University of Birmingham

School of Engineering

Integrated Design Project 3

FINAL GROUP REPORT

Mechanical Engineering

MEng

Team Number	1
Group Number	Mech MEng 1

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Feedback (Compulsory Section)		
Reflecting on the feedback that we have received on previous assessments, the following issues/topics have been identified as areas for improvement:	1	Ensure consistency throughout the report.
	2	Justifying decisions and backing them up
	3	Repetition of irrelevant information
In this assignment, we have attempted to act on previous feedback in the following ways:	1	Keep the formatting consistent throughout the whole report, with labelled figures and tables.
	2	All of briefs points met.
	3	Tables set out clearly to show relevant information
Feedback on the following aspects of this assignment (i.e. content/style/approach) would be particularly helpful to us:	1	Overall Formatting
	2	Use of Figures and Tables
	3	Justification of choices

Abstract

Meng Mechanical Group 1 was tasked with designing and developing the suspension, steering, braking, tyres, and the vehicle dynamics system for the fully electric Toyota Hilux, adapted to compete in the 2028 Dakar Rally. The project aimed to demonstrate the viability of sustainable electric mobility in extreme off-road environments, while maintaining competitive speed, safety, and reliability.

This report presents the design and optimisation of a robust chassis dynamic package suited to desert racing, capable of withstanding extreme terrain while maintaining vehicle stability, control, and comfort. A double wishbone suspension system with dampers was selected to balance ride quality and handling precision, whilst also being sustainable and energy efficient by including regenerative shocks. Steering and braking systems were developed to ensure responsive control and reliability under repeated high-load conditions, while tyre selection and sizing were optimised for traction across sand, rocks, and uneven terrain.

Integration with other subgroups was critical throughout the design process. Close collaboration with the powertrain team informed weight distribution strategies, particularly in accommodating the 1,200 kg battery pack. The suspension geometry was co-developed with the chassis and bodywork team to ensure proper packaging and structural compatibility. The report further discusses wider engineering impacts, including sustainability, safety, human factors, and the commercial potential of an electrified off-road platform.

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Introduction

The Dakar Rally is considered to be the most challenging endurance motorsport events in the world that demands exceptional vehicle resilience, control and reliability. Traditionally dominated by internal combustion engine vehicles, the rally landscape has evolved with the rise of electric vehicles, presenting new engineering challenges and opportunities in terms of drivetrain packaging, weight distribution, and system integration.

One of the critical issues in electrifying off-road vehicles is managing the significant mass of battery packs, which greatly affects a vehicle's weight distribution. Unlike traditional ICE vehicles, where the weight is either biased by the engine placement, electric vehicles have the battery mounted centrally and low in the chassis, effectively mimicking a mid-engine configuration. This setup lowers the centre of gravity and improves lateral stability during cornering and high-speed off-road manoeuvres.

The unique packaging and torque characteristics of electric drivetrains also necessitate revised approaches to suspension and steering design. 'The use of independent suspension systems like double wishbone helps in improving the handling and ride quality of electric all-terrain vehicles' (Samanta et al., 2023). Additionally, 'EV's are heavier and produce more torque than gas-powered vehicles, meaning that tyres need to be more durable while also maintaining low rolling resistance and minimal noise' (Hawkins, 2023). Adhering to the sustainability goals, they should also be considered to disposability and being environmentally friendly.

This report presents the design and development of the suspension, steering, braking, tyres and vehicle dynamics for an electrified Toyota Hilux, exploring technical requirements, innovative technologies, and integration within the overall vehicle architecture, cooperating closely between the other teams.

Aims & Objectives

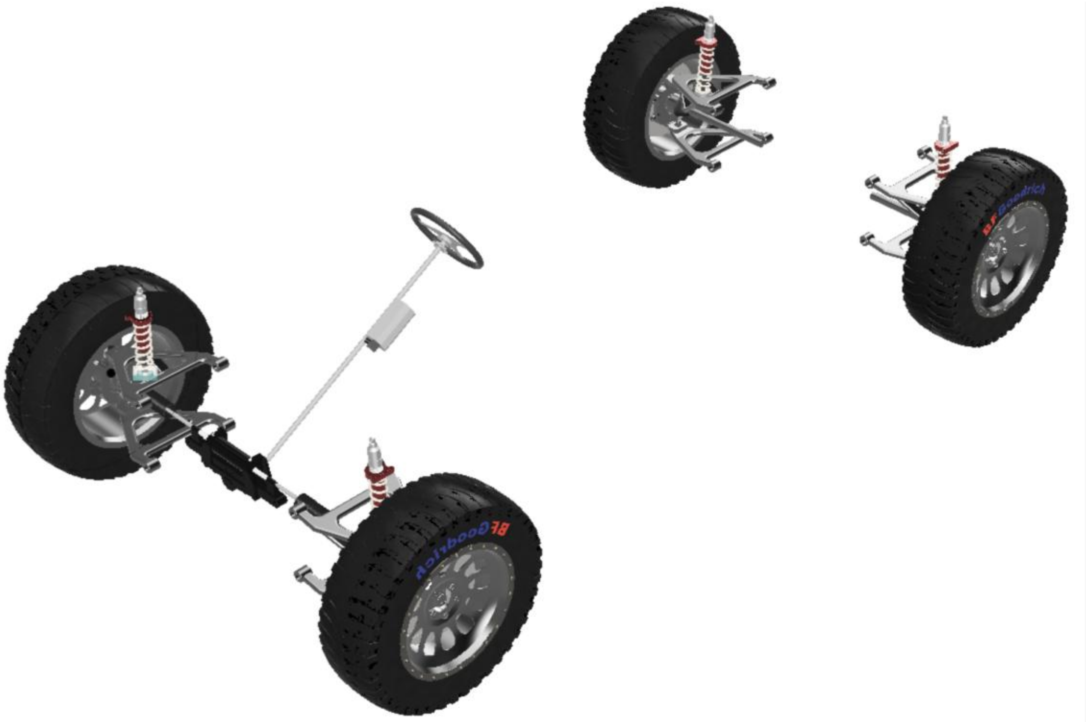


Figure 1: Final Render

Aim:

Design and optimise the suspension, steering, braking, tyres, and vehicle dynamics of a fully electric Toyota Hilux for competitive performance, safety, and reliability in the 2028 Dakar Rally.

Objectives:

1. **Analyse Rally Vehicle Dynamics:**
   Assess necessary adaptations for Dakar Rally conditions.
2. **Research Innovative Technologies**
   Evaluate the pros and cons of integrating innovative technologies in the vehicle.
3. **Calculate and Model Specifications**
   Determine relevant forces, stresses, and dimensions; develop component models.
4. **Validate using Simulations**
   Use FEA and CFD to simulate stresses on the suspension and steering and brakes.
5. **Prototype**
   Create a physical prototype to demonstrate the innovation of the product.
6. **Risk Assessment and Business Case**
   Develop a risk assessment and business case with costs and funding strategies

Group Project Management Strategy

A group management strategy was implemented to ensure efficient collaboration and task delivery throughout the project, with the first thing being to have each member complete the Belbin team role test to assign responsibilities based on individual strengths, shown in Table 1.

Table 1: Group Members and Roles

Member	Role
Diljit Singh (Suspension)	Coordinator
Aqib Mahmood (Steering & Braking)	Complete Finisher
Vinay Mahal (Steering & Braking)	Complete Finisher
Ayodeji Joesph (Steering & Braking)	Specialist
Grant Rogan (Suspension)	Implementor
Keranveer Mann (Steering & Braking)	Team Worker
Sayhan Ali (Suspension)	Specialist
Saeed Ahmed (Suspension)	Team Worker

Following this, the group was divided into topic specific groups, aligning members based on their confidence and technical knowledge to ensure a distributed workload. To maintain communication, the group used:

- Microsoft Teams: for secure document sharing and formal updates.
- WhatsApp Community: for rapid idea exchange and informal discussion.

Weekly meetings, scheduled in addition to IDP lectures, were used to:

- Review completed tasks.
- Assign new actions.
- Highlight any technical issues or dependencies.

Thursday sessions facilitated cross group integration, ensuring compatibility between subsystems like chassis, aerodynamics, and powertrain.

A formal meeting log was maintained (Table 2), allowing clear alignment of goals, tracking of deliverables, and documentation of group progress.

Table 2: Group Meeting Log

Date	**Attendance**	**Tasks to be done**	**Tasks to be completed by the next meeting?**	**Absences**
11/02/25	DS, AJ, KM, SA, SA, GR, VM, AM	Decide what who's going to do what in the project and see what we're going to do with the presentation	Start presentation and research on individual areas.	
10/03/25	DS, AJ, KM, SA, SA, GR	Decide what areas to reduce focus on and split areas to members.	Receive mass calculations and split workload between members.	AM, VM
17/03/25	DS, GR, SA, KM, VM, AJ, AM	Research on component types and where to source them from.	Continue research and start CAD on suspension and steering.	SA
24/03/25	DS, SA, SA, GR, AJ, KM, VM	Continue CAD and finalise research on component types.	Finish vehicle dynamics sections.	AM
28/03/25	DS, AM, VM, AJ, KM	Focus on what should be put into the document and clarify any issues with separate parts	Do FEA on brakes and wishbones. Continue with CAD of steering.	GR, SA, SA
31/03/25	DS, GR, SA	Calculations have to be redone.	Complete calculations and focus on prototype planning.	AM, VM, AJ, KM, SA
04/04/25	DS, GR	Redo calculations for spring and damper	Find suitable spring and damper and finish model	Full group not required
22/04/25	DS, GR, AM, VM, AJ, KM	Discuss making of prototype, finalising and formatting the report.	Do CAD drawings, reduce calculations to tables, redo business case.	SA, SA

25/04/25	DS, GR, VM, KM, AM	Think about how to reduce word count and format into tables.	Add remaining parts and reduce into tables.	SA, SA, AJ
29/04/25	DS, GR, VM, KM, AM, SA, SA, AJ	Finalise report and start formatting.		

Product Design Specification

Table 3: Product Design Specification

N.o	Parameter	Specification
Suspension Design & Kinematics		
1.1	Operation	Double Wishbone Suspension system capable of absorbing large impacts and maintaining control and comfort over rough terrain.
2.1	Performance	Basic Dimensions: - Ride Height = 325mm - Track Width = 1800mm - Wheelbase = 3000mm - Wheel travel = 350mm
2.2		Spring Characteristics: - Natural frequency = 2Hz - Spring stiffness = 90N/mm - Spring travel = 205mm
2.3		Energy Recovery System: - Minimum power recovered 2kW
3.1	Weight	Sprung (per axle) = 895kg Un-sprung (per axle)= 196kg
4.1	Material	Minimum yield strength = Minimum torsional stiffness =
5.1	Expense	Bought-in components: - WP XPLOR PRO 8946 SHOCK = £1,160/unit - K-Tech shock absorber spring = £98.17/unit - Total cost (6 units) = £7,550
5.2		Material cost: - Less than £500
Steering/Braking & Tyres		
6.1	Operation	Ensure reliable braking and steering performance under extreme conditions (top speeds of 204 km/h, rough terrain, mud, water)
7.1	Steering spec	Steering system type, Electro-Hydraulic (EHPS) with high-pressure assist (Rexroth EHP + hydraulic rack)
7.2		Effective steering ratio, 15:1 to 18:1 (optimal balance between stability and low-speed manoeuvrability)
7.3		Turning circle, ≤ 13 m (realistic for Dakar pickup platform; depends on steering angle & arm length)
7.4		Steering input torque at the wheel: ≤ 25 Nm (driver effort at the wheel), with EHPS providing up to 300+ Nm assist at the rack input.
8.1	Weight	Steering System Mass: ≤ 18 kg (including hydraulic rack, assist pump, reservoir, hoses, and mounting hardware)
8.2		Braking System ≤ 40 kg (total, including calipers, discs, pads, and mounting hardware)
8.3		Tires and Wheels ≤60 kg per corner
9.1	Clearance	Sufficient clearance between tyres and suspension/brake components under full articulation; minimum 25 mm clearance from wheel arch to tyre at full bump.
10.1	Operation	Effective and reliable braking system to provide optimal deceleration on various surfaces.

10.2		Hydraulic Disk Brakes
		Tyres - Must maintain performance at temperatures exceeding 50°C surface temperature, typical of desert stages.
10.2		Tyres - Operational pressures must support tuning between 17–31 psi, depending on terrain type, while maintaining safe bead retention — especially during high loads, sidewall deflection, and uneven terrain.
11.1	Brake Performance	Target deceleration - 8 m/s^2 in ideal conditions (Dakar vehicles typically decelerate between 5 - 7m/s^2 on sand.)
11.2		Brake force required ~17,500 N (based on 2,200 kg mass & deceleration of 8 m/s^2)
11.3		Brake circuit layout – Front / Rear split
11.4		Brake bias - Front: 65% / Rear: 35%
11.5		Brake Type: High-performance disc brakes, preferably vented, with reinforced callipers for heat dissipation.
11.6		Brake discs with maximum operating temperature of at 600°C
12.1	Tyres spec	Tyres - Tread compound should balance grip and wear, with a medium compound preferred for mixed stages.
12.2		Open, self-cleaning tread blocks, sidewall protection, durability for high loads and long stages
12.3		Durability - Tyres must withstand at least 500–800 km of mixed terrain per stage without significant loss of performance or structural damage.
Vehicle Dynamics & Weight Distribution		
13.1	Top Speed	204 km/h
14.1	Acceleration	< 6 seconds.
15.1	Max Lateral Acceleration	1.2-1.5g.
16.1	Weight Distribution	50:50 (Front:Rear)

Suspension Design & Kinematics

Suspension Requirements and Selection

The purpose of suspension systems is threefold; to maintain contact between the wheel and road surface, to isolate the shocks generated at the wheel from the driver and to stably transfer the braking and acceleration forces through the body of the vehicle. To ensure this, the following requirements were considered throughout our design:

High Strength Components – Dakar vehicles can momentarily experience forces of up to 16g, equivalent to almost 35 tonnes, producing large moments and shear forces through the components.

Driver Safety – The natural frequency of the suspension system must be considered to design for the highest loads the vehicle will experience. The typical range for rally cars is 1.5-2.0Hz (Rajeev and Sudi, 2019) which balances performance needs with safety considerations.

Stability and Performance – All bought-in parts must meet the requisite standards to enhance performance and ensure safety. To achieve this, any bought components will be sourced from notable World Rally Raid suppliers.

As part of our contribution to Sustainable Development Goal (SDG) 9, we have researched the implementation of energy recovery systems within the suspension. Magneto-piezo harvester modules have shown promise in research environments but are yet to be introduced in racing. We intend to integrate such a system into the shock absorbers to enhance battery longevity and vehicle range.

Alhumaid et al. (2021) used this style of generator to produce 242W at 120km/h on class D roads, for a small-scale electric vehicle. An earlier study by Xie and Wang (2015) found a piezoelectric bar harvester model could recover around 740-980W from a quarter car model, on class D roads, that was 170kg lighter than our model, at 126km/h.

Both studies found that increasing the vehicle speed, road roughness and the size of the piezo components and magnets increased the energy recovered in the generator. Therefore, from the available data, it is extrapolated that our model could utilise a harvester module capable of recovering between 2-5kW of energy per wheel, based on the larger and faster vehicle, maximum average speed of 170km/h, as well as the rougher road surfaces, class E or rougher. This will account for 6.7% of the motor power rating, extending the range of the battery noticeably.

Table 4: Spring Parameters and Values

Parameter	Value
k_s (Spring Stiffness)	$70.7N\,mm^{-1}$
k_t (Tyre Stiffness)	$85{,}000Nmm^{-1}$
k_r (Effective Ride Rate)	$70.67Nmm^{-1}$
f_s (Natural frequency of sprung masses)	$2Hz$
f_u (Natural frequency of unsprung masses)	$4.69Hz$
δ_f (Front static deflection)	$123\ mm$

Table 5: Damping coefficients

(Ns/m)	C_{crit}	C ($\zeta = 0.7 \times C_{crit}$)
Sprung	11,246.9	7,872.8
Un-sprung	5,775.8	4,043.1

Table 6: Rebound/Bump Values

	C(Ns/m)
Rebound	5,623.4
Bump	2,249.4

Materials and Design for Sourced Parts

Spring and Damper

We are using the WP XPLOR PRO 8946 SHOCK and K-Tech 90N 46x220mm spring for our shock absorber system.

This was determined using a quarter-car model and the following calculations:

Spring travel -

$$\Delta Spring = MR \cdot \Delta Wheel = 0.586 \cdot 350 = 205mm$$

Spring stiffness -

$$k_r = (2\pi f_s)^2 \cdot m_s = (2 \cdot 3.14 \cdot 2)^2 \cdot 447.5 = 70.67Nmm^{-1}$$

$$k_s = \left(\frac{1}{k_r} - \frac{1}{k_t}\right)^{-1} = \left(\frac{1}{70.67} - \frac{1}{85{,}000}\right)^{-1} = 70.7Nmm^{-1}$$

Table 7: Assumptions used in sourced parts calculations

Motion ratio	Found from wishbone geometry	
Natural frequency	Taken from research data, see bottom of p. 9	
Tyre stiffness	Found from quarter mass under 16g of acceleration. It can be assumed the tyres will undergo the maximum force with minimal deformation as this is their function	545.5*16*9.81 = 85,622N

Wheel Hub

The wheel hub connects the wheels to the suspension system and ensures safe transmission of forces between the two. A 5-bolt configuration was chosen due to its widespread availability and proven strength, ensuring easy replacement if damaged during competition.

The Ridex 653W0122 Wheel Hub was chosen, made of steel to withstand high torque and stress loads. Ridex offers reliable precision engineering at a competitive cost, making it ideal for long term use.

Refer to Figure 1 for hub selection.

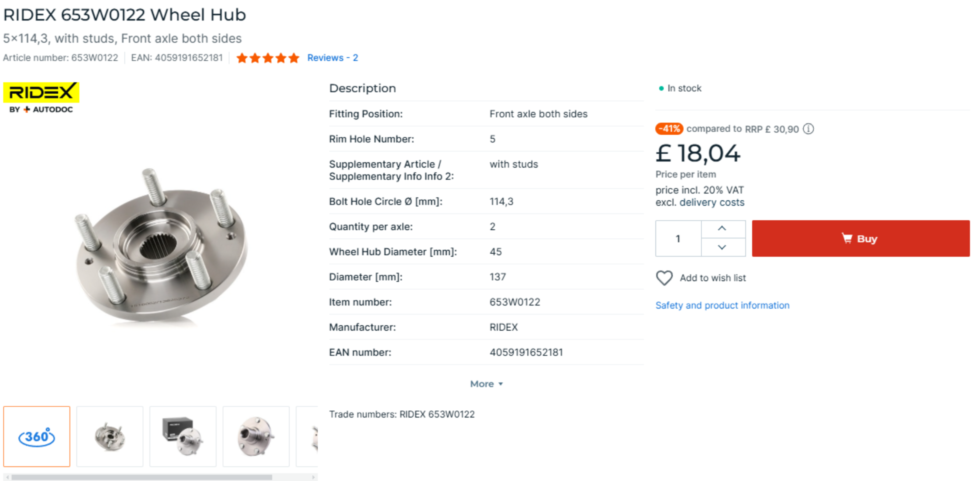


Figure 2: Wheel Hub Manufacturer

https://www.autodoc.co.uk/ridex/13628372?utm_medium=cpc&utm_source=google&tb_prm=21789142049&gshp=1&gad_source=1&gclid=CjwKCAjw47i_BhBTEiwAaJfPps5_NijevOO5HCU8XmAgiygqCR9jj3R3VrQk-2S5Z0Cu4qhfhHm76RoCnRwQAvD_BwE

Materials and Design for Bespoke Parts

The bespoke suspension system is inspired by the X-Raid Mini JCW Buggy, chosen due it proven performance in off-road rally conditions, particularly in endurance events like the Dakar Rally. Core components include the wheel hub, knuckle, wishbones, and the spring and damper assembly.

The wishbones and knuckle are going to be manufactured in-house, whereas the spring and damper and wheel hub are going to be bought in.

Selection of materials was based on the following minimum property requirement:

Table 8: Minimum Material Properties

Property	Value
Tensile Strength	Minimum 500 MPa
Yield Strength	Minimum 350 MPa
Fatigue Strength	Minimum 250 MPa
Density	Minimum 7000 kg/m^2
Young's Modulus	Minimum 70,000 MPa
Impact Toughness	Minimum 100 kJ/m^2
Cost	Maximum £15/kg

Knuckle

The knuckle connects the wishbones to the wheel hub and must withstand substantial torsional and bending loads. Figure 2 shows the selection graph from Granta Edupack.

