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Formula 750 Redesign - Group Project

Group Design Project ACHA5014

Drew Gilasbey, Callum Carter-Davies, Evan Craig

Date (03/2023)

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1. Introduction

This report will cover the planning and progress achieved during the group project and my own roles within the group; the goal of our group project was to redesign the powertrain of the formula 750 car which is currently a solely combustion engine race series to electric, as the Fiat F.I.R.E engines used in formula 750 are becoming increasingly sparse.



Figure 1: Formula 750 race cars on track

https://www.750mc.co.uk/formulae/750-formula/gallery/photos,2015-750-formula-donington-nat_56.htm [1]

Formula 750 is one of the oldest racing formulas still competing today, it has been the proving ground for many of the most renowned motorsports engineers such as Gordon Murray and Colin Chapman therefore it is important to keep the series relevant by updating its powertrain to a more modern alternative.

Our strategy has the potential to shape the future of racing series, given the growing challenges in using combustion engines for affordable racing as fuel costs continue to rise. Moreover, as Motorsports UK shifts towards more sustainable racing, our plan could become increasingly relevant. Notably, the Formula Foundation E series offers a similar model where the race car is purchased directly from the manufacturer in a race-ready state, which would bear a striking resemblance to our plan for the formula 750 electrification. which could then be sold as a ready to race vehicle to customers.

We have been provided with a Falcon Race kits Formula 750 rolling chassis, therefore this project will require concise research and planning into the rules and regulations of the formula 750 race series that will affect which electrical components will be suitable for use on the chassis provided.

The ultimate objective of the project is to convert the formula 750 to an electric race car, with appropriate packaging and bodywork and thermals all conforming to the formula 750, and MsUK rules and regulations.

2. Project organisation

Group members were given a choice of roles covering different aspects of the vehicle to focus on (Table 1).

Table 1 - Project members and their roles.

Roles	Responsibility
Packaging, Cooling, and Battery	Drew Gilasbey
Motor and Control	Callum CD
Transmission	Evan Craig

Drew Gilasbey: I have been allocated the above roles due to my personal results in the Belbin test, my 2 highest attributes in this test were an Implementer and a plant type Belbin role. Implementers excel in planning and executing tasks efficiently, making them valuable in time-sensitive projects. They are also reliable in meeting deadlines. Within a team, implementers are skilled at organizing ideas and turning them into tangible results. These strengths will be critical in the packaging role, it requires development from initial ideas to final products, including cad models, prototypes, and fitting specific requirements.

The Plant Belbin team role is excellent in creative problem-solving, a crucial skill for my role in developing packaging from scratch. This role is also adept at solving challenging problems, valuable in overcoming the difficulty of fitting the packaging perfectly on the chassis.

Callum Carter-Davies: My main Belbin roles were plant and team worker. This suggests I will have a creative approach to coming up with ideas for the project and will work well with a team to implement the results of those ideas. My role in the group is more analytical than creative so it may be more challenging than a more experimental role.

3. Personal Role & Summary of work completed (Drew Gilasbey 2004512)

My role within the group project was packaging, cooling, and battery management. This required concise planning in order to achieve correctly. The tasks carried out and progress throughout the project will be displayed below.

3.1 Packaging

The role of packaging included testing and designing component positioning on the chassis. This was important to the overall project as if we were unable to fit the required components to the formula 750 chassis then the project would be unable to be completed. The total components required included: the Emrax 268 motor, Sevcon controller, 7 BMW i3 battery modules, Battery Management system, and cooling system. This task required precise planning as the formula 750 chassis was designed to accommodate an I.C.E package and therefore isn't necessarily designed for the electrical components planned in our project. This was overcome by using CAD models of the chassis and producing bounding boxes of all the component sizes within the chassis to figure out the correct placement of components.

3.1.1 Component Dimensions

3.1.1.1 IPace Module Dimensions and Positioning

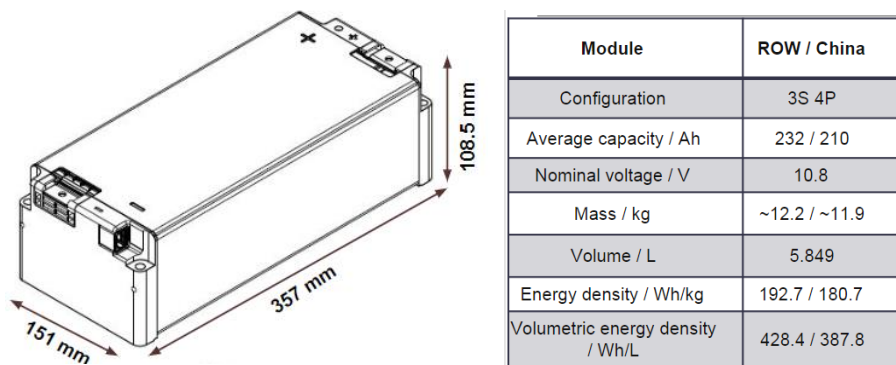
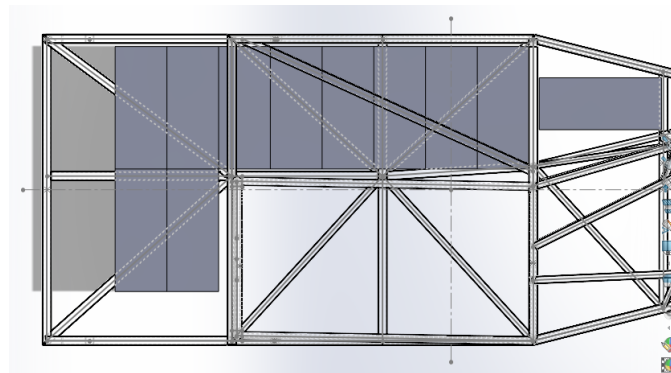
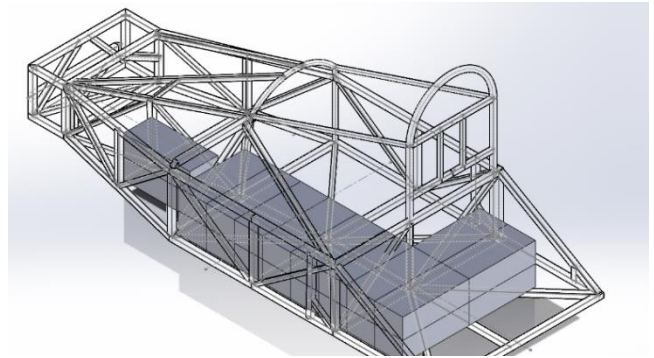
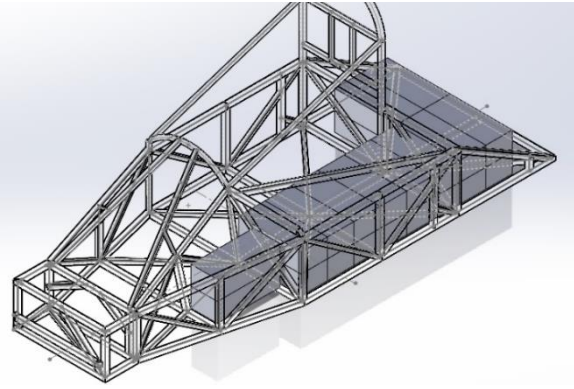


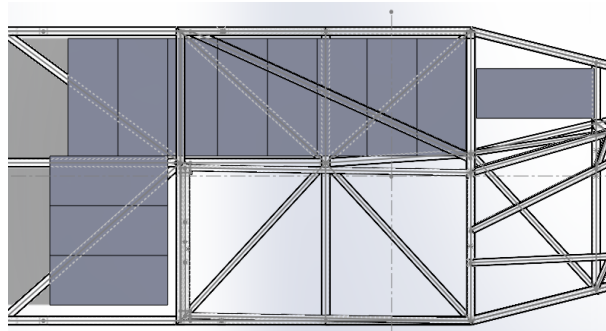
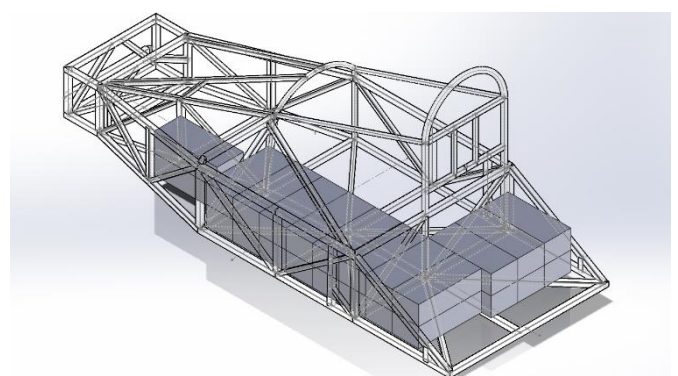
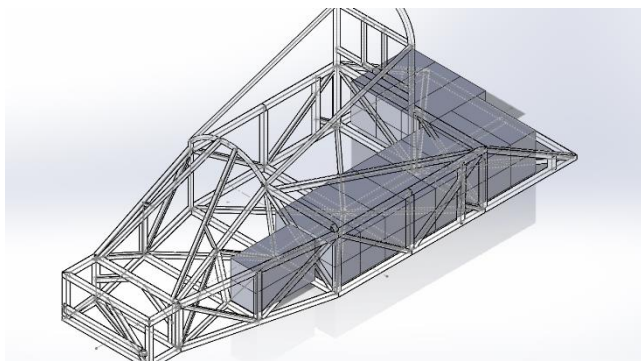
Figure1 (1): Jaguar IPace module dimensions.

Jaguar IPace Modules: weight 12.5kg per module, 26 modules required = 325kg total module weight. This has to be split across 2 battery packs as stated in the regulations [2] **5.28.1. The RESS must be comprised of a maximum of 2 separate traction battery enclosures per vehicle.** Ideally this would be equally distributed possibly having one pack beside the driver and one behind. Some CAD models were produced using SOLIDWORKS in order to visualise the module sizes and their potential positioning on the formula 750 chassis as seen below:



Figures (2): Jaguar I Pace module positioning no.1.

Second Arrangement



Figures (3): Jaguar I Pace module positioning no.2.

It can be seen from the two arrangements of the I pace modules above that the required number of modules will result in a great deal of space taken up on the chassis, this is true of both the module arrangements modelled. The modules are unable to be separated into more than 2 separate clusters on the chassis as stated in the MSUK regulations. Therefore, it was decided to research alternative battery options in order to find batteries with a higher capacitance to reduce the overall total number of modules required.

