Analysis of different powertrain configurations for a formula style electric race car

By Jan Badal

LUND UNIVERSITY

Master Thesis

Main Supervisor: Francisco J. Márquez – Fernández
Main Industrial Supervisor: Filip Kronstrom
Examiner: Martin Tunér

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LUNDS UNIVERSITET
Lunds Tekniska Högskola

Abstract

The aim of this thesis is to provide useful framework for the design of the upcoming electric race car of Lund Formula Student Team. The thesis intends to find the different powertrain concepts on the state of the art. From the configurations, the thesis should provide outcomes of the performance, efficiency, complexity design and cost. Furthermore, the best concept should be find and a simple preliminary design is made.

To compare the different concepts developed, a Matlab code was used, which simulates the vehicle dynamics of the race cars. A Simulink model was be used to analyse the different electric systems and come up with the most efficient solution.

The results of the thesis show that the powertrain configuration that should perform better in a real competition is the design with four motors actuating one in each wheel. The reason behind it, is the abilty of the system to provide different torque at each wheel, known as torque vectoring. By distributing different torque at each wheel the race car is able to create a yaw movement to the body, allowing it to make turns at a higher velocity.

The design shows the different parts composing the powertrain, and how each of the parts was chosen. To conclude the thesis, the four motor's configuration is compared to the LFS20 design in order to explain how this powertrain improves the car results in the overall competition.

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Nomenclature

FS (Formula Student)

LFS20 (Lund Formula Student 2020)

ICE (Internal Combustion Engine Car)

RWD (Rear Wheel Drive)

FWD (Front Wheel Drive)

AWD (All Wheel Drive)

TV (Torque Vectoring)

PMSM (Permanent Magnet Synchronous Motor)

RPM (Rounds Per Minute)

SoC (State of Charge)

DoF (Degrees of Freedom)

G (Gravity constant)

1 Introduction

1.1 Formula Student

Formula Student also known as Formula SAE, is an international competition which aims to inspire and develop young engineering students from all around the world and challenges them to compete to build a single-seat racing car to compete in static and dynamic events, testing its reliability and performance.

The competition prepares undergraduate and graduate engineering students in a variety of disciplines for future employment in mobility-related industries by challenging them with a hands-on, team engineering experience that also requires budgeting, communication, project management, and resource management skills. Students also gain valuable exposure with recruiters from leading companies in the mobility industry, to help land their first engineering job after graduation. **[1]**

The development of the race-car requires substantial work, hence competing universities hold a department where the students from the team work during eight to twelve months to have the car ready for the competitions, normally hold during summer.

Lund Unversity's team has been in the competition since 2006. It started by the name of LURacing, becoming later on Lund Formula Student team. The team had their biggest success on 2010, placing 2nd in the UK competition, and it has had a good run since then. **[2]**

The competition is partnered with the SAE International which is a globally active professional association and standards developing organization for engineering professionals in various industries. SAE international holds the Formula Student Competition among others. Inside the Formula SAE, there are three possible competitions where the teams can sign to, depending on the type of car: Formula Hybrid (hybrid car), Formula SAE (combustion car) and Formula SAE electric (electric car). **[3]**

At the moment the team is stepping up for next year's competitions, and has the aim to compete with both an internal combustion engine vehicle, and an electric one. The electric race car has been developed from scratch, and will be ready for 2020's competition.

The competition holds many events with an evaluation criterion in order to select the winning team. There is a jury which awards a maximum amount of points depending on the event. There are eight events, three static and five dynamics. The relative importance of each event in the whole competition and the points awarded by these are shown in the above figure.

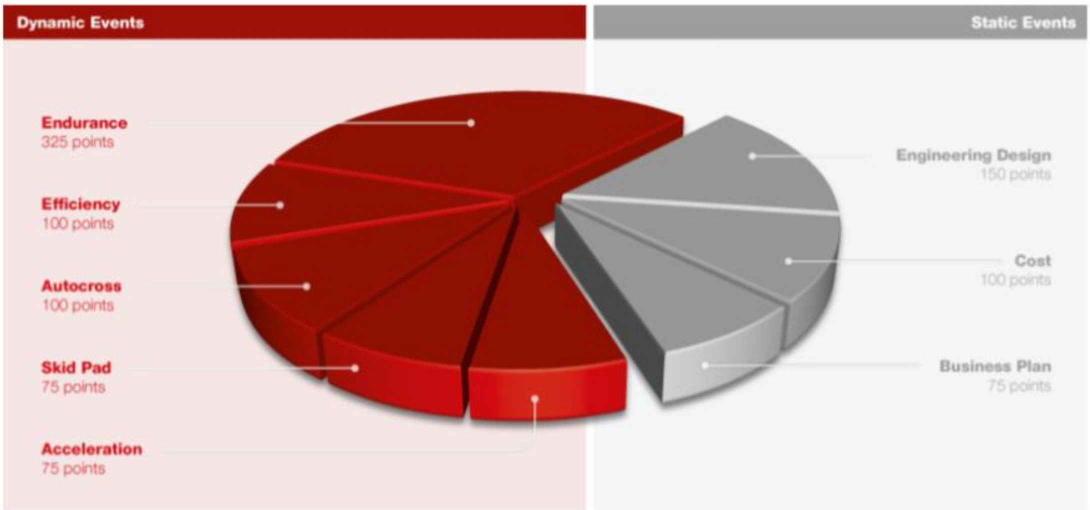


FIGURE 1*:* FORMULA STUDENT COMPETITION EVENTS **[4]**

Endurance (352 points): It consists of a 22km race over 20 laps with a pit stop and driver change at 11 km. This is the most fearful competition for the teams since it requires a high degree of reliability on the developed vehicle fully complete the race. Finishing this race is already considered a win for most teams.

Efficiency (100 points): The efficiency scoring is based on the energy the car consumes during the Endurance race.

Autocross (100 points): This is probably the hardest dynamics event to succeed on. It is a one lap track. It clearly shows which car has the best pace, although the driver also plays a significant role in this race.

Skid Pad (75 points): In this event, the car has to run a full right hand circle and then a full left hand circle. The average time between the two circles is then used to describe the vehicles steady state cornering performance

Acceleration (100 points): This 75 meter's acceleration race shows up the car with the highest acceleration.

Engineering Design (150 points): This is the most prestigious static event. In consists of a 40 minutes' presentation, explaining the whole design process in detail. Winning this event proves that the team has the highest engineering knowledge, and so placing high is really important for the team.

Cost (100 points): The teams are constrained to document the cost of every part of the car, both the parts which are manufactured and ones coming from an external provider.

Business Plan (75 points): The business plan event consists on a 10 minutes' presentation where the teams are to convince a jury of potential investors to invest on the idea of manufacturing 1000 units of the car per year.

1.2 Methodology

To properly show the methodology used in this master thesis, the following graph was developed, which summarizes the whole process:

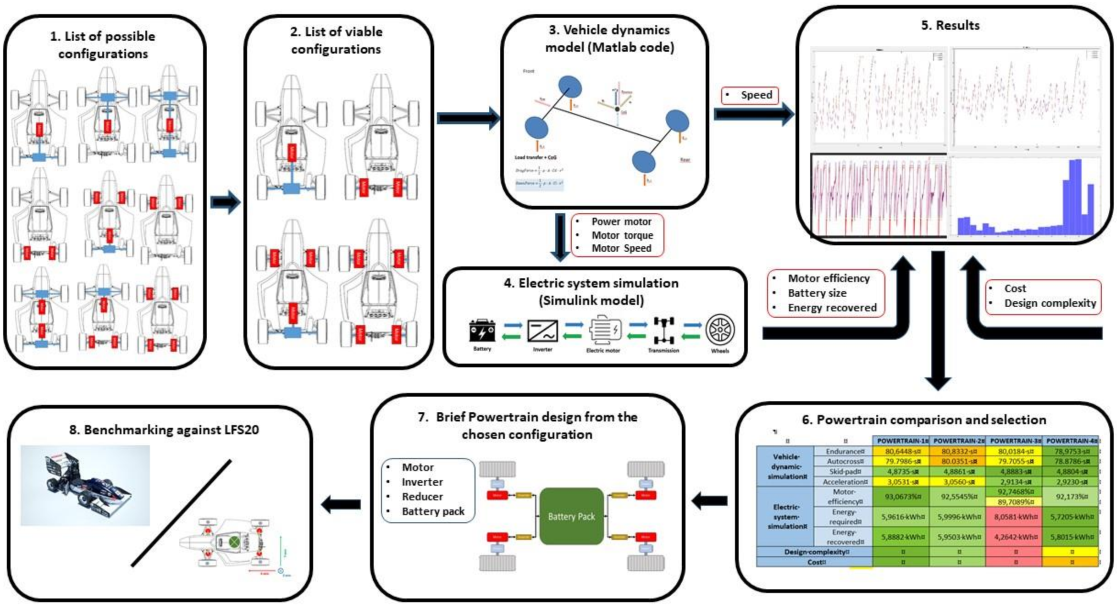


FIGURE 2: *METHODOLOGY OF THE MASTER THESIS*

The first step of this work was to explore the different possibilities that an electric car may offer in terms of the powertrain. Based on both the existing literature from a renown source like SAE International (Society of Automotive Engineers) and the previous experience of Lund Formula Student team, several powertrain configurations have been identified and are considered for the analysis.

Once the number of interesting configurations have been identified, it should be considered if they can be applied to the electric race car from the team. Since this thesis does not include the vehicle as a whole but only the powertrain, the car designed by the team for 2020 will be the starting point, and so the different powertrains considered must be able to fit in it.

Afterwards, the dynamic performance of a vehicle equipped with each of the remaining feasible configurations are simulated. A simulation program in Matlab, written by a member of the Lund Formula Student team is used as the starting point. The code has been adapted in order to model each of the different proposed configurations. This code is required to be able to simulate each of the powertrains in the different races of a FS competition. The main output of the simulation will be the speed, which is used to identify the fastest configuration.

Furthermore, during one of the events of the competition, the efficiency of the vehicle is tested. To properly assess how each powertrain would perform in this event, a second model will be used in order to analyses how efficient is each powertrain. This model will be developed using Simulink. It will use the outcomes of the vehicle simulation code of the power's motor, torque's motor and speed's motor. The efficiency will be compared through the three main outputs of the model: the motor efficiency, the energy used to complete the race and the energy recovered through the race.

Besides the performance evaluation obtained from the simulation models, there are some aspects of each proposed solution that should be evaluated qualitatively in order to have a more comprehensive idea of their potential. These aspects include: design complexity and cost

After comparing all the results for the different viable solutions, the best configuration is chosen and a preliminary design will be provided. This design will include a quick overview of all the components included of the powertrain. For each, a specific component suggestion is made.

As a conclusion to the work, the chosen powertrain solution is benchmarked against the LFS20 in order to highlight the improvements that the proposed solution offers in comparison to the design made by the team for the 2020 competition.

1.3 Goals

The main objectives of this MSc. thesis are:

1) Provide the set of electric powertrain concepts that can fit in an electric race car
2) Provide the set of possible powertrain configurations which could fit in the Lund's Formula Student electric car
3) Build up a software model which allows to compare different powertrain configurations in terms of vehicle dynamics
4) Build up a software model which allows to compare different powertrain configurations in terms power related issues
5) Compare the different solutions, and choose the one with the best outcome
6) Provide a preliminary design from the chosen solution above.
7) Benchmarking between the chosen design and the LFS20 electric car

2 Powertrain configurations

2.1 Prior considerations

A powertrain of a vehicle comprises the main components that generate power and deliver it to the road surface. The main difference between a powertrain and a driveline is that the first one also includes the engine. **[5]**

Combustion cars have a powertrain consisting of an internal combustion engine (ICE) attached to a differential, which provides torque to a shaft or shafts. The shaft delivers the power to the wheels and these to the road surface. Depending on the shaft or shafts at which the motor delivers the power, there are three possible configurations for ICE cars: rear wheel drive, front-wheel drive or all-wheel drive cars which is the same as four-wheel drive.

Internal combustion engines, produce usable torque and power output only within a limited speed range. Moreover, these type of motors are only efficient in a small portion of speed range. Therefore, ICE requires of an attached gearbox that allows the car to run in a wider speed range.

On the other hand, electric motors may not need a reducer, and work efficiently on a higher speed range. By removing the gearbox, the transmission system becomes simpler, thus new opportunities for powertrain configurations come in hand.

This is especially important in a race scenario. Electrified powertrains enable configurations with various motors. The electric motors can be directly attached to the wheels and with a proper control system, the car is able to apply different torque at the wheels from each side of the car. By doing so, the vehicle is able to create a yaw movement without need of steering on the wheels. This phenomenon is called torque vectoring, and it opens up the possibility for faster configurations than the standard ones.

2.2 List of the possible configurations

The configurations explained in the following part are all the possibilities that may be considered for a race car. However, these designs are to be introduced in the Lund's Formula Student electric car, without the need of fully changing the Lund's car design. If the configuration is not adaptable for the LFS20, the powertrain configuration is considered as non-applicable, thus no further analysis is made on it.

1. One motor. Rear wheels driven.

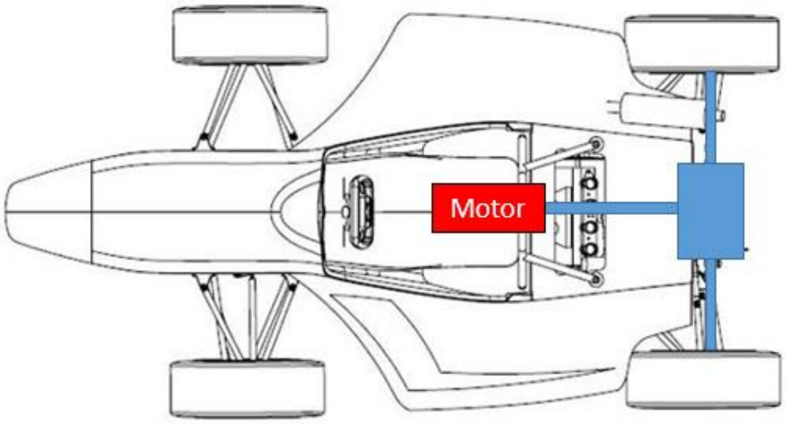


FIGURE 3: *REAR WHEELS DRIVEN POWERTRAIN CONFIGURATION* (*CAR DRAW TAKEN FROM* **[6]**)

This is the typical configuration for an ICE car with RWD. This configuration is the one being used in the next year's race car (LFS20 powertrain design). There exists a full design for this powertrain already, from the electric Lund Formula Student team. **[7]**

However, it is interesting to keep this configuration for further analyses to compare the current design from the team to the potential alternatives **Accepted**

2. One motor. Front wheels driven.

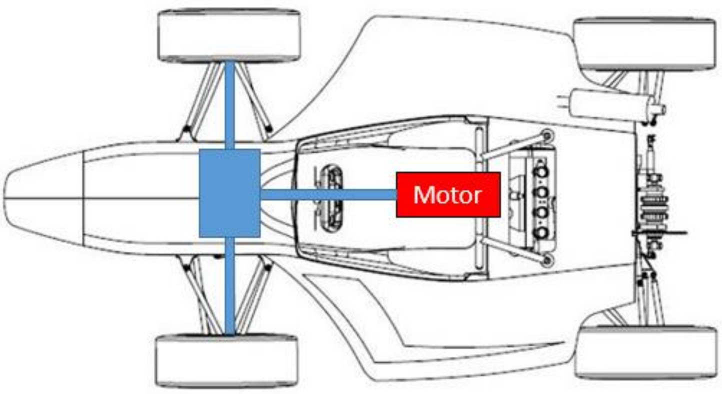


FIGURE 4: *FRONT WHEELS DRIVEN POWERTRAIN CONFIGURATION*

This is the typical configuration for an ICE car with FWD. **[8]**

This is an impossible design to be integrated in the current Formula Student electric car, since there is no room in the front part of the vehicle for a front shaft and a differential. **Eliminated**

3. One motor. All wheels driven.

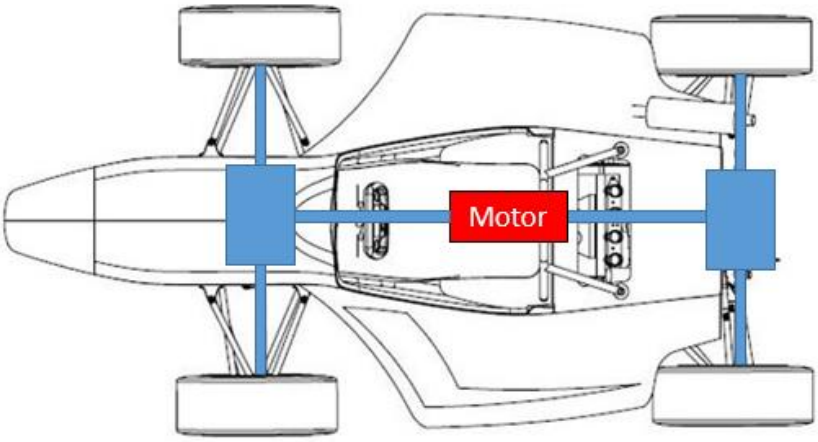


FIGURE 5: *ALL WHEELS DRIVEN POWERTRAIN CONFIGURATION*

It is the typical configuration for an ICE car with AWD or 4WD also called 4x4. This type of car allows the motor to transfer torque to both the front and rear shafts. This torque distribution enables the car to drive on more challenging roads than other car drivelines.

This is an impossible design to be integrated in the current Formula Student electric car, since there is no room in the front part of the vehicle for a front shaft and a differential. **Eliminated**

4. Two motors. Front-and-rear-wheel-independent-drive-type electric vehicle (FRID EV)

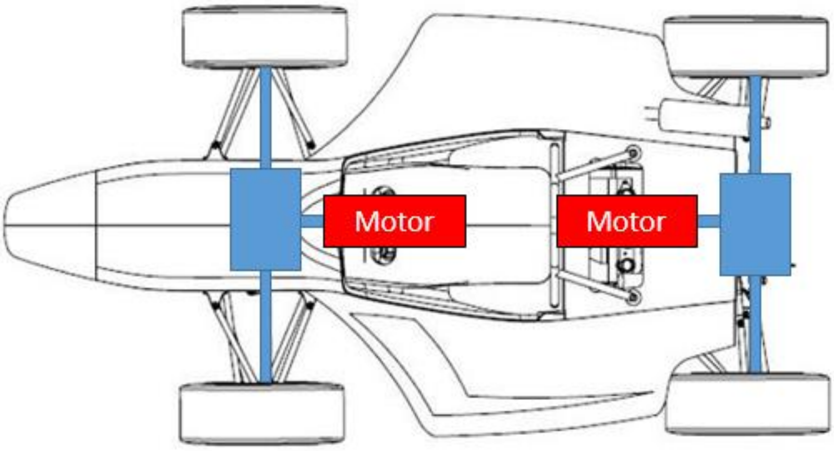


FIGURE 6: *FRONT-AND-REAR-WHEEL-INDEPENDENT-DRIVE-TYPE ELECTRIC VEHICLE POWERTRAIN CONFIGURATION*

This configuration has two individual propulsion systems, one on the front shaft and the other on the rear shaft. The main function of this configuration is to perform efficient acceleration and deceleration on all roads by suitably distributing the driving or braking torques to both shafts, enabling load transfer. **[9] [10]**

This is an impossible design to be integrated in the current Formula Student electric car, since there is no room in the front part of the vehicle for a front shaft. **Eliminated**

5. Two motors. Rear in-wheel motor hubs

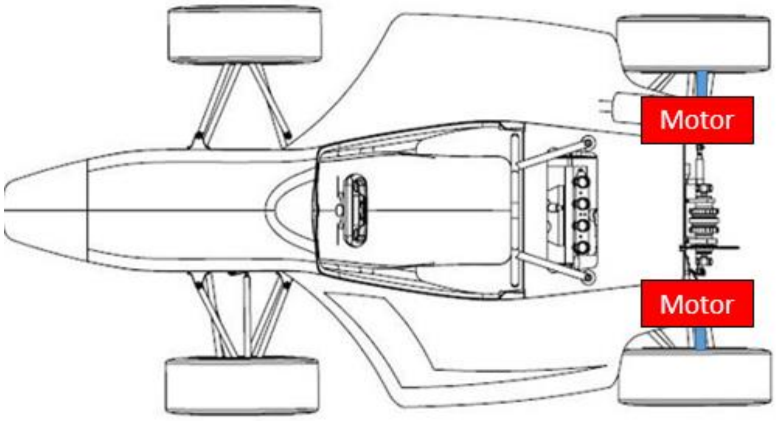


FIGURE 7: *REAR IN-WHEEL MOTOR HUBS POWERTRAIN CONFIGURATION*

This is a configuration with two engines directly distributing torque to each of the rear wheels. Having two independent motors will allow the system for individual torque distribution. **[11] [12] [13]**

This configuration will be further analysed. It will be interesting to check the performance of a vehicle with rear torque vectoring. **Accepted**