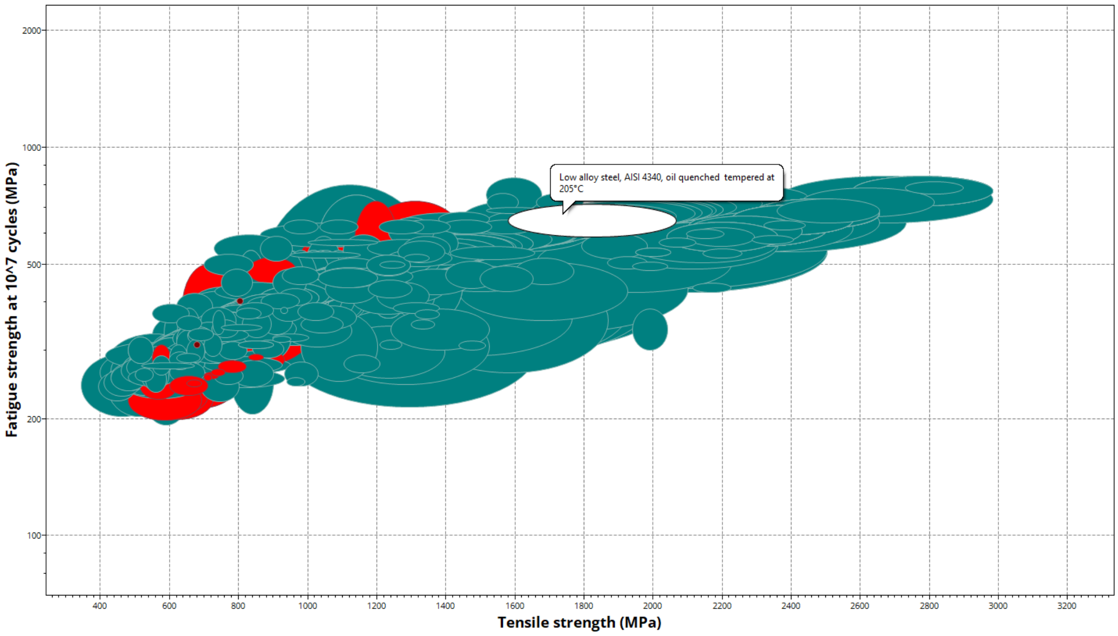


Figure 3: AISI 4340 Fatigue vs Tensile Strength

Table 9: AISI 4340 Material Properties

Property	Value
Tensile Strength	1580-2070 MPa
Yield Strength	1510-1850 MPa
Young's Modulus	205,000-213,000 MPa
Fatigue Strength	589-712 MPa
Elongation	8 - 12 % strain
Toughness	9.58-20.1 KJ/m^2

A higher strength material was chosen since the knuckle undergoes high stress from both wishbones, whilst being oil quenched and tempered means that it has higher toughness and less likely to fail under impact loading. It also has excellent fatigue stress, allowing for prolonged periods of stress on the knuckle.

Load Analysis – Knuckle

In order to validate the material choice, the following calculations for bending stress and moment were done.

Table 10: Upper Wishbone Parameters

Parameter	Values
Steering Torque (T)	15,000 Nm
Radius (r)	0.09 m
Polar Moment of Inertia (J)	0.0001031 m^4
Vertical Force ($F_{vertical}$)	85,622 N
Distance ($d_{vertical}$)	1.085 m
Outer Diameter (D)	0.09 m
Inner Diameter (d)	0.045 m
Section Modulus (S)	0.0001342 m^3

Table 11: Upper Wishbone Calculations

Calculation	Formula	Value
Torsional Stress ($\sigma_{torsion}$)	$\frac{T}{J} \times r$	$\frac{15{,}000}{0.0001031} = 145.49\ GPa$
Bending Moment (M)	$F_{vertical} \times d_{vertical}$	$85{,}622 \times 1.085 = 92{,}900\ Nm$
Bending Stress ($\sigma_{bending}$)	$\frac{M}{S}$	$\frac{92{,}900}{0.0005726} = 162.16\ GPa$
Torsion Safety Factor (SF)	$SF = \frac{Yield\ Strength}{Stress}$	$\frac{1510}{145} = 10$
Bending Safety Factor (SF)		$\frac{1510}{162} = 9$

Both of these values show that there is a high enough safety factor that we don't have to worry about it failing. The front axle has a mounting point for the steering arm, whereas the rear doesn't.

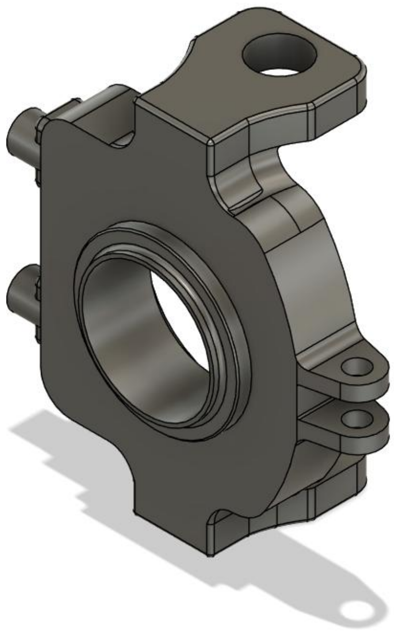


Figure 5: Front Axle Knuckle

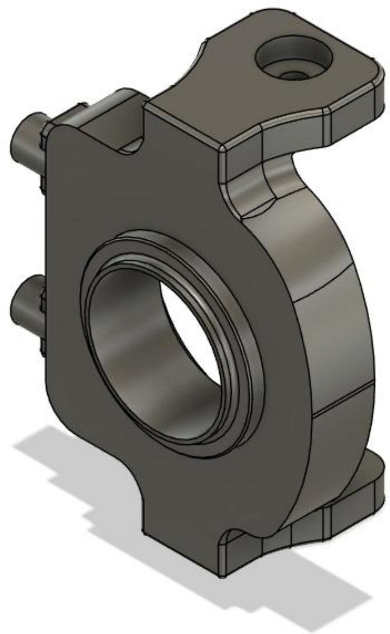


Figure 4: Rear Axle Knuckle

Wishbones

The wishbones serve as the structural links between the wheel hub/knuckle and the chassis. 4130 steel is the material of choice as it has great strength and fatigue resistance, since suspension arms endure significant loads and stress cycles. Figure 6 below shows the fatigue and tensile strength of 4130 steel compared to other metals, with Table 12 showing properties.

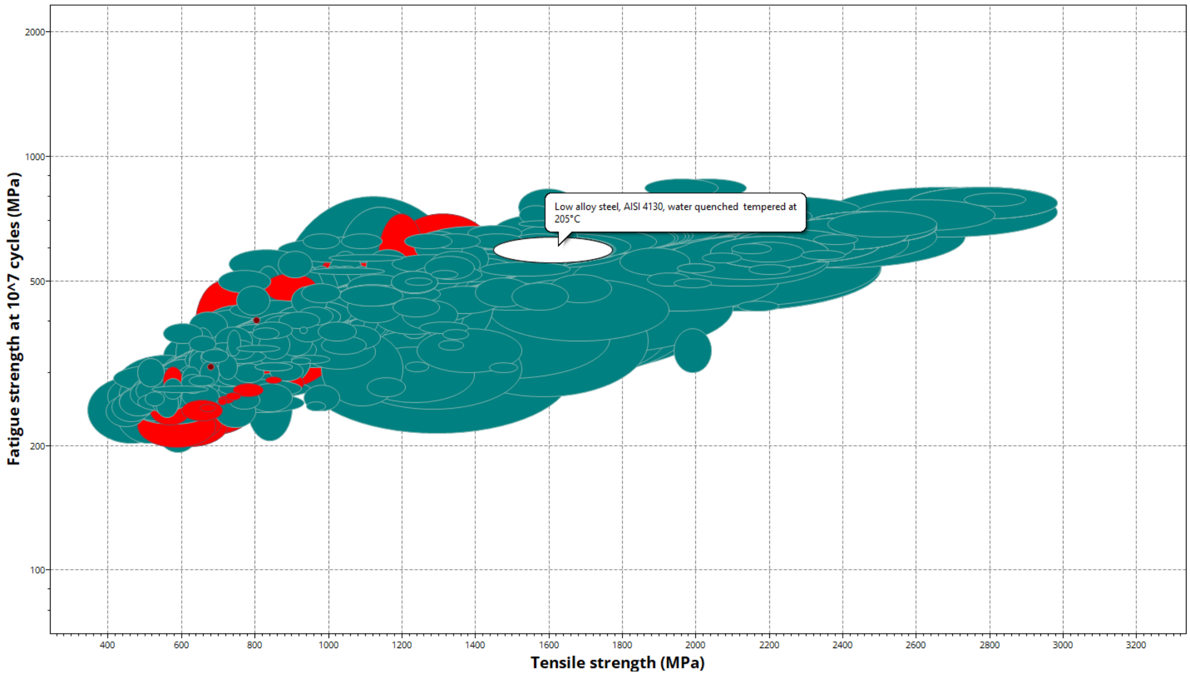


Figure 6: AISI 4130 Fatigue vs Tensile Strength

Table 12: AISI 4130 Material Properties

Property	Values
Tensile Strength	1450 - 1770 MPa
Yield Strength	1320 - 1610 MPa
Young's Modulus	210,000 - 216,000 MPa
Fatigue Strength	555 - 639 MPa
Elongation	8 - 12 % strain
Toughness	4.33-12.6 KJ/m^2

Load Analysis – Upper Wishbone

In order to validate the material choice, the following calculations for bending stress and moment were done.

Table 13: Upper Wishbone Parameters

Parameter	Values
Vertical Force ($F_{Vertical}$)	85,622 N
Vertical Load Distance ($d_{Vertical}$)	1.085 m
Steering Torque ($T_{Steering}$)	3,400 N
Steering Moment Arm ($d_{Steering}$)	0.6 m
Section Modulus (Rectangular) (S)	0.0001471 m^3
Width (w)	0.1 m
Height (h)	0.06 m

Table 14: Upper Wishbone Calculations

Calculation	Formula	Value

Vertical Bending Moment ($M_{Vertical}$)	$F_{Vertical} \times d_{Vertical}$	$85{,}622 \times 1.085 = 92{,}900\ Nm$
Steering Bending Moment ($M_{Steering}$)	$T_{Steering} \times d_{Steering}$	$3{,}400 \times 0.6 = 2{,}040\ Nm$
Total Bending Moment (M_{Total})	$M_{vertical} + M_{steering}$	$2{,}040 + 92{,}900 = 94{,}940\ Nm$
Sectional Modulus (S)	$\frac{hw^2}{3 + (1.8 \times \frac{w}{h})}$	$\frac{0.06 \times 0.1^2}{3 + (1.8 \times \frac{0.06}{0.1})} = 0.0001471\ m^3$
Bending Stress ($\sigma_{bending}$)	$\frac{M_{Total}}{S}$	$\frac{94{,}940}{0.0001471} = 645.41\ GPa$
Bending Safety Factor (SF)	$SF = \frac{Yield\ Strength}{Stress}$	2

The sectional modulus was simplified to a rectangular shape with the width being 0.1 m and the height being 0.06 m, to make estimating the safety factor easier. The safety factor confirms that AISI 4130 is suitable for the wishbones, and since the top wishbone is larger than the bottom, and would take more of the stress, the material is suitable for the bottom one too.

FEA (Upper wishbone FEA)

To validate the structural integrity of the upper wishbone, FEA was performed under the most critical loading scenarios: maximum vertical force during full compression and a static loading condition representing the vehicle at rest. These simulations aimed to assess whether the component would operate within the elastic region of the material chosen.

The wishbones were simulated using AISI 4130 steel, with the appropriate boundary conditions at the pivot, spring mount, and knuckle connection, based off the values calculated from maximum wheel load and spring connection below.

The maximum load is 85,622 N, spring stiffness of 70,700 N/m, and spring travel of 205 mm.

The force applied to the spring mount during full compression is calculated using:

$$Fspring = k \times x = 70{,}700 \times 0.205 = \mathbf{14{,}994\ N}$$

We assume ‘the force distribution in double wishbone suspension typically places 30% of vertical load on the upper wishbone when the spring/damper is mounted there (Srinivasan and Selvaraj, 2021):

$\therefore Fknuckle = 85{,}622 \times 0.3 = \mathbf{25{,}687\ N}$

This is applied to the ball joint mounting point where the top of the knuckle connects.

For static conditions, we assume that it will be 1/6th as ‘for high-performance rally vehicles, static loads are often just 1/6 of the dynamic peak load’ (Wong, 2008):

$Fstatic\ wheel = 85{,}6226 = \mathbf{14270\ N}$

From this we can work out the force on the knuckle, knowing the upper wishbone load being 30%:

$Fstatic\ knucle = 14{,}270 \times 0.3 = \mathbf{4281\ N}$

Assuming that the spring compresses 0.1m under static conditions:

$$Fstatic\ spring = 70{,}700 \times 0.1 = \mathbf{7{,}070\ N}$$

Although only the final simulations are shown, multiple load cases were considered during the process, with various meshing sizes. A single point force was applied at the knuckle joint and damper mounting, with the chassis mounting pointed having a fixed joint.

FEA Results:

Static Loading Simulation

In Figure 8, the highest stress is concentrated around the junction between the 2 arms, which is expected as it's the primary load transfer areas.

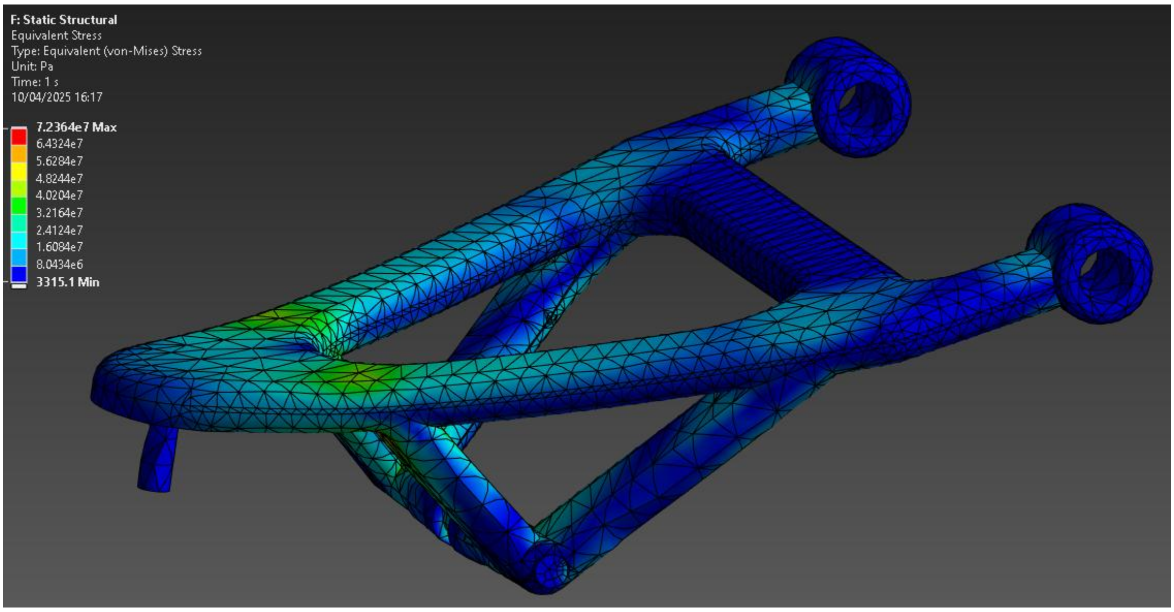


Figure 7: Top Wishbone Static Equivalent Stress

In Figure 9, the maximum deformation occurs at the front of the upright mount, which is consistent with expectations.

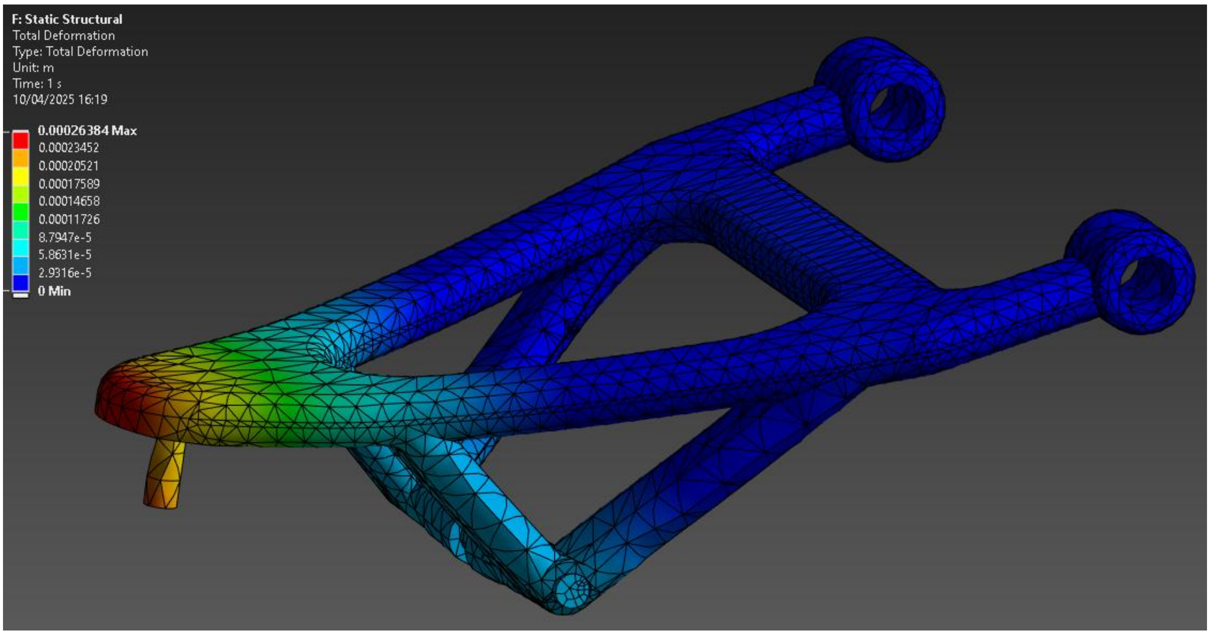


Figure 8: Top Wishbone Static Deformation

Table 15 shows the maximum stress is well below the materials yield strength.

Table 15: Static FEA Wishbone Results

Maximum Equivalent Stress	72,364,000 Pa
Minimum Stress	3315.1 Pa
Maximum Displacement	2.6384e-004 m
Deformation Pattern	No signs of buckling, smooth and continuous across arm.

Max Load

In Figure 9, the maximum stress occurs at the joints from the mounting points to the main wishbone. It is still below the yield strength of the material.

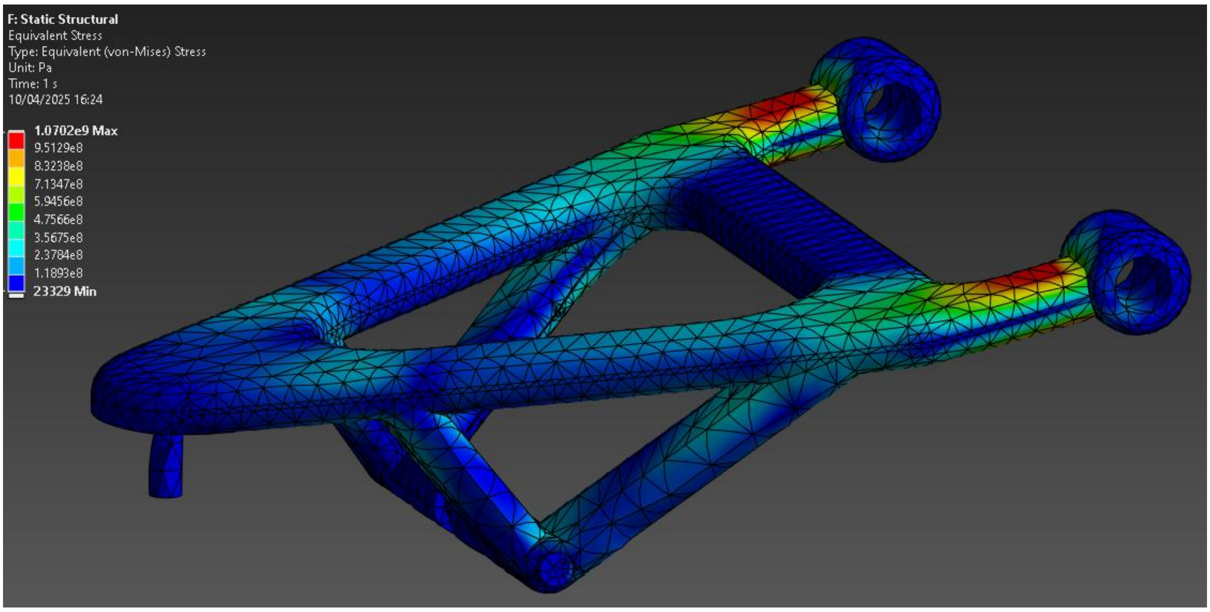


Figure 9: Top Wishbone Maximum Load Equivalent Stress

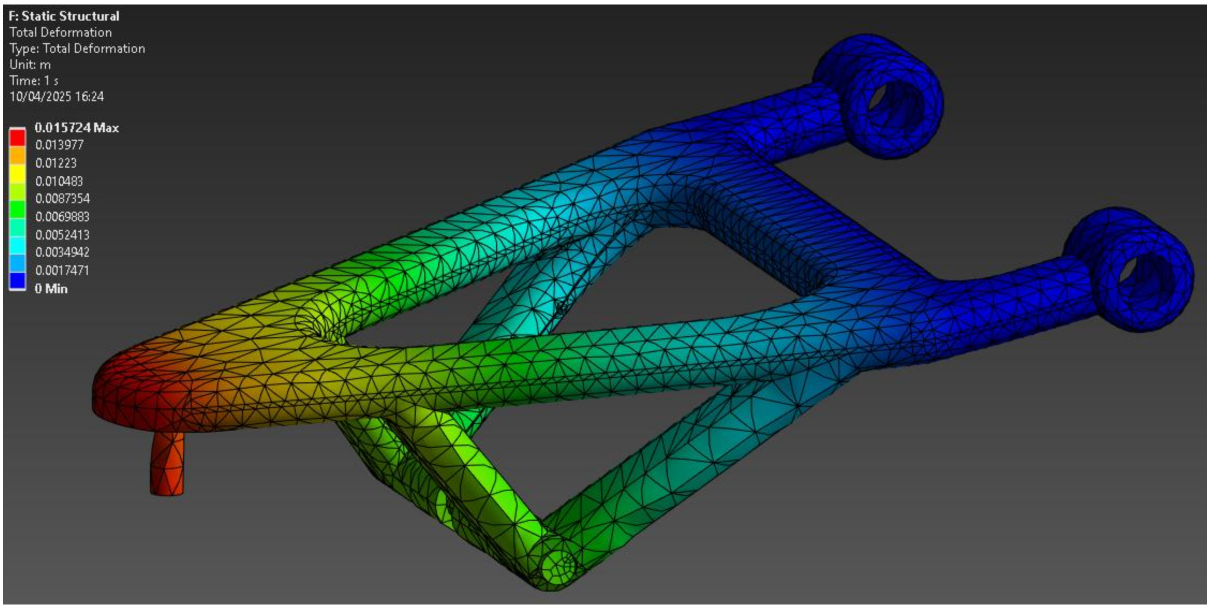


Figure 10: Top Wishbone Maximum Load Deformation

Table 16: Maximum Load FEA Wishbone Results

Maximum Equivalent Stress	1,070,200,000 Pa
Minimum Stress	23329 Pa
Maximum Displacement	1.5724e-002 m
Deformation Pattern	Smooth deformation, no buckling

There are limitations with this simulation, such as simplified geometry, load application, and no joint flexibility. Despite this, the analysis provided a confident evaluation of the wishbone's behaviour under peak loads. The safety factor calculated shows that the values would remain within acceptable limits.