3.1.1.2 BMW I3 Module Dimensions and Positioning

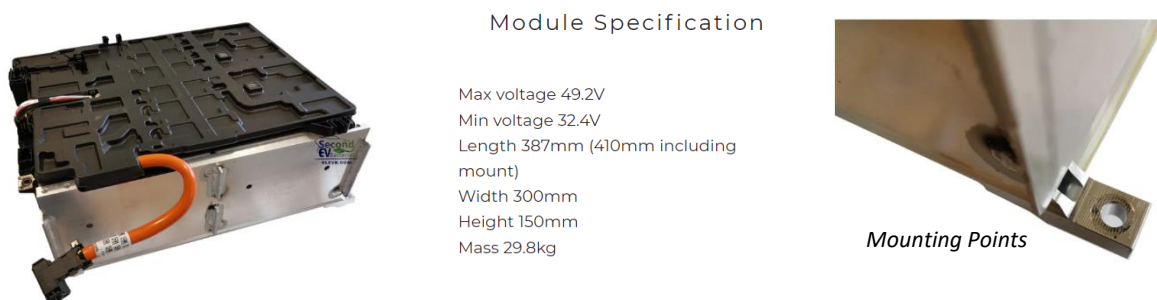
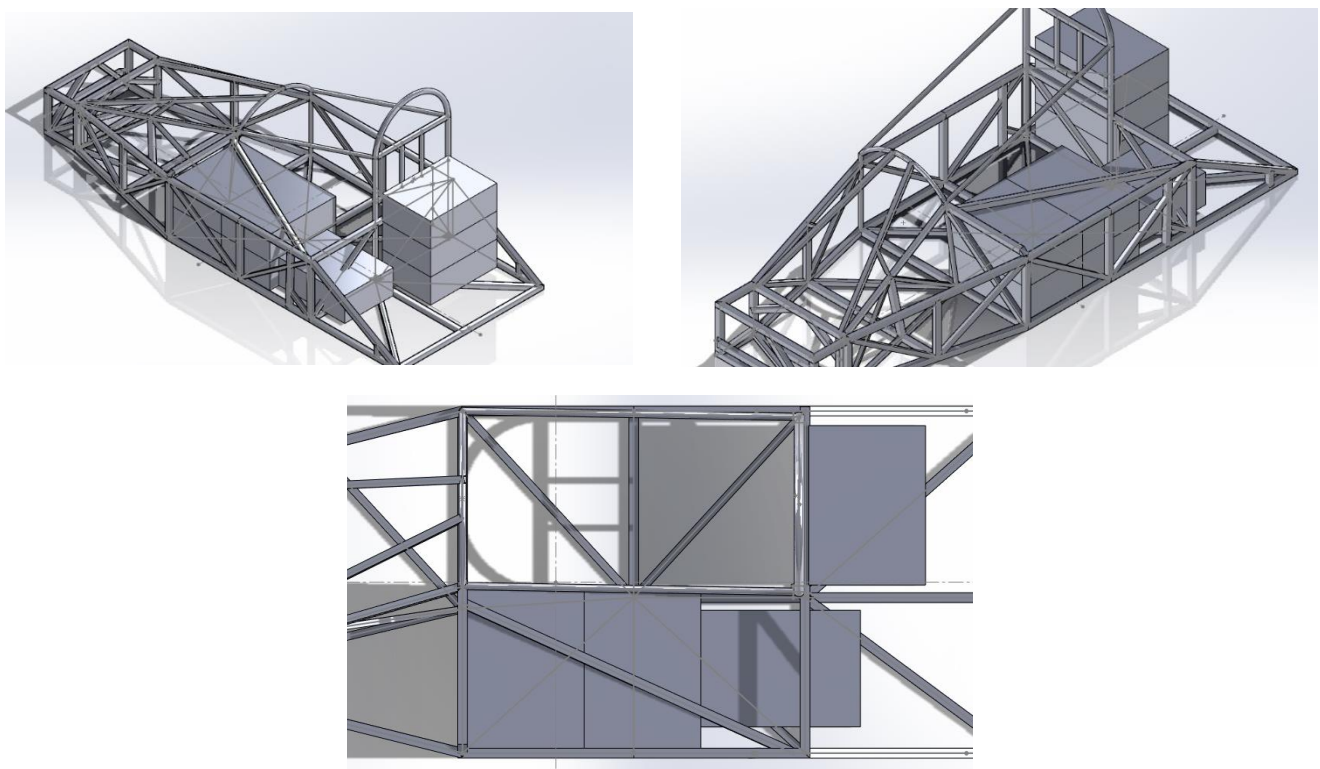
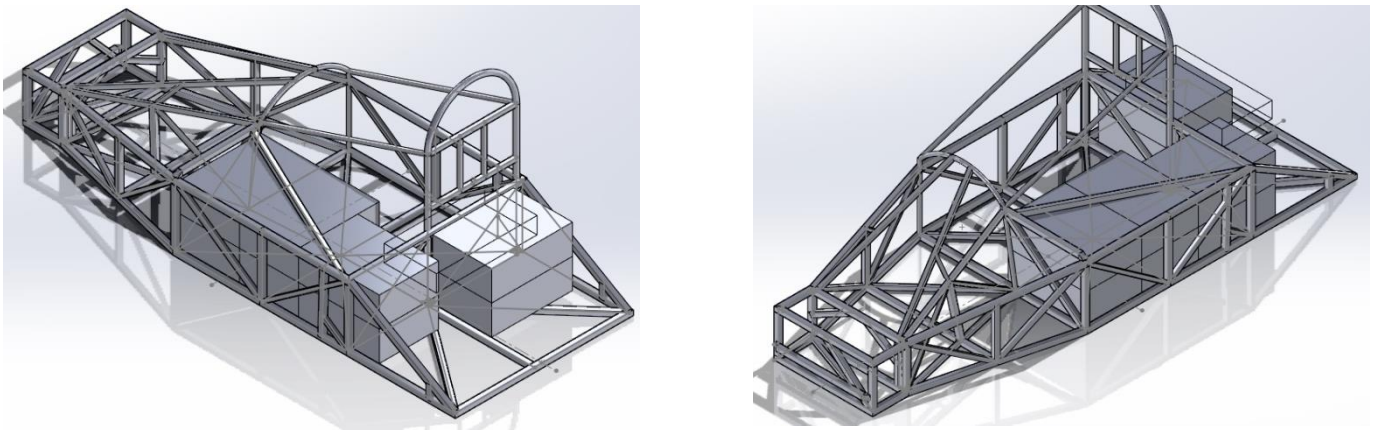


Figure (4): BMW I3 module dimensions <https://www.secondlife-evbatteries.com/collections/ev-modules-packs/products/bmw-i3-120ah-module> [3]

It was decided that BMW I3 modules would be more suited to the application as they possess a higher capacitance at a lower weight and therefore only 7 or 8 modules would be required at a weight of 208.6kg and 238.4kg respectively. CAD models were produced in order to work out where best to position these modules on the chassis:



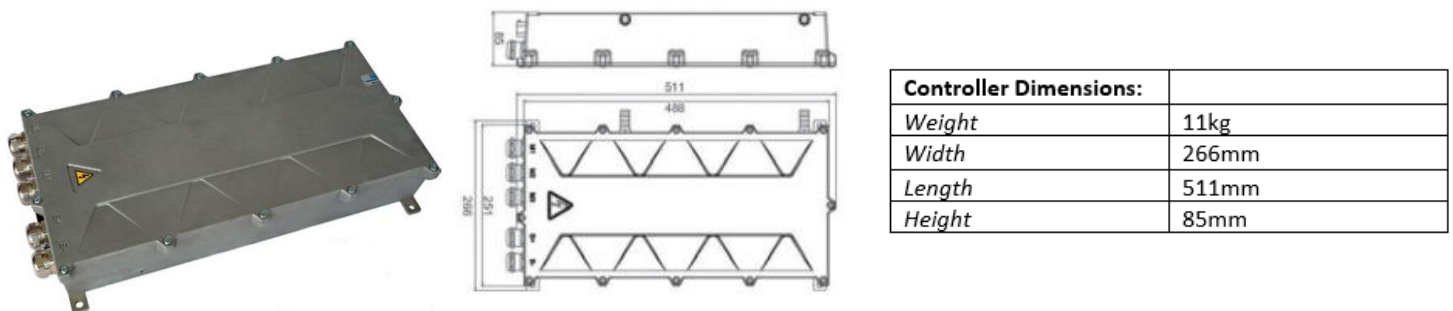
Figures (5): BMW I3 module positioning no.1.



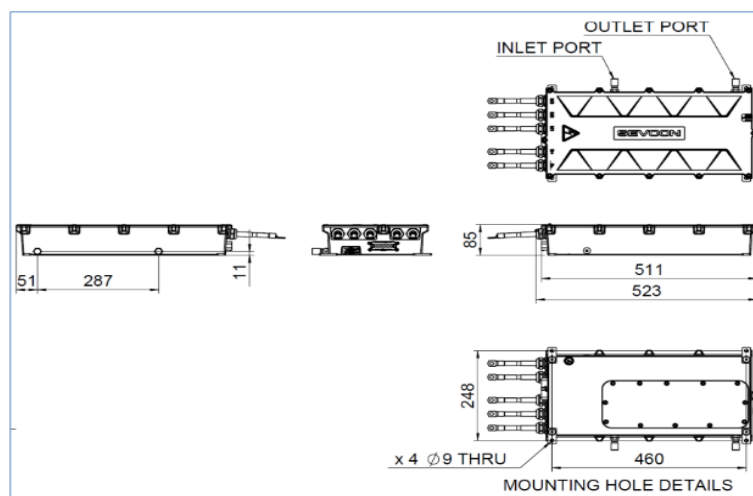
Figures (6): BMW I3 module positioning no.2.

It was decided at this stage that the best arrangement for the modules would be option number 2 as this leaves the most space on the chassis for the other components. Additionally, the first option has a stack of 3 modules positioned behind the drivers head and this is not desirable as would increase the centre of gravity height of the vehicle, and in turn would negatively affect the handling.

3.1.1.3 Controller dimensions and Positioning



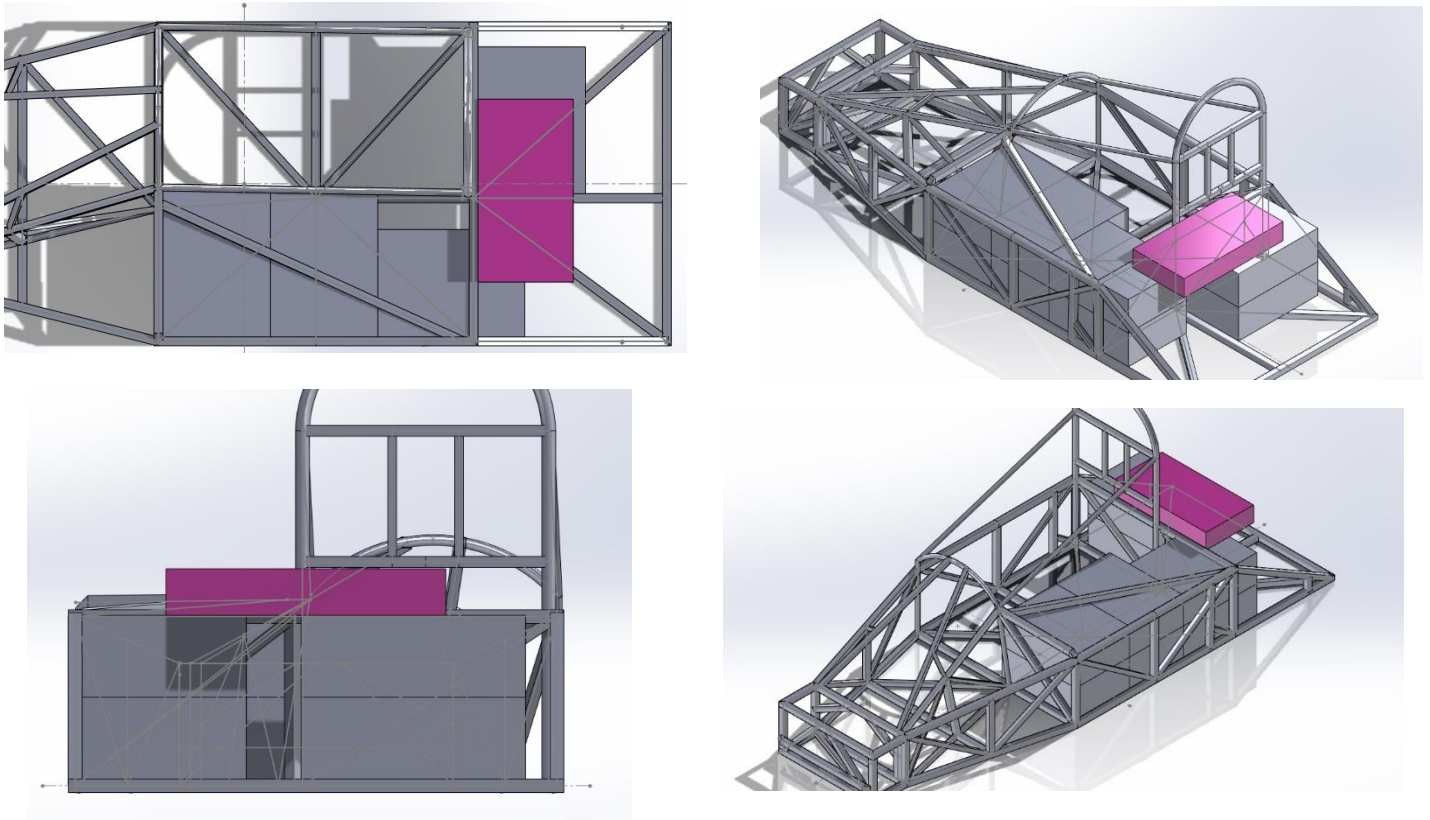
Figures (7): controller dimensions



The inverter should not be used as a stressed member.
Flatness of mounting surfaces: <0.2mm