6. Two motors. Front in-wheel motor hubs

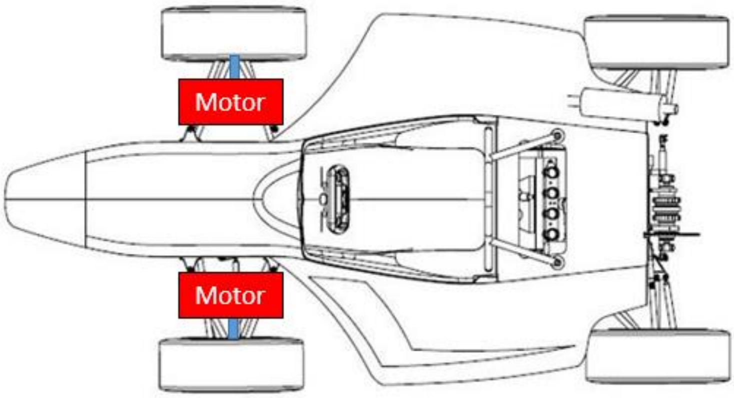


FIGURE 8: *FRONT IN-WHEEL MOTOR HUBS POWERTRAIN CONFIGURATION*

This is a configuration with two motors directly distributing torque to the front wheels. The individually controlled engines on the front would provide the vehicle of front TV (torque vectoring). **[14]**

However, Lund's team mentioned that this design would perform understeering. It is not the most suitable configuration and even though it might be useful to analyse front torque vectoring, this configuration is unlikely to be designed in a real race car scenario. **Eliminated**

7. Three motors. One motor in the front shaft and two rear in-wheel motor hubs

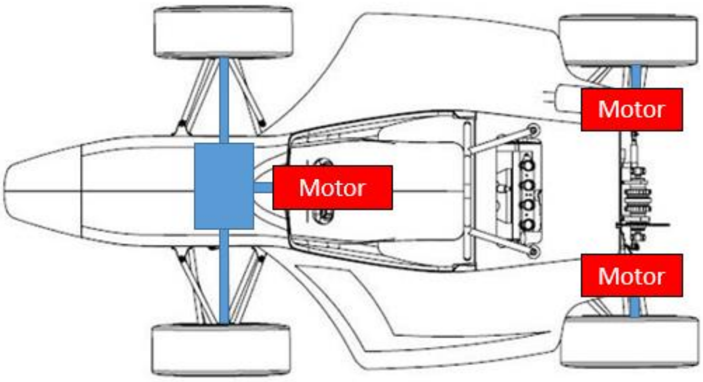


FIGURE 9: *ONE MOTOR IN THE FRONT SHAFT AND TWO REAR IN-WHEEL MOTOR HUBS POWERTRAIN CONFIGURATION*

This design allows a configuration which can either distribute torque to the rear part and the front and also to the left and right sides by distributing the torque with the rear wheel motors **[15] [16].**

However, this design cannot be applied to the Lund's car due to the lack of room in the front for a front shaft. **Eliminated**

8. Three motors. One motor in the rear shaft and two front in-wheel motor hubs

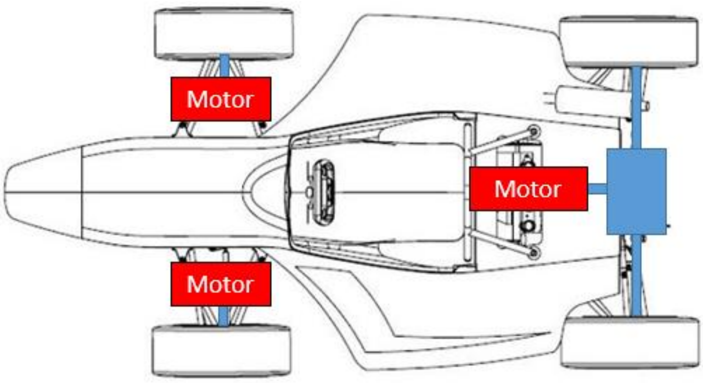


FIGURE 10: *ONE MOTOR IN THE REAR SHAFT AND TWO FRONT IN-WHEEL MOTOR HUBS POWERTRAIN CONFIGURATION*

This design allows a configuration which can either distribute torque to the rear part and the front and also to the left and right sides by distributing the torque with the front wheel motors. **[17]**

This design is suitable for Lund's Formula Student electric car. Furthermore, the design will allow comparing another type of torque vectoring by using a front torque distribution. **Accepted**

9. Four motors. Four in-wheel motor hubs

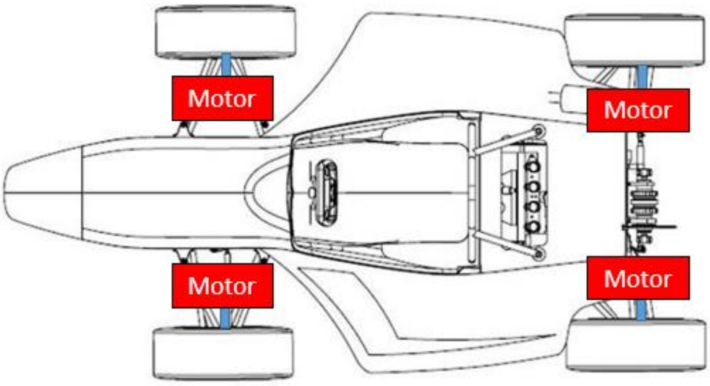


FIGURE 11: *FOUR IN-WHEEL MOTOR HUBS POWERTRAIN CONFIGURATION*

This is the most popular design when using in-wheel motors. It is the most used design at the moment for electric race cars on the FS Competition. It allows the car to have a perfect torque distribution among all the wheels, enabling both load transfer and torque vectoring. **[18] [19] [20**

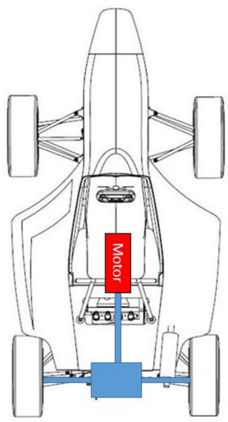
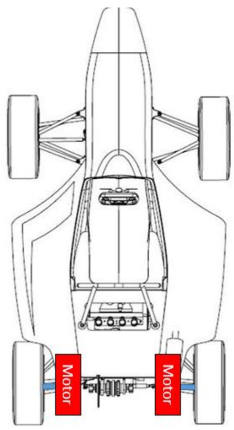
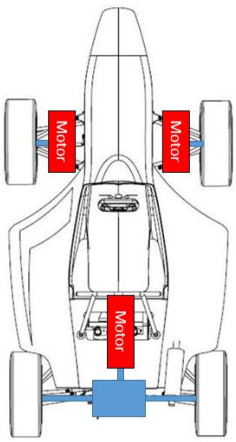
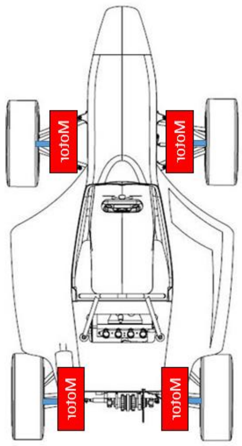
This design can be implemented. It will also be interesting to compare this configuration with all-wheel's torque distribution with the other two types of torque vectoring. **Accepted**

2.3 Accepted powertrains

Finally, there are four main designs which will be compared in the following analysis. The designs are listed below. From now on, they will be referred to as Powertrain 1, Powertrain 2, Powertrain 3 and Powertrain 4.

The number of the Powertrain configuration is equal to the number of motors of the design.

TABLE 1: *VIABLE POWERTRAIN CONFIGURATIONS CONSIDERED FOR LFS*

Powertrain 1	Powertrain 2	Powertrain 3	Powertrain 4
Rear wheels driven	Rear in-wheel motor hubs	One motor in the rear shaft and two front in-wheel motor hubs	Four in-wheel motor hubs
			

3 Vehicle Dynamics Simulation

In order to compare the performance on each race of the FS Competition, a software code was developed to simulate the vehicle dynamics for each powertrain concept chosen.

The simulation was written in Matlab, a numerical computing software, which allows to introduce all the equations describing the vehicle dynamics for each powertrain. The Lund Formula Student team already has a program, which was used to simulate the combustion car. This program has been adapted to this project in order to accurately represent the different electric cars. Additionally, the simulation code in Matlab provides fast execution times, allowing to simulate one powertrain alternative in about 20 seconds.

3.1 Theory of the model

To simulate the handling of a vehicle, different types of vehicle models are possible, depending on the desired level of detail and the task at hand. Mainly the different approaches may be divided according to the Degrees of Freedom (DoF) the model involve. In the next table, different model types are shown:

TABLE 2: *VEHICLE DYNAMICS MODEL TYPES ACCORDING TO THE NUMBER OF DOF* **[23]**

Model type	Degrees of freedom
Single track model, linear	2
Single track model, nonlinear	3-7
Twin track model	14-30
Complex multibody system model	>20
Finite-Element model	>500
Hybrid model	>500

The aim of this thesis, is to compare the performance in a race, and not to directly comprehend what is happening in every part of the vehicle at each time step. This model will also compare the different proposed concepts in each of the races, thus, it requires a program with a rather fast execution.

The race car considered in this work, can be modelled simulating the vehicle longitudinal motion and the lateral motion, which requires 2 DoF. The model will be based in a single track linear model. This model will require less DoF than others, hence some further assumptions have to be made:

- All types of strain will be neglected (vehicle suspensions is not considered in the model)
- All lifting, rolling and pitching motion will be neglected (only yaw rate is considered)
- The vehicle's mass is assumed to be concentrated at the centre of gravity.
- The slip angle of the tire will be neglected.
- The wheel-load distribution between front and rear axle is assumed to be constant.
- The longitudinal forces on the tires, resulting from the assumption of a constant longitudinal velocity, will be neglected.

Another thing that differentiates this model is the way the wheels are modelled. Single track models normally use a bicycle model, considering the front and rear tires to be represented as one single tire in each axle. In this case, a single wheel model is used, based on a g-g diagram, which is explained in next section

3.2 GG diagram

In its simplest terms, a race circuit may be thought of as a number of segments each composed of a corner, a straight and a corner. Each corner has a certain curvature, and there is an optimal acceleration profile for each corner. An example of a curve is on the following image.

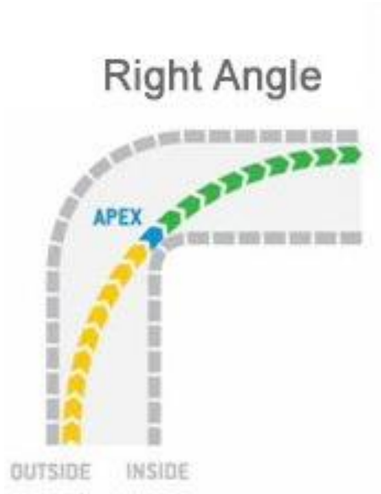


FIGURE 12: *RIGHT APEX TURN* **[24]**

In every corner, one should arrive to the curve at maximum possible velocity, and then start braking until it reaches the apex point, which is the exact corner of the curve. At this point the car fully accelerates again until it finds the next apex point where it needs to decelerate and so on. An important principle of circuit racing is that velocity should never be constant, unless limited by the maximum speed of the vehicle.

In engineering terms, velocity is a vector quantity because it possesses both a magnitude (speed) and a direction. It may be represented by an arrow whose length, to some arbitrary scale, corresponds to the speed and whose direction is given by the arrow's orientation. This velocity arrow is constantly changing in length and direction, that is why race car requirements are best expressed in terms of acceleration.

In terms of racing there are two main accelerations:

- **Longitudinal acceleration**: Acceleration in a straight line. This is the length change of the velocity vector. Acceleration when positive and deceleration or braking when negative. The longitudinal acceleration is the variation of velocity divided by the time between the two velocity points ($\Delta V/ \Delta t$). It is common to express the acceleration in gravitational field units (g). So the expression would be $a_{longitudinal}= \Delta V/g\cdot \Delta t$.

- Cornering acceleration or **lateral acceleration**: This acceleration is associated with the change in direction of the velocity vector with time. The lateral acceleration is given by the relation v^2/R where v is the speed in meters per second and R is the radius of the path in meters. In gravitational field units, the expression would be $a_{lateral}= v^2/g \cdot R$.

Acceleration components and resultant accelerations can also be used to see a race car circuit performance:

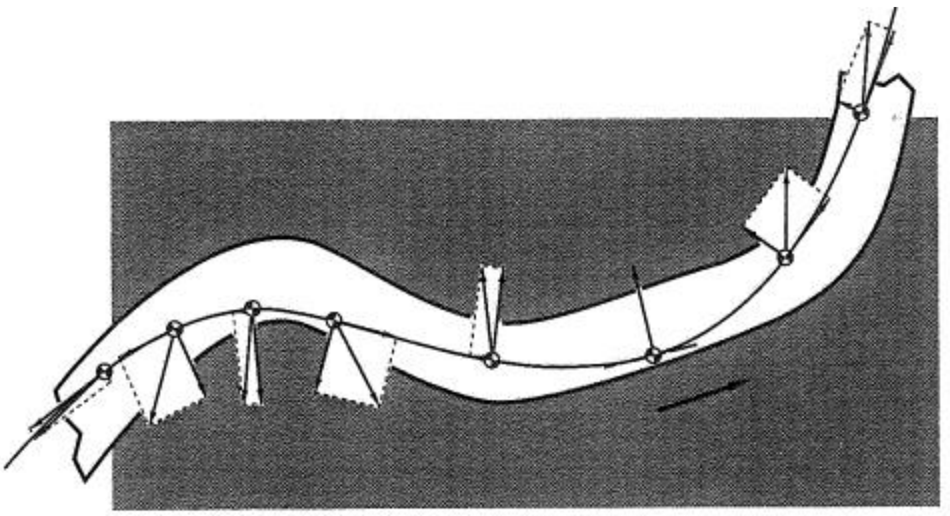


FIGURE 13: *RACE PERFORMANCE IN ACCELERATION TERMS* **[25]**

Figure 13 shows a car's behaviour throughout a random track. The arrows represent the acceleration of the car, being the ones parallel to the trajectory the longitudinal acceleration, the perpendiculars the lateral acceleration and the sum of both the total.

The resultant acceleration vector as the vehicle progresses along a circuit has led to the concept of the g-g diagram, which represent the acceleration points of the car. An example of this representation is shown in the next image:

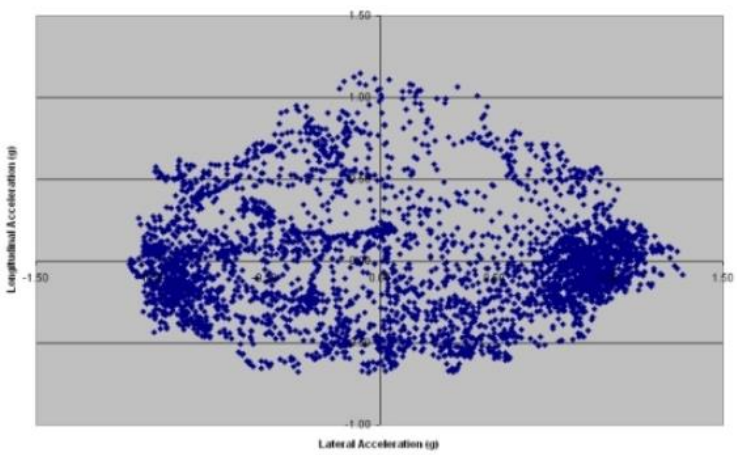


FIGURE 14: *REPRESENTATION OF A G-G DIAGRAM* **[26]**

This diagram is used to show the acceleration vector during the whole race. In the above figure, one can see the limits of the acceleration vector, being ellipse shaped. The g-g diagram has a second use that has not been mentioned yet: it allows us to determine the manoeuvring area utilized by the vehicle in a comprehensive series of tasks.

This is the key of the vehicle dynamics simulation program. By using a closed loop code, the maximum longitudinal and lateral acceleration at each car speed can be found. The limits of both accelerations depend on the maximum grip limit of the tire. This introduces the tire friction circle. **[25]**

The idea is that no matter what combination of steering and braking/driving torques are applied to a wheel, the maximum horizontal force that the tire can produce is limited by the tire or road friction coefficient times the load on the wheel.

Conceptually, the tire friction circle or ellipse, can be applied to the whole automobile by collapsing the four wheels into a single equivalent car/road interface. It means, that each of the friction tire circles applied to the four wheels can be modelled as a single wheel, by summing the horizontal forces of each wheel to a single one.

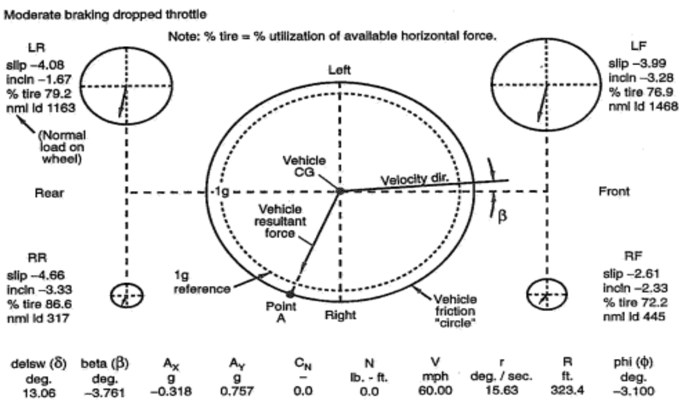


FIGURE 15: *VEHICLE FRICTION CIRCLE/ELLIPSE* **[25]**

In this case, the friction force is considered both lateral and longitudinal. This means that there are different values for the friction coefficient for both directions (μ_{yy} and μ_{yx} are different) and thus the tire friction circle becomes an ellipse.

The goal is to find the g-g diagram of the vehicle, to acknowledge the dynamics limit of it. The first step is to find the maximum positive longitudinal acceleration capacity of the vehicle at different velocities, considering lateral acceleration to be 0. It is considered that the vehicle can run at 120 km/h at maximum. From 0 km/h to 120 km/h, the car is considered to run at 14 different velocities, thus 14 acceleration points are calculated with the following method:

1) Find the wheel load distribution for a combination of speed and acceleration. The first guess of the acceleration is the maximum force that the motors can deliver at the considered speed. The wheel load distribution is calculated with the free body diagram showed in **Figure 16**.

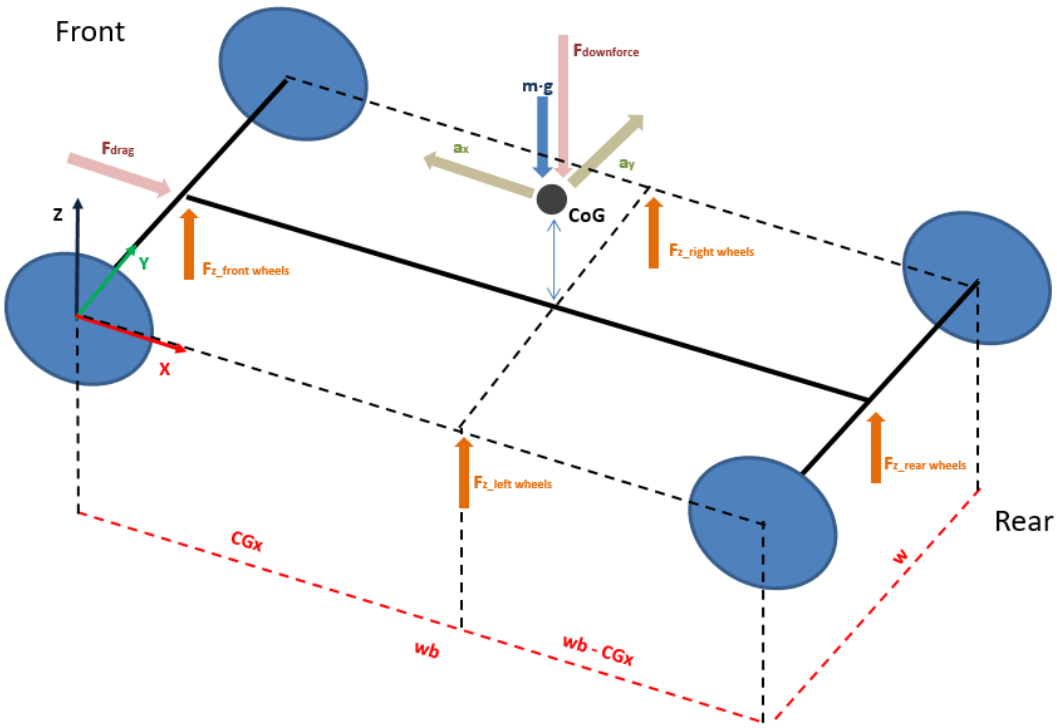


FIGURE 16: *FORCES ON THE VEHICLE DIAGRAM*

With a_x as the longitudinal acceleration input (vehicle force divided by mass) and a_y the lateral acceleration input (which is 0 on this case), the load of front/rear and right/left wheels is calculated, by applying the equations of forces in equilibrium:

$$DragForce = \frac{1}{2} \cdot \rho \cdot A \cdot Cd \cdot v^2$$

$$DownForce = \frac{1}{2} \cdot \rho \cdot A \cdot Cl \cdot v^2$$

$$F_{front\ wheels} = \frac{m \cdot g \cdot CG_x + DownForce \cdot x - DragForce \cdot Z - m \cdot a_x \cdot CG_z}{wb}$$

$$F_{rear\ wheels} = m \cdot g + DownForce - F_{front\ wheels}$$

$$F_{right\ wheels} = \frac{m \cdot g \cdot CG_y + DownForce \cdot CG_y - m \cdot a_y \cdot CG_z}{w}$$

$$F_{left\ wheels} = m \cdot g + DownForce - F_{right\ wheels}$$

In these equations the left tire is considered as the coordinates origin and ρ is the air density, A is the frontal area of the car, Cd and Cl are the drag and lift coefficients, v is the velocity, m is the mass of the vehicle, g is gravity, CG_x, CG_y and CG_z are the distance to the centre of gravity, X and Z are the considered attacking points of the downforce and dragforce, w is the track width and wb is the wheel base.