Manufacture of Suspension

The bespoke wishbones and steering knuckle were manufactured in-house due to their complex geometries, high-load demands, and prototype-specific tolerances. Dakar Rally conditions can exceed 16g vertical impact, requiring precise material selection and fabrication control.

Wishbones

Fabricated from AISI 4130 steel for its excellent fatigue resistance and weldability, the wishbones use a rectangular cross-section for simplicity and strength.

Manufacturing Process:

1. Cutting & Preparation: Tubes and plates are cut to length, cleaned, and fish-mouthed for accurate joints.
2. Jig Setup: Components are aligned in a custom jig to maintain geometry.
3. TIG Welding: Precision welding ensures structural strength with minimal distortion.
4. Machining: CNC milling is used for spring mounts and bush housings.
5. Finishing: Parts are bead-blasted and optionally powder-coated.

Knuckle

Machined from AISI 4340 billet steel, the knuckle was chosen for its high yield strength and fatigue resistance under dynamic off-road loading.

Manufacturing Process:

1. Billet Shaping: Rough turning and CNC milling define the profile and mounting faces.
2. Drilling/Boring: Precision holes are machined for ball joints and bushings.
3. Heat Treatment: Oil quenching and tempering improve strength and durability.
4. Inspection: Critical dimensions are verified using calipers and bore gauges.

Assembly & Fixtures

MDF jigs were laser-cut to hold wishbones during welding, and soft jaws were used in CNC vises to protect surfaces. Final alignment was validated with spherical bearings before surface finishing.

Steering System, Braking Systems & Tyres

Steering

Figure 11: Rack and Pinion Steering Model

Table 17 outlines the selected steering system components, detailing specifications aligned with Dakar Rally performance requirements.

Table 17: Steering Parameters

Component	**Specification**
Steering Rack (1):  	**Product:** Howe Performance 2.5 Trophy Truck Rack - 113 3.0 X 6.0 TRAVEL POWER RACK W/S 1.5:1 **Description:** Heavy-duty rack designed for off-road racing applications. **Estimated Cost:** $1,200–$1,500 USD **Source:** Race Ready Products. *Assumptions:* *Rexroth EHP pump (6) output P = 120 bar = 12000 kPa* *Rack Piston area = 2.5 in^2 = 0.00161 m^2* *Steering Arm Length = 510 mm* *Pinion radius (r) = 0.017 m* *Calculations:* *Rack Force (hydraulic output),* $F = P \times A = 12000 \times 0.00161 = 19.3\ kN\,(rack)$ *~19,300 N (based on 120 bar)*

	Pinion Torque (at rack input), $T = F \times r = 19300 \times 0.017 = 328\ Nm\ (pinion)$ *~328 Nm* *Steering Angle (outer wheel),* *~25°* *Steering Angle (inner wheel),* *~40–42° (with Ackermann)* *Wheel Torque at Knuckle,* $T = 19300 \times 0.51 = 9843\ Nm\ (wheels)$ *~9843 Nm per side*
Tie Rods & Heim Joints (2): 	**Product:** Kartek Off-Road Custom Made 4130 Chromoly Tie Rods **Description:** Custom-made tie rods for 7/8” Heim joints, suitable for heavy-duty applications. **Estimated Cost:** $300–$400 USD per pair **Source:** Racegear WA
Adjustable Steering Column (3): 	**Product:** Kreger Fabrication Adjustable Tilt Steering Column **Description:** Designed for 3/4” steering shafts, suitable for custom applications. **Estimated Cost:** $150–$300 USD **Source:** Kartek
Quick Release Steering Hub (4): 	**Product:** Sparco Bolt-On Quick Release Hub Kit 6x70 mm **Description:** Allows for rapid steering wheel removal, enhancing driver accessibility. **Estimated Cost:** $150–$200 USD **Source:** MSAR

Steering Wheel (5): 	**Product:** Sparco R383 Champion Steering Wheel **Description:** Designed for rally applications, providing excellent grip and control. **Price:** From £110.00 (Excl. VAT) **Source:** GSM Performance / BOFI Racing
Electro-Hydraulic Power Steering Pump (6): 	**Product:** Rexroth EHP Series Electrohydraulic Pump **Description:** Reliable electro-hydraulic pump with a 1-year warranty. **Estimated Cost:** Approximately \$1,000–\$1,200 USD **Source:** Alibaba - Rexroth EHP Series *<u>Power Steering Pump Kit (Alternative):</u>* *Product: KRC Cast Iron SBC Pump Kit* *Description: Includes pump, mounting brackets, spacer kit, and reservoir tank.* *Estimated Cost: \$360 USD* *Source: Speedway Motors*
Additional Steering Components (Local Assembly):	Hydraulic Hoses (AN-6/8) - High-pressure rated hoses Fittings & Adapters - AN-to-rack and AN-to-pump fittings Reservoir (Remote) - Compatible with EHPS system Steering Fluid Cooler - Suitable for high-performance applications
Additional Extra Components (Local / Online):	U-Joints & Shaft Couplers - Compatible with steering column and rack

	Firewall Bearing/Grommet - Suitable for steering column pass-through Column Support Brackets - Custom fabricated or off-the-shelf Mount Brackets (Pump & Reservoir) - Custom fabricated or off-the-shelf Power Wiring & Relay - High-current rated components Steering Fluid - High-temp synthetic steering fluid (e.g., Red Line, Motul)

Table 18 compares the design requirements to the actual specifications achieved by the custom steering system.

Table 18: Design Requirements

Parameter	Required (Target Spec)	Actual (Your System)	Status
Driver Input Torque @ Wheel	≤ 25 Nm	~10–20 Nm	Meets
Rack Assist Torque (Pinion)	≥ 300 Nm	~328 Nm (from Rexroth EHP @ 120 bar)	Exceeds
Steering Force at Rack	≥ 15,000 N	~19,300 N	Exceeds
Torque at Knuckle (Wheel)	≥ 8,000–10,000 Nm (typical Dakar)	~11,580 Nm (with 600 mm steering arms)	Exceeds
Steering Angle (inner wheel)	≥ 38–42°	~40–42° (with Ackermann)	Meets
Rack Travel	6–7 inches	6 inches (152 mm)	Meets
Rack Ratio	≤ 2.0:1 (faster preferred for Dakar)	1.5:1 (fast)	Meets
System Type	Electro-Hydraulic (EHPS)	EHPS (Rexroth + hydraulic Howe rack)	Correct

Table 19 presents the estimated cost breakdown of each component, leading to a total system cost projection.

Table 19: Steering Cost Breakdown

Component	Selected Part / Description	Estimated Cost
Steering Rack	Howe Performance HOW-113 (3.00” pinion, 6” travel)	£2,864
Electro-Hydraulic Pump (EHP)	Rexroth EHP, 120 bar (from Alibaba / motorsport vendors)	£880
Hydraulic Hoses & Fittings (AN-6/8)	High-pressure PTFE lines + adapters + returns	£280
Fluid Reservoir	Remote high-temp reservoir with filter	£120

Steering Fluid Cooler	Off-road rated (Setrab or equivalent)	£120
Steering Column	Kreger Fabrication adjustable column (for 3/4" shaft)	£200
Quick-Release Hub	Sparco bolt-on 6x70mm	£144
Steering Wheel	Sparco R383 Suede, 330 mm	£112
Tie Rods + Heim Joints	Kartek 4130 chromoly custom rods (7/8" or 3/4")	£240
U-joints + Column Hardware	Flaming River or Sweet MFG joints and couplers	£160
Mounting Brackets (Rack + Pump)	Fabricated in-house or outsourced	£120
Power Wiring + Fuse/Relay for EHP	High-current cable, weatherproof relay, inline fuse	£80
Steering Angle Sensor (optional)	CAN-output sensor for telemetry or VCU	£120
TOTAL COST		**£ 5200**

Braking

The braking system was designed to offer reliable performance and safety in extreme Dakar Rally conditions. The main specifications were:

- Resist vehicle speeds of up to 204 km/h
- Optimise braking power without locking up the wheels and losing traction on a variety of off-road surfaces (sand, gravel, rocks)
- Maintain wear resistance through endurance stages (~400–1000 km/day), with minimal replacement of brake pads and rotors (Timeout Racing, 2025)
- Regulate high brake disc temperatures (up to 400–600°C) with minimal fade and keep the rotor material under failure points such as thermal cracking, excessive warping, or fatigue (apracing.com, n.d.)

Table 20 notes that values used to calculate the braking parameters.

Table 20: Braking Parameter Values

Mass (kg)	Wheelbase (m)	Front axle to COG (m)	Rear axle to COG (m)	COG height (m)	Projected area (m^2)	Wind speed (km/h)	Wind density (kg/m^3)	Coefficient of air resistance	Max speed (km/h)
2182	3.085	1.5425	1.5425	0.8101	3.231	18.5 (5.15 m/s)	1.225	0.394	204 (56.67 m/s)
Coefficient of friction	Gravity (m/s^2)	Incline angle	Pedal ratio	Master Cylinder Diameter front (m)	Master Cylinder Diameter rear (m)	Piston area front (m^2)	Piston area rear (m^2)	Brake disc coefficient front	Brake disc coefficient rear
0.6	9.81	0	5	0.0254	0.0254	0.004913	0.003267	0.52	0.42
	Brake disc effective radius front		Brake disc effective radius rear		Tire SLR front		Tire SLR rear		
	0.1575		0.14575		0.4699		0.4699		

Table 21 shows the ideal braking curve equations, which are shown on the graph in Figure 11.

Table 21: Ideal Braking Curve Equations

x-axis	$F_{x,f,ideal} = \left(1 - \psi + \frac{CoG_h}{l} a\right) aW$
y-axis	$F_{x,r,ideal} = \left(\psi - \frac{CoG_h}{l} a\right) aW$

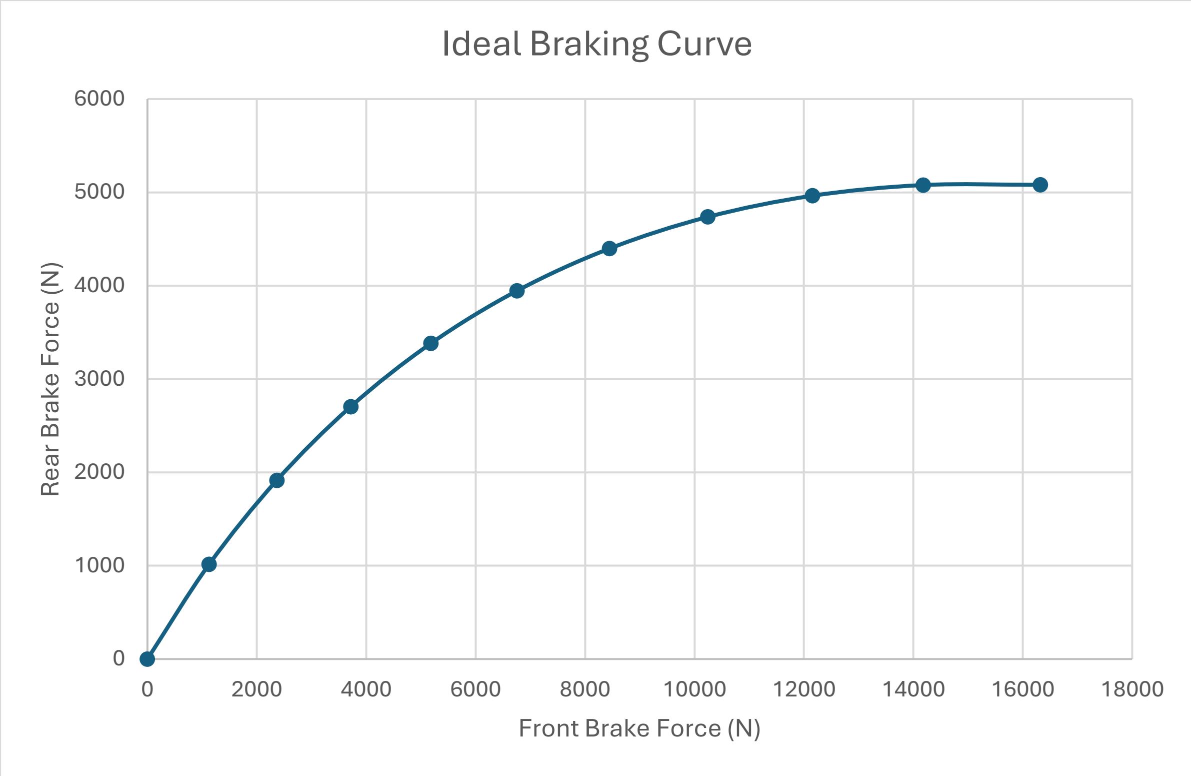


Figure 12: Front vs Rear Brake Force

Table 22 shows the calculations done in order to find suitable brakes for our vehicle.

Table 22: Braking Calculations

Brakes Stopping Force			
Front	$F_{input} = 333.54$ N $F_{cylinder} = F_{input} n_{brake\ ratio}$	333.54×5	1667.7 N
	$P_{f,cylinder} = \left(\frac{F_{cylinder}}{\pi d^2_{cylinder}}\right) \Big/ 2$	$\left(\frac{1667.7}{\pi \times 0.0254^2}\right) \Big/ 2$	3291248.05 Pa
	$F_{x,f,clamping} = P_{cylinder} A_{piston}$	$3291248.05 \times 0.004913$	16169.90 N
	$F_{x,f,total\ clamping} = 4 F_{x,f,clamping}$	4×16169.90	64679.61 N
	$F_{x,f,braking} = \mu_{pad} F_{x,f,total\ clamping}$	0.52×64679.61	33633.40 N
	$F_{x,f} = \frac{r_{f,disc}}{h_{tire}} F_{x,f,braking}$	$\frac{0.1575}{0.4699} \times 33633.40$	11273.16 N
Rear	$F_{input} = 333.54$ N $F_{cylinder} = F_{input} n_{brake\ ratio}$	333.54×5	1667.7 N

	$P_{r,cylinder} = \left(\frac{F_{cylinder}}{\pi d_{cylinder}^2}\right) \Big/ 2$	$\left(\frac{1667.7}{\pi \times 0.0254^2}\right) \Big/ 2$	3291248.05 Pa
	$F_{x,r,clamping} = P_{cylinder} A_{piston}$	$3291248.05 \times 0.003267$	10752.51 N
	$F_{x,r,total\ clamping} = 4F_{x,r,clamping}$	4×10752.51	43010.03 N
	$F_{x,r,braking} = \mu_{pad} F_{x,r,total\ clamping}$	0.42×43010.03	18064.21 N
	$F_{x,r} = \frac{r_{r,disc}}{h_{tire}} F_{x,r,braking}$	$\frac{0.14575}{0.4699} \times 18064.21$	5603.02 N
Braking Force and Acceleration of Braking System			
Total	$F_{x,total} = F_{x,f} + F_{x,r}$	$11273.16 + 5603.02$	16876.18
Acceleration	$a_{x,ovr} = \frac{F_{x,total}}{M}$	$\frac{16876.18}{2182}$	7.73 ms^{-2}
Rolling Resistance			
	$f_r = 0.0136 + .40 \times 10^{-7} v_x^2$	$0.0136 + .40 \times 10^{-7} 185.5^2$	0.0150
Front	$N_f = \frac{Mgc}{L} + \frac{Ma_x h}{L}$	$\frac{2182 \times 9.81 \times 1.5425}{3.085} + \frac{2182 \times 7.73 \times 0.8101}{3.085}$	15131.83 N
	$R_x = f_r N_f$	0.0137×15131.83	226.62 N
Rear	$N_r = \frac{Mgb}{L} - \frac{Ma_x h}{L}$	$\frac{2182 \times 9.81 \times 1.5425}{3.085} - \frac{2182 \times 7.73 \times 0.8101}{3.085}$	6273.59 N
	$R_x = f_r N_r$	0.0137×6273.59	93.96 N
Drag Force			
	$F_d = \frac{\rho}{2} C_D A_f v_x^2$	$\frac{1.225}{2} \times 0.394 \times 3.231 \times 51.52^2$	2069.35 N
Overall Deceleration, stopping distance, and stopping time			

	$F_{ovr} = \mu(F_{x,f} + F_{x,r}) + R_{x,f} + R_{x,r} + F_d + Mgsin(\theta)$	$0.6 \times (11273.16 + 5603.02) + 226.62 + 93.96 + 2069.35 + 2182 \times 9.81 \times sin(0)$	12515.64 N
	$a_{x,ovr} = \frac{F_{ovr}}{M}$	$\frac{12515.64}{2182}$	$5.74\ ms^{-2}$
	$s = -\frac{v_0^2}{2F_{ovr}}M$	$\frac{56.67^2}{2 \times 12515.64} \times 2182$	$279.92\ m$
	$t = \frac{v_0}{F_{ovr}}M$	$\frac{56.67}{12515.64} \times 2182$	$9.88\ s$

Figure 12 shows the layout of the braking system.

Pedal Box (Tilton 850-Series)

↓

Master Cylinders (Tilton 78-Series 25.4mm / 25.4mm)

↓

AN-3 Stainless Steel Braided Brake Lines

↓

Brake Bias Adjuster + Balance Bar

↓

Brembo 6-Piston Front Calipers / Brembo 4-Piston Rear Calipers

↓

Brembo Brake Pads (RB 340 Front / RB 330 Rear)

↓

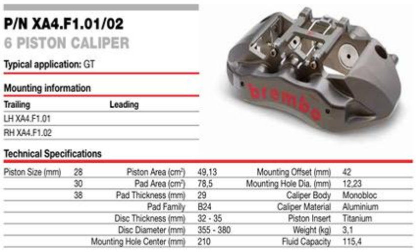
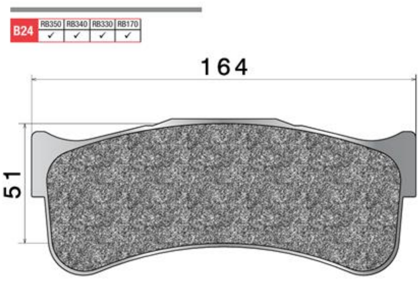
Floating High-Carbon Cast Iron Discs (380mm Front / 355mm Rear)

Figure 13: Braking System Layout

Table 21, 22 and 23 show the components that were selected based off the calculations done above.