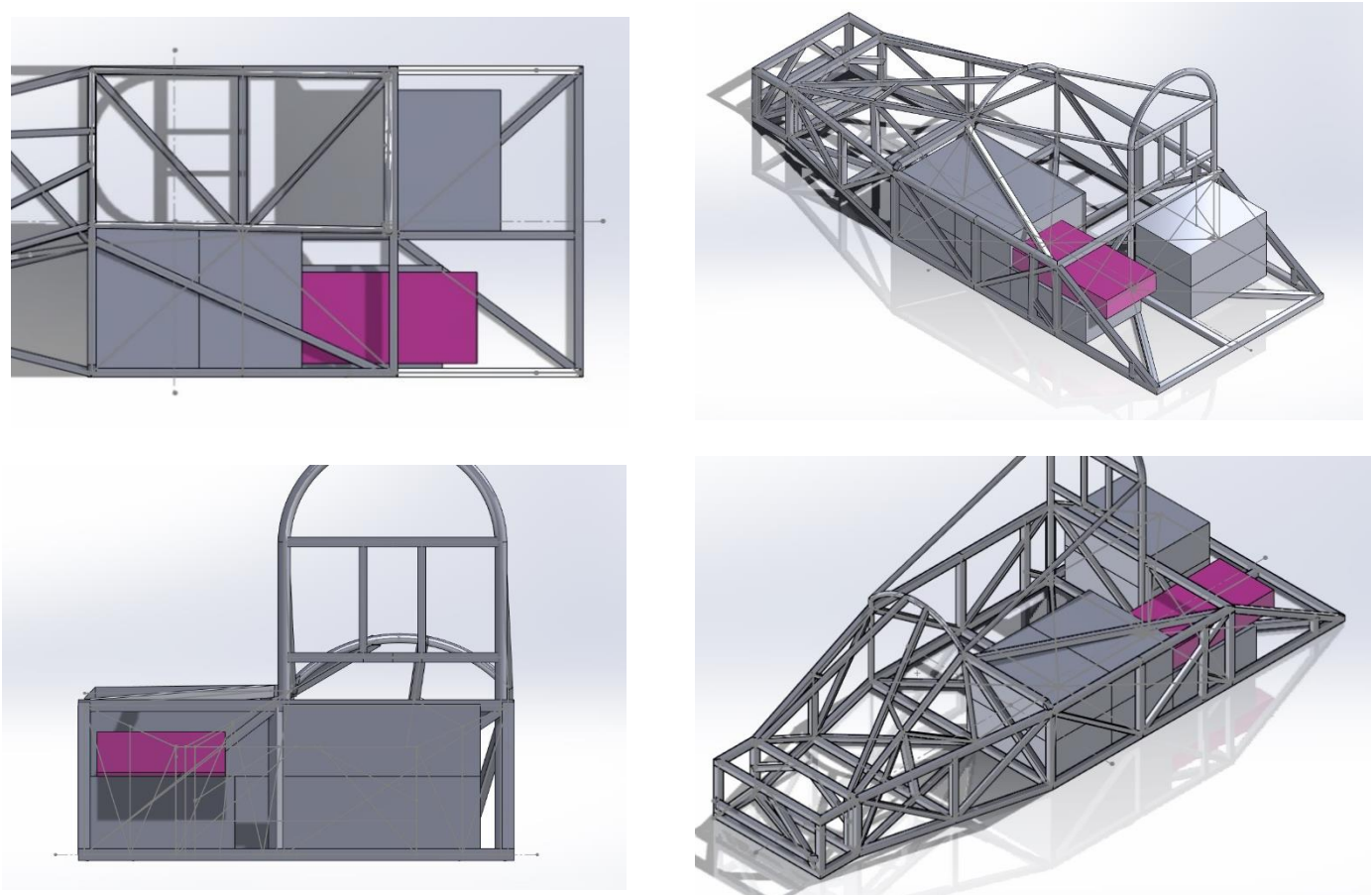
Figure (8): controller instillation diagram

A CAD model of the controller was produced using the given dimensions and was positioned on the chassis with the battery modules in a couple of different arrangements as seen below:



Figures (9): BMW I3 module positioning no.2 with controller mock-up (in Pink). 8 Modules

Second Arrangement



Figures (10): BMW I3 module positioning no.2 with controller mock-up (in Pink). 7 Modules

It was decided that 7 battery modules would be sufficient for the race and therefore the final controller positioning model only contains 7 modules, this was beneficiary towards the controller positioning as it now can be placed lower to the chassis which would reduce the centre of gravity height of the vehicle this is important as the aim is for the vehicle to be able to race and a low centre of gravity height is important to reduce the roll centre height and increase the vehicles handling characteristics.

3.1.1.4 Emrax Motor dimensions and Positioning

Emrax 268 motor is to be used, a CAD drawing with dimensions can be seen below:

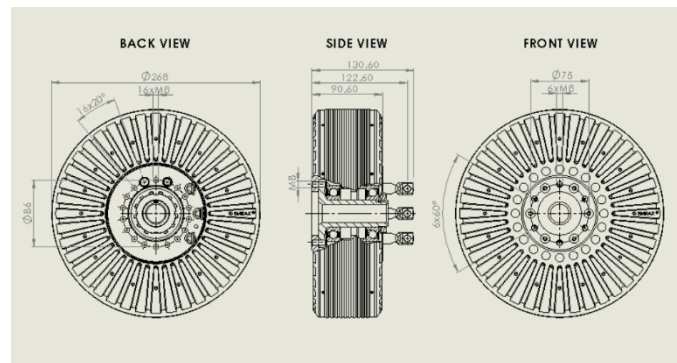
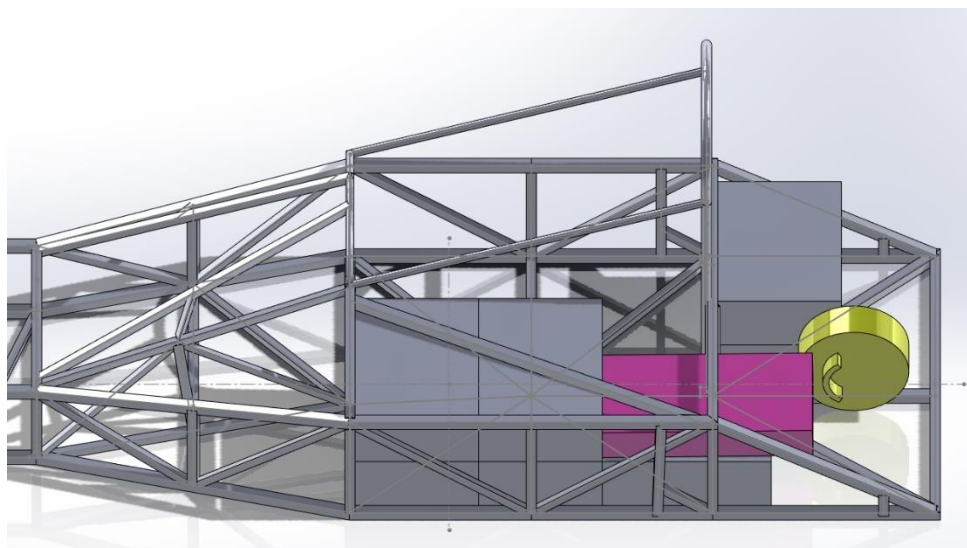


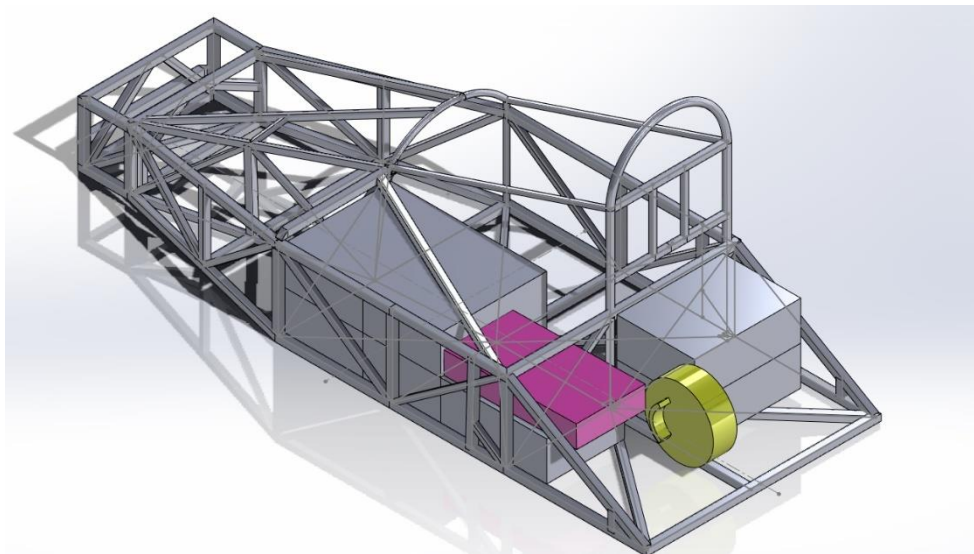
Figure 9: EMRAX 268 pictures

Figure (11): Emrax motor dimensions: Manual for EMRAX Motors. Version 5.4. [4]
March 2020

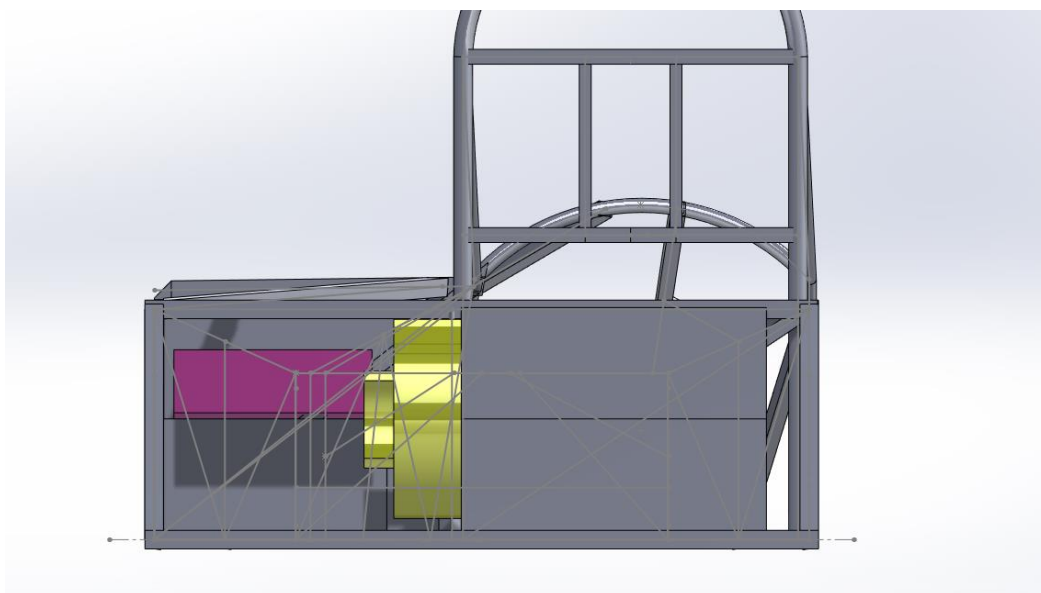
With the controllers positioning decided on the chassis it was clear to see that the motor can be installed in the most optimal location in this case the rear of the chassis. This is also the final position of all components on the chassis.



Figures (12): BMW I3 module positioning no.2 with
controller mock-up (in Pink), And motor (in gold)



Figures (13): BMW I3 module positioning no.2 with controller mock-up (in Pink), And motor (in gold)



Figures (14): BMW I3 module positioning no.2 with controller mock-up (in Pink), And motor (in gold)

3.2 Cooling

Cooling tasks entailed calculating the required amount of cooling needed in order to run the car and whether to solely liquid cool the components or some components can be run on combined cooling of liquid and air, radiator sizing, and radiator placement to best to cool the components, battery modules, controller etc.

It was decided that the best option for cooling would be to solely liquid cool the components as from the cad models above its clear to see that many of the components would be out of direct airflow and therefore would not be sufficiently cooled by air cooling or combined cooling. In addition to this the Emrax motor possesses a higher IP rating when solely liquid cooled at IP65 as opposed to IP 21 for air cooled and combined cooling as seen in the figure below.

EMRAX
INNOVATIVE E-MOTORS

Manual for EMRAX Motors / Generators | Version 5.4 | 5th of March 2020

EMRAX 268 Technical Data Table

Type	EMRAX 268 High Voltage			EMRAX 268 Medium Voltage			EMRAX 268 Low Voltage*		
Technical data									
Air cooled = AC Liquid cooled = LC Combined cooled = Air + Liquid cooled = CC	AC	LC	CC	AC	LC	CC	AC	LC	CC
Ingress protection	IP21	IP65	IP21	IP21	IP65	IP21	IP21	IP65	IP21

Figure (15): Manual for EMRAX Motors. Version 5.4. March 2020 [4]

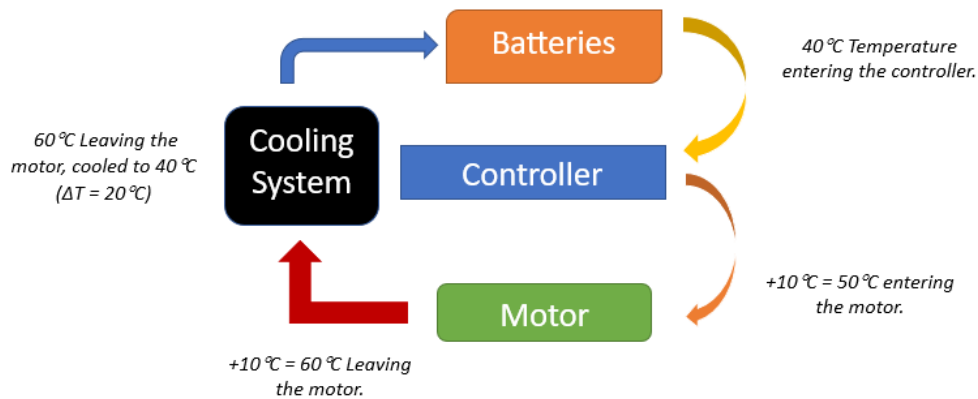


Figure (16): diagram explaining the cooling requirements.