It should be noted that when the vehicle is accelerating the load is higher on the rear wheels, while when braking the front wheels have to handle a higher force.

2) The load at each wheel is then calculated, considering a constant front load transfer of 0,45 being a bit lower than in the rear shaft (0,55) which are calculated as:

$$Front\ load\ transfer = \frac{CG_x}{wb}$$
$$Rear\ load\ transfer = 1 - Front\ load\ transfer$$

The side load transfer is calculated as the difference between the wheel load of left wheels minus the wheel load of right ones:

$$Side\ load\ transfer = \frac{F_{left\ wheels} - F_{right\ wheels}}{2}$$

Now, the load at each wheel can be calculated applying load transfer:

$$F_{rear\ right\ wheel} = \frac{F_{rear\ wheels}}{2} - Rear\ load\ transfer \cdot Side\ load\ transfer$$
$$F_{rear\ left\ wheel} = \frac{F_{rear\ wheels}}{2} + Rear\ load\ transfer \cdot Side\ load\ transfer$$
$$F_{front\ right\ wheel} = \frac{F_{front\ wheels}}{2} - Front\ load\ transfer \cdot Side\ load\ transfer$$
$$F_{front\ left\ wheel} = \frac{F_{front\ wheels}}{2} + Front\ load\ transfer \cdot Side\ load\ transfer$$

The model considers that the vehicle cannot tilt, thus if a negative load is calculated on a wheel, it is changed to the minimum load (1 newton).

3) The total tire limit of the vehicle considering the four wheels collapsed in one, is calculated as the sum of the load at each wheel multiplied by the friction coefficient.

4) Lastly, it is checked if the vehicle tire limit is higher or lower than the force applied by the engine of the vehicle. If it is lower, that means we are grip limited, and a recalculation has to be made. The process is to go back to step one, considering as input force the tire limit. However, if the tire limit is higher than the engine force, that means that we are not grip limited, as so, the maximum longitudinal acceleration at this velocity is the engine force divided by the vehicle mass.

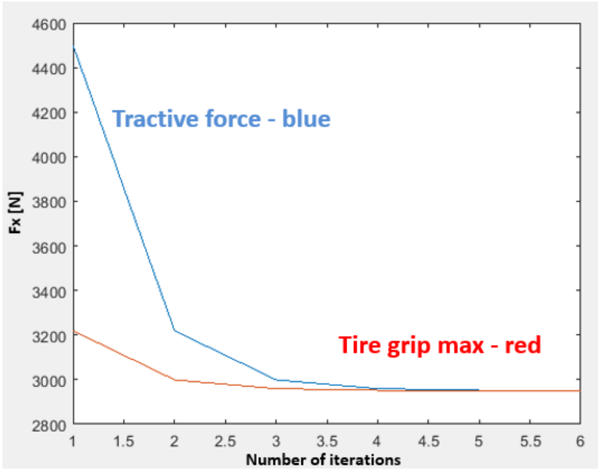


FIGURE 17: *CONVERGENCE OF THE VEHICLE LOAD AT A CERTAIN SPEED*

Figure 17, shows the iteration process to find the maximum force of the vehicle speed at a certain velocity. The blue line is the force from the engine, while the red one is the tire grip limit.

Once the positive longitudinal acceleration is calculated at each velocity, the process is repeated for negative longitudinal acceleration (again, considering lateral acceleration to be 0). This will provide the results of the braking capacity of the vehicle. Now, it is considered that the vehicle can apply as much force as the tire grip limit, hence no iterations have to be made. It should be highlighted that the free body diagram from **Figure 16**, has the longitudinal acceleration pointing the other way when the maximum braking points are calculated (negative acceleration)

With both the maximum negative and positive longitudinal acceleration, the lateral acceleration at each velocity and longitudinal acceleration has to be calculated. A 14x14 matrix is created, which includes all the different possible longitudinal acceleration points that can be achieved:

	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	-13.7265	-11.4387	-9.1510	-6.8632	-4.5755	-2.2877	0	1.8861	3.7723	5.6584	7.5446	9.4307	11.3168	13.2030
2	-13.7851	-11.4876	-9.1901	-6.8926	-4.5950	-2.2975	0	1.8943	3.7886	5.6830	7.5773	9.4716	11.3659	13.2603
3	-13.9605	-11.6338	-9.3070	-6.9803	-4.6535	-2.3268	0	1.9188	3.8376	5.7564	7.6752	9.5940	11.5128	13.4316
4	-14.2509	-11.8758	-9.5006	-7.1255	-4.7503	-2.3752	0	1.9593	3.9186	5.8779	7.8372	9.7965	11.7559	13.7152
5	-14.6536	-12.2113	-9.7690	-7.3268	-4.8845	-2.4423	0	2.0155	4.0309	6.0464	8.0619	10.0774	12.0928	14.1083
6	-15.1646	-12.6371	-10.1097	-7.5823	-5.0549	-2.5274	0	2.1010	4.2021	6.3031	8.4042	10.5052	12.6063	14.7073
7	-15.7791	-13.1493	-10.5194	-7.8896	-5.2597	-2.6299	0	2.0475	4.0950	6.1426	8.1901	10.2376	12.2851	14.3326
8	-16.4915	-13.7429	-10.9943	-8.2457	-5.4972	-2.7486	0	1.9358	3.8717	5.8075	7.7434	9.6792	11.6151	13.5509
9	-17.2951	-14.4126	-11.5300	-8.6475	-5.7650	-2.8825	0	1.8242	3.6483	5.4725	7.2967	9.1209	10.9450	12.7692
10	-18.1372	-15.1143	-12.0915	-9.0686	-6.0457	-3.0229	0	1.6605	3.3211	4.9816	6.6422	8.3027	9.9633	11.6238
11	-19.0095	-15.8412	-12.6730	-9.5047	-6.3365	-3.1682	0	1.4837	2.9674	4.4512	5.9349	7.4186	8.9023	10.3860
12	-19.9226	-16.6022	-13.2817	-9.9613	-6.6409	-3.3204	0	1.3494	2.6988	4.0482	5.3977	6.7471	8.0965	9.4459
13	-20.8620	-17.3850	-13.9080	-10.4310	-6.9540	-3.4770	0	1.2381	2.4763	3.7144	4.9525	6.1907	7.4288	8.6670
14	-21.8114	-18.1762	-14.5410	-10.9057	-7.2705	-3.6352	0	1.1283	2.2567	3.3850	4.5134	5.6417	6.7700	7.8984

FIGURE 18: *LONGITUDINAL ACCELERATION TABLE OF THE VEHICLE*

The first column is the maximum braking acceleration, the last column is the maximum positive acceleration and the 7th column includes only zeros. The rest columns are filled with equally distributed differences with the middle column. Rows 1 to 14 are the different velocities at which the vehicle may run, being 14th the highest (120 km/h).

For each different longitudinal acceleration and vehicle velocity, the maximum lateral acceleration has to be calculated. It is done with the following method:

1) For a first guess, the lateral acceleration is calculated with the ellipse tire model, which has the following equation

$$AyTire = \sqrt{AxTire^2 \cdot \frac{\mu_{yy}{}^2}{\mu_{yx}{}^2} - \mu_{yy}{}^2 \cdot \frac{Fz^2}{m}}$$

Where AxTire is the longitudinal acceleration point, μ_{yy} and μ_{yx} are friction coefficients, m is the mass and Fz is the vertical force on the vehicle. For the first guess, Fz is calculated as the weight of the vehicle plus the downforce.

2) With AyTire and AxTire being known accelerations, the load at each wheel is calculated, by applying the equilibrium equations on the previously showed free body diagram (**Figure 16**) and applying the load transfer the same way.

3) The longitudinal force applied to each wheel is calculated multiplying the proportion of wheel load per the total longitudinal force of the vehicle. Using the friction ellipse, the lateral acceleration at each wheel is found, using the following equation:

$$AyTire = \sqrt{\mu_{yy}{}^2 \cdot \left(\frac{Fz}{m}\right)^2 - AxTire^2 \cdot \frac{\mu_{yy}{}^2}{\mu_{yx}{}^2}}$$

In the equation, μ_{yy} and μ_{yx} are friction coefficients, Fz is the wheel load, m is the mass of the vehicle and AxTire is the longitudinal acceleration proportion.

4) The sum of the lateral acceleration at each wheel is the total lateral acceleration of the vehicle. The calculated value is compared to the lateral acceleration input of step 1). If the difference is considerable (more than 0,1% on this model), the lateral acceleration is recalculated, with the acceleration found being the new input.

The process is finished when the 2nd 14x14 matrix is completed, being the lateral acceleration limits. With it, a surface plot can be drawn, which shows the limits of lateral and longitudinal acceleration of the vehicle.

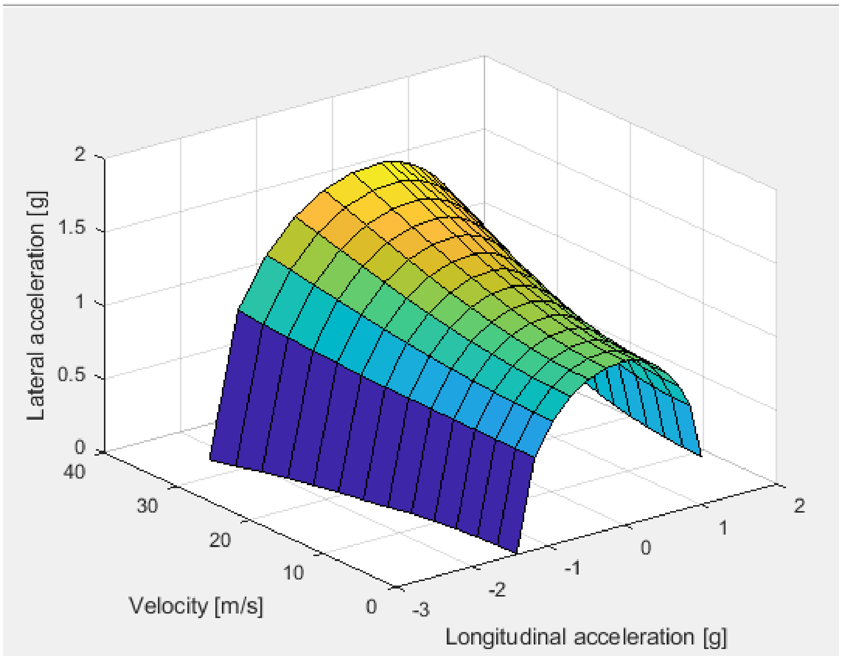


FIGURE 19*: G-G VEHICLE DIAGRAM*

3.3 Torque vectoring

One of the main reasons why a race car with multiple motors might perform better is due to the torque vectoring. Torque vectoring is the term introduced by the company Ricardo to identify a driveline device capable of controlling the magnitude and direction of torque to influence traction and vehicle dynamics, and is accurately described as a variable torque bias coupling **[27]**

The idea of torque vectoring driveline systems is that these are capable of distributing drive torque in controllable proportion to each of the wheels. Generally, a vehicles reaction to steering input is to change direction, also referred as yaw. The yaw rate in conventional vehicles is only controlled steering, but the ability to independently regulate torque at each wheel opens up for the possibility of active regulation of yaw response. This concept is illustrated by the next figure:

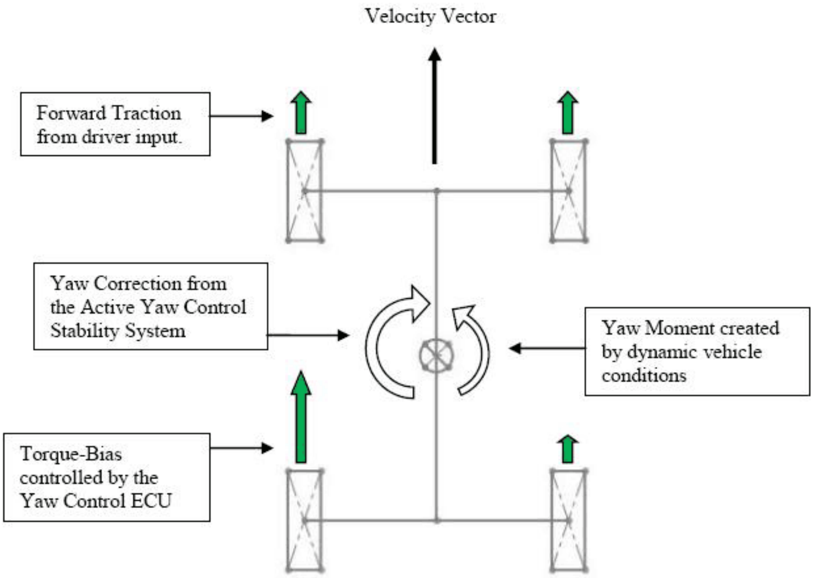


FIGURE 20: *YAW CORRECTION FROM TORQUE VECTORING PHENOMENA* **[28]**

In **Figure 20** it is shown that when different torque is applied at each of the wheels, the total yaw rate is incremented over the centre of the vehicle, allowing the vehicle to turn without direction steering on the wheels.

This concept is best suited to electric vehicles because of the more instantaneous torque response and the ability to more easily achieve independent power output at each wheel. **[29]**

The ability of creating a yaw moment can be done through the right-and-left torque vectoring which is possible in different types of powertrains:

- **Front wheel driven driveline (FWD):** When only front wheel's torque may be controlled. This is the case for powertrain 3.
- **Rear wheel driven driveline (RWD):** When rear front wheel's torque may be controlled. This is the case for powertrain 2.
- **All wheel driven driveline (AWD)**: When all wheel's torque may be controlled. This is the case for powertrain 4.

But this concept is beyond the yaw rate fast response. Studies state that the total maximum cornering force of the right and left wheel's increases. This occurs because the system optimizes the driving force assignment between the right and left wheels.

This has a direct impact to the G-G diagram explained in the previous chapter. The tire limits on longitudinal and lateral acceleration may be increased when torque vectoring is applied. Conventional vehicles produce yaw only by steering the wheels in a certain way, and the grip limit dictates the maximum yaw rate of the vehicle. By applying torque vectoring, the extra yaw rate allows the grip limit to be higher, or in other words, apply a higher lateral acceleration at the same speed as a conventional vehicle. **[20] [30]**

Researchers suggest that this has a direct impact in the G-G diagram, improving the vehicle dynamic limits. **Figure 21** shows how these limits may be modified at a certain speed:

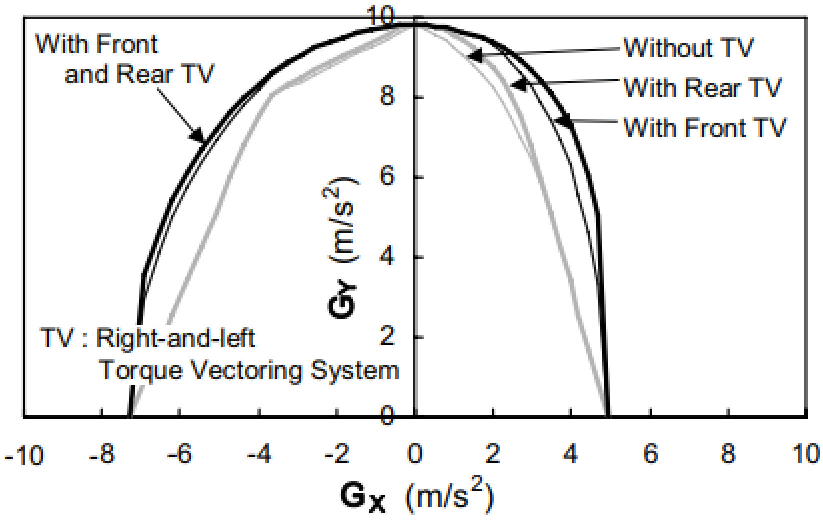


FIGURE 21: *VEHICLE DYNAMIC LIMITS MODIFICATION WITH TORQUE VECTORING* **[30]**

To include torque vectoring in the Vehicle dynamics simulation model, a function was generated. Firstly, a GG plot is generated as explained in the previous section. This plot is modified to increase its limits, as explained in the following paragraph

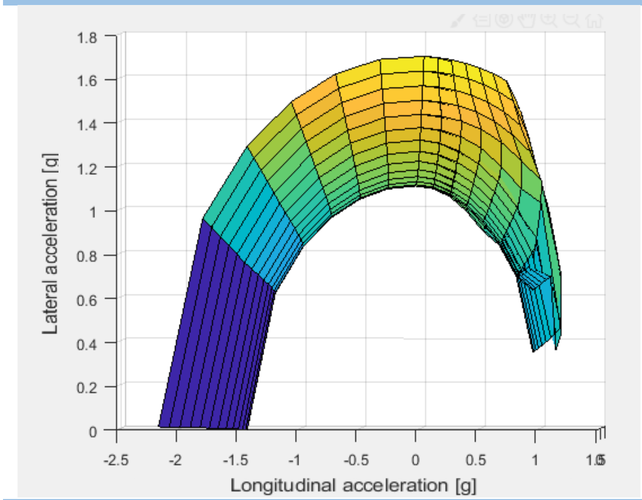
Similar to the previous code explained, the first step is to calculate the load on each wheel at a certain speed. By multiplying the load by the friction coefficient the horizontal force from each wheel is calculated. If this force is lower than the maximum force that the motor can apply, it means that there is still some margin for torque distribution. The difference between the two, is the torque that may be distributed.

Since the maximum vehicle friction is represented by the sum of the horizontal forces of all wheels, the torque that we are able to distribute can be added to the outer wheel and be subtracted from the inner wheel. The total force will keep balanced, and the total tire force will be inside the grip limit.

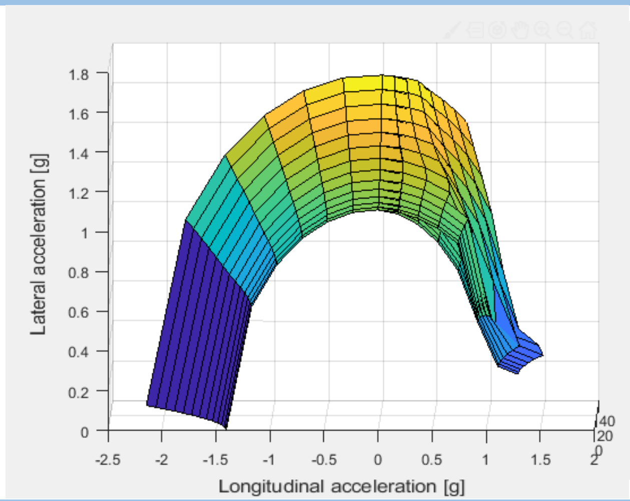
This process is repeated for all the velocity points. The GG diagrams are modified to the ones shown in the following picture, overlapped to the standardly generated ones. Powertrain 1 is not shown because the configuration lacks torque vectoring capacity

TABLE 3: *GG DIAGRAM BEFORE AND AFTER APPLYING TORQUE VECTORING*

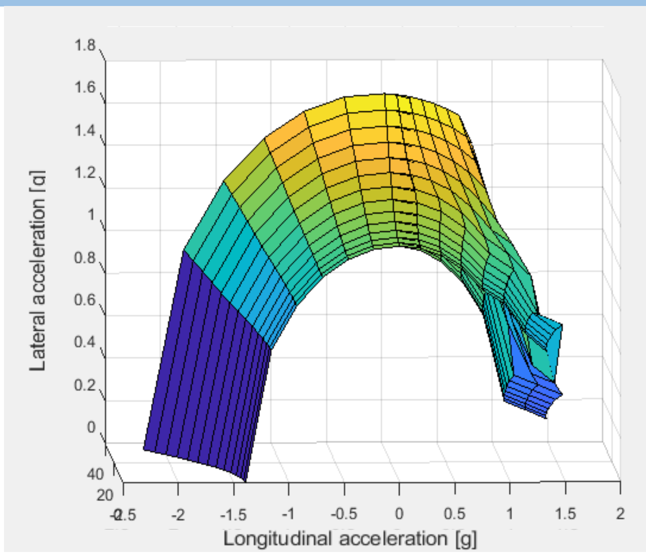
POWERTRAIN 2



POWERTRAIN 3



POWERTRAIN 4



It can be seen that the diagram changes the limits significantly on Powertrain 4. This is caused by the possibility of distributing torque among all the wheels. Both outer wheels and both inner wheels are able to modify the torque applied to have larger limits and then be a faster car on some turns.

