentioned in section 8.7.1.2, inertia values for the wheels and their carriers have been left at the values of those in the FS_RaceCar_4.0 model. The wheel inertia values will have a slightly larger effect on this model because here the wheels are 2.21kg less than the FS_RaceCar_4.0 wheels.
8.7.2.3 Suspension

The front and rear springs for BR-16 were set as characteristic values, with the front spring having a stiffness of 35000 N/m and the rear spring with a stiffness of 40000 N/m. The front spring was taken to have a free length of 0.218m and the rear 0.208m. BR-16 does not yet contain a secondary spring system so this was set as unspecified, although for aerodynamics purposes it is likely there will be a secondary spring for competition. Dampers were defined using a look up table for the front and rear when under compression and rebound as shown in Table 9.7.8 below. Both the front and rear dampers were defined using the same push and pull damping values.

<table>
<thead>
<tr>
<th>Damping (Ns/m)</th>
<th>Velocity (m/s)</th>
<th>Force (N)</th>
<th>Damping (Ns/m)</th>
<th>Velocity (m/s)</th>
<th>Force (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Push</td>
<td>0</td>
<td>0</td>
<td>Pull</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>0.025</td>
<td>83.35653</td>
<td></td>
<td>0.025</td>
<td>137.2931</td>
</tr>
<tr>
<td></td>
<td>0.05</td>
<td>153.9644</td>
<td></td>
<td>0.05</td>
<td>166.7131</td>
</tr>
<tr>
<td></td>
<td>0.075</td>
<td>183.3844</td>
<td></td>
<td>0.075</td>
<td>196.133</td>
</tr>
<tr>
<td></td>
<td>0.1</td>
<td>196.133</td>
<td></td>
<td>0.1</td>
<td>211.8236</td>
</tr>
<tr>
<td></td>
<td>0.125</td>
<td>212.8043</td>
<td></td>
<td>0.125</td>
<td>225.553</td>
</tr>
<tr>
<td></td>
<td>0.15</td>
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<td></td>
<td>0.15</td>
<td>245.1663</td>
</tr>
<tr>
<td></td>
<td>0.175</td>
<td>234.3789</td>
<td></td>
<td>0.175</td>
<td>259.8762</td>
</tr>
<tr>
<td></td>
<td>0.2</td>
<td>243.2049</td>
<td></td>
<td>0.2</td>
<td>270.6635</td>
</tr>
<tr>
<td></td>
<td>0.225</td>
<td>249.0889</td>
<td></td>
<td>0.225</td>
<td>284.3929</td>
</tr>
<tr>
<td></td>
<td>0.25</td>
<td>254.9729</td>
<td></td>
<td>0.25</td>
<td>304.0062</td>
</tr>
</tbody>
</table>

Table 8.32 – Damping for BR=16 front and rear dampers

Buffers were also defined using look up tables. The buffers for the front and rear dampers had the same push/pull characteristics, shown in Table 9.7.9 below:
### Table 8.33 – Buffer parameters for BR-16

<table>
<thead>
<tr>
<th>Compress (m)</th>
<th>Force (N)</th>
<th>Compress (m)</th>
<th>Force (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Push</td>
<td></td>
<td>Pull</td>
<td></td>
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<td>0</td>
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</tr>
<tr>
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</tr>
<tr>
<td>0.02</td>
<td>0</td>
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<td>0</td>
</tr>
<tr>
<td>0.03</td>
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<td>0.04</td>
<td>0</td>
</tr>
<tr>
<td>0.04</td>
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<td>0.05</td>
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</tr>
<tr>
<td>0.05</td>
<td>81400</td>
<td>0.07</td>
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</tr>
<tr>
<td>0.06</td>
<td>81400</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.07</td>
<td>81400</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The length of vertical wheel travel from which the buffer is activated was taken to be 0.04m in the compression direction and -0.04m in the rebound direction.

The ARB for the front was set to a stiffness of 107130.51 N/m and the rear was set to 21767.997 N/m.

For Kinematics data of BR-16 FS_RaceCar_4.0_front.skc and FS_RaceCar_4.0_rear.skc were used. This means that the kinematics data will be slightly different in the vehicle simulation. Kinematics and simulation for the car are currently under investigation (see section 8.7.1.3).

The Compliance, wheel bearing and external forces were not specified for the simulation of BR-16.

#### 8.7.2.4 Steering

Steering was defined by the use of steering angle with no power steering for the BR-16 model. The rack travel to steering pinion angle was defined as 73.23 rad/m.
8.7.2.5 Tyres

For the BR-16 model the example tyre file, FS_195_50R13, was used (see section 8.7.1.5).

8.7.2.6 Brakes

For this model an external file was used. This was the example file provided by IPG which includes ESP simulation and one master cylinder (see section 8.7.1.6).

8.7.2.7 Powertrain

![Torque curve for BR-16](image)

*Figure 8.7.4 – Torque curve for BR-16*

Figure 9.7.4 shows the torque curve from engine dynamometer results of BR-16, which was programmed into IPG CarMaker.

The fuel consumption data for the car was deactivated for this project as the fuel consumption of the car is not a top priority for racing purposes. Fuel tank data was parameterised, with volume set to 50L and initial filling percentage set at 80%. The fuel density was 0.75 kg/l [110] and the heat value of the fuel 11.3 kWh/kg [111] – the approximate values for 97 RON petrol. See section 8.7.1.7 for reasons why the fuel tank was set to such a large volume.
The clutch was left at the example values provided by CarMaker for the FS_Racecar_4.0 model.

Gear ratios were set to those shown in Table 9.7.10 and the synchronisation time set to 100ms.

<table>
<thead>
<tr>
<th>Gear</th>
<th>Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.5</td>
</tr>
<tr>
<td>2</td>
<td>1.579</td>
</tr>
<tr>
<td>3</td>
<td>1.272</td>
</tr>
</tbody>
</table>

Table 8.34 – Gear ratios for BR-16

The overall drive ratio was set to 3.167. The differential was set to rear viscoelastic as this model has the capability to transfer torque when there is a speed difference between both wheels, so it is suitable for modelling an LSD as used in BR-16 [60].

8.7.2.8 Aerodynamics

For these simulations the aerodynamics function was turned off.

8.7.3 Results of Simulation

For each vehicle simulations were completed around Bruntingthorpe Kart track. This data were then compared against test data from the car. Currently there is only some testing data available for BR-XV, as BR-16 is still in the manufacturing process. Therefore any BR-16 simulations are expectant data only and have yet to be validated.
Figure 9.7.5 shows a map of Bruntingthorpe kart track with labelled corners to assist the readers of this report to understand the graphs. Figure 9.7.6 below shows an example speed graph. Assuming that the points of lowest velocities of data are the points at which the driver hits the apex of the corner, the general pattern below can be seen:
8.7.3.1 BR-XV
Simulation results for BR-XV were taken over 300 seconds around Bruntingthorpe, giving just over 9 lap’s worth of data for comparison against test data. Test data used for comparison was from 5\textsuperscript{th} March 2015 as previous test data did not include a variety of sensors.

8.7.3.1.1 Driver Optimisation of Lap times
This set of simulations was carried out by a Level 3 student, Alexander Boot, using the parameterised vehicle model as detailed in this report. In these simulations driver characteristics were edited in order to provide a more realistic driving simulation with results closer to those of the Brunel Racing drivers.

8.7.3.1.2 Race Driver – Accelerations, g-g diagram
The race driver function was selected for these simulations and edited to determine which driving characteristics provided lap times closest to those achieved during testing. The standard race driver feature provides a g-g diagram as shown in Figure 9.7.7 below:
For Alexander’s simulations, the optimum lap times achieved were using an acceleration exponent of 49 and a deceleration exponent of 49 instead of the standard setting of 1g for both acceleration and deceleration. This provided an average lap time across 19 laps of 30.492s when all other driver characteristics were set as standard.