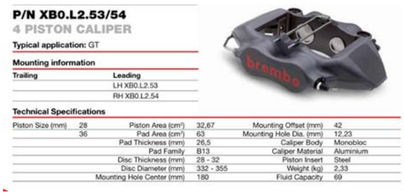
Front Brake Components

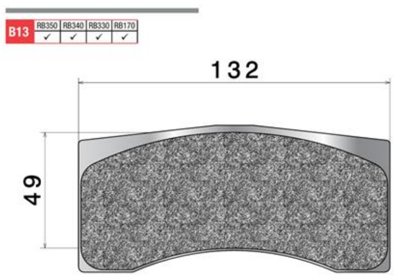
Table 23: Front Braking Components

Component	Description	Qty	Cost Per Unit	Specification
Callipers	Brembo 6 Piston Caliper – XA4.F1.01/02 	2	£4,174.05 6 PISTON CALIPER P/N XA4.F1.01/02 - Track Formula	Brembo_Racing AutoCatalogue.pdf
Brake Disc	Kit UPGRADE Brake disc GT | TY5 10C.9022A 380 x 34 mm 	2	N/A Approx - £612.00 (Based on Brembo - 380mm Curved Vein Slotted Rotors > GSM BigBrakes4u)	Brembo_10C.9022A (1).pdf
Brake Pads	RB 340 Pad Area (cm2) 78,5 Pad Thickness (mm) 29 Pad Family B24 	4	£502.59 BREMBO Racing Pad 6P GT B2429340 - 107A46934	Brembo_Racing_Brake_Pads_Catalogue_June-21 (1).pdf
Master Cylinder	Tilton 78 Series Master Cylinder 1” (25.4 mm) Bore Size 	1	£335.20 Tilton 78 Series Master Cylinders - Tilton Brake Master Cylinders - Competition Supplies	78-Series-MC-installation-dwg

Rear Brake Components

Table 24: Rear Braking Components

<table>
<tr><th>Component</th><th>Description</th><th>Qty</th><th>Cost Per Unit</th><th>Specification</th></tr>
<tr><td>Callipers</td><td>Brembo 4 Piston Caliper – XB0.L2.53/54

</td><td>2</td><td>£1,860.43

4 PISTON CALIPER P/N XB0.L2.53/ 54 - Track Formula</td><td>Brembo_Racing AutoCatalogue.pdf</td></tr>
<tr><td>Brake Disc</td><td>BRAKE DISC GT | TY5 202.8010A 355 x 32 mm
</td><td>2</td><td>N/A - £295.67 - Based on (09.B781.13 BREMBO TWO-PIECE FLOATING DISCS LINE Brake disc 355x32mm, 5, perforated/vented, two-part brake disc, Coated, High-carbon | AUTODOC price and review)</td><td>Brembo_202.8010A.pdf</td></tr>
<tr><td>Brake Pads</td><td>RB 330
Pad Area (cm2) 63
Pad Thickness (mm) 26,5 Pad Family B13

</td><td>4</td><td>£402.29

BREMBO Racing Pad 4P rear GT B1326330 - 107A469A3</td><td>Brembo_Racing_Brake_Pads_Catalogue_June-21 (1).pdf</td></tr>
<tr><td>Master Cylinder</td><td>Tilton 78 Series Master Cylinder 1” (25.4 mm) Bore Size</td><td>1</td><td>£335.20</td><td>78-Series-MC-installation-dwg</td></tr>
</table>

			[Tilton 78 Series Master Cylinders - Tilton Brake Master Cylinders - Competition Supplies]()	

Other Braking Components

Table 25: Other Braking Components

Component	Description	Qty	Cost Per Unit	Specification
Brake Fluid	P/N 04.8164.11 LCF 600 PLUS Brake Fluid - 500cc 	10	£31.04	[Brembo Racing Brake Fluid LCF 600 500ml Bottle - 04816411 | Design 911]()
Pedal Box	850-Series 2-pedal (Brake & Throttle) Underfoot Pedal Assembly with Slider System 	1	£3487.26	[850-Series 2-pedal (Brake & Throttle) Underfoot Pedal Assembly with Slider System - Tilton Engineering]()
Bias Adjuster Cable	Tilton Billet Aluminium Bias Adjuster Cable 72-408 	1	£229.34	[Tilton 72-408 Billet Aluminium Bias Ajuster Cable - Competition Supplies]()
Throttle Linkage	Tilton Throttle Linkage for 72-616 Pedal Assembly – Mount and 90-degree Penny and Giles Sensor	1	£356.30	[Tilton Throttle Linkages for Tilton Underfoot]()

				Pedal Assemblies - Tillton 600 and 800 Series Pedal Box - Competition Supplies
Brake Lines	AN-3 Stainless Steel Braided Brake Lines (PTFE) – Clear 	10	£10.85 / m	AN-3 Stainless Steel Braided Brake Lines (PTFE) - Clear - Track Formula
Reservoir	Tilton 3 Chamber Fluid Reservoir – 7/16 UNF 	1	£180.12	Tilton 3 Chamber Fluid Reservoirs - Official Tilton UK Distributor | Competitionsupplies.com
Banjo Inlet Adapter Kit	APS Banjo Inlet Adaptor kit Suit Tilton Reverse Cylinder Pedal Assemblies 	1	£46.02	Buy APS Banjo Inlet Adaptor kit Suit Tilton Reverse Cylinder Pedal Assemblies from Competition Supplies - Worldwide Shipping Available
Brake Bias Balance Monitor	Brake Bias Adjuster	1	£250.80	Buy Monit Brake Bias Adjuster from Competition Supplies - Worldwide Shipping Available

				
Handbrake	2. HGK Billet Hydraulic Handbrake with Tilton Master Cylinder ¾" with Base Plate for Mounting 	1	£650.94	HGK Handbrake With Tilton Master Cylinder – HGK Shop

Component Justification

Brake Discs:
Our design chose the Brembo 2-piece floating discs with a TYP 5 slotted pattern. This type of brake disc offers great initial bite and consistent braking performance by allowing efficient evacuation of gases and debris from the pad-disc interface. The discs are vented internally with curved vanes, which enhance airflow and cooling efficiency, reducing the risk of thermal cracking and fade during long high-temperature operation. The 2-piece Brembo brake discs are made up of high-carbon cast iron rotor with precision milled billet aluminium Bell to reduce unsprung and rotating weight, which benefits handling, acceleration, and vehicle dynamics (Racer Products, 2025).

Brake Calipers and Pads:
Brembo calipers (6-piston front, 4-piston rear) are used for their lightweight forged aluminium construction, providing high stiffness and uniform pad pressure distribution.
Front RB 340 and rear RB 330 pads are selected for their high frictional value ($\mu \approx 0.52$–0.42), low wear rate (2.3 μm/stop) and excellent thermal stability (Bremboparts.com, 2025)

Brake Hoses:
AN-3 stainless steel braided PTFE hoses are used to minimise volumetric expansion upon pressure, giving enhanced pedal feel and brake efficiency under harsh temperatures above 260°C. (Trackformula.co.uk, 2017)

Pedal Box and Master Cylinders:
The Tilton 850-Series underfloor pedal box with 78-Series master cylinders (25.4 mm bore

front and rear) includes an adjustable balance bar (pedal ratios 4.8:1 -6.1:1), allowing dynamic front-rear bias adjustments to suit a variety of terrain conditions. (Tilton Engineering, 2025)

Sustainability and Serviceability

Our design has incorporated a 2-piece floating discs to allow individual replacement of the cast iron rotor rings, enhancing component life and reducing waste. High carbon content cast iron is recyclable to the degree, rendering it sustainable in engineering use. The modular design of the Tilton pedal box also enhances maintenance access and part replacement, allowing for extensive rally vehicle operations.

Brake Line and Fitting Specification

Rear and top outlet ports on Tilton 76-Series master cylinders are compatible with AN-3 fittings. AN-3 braided stainless steel brake lines were selected due to their compatibility with both Brembo calipers and Tilton master cylinders. Brembo calipers typically require M10x1.0 or M10x1.25 banjo fittings and are secured with crush washers for leak-free connections.

The brake lines used will be correctly sized to permit suspension travel and steering angles without tension. Routing and clamping are set up carefully to prevent abrasion and fatigue. The system will be well bled after assembly to remove air and ensure optimum braking performance.

Front Brake Disc/Pad FEA Simulations and Analysis

For the FEA Simulations, the front brake disc was modelled on Fusion 360 as shown below. Simplifications were made to help reduce computational costs and decrease simulation times. A balance of accuracy and efficiency was done through the following simplifications:

- Removing the brake pad backing plates
- Removing the internal bell and bobbins
- Sharp corners on vanes without fillets or chamfers

Figure 14 shows the original brake disks model with Figure 15 being the simplified model used for FEA.

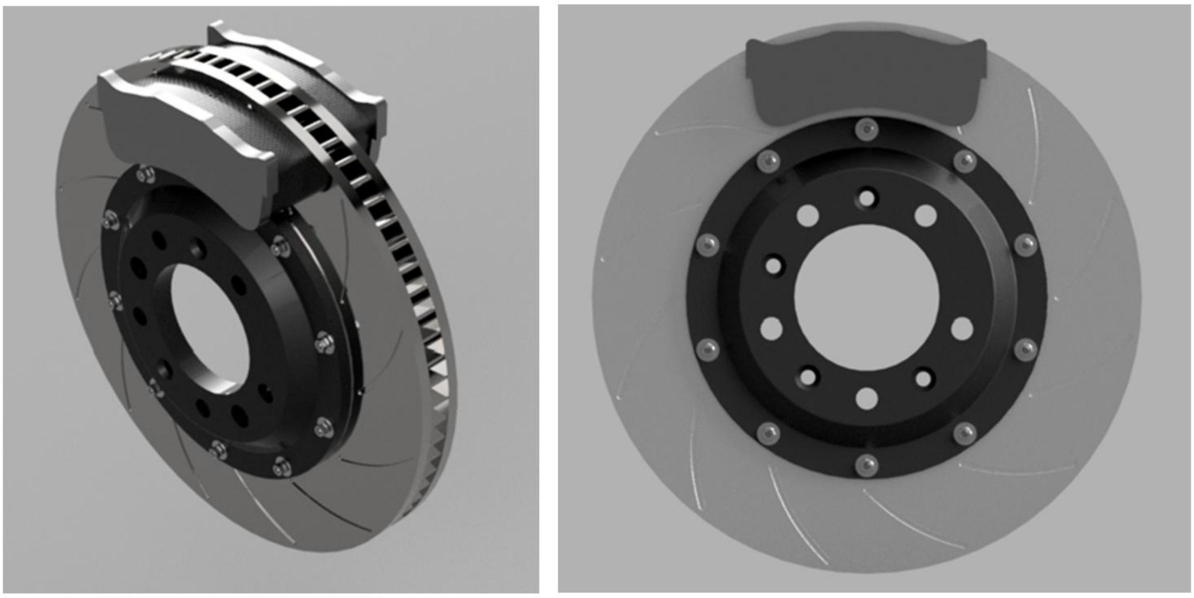


Figure 14: Original Brake Disks Model

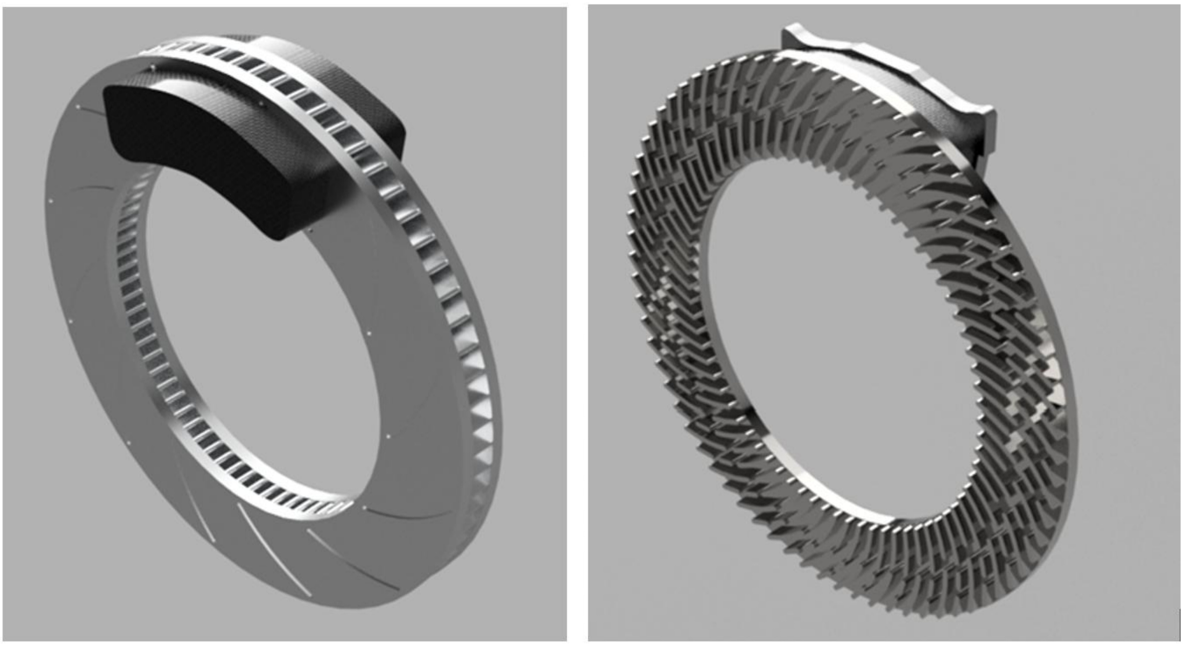


Figure 15: FEA Simplified Brake Disks Model

FEA Setup

A coupled thermal-structural analysis was performed using the following setup (Table 26):

Table 26: FEA Brake Disks Material Setup

Material	Min Yield Strength (MPa)	Min Tensile Strength (MPa)	Density (kg/m^3)	Young's Modulus, E (MPa)
Ductile Cast Iron Grade 60-40-18	329	461	7150	169,000

- **Contacts**: Frictional (μ = 0.52) with small sliding enabled
- **Loads**:
    - **Maximum Braking**: 16169.9 N clamping force, 600°C at pad tracks, rotational velocity 131 rad/s (anticlockwise)
    - **Average Dakar Braking**: 8000 N clamping force, 300°C at pad tracks, rotational velocity 70 rad/s (anticlockwise)
- **Convection**: 100 W/m²K cooling applied to cooling vanes
- **Boundary Conditions**: Remote displacement was applied at the rotor's mounting face to realistically simulate hub constraints, allowing rotation about the disc axis while preventing translational motion to avoid rigid body errors.

FEA - Max Clamping Force + Max Speed + High Temperature

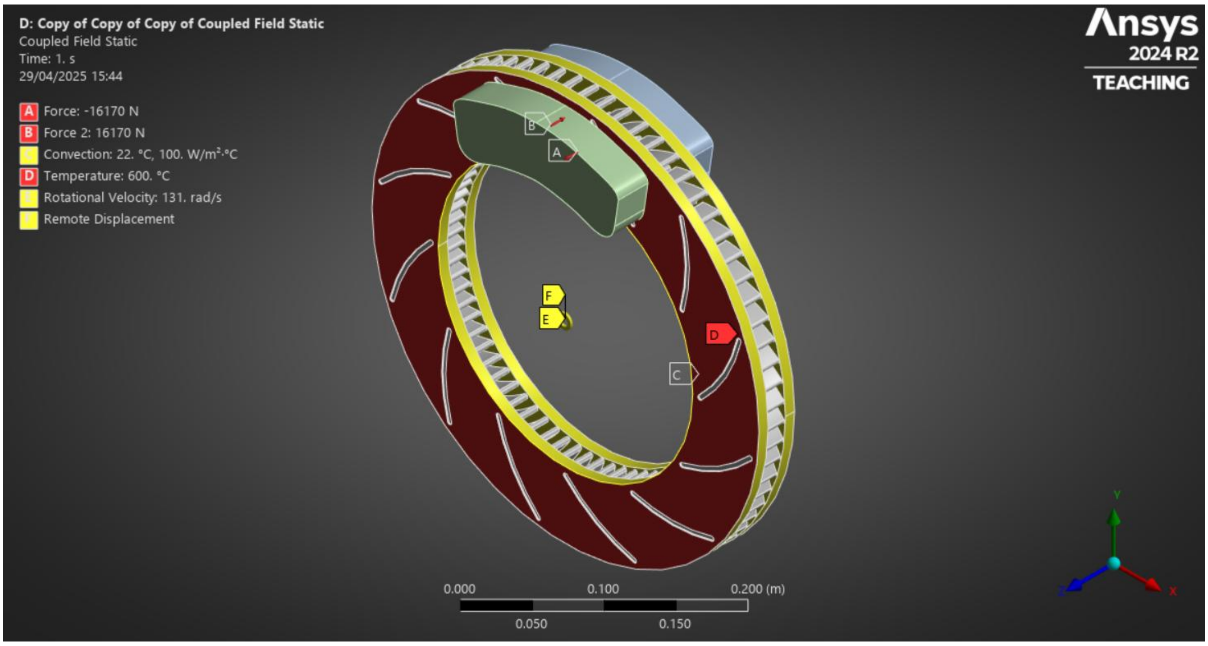


Figure 16: Max Clamping Force + Max Speed + High Temperature Setup

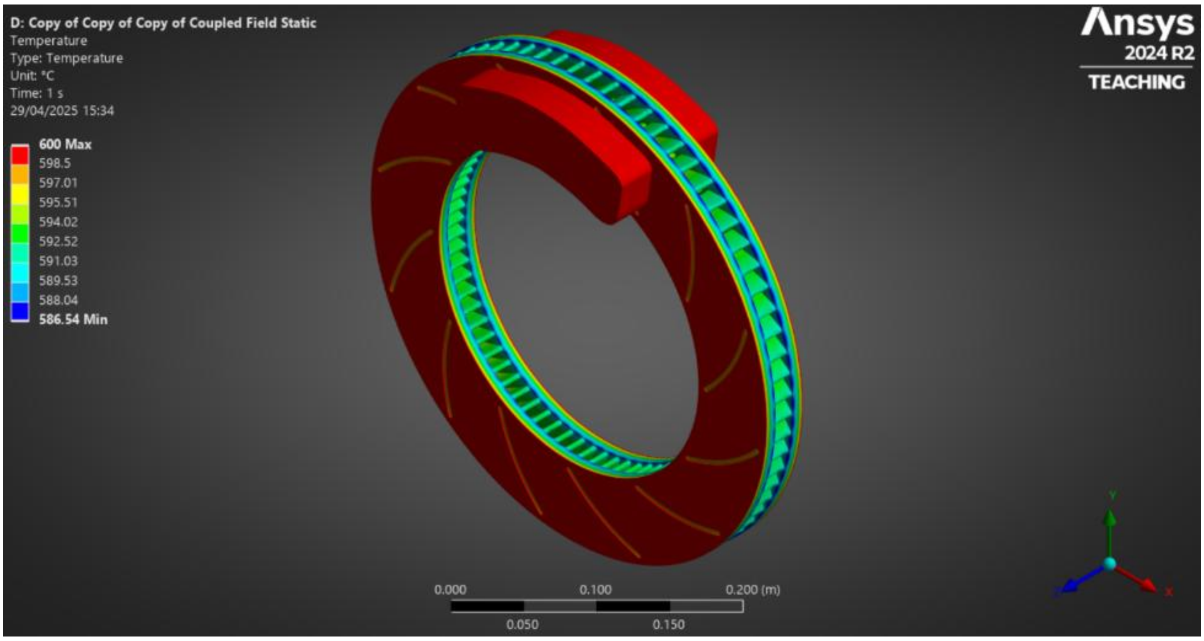


Figure 17: Max Clamping Force + Max Speed + High Temperature Results

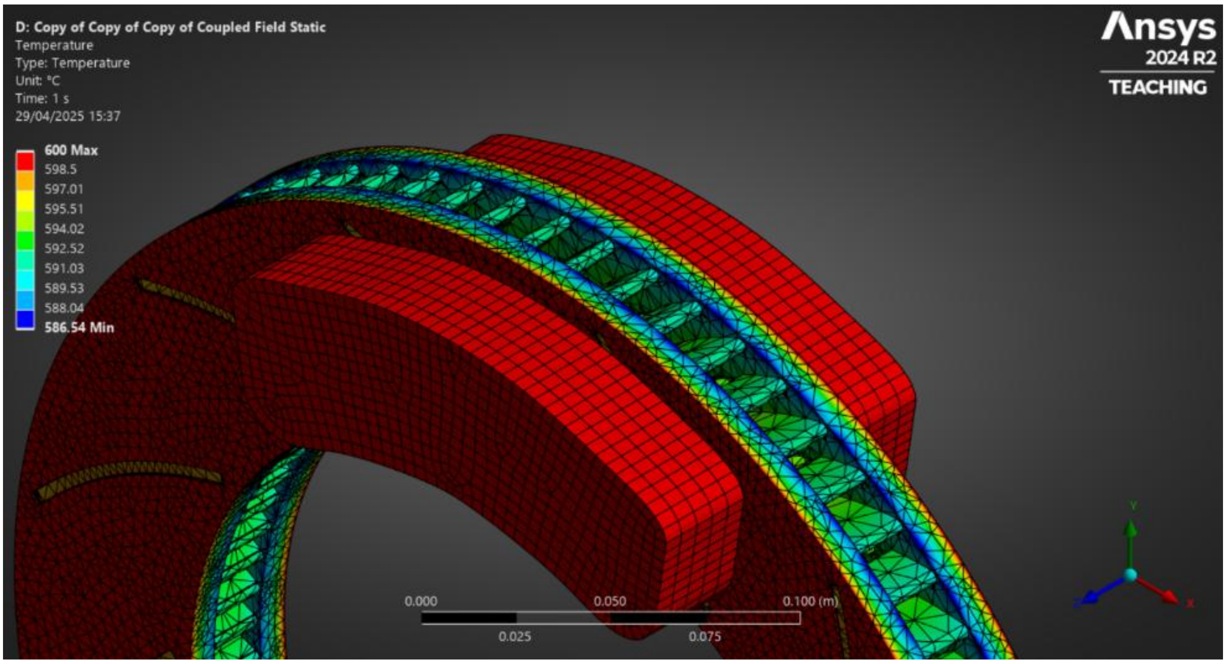


Figure 18: Max Clamping Force + Max Speed + High Temperature Mesh

Temperature Distribution: The brake disc-pad interface reached a peak temperature of 600 °C. The temperature was lower near the inner vanes with a minimum temperature of 586.5 °C. This confirmed effective thermal conduction and justified the selection of Brembo's internally vented Type 5 slotted rotors.