The other two also increase the lateral acceleration limits, but the results are not as notable as for Powertrain with 4 in-wheel motors.

It should also be highlighted that none of the powertrains increase the limits on negative longitudinal acceleration. This is because the braking system was modelled as if the capacity of braking is directly dictated by the grip limit and not by the motor force. That means that there is no margin for the motor to distribute torque to make a faster turn when decelerating.

Although it is a bit too soon to jump into conclusions, it can already be seen the good results from in-wheel motors driveline.

3.4 Functionality of the Vehicle Dynamics Simulation Model

The Matlab code was originally developed by a former team member from the Lund Formula Student. It was developed to simulate the combustion engine car and check how good it would perform on the competition beforehand. The program has been adapted during this project for the electric car, and has to be able to simulate each of the configurations on each of the dynamic events of the FS competition.

The code includes up to 25 functions and more than 20 scripts. As previously explained the dynamics are based on the previously explained GG diagram.

The program to simulate the vehicle dynamics follows the structure:

1) Load files and add input variables

The first step is to load all the files. These files contain the information of the motor curves (motor torque and motor speed points) and the track information. The track files are text files with two columns, the first one indicating the length of an increment and the second one the radius of it. This files are taken from the 2012 Hockenheim FS competition, Germany. They are available on OptimumG's website.

The input variables are all those parameters that will be used. They include the physics variables (gravity and friction coefficients), aerodynamics (drag and lift coefficients) and all the car variables like mass, frontal area, wheel radius, centre of gravity.

All the car variables are based on the LFS20' electric car, and are adapted depending on the chosen Powertrain. A recalculation of mass and centre of gravity is done through and estimation of the components that are needed on each powertrain. For example, from Powertrain 1 to Powertrain 4, the differential, central motor and inverter are removed, and four motors, four inverters and four reducers are added.

2) Race Type

The race type is then chosen. Endurance and autocross races are created with the loaded files. For the skid pad race and acceleration, the tracks are created with a function.

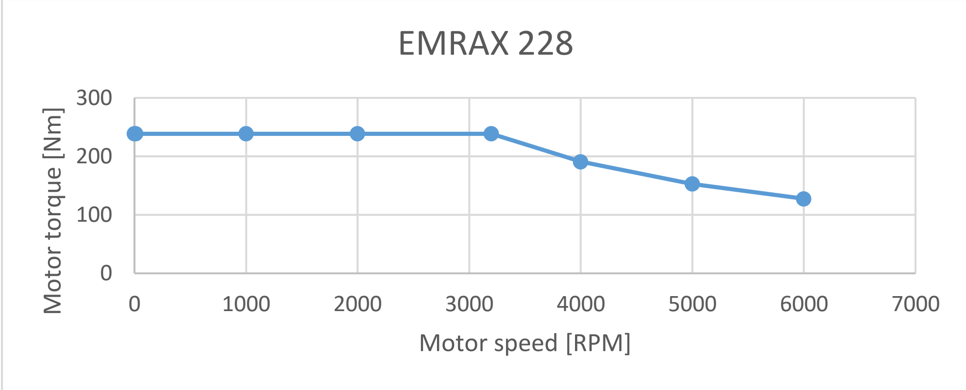
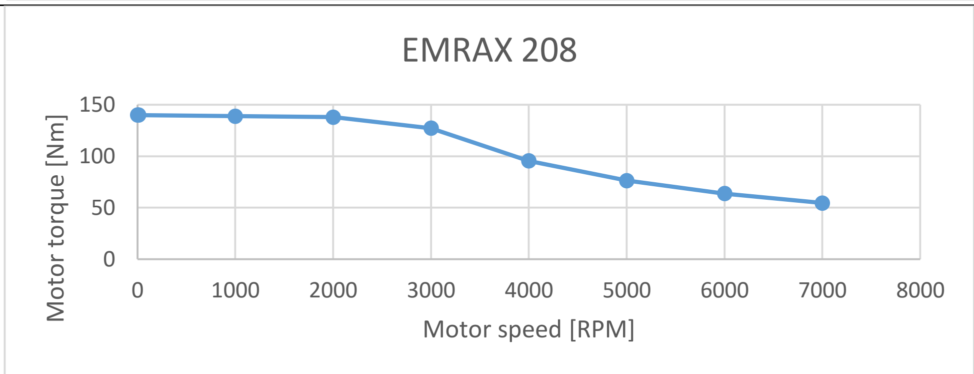
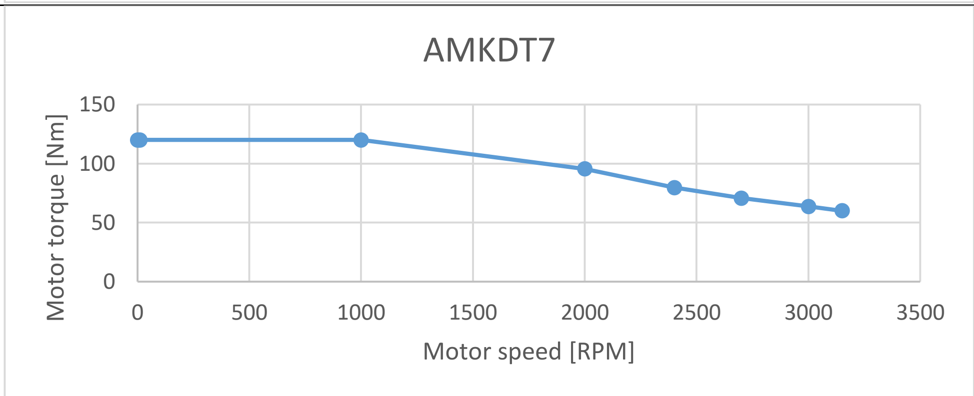
3) Load Engine

The next step is to load the engine data. A previous research was done to find the proper engines. The main brands that were considered to determine the appropriate motor for each configuration were: Emrax and AMK.

The motors were dimensioned according to a FS Competition rule for electric cars, which states that the maximum output of power has to be 80kW. As so, the combination of engines of the configuration is required to have a power output close to 80kW. This way, the efficiency from the motors are maximized.

Once the motor was chosen the torque- RPM curve was elaborated, limiting the load of the power for each case. Three motors were chosen for this project, shown in the next figure:

TABLE 4: *TABLE SUMMARIZING ENGINES CHOSEN*

Motor	Power load limit	Powertrain	Torque-RPM curve
EMRAX 228	80 kW	Powertrain 1 Central motor	
EMRAX 208	40 kW	Powertrain 2 Wheel hub motors, Powertrain 3 Central motor	
AMKDT7	20 kW	Powertrain 3 Wheel hub motors, Powertrain 4 Wheel Hub motors	

Once the motors have been determined, the calculation of the differential ratio is done. This ratio cannot be higher than 10, since then the reducer would require a more complex design. This was also a limitation when looking for suitable motors.

To calculate the differential ratio, it was estimated that the vehicle reaches 120km/hand that this speed is reached at the highest motor speed point of the torque curve.

Now, the tractive effort vector is found. This vector describes the maximum force that the vehicle can do at each speed. The lines of the code which include both the differential ratio calculation and the tractive effort vector are shown in the following image:

```
finalDriveRatio = (AMKDD15_rpm(length(AMKDD15_rpm))*pi/30)/(v(length(v)))*wheelRadius;
for vPosition=1:length(v)
    current_rpm = v(vPosition)*finalDriveRatio/2/pi/wheelRadius*60;
    torque_engine = interp1(AMKDD15_rpm,AMKDD15_torque,current_rpm); %MOTOR MODIFICATION
    torque_wheel = torque_engine*finalDriveRatio; %We consider efficiency 1, since the mo
    tractiveEffortVector(vPosition) = 4*torque_wheel/wheelRadius*driveEfficiency; %Maximu
end
```

FIGURE 21: *CALCULATION OF THE TRACTIVE EFFORT VECTOR AND THE DIFFERENTIAL RATIO FOR POWERTRAIN 4*

From the differential ratio considered on the reducer, a transmission efficiency was estimated. This efficiency was estimated through the gearbox need, which was considered to be about 98% for all designs. **[31]**

4) GG-Plot and torque vectoring

The next part of the program consists in creating the GG diagram for each velocity. The limits of the diagram are expanded by the torque vectoring code. This part was explained in the previous sections.

5) Run Lap

The next step is to run the lap. The main goal of this part is to declare the velocity of the car at each track step.

Firstly, the number of track points are increased. For a better definition of the race, the track points are converted into smaller increments to make a smoother track. From this increments and with the information of the radius curve of each, the apexes points are determined. The apex points correspond to the increments where the car changes from braking to throttling.

To calculate the speed of the car at each time step, it is first required to calculate the proper velocity at which the apexes should be run. The speed calculation on apexes is simpler, since the longitudinal acceleration at these points is zero. With the radius at each increment, the proper lateral acceleration can be calculated with the following equation:

$$v = \sqrt{F_y \cdot \frac{CornerRadius}{m}}$$

This is the same math explained before to identify the lateral acceleration ($a_{lateral} = v^2/g \cdot R$). The shown equation requires two unknown variables, F_y (lateral force from the car) and v (velocity). For a first guess, the maximum velocity is used. If the maximum lateral grip force required at that velocity is higher than the limit of the lateral acceleration (from the GG-plot) a lower velocity is used for a new guess. The program iterates from the maximum velocity to the proper maximum velocity that can be run at that apex point.

Once the speed at the apex points is known, the following and previous velocity points are calculated considering an increment of full throttle or full braking depending on the case. The speed points are calculated taking into account the radius of each and the velocity of the previous one. If the next points of an apex point of accelerating have a lower velocity than the previous points of another apex, the minimum of the two is taken. Then the velocity of all the points between these two apexes is known. The next figure explains this process better:

Radius of the point	8.2808	8.2808	0
0.0490	0	8.2949	0
0.0490	0	8.3153	0
0.0490	0	8.3404	0
0.0490	0	8.3686	0
	[…]		
0.0624	0	14.3452	0
0.0624	0	14.3895	0
0.0624	0	14.4336	14.4393
0.0624	0	14.4776	14.3779
0.0624	0	14.5215	14.3162
	[…]		
0.1949	0	14.8700	11.1408
0.1949	0	14.8599	11.0521
0.1949	0	14.8497	10.9698
0.1949	0	14.8396	10.8955
0.1949	10.8319	14.8295	10.8319

FIGURE 23: *MATLAB CODE VELOCITIES BETWEEN APEX POINTS (GREEN ARROW MEANS ACCELERATION POINTS AND RED ARROW MEANS DECELERATION POINTS)*

In **Figure 23** an example of the velocities at which the vehicle runs are shown. From the first apex (green squared point), the vehicle accelerates as much as possible considering the maximum lateral acceleration from the GG plot limits and calculating the maximum velocity with the radius from each point (left column). From the next apex (black squared point), the previous speed points are calculated with the vehicle full braking and with the limit considerations again. Then when one of the velocity columns is higher than the other, the minimum of both is taken as the appropriate the velocity. That means that in **Figure 23**, the vehicle will accelerate from the apex point to 14,4336 m/s (blue squared point) and then start braking until it reaches the next apex point. From this point, the vehicle will go full throttle again.

6) Data results and plots

The rest of the variables can be calculated through the velocity at which each track increment is run at. With the already known variables, the calculation for the rest is shown in the following table

TABLE 5: *DATA OUTPUT CALCULATIONS*

Variable	Letter abbreviation	Formula to calculate
Velocity, increment length and radius	v, l, r	Known
Wheel radius, differential ratio, Drag coefficient, frontal area, air density	wr, I, Cd, A, ρ	Parameters defined in the first step of the code
Time increment of the point	t	$t = \frac{l}{v}$
RPM motor	rpm	$rpm = \frac{v \cdot i \cdot 60}{2 \cdot \pi \cdot wr}$
Longitudinal acceleration	a_{long}	$a_{long} = \frac{\Delta v}{\Delta t}$
Lateral acceleration	a_{lat}	$a_{lat} = \frac{v^2}{r}$
Drag force	F_{drag}	$F_{drag} = \frac{1}{2} \cdot Cd \cdot A \cdot \rho \cdot v^2$
Motor torque	τ	$\tau = \frac{(a_{long} \cdot m + F_{drag}) \cdot r}{i \cdot numb\ of\ motors}$
Power output of the motor	P	$P = \frac{\tau \cdot rpm \cdot 2 \cdot \pi}{60}$

The outputs of the Endurance race are used for the following model, the Electric System Simulation model. This model requires three main inputs: The speed of the motor, the torque of the motor and the power output of the motor.

4 Electric System Simulation

4.1 System design

In order to analyse the dynamic event of the Efficiency (an event which can award up to 100 points), a Simulink model was created from scratch. As explained before this event is bound to the Endurance race. The power consumption of the vehicle is studied during the Endurance race. It is also important to highlight that the capacity of recovering energy by braking with the electric motor (regenerative braking) will also mean a more efficient car.

Still, the aim of this part is not only to check the efficiency of the vehicle, but also to be able to quantify the total energy that must be delivered by the battery. As so, the battery pack will be quantified in terms of minimum energy stored. The battery cells model and brand have already been selected, hence the number of cells will be decided from the results of this model. Furthermore, the current supplied by the battery needs to be monitored, in order to design the battery pack depending on the limitations of the chosen battery cells.

This project is about powertrains which consist of four main parts of an electric vehicle. The electric traction motor, the transmission (drive shafts, gears and differentials), the inverter and the battery pack.

The motion of a vehicle happens thanks to the force delivered to the road, so basically the force that the tires make on the surface. This energy ultimately delivered to the wheels goes through several steps before arriving to the wheels. The next figure illustrates the energy path:

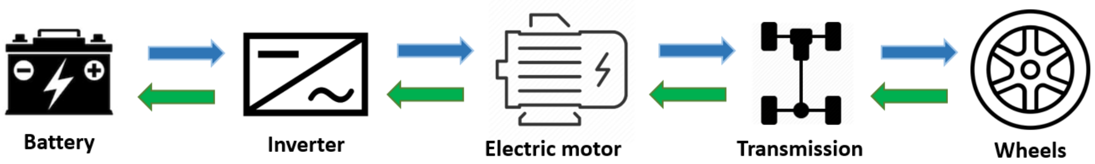


FIGURE 24: *ENERGY CONVERSION THROUGH THE POWERTRAIN*

Each of these elements has a certain efficiency, which means that not all energy that is delivered by the battery will arrive to the wheels. The aim of this Simulink model of the electric system is to analyse the power loss in each of these elements of the powertrain. The transmission is the only part that is not modelled here, since it was already taken into account in the vehicle dynamics simulation code, which provides the necessary input data to the electric system model.

The functionality of the model is shown in the following figure:

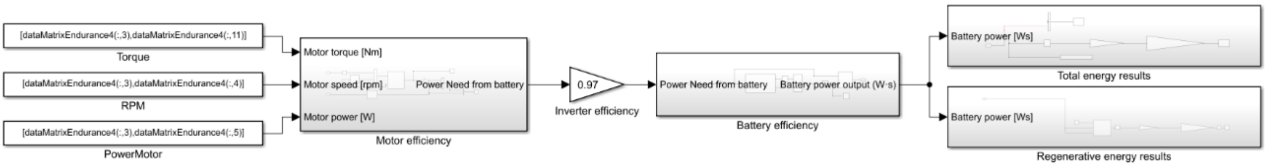


FIGURE 25: *ELECTRIC SYSTEM SIMULATION MODEL*

As it can be seen in the **Figure 25** the model works backwards to be able to know how much energy the battery needs to deliver. In other words, the model acquires the power need from the motor, and then calculates the amount of power that is lost in the system. Adding this loss to the power need from the motor we are able to calculate the need of energy storage in the battery.

The electric system Simulink model begins with the data generated from the vehicle dynamics simulations. It uses three inputs, the motor torque, the motor speed and the motor power. The first two inputs are used to recognize the working point of the motor in the motor efficiency map. With the efficiency the power lost in the motor is calculated. The inverter is considered to have a constant 97% efficiency at any power from the motor.

A battery model was generated to identify the power lost on the battery pack. This model is explained deeper on the following sections.

On the right part of the **Figure 25** the results are analysed. The upper block refers to the Total energy results, which is basically a comparison of the total power output from the battery and the total power that the car delivers to the wheels. Briefly, the total power versus the total power minus the power lost. With this power we can easily know the total energy need for each of the powertrains, just by integrating the power, since the power is the rate of using energy.

The bottom block corresponds to the Regenerative energy results, which basically provides us the powertrain which is able to recover more energy.

4.2 Efficiency map

By using the vehicle dynamics simulations model data, we are able to know how much power the motor needs to deliver to the wheels. However, the motor itself has some losses. To identify these losses, we have to keep in mind that the motors that were designed for the powertrain configurations were permanent magnet synchronous motors.

The losses on the motor are modelled with the use of an efficiency map. This map provides the efficiency at which the motor operates at a certain speed and torque level. Such maps are normally given by the motor manufacturer. In this project, different motors were used but the servomotor from AMK Group did not include its efficiency map. Therefore, it was required to create an efficiency map from scratch.

The motor lacking an efficiency map is an AMK DT7-75-20-xxW-3500. Based on the technical data provided on the catalogue we are able to find the types of losses there exist and quantify these. AMK Group provides the following information:

Motor type	Standstill data		Rating data					Maximum data		Electrical data		Mechanical data				
	M_0 [Nm]	I_0 [A]	M_N [Nm]	P_N [kW]	I_N [A]	n_N [rpm]	k_T [Nm/A]	M_{max} [Nm]	I_{max} [A]	L_{tt} [mH]	R_{tt} [Ω]	n_{max} [rpm]	J [kg cm²]	L [mm]	L_{BR} [mm]	m [kg]
DT7-75-20-xxW-3500	75	51	66	21	48	3000	1.48	120	99	1.25	0.294	3400	55	298	342	3

FIGURE 26: *TECHNICAL DATA OF THE AMK DT7-75-20-XXW-3500 MOTOR* **[39]**

There are two main type of losses that may be identified:

- **Winding loss**: Losses are produced when the current flows through the three phases winding, so it is called winding loss. The resistance of each phase is recognized as a fixed value, as it can be seen on **Figure 26**, R_{tt} equals 0,294 Ω. In the rating data from the above mentioned table it is also mentioned the torque to current ratio, which corresponds to k_t=1,48 Nm/A. With this data the windings losses can be written as a simple resistance loss, where $P_{\_cu}=R\cdot I^2$ and $I=T\cdot k_t$ ($P_{\_cu}$ is the windings loss, R is the electrical resistance, I is the current, T is the torque of the motor and k_t is the constant that links torque and current). This power loss is shown on the left side of **Figure 27**. **[32]**

- **Iron loss**: There are several different methods to calculate the iron losses in a motor, based on the distribution of magnetic induction. The total iron losses are the sum of the hysteresis and eddy currents loss components:

Hysteresis losses are related to the movement of the magnetic micro-domains in a ferromagnetic material, when they reorient in the presence of an external magnetic field. **[33]**

Eddy current losses are triggered by the external variable magnetic field, which causes an induced current in the iron core of the machine, since it is a conductive material. Since there is no information about the iron losses in the technical data sheet of the motor, the modelling of these losses were based on previous motor design experience; when operating at the rated power (rated torque and speed), hysteresis and eddy current losses were estimated to 0,5% and 1% of the rated power respectively.