8.7.3.1.3 Race Driver – Corner cutting coefficient

This is a setting of IPG Driver whereby 0 means that the driver remains in the centre of the track and 1 means that the driver uses all of the track width to determine the optimal racing line [61]. Alexander’s simulations investigated a range of corner cutting coefficients from 0.1 to 0.97. After 0.97 the driver of the BR-XV model proceeded to run off track on multiple occasions. The corner cutting coefficient found to provide the maximum improvement in lap time was 0.97 which provided an average lap time of 31.237s when all other driver characteristics were set as standard.
8.7.3.1.4 Speed vs. time graph

Looking at Figure 8.7.8 it is possible to determine whether the lap times match up for simulation and test data. Following the point where each vertical line is marked there are some distinct areas of the graphs. After the apex of turn 1 the vehicles increase their speed. The test data clearly reaches a higher peak speed of 86 km/h than the simulation data which reaches only 55 km/h. The speed then drops a little for the slight turn after turn 1 and then increases again before slowing down for turn 2. The physical test data shows that the driver brakes much harder coming up to turn 2 but as the simulation vehicle was travelling at a lower speed into that turn the driver did not need to brake much before entering the corner. The simulation data then increases smoothly in the run up to turn 3. This isn’t the same for the physical test data, as the driver clearly lifts off the throttle. This could be to facilitate a downshift prior to turn 3. Again, the test data shows that the real driver reaches a higher peak speed than the simulation driver between turn 2 and turn 3. In the run up to the hairpin the real test data shows that nearly the same peak speed is reached as was reached between turn 2 and turn 3. This is the same for the simulation data for the initial
lap shown although this begins to drop in the second lap. For both sets of data the hairpin is the point on the track where the vehicle speed is at its lowest. The simulation driver brakes earlier than the real driver for all turns up to and including this point, also slowing down to 30 km/h whereas the physical test data shows that BR-XV slows down to around 43 km/h. Subsequently, here is where the simulation begins to slow down each lap as the simulation car consistently loses time on the hairpin.

There is then another drop in speed as both vehicles enter turn 5, after which the simulation data increases constantly to its peak speed reached around track of 80 km/h. The physical test data shows a slightly higher peak speed is reached although fluctuations in the vehicle speed show that the driver lifts off the throttle for the slight curve in the track before returning back to turn 1.

8.7.3.1.5 G-G Diagram

![BR-XV g-g Plot](image)

**Figure 8.7.9 - BR-XV g-g diagram comparison**

Looking at Figure 9.7.9 above it is clear that both the simulation data and test data exhibit the same g-g pattern. It is noticeable that the test data has a broader range of both lateral
and longitudinal acceleration. The longitudinal acceleration is limited by the vehicle model and the lateral acceleration increases when travelling at a faster speed around a bend whilst maintaining grip, supporting the graphs of speed in the previous section in showing that the vehicle model is slower than the real car. IPG Driver is also highly consistent whereas a real driver (not at professional level as per Formula Student regulations) is not always consistent.

8.7.3.1.6 Longitudinal acceleration vs. time graph

![BR-XV Longitudinal Acceleration vs. time graph](image)

Figure 8.7.10 - BR-XV Longitudinal acceleration with time comparison

Longitudinal and lateral accelerations can be further analysed by breaking each acceleration down and plotting them with time. Figure 9.7.10 shows the longitudinal acceleration with time over a few laps. The physical test data results show higher peak longitudinal accelerations than the simulation data, supporting the speed vs. time graph shown earlier in the report which exhibits slower speeds reached by the vehicle model and a less steep gradient. The simulation data also suggests that longitudinal deceleration values are much greater than those from physical test data at certain points around the track, namely the deceleration into turn 1 and also into the hairpin. The vehicle model reaches similar peak speeds into turn 1 and slows down to a speed slower than that of the real world data,
supporting this theory. The simulation also slows down a great deal more than the real vehicle for the hairpin, also supporting this data.

8.7.3.1.7 Lateral acceleration vs. time graph

![BR-XV Lateral Acceleration vs. time graph](image)

*Figure 8.7.11 - BR-XV Lateral acceleration with time comparison*

Figure 9.7.11 shows the comparison of lateral acceleration over the same few laps. This suggests that the physical test data and simulation data results are very similar, reaching similar peaks for the corners and following the same pattern. Generally, the simulation data reaches lower peak lateral accelerations, with the exception of a few turns on the track although this is not consistent. For some turns this is most likely due to the fact that the simulation vehicle is reaching lower speeds in corners, so it is expected to have slightly lower lateral acceleration values than that of the physical test data.

8.7.3.2 BR-16

8.7.3.2.1 Driver Optimisation of Lap times

For BR-16 simulations, driver characteristics were changed slightly as they were too harsh to work with the BR-16 vehicle model.
8.7.3.2.2 Race Driver – Accelerations, g-g diagram

For the g-g diagram in Race Driver, the optimum lap times achieved were using an acceleration exponent of 5 and a deceleration exponent of 5. Any higher values resulted in the vehicle leaving the track after 3 laps.

8.7.3.2.3 Race Driver – Corner cutting coefficient

For BR-16 simulations the corner cutting coefficient was set to 0.9, for the reasons described above in section 8.7.3.2.2.

8.7.3.2.4 Speed vs. time graph

![BR-16 Speed vs. time graph](Figure8.7.12.png)

Figure 8.7.12 - BR-16 Speed with time comparison

When compared with the same data used for comparison against BR-XV, there are a few differences which can be noted (see Figure 9.7.12). In the portion of track before turn 1, the peak speed of the vehicle model is very close to the peak speed achieved during testing. The speed achieved by the vehicle model between turn 1 and 2 is also much lower than that achieved by the real vehicle. However, the acceleration of the vehicle model is close to if not
better than that of the physical test data as it reaches nearly the same peak speed between turns 2 and 3 and between turn 3 and the hairpin. As with BR-XV simulation data, the main time lost per lap is at the hairpin turn of Bruntingthorpe. The driver model used for the BR-16 simulation also provides data which shows a more similar pattern to that of the real driver of the car. This is particularly noticeable just before turn 1 where both the simulation data and the physical data show that the driver lifts off the throttle on the slight curve prior to the turn.

8.7.3.2.5  G-G Diagram

![BR-16 g-g Plot](image)

*Figure 8.7.13 - BR-16 g-g diagram comparison*

It can be observed from Figure 9.7.13 that the points from BR-16 simulation data exhibit a range closer to that of the test data, producing a wider pattern than the simulation results from BR-XV. The same shape of the diagram is still shown indicating that the vehicle characteristics are close to those of the real vehicle.
8.7.3.2.6  Longitudinal acceleration vs. time graph

Longitudinal acceleration values for BR-16 can be seen in Figure 9.7.14. These values are closer to those of the test data than for the BR-XV model, although in some instances the longitudinal acceleration capability of the vehicle model is greater than the real vehicle. This can be seen just before the car hits the hairpin, upon which it decelerates very rapidly. The simulation data also reaches higher peak acceleration just before turn 1.
8.7.3.2.7  Lateral acceleration vs. time graph

Figure 8.7.15 - BR-16 Lateral acceleration with time comparison

Lateral acceleration values for BR-16 are also extremely close to the test data values as shown in Figure 9.7.15. Peak lateral accelerations are all close with the exception of the maximum lateral acceleration reached just after turn 5. Here the simulation data shows a much lower lateral acceleration than the physical test data. This could be because the vehicle model exits turn 5 at a much slower speed than that of the real vehicle so it will not reach the same peak lateral accelerations in a corner. The physical test data also reaches a higher lateral acceleration in the run up to the hairpin than the simulation data, again indicative that the simulation vehicle is travelling round the track at a lower speed.
8.7.3.3 BR-XV Vehicle Model vs. BR-16 Vehicle Model

8.7.3.3.1 Speed vs. time graph

Comparing both vehicle models using Figure 8.7.16, there are a few noticeable characteristics. The primary feature is that the acceleration of the BR-16 model is better than that of the BR-XV model, with steeper line gradients when the vehicle increases speed. This also gives BR-16 a slightly higher peak speed of 84 km/h whereas the BR-XV model only reaches 80 km/h. Both models begin to suggest that the driver is accelerating and braking at the same times on track, until the hairpin is reached. At this point the BR-16 model does much worse than the BR-XV model, slowing down to 18 km/h whereas the BR-XV model only slows to 30 km/h.
8.7.3.3.2 G-G Diagram