The required amount of cooling was calculated by finding the efficiencies of the components which required cooling each of the components had an estimated efficiency of 95% [13] and three components total required cooling therefore there would be a total inefficiency of 15% or 5% per component. The total power output will be 100kW therefore each component loses 5kW worth of energy as heat 5kW is equal to 5000J/s. Water has a specific heat capacity of 4,184 Joules of heat for the temperature of one kilogram of water to increase 1°C. [5]. Consequently dividing 5000(j/s) with

the SHC of water 4,200 (rounded up) = 1190 or 1.2L and then dividing by the temperature change from each component (ΔT) 10°C as seen in *figure (16)*, that produces 0.12L/s therefore each component requires 0.12L/s of water flow meaning a total of 0.36L/s or 21.6L/m.

3.2.1 Water pump

An adequate water pump system was investigated in order to produce the 21.6L/m of cooling required. Using the table below it was decided that the best form of water pump would be that of a motorbike between 500cc and 1000cc as it possesses a flow rate of 23L/m. The required flow rate is 21.6L/m so this is ideal. The EBP15 was not chosen as its flow rate would not be sufficient to cool the components at 15L/m.

DAVIES CRAIG APPLICATION GUIDE

APPLICATION	PUMP MODEL	PUMP ONLY 12V	PUMP KIT 12V	PUMP COMBO 12V
ENGINES UP TO 2.0L (STANDARD)	EW P80	8105 OR 8109	8005 OR 8009	8907 OR 8919
ENGINES FROM 2.0-3.5L (STANDARD)	EW P115	8125 OR 8140	8025 OR 8040	8930 OR 8950
ENGINES FROM 3.5L - 5.0L (STANDARD)	EW P130	8180	8080	8990
ENGINES 5.0L OR GREATER (STANDARD)	EW P150	8160	8060	8970
MODIFIED ENGINES SUPERCHARGED/TURBO	EW P150	8160	8060	8970
DRAG RACING, DRIFTING & SPEEDWAY	EW P150	8160	8060	8970
OFF ROAD / 4WDS	EW P130	8180	8080	8990
MOTORBIKE UP TO 500CC (STANDARD)	EB P15	9002	9001	-
MOTORBIKE 500CC - 1000CC (STANDARD)	EB P23	9051	9050	-
MOTORBIKE 1000CC - 2000CC (STANDARD)	EB P25	9025	9025	-



Figure (17): Table of water pump options: <https://www.demon-tweeks.com/uk/davies-craig-ewp80-electric-water-pump-dcp8005/> [6]

Figures (18 & 19): The chosen water pump EBP23: <https://daviescraig.com.au/product/ebp23-electric-booster-pump-kit-12v-9050> [7]

The water pump chosen comes with all appropriate fittings and fasteners within the package, the manufacturer states that the water pumps intended usage is for “solar pump applications, water cooled motorcycle, go-kart engines, turbo air / water intercoolers & household irrigation.” [4] this is ideal for the application within the formula 750 as the heat energy dissipation is similar to that of a motorcycle.

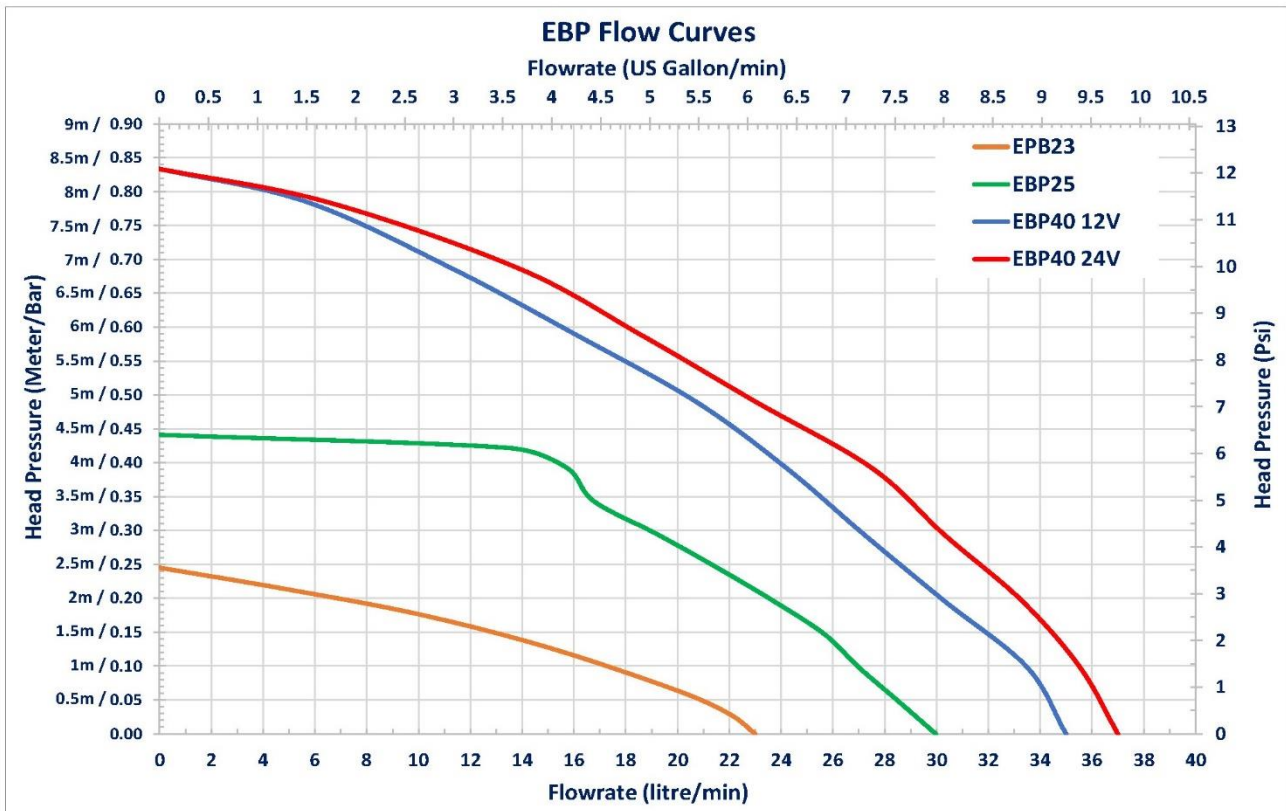


Figure (20): graph showing flow rate of the chosen water pump EBP23 in orange:
<https://daviescraig.com.au/product/ebp23-electric-booster-pump-kit-12v-9050> [6]

3.2.2 Radiator Sizing

Specific heat capacity of water – 4,200

Coolant mass flow rate – 0.1 kg s

Coolant temperature leaving the controller – 60°C (333k)

Coolant temperature needed back into the battery - 40°C (313k)

Air flow through radiator at average speed – Mass air flow rate =

$V \cdot A \cdot \rho = 40\text{m/s} \cdot 0.011 \text{ (frontal area of the airbox)} \cdot 1.293\text{kgm}^3 = 0.65 \text{ rounded to } 0.70 \text{ kg s}$

Ambient air temperature - 20°C (293k)

Overall heat transfer coefficient – 300 W m⁻² K

Using the following equation, the heat loss, or energy a second, or power loss from the coolant due to temperature change can be calculated:

$$Q = \dot{m} c \Delta T$$

$$Q = 0.1 \cdot 4.200 \cdot (60 - 40)$$

$$Q = 0.1 \cdot 4.200 \cdot 20$$

$Q = 8400\text{W}$ or 8.5kW (transferred to the air)

$8.5\text{ kW} \times 3$ (heat lost through a motorcycle radiator 30%) = $25.5\text{kW} = 34.2\text{hp}$

With this information an appropriate radiator for a vehicle/motorcycle of around 35hp was researched.

The Honda CRF250R was chosen as it produces 37 hp as seen in the graph below, and components are readily available for this motorcycle.

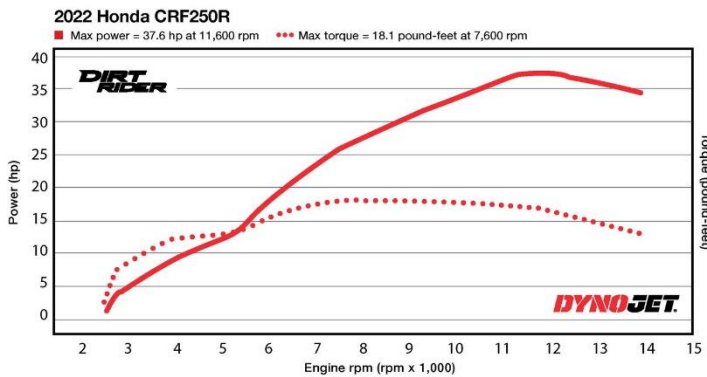


Figure (21): graph showing engine hp versus rpm:

<https://www.dirt rider.com/story/tests/honda-crf250r-dyno-test-2022/#:~:text=With%2037.6%20hp%20and%2018.1,peak%20th n%20the%20%2721%20model.> [8]



Figure (22): Honda CRF250R:

<https://www.honda.co.uk/motorcycles/range/off-road/crf250r/overview.html> [9]

Mishimoto produce an aftermarket aluminium radiator for the specified motorbike model number: MMDB-CRF250R-10LX available at: <https://www.mishimoto.co.uk/honda-crf250r-aluminium-dirt-bike-radiator-10-left-braced.html> this will be perfect for the application within the formula 750 race car as it conforms to the requirements aforementioned and is readily available for purchase. The best location for the mounting of the radiator would be behind the driver's head receiving air through the airbox. This can also be mounted on top of the battery modules.



Figure (23&24): Honda CRF250R radiator: <https://www.mishimoto.co.uk/honda-crf250r-aluminium-dirt-bike-radiator-10-left-braced.html> [10]

3.3 Battery selection

Batteries were researched and selected to produce 100kW of power over 20 minutes of racing, the Jaguar I pace batteries [11] required 26 modules at a cost of £10,348 total, in order to achieve this as seen in section 3.1.1.1. The I pace modules proved very difficult to fit onto the chassis as there is a large number of them required and their overall weight was very high. Thus, as aforementioned it was decided to use BMW I3 modules instead of the I pace modules as they are newer batteries and posses a higher capacitance at a lower weight. A total of 7 modules will be required with a total weight of 209kg this is over 100kg lighter than the I pace modules, additionally only requiring 7 modules frees up a lot of space on the chassis for other necessary components as seen in section 3.1.1.2. The BMW modules are also significantly cheaper as 7 modules will cost £6,993 total. [3]

The batteries will also require a battery management system or BMS this can be purchased from the same location as the batteries them selves for a cost of £495.



EVS-BMS are fully compatible, including balancing, with the following OEM batteries. Details on limitations due to OEM firmware limits will be published in the [user manual](#).

1. VW E-Golf
2. VW GTE
3. VW ID3 and ID4 (MEB) 12S and 8S modules
4. BMW Hybrid - Gen 1 (9 kWh) and Gen 2 (12 kWh)
5. BMW I3
6. Mitsubishi Outlander - LEV40 & LEV46 modules
7. Chevy Volt - Gen 1
8. Tesla Model S/X

Figure (25): BMS suitable for the I3 modules: <https://www.secondlife-evbatteries.com/collections/accessories/products/evs-bms-simpbms-battery-management-system> [12]

6 Summary of Work: Callum Carter-Davies

My areas of focus were on the motor, controller, and transmission of power to the rear differential. This involved analysing the performance of the existing ICE vehicle, researching the function of existing parts, designing mountings for those parts, and determining the expected performance of the vehicle.

6.1 Rules & Regulations

To be eligible for use in motorsport in the UK the vehicle had to follow the regulations set out by the governing body Motorsport UK, in the MSUK Blue Book 2022, which defined the vehicle as a bespoke BEV. As the vehicle was intended for use in the Formula 750 championship it was also essential to follow the restrictions found in the Formula 750 rulebook, with the exception of the rules that would no longer apply such as requirements for the engine.

Restrictions related to my role in the project can be seen in Table 2.

Table 2 - Applicable Restrictions from MSUK and F750.

Regulating Body	Restriction
Motorsport UK	DC voltage limit of 450 V
	Maximum output power of 250 kW
	Maximum of 2 motors with combined output
	Only 1 driven axle
	Must have reverse gear
	Must have a manual isolator switch
	Must have an automatic isolator
	Motor can't be hub mounted
	No Torque Vectoring
	Must have status indicator light
Formula 750	Driver controlled gear selection

6.2 Analytical Work

6.2.1 Drive Cycle Analysis

A drive cycle based off previous data recorded at Donnington Park National circuit was analysed using MATLAB (MATLAB, 2022) to determine a baseline for the performance of the vehicle, which the EV conversion would then be designed to match or exceed. The MATLAB code can be seen in Appendices 1,2,3.