The following image shows the winding losses and the iron losses, respectively:

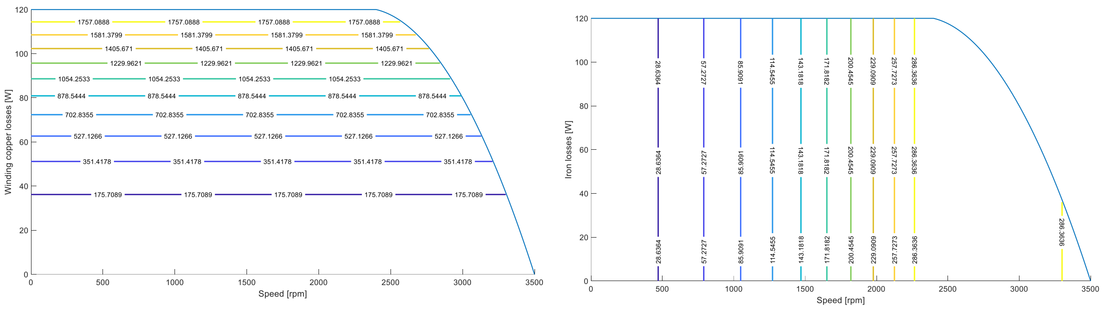


FIGURE 27: *WINDING LOSSES IN THE LEFT AND IRON LOSSES IN THE RIGHT FOR THE MOTOR AMK DT7-75-20-XXW-3500*

By adding both of these losses and comparing them to the power output from the motor at each point, we are able to draw the efficiency map:

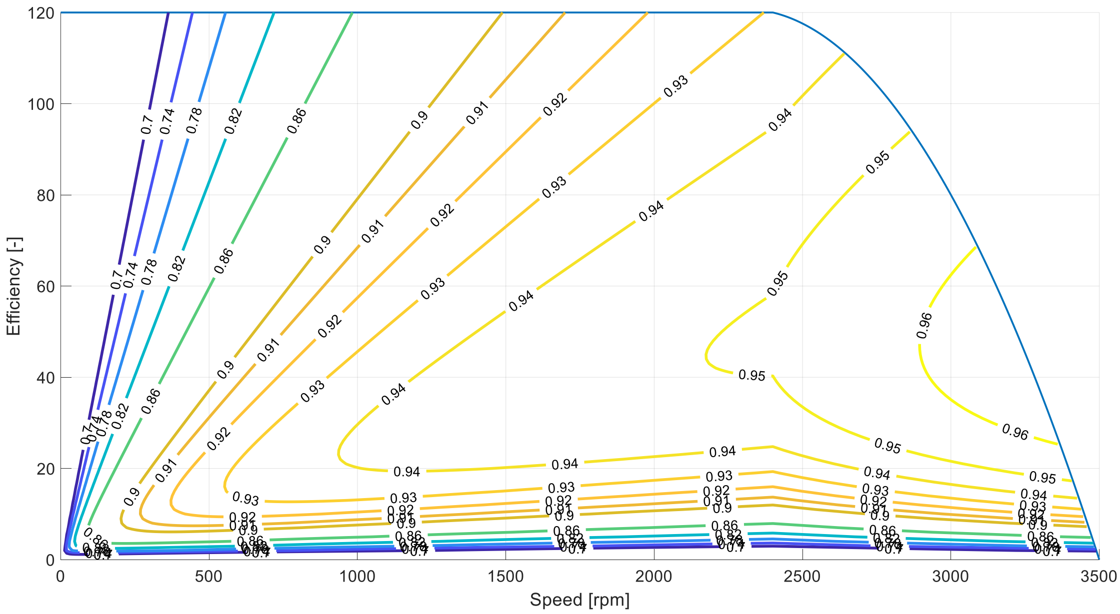


FIGURE 28: *EFFICIENCY MAP OF THE AMK DT7-75-20-XXW-3500 MOTOR*

4.3 Motor efficiency

Once the efficiency maps are created, they are integrated into the electric system model in Simulink.

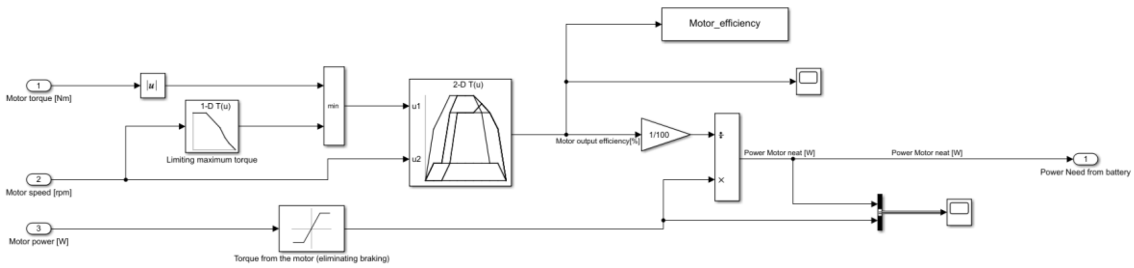


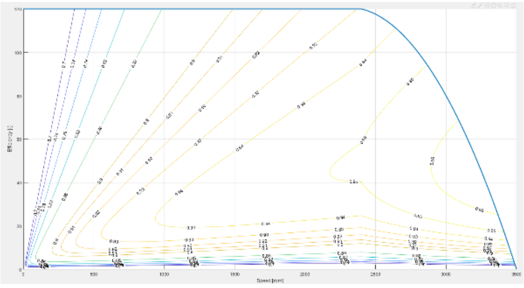
FIGURE 29: *MOTOR EFFICIENCY SIMULINK MODEL*

The first step is the "Limiting maximum torque" block. This block enables the model to filter the torque demands which overcome the maximum torque that the motor can deliver. This may happen when the car is braking, where the braking force of the vehicle is higher than what the motor can provide. In this case, conventional friction brakes must be applied too.

The "Efficiency map" block, is a 2 dimensional table representing the efficiency map as created in the previous section. In **Table 6**, placed below, a comparison between the efficiency maps and the corresponding table can be seen. It should be noted that the first efficiency map was the one that was self-made, while the other two are obtained from the manufacturer's website.

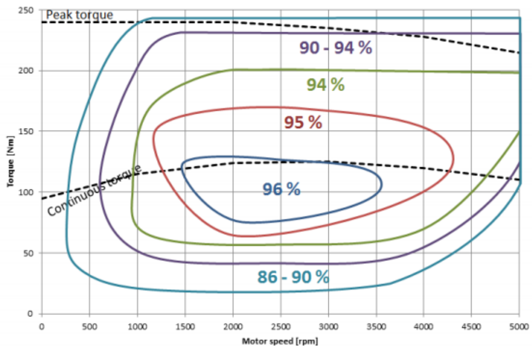
TABLE 6: *MOTOR EFFICIENCY MAP (LEFT), MOTOR EFFICIENCY TABLE (RIGHT). ALL THE GRAPHS ARE TORQUE/RPM, BEING TORQUE THE Y AXIS AND RPM THE X AXIS.*

AMK DT7-75-20-xxW-3500



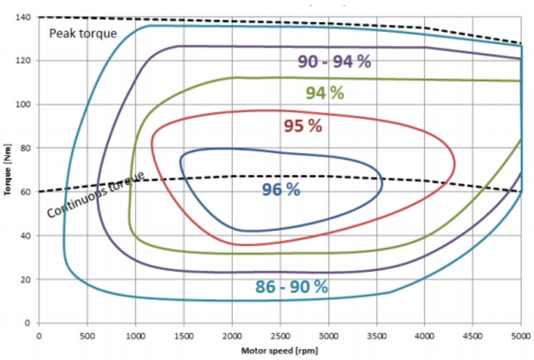
120	55	76	86	90	92	93	94	94
100	61	79	88	91	93	94	94	95
80	64	82	90	92	93	94	95	96
60	70	86	91	93	94	95	96	96
40	76	89	93	94	94	95	96	96
20	80	92	94	94	93	93	94	95
0	88	89	88	88	87	89	87	87
	0	500	1000	1500	2000	2500	3000	3500

EMRAX 228



140	86	86	86	86	86	86	86	86
120	86	88	92	92	92	92	92	92
100	86	88	94	94	94	94	94	94
80	86	88	94	96	95	95	95	95
60	86	88	94	96	96	94	94	94
40	86	88	94	94	94	88	88	88
20	86	88	88	88	88	86	86	86
0	86	86	86	86	86	86	86	86
	0	1000	2000	3000	4000	5000	6000	7000

EMRAX 208



250	86	86	86	86	86	86	86	86	86	86	86
225	86	88	92	92	92	92	92	92	92	92	92
200	86	88	94	94	94	94	94	94	94	94	94
175	86	88	94	96	95	95	95	95	95	95	95
150	86	88	94	96	96	96	94	94	94	94	94
125	86	88	94	96	96	96	94	94	94	94	94
100	86	88	94	96	96	96	94	94	94	94	94
75	86	88	94	94	94	94	88	88	88	88	88
50	86	88	88	88	88	88	86	86	86	86	86
25	86	88	88	88	88	88	86	86	86	86	86
0	86	86	86	86	86	86	86	86	86	86	86
	0	500	1000	1500	2000	2500	3000	3500	4000	4500	5000

The bottom input “Motor power” from the Simulink model seen in **Figure 29**, is followed by a block used to limit the power delivered by the motor. As explained in the previous sections, the motor is limited by the FS rules competition, which dictates that the motor output can be maximum 80kW in total. This block limits the power delivered from the motor so that in the case that the car is braking, the total does not exceed the motor power capacity. **[34]**

4.4 Battery efficiency

With the power output from the motor, and with the inverter efficiency also included, it is time to take a look at the final part of the model, the battery model. There are several methods to model a battery with different degrees of complexity depending on the purpose of the model. A simple modelling approach which gives sufficiently accurate results for our analysis consists of an ideal voltage source and an internal resistance. **[35] [36]**

The battery is considered to be a source of power, providing power to an external load. The battery is modelled by an ideal voltage source and an internal resistance, where some of the power is lost. The following image is the representation of this model:

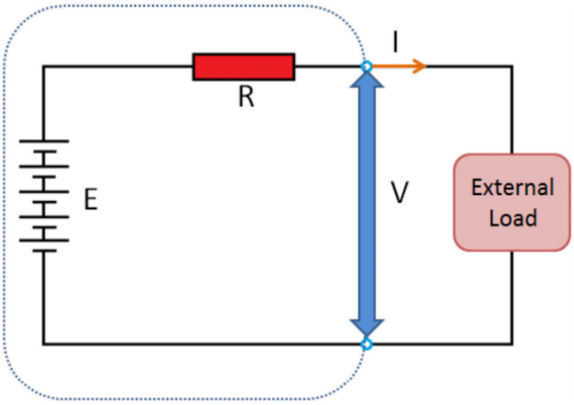


FIGURE 30: *SIMPLE INTERNAL RESISTANCE MODEL* **[36]**

In **Figure 30** R represents the internal resistance of the battery modelled, V is the battery voltage at the terminals, and I is the battery current, and E is the energy source (at potential V_{oc}) of the battery.

Then just by applying Kirchhoff laws the power lost by the impedance can be calculated, as well as the power that has to be delivered by the battery:

$$\mathrm{P} = \mathrm{I} \cdot \mathrm{V}_{OC}$$
$$\mathrm{P}_{loss} = \mathrm{I}^2 \cdot \mathrm{R}_{int}$$

The power output, or in the case of regenerative braking, input for the battery, can be calculated by:

$$P_{battery} = P + P_{loss}$$
$$P_{battery} = \mathrm{I} \cdot \mathrm{V}_{OC} + \mathrm{I}^2 \cdot \mathrm{R}_{int}$$

Where P is the power needed by the external load. Since the current is the only unknown parameter, it can be calculated from the previous equation as:

$$I = \frac{-V_{OC}}{2 \cdot R_{int}} + \sqrt{\frac{{V_{OC}}^2 - 4 \cdot R_{int} \cdot P_{battery}}{4 \cdot {R_{int}}^2}}$$

The battery model implemented in Simulink is presented in **Figure 31:**

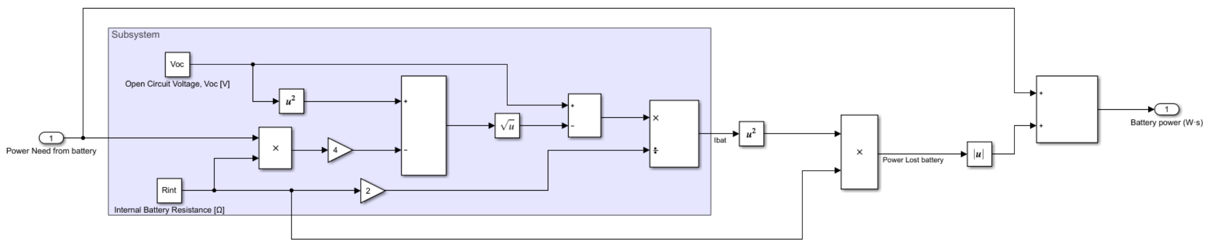


FIGURE 31: *SIMPLE INTERNAL RESISTANCE SIMULINK MODEL*

There are two numerical variables in this model, the Open Circuit Voltage (V_{oc}) and the Internal Battery Resistance (R_{int}). For the first one, the V_{oc} is equal to 518 V. This value corresponds to the voltage of the LFS20 battery. To estimate the value of the internal resistance, it was considered that the maximum loss of the battery was 10% of power at the maximum power rated points.

What is really interesting about this model is that the current supplied by the battery is also obtained. With the behaviour of the current in the battery, the design of the battery pack can be made. It is important that the current does not exceed the maximum discharge rate when it is acting as a motor, and it cannot exceed the maximum charge rate when acting as a generator.

4.5 Battery sizing model

The last part of the model consists of adapting the results acquired.

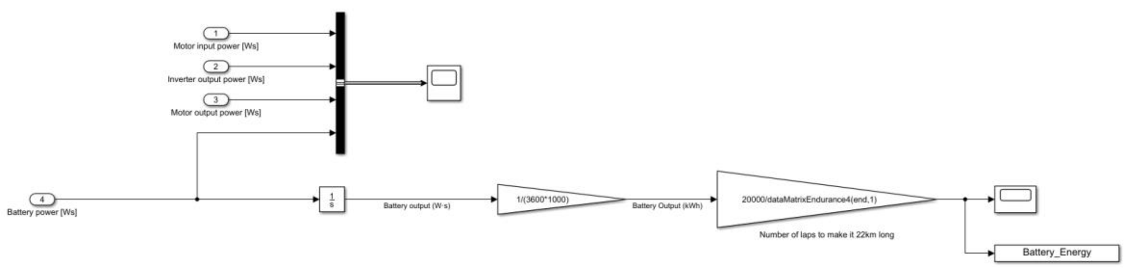


FIGURE 32*: BATTERY SIZING MODEL*

This part of the model includes two main results. The first scope is used to compare the different losses throughout the different powertrain parts, and is shown in **Figure 32**. Basically it shows how much power is lost during the outcome of the model.

The second part is used to model the battery itself. Integrating the total power output from the battery, the total energy that the battery has to deliver is calculated. Since the Endurance race modelled in the vehicle dynamics simulation only includes 1 lap, which is approximately 1,6 km long, the obtained energy consumption must be scalped up accordingly, and converted to KWh for simplicity. This will give the required battery size for the simulated powertrain concept.

4.6 Battery energy recovering model

In order to compare the capability to recover energy from the different alternatives, the following blocks are added to the model:

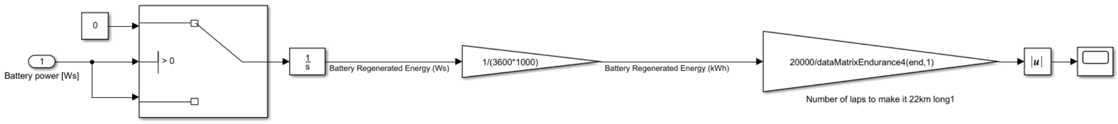


FIGURE 33: *BATTERY ENERGY RECOVERING MODEL*

This section of the model is similar to the previous one, but it has a switch to only account for when the battery power output is negative (the battery is charging with the braking energy). Then again, the last blocks are used to scale the results for a 22km race.

5 Simulations and results

As stated in the introduction part the main goal of this project is to evaluate each of the powertrains considered and decide which one is the most likely to be implemented in a real scenario. In this part, the results of the software simulation are shown, and compared among them. Also a qualitative part (which does not involve numeric data) is used to compare each of the powertrains. In the end, the powertrain with the best results is selected, and a preliminary design is proposed in the last part of the work.

5.1 Performance (Vehicle dynamics simulation results)

In order to analyse the performance of each of the drivetrains, the Matlab program described in the Vehicle Dynamics Simulation is used. As explained previously, the most important results from this simulation is the speed at which the vehicle runs the race, so basically the purpose is to recognize the fastest design.

It is also important to note that the FS competition involves different races. Each of the races results contribute differently to the overall score of the team, therefore the comparison of the performance in each of the events will also be following this criterion.

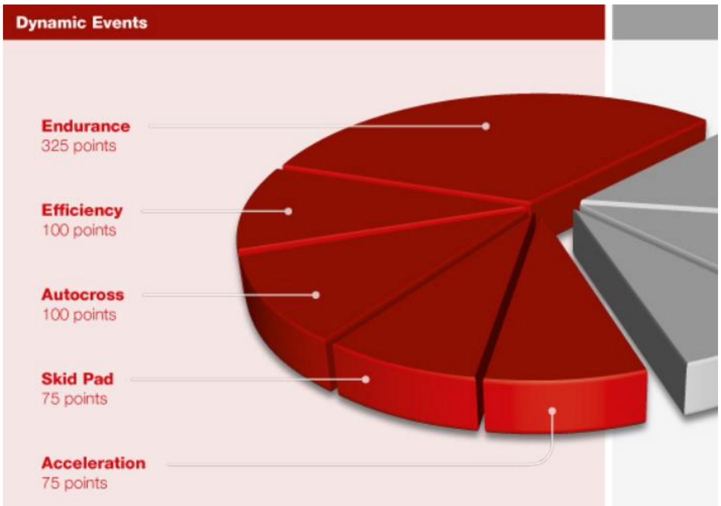
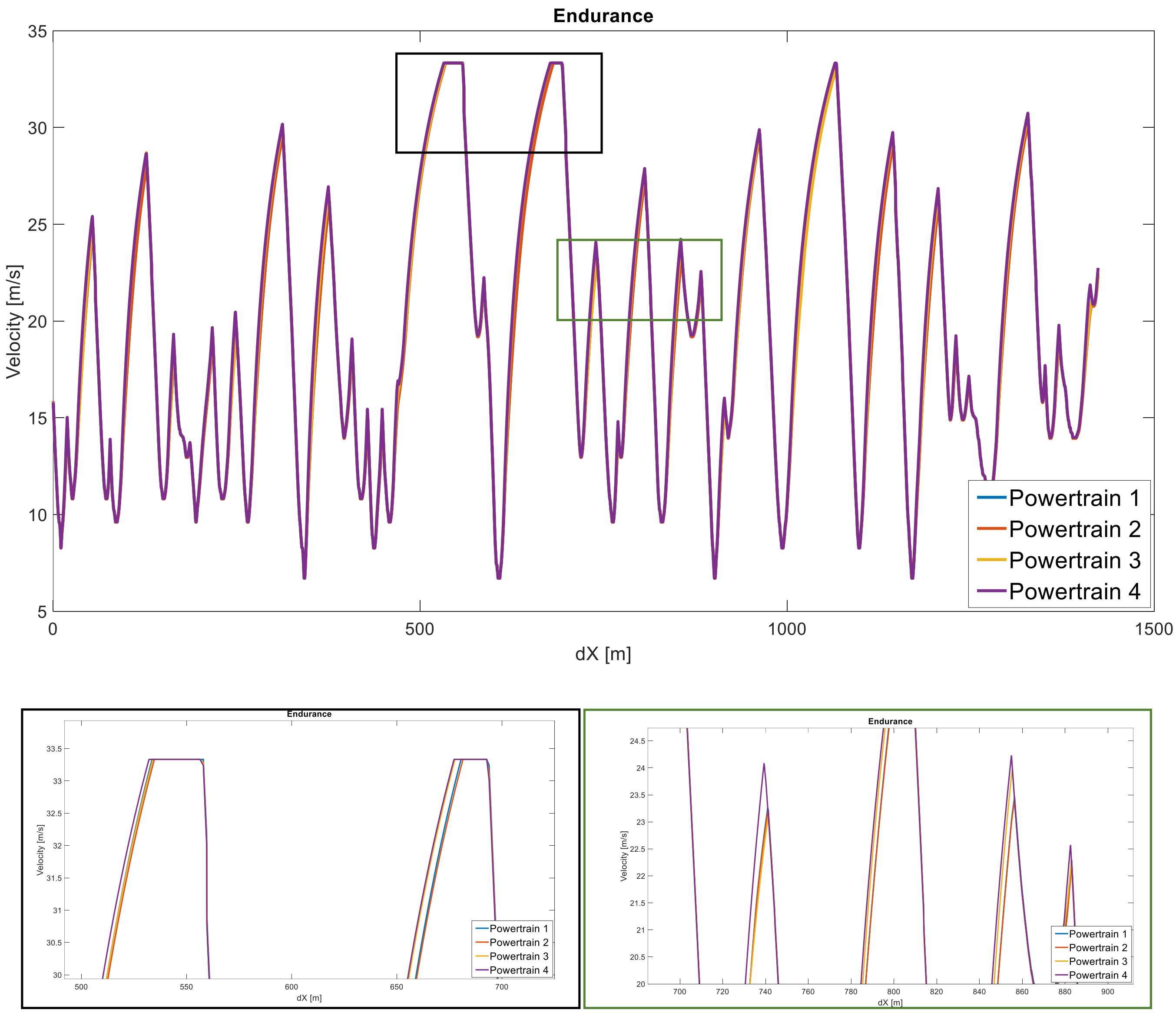


FIGURE 34: *DYNAMIC EVENTS FROM FS COMPETITION AND POINTS AWARDED FROM EACH* **[4]**

5.1.1 Endurance

This is the 22 km race. Since it is the largest, it awards the highest amount of points. The lap data of this track was taken from the competition held in Hockenheim on 2012, in Germany. The race track has about 1,57 km, thus it takes 14 laps to complete the race. The model, only simulates one lap, and since the total circuit involves 14 laps, it is considered that the run lap is not the starting one. Hence the starting velocity is not 0.

The results of this lap are shown in the following image:



Endurance	Powertrain 1	Powertrain 2	Powertrain 3	Powertrain 4
Mean speed	15,9 m/s	15,8 m/s	15,9 m/s	16,07 m/s
Time for one lap	79,8 s	80,0 s	79,7 s	78,9s

FIGURE 35: *ENDURANCE RACE RESULTS*

From the results it is really clear that the fastest vehicle is the one featuring Powertrain 4. The difference with the others is significant, being almost one second faster. Considering that the race takes 14 laps, the difference would be even bigger.