Deformation

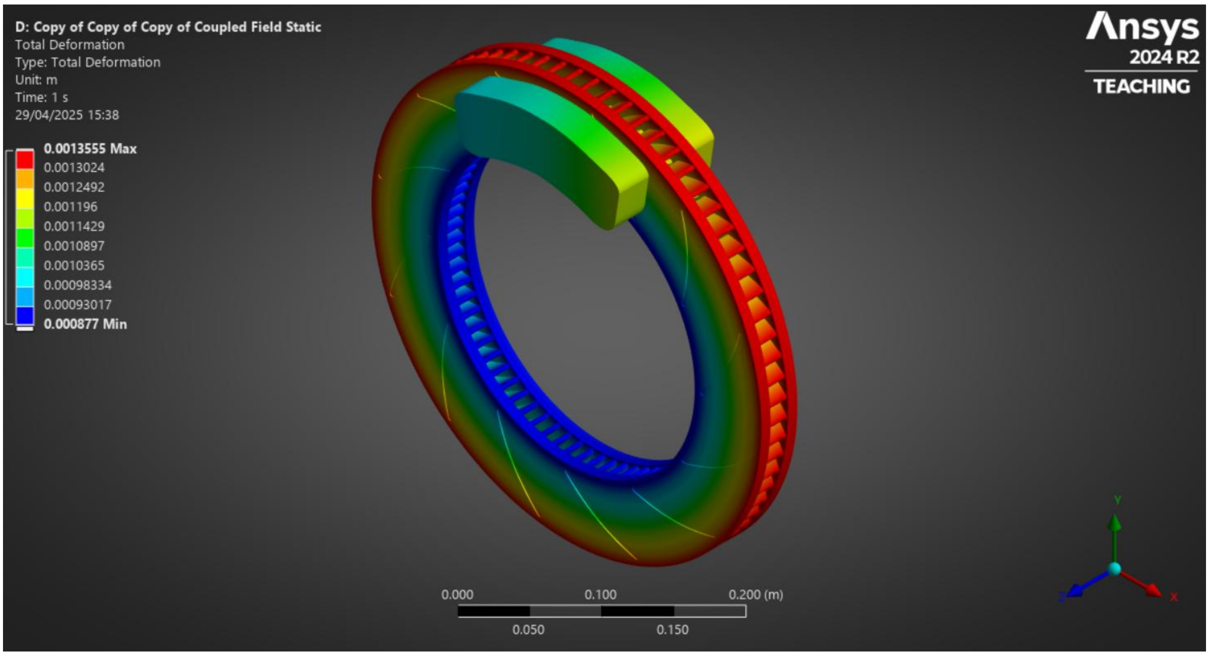


Figure 19: Max Clamping Force + Max Speed + High Temperature 3/4 Deformation

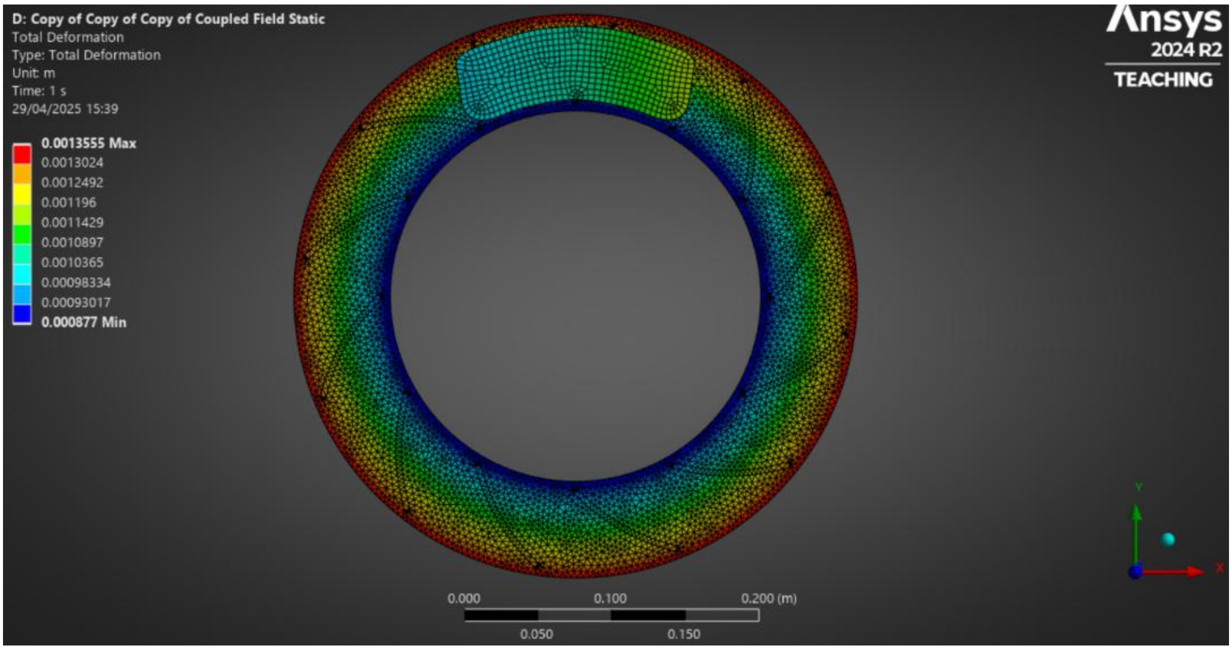


Figure 20: Max Clamping Force + Max Speed + High Temperature Side View Deformation

Deformation: The highest total deformation was 1.36 mm, primarily on the disc outer edge. This shows that the high-carbon cast iron disc resists significant warping under thermal and structural loads.

Von Mises Stress

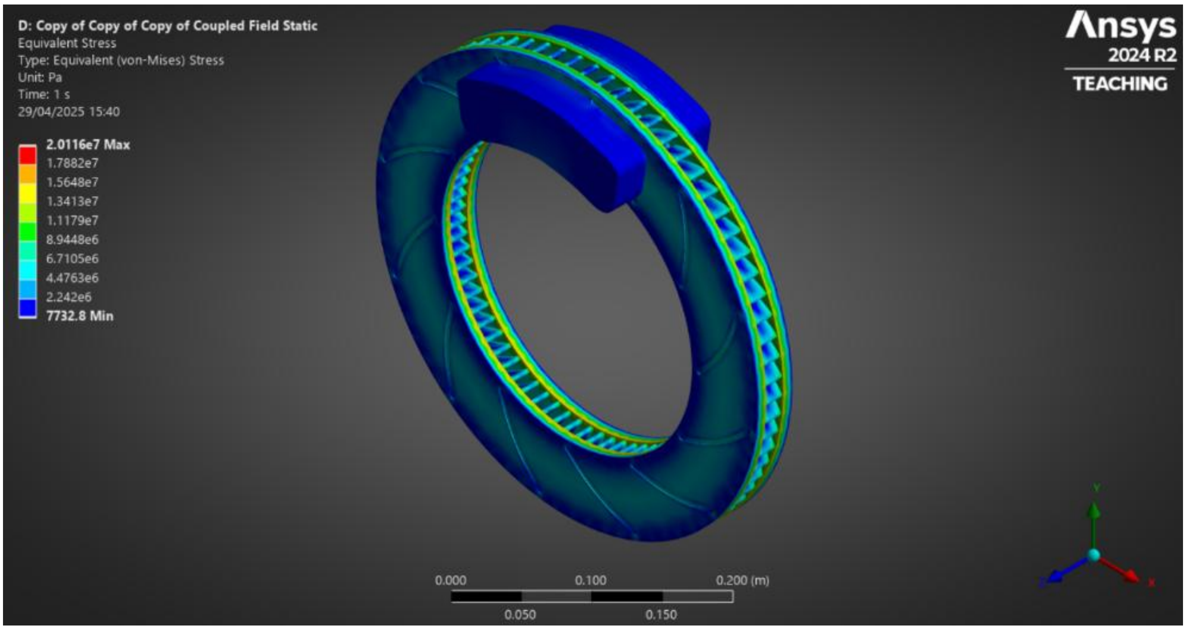


Figure 21: Max Clamping Force + Max Speed + High Temperature 3/4 View Von Mises

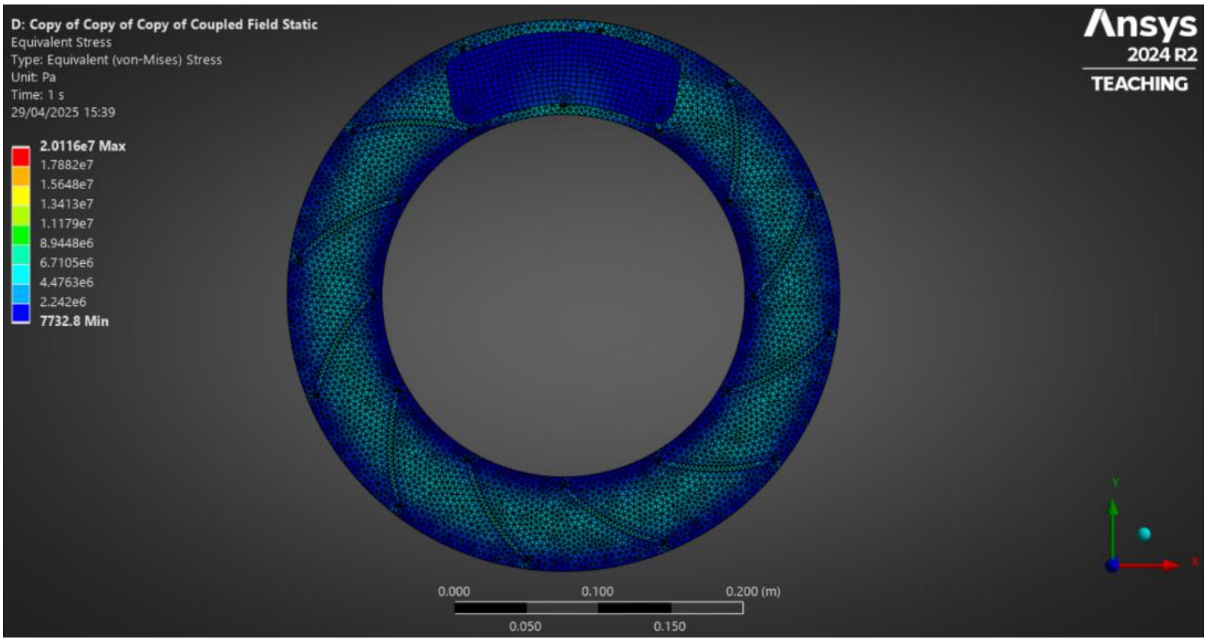


Figure 22: Max Clamping Force + Max Speed + High Temperature Von Mises Stress

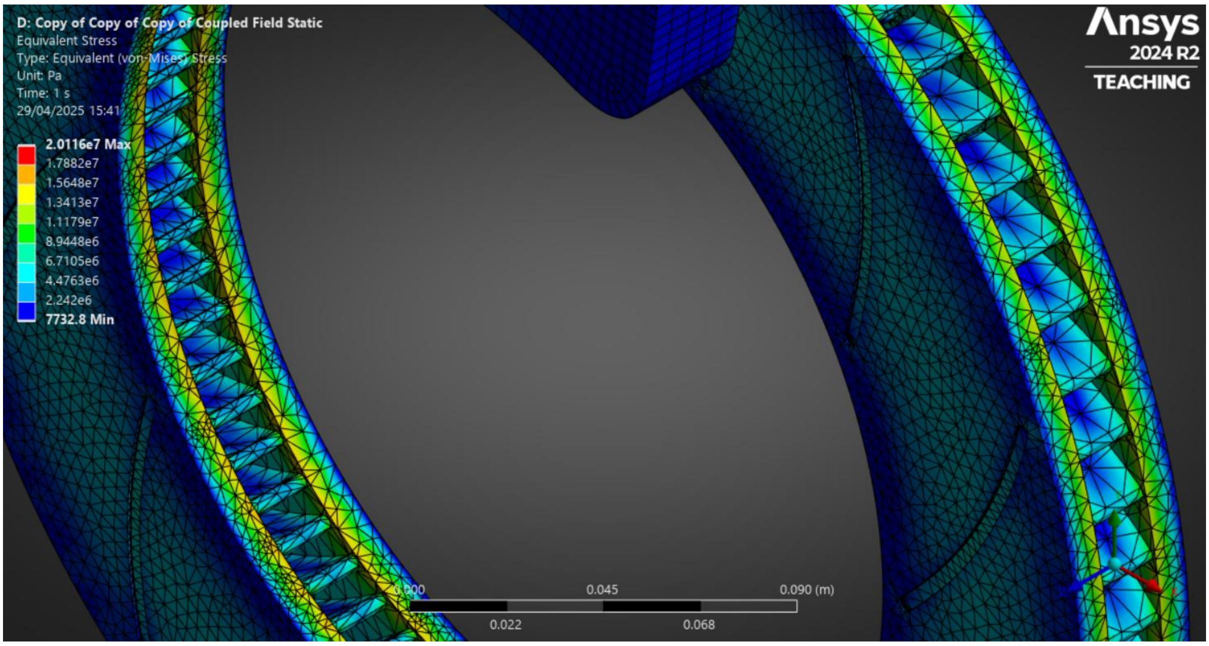


Figure 23: Max Clamping Force + Max Speed + High Temperature Mesh Von Mises

Stress (von Mises): The maximum equivalent stress was around 20.1 MPa, near the inner cooling vanes. This is well below the ultimate tensile strength of high-carbon cast iron (400–500 MPa).

FEA - Average Clamping Force + Average Speed

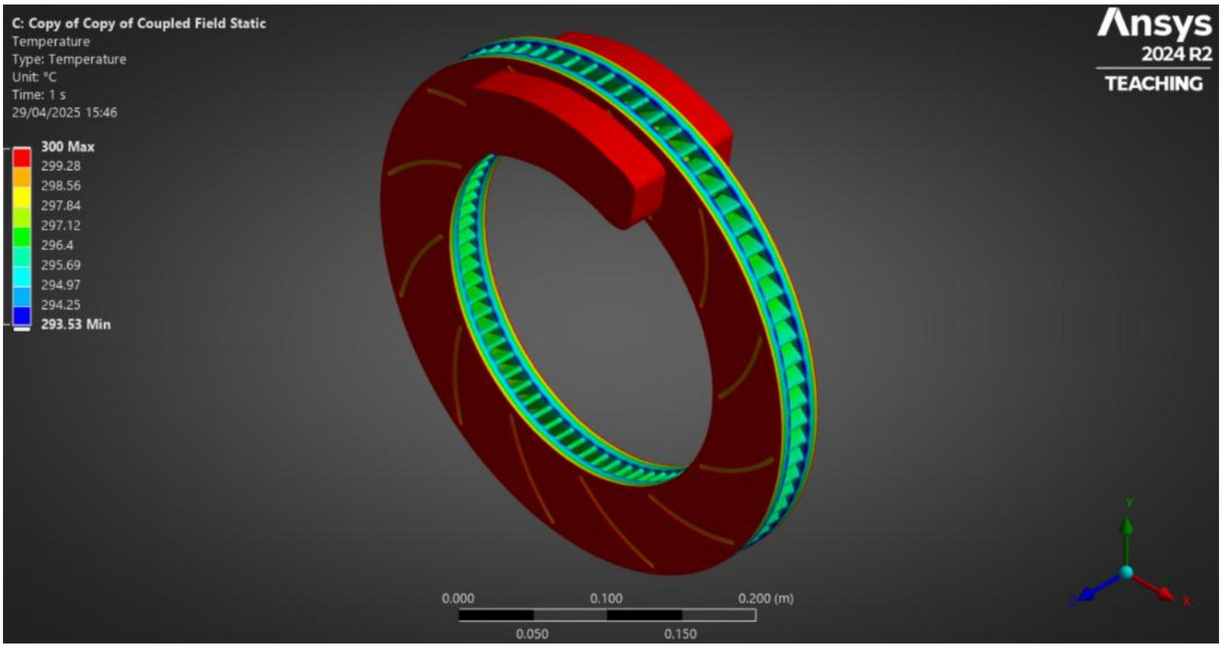


Figure 24: Average Clamping Force + Average Speed Temperature

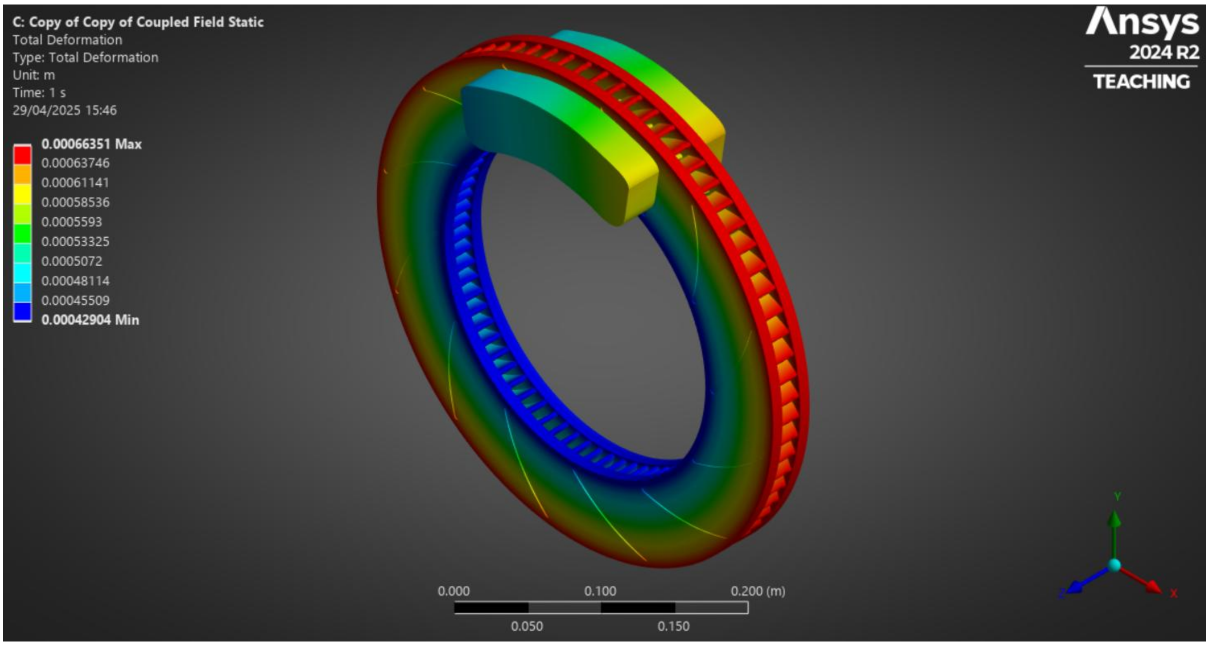


Figure 25: Average Clamping Force + Average Speed Deformation

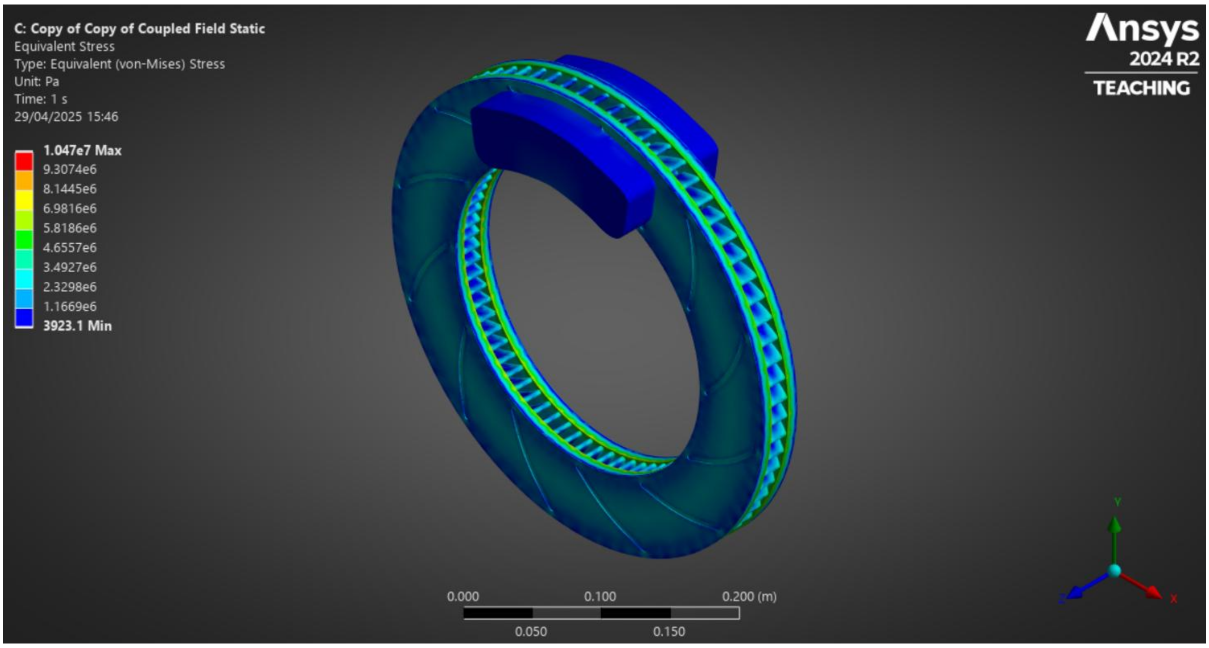


Figure 26: Average Clamping Force + Average Speed Von Mises

Coupled structural-thermal simulations were performed under two scenarios (Table 27):

Table 27: Braking Disks FEA results

Scenario	Temp (°C)	Max Deformation (mm)	Max Von Mises Stress (MPa)
Maximum Braking (56.67 m/s)	600	~1.35	~20.11
Average Braking (30.28 m/s)	300	~0.66	~10.47

Discussion

Stress concentrations were observed near vane roots, which could be due to CAD simplifications (e.g. absence of fillets and chamfers). In a detailed design with smoother transitions, lower peak stresses would be expected. The structural analysis remained within safe limits of cast iron, with minimal thermal distortion and stress levels well below failure thresholds. This proves the disc's suitability for Dakar endurance use. Under worst-case braking, the max Von Mises Stress is 20.11 MPa, which is well below the yield strength (329 MPa) and ultimate tensile strength (461 MPa) of high-carbon cast iron, confirming a high safety factor (>20) under worst-case braking.

Regenerative Braking

Regenerative braking was excluded due to Dakar-specific demands. The rally involves extended coasting, throttle modulation, and terrain-induced deceleration, offering limited opportunity for energy recovery. Incorporating regenerative braking in the hydraulic system would introduce complexity, packaging difficulties, and high-temperature, dust, and vibration failure. Instead, we decided on a mechanical system optimised for cooling performance, lightweight components, and reliable braking. Energy regeneration is still achieved through the regenerative suspension system, which has a consistent output without compromising on braking performance. This option aligns with our focus on simplicity, maintainability, and reliability.