The recorded velocity had to be filtered to remove peaks in the data caused by interference, this noise was reduced using the 'movmedian' function (Figure 26). The maximum velocity of the vehicle during the lap was found to be 48.6 m/s, and the minimum velocity during the lap was 27.9 m/s. This was deemed to be the target range of the motor's operation to maximise efficiency, reducing losses and allowing for a reduction in battery mass.

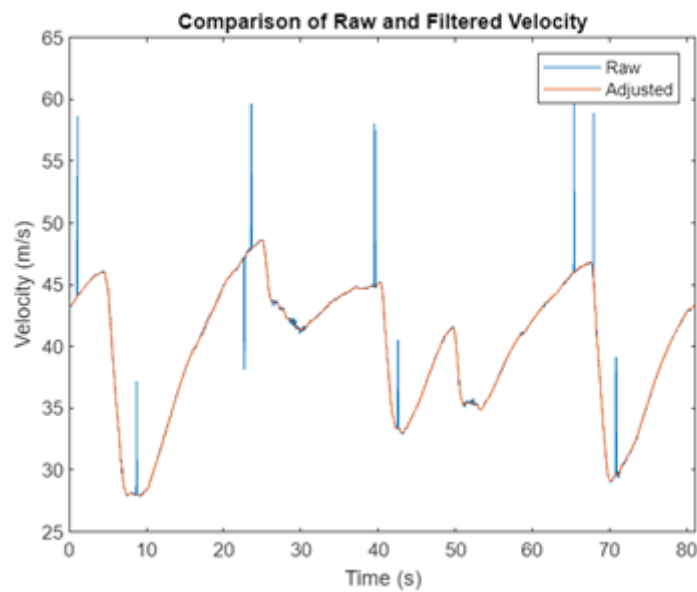


Figure 26 - Comparison of raw and filtered recorded velocity.

The acceleration of the vehicle was then calculated by dividing the differential of velocity by the differential of time (Figure 27).

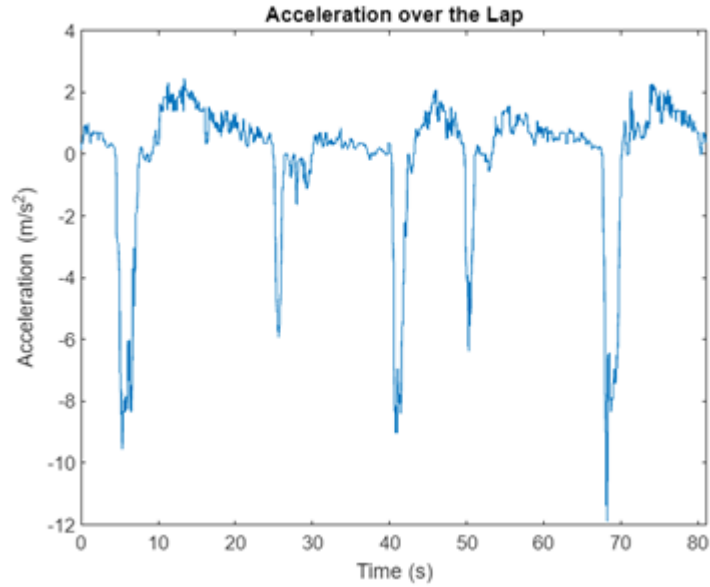


Figure 27 - Acceleration of the vehicle.

The resistive force against the vehicle's motion was comprised of the combination of air resistance, gradient resistance, and rolling resistance (Figure 28).

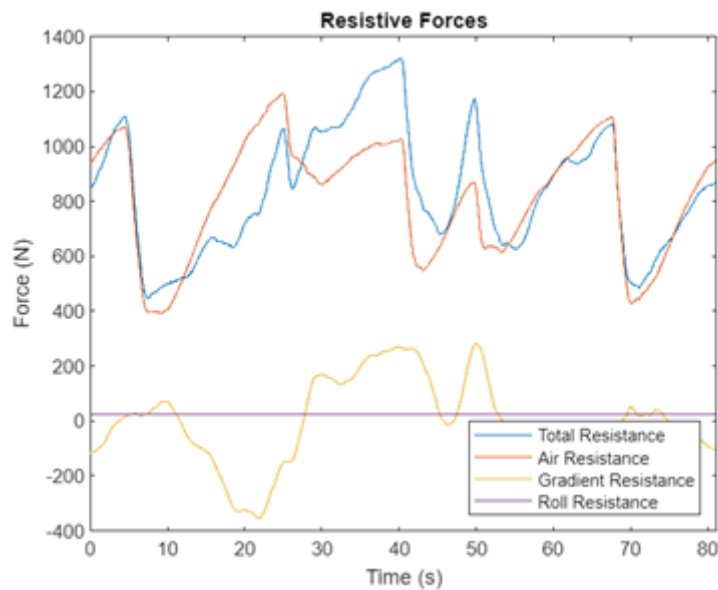


Figure 28- Resistive forces over the lap.

The surplus effort was determined by multiplying the acceleration by the mass of the car and the tractive effort was calculated from the sum of the resistive forces and the surplus effort. Multiplying the tractive effort by the wheel radius gave the torque at the wheel with the maximum wheel torque being 453 N·m.

The torque ratio between engine and wheel was estimated and then narrowed down by comparing the wheel speed to the velocity of the vehicle, this was assumed to be correct once the graphed curves

showed the velocities matched (Figure 29). The torque ratio for 3rd and 4th gears including final drive were found to be around 4.94 and 4.18 respectively.

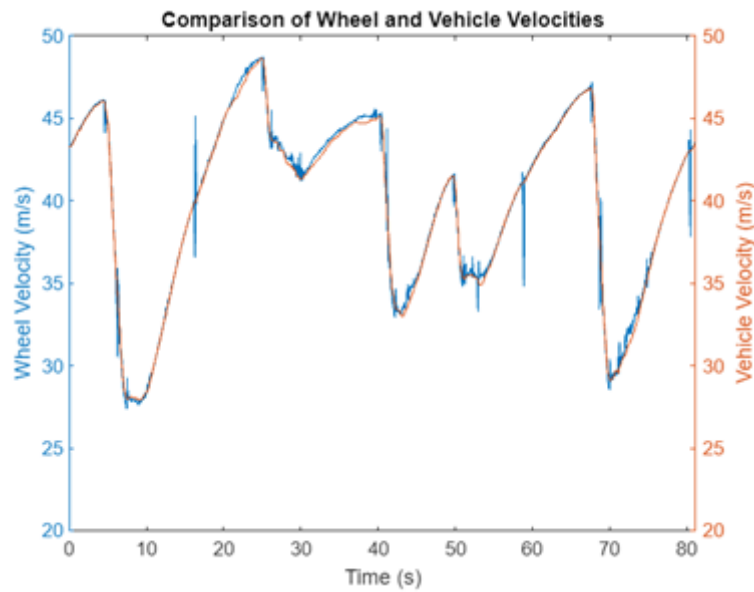


Figure 29 - Comparison of wheel and vehicle velocities.

The torque at the wheel was then multiplied by the angular velocity of the wheel to determine the power output of the engine (Figure 30), with the maximum determined to be 68 kW. Due to drivetrain inefficiencies causing a loss of power between the engine and the wheels, it is likely that the actual power output of the engine is higher than this. This gave a power-to-weight ratio of 138.1 W/kg. The power output was typically between 50 and 68 kW excluding braking zones.

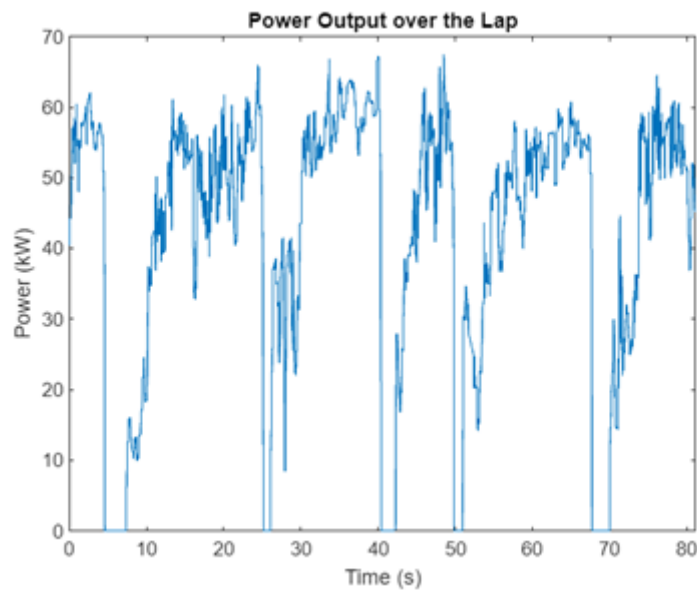


Figure 30 - Power output over the lap

The analysis also provided information regarding energy usage allowing for an estimation of energy usage and potential for energy restoration from braking, used for the selection of batteries.

The work done per lap at Donnington was 3.397 MJ, giving an energy requirement of approximately 50.25 MJ per 20-minute session. It was predicted that the EV vehicle would have a mass of 788 kg, 61% more than the mass of the ICE vehicle. Using this mass, it was found that this additional mass would increase the energy requirement up to 3.972 MJ per lap and 58.77 MJ per session, however the capacity of the rechargeable energy storage system could be reduced through the use of KERS.

6.3.2 Calculating Transmission Ratio

The ratio of the transmission from motor to differential was calculated by comparing the maximum motor speed of 4500 rpm against the highest expected vehicle velocity of 50 m/s. The transmission ratio could be lowered further to maintain motor efficiency in the 96% zone at higher vehicle speeds as it is likely to exceed the limit of grip of the tyres. The maximum transmission ratio required was then calculated:

$$C_{Wheel} = \pi * D = 0.52\pi \text{ m}$$

$$rpm_{Output} = \frac{60 * V_{max}}{C_{Wheel}} = \frac{60 * 50}{0.52\pi} = 1836.4 \text{ rpm}$$

$$Transmission \text{ Ratio} = \frac{rpm_{Input}}{rpm_{Output}} = \frac{4500}{1836.4} = 2.45$$

This determined the ratio to be 2.45 or lower to maintain the maximum speed of 50 m/s.

With the optimal transmission ratio determined, Gates Design Power (Gates Design Power, 2023) was used to assist in the selection of a belt drive system. The input parameters were a motor speed of 4500 rpm, 100 kW power, and centre distance of 350mm $\pm$ 10%, the target output speed was 1836.4 rpm $\pm$ 10%. The variances were used to find products that fell outside of the initial specification but would still function well when implemented. The target rpm did not have a buffer as 50 m/s was the minimum top speed required to meet the target criteria (Figure 31).

The selected drive belt was a Poly Chain GT Carbon 8MGTC-1200-36, which is a synchronous belt with an 8mm pitch (Gates, 2019). The belt would be used in conjunction with a 32-tooth sprocket mounted to the motor side(sp-spareparts.com, 2020), and a 75-tooth driven sprocket (sp-spareparts.com, 2020) attached to the differential, giving a ratio of 2.34. This would reduce the torque output but allow a faster output speed of 1923 rpm and the vehicle's velocity to reach 52.36 m/s at a motor speed of 4500 rpm. The belt tension force under 100 kW operation would be approximately 6670 N.

Prod Line	Belt	Driver# (Teeth)	Driven# (Teeth)	dN RPM	Δ RPM	ODR	Ref. Cost	CD (mm)	Δ CD (mm)	Belt Pull (N)	$\uparrow$ Width (mm)
Poly Chain GT Carbon	8MGTC-1200-36	32	75	1,920.00	83.00	0.00	1.01	382.07	32.07	6669.87	46.31

Figure 31 - Results from the Gates Design Software

One example of a belt driven EV is the Zero SR/S Z-F electric motorcycle. It has a peak torque of 190N·m and peak power of 82 kW and uses a 14mm polychain HTD carbon belt with a 4.5 ratio to drive the rear wheel (zeromotorcycles.com, 2023). It has a combined curb mass and carrying capacity of 454 kg spread over 2 wheels. As the weight per wheel is similar to the expected weight per wheel of the Formula 750 and the peak torque of the Formula 750 is approximately double, the size of the chosen belt was reasonable.

6.2 Part Selection

6.2.1 Controller

The controller in an electric vehicle links the traction battery to the motor, supplying the motor with power by using pulse width modulation to control the motor's speed. Input pins are used to provide the controller with information on the temperature, speed and position of the motor through the use of sensors, such as thermistors and encoders. The driver can interact with the controller by using devices such as switches or pedals to make a request to the controller (Idaho National Laboratory, no date).