The main reason for being the best powertrain can be seen in the green rectangle, on the bottom right picture. The vertex points are the apexes, where the car goes from accelerating to decelerating. These vertices are higher on the purple line (Powertrain 4), which means that the car is able to reach a higher velocity on cornering phases.

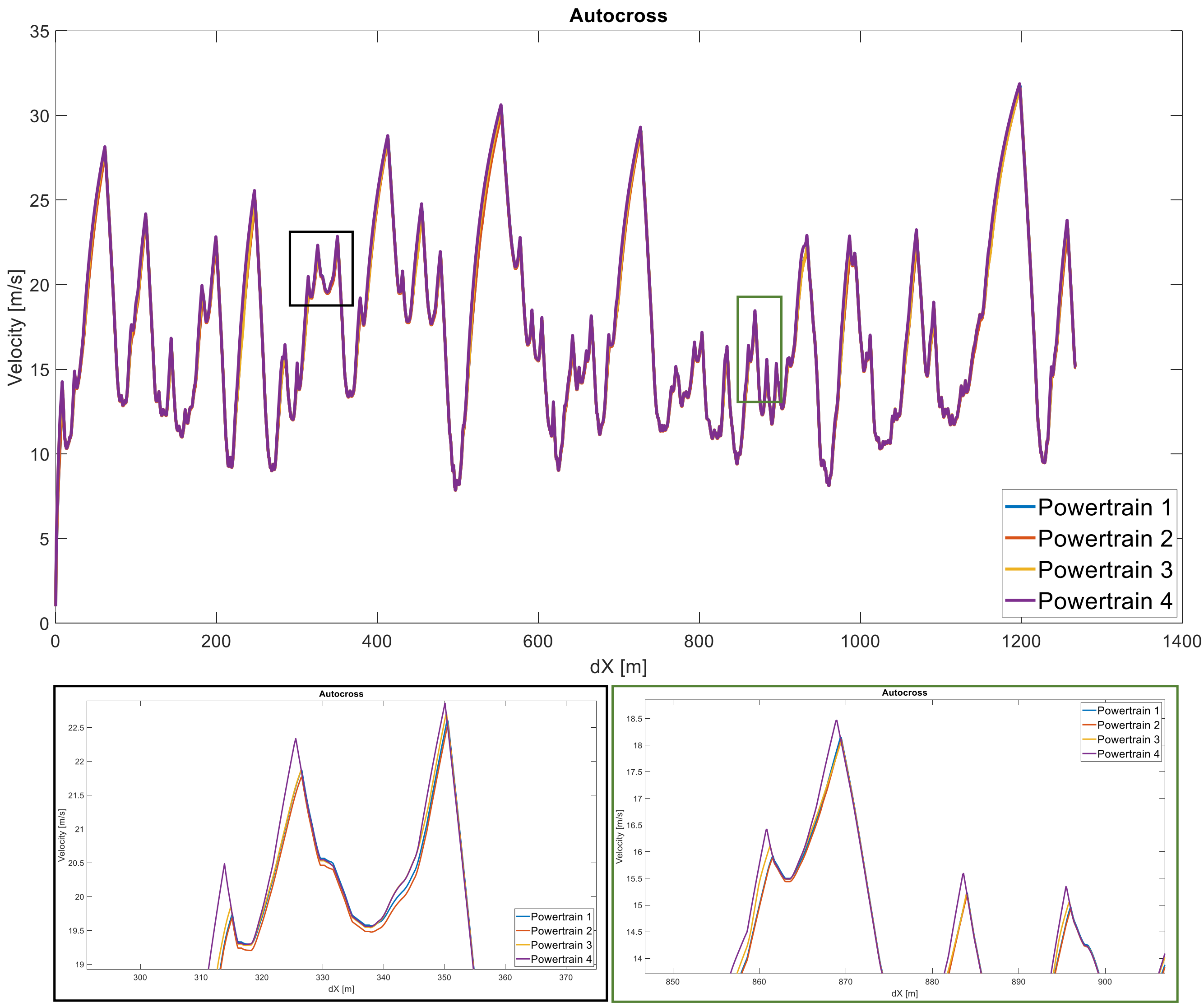
However, the design which is able to reach a high velocity faster is Powertrain 2. This is mainly caused to a bit over dimensioned motors. Furthermore, it gets punished by this over dimensioned system placing last on this race. This fact is highlighted on the black rectangle on the above graph.

The results clearly show that Powertrain 4 is the fastest design, with the higher mean speed. It is important to know, that the main difficulty of this race is to entirely complete it. This is a software simulation, where only the vehicle dynamics are taken it to account. The reliability of the car is not tested; thus this simulation is not enough to conclude that this design would be the best for the race.

5.1.2 Autocross

The Autocross race consists of a one lap track. The outcome of this race depends mainly on the driver's skill and the vehicle dynamics. Considering that the driver skills cannot be controlled in a software simulation, one can say that the results of this race will be very relevant to consider which is the design with the best vehicle dynamics.

The simulation consists of a single lap of an autocross race from a competition celebrated in 2012. It can be seen that, in contrast to the Endurance race, the car does start steady. In the following figure, the results of the simulation are shown:



Autocross	Powertrain 1	Powertrain 2	Powertrain 3	Powertrain 4
Mean speed	16,0 m/s	15,9 m/s	16,0 m/s	16,1 m/s
Time for one lap	79,4 s	79,73 s	79,4 s	78,5 s

FIGURE 36 *AUTOCROSS RACE RESULTS*

The results are similar to the previous event. Powertrain 4 places first again. The difference between the winner and the rest is a bit tighter, but still not negligible. Again, Powertrain 4 reaches the finish line eight tenths of a second faster.

The reason behind being the winner design again is similar to the previous race. The purple line is higher on almost the whole race. Repeatedly, the apex points are reached on a higher velocity, which consequently mean a faster car on track points near the apexes. This fact can be seen in both green and black rectangles on the above figure.

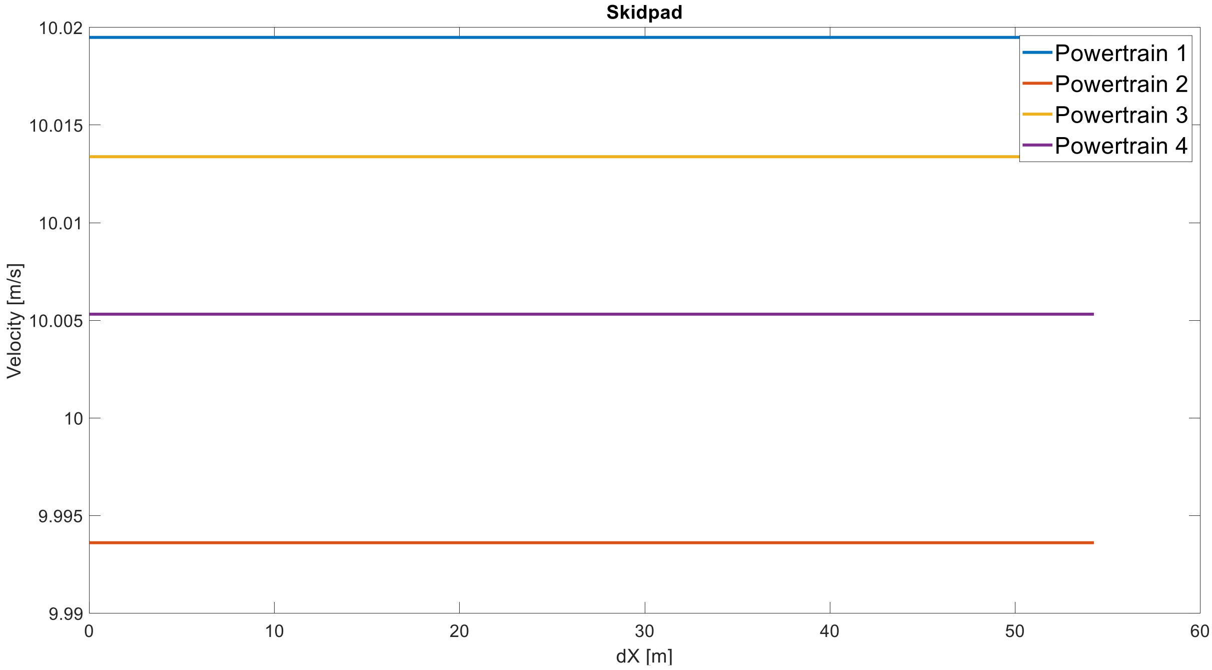
The good results for this powertrain are directly related to the gg diagram explained previously in chapter 3.2, which is the basis of the software model. Having a wider diagram has a direct influence in the results, enabling the model to reach a higher velocity on cornering points, where the lateral acceleration is higher for the same velocity, or in other words, reach a higher velocity on a demanding hard turn.

As explained in the introduction of this race, it can be considered than the winner of this race will be considered the alternative with the best vehicle dynamics. Thus, powertrain 4 is so far the most prepared for tougher tracks.

5.1.3 Skidpad

The Skidpad race is a race where the car has to run a full right hand circle and then another circle in the other way. The simulated model consists of a track that has an increment with a constant radius. The software calculates the optimal velocity at which a car runs on a turn for a certain radius. Since the track is a whole circle, the radius of each step is the same. It is considered again that the car does not start steady.

The results for this race are shown in the below picture



Skidpad	Powertrain 1	Powertrain 2	Powertrain 3	Powertrain 4
Mean speed	10,02 m/s	9,99 m/s	9,99 m/s	10,01 m/s
Time for one lap	4,87 s	4,89 s	4,89 s	4,88 s

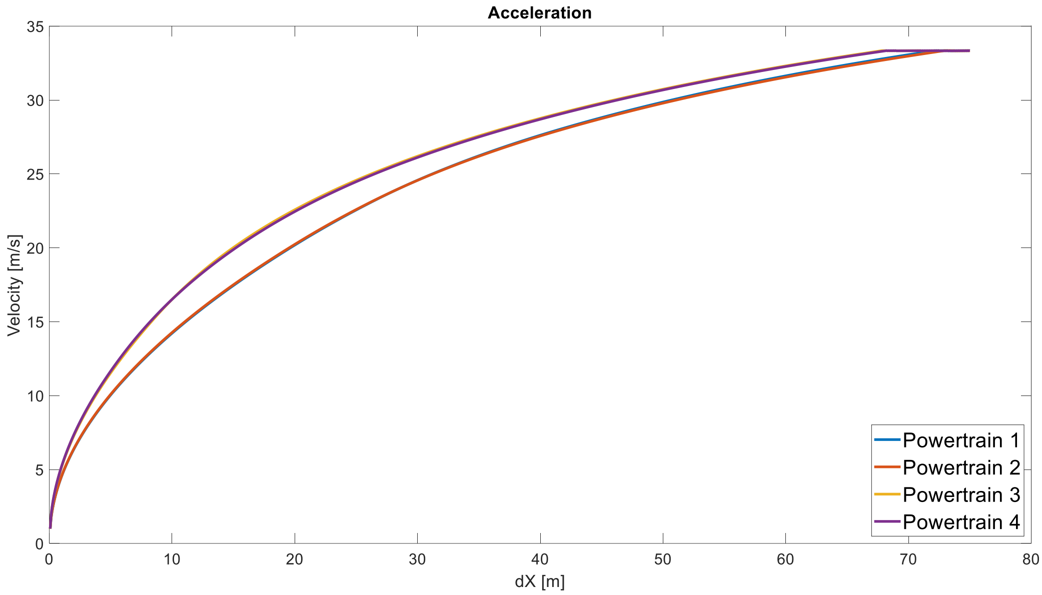
FIGURE 37: *SKIDPAD RACE RESULTS*

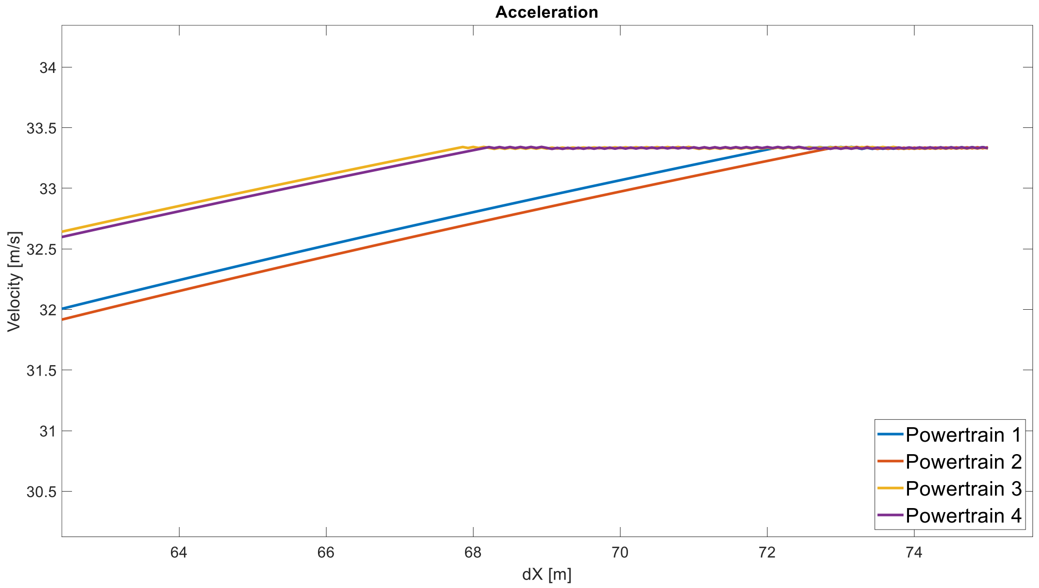
This race shows a different winner than the others. The results however, are really close to each other. Although Powertrain 3 is the winner of this race, there is no real conclusion that can be withdrawn from this simulation, since the difference among all powertrains is lower than 0,01 seconds.

5.1.4 Acceleration

The acceleration race consists of a 75 meters straight run. The winner of the race will show the powertrain with the biggest tractive system and/or the lighter design.

The results of this race are shown in the following graphs:





Acceleration	Powertrain 1	Powertrain 2	Powertrain 3	Powertrain 4
Mean speed	24,6 m/s	24,5 m/s	25,7 m/s	25,7 m/s
Time for one lap	3,05 s	3,06 s	2,91 s	2,92 s

FIGURE 38: *ACCELERATION RACE NUMERICAL RESULTS*

For the Acceleration race, the fastest car is Powertrain 3. As seen **Figure 37**, the fastest car during the acceleration race is the one which reaches the maximum velocity of 120km/h or 33,33 m/s first. In this case both Powertrain 3 and 4 are 0,15 seconds faster than the other two.

It was said that the explanation to be the leader of this race was either the propulsion system or the weight of the vehicle. Taking a look at the weight of each powertrain:

TABLE 7: *WEIGHT OF EACH CAR DESIGN*

	Powertrain 1	**Powertrain 2**	**Powertrain 3**	**Powertrain 4**
Mass	302,7 kg	308 kg	311 kg	314,7 kg

It can be seen that the results are inversely proportional to the mass of each vehicle. This means that the influence of the force from the motors is more relevant than the weight. Thus, Powertrain 3 has the higher tractive force, followed closely by Powertrain 4.

5.2 Power efficiency (Electric system simulation results)

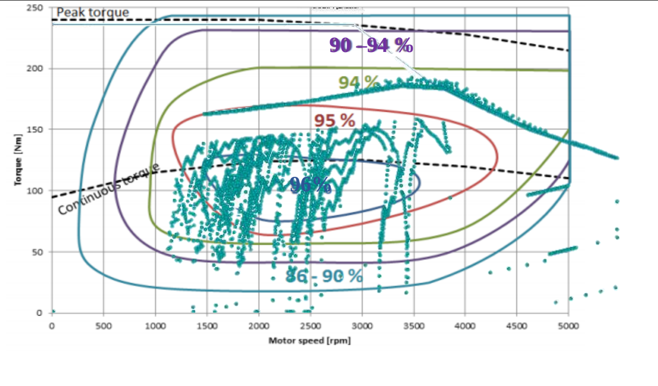
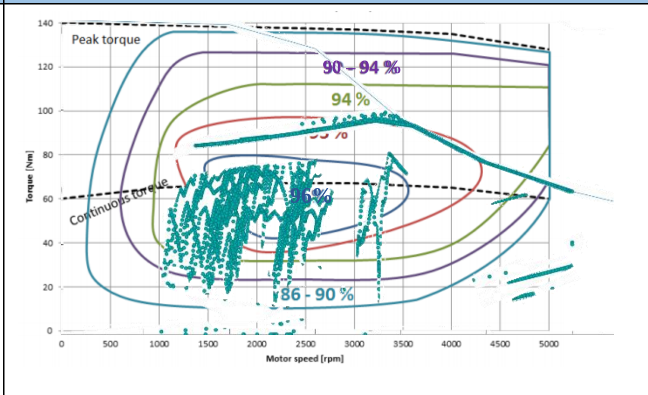
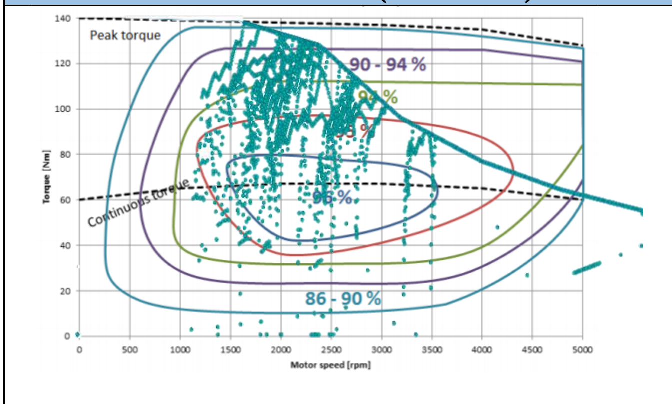
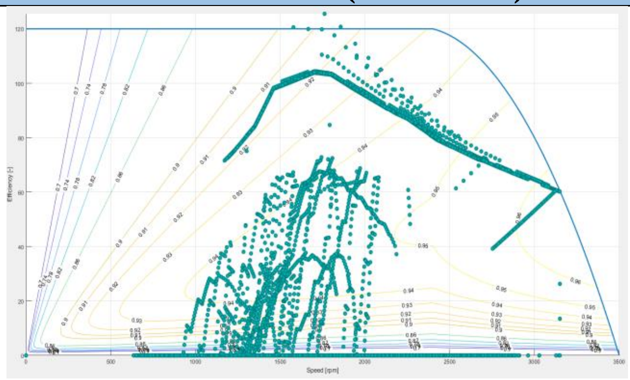
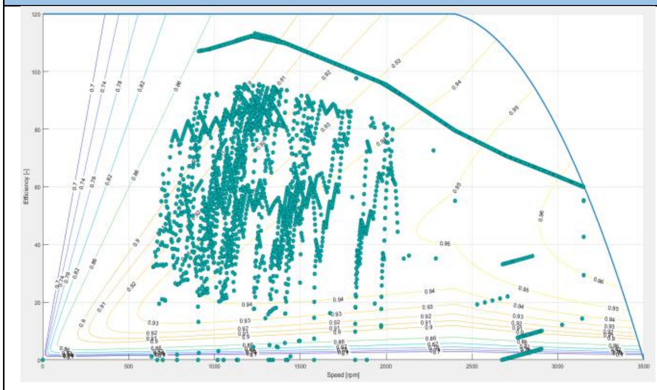
5.2.1 Motor efficiency

In this part, it will be analysed how efficient are the motors in each of the powertrains. To do so, the efficiency maps presented in section 4.3 are used. The results show the working points at each time step, overlapping the working points on the efficiency maps.

The race used to test this motor efficiency is the endurance race. The efficiency event results are based on the energy consumption (total energy consumed from the battery), that is why if the motor runs at points with higher efficiency the energy consumption will be lower, and so more points will be awarded at this event.