![BR-XV vs BR-16 g-g plot](image)

**Figure 8.7.17 – g-g diagram comparison of two vehicle models**

Looking at the g-g diagrams for each vehicle model in Figure 9.7.17, both exhibit a similar pattern. Where BR-XV has a wider range of lateral acceleration, BR-16 has a wider range of longitudinal acceleration.
8.7.3.3 Longitudinal acceleration vs. time graph

Looking at the longitudinal acceleration with time graph for each simulation shown in Figure 9.7.18, this better illustrates the fact that the BR-16 model has higher peak longitudinal acceleration values than the BR-XV model. It can be seen on the hairpin that the driver in the BR-16 model appears to reach the turn, begins to lift off the brake and then has to put the brakes on harder again. This is the slight fluctuation seen in the third trough after each lap start line. This could be the reason why the BR-16 model is losing time on the hairpin, as the driver tries ease off the brakes too early and then has to put the brakes on harder to ensure the car makes it round the hairpin.
8.7.3.4 Lateral acceleration vs. time graph

Looking at the lateral acceleration graphs in Figure 9.7.19 it can be seen that similar peak lateral accelerations are reached by both vehicle models. The slight fluctuation in the third peak from the lap start line is indicative of the driver model in BR-16 not using enough braking force on the way out of the hairpin and then having to brake again, as mentioned in the previous section 8.7.3.3.3 of this report. The BR-XV model also reaches greater values of negative lateral acceleration than the BR-16 model.

8.7.4 BR-16 Predicted Sprint/Endurance event performance

As the BR-16 model had closer characteristics to those of the test data, it would be pertinent to simulate BR-16 around the FSUK endurance track to see how it would perform. Figure 9.7.20 below shows a map of the Endurance track from last year’s event, which was programmed into CarMaker in segments in order to simulate the BR-16 model competing in this event.
The simulation was set to run from the sprint start line as shown in Figure 9.7.20 above. The simulation ran for 300 seconds as with Bruntingthorpe simulations. Chicanes had to be programmed into the track using the “cone alley” feature of IPG CarMaker to ensure that the driver did not just take the racing line and run straight down the middle.
8.7.4.1 Driver Optimisation

For these simulations the driver settings were changed back to those used for BR-XV simulations (see section 8.7.3.1.1).

8.7.4.2 Speed vs. time graph

![BR-16 Predicted Endurance Speed vs. time graph](image)

Looking at the two laps shown in Figure 9.7.21 above, the driver in the simulation is again extremely consistent. He reaches a peak velocity of approximately 119 km/h before having to brake for the corner just after the marked endurance start line. There is a slight fluctuation in vehicle speed as the driver slows down for the Mercedes AMG HPP complex. Speed then increases a little as the car passes through the Mercedes AMG HPP complex and then decreases again in preparation for the chicanes on the SAE straight. The vehicle speed increases again before the driver has to brake ready for Maggott’s hairpin. The vehicle then accelerates through the Jaguar Chicane and Land Rover Esses, reaching a peak speed of approximately 132 km/h. The driver brakes hard ready for the second set of chicanes and accelerates again afterwards. The driver then applies the brakes a little just before entering Copse corner and accelerates a little more before braking for the National hairpin.
The speed graph shows that the initial lap is completed in 51.5 seconds and the second in 55.3 seconds. Looking at the fastest sprint time from FSUK 2014 of 47.258 seconds [114], this is a fairly reasonable time and would suggest that BR-16 will do well in the Sprint and Endurance events at FSUK. The slower lap time on the second lap could be where the driver brakes a bit more into the Carter’s, which is only noticeable in the second lap data. It is because of these two troughs prior to the first large peak on the graph, that lap time is lengthened for successive laps.

The driver model could be refined a little more in order to speed the vehicle up on the hairpin as although this isn’t progressively slowing the car down each lap, it is an area where improvements to lap time could be made.

8.7.4.3 G-G Diagram

![BR-16 Predicted Endurance g-g Plot](image)

Looking at Figure 9.7.22, the g-g diagram of BR-16 around the Endurance track shows that the lateral acceleration of the car has a broad range from around 1.2g to -1.2g. This is similar
to the g-g diagram for BR-16 around Bruntingthorpe although the chicanes and tighter corners present in the Endurance track have provided more data points towards the middle portion of the g-g diagram rather than close to the outside edge as with the Bruntingthorpe data.

The longitudinal acceleration data is not as broad as the values of lateral acceleration although they have a wider range than the data from BR-16 around Bruntingthorpe, ranging from around 0.6 to -1.3g.

8.7.4.4 Longitudinal Acceleration vs. time graph

Looking at the longitudinal acceleration with time graph shown in Figure 9.7.23 the two peak points of longitudinal acceleration correspond to the run up to Mercedes AMG HPP Complex and the second set of chicanes, confirming the speed vs. time graph. The lowest troughs seen each lap correspond to Maggott’s hairpin, the second set of chicanes and the national hairpin. This is not surprising as both hairpins are very tight corners and the greatest deceleration occurs just after peak speed is reached at the Land Rover Esses. The
fluctuations in longitudinal acceleration between 11 and 17 seconds in the first lap and at this stage in laps thereafter is indicative of the corner just after the endurance start line up to the Mercedes AMG HPP Complex where the driver is accelerating slightly but having to lift off the throttle to prevent running off track.

8.7.4.5 Lateral Acceleration vs. time graph

The graph for lateral acceleration with time (shown in Figure 9.7.24) has a greater number of peaks and troughs than any other lateral acceleration graph presented in this report due to the number of bends present in the Endurance track. This results in peak positive and negative lateral acceleration regularly being reached around the track especially in the run up to the hairpin where the majority of the chicanes occur.

There is one noticeable peak which is a little wider than the others, which indicates the vehicle passing through the Jaguar Chicane, where peak speed is reached. The wide trough after this peak is a result of the driver steering slightly in order to get through the Land Rover Esses. The peak after that point is representative of the entry to the second set of
chicanes. The driver enters the first chicane faster than successive chicanes which explains the initial larger peak and subsequent smaller peaks.
9 BR-16 Manufacture

9.1 Unsprung (EJ)

9.1.1 Hubs

The drawing for the front hubs was released on February 26th 2015 and the hubs went into manufacture in-house in the next couple of days. Figure 9.1.1 shows the first hub in the finishing stages of the milling operations.

![](image)

*Figure 9.1.1 First Hub in Manufacture after all Mill Profiling operations*

The hub in Figure 9.1.1 became the test hub for all machining operations after one of the turning operations was miss-programmed and cut a significant portion of the hub which was required. This meant we needed to machine another hub for the vehicle, but did mean we could run all the programmes on one hub to make sure they all worked before running them on the other hubs.
9.1.2 Steel Wishbones

Wishbones are critical parts that need to be manufactured very close to the suspension geometry that was designed to ensure the vehicle handles as expected. It is also important that the manufacturing process is repeatable so that the left and right side of the car are symmetrical.

The jig plate for this year was made from a thick steel plate and the holes were centre-drilled on a 2.5 axis Mill with 4 decimal place accuracy. The holes were then drilled using a pillar drill with two people holding the plate allowing the plate and drill bit to meet in the centre of the desired hole. The pegs to hold the bearings and inserts in for welding were then designed such that the centre of the bearings were all 25.4mm (1”) away from the surface of the jig plate, ensuring that the welding of the wishbones was all done in plane. The housings for the staked bearings (Figure 8.2.19) were sent off to a water-jet cutting company and the rod-end tube inserts were made in-house. Figure 9.1.2 shows the CAD assembly of the Wishbone Jig Plate complete with all the bearing pegs in their correct place.
9.2 Chassis (GG)

9.2.1 Rear Frame

Manufacturing of the first rear frame (Figure 9.2.1) was conducted in the 2nd week of December 2014, this frame was designed for the “lowered engine project”, sadly this project has failed to deliver satisfactorily so the first rear frame is likely to now not be used. The second frame is due to be constructed during week 30.