The controller used in the vehicle is a Sevcon Dragon8 Gen 4 size 10 motor controller. It is capable of receiving an input between 128 and 700 V DC, at up to 200 A continuously, 400 A for 120 seconds, and 450 A for 30 seconds (Sevcon, no date). The controller begins to limit current once it reaches a specified temperature (Figure 32) in order to prevent damage from overheating, therefore removal of heat from the controller is essential to maximising power output.

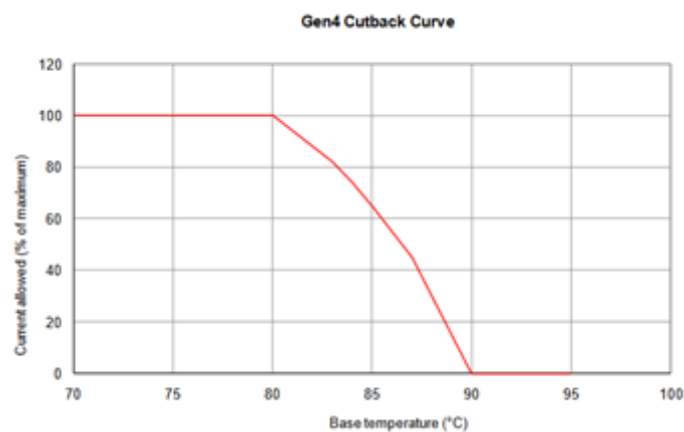


Figure 32 - Cutback curve limits the current output based on temperature (Sevcon, no date).

The torque of the motor at low speeds can be limited through the controller in order to give the driver more fidelity around the point at which the vehicle loses traction. The behaviour of the controller can be set up and configured using the DVT configuration software, allowing for the mapping of the accelerator pedal as well as defining whether the application of the pedal will request torque or speed (Sevcon, no date).

6.2.2 Wiring Layout

The planned wiring layout for the controller (Table 2). A 12 V auxiliary battery is used as a low voltage power source. The B- and B+ connect the battery and the controller with contactors, allowing the battery to be disconnected from the circuit.

The encoder used to measure the speed of the motor is the RM44 rotary magnetic encoder which generates 2 sin waves, having a 90 degrees offset between them (*RM44 Rotary Magnetic Encoder*, 2023).

The accelerator pedal being below a certain threshold would trigger KERS, this would slow the car as a parallel system alongside the friction brakes. Having the brakes set in series could increase the energy restored, requiring less batteries, but could also lead to unstable braking behaviour during the crossover from KERS to friction brakes at the blending point.

Table 3 - Wiring Plan

Pin	Device	In/Out	Additional Information
1	Key Switch	-	Supplies power from 12V battery
2	Contactor Out	Out	Provides voltage to low side of contactor to open it, allowing power flow from battery.
3	Contactor Out	Out	
4	Contactor Out	Out	
5	Accelerator Potentiometer	In (Analogue)	Used for torque request pedal. Provides a reading of 0 to 9.5 V with a 0.5 V buffer from 9.5 to 10 V for safety reasons.
10	CAN Ground	-	
11	CAN High	In/Out	Used to communicate with other devices over CANbus.
12	CAN High	In/Out	
13	Control 0 V	-	Ground connection for the low voltage circuit
14	Reverse Switch	In (Digital)	
19	Encoder - SIN	In (Analogue)	Each rotation of the motor creates a 5 V pulse on SIN and COS wires. The frequency of these pulses tells the controller the speed of the motor.
20	Encoder - COS	In (Analogue)	
21	Encoder Supply	-	5 V supply to the encoder
22	CAN Low	In/Out	Used to communicate with other devices over CANbus.
23	CAN Low		
24	Protected key-switch	-	Closes contactor to supply power from traction battery to motor.
25	Forward Switch	In (Digital)	
26	Foot Switch	In (Digital)	A button that must be pressed for the accelerator pedal to function.
28	Motor Thermistor	In (Analogue)	Provides the controller with information on the motor temperature.
29	Accelerator Pedal Supply	Out	Supply voltage for the accelerator pedal potentiometer.
34	CAN Termination		Bridges across the CANbus connection to end it.
35	CAN Supply		Supplies the CANbus with 24 V.

6.2.3 Motor

The motor chosen for the vehicle was an EMRAX 268 medium voltage which is an inner stator, axial flux motor and has a mass of 21.9 kg. Having an inner stator allows for easier liquid cooling of the stator, cooling the motor is vital to ensure the torque output is not limited to prevent overheating.

The motor has a maximum voltage of 650 V DC however regulations regarding EV vehicles in the Motorsport UK bluebook limits this to 450 V DC (Emrax, 2020). Combined with the 450 A maximum current output of the controller, the maximum power output of the motor is 200kW and the maximum continuous power output of the motor is 107 kW.

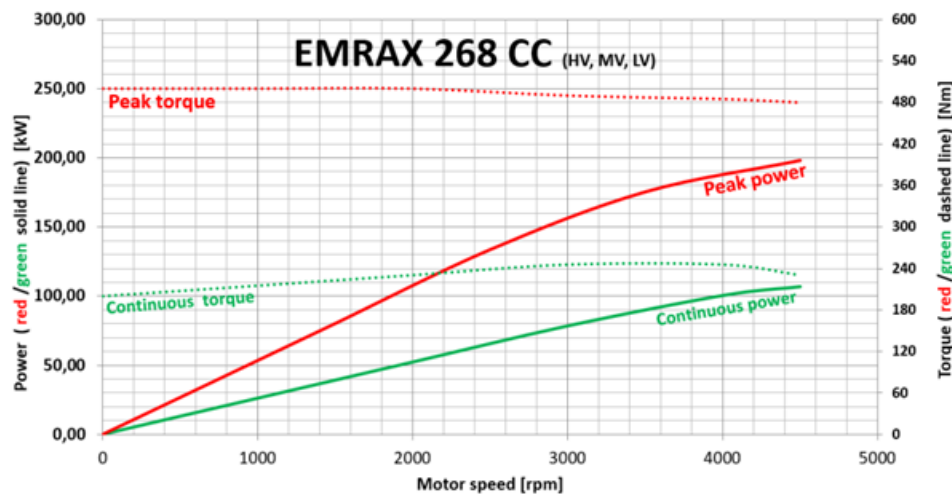


Figure 33 - Emrax 268 CC torque and power map (EMRAX, 2020).

This motor provides 1.3 N·m of torque per Amp rms supplied and can operate with this torque output across the majority of its 4500-rpm range, with a slight drop in torque as it approaches 4500 rpm (Figure 33). It can operate with a supplied current of 200 Amps continuously, producing 260 N·m of torque before accounting for inefficiency. However, the current can be increased up to 400 Amps for 2 minutes, and 450 Amps for up to 30 seconds providing it is cooled adequately (Emrax 2020).

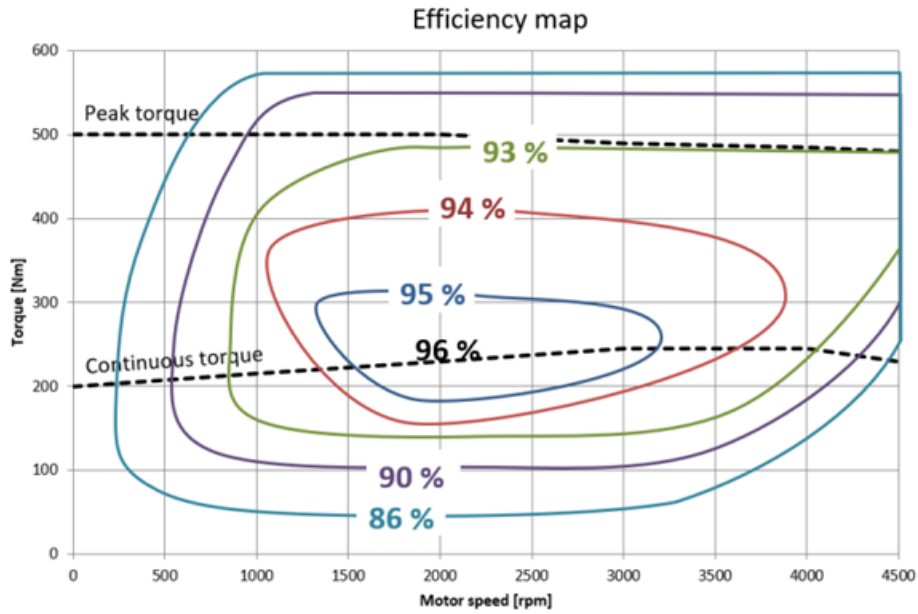


Figure 34 - Efficiency map of emrax 268 motor (EMRAX, 2020).

As the traction battery will be supplying roughly 309.4 V (Drew Gilasbey, 2023) at 200 A continuously, a continuous motor output power of 59.4 kW is possible at 96% efficiency at motor speeds of 1500 to 3000 rpm (Figure 34), between a vehicle velocity of 17.5 m/s and 34.9 m/s. The output power can be increased for short durations by increasing the current to 450 A to achieve a peak power of 125.3 kW at 90% motor efficiency. This power will be slightly reduced due to losses through the transmission.

6.4 Modelling Work

The motor mounting would be particularly important due to the large forces acting on it. To calculate the reaction torque acting through the motor mounts a worst-case scenario was created. The reaction torque was calculated using the entire estimated weight of the car, 7730 N, acting on the rear wheels as a normal force n , and using tyres with a grip coefficient μ of 2, and a transmission ratio of 2.34.

$$F = \mu * n = 2 * 7730 = 15460 \text{ N}$$

$$Torque_{Axle} = F * r = 15460 * 0.26 = 4020 \text{ N} \cdot \text{m}$$

$$Torque_{Motor} = \frac{Torque_{Axle}}{Ratio} = \frac{4020}{2.34} = 1718 \text{ N} \cdot \text{m}$$

The motor mount had to be designed to withstand the torque applied to it for many cycles as a high torque would be applied to it due to harsh acceleration and braking from being used in a racing situation. Bushings could be used between the motor and the frame fixed mount to absorb and mitigate the effects of vibrations on other parts of the vehicle (Kruse and Chauvet, 2020).

The motor mount is made from AISI 1020 square steel tubing due to its good weldability and yield strength. The yield strength of the steel is 294 MPa with an ultimate tensile strength of 394 MPa (Azom.com, 2019). The square tubing is 25 x 25 mm and has a thickness of 3mm. A high carbon steel could be used instead, allowing for a reduction in wall thickness.

The 1st design (Figure 35) of the motor mount was connected to the chassis via the centre longitudinal beam and the centre vertical beam, which would be ideal for mass distribution. This had to be changed once it was apparent that the belt would be in contact with the frame of the vehicle and the motor mount would have to be situated further back. The maximum stress in this design was $5.41 \times 10^7 \text{ N/m}^2$.

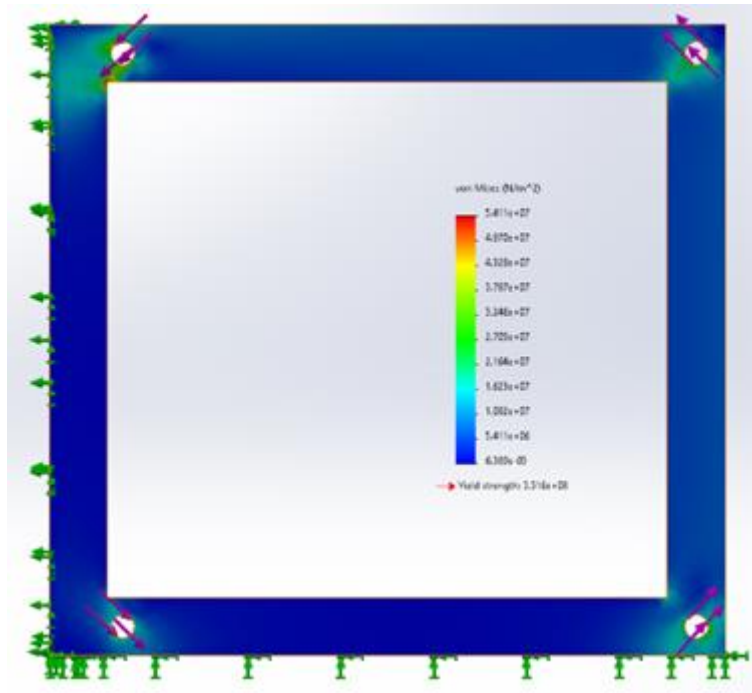


Figure 35 - 1st design of motor mount

Following this, the 2nd design (Figure 36) of the motor mount was created and was only attached via the centre longitudinal beam. Fillets were added to remove the stress concentrations at the inside corners. This design had higher von Mises stresses of up to 2.68×10^8 N/m² due to the lack of support from the vertical centre beam due to the mount having to be located further back in the vehicle.