Tyres & Wheels

Figure 27: BFGoodrich All-Terrain T/A KDR2+

Table 28: BF Goodrich All-Terrain T/A KDR2+

Product	*BFGoodrich All-Terrain T/A KDR2+*		
Description	This tire is designed to deliver enhanced grip, improved steering response, and superior traction, particularly on muddy terrains, without compromising overall performance. It retains optimal efficiency on hard-packed or sandy terrains, climbing dunes, or navigating aggressive, rocky trails in extreme temperatures		
Dimensions	37x12.50 R17		
Compound	Medium		
Pressure Ranges	Sand Dunes	Gravel / Dirt	Rocks
	17 – 19 psi	29 – 31 psi	27 – 30 psi
Wheel Nut Torque	110–120 Nm		
Weight	< 36 kg		
Estimated Cost	$652 - $704 USD per tyre		
Source	Desert Rat - https://www.desertrat.com/series-222299-bf-goodrich-racing-all-terrain-t-a-kdr2.html		

The selection of the BFGoodrich All-Terrain T/A KDR2+ tyres aligns with the PDS requirements for traction, durability, and weight under extreme off-road conditions. The 37-inch diameter improves ground clearance, while the 12.5-inch width increases surface contact, enhancing grip and stability on soft and rocky terrain. Compatibility with 17-inch rims ensures clearance for 380 mm brake discs and 6-piston callipers.

The medium compound provides a balance between grip and longevity, as required for mixed Dakar stages. Pressure ranges from 17–31 psi enables terrain-specific tuning without compromising bead retention or safety. The open, self-cleaning tread blocks support traction in muddy or loose surfaces, and reinforced sidewalls enhance durability across stages of 500–800 km.

A weight of under 36 kg keeps the tyre within the 60 kg per corner PDS limit when paired with a suitable wheel. The recommended wheel nut torque of 110–120 Nm matches the Ridex 5-nut hub for secure high-load operation.

Figure 28: Black Rock Cage Beadlock Dark Bronze Wheel

Table 29: Black Rock Cage Beadlock Dark Bronze Wheel Specification

Product	Black Rock Cage Beadlock Dark Bronze Wheel
Description	Built off-road tough and engineered for heavy-duty use, the Black Rock Cage Beadlock features a functional bead lock ring designed to maintain tyre stability at low pressures, while delivering impressive strength and reduced weight for extreme off-road racing conditions
Dimensions	17x9
Bolt Pattern	5x114.3
Offset	+12 mm
Load rating	2,700 – 3,200 lbs per wheel
Bolts Torque requirements	27 – 34 Nm
Weight	< 18 kg
Estimated cost	$499 per wheel
Source	CNC Wheels - https://cncwheels.com.au/shop/4x4-rims/black-rock-cage-beadlock-dark-bronze-17x9-5x114-3-wheel/

The Black Rock Cage Beadlock wheels were selected to meet Dakar Rally demands for strength, reliability, and safety. The 17x9 size and 5x114.3 bolt pattern ensure compatibility with the Ridex hub, while the +12 mm offset balances track width and wheel arch clearance for optimal steering stability. Weighing under 18 kg, they help keep total unsprung mass within the 60 kg per corner limit when paired with the selected tyres.

The bead lock ring secures the tyre bead at low pressures (17–19 psi), maintaining traction on sand without risk of de-beading. With a load rating of up to 3,200 lbs, they withstand repeated off-road impacts. Their lightweight construction and reinforced design support durability and performance across sand, rock, and gravel stages, making them a robust choice for long, high-impact rally conditions.

Vehicle Dynamics

Mass Distribution & Weight Balance

The estimated total vehicle weight is 2,182 kg, with the battery being 1,200 kg (50% of total mass). Unlike internal combustion engines, which have engine mass concentrated towards the front of real, battery placement in an electric rally truck plays a crucial role in optimising handing, stability and centre of gravity.

The battery should be positioned centrally and as low as possible. This lowers the centre of mass, therefore reducing body roll and handing is improved at high speeds.

Table 30 shows the weight of the essential components of the vehicle and how much of the vehicle's weight they accumulate to.

Table 30: Mass Breakdown

Component	Weight (kg)	Percentage	Notes
Battery Pack	1,200	55	~50% of total vehicle mass
Chassis & Frame	480	22	Roll cage, crash structure, battery protection
Steering System	12	0.5	Lightweight rally-specific steering
Braking System	40	1.8	High-performance rally braking system
Suspension System	130	6	Front & rear double wishbone setup, dampers, control arms
Wheels & Tyres	210	9.6	4 main wheels + 2 spare
Bodywork	50	2.3	Lightweight composite carbon fibre/fibreglass
Electrical Systems	10	0.5	Motor controllers, battery management, telemetry systems
Miscellaneous Components	50	2.3	Driver seat, dashboard, reinforcement structures
Total Estimated Mass	2,182	100	

The target front-to-rear weight would be 50:50. As very few electric rally vehicles have been made, we can reference this rally pick-up to a mid-engine car. 'A mid-mounted engine distributes the weight across the car's four tires, allowing them to grip the road at their maximum potential during cornering, braking, and accelerating' (Ewing, S. (2022)). Positioning the battery forwards or backward to fine tune the balance will allow for the easiest weight management strategy.

Centre of Mass

The centre of mass (CoM) is one of the most factor affecting handling. Since the battery makes up 49%, positioning it low will greatly impact how the vehicle handles. It should be placed 500 mm from the ground. This will be lower than the stock off-road Hilux, which is 600 mm.

Cornering & Stability

In off-road, stability over rough terrain is more important than cornering speed. The handing should be predictable, and the body should not roll a significant amount when going round corners.

A lower vehicle can handle corners a lot better than a tall one. If you can't lower the height of the vehicle, you can increase the width. Having a wide stance allows better stability. In this case, we want the track width to be 1,750 mm. Aerodynamic stability can also improve cornering, which can be done through using air ducts to allow for optimal air flow and reduced drag.

Suspension & Ride Comfort

Camber Angle (Tilt of wheels. Negative chamber enhances cornering grip): -3° to -1.5°

Caster Angle (Steering axis tilt. Positive caster improves stability): 6° to 8°

Toe Angle: Toe in (0.5° to 1°)

Wider Engineering Applications

Sustainability

Using lightweight materials will help reduce the overall environmental impact, which links to sustainable goal 12, by improving efficiency in other parts of the car. Regenerative braking is also a factor that could contribute to this improved energy efficiency by recovering energy when the vehicle is decelerating.

On the flip side, rally tires can produce microplastics that can accumulate in soil and water, that can affect the wildlife especially in remote areas, linking to sustainable goal 15. Furthermore, rally components often need replacing due to wear and tear, further contributing to the carbon footprint. However, our goal is to reduce this as much as possible by increasing the durability of these parts.

Commercial

From a commercial standpoint, advanced systems can add significant value to the vehicle. These improvements can be marketed as high-performance upgrade kits, generating interest from aftermarket suppliers or OEM partners. Better performance also enhances the appeal to both sponsors and consumers. As this is not a mass production vehicle, the market is niche.

Ethical

Ethical responsibility is shown by prioritising driver safety through responsive braking, stable steering, adjustability, and durable systems. A modular approach is adopted to allow for easy repair and reuse supporting resource conscious engineering.

Legal

These systems must comply with FIA and Dakar Rally organisers regulations. Making sure that the components meet the safety and security requirements, and do not break any legal rules. There is a risk of infringement with existing patents if proprietary technologies are used without proper research, which could introduce issues further on in the design and manufacturing process, wasting essential research and designing time.

Human Factors

This whole design prioritises driver comfort, safety, and longevity, as well as the overall goal of this project. A well-balanced system reduces fatigue by ensuring ergonomic seating and steering, and responsive and reliable braking and steering. A setup that only prioritises racing, especially with endurance races like these, will affect the driver and could cause safety problems that will not lead us to secure future sponsors and could ruin the reputation of the brand.

Risk Assessment

Table 31: Risk Assessment

Risk	Effect	Severity	Frequency	Risk Level (Severity × Frequency)	Current Risk Management
Incorrect suspension design	Poor ride quality, reduced handling stability, potential vehicle failure	4	3	12 (Medium-High)	Conduct FEA simulations and testing before finalising the design. Consult with experts on geometry.
Steering system misalignment	Unsafe steering response, loss of vehicle control	5	2	10 (Medium)	Use precise tolerances in design. Simulate steering forces and verify calculations.
Brake system failure/miscalculation	Ineffective braking, increased stopping distance, potential crashes	5	2	10 (Medium)	Validate brake system calculations. Use fail-safe designs and redundancy measures.
Uneven weight distribution	Poor handling, increased tire wear, potential rollover risk	4	3	12 (Medium-High)	Optimise weight distribution using simulations and real-world benchmarks.
Tire selection errors (wrong compound/tread design)	Poor grip, increased wear, reduced performance in off-road conditions	3	4	12 (Medium-High)	Research and compare rally tire data. Select a tested tire compound for terrain conditions.
Simulation errors in vehicle dynamics models	Misleading results leading to poor design choices	3	3	9 (Medium)	Cross-verify results using multiple software tools (e.g., MATLAB, Simulink,

					ADAMS). Peer review results.
Component failure due to excessive loads	Suspension, steering, or brake failure leading to safety hazards	5	2	10 (Medium)	Apply safety factors in calculations. Test materials under expected load conditions.
Communication failure between teams	Mismatched components, inefficiencies in integration	3	4	12 (Medium-High)	Regular team meetings, shared documentation, and defined interfaces between subsystems.
Manufacturing/assembly errors	Poor fitment, misalignments, compromised performance	4	3	12 (Medium-High)	Create detailed CAD models and ensure accurate tolerances before manufacturing.
Testing safety hazards (vehicle testing environment risks)	Injury risk, property damage, unexpected failures	5	2	10 (Medium)	Follow strict testing procedures, wear PPE, and conduct low-risk simulations before real-world tests.

Business Plan

Executive Summary

Our business is a specialised motorsport engineering consultancy focused on electrifying and upgrading vehicles for extreme off-road endurance events. While the wider team prepares a fully electric Toyota Hilux for the 2028 Dakar Rally, our subgroup concentrates on suspension, steering, braking, tyres, and vehicle dynamics. Toyota has tasked us with adapting the vehicle for their new solid-state battery, showcasing its durability and performance. As an internal R&D unit, we're designing a subsystem to handle extreme terrain and vertical loads while regenerating energy, highlighting electric propulsion's sustainability. We aim to patent key technologies, like our regenerative wishbone suspension, and license them to OEMs, motorsport teams, and Tier 1 suppliers, creating scalable IP-based revenue. Our division operates semi-independently, enabling direct supplier and partner relationships.

Market Opportunity

The electric motorsport industry has seen growth with Formula E and Extreme E, which has driven public and corporate interest in clean mobility. According to the International Energy Agency (2023), 1 in 5 cars sold globally were electric, with the amount rising each year, as shown in Figure 12.

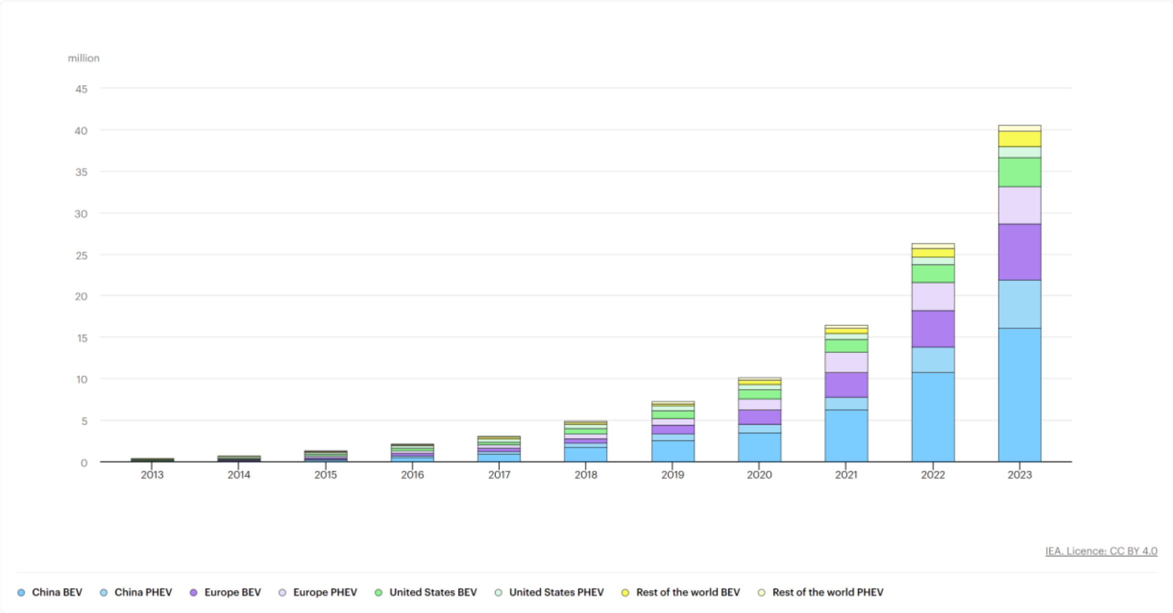


Figure 29: Electric Car Sales 2013-2023

However, this growth is concentrated with commercial vehicles, with limited representation of off-road performance markets. Our group's focus addresses a gap in the market, electrified off-road performance vehicles with suitable kinematics adaptable for both OEMs and aftermarket applications.

Whilst our immediate collaboration is with Toyota, future clients may include:

- OEM R&D teams
- Privateer rally teams
- Electric motorsport divisions
- Low volume manufacturers

Capital Investment and Fabrication Capabilities

Investing in our own fabrication and machining capabilities eliminates outsourcing lead times and enables rapid prototyping for future clients. The one-off nature of this prototype justifies a small-scale, precision-focused workshop setup.

Key Capital Investments:

CNC Milling Machine (3-axis) – Essential for machining precise mounting interfaces.

CNC Lathe – Required for knuckle rough turning and profile generation.

TIG Welding Rig – High-control welding for fatigue-critical joints.

Laser Cutter (MDF Jigs) – For quick fabrication of alignment fixtures.

Inspection Tools – Digital calipers, bore gauges for dimensional checks.

These machines support a lean manufacturing model suitable for low-volume, high-performance suspension systems. Future work could include expanding this capability to short-run motorsport part production.

Costs

The full costs to manufacture the suspension, steering, braking and tyres is shown in Table 32.

Table 32: Total Costs

Component	Supplier	Quantity	Cost (£)
AISI 4130 Steel Tube (wishbones)	Custom	4	80
AISI 4340 Steel Billet (knuckle)	Custom	50	100
Spring and Damper	K-Tech and WP	6	7,500
Braking	Brembo and Tilton	4	23,800
Steering	Howe Performance	1	5,200
Tyres	CNC and Desert Rat	5	6,000
Manufacturing for Suspension			255
Labour			3,200
Total			46,135

Investment

To bring the electric Dakar Rally project to life, an estimated budget of £2–3 million will be raised through a combination of original equipment manufacturer (OEM) partnerships, private motorsport teams, competitive grant funding, and cutting-edge technology collaborations. This funding will cover vehicle design and engineering, advanced component integration, off-road endurance testing, and complete participation in the Dakar Rally.

Table 33: Source of Investment

Source	Example Entities	Rationale
EV OEMs	BYD, Geely Auto Group, XPeng	Provide funding, technical components, and branding in exchange for global exposure and technology validation in extreme racing environments.
Private Race Teams	AF Corse, UK-based endurance teams	Offer operational support and racing expertise, potentially contributing funding and logistics in exchange for competitive entries and brand presence.
Motorsport Grants	Motorsport UK Development Fund	Public sector grants supporting innovation, sustainability, and UK motorsport leadership, with up to £1 million available to successful applicants.
Technology Partnerships	Koenigsegg (David Inverter), Rimac, Williams Advanced Engineering	Advanced EV systems supplied in-kind or at reduced cost to promote

		their capabilities under Dakar conditions.
Battery Research Bodies	Faraday Institution	Public-private research backing for battery R&D, enabling enhanced performance, safety, and endurance solutions.

The OEMs targeted each align with the project's objectives. BYD, a global leader in EVs, is expanding into Western markets and would benefit from Dakar visibility. Geely is evolving its EV offerings through brands like Polestar and Zeekr and would gain from showcasing endurance capabilities. XPeng, focused on high-performance EVs and growing its European footprint, shares the project’s innovative ethos.

We’ll partner with UK race teams like AF Corse for expertise and credibility and apply for Motorsport UK grants of up to £1M. Tech partners such as Koenigsegg may supply components like the ‘David’ inverter in exchange for validation and exposure. Support from the Faraday Institution could aid battery development. Initial outreach will target OEMs, race teams, and grants, with full development starting once funding is secured—aiming to field the first all-electric vehicle to finish the Dakar Rally.

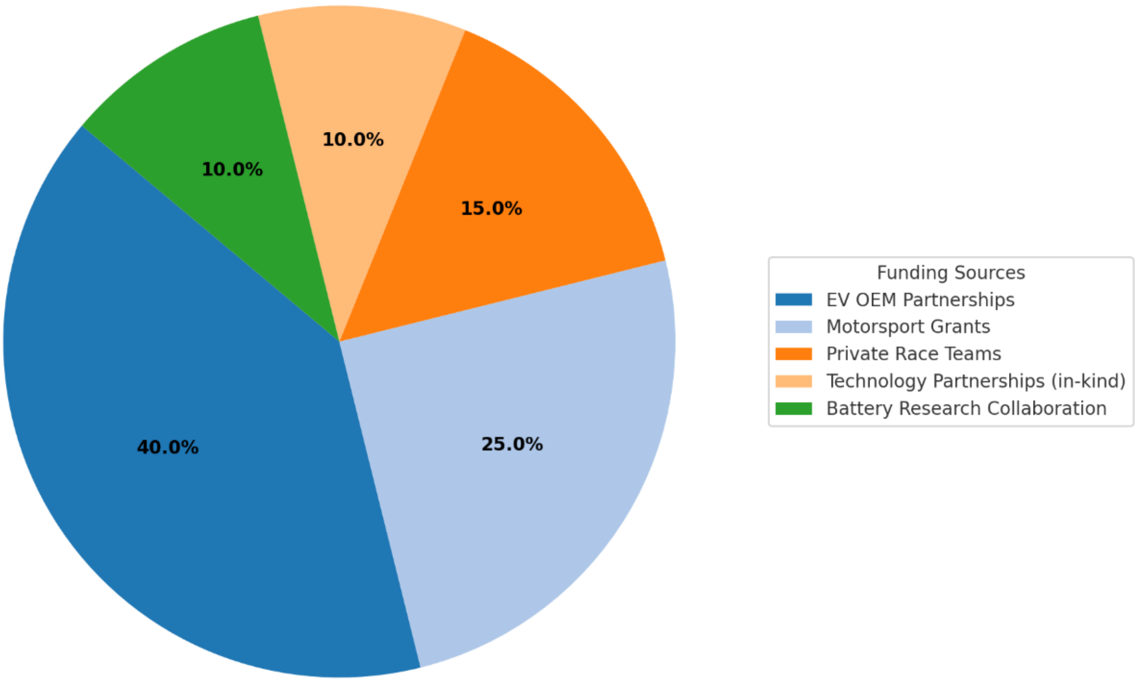


Figure 30: Summary of investments in a pie chart

Future Growth Plan

Table 34 shows the plan we intend to follow until the final reveal of the finished vehicle.

Assuming:

OEM Deals: Starts at £50k-£70k/year with increase as credibility builds.

IP Licensing: ~£10k-£100k/year from 2027, depending on IP strength

Kit Sales: Ramp from 10 kits to 100+ by 2024

Consulting & Services: Add on contracts for training, integration, and engineering (+10% revenue from 2028)

Table 34: Future Plan with Profit and Losses

Year	Milestone	Development & Labour (£)	Materials (£)	R&D Services (£)	Sales Revenue (£)	OEM Revenue (£)	IP & Services (£)	Net Profit/ Loss (£)
2025	Setup prototyping lab, develop prototypes	20,000	26,135	5,000	0	0	0	-51,135
2026	Supply first full system for Hilux prototype	25,000	30,000	5,000	10,000	100,000	0	+50,000
2027	Offer low volume sales of custom parts	20,000	15,000	5,000	50,000	150,000	10,000	+170,000
2028	Partner with OEMs to co-develop systems for EV trucks.	30,000	20,000	10,000	100,000	150,000	25,000	+215,000
2035	Full Commercialisation + Licensing	40,000	30,000	25,000	200,000	250,000	75,000	+430,000
2040	Global Marketing & System Consulting	60,000	40,000	50,000	500,000	300,000	100,000	+750,000

Team Integration

The project consisted of a lot of parts which various groups had to contribute to. This meant working with other groups to ensure that the designs were coherent. The figure below shows how the information was shared within the group.

A WhatsApp community was also set for the whole team. Within that community, separate chats were made for each group and one for just the team leaders in order to set any group leader meetings.

Group leader meetings were set every Monday to catch up on anything and also convey any problems that may be happening or set any deadlines.