Manufacture of the frame has been massively simplified this year through the use of on-site welding, well designed jigs, and the use of machine profiled tubes (Figure 9.2.2); further
details of which can be found in the portfolio of evidence. All of the above have also led to a much tighter control on geometric tolerances.

Figure 9.2.2. Fit-up of machine profiled tubes

9.2.2 Monocoque

It was originally intended for the monocoque to be folded around the time of the Christmas holidays, the inner skins were water-jet cut ahead of this time in preparation for bonding of the panels, however ultimately the monocoque was placed on hold 2 weeks prior to this date when changes in the rules were discovered rendering the design illegal. Following a protracted physical testing program and several rules clarifications it is believed that the chassis is now legal and a revised SES is being submitted. It is intended to complete chassis manufacture prior to the exam season.
9.3 Aerodynamics (GM)

The final goal of this project is to manufacture the designed components with adequate strength and good surface finish. There are several ways of manufacturing a carbon airfoil however the tools at Brunel Racing’s disposal limit the choice between available techniques. Not all the components are going to be manufactured in same way: depending on the force produced by the component and its dimension the more appropriate manufacturing technique will be utilized.

9.3.1 Front Wing Mainplane

The front wing mainplane is the largest component to be manufactured. The first consideration is that the mainplane is the first component at the front of the car that could be in contact with objects (such as cones) during the dynamic events at competition. The strength of the component has hence to be high in order to prevent permanent damage: the adoption of a RohaCell core is the best solution to the problem. The downside of using a RohaCell core is that the foam has to be outsourced for machining. In fact due to the high toxicity of the material the machining operation has to be carried out in an environment with professional air extraction systems. This machining operation is very costly.
Due to the great span of the front wing main plane two issues are arising: the limited capabilities of the machinery available at Brunel to machine the tooling block to create molds and the size of the oven for carbon curing. In fact the machines at Brunel can only work with components of a maximum size of 1000*500mm. The front wing main plane has a span of 1360mm and a chord length of 525mm.

The first issue is solved by machining the mold in two parts of 680x525mm and subsequently gluing the parts together to create a single wing mold. The restriction imposed by the size of the oven can be overcome by wet laying carbon instead of using Pre-Preg fibre: with the wet layup technique the material is left to cure at atmospheric conditions instead of at a controlled pressure and temperature (the over is not required).

A picture of the machined TRELLEBORG tooling block mold can be visualized below:
9.3.2 Rear wing mainplane

The rear wing mainplane is the second largest aerofoil to be manufactured. The strength of the component has to be adequate to transfer the load to the wing mounts without failing. The best solution is to save weight and provide secure attaching points to the mounts is to manufacture a hollow wing featuring a structured made of ribs.

Ribs are the structural crosspieces that make up the framework of the wing. They usually extend from the wing leading edge to the trailing edge of the wing. The ribs give the wing its cambered shape and transmit the load from the skin. [115]

A typical wing structure made with ribs can be seen below:

![Figure 9.3.3 Wing Ribs](image)

The ribs will be positioned strategically: the wing mounts struts will be attached to the wing mainplane rather than the endplates. By positioning larger and stronger ribs below each clevis on the mainplane it is possible to securely attach the wing mounting structure without the risk of creating high localized stresses on the weaker carbon skin.
Again due to the large dimension of the mainplane, the mold will be made in 2 pieces and the wet-layup technique will be utilized to overcome the limit imposed by the size of the oven.

**9.3.3 Wing Flaps**

All of the wing flaps are of moderate dimensions and are not subjected to high forces. For this reasons the tooling block mold can be machined in one single piece. This also means that the oven available in the workshop can be utilized for curing the Pre-Preg Carbon fibre. This saves substantial time since the lay-up technique of Pre-Preg fiber is considerably faster than wet-laying.

The flaps will not contain a RohaCell core but instead structural ribs will be introduced after the Pre-Preg curing process in order to maintain the weight of the component relatively low.

The layup consists in two ply of TenCate E722 205gsm HS Carbon Pre-preg at 0° and 90° in order to create a quasi-isotropic" laminate which effectively creates uniform in plane properties.
9.4  Driver Controls & Electronics (TM)

9.4.1  Brake System

Solid discs will be made along with a drilled-hole design. Ground 4130 steel will be bought and then sent out to be water-jet cut. This has two advantages; cost and quality control. Since the team has access to water-jet cutting from a sponsor it is cheaper to buy disc material and send the designs out with the material to be cut. Furthermore with previous brake disc quality issues; material can checked in-house before being sent for manufacture.

9.4.2  Pedal Box

The pedal box base and brake pedal is to be made from 1060 aluminium alloy. These components can be water jet cut.

The other key component in the pedal box is the accelerator pedal. The accelerator pedal will have the upright and footplate skins cut from 3-ply CFRP and the upright and footplate cores cut from Rohacell 51 in house on a band saw. Components will be jigged and glued together. The jig is currently in the design phase and is likely to be made from MDF. The jig is to ensure the uprights are positioned correctly against the rear of the footplate.

9.4.3  Steering Column

The steering column will be 5/8” tube made from 4130 steel. This will bought in and cut in house. The column will be welded to the universal joints by one of the Motorsport Centre technicians.

9.4.4  Loom Materials
Before the wiring loom can be manufactured an accurate calculation of wire lengths is required. Using the CAD representation of the vehicle and the routing map, lengths can be measured and recorded in a spreadsheet. An excerpt from this spreadsheet is shown for the rear frame in Figure 9.4.1.

9.4.5 Loom Construction

Before build of the loom begins all measurements will be checked on the chassis. These lengths will be laid out on a board. As each section of the loom is made it will be stuck to the board. Any adjustments can be made such as moving junctions and routing before it is all put together and put into the car.
9.5 Innovative Design Solutions (JS)

9.5.1 DRS and Full Active System

The DRS and Full Active System share the same hardware and this hardware is discussed in this section. The system uses two primary sets of hardware for the operation of active suspension and DRS/active aerodynamics; forced actuators and electromagnetic servos respectively. The selection of the primary hardware, the secondary components and the operational design is discussed below.

9.5.1.1 Active Suspension

The primary hardware used to operate the active suspension are forced actuators, with an actuator used at each corner of the suspension. The actuators are used to exert an external force into the suspension system as directed by the system software. A number of types of forced actuator are available for this operation; including, electro-mechanical, hydraulic and pneumatic. The most appropriate type of forced actuator was concluded to be a pneumatic actuator as this provided a great enough force for the operation required with minimal weight implications. Full research into the different system types is discussed in the Portfolio of Evidence.
The specification of the forced actuator required for use with the active system was determined by analysis of the data gained from simulating the full system about the skid pad track and another test track designed with a higher number of corners and straights to imitate real world use of the system. The force input of the force actuators required by the system was recorded and the cases are shown in Figure 9.5.1 and Figure 9.5.2.

Figure 9.5.1: Force Input required by Forced Actuators over 20s period of Skid Pad Event
Data for maximum force requirement across the two graphs can be evaluated and this value will govern the selection of a specific actuator specification. The maximum force output of a double acting pneumatic force actuator is calculated using the equation below:

\[
F = p \times \frac{\pi \times (d_1^2 - d_2^2)}{4}
\]

*Equation 9.5.1: Force output for pneumatic actuator*

where; \(d_1\) = full bore piston diameter (m)  
\(d_2\) = piston rod diameter (m)

The second governing factor is the actuator length. The overall length of the actuator when fully extended and when fully retracted must be greater than that of the suspension spring.
that the actuator is positioned parallel to. This is to prevent the actuator reaching its minimum or maximum length before the spring and so potentially resulting in failure of the actuator. The minimum length when retracted and maximum length when extended between mounting points of the springs used by Brunel Racing were assessed to be 143mm and 200mm respectively.