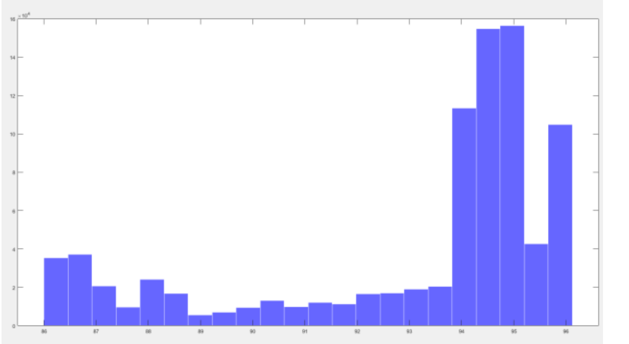
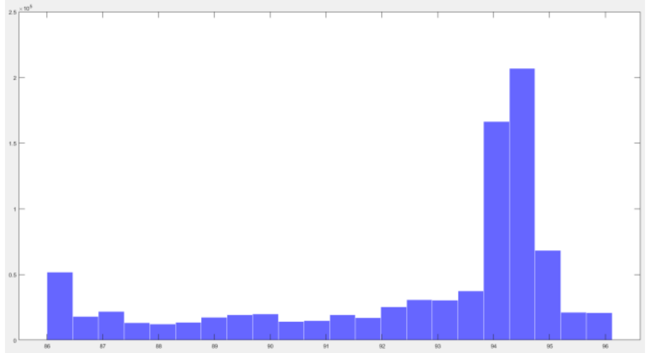
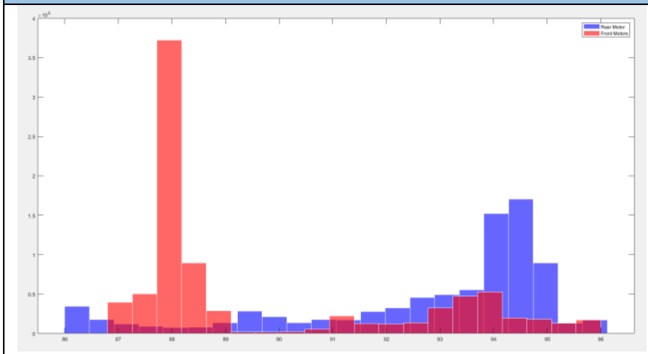
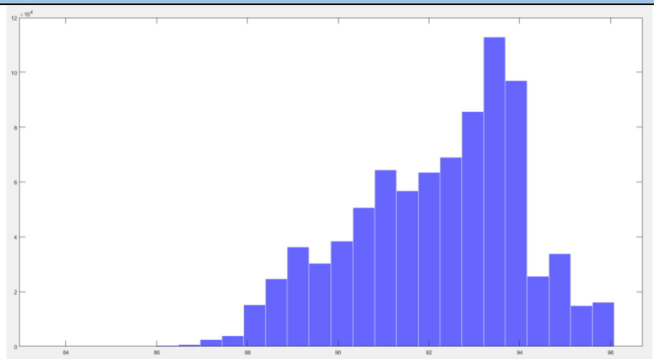
The results for each powertrain are shown in the following graphs:

TABLE 8: *TORQUE VS MOTOR SPEED WORKING POINTS FOR EACH POWERTRAIN*

<table>
<tr><th>POWERTRAIN 1</th><th>POWERTRAIN 2</th></tr>
<tr><td>
</td><td>
</td></tr>
<tr><th>POWERTRAIN 3 (rear motor)</th><th>POWERTRAIN 3 (front motors)</th></tr>
<tr><td>
</td><td></td></tr>
<tr><th>POWERTRAIN 4</th><th></th></tr>
<tr><td></td><td></td></tr>
</table>

To have a clear picture of how much time the motors spend at the different efficiency levels, some histograms are plotted as it can be seen in **Table 9**.

TABLE 9: *MOTOR EFFICIENCY FOR EACH POWERTRAIN*

POWERTRAIN 1	POWERTRAIN 2
	
Efficiency mean = 93,1%	**Efficiency mean = 92,6 %**
POWERTRAIN 3	**POWERTRAIN 4**
	
Efficiency mean front motors= 89,7% **Efficiency mean rear motor= 92,8%**	**Efficiency mean = 92,2%**

The following table summarises the efficiency of each motor when included in a certain powertrain configuration. The last column shows the overall efficiency of the motors in the powertrain.

TABLE 10: *MOTOR EFFICIENCY FOR EACH POWERTRAIN*

	Emrax 208	Emrax 228	AMK DT7	Mean Efficiency
Powertrain 1	-	93,1%	-	**93,1%**
Powertrain 2	92,6%	-	-	**92,6%**
Powertrain 3	92,8%	-	89,7%	**91,2%**
Powertrain 4	-	-	92,2%	**92,2%**

From these results, it is clear, that the Emrax motors work at a higher efficiency. Taking a look at the efficiency of the front motors in powertrain 3, it can be clearly seen, that the way this configuration was proposed is not the most optimal. As explained before, the rear motor at this configuration works at full load, while the ones in the front are only used if necessary.

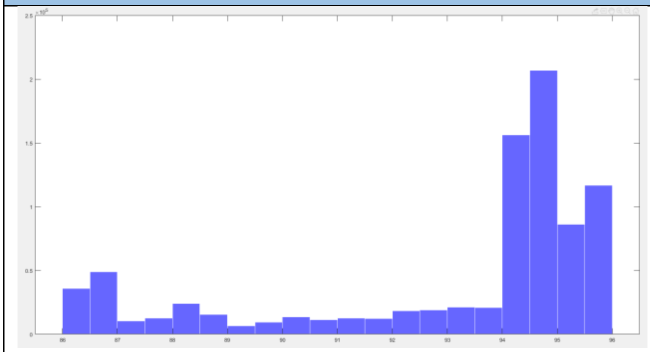
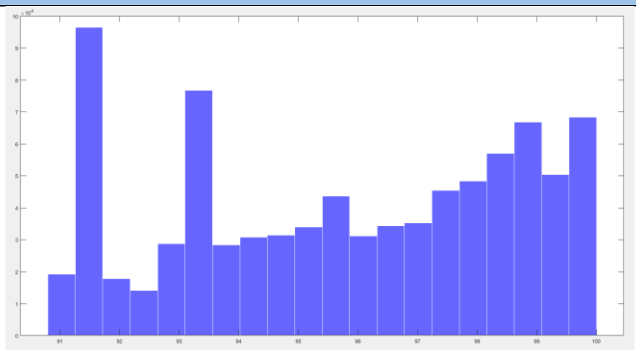
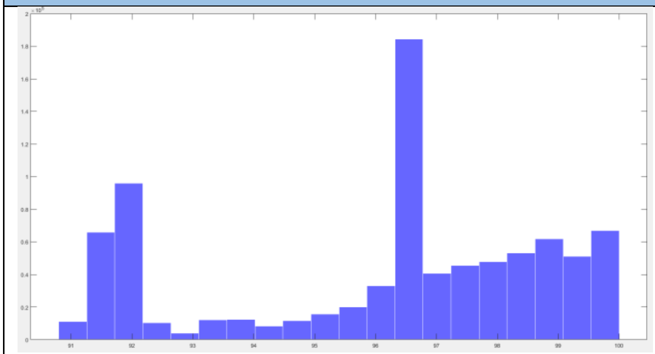
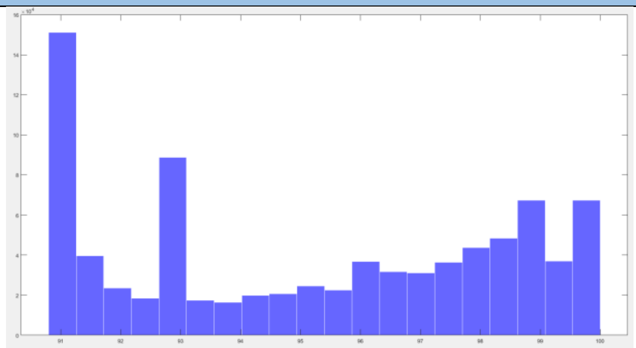
It can be clearly seen that powertrain 1 is the best configuration in terms of efficiency, this is probably caused by the fact that its efficiency map is very well suited for the case.

5.2.2 Battery efficiency

In order to have an overall look of the powertrain efficiency, the battery efficiency was modelled for each powertrain. As said previously, the battery efficiency model is based on the loss of an internal resistance.

In the following table, the battery efficiency results are shown:

TABLE 11: *BATTERY EFFICIENCY OF EACH POWERTRAIN*

POWERTRAIN 1	POWERTRAIN 2
	
Efficiency mean = 95,8%	**Efficiency mean = 95,8%**
POWERTRAIN 3	**POWERTRAIN 4**
	
Efficiency mean = 96,1%	**Efficiency mean = 95,2%**

It can be seen from the results that the most efficient model in terms of power battery is Powertrain 3. However, the difference among the different configurations is not really high.

5.2.3 Battery size

In this section, the energy needed by each of the different powertrains are shown. The following graphs represent the total energy needed from the battery at each time step. The results will be used to identify the total energy needed to complete the endurance race, hence the total energy that the battery is required to store.

When the curves shown in the different graphs have a negative slope (decreasing the energy needed) it means that the vehicle is regenerating energy at that particular time instant. A positive slope occurs when the battery is discharging and proving energy to the traction system

TABLE 12: *ENERGY REQUIRED TO COMPLETE THE ENDURANCE RACE BY EACH POWERTRAIN*

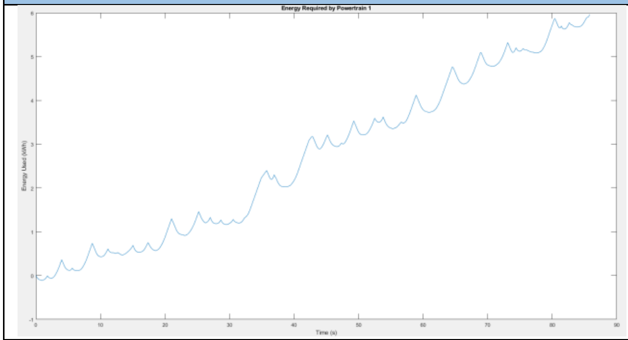
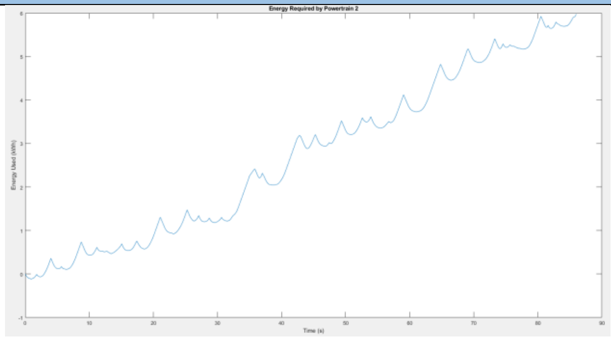
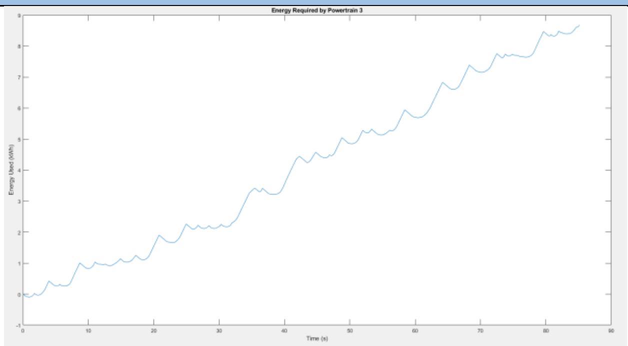
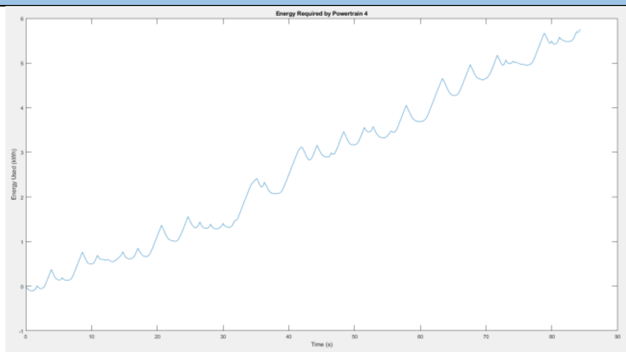
POWERTRAIN 1	**POWERTRAIN 2**
	
Energy required =5,96 kWh	Energy required = 6,00 kWh
POWERTRAIN 3	**POWERTRAIN 4**
	
Energy required = 8,06 kWh	Energy required = 5,72 kWh

Table 12 shows best results for powertrain 4, closely followed by powertrain 1 and powertrain 2. On the other hand, powertrain 3 results are really bad, requiring an amount of energy about a 25% higher than in the rest of the cases. The results show that the size of the battery required for Powertrain 3 would be enormous, and maybe make it non-viable.

Conversely, powertrain 4 would need a smaller battery than the rest of the cases, making it then the most efficient overall system. Although it has the highest internal resistance in the battery model, and the worst efficiency in the motor (compared to the other two closer configurations) it has the best results. This can be caused by two main reasons:

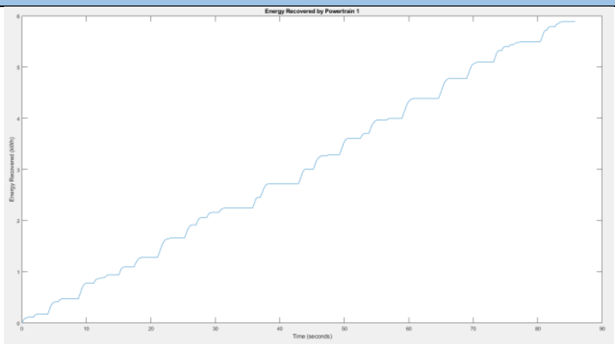
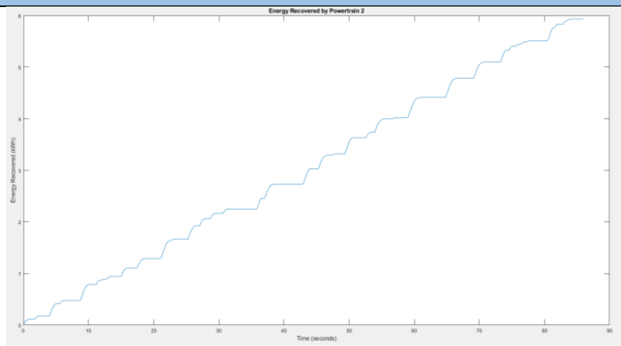
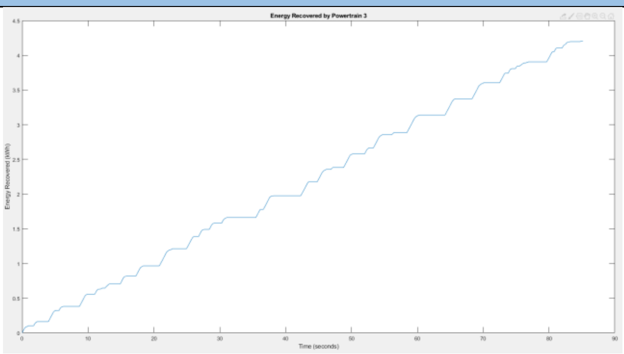
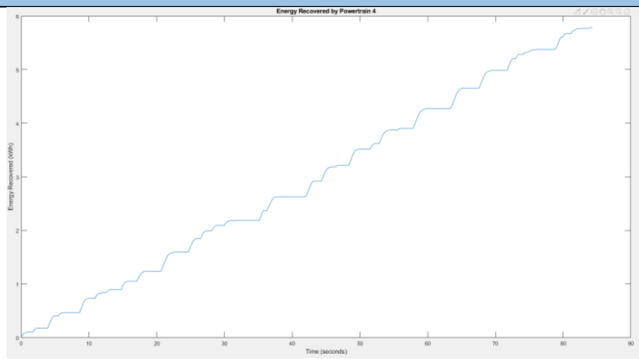
- **The timing of this vehicle is lower than in the other cases**. As shown in the results from the performance during the Endurance race, this configuration is one second faster than the others. Taking into account that this model requires the 14 laps to complete the 22km, means that it is 14 seconds faster than the others. This decrease on time is not negligible, and so the energy needed to complete the race is lower.

- **The recovery capacity for this configuration is higher**. As a second possibility it might happen that this configuration has a higher capacity for recovering energy. If this was the case, the total amount of the energy used would be lower if the car is able to recover it faster. A slightly deeper analysis on this is made in the following lines.

5.2.4 Motor energy recovery

In this part, the comparison of the powertrains is done in terms of the energy recovering capability of the motors.

TABLE 13: *ENERGY REQUIRED TO COMPLETE THE ENDURANCE RACE BY EACH POWERTRAIN*

POWERTRAIN 1	POWERTRAIN 2
	
Energy Recovered= 5,89 kWh	Energy Recovered= 5,95 kWh
POWERTRAIN 3	**POWERTRAIN 4**
	
Energy Recovered=4,26 kWh	Energy Recovered=5,80 kWh

It can be seen, that Powertrain 2 is able to recover more energy than the others. This is probably caused by the fact that, the engines used for this powertrain have higher rated torque than in other cases. This means that in this powertrain, the vehicle acceleration is not limited by the power of the motor but by the grip of the tire.

It also has to be pointed out that the first two powertrain concepts have a higher recovering capacity, probably caused by the higher efficiency of the motor. Powertrain 3 is the worst on recovering, probably because it has the lowest motor efficiency.

5.2.5 Powertrain Efficiency

The purpose of this section is to analyse the overall efficiency results obtained from the electric system simulation. It intends to summarize how much of an influence each part's losses have in the overall system. That is why only Powertrain 4 was plotted.

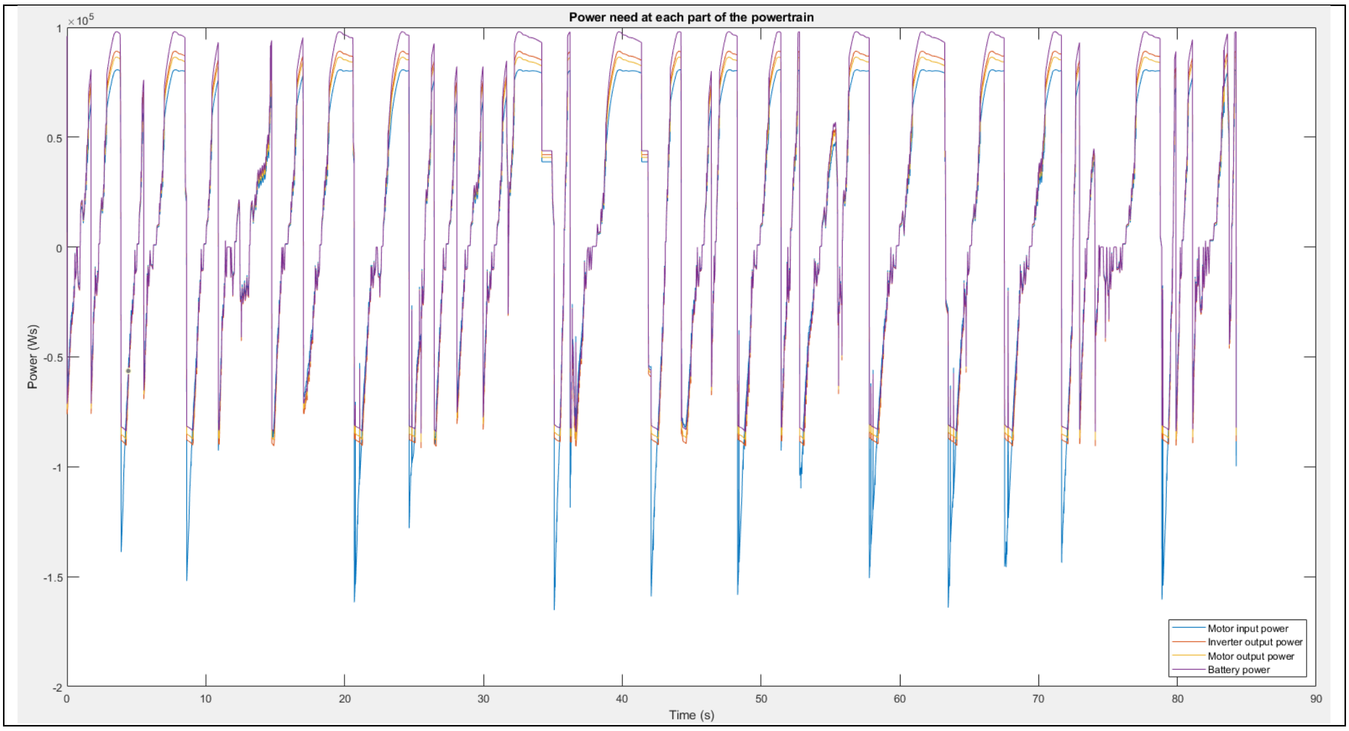


FIGURE 39: *POWER NEED AT EACH PART OF THE POWERTRAIN 4*

In **Figure 39** the input and output power of each of the powertrain parts was plotted. Of course, the output of a previous part is the input of the following one.

It should be pointed out, that the difference between each of the lines corresponds to the loss of power in a concrete part:

- **Motor losses**: Difference between blue line and yellow line
- **Inverter losses**: Difference between yellow line and red line
- **Battery losses**: Difference between red line and purple line

The results of the efficiency from each powertrain item is shown in the following table:

TABLE 14: *EFFICIENCY OF EACH COMPONENT OF THE MODELLED POWERTRAIN BY EACH CONFIGURATION*

	POWERTRAIN 1	POWERTRAIN 2	POWERTRAIN 3	POWERTRAIN 4
Transmission	98%	98%	98%	98%
Motor	93,1%	92,6%	89,7% 92,8%	92,2%
Inverter	97%	97%	97%	97%
Battery pack	95,8%	95,8%	96,1%	95,2%
TOTAL	**84,8%**	**84,3%**	**83,3%**	**83,4%**

It can be seen that both the Powertrain 1 and 2 have a better overall efficient system. Powertrain 3 gets punished by the low efficiency on the motor, while Powertrain 4 also places below on both the motor efficiency and the battery efficiency.

5.3 Design complexity

The qualitative aspects of each powertrain are tricky to analyse and reach any conclusions. However, a comparative assessment can be made to determine which of the powertrains requires a harder design effort.