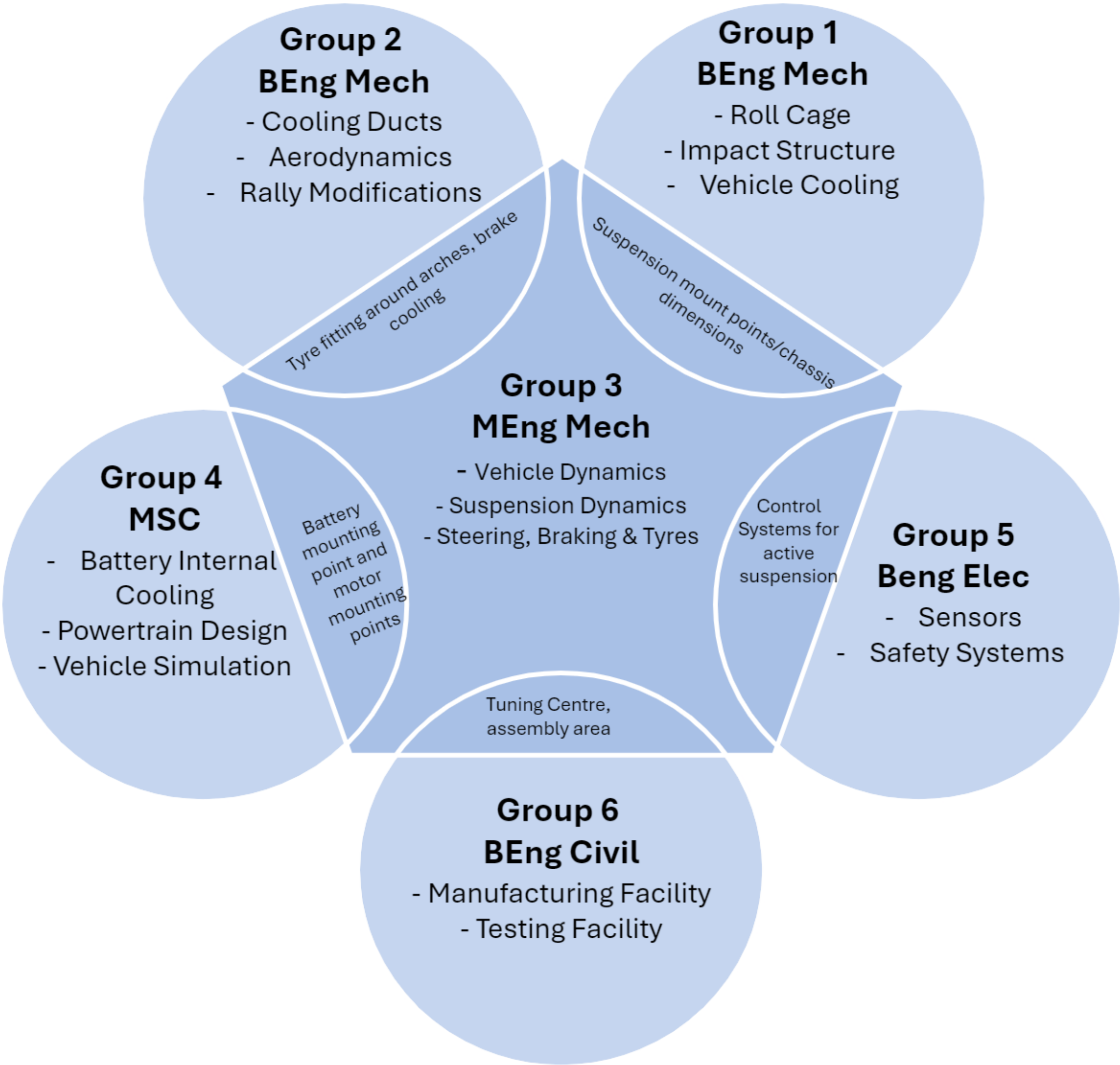


Figure 31: Team Integration Chart

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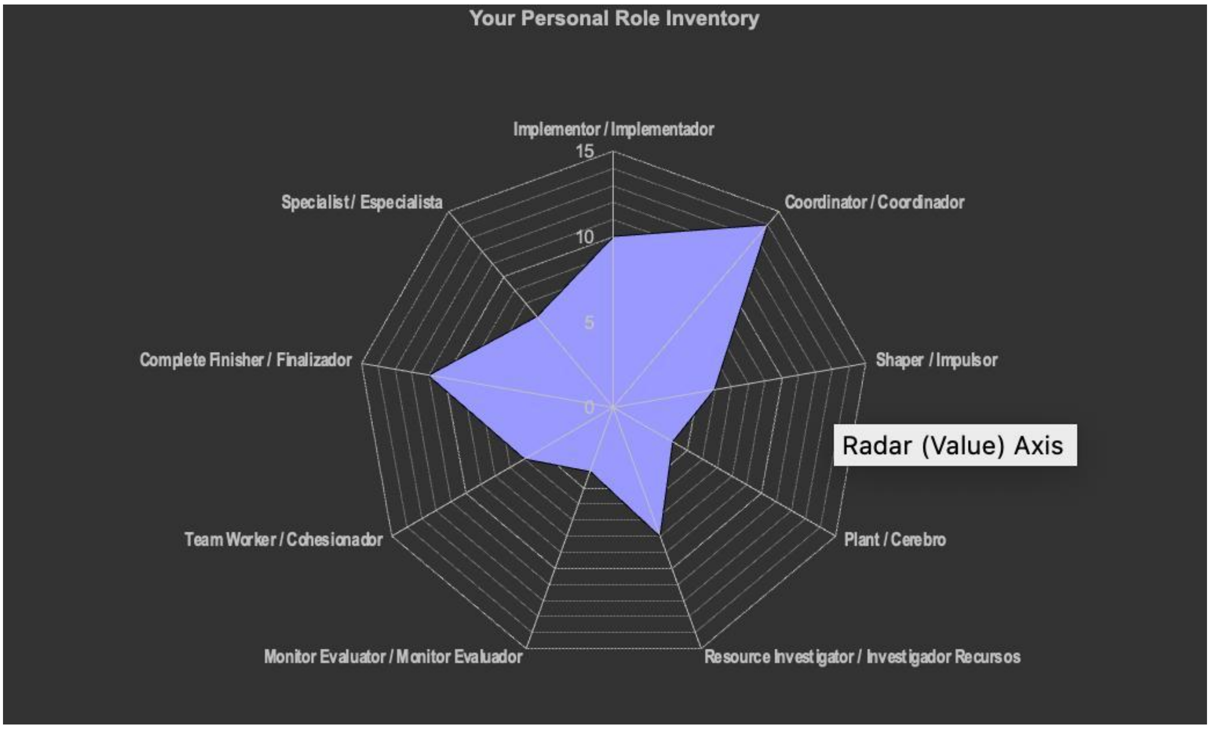
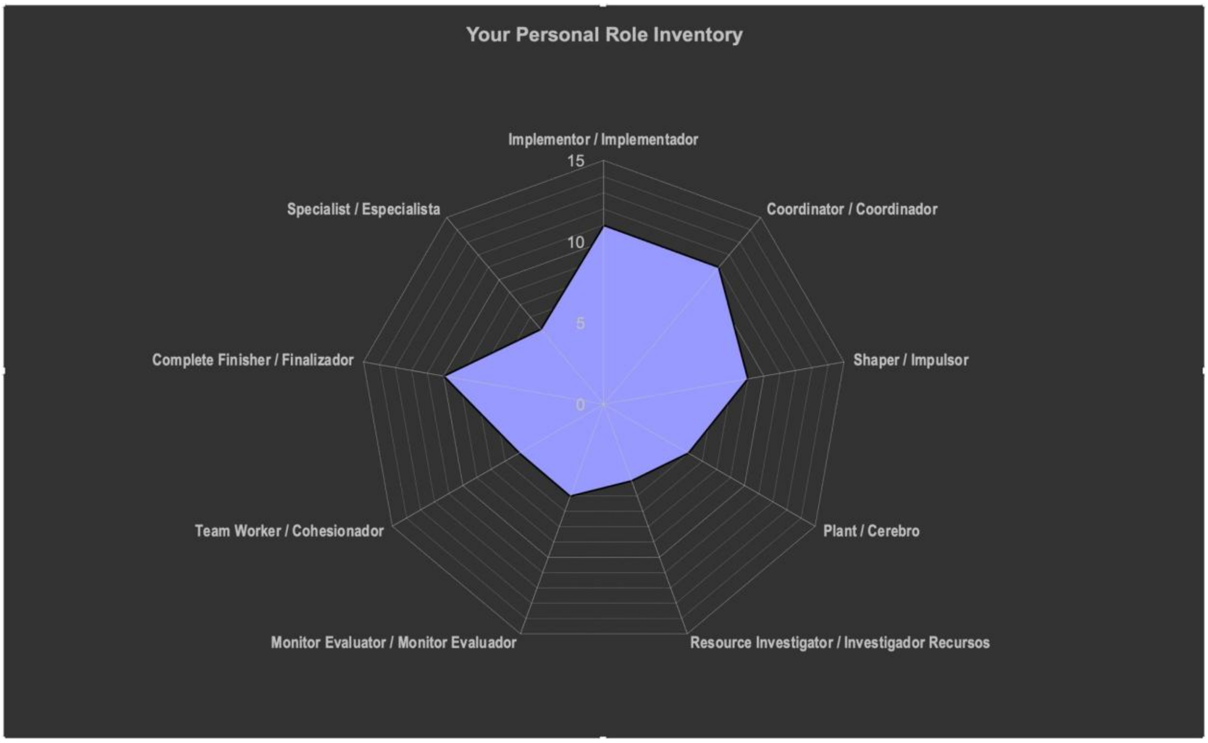
Appendix

<u>Belbin Tests</u>

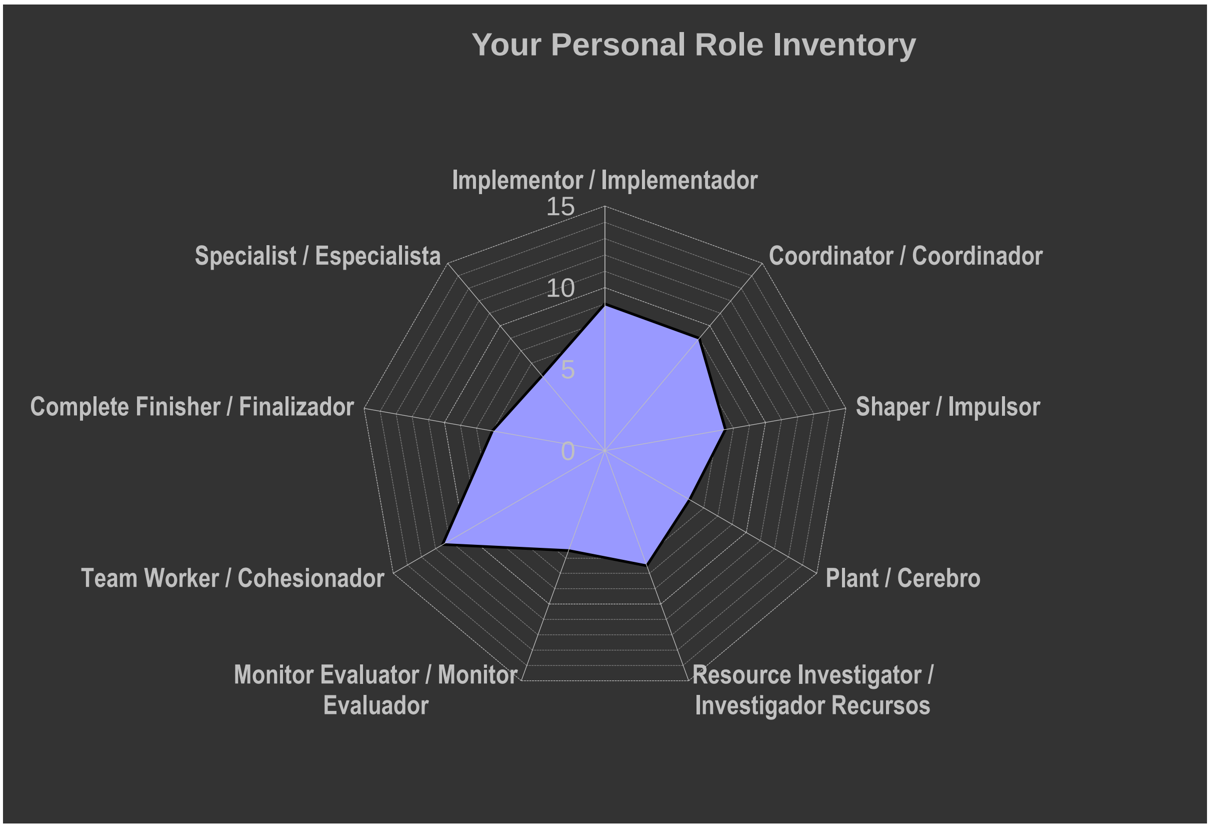
Aqib Mahmood



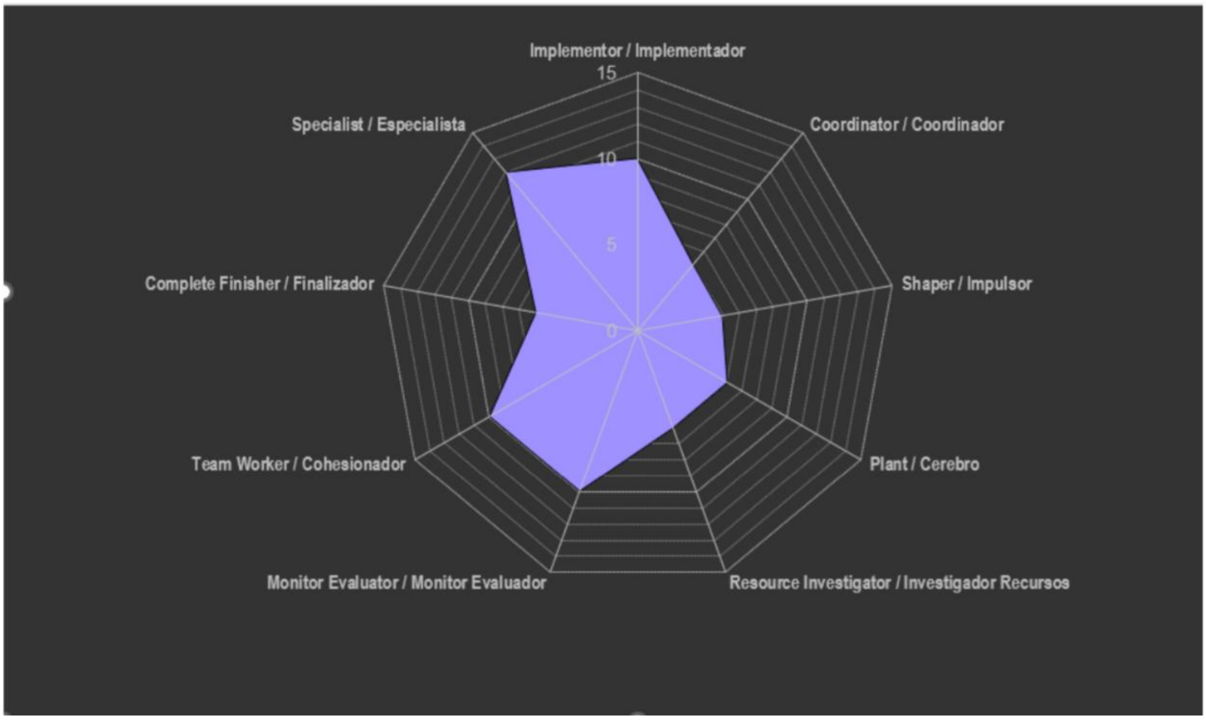
Grant Rogan



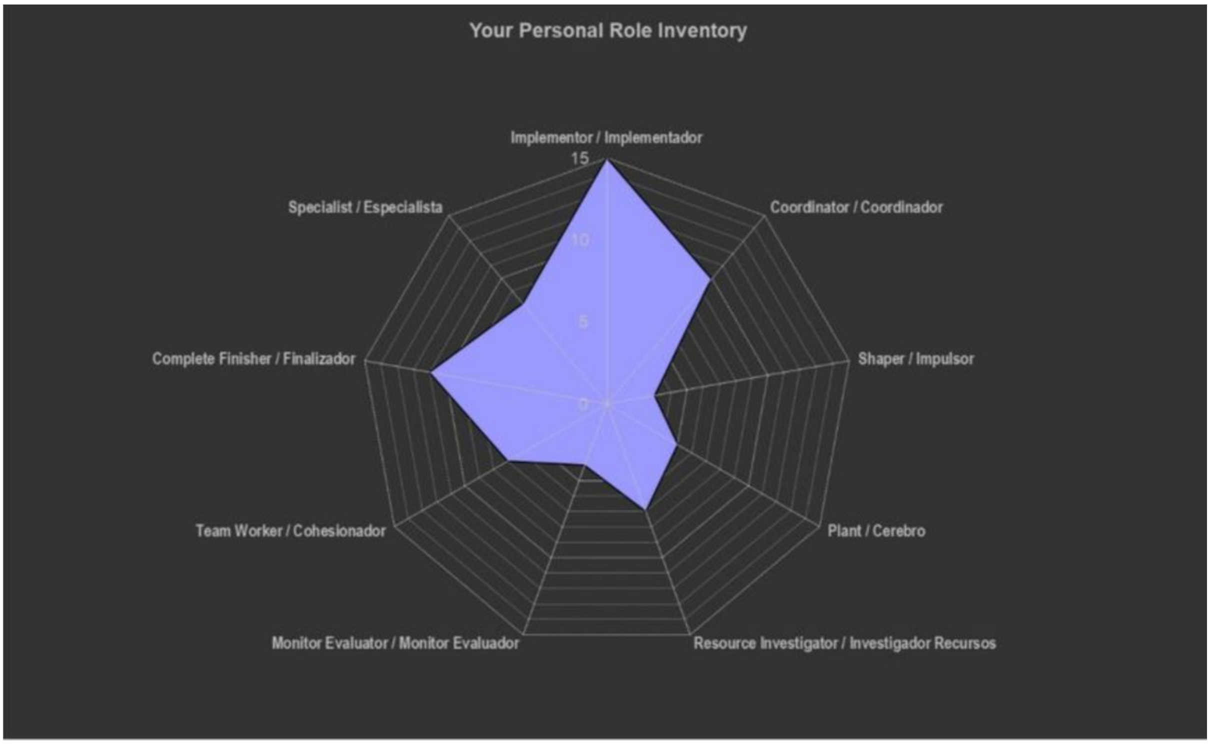
Keranveer Mann



Ayodeji Joseph



Vinay Mahal


Your Personal Role Inventory
Implementor / Implementador
Coordinator / Coordinador
Shaper / Impulsor
Plant / Cerebro
Resource Investigator / Investigador Recursos
Monitor Evaluator / Monitor Evaluador
Team Worker / Cohesionador
Complete Finisher / Finalizador
Specialist / Especialista
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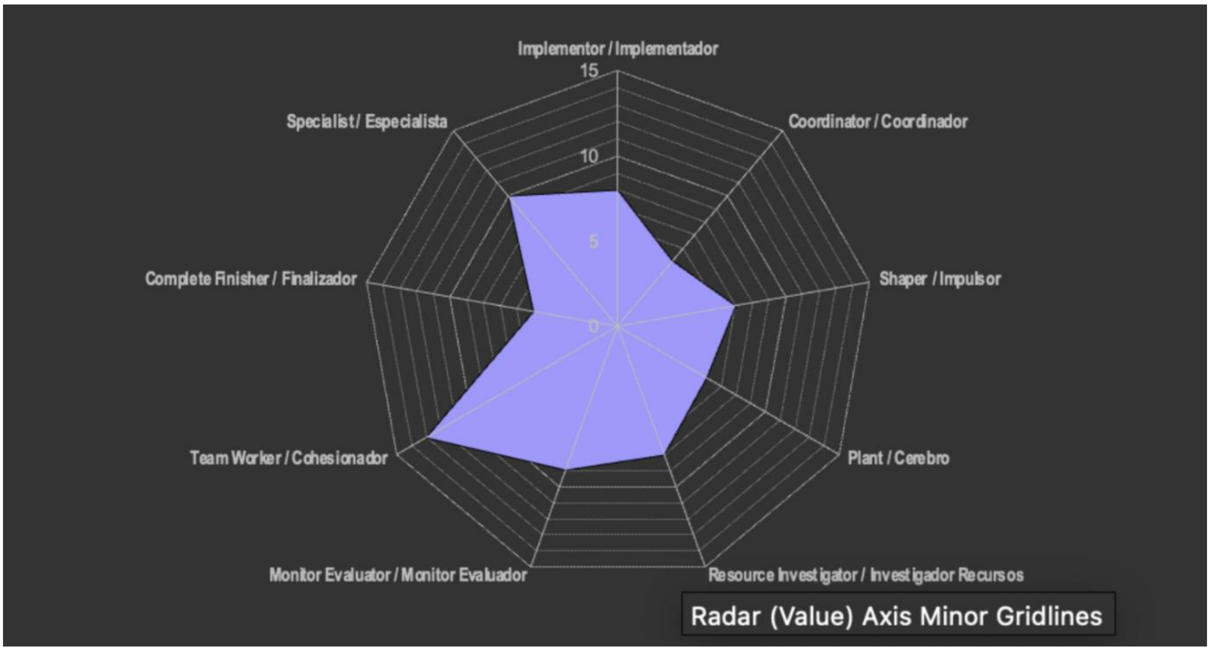
Saeed Ahmed