Figure 9.5.3: SMC C8S N25mm/100mm Force Actuator
Based on the required specifications of the forced actuators the hardware selected is shown in Figure 9.5.3, with critical specification shown in Figure 9.5.4. The pneumatic system also requires; solenoid valves for control of actuator output, control computer and high pressure reservoir. A full system diagram design specifically for BR-16 is displayed in Figure 9.5.22.
The maximum force output of the forced actuators was found to be 471N, a value greater than the maximum required input of the system for the two cases discussed.

9.5.1.1.1  Suspension Rocker Design

Modification must also be made to the suspension system rockers to allow for the additional mounting of the forced actuators. For system testing purposes modifications were made to both suspension rocker designs of BR-XV and BR-16. The positioning of the mounting points of the actuators was made to ensure a parallel alignment of the actuator to the spring throughout the full motion of the rockers. The design modifications are shown in Figure 9.5.6 and Figure 9.5.7.
Figure 9.5.6: BR-XV Rocker re-design for Forced Actuator mounting

Figure 9.5.7: BR-16 Rocker re-design for Forced Actuator mounting
To evaluate the changes to the structural strength of the rockers post the discussed design adjustments and to certify the safety factor of the new rockers is above that of component failure, FEA analysis of the rockers were undertaken. Prior to FEA analysis the forces acting on the rocker designs must be resolved to find the force components to apply to the FEA model. Figure 9.5.9 illustrates the forces acting on the rocker designs.

Figure 9.5.9: Forces acting on BR-XV modified rocker design

Figure 9.5.8: Forces acting on BR-16 modified rocker design
No forces were set to act in the x-direction as the rockers can be presumed to act perpendicular to this axis.

Results of the FEA analysis are shown in Figure 9.5.10 and Figure 9.5.11. The results show the safety factors for both rocker designs is greater than 1.2 and so is determined as safe for use without risk of failure.

*Figure 9.5.10: Safety Factor analysis of BR-XV modified rocker with actuator force*
The overall mass of the Active Suspension and redesign rocker components for BR-16 is displayed in Table 9.1.

<table>
<thead>
<tr>
<th>Component</th>
<th>Quantity</th>
<th>Mass (g)</th>
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<tbody>
<tr>
<td>Modified Rocker</td>
<td>4</td>
<td>202.66</td>
</tr>
<tr>
<td>Forced Actuator</td>
<td>4</td>
<td>286</td>
</tr>
<tr>
<td>Solenoid Valve</td>
<td>4</td>
<td>60</td>
</tr>
<tr>
<td>High Pressure Reservoir</td>
<td>1</td>
<td>254</td>
</tr>
<tr>
<td>Electrical Components</td>
<td>-</td>
<td>&lt;100</td>
</tr>
<tr>
<td>Total Mass</td>
<td></td>
<td>&lt;2548.64</td>
</tr>
</tbody>
</table>

*Table 9.1: BR-16 Active Suspension components mass*
9.5.1.2  DRS and Active Aerodynamics

Similar primary hardware will be used in the operation of the DRS and Active Aerodynamic system as that used in the Active Suspension system and so a selection between the different types of actuation is made specific to this operation. As previously discussed the total weight and complexity of a hydraulic system does not out weight the performance benefits and so makes it inappropriate for use in a Formula Student Car. Although found to be most suited for the Active Suspension system and used by a number of competitor Formula Student teams for DRS, the data shown in Table 9.2 indicates the improved force output is not required for this operation and so the weight deficit over electro-mechanical servos cannot be justified. The data also highlights the maximum required torque to rotate the wings elements about their pivot points is within the operational specifications of an electro-mechanical servo and so this is considered the most suited hardware.

Figure 9.5.12: Example of pneumatic DRS operation - Monash University 2013
Prior to selecting a specific electro-mechanical servo for use in the system, the mechanical design of the servo-wing element mechanism must be evaluated. The design is optimised for; high mechanical advantage of the servo over the torque required for element rotation, minimal weight, and minimal aerodynamic profile. The final mechanism design for the front wing is shown in Figure 9.5.13, Figure 9.5.14, Figure 9.5.15, Figure 9.5.16, Figure 9.5.17 and Figure 9.5.18. Similar designs are used for the side and rear wings.
Figure 9.5.14: Mechanical Linkage diagram, continuous line denoting rest position, dashed denoting DRS position

Figure 9.5.13: Front wing functional internal mechanism assembly
Figure 9.5.16: Front Wing DRS assembly (transparent)

Figure 9.5.15: Front Wing DRS assembly (exploded components view)
Figure 9.5.17: Front Wing DRS components

Figure 9.5.18: Front Wing DRS mechanism design, orientated to DRS position
The mechanical linkage design between the servo and the pivot point of the wing element must consider the following design specifications:

- The linkage must allow for full rotation of the wing element from the element’s rest position to the full DRS position.
- Rotational of the wing element must be achieved within the operational allowances of the servo.
- Positive mechanical advantage and velocity ratio must be achieved by the servo over the wing element.

These specifications will determine the length of the connecting rod and the initial mounting angles of the lever arms at the servo and wing element. The angle between the lever arms will also be dependent on the prevention of component collision thought the articulation of the mechanism. The calculations to determine the angle output of the servo for desired wing element angle, mechanical advantage and velocity ratio are shown below:

Equation 9.5.2: Mechanical advantage

\[
\begin{align*}
\sum a \cos \theta_2 + b \cos \theta_3 - c \cos \theta_4 - d \cos \theta_1 &= 0 \\
\sum a \sin \theta_2 + b \sin \theta_3 - c \sin \theta_4 - d \sin \theta_1 &= 0
\end{align*}
\]
Above equations are solved simultaneously to obtained equations for \( \theta_4 \), the output angle of the element for given input values of \( \theta_2 \):

\[
\theta_4 = 2 \times \tan^{-1}\left(\frac{-B \pm \sqrt{B^2 - 4AC}}{2A}\right)
\]

*Equation 9.5.3: Output angle*

Where:

\[
A = \cos \theta_2 - \frac{d}{a} - \frac{d}{c} \cos \theta_2 + \frac{a^2 - b^2 + c^2 + d^2}{2ac}
\]

\[
B = -\sin \theta_2
\]

\[
C = \frac{d}{a} - \left(\frac{d}{c} + 1\right) \cos \theta_2 + \frac{a^2 - b^2 + c^2 + d^2}{2ac}
\]

Mechanical Advantage between the servo arm and the element arm is calculated using the following equation:

\[
m_a = \frac{a \sin \alpha}{c \sin \beta}
\]

*Equation 9.5.4: Mechanical advantage between servo arm and element arm*

Where:

\[
\alpha = \theta_2 - \theta_3
\]

\[
\beta = \theta_4 - \theta_3
\]

Velocity Ratio between the servo arm and element arm is then calculated using:

\[
v_r = u \times m_a
\]

*Equation 9.5.5: Velocity ratio*
Where:

\[ u = \text{servo velocity} \]

The data gained from the above equations will be used to select an electro-mechanical servo that will meet the requirements of the system. The key data that must be considered is the required torque output of the servo, operational velocity and total weight. The chosen servo for use in the system, and its specifications are shown in Figure 9.5.20 and Figure 9.5.21.

**Figure 9.5.20: MKS HV767 Titanium Servo**

<table>
<thead>
<tr>
<th>Modulation:</th>
<th>Digital</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torque:</td>
<td>6.0V: 347.2 oz-in (25.00 kg-cm)</td>
</tr>
<tr>
<td></td>
<td>7.4V: 430.5 oz-in (31.00 kg-cm)</td>
</tr>
<tr>
<td>Speed:</td>
<td>6.0V: 0.15 sec/50(^\circ)</td>
</tr>
<tr>
<td></td>
<td>7.4V: 0.12 sec/50(^\circ)</td>
</tr>
<tr>
<td>Weight:</td>
<td>1.79 oz (79.0 g)</td>
</tr>
<tr>
<td>Dimensions:</td>
<td>Length: 1.57 in (39.0 mm)</td>
</tr>
<tr>
<td></td>
<td>Width: 0.79 in (20.1 mm)</td>
</tr>
<tr>
<td></td>
<td>Height: 1.77 in (45.0 mm)</td>
</tr>
<tr>
<td>Motor Type:</td>
<td>Coreless</td>
</tr>
<tr>
<td>Gear Type:</td>
<td>Titanium</td>
</tr>
<tr>
<td>Rotation/Support:</td>
<td>Dual Bearings</td>
</tr>
</tbody>
</table>

**Figure 9.5.21: MKS HV767 Servo Specifications**
The overall mass of the DRS and Active Aerodynamics system for the front wing is displayed in Table 9.3.