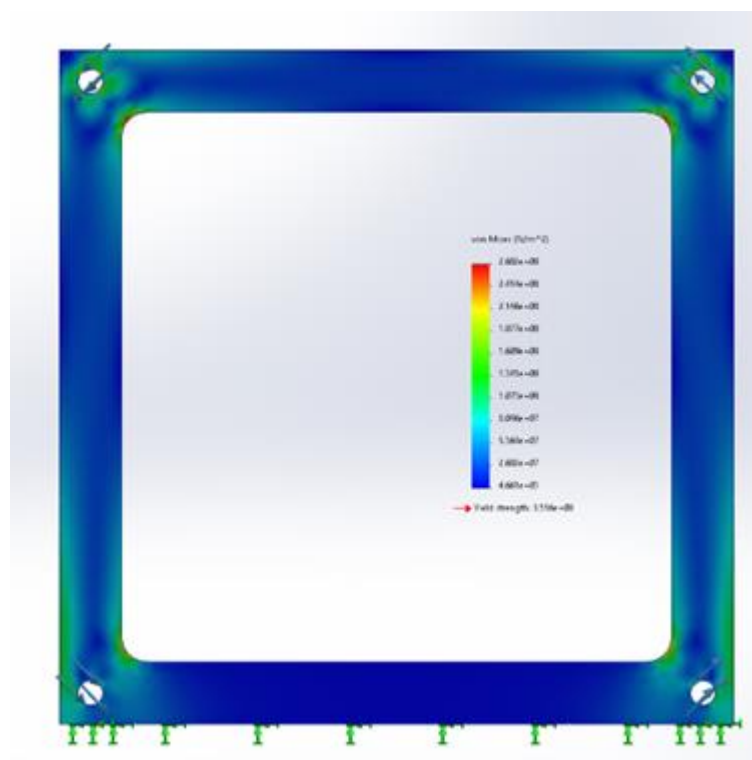


Figure 36 - 2nd design of motor mount

The 3rd design (Figure 37) reconnected the mount to the vertical central beam with a longitudinal beam to spread the load through the vehicle. This lessened the stress concentrations in the corners of the square down to a maximum of $3.88 \times 10^7 \text{ N/m}^2$. This can be seen mounted to the vehicle in Figure 38.

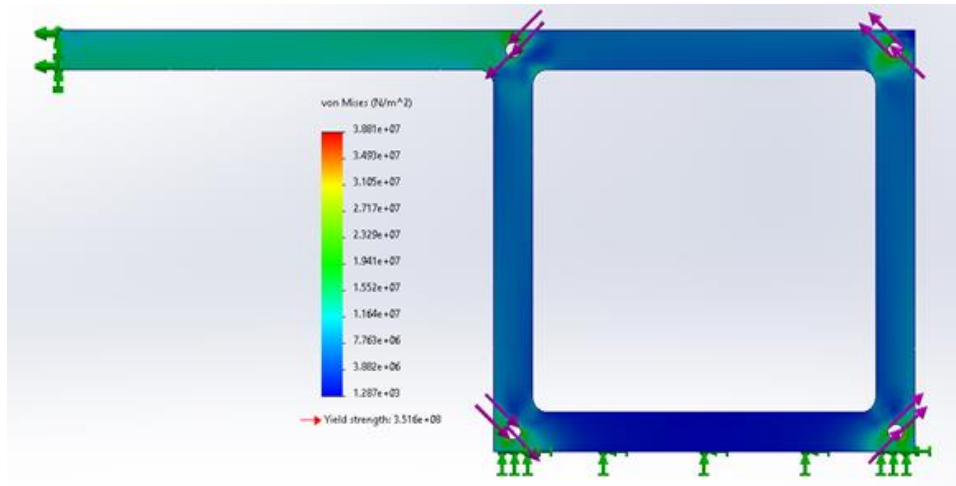


Figure 37 – 3rd design of the motor mount.

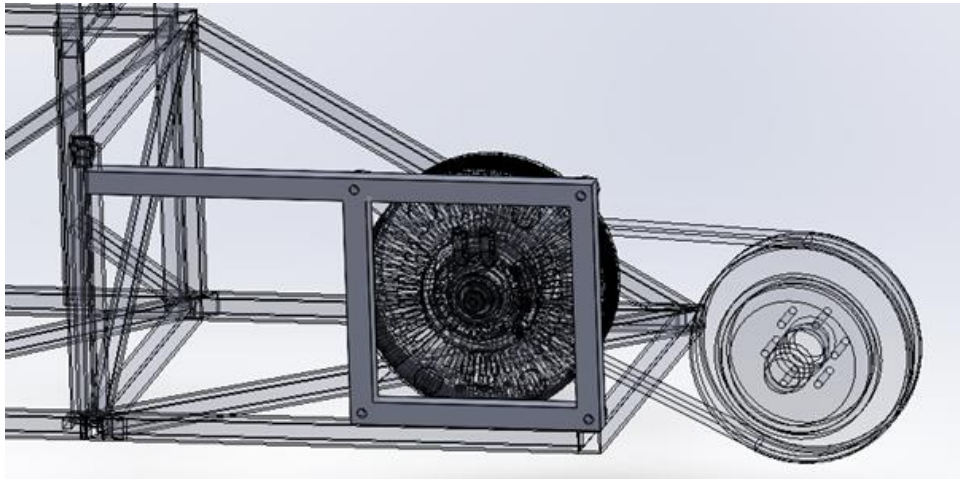


Figure 38 - Motor mount attachment.

6.5 Results

A comparison of the ICE and EV versions of the vehicle shows the targets set at the beginning have been met. The expected mass of 788 kg would give the EV vehicle a power to weight ratio of 159 W/kg. However, the EV can only maintain that power for a limited time depending on heat dissipation.

Table 3 - Comparison of ICE and EV performance

Performance Aspect	ICE	EV
Peak Power (kW)	68	125.3
Peak Engine Torque (N·m)	95	500
Peak Wheel Torque (N·m)	453	1170
Power to Weight Ratio (W/kg)	138.1	159

7 Summary of Work Completed (Evan Craig)

My primary responsibility was to research and evaluate a differential and bearing setup that could be mounted to the chassis. After extensive research, I determined that an open differential would have to be used as the regulations do not allow locked or limited slip differentials. Further research showed the differential would require a bearing setup consisting of tapered roller bearings as they would be the best option. I also calculated the driveshaft lengths so that custom driveshafts can be made to fit the differential. The differential was then mounted to the chassis and connected to the electric motor via a rubber belt.

In order to ensure that the chosen differential and bearing setup complied with the regulations of the racing competition, I reviewed the guidelines and specifications to find suitable components. After a thorough evaluation of several different types of bearings, including ball bearings and cylindrical roller bearings, I determined that a tapered roller bearing setup would be the most appropriate for our project. To ensure that the selected setup was effective, we carried out experiments to test its performance and durability under varying conditions. In addition, I worked closely with the other members of the team to ensure that the driveshaft lengths were accurately measured and custom-made to fit the differential. This involved detailed calculations and precision measurements to ensure that the driveshafts were the correct length and had the necessary strength to withstand the forces generated by the electric motor and differential.

In addition to selecting the appropriate differential and bearing setup, we also had to determine the best location to mount the differential on the chassis. We considered several factors, including weight distribution and available space. After analysing the chassis and taking into account the weight of the electric motor, battery pack, and other components, we determined that the optimal location for the differential was towards the rear of the chassis. Once we had determined the best location for the differential, we were able to calculate the centre distance between the electric motor and the differential. This was necessary to determine the appropriate belt size for the rubber belt that would connect the electric motor to the differential. Overall, selecting and mounting the appropriate differential and bearing setup was a crucial component of the project. Through research and experimentation, we were able to determine the most effective solution that would provide the necessary power and torque to drive the vehicle. Additionally, by carefully considering weight distribution and other factors, we were able to optimize the performance of the vehicle and ensure that all components were working together seamlessly.

To calculate the force generated through the bearings and mount of the differential by the electric motor, several factors needed to be taken into consideration. First, we needed to calculate the effective radius of the tire. Using the peak RPM of the electric motor and the known speed of the vehicle, we determined the effective radius to be 0.855 meters. Next, we converted the torque of the electric motor from Nm to kg/m and the power from kWh to kW. Finally, we used the distance between the electric motor and differential and the efficiency of the rubber belt to calculate the force generated through the bearings and mount of the differential. Our calculations showed that the electric motor would generate a force of approximately 526.06 kg through the bearings and mount of the differential, assuming a belt efficiency of 95%.

After the differential was successfully mounted to the chassis, we connected it to the electric motor via a rubber belt. The belt efficiency was also taken into consideration during the design process to ensure maximum power transfer from the motor to the differential.

7.1 Differential Forces Calculation

If the distance between the electric motor and the differential is 0.45 meters, and the peak RPM of the electric motor is 1600, we can calculate the speed of the vehicle as follows:

$$V = (2 * \pi * R * \text{RPM}) / 60$$

where V is the speed of the vehicle, R is the effective radius of the tire, and RPM is the peak RPM of the electric motor.

Assuming the speed of the vehicle is 180 km/h at the peak RPM of the electric motor, we can calculate the effective radius of the tire as follows:

$$R = (V * 1000 * 60) / (2 * \pi * \text{RPM})$$

$$R = (180 * 1000 * 60) / (2 * \pi * 1600)$$

$$R = 0.855 \text{ meters}$$

Now we can calculate the force generated through the bearings and mount of the differential as follows:

$$\text{Force} = (\text{Torque} / \text{Distance}) / \text{Belt Efficiency}$$

First, we need to convert the torque from Nm to kgm:

$$2200 \text{ Nm} / 9.81 \text{ m/s}^2 = 224.09 \text{ kgm}$$

Next, we convert the power from kWh to kW:

$$100 \text{ kWh} * 1000 \text{ W/kW} = 100,000 \text{ kW}$$

Finally, we can calculate the force generated through the bearings and mount of the differential:

$$\text{Force} = (224.09 \text{ kgm} / 0.45 \text{ meters}) / 0.95 = 526.06 \text{ kg}$$

Therefore, the electric motor with 2200 Nm of torque and a power of 100 kW, connected to the differential via a rubber belt with a distance of 0.45 meters, would generate a force of approximately 526.06 kg through the bearings and mount of the differential, assuming a belt efficiency of 95%.

7.2 Bearing Analysis

There are several types of bearings available on the market, and each one has its unique advantages and disadvantages. One commonly used bearing type is the ball bearing, which is known for its low friction and ability to handle high radial loads. Ball bearings are often used in applications where speed is essential, such as in high-performance engines and turbochargers.

Another type of bearing is the cylindrical roller bearing, which is suitable for heavy radial loads and high speeds. Cylindrical roller bearings are commonly used in high-performance engines and racing cars because they can withstand high loads and operate at high speeds. However, they are not as effective as tapered roller bearings when it comes to handling axial loads.

Tapered roller bearings, on the other hand, are designed to handle both radial and axial loads. They are commonly used in automotive applications where high radial and axial loads are present, such as in the differential setup of racing cars. Tapered roller bearings provide superior load-carrying capacity and are more reliable than other bearing types under high-speed and high-load conditions.

In racing applications, the selection of the appropriate bearing is crucial as it can affect the overall performance and reliability of the vehicle. Tapered roller bearings are often the preferred choice for racing applications due to their ability to handle high radial and axial loads and operate at high speeds. Their superior load-carrying capacity and reliability make them suitable for use in critical areas such as the differential, where high loads and speeds are present. Additionally, tapered roller bearings are less prone to failure and require less maintenance than other bearing types, making them an ideal choice for racing applications where reliability and performance are essential.

7.3 Driveshaft Analysis

Calculating and measuring the driveshaft length was a crucial step in the project. After selecting the appropriate differential and bearing setup, I needed to determine the length of the driveshaft that would connect the differential to the wheels. This involved taking precise measurements of the distance between the differential and the wheels, as well as accounting for the angle at which the driveshaft would be mounted. To accurately calculate the length of the driveshaft, I also had to consider any modifications that had been made to the chassis, such as changes to the suspension or steering system. This was necessary to ensure that the driveshaft was the correct length and could handle the amount of torque and power that would be generated by the electric motor. Once I had determined the appropriate length for the driveshaft, I also had to consider the type of material that would be used to construct it. Carbon fibre was ultimately chosen for its lightweight and high-strength properties, which are ideal for racing applications. In addition to selecting the appropriate material, I also had to ensure that the driveshaft was balanced to prevent any vibration or wobbling that could affect the performance of the vehicle.

8 Project Review

The purpose of the project was to plan the adaptation of a formula 750 race car from ICE to EV. The result of the project was partly successful, as we determined it may be possible to create an EV adaptation of the formula 750 with similar performance. However, the project fell behind the schedule detailed in the Gantt chart plan made at the start of the project (Figure 39) which led to milestones being missed. This was partly due to not having prior experience in planning projects, overlooking aspects of the development, and members of the group leaving the project.