Implementor / Implementador
Coordinator / Coordinador
Shaper / Impulsor
Plant / Cerebro
Resource Investigator / Investigador Recursos
Monitor Evaluator / Monitor Evaluador
Team Worker / Cohesionador
Complete Finisher / Finalizador
Specialist / Especialista
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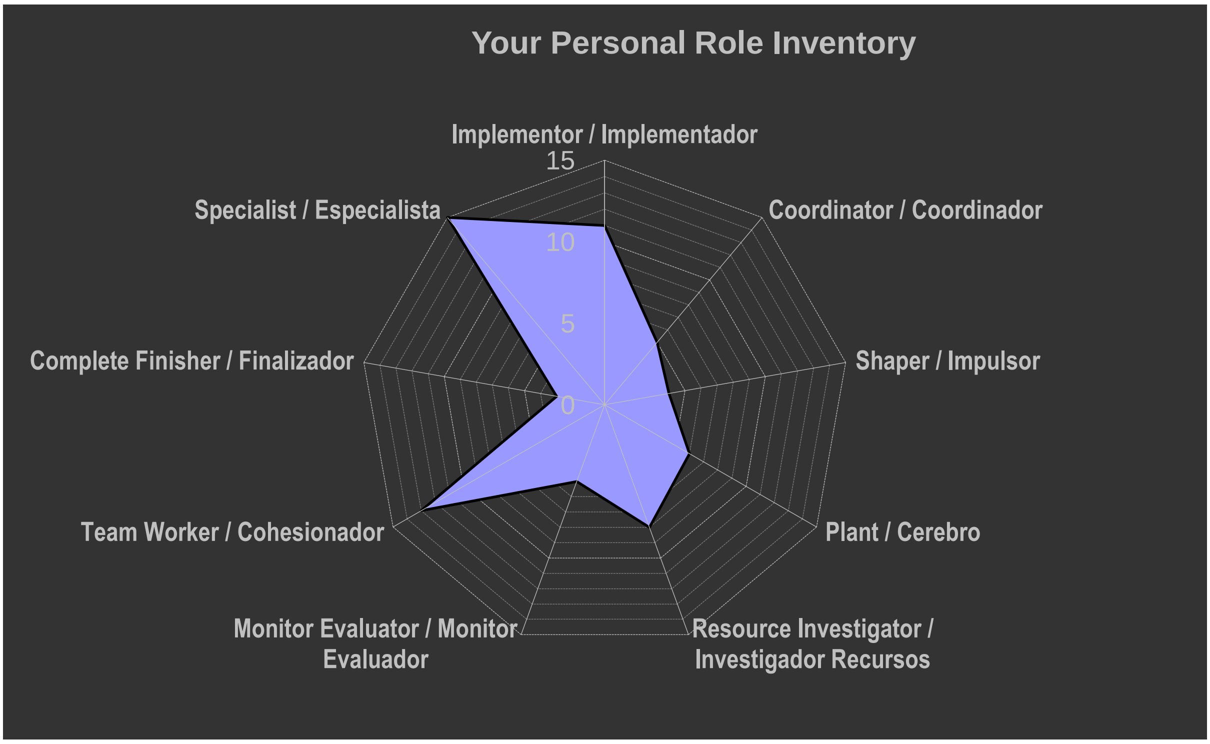
Radar (Value) Axis Minor Gridlines

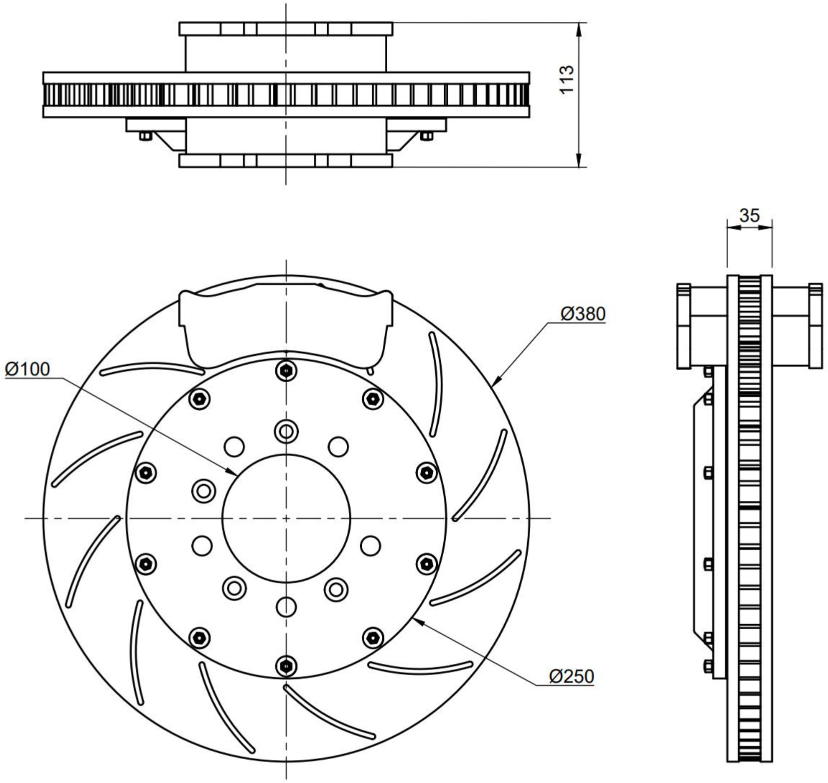
Sayhan Ali

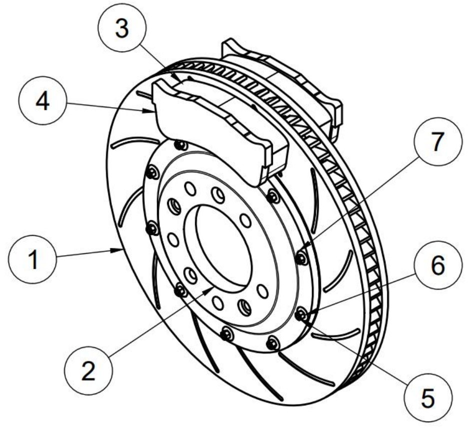

Your Personal Role Inventory
Implementor / Implementador
Coordinator / Coordinador
Shaper / Impulsor
Plant / Cerebro
Resource Investigator / Investigador Recursos
Monitor Evaluator / Monitor Evaluador
Team Worker / Cohesionador
Complete Finisher / Finalizador
Specialist / Especialista
15
10
5
0

Assumption Calculations

- Motion ratio = 1:1 (spring rate is equal to wheel rate)
- Damping is linear despite real-world progressive behaviour
- Progressive spring behaviour is approximated as linear for baseline calculations
- Tire stiffness derived from maximum estimated force
- Damping ratios chosen to represent realistic Dakar rally configurations
- Rebound: Bump damping ratio assumed as 2.5
- Gravitational acceleration (g) = 9.81 m/s^2

Drawings

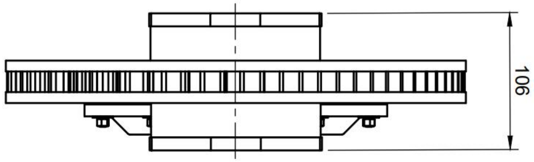


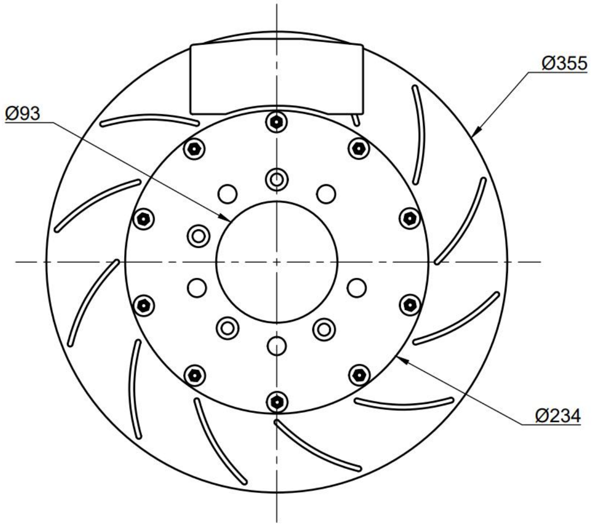


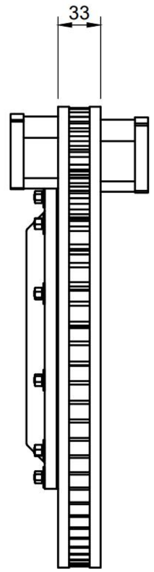
ISOMETRIC VIEW
SCALE 1:3

Parts List			
Item	**Qty**	**Part Description**	**Material**
1	1	Brake Disc Rotor	Iron, Gray Cast ASTM A48 Grade 60
2	1	Brake Disc Bell	Aluminium 7075
3	2	Brake Pad	RB 340
4	2	Brake Pad Backing Plate	Stainless Steel
5	10	DIN 6912 - M5 X 25 Steel 4.6 Plain	Steel 4.6, Plain
6	10	NF EN 24034 - M5 Steel 6 Plain	Steel 6, Plain
7	10	NF E25-514 - L5 Stainless Steel A5 Plain	Stainless Steel A5, Plain

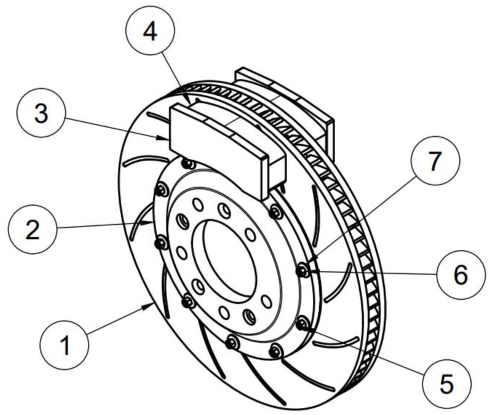
Scale: 1:3	Component Mass: 16.8 kg	Created by: Vinay Mahal 26/04/2025	Group 4
		Document type: Assembly Drawing	Document status: Final Drawing
		Dimensions in Milimetres 0 Decimal Place ±0.50 1 Decimal Place ±0.25	Component: Front Brake Disc and Pads
			Rev. / Date of issue / Sheet: 1/1





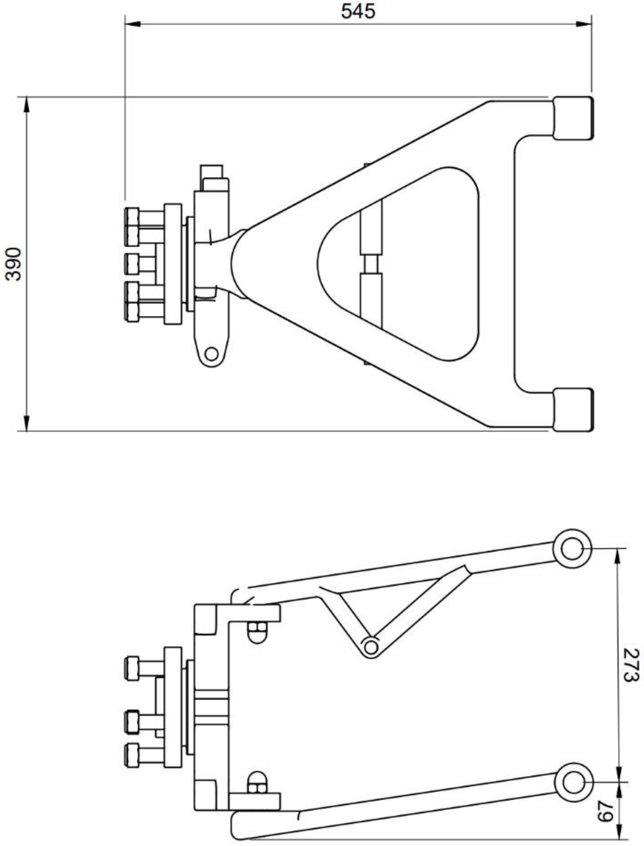


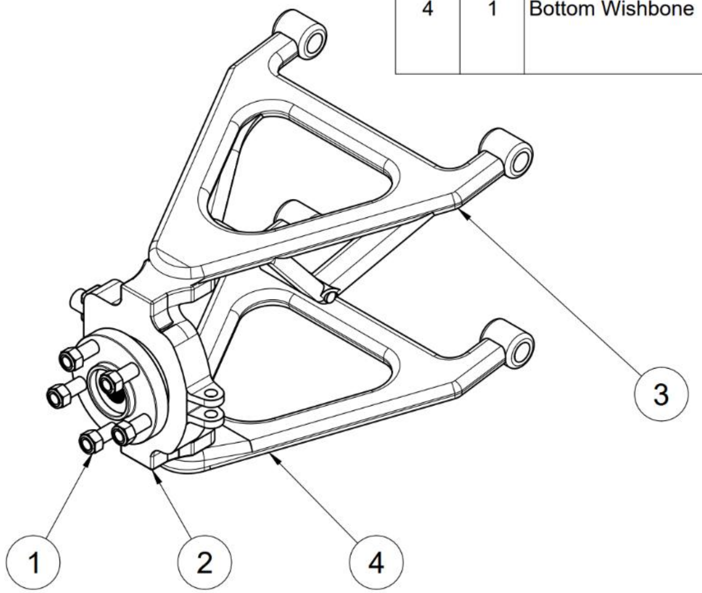
Parts List			
Item	**Qty**	**Part Description**	**Material**
1	1	Brake Disc Rotor	Iron, Gray Cast ASTM A48 Grade 60
2	1	Brake Disc Bell	Aluminium 7075
3	2	Brake Pad	RB 330
4	2	Brake Pad Backing Plate	Stainless Steel
5	10	DIN 6912 - M5 X 25 Steel 4.6 Plain	Steel 4.6, Plain
6	10	NF EN 24034 - M5 Steel 6 Plain	Steel 6, Plain
7	10	NF E25-514 - L5 Stainless Steel A5 Plain	Stainless Steel A5, Plain



ISOMETRIC VIEW
SCALE 1:3

Scale	Component Mass	Created by	
1:3	14.2 kg	Vinay Mahal 26/04/2025	Group 4
		Document type: Assembly Drawing	Document status: Final Drawing
		Dimensions in Milimetres 0 Decimal Place ±0.50 1 Decimal Place ±0.25	Component: Rear Brake Disc and Pads
			Rev. / Date of issue / Sheet 1/1

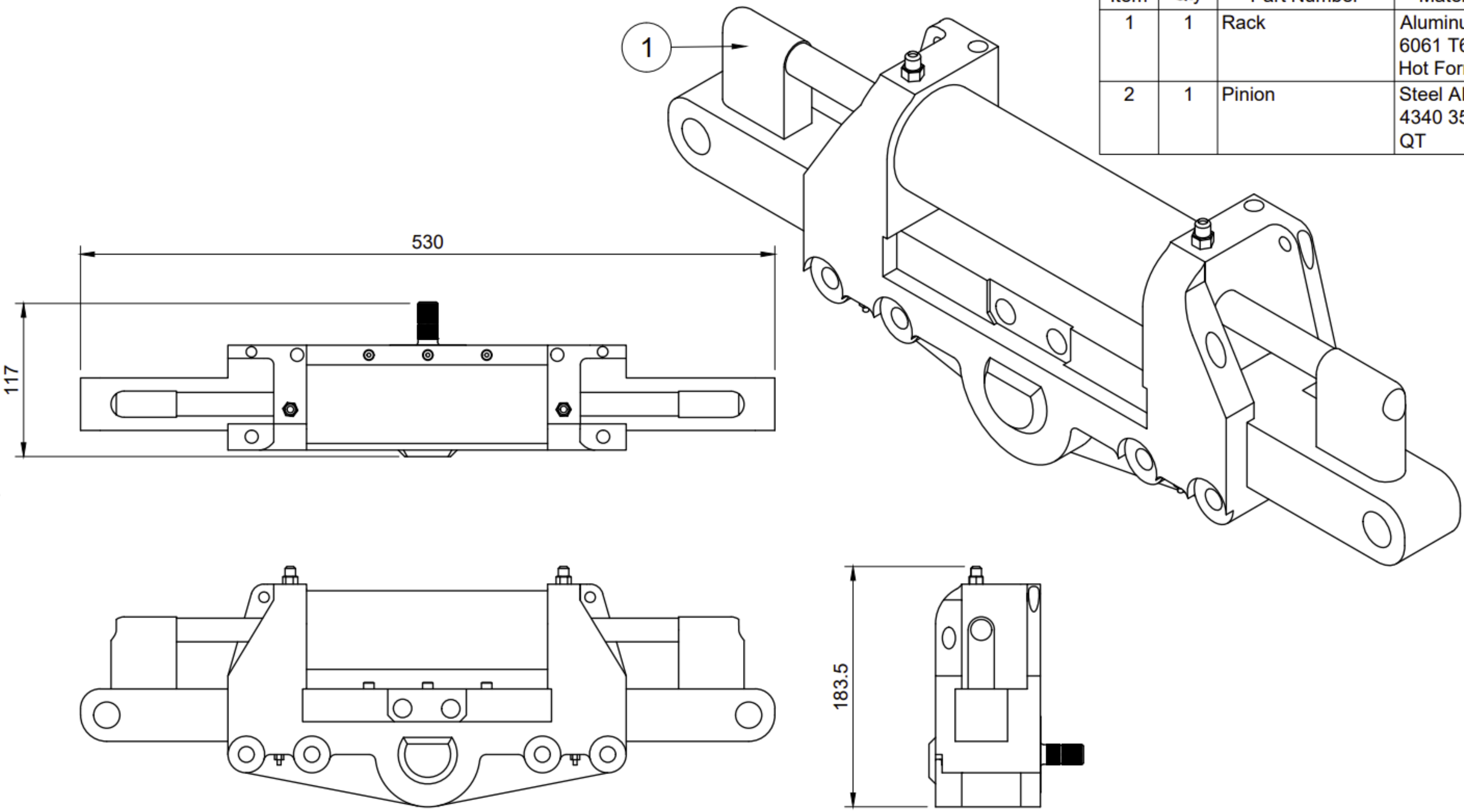




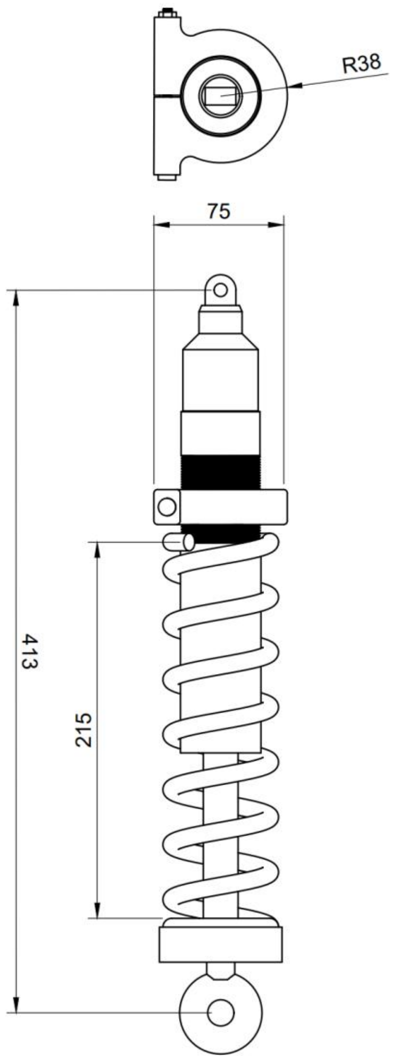
Parts List			
Item	Qty	Part Number	Material
1	1	Wheel Hub	----
2	1	Knuckle	Steel AISI 4340 350A QT
3	1	Top Wishbone 2	Steel AISI 4130 366 QT
4	1	Bottom Wishbone	Steel AISI 4130 366 QT

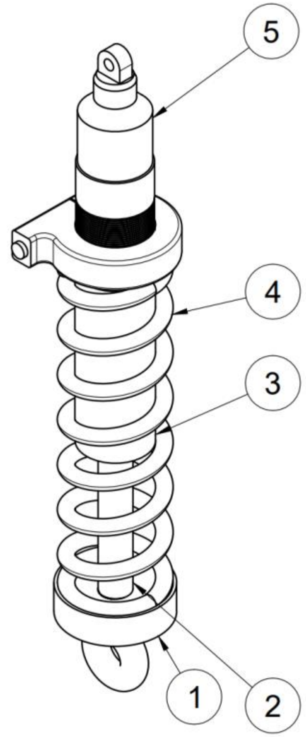
Scale	Component Mass	Created by		Group 4
1:5	9.7 kg	Dilijt Singh	22/04/2025	
		Document type		Document status: Final Drawing
		Dimensions in Milimetres 0 Decimal Places ±0.50 1 Decimal Place ±0.25		Component: Wishbones and Knuckle
				Rev. / Date of issue / Sheet: 1/1

Parts List			
Item	Qty	Part Number	Material
1	1	Rack	Aluminum 6061 T6 80 Hot Formed
2	1	Pinion	Steel AISI 4340 350A QT



Scale	Component Mass	Created by		
1:3	9.5 kg	Dilijt Singh	25/04/2025	Group 4
		Document type: Assembly Drawing		Document status: Final Drawing
		Dimensions in Milimetres 0 Decimal Places ±0.50 1 Decimal Place ±0.25		Component: Rack and Pinion
				Rev. / Date of issue / Sheet 1/1





Parts List			
Item	Qty	Part Number	Material
1	1	Lower Mount	Aluminum, Anodized Red
2	1	Piston	Steel
3	1	Cylinder	Aluminum, Anodized Blue
4	1	Spring	Stainless Steel 316L
5	1	M5x90mm Bolt	Steel

Scale	Component Mass	Created by	Group 4
1:2.5	3.3 kg	Dilijt Singh 29/04/2025	
		Document type: Assembly Drawing	Document status: Final Drawing
		Dimensions in Milimetres 0 Decimal Places ±0.50 1 Decimal Place ±0.25	Component: Spring and Damper
			Rev. / Date of issue / Sheet 1/1