<table>
<thead>
<tr>
<th>Component</th>
<th>Quantity</th>
<th>Mass (g)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electro-mechanical servo</td>
<td>4</td>
<td>79</td>
</tr>
<tr>
<td>Main Element Casing</td>
<td>1</td>
<td>326.6</td>
</tr>
<tr>
<td>Second Element Casing</td>
<td>1</td>
<td>195.7</td>
</tr>
<tr>
<td>Second Element Opposing Casing</td>
<td>1</td>
<td>181.2</td>
</tr>
<tr>
<td>Servo Housing</td>
<td>1</td>
<td>53.4</td>
</tr>
<tr>
<td>Servo Arm</td>
<td>1</td>
<td>29</td>
</tr>
<tr>
<td>Carbon Linkage Arm</td>
<td>1</td>
<td>15.1</td>
</tr>
<tr>
<td><strong>Total Mass</strong></td>
<td></td>
<td>730</td>
</tr>
</tbody>
</table>

*Table 9.3: Active Aerodynamics components mass*

9.5.1.3 Full System Control Hardware

Control and activation of the full system is dependent on driver input and so the hardware required for driver UI must be considered. The primary driver UI with all car system is the steering wheel and GEMS display as discussed in 8.6.2.2. This hardware will be integrated into the full active system with activation over mode selection and DRS activation controlled by steering wheel toggle switches/push-hold buttons and feedback displayed on the GEMS display.
The specific active system GEMS display screen will provide the driver with feedback regarding the mode currently selected, ON/OFF control of the DRS. Current use of each force actuator and orientation of each rotational wing element.

The full system integration within BR-16, including pneumatic plumping lines, hardware positions and electrical connections is displayed in Figure 9.5.22.
The force actuators are represented in red, solenoid valves in dark blue, high pressure reservoir in green, airlines in light blue, system computer in yellow and electrical data lines in orange.
9.6 Manufacturing Plan

Similar to the project plan, Microsoft Project was used to create a manufacturing plan. This estimates the amount of time that a part will require to be manufactured either in-house or outsourced based on previous experience and after consultation with workshop technicians.
10 Conclusions

10.1 Project Management (GG, EJ, CE)

10.1.1 Project Planning

Top level decisions were identified early in the year. These formed the basis of the original project plan. During the year as the design phase progressed, deficiencies in this plan were exposed, which led to the need for a new document to be created with more realistic project milestones. The methods used to construct this original plan can be criticised as they did not allow sufficient detail of the design and manufacturing process to be included. The new project plan allowed the team to follow a more accurate critical path to be able to achieve project completion on schedule.

10.1.2 Budget and expenditure Tracking

Tracking the expenditure of the entire project was a new task for the team to adopt this year which will be better executed in following years when they have budgets and can plan expenditure better.

There were many rule changes this year which delayed the development of major components, such as the chassis. This did not directly have an effect on the expenditure, but did delay the design of the chassis and components which interact with the chassis.

The planned expenditure was very optimistic and the team were unable to design the parts and specify materials for the items and materials to be purchased when planned. Many of the parts however came in under budget (those ordered before 15th March) and allowed the project to be under budget by several thousand, which allowed the team to reallocate budget, for example, from competition hotel to machining large vehicle components.
10.2 Vehicle Testing (CE)

In total, four test sessions were carried out through the year using BR-XV. At these, nobody was seriously injured and the vehicle was not involved in any incidents, so this aspect of testing can be considered a success. A number of persistent issues arose at several of the tests, most notable the battery charging problems. Work back at the Motorsport Centre was carried out to rectify these, but in most cases a temporary fix was achieved, only for the issue to occur again. The test sessions generated useful data for the members of the team that required it, which was able to lead to development of future systems and inclusion in level 3 and 5 dissertations. Weather conditions at each test day were generally adequate, allowing each program to be completed when the vehicle did not encounter any mechanical problems that terminated running for the day. Procedures developed at these test days will be carried forward to test sessions carried out in June using BR-16 once it has been constructed.

10.3 Tyres (CE)

The current 13” tyres and a variety of 10” options have been evaluated over a number of different performance metrics so that an informed decision on wheel size can be made. The 10” wheel appears to be weaker in some respects, but in combination with a lower unsprung mass, it should be able to build on its strengths and allow the vehicle to perform well.

10.4 Unsprung (EJ)

The suspension this year was a redesign and recalculation of all major components, resulting in new concept parts, the majority of which were carried through before. The new suspension geometry with its changeability will aid the car well in tuning and set-up, but
also throughout the cars lifecycle, allowing future development on the suspension by testing the car in these different positions to design new suspension geometry.

By switching to the 10” wheel Rim, the 3-Stud wheel bolt pattern, integrated brake disc and wheel mounting flange on the hub, 3rd/heave springs and many ARB settings, there were many packaging exercises performed which meant every part on the suspension was drawn in CAD to check clearances. The hubs and uprights had the most design development in terms of design hours and it shows in the resultant mass reduction in these parts; a 50% reduction in front hub mass.

The packaging of the car was overall improved, which in part was down to the number of components constructed in the full vehicle model in CAD, and partly down to the improved design of each part, both inboard and outboard.

All of the suspension team members contributed to the project and all Level 3 design projects were conducted by the individual, with guidance and support from the Level 5 department manager and management team where relevant.

10.5 Chassis (GG)

Whilst it is felt that the use of a monocoque construction will be beneficial to the dynamic behaviour of the vehicle for the reasons set out in section 2.6.4 the mass benefits of the design have not been realised to their full potential due to changes in the regulations. The chassis is still however likely to be lighter than the spaceframe of BR-XV and will be significantly stiffer. What was anticipated to be the lightest monocoque tub constructed by BR by around 3.5kg is ultimately likely to be heavier than many due to the level of unintended reinforcement required. It is however worth noting that none of the panels previously used in BR-XI, 12 or 13 would be legal for the current SES requirements.
Given that test derived UTS is now the biggest limiting factor on the design, the panel which was designed is now at completely the wrong end of the spectrum. A panel was designed for minimum mass with maximum EI whilst providing sufficient shear strength. As such a very thin skinned panel was designed with very thick core.

It was found that test derived UTS values vary hugely from the data-sheet facing skin mechanical properties, with this now required to be used to prove legality, hand calculations and FE modelling are no longer suitable for specification of a panel, the panel testing now needs to be one of the first steps in the design of the monocoque structure.

The poor UTS values are in a large part due to the unpredictable nature of the honeycomb core material. The test derived UTS values assume a skin failure however the panel designed would fail via a core failure; whilst this has been the case for all panels designed since BR-X, the previous panels would have experienced skin and core failure at more similar loads, whereas the panel designed for BR-16 had relatively low facing skin stress at the predicted failure load.

Whilst core failure was predicted by hand calculations, the predicted core stress was typically around half the published capability of the core material in its weakest direction and closer to a quarter of the stronger direction. It was found that the thick panel specified was too stiff to bend in the typically documented manner; with skin separation rather than skin area begin responsible for so much of the panel’s stiffness, the panel was not able to deform significantly before the load applicator would begin to crush the core leading to failure.
As the skins were so thin, they were insufficiently stiff in isolation to distribute the load across an area of core; once the core begins to crush the panel will no longer take additional load. This behaviour was backed up by the apparent common displacement of the load applicator at the point of panel failure regardless of skin thickness.