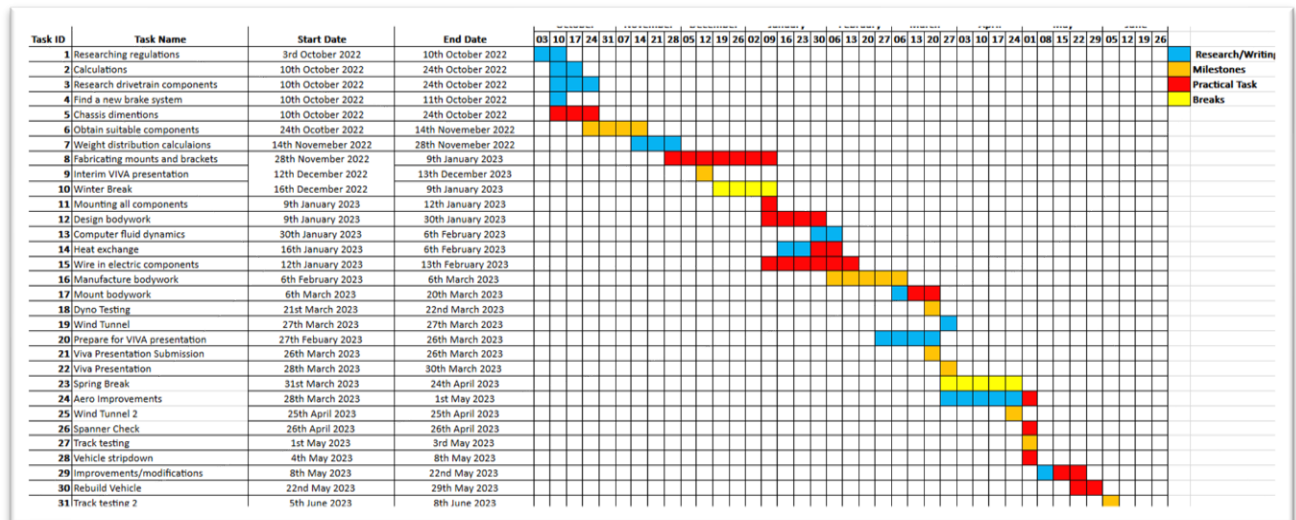


Figure 39 – Gantt chart detailing the steps of the project.

Due to some members of the team leaving the project, the group's morale and motivation towards the project was affected, as well as creating uncertainty as to which members would cover the additional workload. This led to the remaining group members taking on additional tasks which resulted in less time being available for their specified roles. This resulted in slow progress as each remaining group member was stretched thinly between multiple roles on the project and could not solely put effort into one single task. This also resulted in a lower quality of the work produced than was expected for the same reason.

In an ideal world the project would have been given extra support in the form of additional team members in order to pick up where the vacant team members left off. This would have kept everything on schedule and would have progressed the work much further overall. Additionally, the remaining group members work would have been able to be much more thorough as they'd be fully focused on their designated tasks.

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10 Appendices

Appendix 1 - MATLAB script pt.1

Formula 750 Drivecycle

Read Data from Excel

```
clc, clear all
```

Comment out the track not being used.

Anglesey Wet

```
% Time = readmatrix('Qualifying Anglesey Wet.csv', 'Range', 'A18:A8416');
% V_kmh = readmatrix('Qualifying Anglesey Wet.csv', 'Range', 'D18:D8416');
% V_ms = V_kmh/3.6;
% E_rpm = readmatrix('Qualifying Anglesey Wet.csv', 'Range', 'C18:C8416');
% Distance = readmatrix('Qualifying Anglesey Wet.csv', 'Range', 'B18:B8416');
% Gear = readmatrix('Qualifying Anglesey Wet.csv', 'Range', 'H18:H8416');
% Gear = floor(Gear); % Rounds down the gear

% n = size(Time,1); % size of the time matrix
```

Donington

```
Time = readmatrix('Donington Qualifying.csv', 'Range', 'A18:A8416');
Velocity_kmh = readmatrix('Donington Qualifying.csv', 'Range', 'D18:D8416');
Velocity_ms = Velocity_kmh/3.6; % Convert velocity to m/s
EngineRPM = readmatrix('Donington Qualifying.csv', 'Range', 'C18:C8416');
Distance = readmatrix('Donington Qualifying.csv', 'Range', 'B18:B8416');
Slope = readmatrix('Donington Qualifying.csv', 'Range', 'U18:U8416');
Gear = readmatrix('Donington Qualifying.csv', 'Range', 'G18:G8416');
Gear = floor(Gear);

n = size(Time,1); % size of the time matrix
xtime = n/100; % time used for graph x axis
```

Process Data

Smooth Velocity

```
VelocityAdjusted = movmedian(Velocity_ms, 40); % smooth out velocity to remove peaks

figure('Name', 'Velocity Smoothed')
plot(Time, Velocity_ms, Time, VelocityAdjusted)
title 'Comparison of Raw and Filtered Velocity'
legend('Raw', 'Adjusted')
ylabel('Velocity (m/s)')
xlabel('Time (s)')
xlim([0 xtime])
max(VelocityAdjusted)
```

Calculate Acceleration

```
dV = gradient(VelocityAdjusted);
dT = gradient(Time);
Accel = dV./dT;
Accel = movmedian(Accel, 40);
max(Accel)

figure('Name', 'Acceleration')
plot(Time, Accel)
title 'Acceleration over the lap'
ylabel('Acceleration (m/s^2)')
xlabel('Time (s)')
xlim([0 xtime])
```

Distance Travelled

Verifying the calculated distance travelled is similar to the actual track length

```
Dist = cumsum(VelocityAdjusted * (Time(2) - Time (1)));
LapDist = max(Dist) % Distance of the lap in metres
```

Specifications

```
CarMass = 488; % Mass of the car in kg
FrontalArea = 1.2; % Frontal area of the car in m^2
CoDrag = 0.7; % Coefficient of drag
CoRR = 0.005; % Coefficient of roll resistance
CoGrip = 1.5; % Coefficient of grip between tyre & track
WheelRadius = 0.26; % Wheel radius in m

I_wheel = 2.7; % Polar moment of inertia of wheels & axles in kgm^2
I_Prop = 0.05; % Polar moment of inertia of propshaft
I_Engine = 0.7; % Polar moment of inertia of engine

G3_Ratio = 1.3;
G4_Ratio = 1.1;
```

Appendix 2 - MATLAB script pt.2

```
FD_Ratio = 3.8;
eff_FD = 1; % Final drive efficiency
eff_G = 1; % Gear efficiency

AirDensity = 1.202; % air density kg/m^3
g = 9.81; %gravitational acceleration

FuelEnergy = 44000000; % Calorific value of 1 litre of petrol in J
EngineEfficiency = 0.3; % Efficiency of Engine generating work from fuel
```

Resistance Calculations

Air Resistance

```
AirResistance = 0.5 * AirDensity * FrontalArea * CoDrag * (VelocityAdjusted.^2);
```

Roll Resistance

```
CarWeight = CarMass * g;
RollResistance(1:n) = CoRR * CarWeight;
RollResistance = RollResistance';
```

Gradient Resistance

```
GradientResistance = CarWeight * sin((Slope - 0.076046)*pi/180); % Gradient resistance calculation with an adjustment to account for
GRTotal = (cunsun(CarWeight * sin((Slope - 0.076046)*pi/180))); % Changing the number following slope to get a zero for the slope, na
GRTotal(1)
GRTotal(8113)
figure('Name','GradientResistance')
plot(Time,GRTotal)
xlim([0 xtime])
```

Inertia Resistance

```
GearRatio = 0;

% Creating a matrix of which gear ratio is in use at which time
for i = 1:n
    if Gear(i) == 2
        GearRatio(i) = G2_Ratio;
    elseif Gear(i) == 3
        GearRatio(i) = G3_Ratio;
    elseif Gear(i) == 4
        GearRatio(i) = G4_Ratio;
    else
        GearRatio(i) = 0.001;
    end
end
GearRatio = GearRatio';

% Find ratios by matching distance wheel covers in 1 second to
% velocity recorded by gps assuming no loss of traction, adjusting FD and
% gear ratios until the velocities are equal (or close enough).
WheelRPM = EngineRPM ./ (GearRatio * FD_Ratio); % Wheel rpm determined by gear and fd ratio
WheelVelocity = WheelRPM * pi * 2 * WheelRadius / 60; %Distance rolled by the wheel per second

figure('Name','Wheel Velocity vs Car Velocity comparison')
title('Comparison of Wheel and Vehicle Velocities')
xlabel('Time (s)')
yyaxis left
plot(Time,WheelVelocity)
ylabel('Wheel Velocity (m/s)')
ylim([20 50])
xlim([0 xtime])
yyaxis right
plot(Time,VelocityAdjusted)
ylabel('Vehicle Velocity (m/s)')
ylim([20 50])
```

Combined Resistance

```
TotalResistance = AirResistance + RollResistance + GradientResistance; % Total Resistance
TotalResistance(TotalResistance < 0) = 0;

figure ('Name','Resistance Graph')
title 'Resistances encountered during the Lap'
plot(Time, TotalResistance, Time, AirResistance, Time, GradientResistance, Time, RollResistance)
legend('Total Resistance', 'Air Resistance','Gradient Resistance','Roll Resistance', 'Location', 'southeast')
xlim([0 xtime])
title('Resistive Forces')
ylabel('Force (N)')
xlabel('Time (s)')
```

Tractive Effort

```
TractiveEffort = (CarMass * Accel) + TotalResistance;
```

Appendix 3 - MATLAB script pt.3

Torque

```
WheelTorque = (TractiveEffort * WheelRadius); % Nm
WheelTorque(WheelTorque < 0) = 0;
WtMax = max(WheelTorque) % maximum wheel torque Nm
```

Calculate engine torque without accounting for inefficiency of the drivetrain

```
EngineTorque = WheelTorque ./ (GearRatio * FD_Ratio); % Nm
ETMax = max(EngineTorque)
```

Power

```
WheelAngVelocity = 2 * pi * (VelocityAdjusted / (pi * 2 * WheelRadius)); % Angular Velocity of Wheel rad/s

Power = WheelTorque .* (WheelAngVelocity) * eff_G * eff_FD;
Power(Power < 0) = 0;
PowerMax = max(Power);

fprintf('Maximum Power (kW) = %d kW',cell(PowerMax/1000))
plot(Time,Power/1000)
xlim([0 xtime])
xlabel('Time (s)')
ylabel('Power (kW)')
title('Power Output over the Lap')

PowerWeightRatio = PowerMax/CarMass % W/kg
fprintf('Power to Weight ratio: %d hp/kg', PowerWeightRatio*0.00134102)
```

Surplus Effort

```
SurplusEffort = TractiveEffort - TotalResistance; % Tractive Effort minus resistances
SurplusEffortPositive = SurplusEffort;
SurplusEffortPositive(SurplusEffort < 0) = 0;
figure('Name','SE graph')
yyaxis left
plot(Time,SurplusEffortPositive)
yyaxis right
plot(Time,Accel)

xlim("auto")
ylim([-2 5])
```

Braking Force

```
BrakeForce = -SurplusEffort;
BrakeForce(BrakeForce < 0) = 0; % Only braking when negative SE is less than 0
figure('Name','BrakeForce')
yyaxis left
plot (Time,VelocityAdjusted)
ylim([0 70])
title 'Velocity and Braking Forces'
yyaxis right
plot(Time,BrakeForce)
BrakeDecel = BrakeForce / CarMass
plot(Time,BrakeDecel)
```

Work Done

```
WorkDone = TractiveEffort .* (VelocityAdjusted * (Time(2) - Time (1))); % Work Done in J
WorkDonePositive = WorkDone;
WorkDonePositive(WorkDonePositive < 0) = 0; % Remove negative work done as this
WorkDoneCumulative = cumsum(WorkDonePositive); % J
WorkDoneTotal = max(WorkDoneCumulative) % J

WorkDoneRace = WorkDoneTotal * (1200/Time(n))
figure('Name','WorkDone')
yyaxis left
plot (Time, WorkDone)
yyaxis right
plot(Time, VelocityAdjusted)
title('Work Done over the Lap')
title ('Cumulative Work Done over the Lap')
%Checking if the result is realistic
FuelUsage = WorkDoneTotal/(FuelEnergy * 4.544 * EngineEfficiency) % How many litres per lap in gallons
MPG = (LapDist / 1600) / (FuelUsage) % Lap distance to miles divided by fuel used in gallons to give MPG
```

Kinetic Energy

```
KE = 0.5 * CarMass * VelocityAdjusted .* VelocityAdjusted;

figure ('Name','Kinetic Energy and Velocity Graph')
yyaxis left
plot (Time, KE)
```