Thicker skins did however allow for more load to be taken by the panel due to the better distribution of load into the core from the stiffer skin, however this benefit was offset by the increased thickness and thus cross-sectional area reducing the test derived UTS.
10.6 Aerodynamics (GM)

Figure 10.6.1 Final render of the 2015 Aerodynamic package

<table>
<thead>
<tr>
<th></th>
<th>Lift</th>
<th>Drag</th>
<th>Moment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front</td>
<td>250N</td>
<td>50N</td>
<td>41 N/m</td>
</tr>
<tr>
<td>Wing</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
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</table>

<table>
<thead>
<tr>
<th></th>
<th>Code</th>
<th>Alpha</th>
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<tr>
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<td>13/-35mm</td>
</tr>
<tr>
<td></td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rear</td>
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<td>Chord</td>
<td>Height</td>
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<td>Sidewing</td>
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<td>Alpha</td>
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<td>Height</td>
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<td></td>
<td>Flap</td>
<td>N/a</td>
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</tr>
</tbody>
</table>

Figure 10.6.2: Aerofoil specification and Final results at 11.1m/s
The outcome of the project was overall positive. The aerodynamics package develops a total of 250N of downforce at 11 m/s. Even though this is less than half of the downforce target set at the beginning of the year it can be considered an extremely positive result. In fact this year’s work on aerodynamics:

- Sets a benchmark design for the new FSAE aerodynamics regulations
- Generates almost the same downforce (456N compared to 537N at 15m/s) as the previous design although new strict regulations were applied.
- Produces a third of the drag (109N compared to 313N @ 15m/s) of drag compared to last year design
- Improves the lift to drag ratio from 1.71 (BR-XV) to 4.18 (BR-16)
- Studied deeply aerodynamics devices mounting methods, providing an early and detailed design of such components.
- Resolved the problem of the pitching moment by having an adequate distribution along the vehicle of the extra aerodynamics forces and their locations

It is also worth mentioning that the downforce target was set well before the release of the new aerodynamics regulations. This target was also set on a vehicle dynamics point of view rather than an aerodynamics point of view: by creating 670N of downforce at 11.1 m/s the car would have probably (and unrealistically) won the skid-pad event with no difficulties.

One of the greater achievements of this year’s project was to successfully perform an aerodynamic track test for the first time in Brunel Racing history. The outcome of the test allowed to:

- Successfully evaluate the Coefficient of Drag or BR-XV
• Successfully evaluate the downforce generated by BR-XV
• Measuring the pressure distribution on the rear mainplane of BR-XV
• Verify the validity of the CFD model used at Brunel Racing
• Give a guidance for future managers on how to perform aerodynamics track testing
• Improve the quantity and quality of measurements tools at Brunel Racing disposal

10.7 Driver Controls & Electronics (TM)

10.7.1 Brake System

Calculations have shown that the brake lines have been over specified in the past and that smaller lines should be able to hold the required braking pressures.

10.7.2 Pedal Box

The pedal box design meets the tight packaging restrictions. The accelerator pedal not only withstands 1000N of force but also meets its objective of a mass of 270g or less. The pedal itself has a mass of 55g due to its CFRP-Rohacell sandwich composite construction.

![Figure 10.7.1 Pedal box in situ](image)
The pedal box also meets its objective. It allows adjustment for a range of drivers from 5th female to 95th percentile male. It also meets its target of reducing the mass of the pedal box by 10%. The estimated mass reduction is about 50% greater than the design in BR-XV. Figure 10.7.1 shows the pedal box in situ. Note the mounting holes are not shown on the chassis but the pedal box is positioned for a 95th percentile male driver.

10.7.3 Steering Column

The original steering column geometry was not suitable for application. After the geometry was altered the steering column meets the packaging restrictions.

10.7.4 Driver’s Cell

Proposed switch locations have largely been dictated by packaging. However, use of GEMS display and electronic steering boss means more information can be displayed to the driver so that there is no need for indicators behind the wheel. The seating will be formed by using epoxy foam when all components in the cockpit are finalised and in place.

10.7.5 Electronics

Increase in sensors will lead to an increase in loom mass. Steps have been taken to reduce this impact by the use of smaller gauge wire. Clear routing plans and wire length prediction should ensure a neat loom. In turn, this should make it easy to trace any electrical issues and therefore meet the objectives. The ECU and data logger have been located in the foot well of the monocoque. This is purely from packaging restrictions. Fortunately, in this location they are easily accessible and meet the objective (just not purely by design).
10.7.6 Level 3 Management

Of the three level 3 students, two have fully engaged in the process and have provided (with guidance) helpful research and designs to be implemented on BR-16. Contact was lost with one level 3 and despite substantial effort in re-establishing contact, no work has been demonstrated to the manager. The research project undertaken by this level 3 is of little impact to BR-16 but will be executed by the manager if required at a later date. See Driver Controls and Electronics Portfolio for management information.

10.8 Innovative Design Solutions (JS)

The Innovative Design Solutions sub-team has explored a number of avenues for innovative design within Formula Student, studying future design concepts, application of advanced systems for the formula and the inclusion of advanced design methods for Brunel Racing. The result is the successful design and development of an upgrade package for BR-16.

The field of Active Aerodynamics, not previously explored by Brunel Racing, has been developed for use on a Formula Student car, tackling the low operating velocities and requirement of varied aerodynamic characteristics for each Formula Student dynamic event. The system has been developed with integration into the current BR-16 aerodynamic package design, using all available aerodynamic components for improved aerodynamic performance. Drag Reduction Systems have further been incorporated into the Active Aerodynamics system, utilising the same hardware and enabling further control of the aerodynamic behaviour of the car during certain operating conditions.

Designed and developed in coherence with the Active Aerodynamics system, an Active Suspension has created further control and improvement of the vehicle’s handling, resulting
in the development of a fully active system for BR-16. The system has proved through simulation testing to dramatically improve the handling and performance of BR-16, with up to 83% reduction in roll angle and 50% reduction in pitch during the Formula Student Skid Pad Event.

The fully active system has been designed with full driver UI to allow for control over system modes, testing and activation during operation of the car. The system also has specifically designed fail safes to prevent component failure and prevent vehicle system damage.

The upgrade package also features the development of an engine cover, designed to reduce the production of drag behind the cockpit and within the engine bay. The final design iteration significantly reduces drag in this region, improving the overall lift-to-drag ratio of BR-16 by 0.61 and refining the flow of fluid over the rear wing main plain.

Although not discussed in this report, two Level 3 projects were managed within the sub-team, with a similar focus on innovative design and future development. The projects resulted in the successful development of wing structure design, enabling improved wing element structure and overall component weight reduction. These designs will provide the following Brunel Racing teams with greater performance benefits and the basis for continuous advancement.

10.9 Vehicle Simulation (CS)

It is clear from simulation results that when compared to test data results, although results for BR-XV were close to those of the physical test data, they were not as close as desired. This could be a result of poor vehicle parameterisation, poor driver parameterisation or lack of test data for comparison.
BR-16 on the other hand seemed to provide similar patterns of data and similar peaks, generally being closer to the physical test data. The only exception is that the simulation car was much slower around the hairpin than both the BR-XV model and the physical test data. This could be a result of the driver parameterisation as opposed to other factors, because the driving characteristics had to be changed to less aggressive to ensure BR-16 completed the simulation around Bruntingthorpe kart track. The BR-16 vehicle model also reached higher peak speeds on track than the BR-XV model, with an improved acceleration.

Simulation of BR-16 around the FSUK Sprint/Endurance track suggested that the car would do well in this event, although physical test data is yet to be obtained for BR-16 as it is still at the manufacturing stage. As the model has not been validated it is difficult to suggest that the values achieved by the simulation car are accurate. Although based on the comparison against test data around Bruntingthorpe kart track, it suggests that generally the vehicle model is slower than in a real test and has difficulty with tight corners. This would imply that BR-16 has the capability to improve on the sprint/endurance lap time set by the BR-16 simulation vehicle at FSUK 2015. This is especially the case when remembering that the BR-16 vehicle model still has 13 inch wheels rather than 10 inch ones, which should also yield an improved lap time.

Simulation has been a useful tool for the creation of a vehicle model, although further refinement of the model and simulation conditions is required. A larger amount of test data could also prove an advantage in validation of the vehicle model.
11  Further Work

11.1  Project Management (GG, EJ, CE)

11.2  Marketing and Sponsorship (GG)

11.3  Vehicle Testing (CE)

11.3.1  Load cases gathered during BR-16 testing to inform various design aspects

During the design stage of many components where FE analysis is conducted, load cases have to be created to simulate real-world conditions. In some cases, particularly for the suspension, the application of strain gauges to wishbones etc. would be of great benefit when it comes to designing components for the next year.

11.4  Tyres (CE)

11.4.1  TTC data/track comparison

The TTC data was gathered in very much idealised conditions. The team would benefit greatly from conducting some track testing to determine such parameters as the lateral and longitudinal friction coefficients and normal loads acting on the tyre. This would allow the TTC data to be scaled correctly to give a more realistic interpretation of tyre behaviour.

11.4.2  18.0x6.0 vs. 18.0x7.5

In the next year, the TTC is planning to conduct another round of tyre testing with the hope of testing another of the tyres that will be of interest to the team; the Hoosier 18.0x7.5 R25B.
11.5 Unsprung (EJ)

The use of test data to analyse kinematics around a lap and post-process the kinematic data in Matlab is a useful tool and I would recommend it for future years. This could be improved by using data from competition, allowing the kinematics to be run through several tracks.

Despite trying to test and gain Roll gradient data this year, it was only possible to pull the data from track tests. Ideally a roll gradient test should be done on a dry skid-pad with varying ARB settings and compare to calculated data to theory.

Compliance in Suspension is something which is not wanted, however it will always be there and minimising it is important when improving vehicle dynamics. By replacing the dampers with mock dampers and loading the car, a plot of camber with ride height change can be made. By replacing the dampers, this graph will show how much compliance there is in the hubs, upright, bearings and wheels.

With completely new suspension geometry and smaller diameter tyres, there will be need and want for improvement in the kinematics and the control of the suspension. Designing this will be aided by the increased number of vehicle dynamics sensors on the car, allowing for a more in-depth understanding of what is happening to the car.

11.6 Chassis (GG)

11.6.1 Full Car Manufacturing

As the chassis manufacture has been delayed due to legality issues much of the chassis remains to be manufactured. It is hoped that a significant portion of this work will be completed by the start of Term 3.
At the time of writing the SES has been successfully completed as have the mandatory crash tests and IAD submission.

### 11.6.2 Torsional Stiffness Validation

One of the projects involved the manufacture of a rig to allow torsional stiffness to be correctly measured at the wheels. At the time of writing the rig is around 50% complete and likely to be completed in the coming weeks, ultimately due to the delays in putting BR-16’s chassis and some suspension components into manufacture the car cannot be tested yet, however once completed this data will be delivered at the competition.

### 11.6.3 Panel Specification for Future Cars

The bulk of the chassis work undertaken as part of this project was unexpected and not in the area expected, unsurprisingly the biggest area for future development is again the panel, the direction taken for the panel construction for BR-16 proved ultimately to be as far as possible from the ideal solution when the 2015 regulations were published particularly due to the test derived UTS requirements.

Earlier work looked at the use of CFRP skins, however given the poor shear strength of CFRP it was determined that around twice the thickness of skin would be required negating any mass benefits although increasing stiffness [116]. Now that it has been determined that a thinner overall panel and a higher skin-core thickness ratio is required, the use of CFRP skins may be the most optimal solution under the current regulations.

It is therefore recommended that thinner cored panels are assessed with thicker skins but using CFRP to minimise mass owing to its density of 1600kg/m³ in comparison to 2700kg/m³ for aluminium. As the values used must be taken from test derived values, it is advised that
standard core thicknesses are used as these can be purchased in smaller quantities for testing purposes prior to full-car quantities being bought. It is likely that the thickness of skins now required to have sufficient EI and UTS together will mean that the shear test, once the defining problem for previous authors, will be the formality.

11.6.4 Hard Point Development

Although the integrated hard-point approach utilised is possibly the best solution on a technical level it has proven to be highly problematic in practice. The required thickness meant extensive machining work was required, holding up manufacture of the honeycomb panels.

From a morale point of view, the absence of a chassis within the workshop has been a huge blow to the project. It is advised that the previously used “top hat” method is used once more, or that standard thickness inserts are used with standard thickness core to allow rapid water-jet cutting of inserts, rather than time and resource intensive machining operations used for BR-16.

11.6.5 Crash Structure

Whilst one of the projects for BR-16 intended to develop a composite crash structure, the project ultimately failed to make sufficient progress to enable testing in time for competition submissions. It is however felt that this is probably the correct route to pursue for the future; as such the tooling for the nose structure is intended to be suitable for use in manufacturing a composite crash structure for test purposes next season.
11.7 Aerodynamics (GM)

The 2015 aerodynamics manager suggest to:

- Develop a customised front wing main plane aerofoil designed for ground effect
- Design and develop a light-weight under tray
- Explore new manufacturing methods to reduce costs
- Perform CFD simulations at different dynamics vehicle state (e.g. Yaw, Pitch)
- Improves CFD simulations by exploiting the GPU solving capabilities
- Perform a full car wind-tunnel test

11.8 Driver Controls & Electronics (TM)

11.8.1 Brake System

Further work needs to be conducted on brake disc design. At a future test day on BR-16 data can be gathered on disc temperature using brake temperature sensors. The test will also consist of discussion with the driver(s) concerning brake feedback before a final design decision is made.

11.8.2 GEMS Display

It is likely that further work will be conducted on the design and implementation of extra features for the GEMS display. The screen has a feature that enable messages (via SMS) to be displayed on screen. This can be used to communicate vital information to the driver or ask him to come in.

11.8.3 Telemetry

Live data transfer from the car can be initiated by using two nano-stations. One nano-station situated in the car will be able to send data to the other (in the pits for example)
when in range. This information can be relayed to the driver if any urgent actions are required.

11.9 Innovative Design Solutions (JS)

As previously discussed, the extensive design phase planned for the sub-team has resulted in physical testing of certain systems later into the final development stages of BR-16. Furthermore with the reliance on components specifically integrated into BR-16 and not utilised by BR-XV, physical testing of the fully active system cannot be performed until BR-16 is built and fully operational. Final optimisation of the fully active system, the calibration of the aerodynamic forces produced by the wing elements and the final setup for each Formula Student dynamics events will be carried out when BR-16 is ready for final testing.

The system optimisation will require an extensive testing procedure, with particular vehicle handling characteristics monitored, and weighting values of the system function altered until optimal performance is achieved. This process will require data from a number of event scenarios recorded by BR-16. Similar procedures will be required for the calibration of the Active Aerodynamics, with testing of BR-16 travelling at varied velocity to allow for the orientation of each wing element in all allowable instances. This testing will provide the system with highly accurate data relating to the request for aerodynamic force input.

The final design iteration of the engine cover design for BR-16 was developed during the design phase of other sub-systems, in particular powertrain sub-systems, and so as previously discussed, the design will have to be modified to ensure cohesion with the final design iterations of the powertrain components. Once finalised, the engine cover will also be tested for heat transfer from the powertrain, with any amendments made to prevent overheating and ensure constant fluid ventilation.
11.10 Vehicle Simulation (CS)

To further refine the vehicle model and make it more accurate, the input of kinematics data needs to be investigated. Accurate kinematics data must be gained whether by calculation or by physical test to ensure that the vehicle dynamic behaviour is as expected.

Investigation of the Simulink for CarMaker tool would also be a benefit to the vehicle model. This would enable the input of an accurate clutch, launch control and traction control model.

The use of a user parameterised driver as opposed to the racing driver used in this project may also be of use, as it provides the user with the capability to define driver characteristics. IPG Driver could be set to exhibit the characteristics of one of the test drivers in order to closer match the physical test data.

Physical test data must also be obtained in order to validate the BR-16 vehicle model once the vehicle has been manufactured.
12 References


[38] Brown and Miller Racing Solutions, Interviewee, [Interview]. 4 November 2014.


[68] Bosch, “MS4 Sport ECU Function sheet,” e.n., e.l., 2012.

[69] Bosch, “C60 Data Logger,” e.n., e.l., 2012.