To properly compare the design complexity, three main aspects will be considered:

- **The differential ratio:** One of the most complex parts of the powertrain will be the reducer. Each of the powertrain have a distinct differential ratio, requiring a individually designed reducer. Higher ratios will require of more complex reducer types, likely with a higher number of steps.

  It should also be considered the fact that powertrains which have motors on the shaft would require a differential. Moreover, the differential would be more complex if the powertrain design requires a higher transmission ratio.

- **The battery size pack**: The second important aspect to highlight is the battery size required. The battery is the part which most likely will be the biggest in the overall design in an electric car. A large battery pack will make it harder for other parts to fit in which would hinder the designing phase.

- **The number of parts the powertrain needs:** The third caspectconsidered, is the number of parts for each powertrain. Obviously a single motor would be simpler to design than a powertrain which involves more parts.

According to all these three aspects, a table summarising the design complexity of the different alternatives is presented:

TABLE 15: *DESIGN COMPLEXITY SUMMARY*

	Powertrain 1	Powertrain 2	Powertrain 3	Powertrain 4
Differential ratio	4,17	4,84	Rear=4,84 Front=2,18	2,18
Battery size	5,96 kWh	6,00 kWh	8,06 kWh	5,72 kWh
Parts of the powertrain	1 motor, 1 inverter, 1 gear reducer + differential	2 motors, 2 inverters, 2 gear reducer	3 motor, 3 inverter, 3 gear reducer + differential	4 motors, 4 inverters, 4 gear reducer

It is really clear that Powertrain 1 requires the simplest design. Then, probably Powertrain 2 is a bit simpler than 4, due to the fact that it requires less parts. However, the reducers needed for Powertrain 4 are likely to be simpler and smaller than for powertrain 2, due to the lower ratio. Powertrain 4 should then go next, and finally Powertrain 3 is the worst scoring poorly in all concepts.

5.4 Cost

The cost of each powertrain is not easy to estimate. Since this project only presents a preliminary design, and lots of changes would be made from the decisions made here and the real case, it was never the goal to provide a whole budget for it. However, a qualitative assessment of the cost allows to classify the different powertrain alternatives from the cheapest to the more expensive one. The order considered was:

1) **Powertrain 1:** The simpler powertrain is also the cheapest by far. This car is the most similar one to the combustion engine car already designed for previous years. Thus, besides the lower amount of parts required in this car, some of the parts might be taken from old cars from the team. This can be done for all the different designs, but definitely this is the car which would have more commonalities with older versions.

2) **Powertrain 2:** Powertrain 2 would be the second cheapest configuration. This version doesn't require that many new parts, and so the most expensive would be to purchase a second motor and a second inverter.

3) **Powertrain 4:** The configuration with four in-wheel motors would be cheaper than Powertrain 3. Although this design requires four motors, inverters and reducers, it does benefit from the shortest battery pack. By requiring a lower amount of battery cells, this configuration becomes cheaper than Powertrain 3.

4) **Powertrain 3:** Powertrain 3 would be the most expensive. It would require different motors and inverters for each part of the vehicle (front and rear). Probably different brands will be chosen since the needs are different on rear and front parts. Also the large battery pack would result in this being the most expensive design.

5.5 Overall results and powertrain selection

Before showing the overall results and jump into the conclusion of which the best powertrain is, it is important to mention that these results are obtained through software simulations based on simplified models (in the most relevant part). As so, even though the results should be representative of the real system, there is a number of uncertainties that might affect the selection of the best alternative.

Furthermore, the qualitative analysis gives a general idea of which would be the most complex system or the most expensive one, but a deeper analysis would be needed to correctly assess which is the best configuration.

It is also important to know that the results of the Electric system simulation may influence the vehicle dynamics simulations. A larger battery would suppose a higher mass for the vehicle, hence also a different location of the centre of gravity, which in turn would change the results of the vehicle dynamics simulation. Moreover, some of the parts like the reducer or the inverter were only chosen at the time of proposing a preliminary design, so they were unknown during the simulation phase and therefore they were included in the models in a simplified way. Although their influence in the results is somewhat limited, in order to have more accurate simulation results the right components should be included in the model, and a new iteration of the simulation should be run

It should be highlighted that the overall system efficiency presented is the combination of the efficiencies of each item in the powertrain modelled (battery pack, inverter, motor and transmission system). The results are shown in the following table:

TABLE 16: *OVERALL RESULTS AMONG POWERTRAIN CONFIGURATIONS*

		POWERTRAIN 1	POWERTRAIN 2	POWERTRAIN 3	POWERTRAIN 4
Vehicle dynamic simulation	Endurance	80,7 s	80,8 s	80,0 s	79,0 s
	Autocross	79,8 s	80,0 s	79,7 s	78,9 s
	Skid pad	4,87 s	4,89 s	4,88 s	4,88 s
	Acceleration	3,05 s	3,06 s	2,91 s	2,92 s
Electric system simulation	Powertrain efficiency	84,8%	84,3%	83,3%	83,4%
	Energy required	5,96 kWh	6,00 kWh	8,06 kWh	5,72 kWh
	Energy recovered	5,89 kWh	5,95 kWh	4,26 kWh	5,80 kWh
Design complexity					
Cost					

It is not difficult to determine which the worse powertrain is, after seeing this numbers. Powertrain 3 places 3rd and 4th in many of the events, and also in the qualitative analysis. The conclusion is clear; this is not the most optimal configuration to be designed. However, the results might be poor because of the assumption that the rear motor is always working at full load, while the other two are used only if necessary. Another simulation with a different power distribution strategy might be required. However, since the results were really distant from all other configurations, it seems unlikely that it would never reach the best outcome results.

Powertrain 2 is the configuration which uses torque vectoring at rear wheels. It is the configuration with the second worst results in an overall analysis. This design does not place first in a single event. Probably the motors are a bit over dimensioned and that may penalize the results of the design.

In regard of the other two alternatives, the numbers are pretty close. Nonetheless, one can take a closer look at the differences between both, and realise that Powertrain 1 scores worse during the competition, but it is a simpler design and a cheaper one. It can be said that both designs do work and give good results, hence the decision on the best powertrain configuration depends on what the Formula Student team wants.

Of course, the simulation results for Powertrain 4 are better, so it all comes down to whether the budget for the design can be accepted by the team and if the design is too complex or not.

This configuration requires an extensive previous study of the design, not only because the powertrain is completely different from the design of the previously developed race cars, but also because a well suited control system is required. However, this is a really challenging aspect, and also provides high rewards, according to the results of the project. It should also be noted, that one of the events of the competition is the engineering design., so the proper study of the design would mean a higher point scoring on this static event.

It is suggested that the team should make a cost study. If Powertrain 4 configuration is achievable in terms of money, according to this project, that is the configuration that should be studied and developed, and presented for the upcoming designs of electric race cars.

However, if the budget is not feasible, the team should consider Powertrain 1 as the selected design, which has pretty good results, and the cost and design complexity does not pose major difficulties.

6 Powertrain design

6.1 Parts of the Powertrain

A drawing of the designed powertrain for the chosen configuration is shown in the below figure:

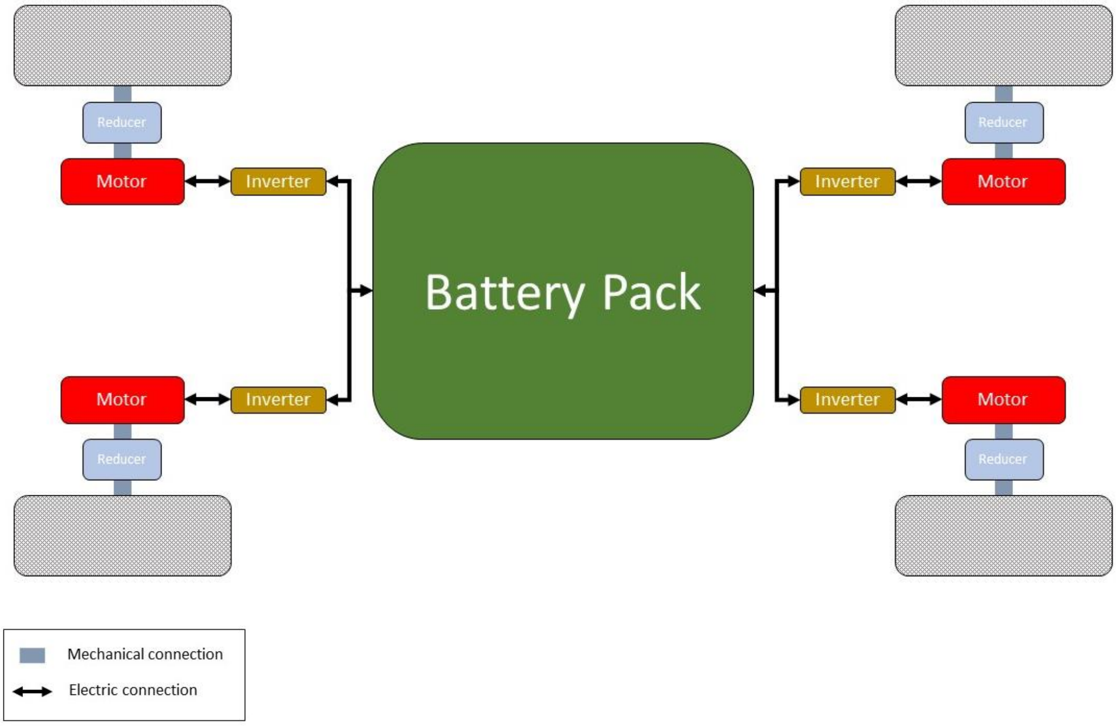


FIGURE 40: *POWERTRAIN 4 OVERVIEW*

As it can be seen, the powertrain consists of 4 main parts: The battery pack, the inverter, the motors and the reducers.

6.2 Motor

As it can be seen, each of the motors is attached to a wheel with a small shaft and a gearbox reducer. Although one of the main advantages of this design is the possibility of directly attach the motor to the wheel, this was not the case due to the available motors characteristics.

A motor directly attached to the wheel would have the main advantage of not requiring a reducer.

However, a motor of this kind would need a torque curve that works at an specific torque and speed range. The designed car is supposed to run at up to 120 km/h or in other words at 33,33 m/s. With a wheel radius of 0,22 meters, the motor would require to have a speed range from 0 to 1470 rpm at the end-range to work at an efficient rate.

Considering this perfect torque curve, this motor should require to work at maximum torque for at least 2/3 of the speed range. With these numbers, and considering a maximum power from the motor of around 20kW (which is the optimal for this case), the maximum torque required would be around 191 Nm.

After a market survey, no motor satisfying the previous requirements was found. The decision was to look into similar motors and choose one a bit over dimensioned for the case. However, no one was really satisfactory, either making the system too heavy or having too high maximum power and making it non efficient for the case.

The final thought was to look for a motor as lighter as possible and with a slightly larger power output as the required for the competition (around 20kW), even thought that would most likely require a gear reducer. The LFS team also believed that a liquid-cooled motor would fit better in their design.

Additionally, the gear ratio of the required reducer should be reasonable, in order to minimise its deign complexity.

The chosen motor is manufactured by **AMK Group**, which is one of the main manufacturers of servomotors, which are really adaptable for wheel motors. This brand also has a whole automotive department in which a part of it works for electric Formula Student teams. They also offer a whole sponsorship for using their motors.

The model of the motor is **DT7-75-20-xxW-3500**. IT belongs to the family of high torque-servomotors:

FIGURE 41: *SERVOMOTOR BRAND AMK, MODEL DT7-75-20-XXW-3500* **[39]**

6.3 Inverter

One of the main parts of an electric powertrain is the inverter. The inverter is in charge of distributing the power between the battery pack and the motor, and vice versa.

An inverter also receives the name of a DC/AC converter, converting the DC current from the battery and supplying it to the motor. One of the difficulties of an electric powertrain when using AC motors, is that the energy received (if accelerating) or given (if decelerating) by the electric motor uses alternate current (AC). The generator (the battery pack in this case) works with direct current (DC) instead. The inverter is in between the two and supplies the right kind of power to each of the parts.

The inverters needed in this case should have an output power of up to 20 kW and work continuously at least at 350 VDC on the battery side.

Lund Formula Student team already has a sponsorship agreement with **Sevcon Electrification Partner**. The company has an inverter which fits perfectly with the desired parameters.

The model of the inverter is a **Gen4 HVLP**

FIGURE 42: *INVERTER BRAND SEVCON, MODEL GEN4 HVLP* **[40]**

6.4 Design of the reducer

As explained previously, due to the difficulty of finding a suitable motor that could be directly attached to the wheels, a transmission between both is required. Hence, this is a required part on the design of the powertrain.

The transmission is required to have a ratio around 2,18. This means that the motor speed curve is 2,18 times the velocity range of the vehicle.

The transmission between each motor and the corresponding wheel can be done through different mechanical systems, like a chain or a belt drive. For this mechanical design, high accuracy in both manufacture and mounting is required. Hence the most optimal solution is to use gear drives which is the most compact and efficient solution for a reducer of this kind.

Gear trains come up as a great solution to solve specific mechanical challenges. Based on applications different concepts of gear trains have been developed, for example bevel gears transfer torque on non-parallel axis of rotation, rack and pinion convers rotational movement into linear. The most optimal design for our transmission system requires the motor and the wheel shaft be addressing the same axis. In other words, the gear type will be gears with parallel axis from which we can distinguish three kinds, depending on the shape of the tooth:

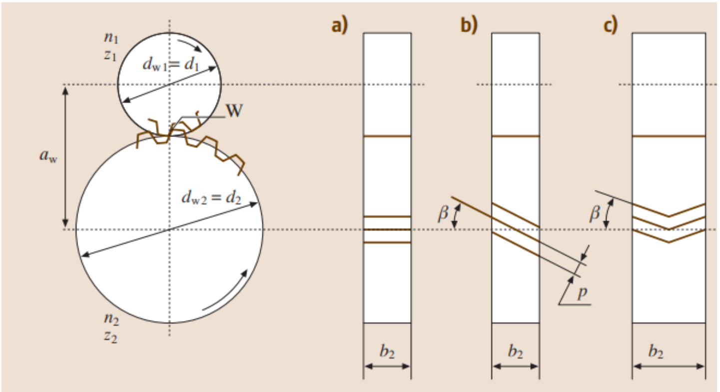


FIGURE 43: *PARALLEL AXIS GEARING WITH EXTERNAL TOOTHING. (A) STRAIGHT TOOTHER/SPUR, (B) HELICAL, (C) HERRING-BONE TAKEN FROM PAGE 334 OF SPRINGER HANDBOOK OF MECHANICAL ENGINEERING* **[37]**

A) **Spur gears** are simpler in construction and manufacturability, and have an excellent precision rating. On the downside, this gears are noisy and have a limited distance between the axes.

B) **Helical gears** are smoother and quieter than spurs, besides having a higher load carrying capacity. However, they require thrust bearings, and are more expensive.

C) **Herring-bone gears** are used for high power transmission capacity systems. They require an equally distributed load and also have the highest manufacturability cost of the three.

The transmission ratio of 2,18 can be accomplished using any of these systems with a single-phase set of gears. Hence the choice will not be affected by the maximum transmission ratio of each gear type.

For such an important piece of the car, the material used is of great concern. Steel is the best candidate to withstand the friction and impact stresses in high performance automotive transmission systems. One can say that the cost of manufacturing four of these pieces (one for each motor), would be really high. The conclusion is that the trading off the noise and a smoother piece for a much higher budget is not worth. Thus, the best mechanical solution is to use spur gears.

The transmission ratio is not only the angular speed reduction between the driving gear and the conducted one, but also in the designing phase, it is ratio between the number of teeth from each. With a needed transmission of 2,18, a single phase gear is enough to accomplish the desired outcome for the reducer. It is very unlikely to provide of a gear transmission with the needed value. Three main solutions may be discussed:

1) **Z1= 218** (number of teeth of the conducted gear), **Z2=100** (number of teeth of the driving gear). This accomplished a ratio of 2,18. However the size of such a pinion and wheel would require to be enormous. X
2) **Z1= 22, Z2=10.** The ratio of this design is 2,2. It is not the desired outcome, and the low number of tooth would make the gears suffer a lot from fatigue. X
3) **Z1= 87, Z2=40.** This design allows a gear ratio of 2,175 which is almost equal to the desired 2,18. The number of tooth seems acceptable to not be rather sizable and not suffer too much from fatigue. √

Using the design mentioned, the result of the gear transmission would be similar to the one in the following picture.

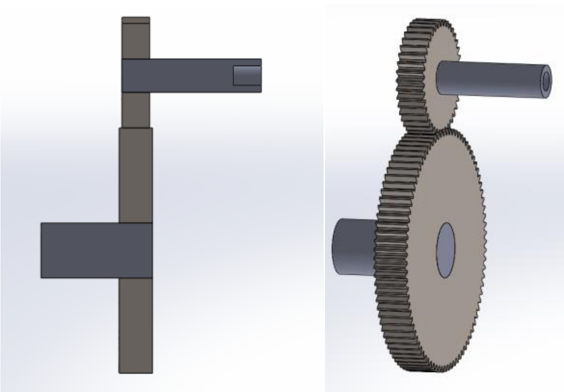


FIGURE 44: *SIMPLE MODEL OF THE GEAR TRANSMISSION*

This is a really simple model, and should be improved to introduce it in a real powertrain. The upper gear (drive gear) is connected to a shaft directly connected to the motor. The bottom one corresponds to the conducted gear and its shaft would be connected to the wheels.

6.5 Design of the battery pack

The battery pack is the part of the powertrain of a race car in charge of storing and delivering the energy necessary to complete the races. In a FS scenario, the electric cars are able to recharge the batteries in between the different races.

Of course the most challenging race in terms of energy management is the endurance race. The results from the simulation program are used for this part.

In this case, instead of checking the results and then searching for the proper provider, the strategy was different. It was decided that a battery can be designed according to the number of cells and the structure of the battery pack.

The approach will be the following:

1. The total number of cells will be dictated by the energy needed for the race. The total energy divided by the energy stored per cell will be the number of cells needed.
2. The current for each cell cannot be higher than the maximum charge and discharge rate. The current will therefore determine the number of parallel cell stacks.
3. The voltage on each cell should not be higher than the nominal voltage.

The cell chosen was the same one as the team wants to use in their first electric race car design, which is the model **SLPBA875175** whom provider is **Melasta.** This cell has the following specifications:

TABLE 17: *BATTERY CELL SLPBA875175 SPECIFICATIONS*

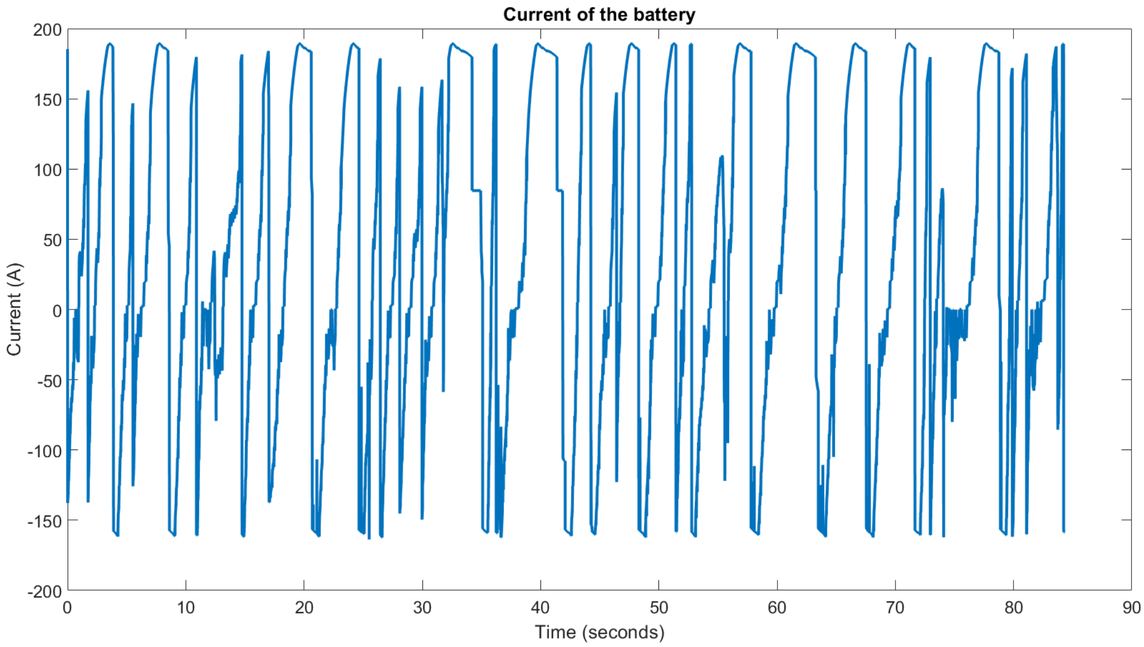
Energy store	Peak discharge current	Max continuous Discharge current	Nominal voltage
55,5 Wh	225 A	150 A	3,7 V

FIGURE 45: *BATTERY CELL BRAND MELATA, MODEL NUMBER SLPBA875175* **[41]**

As the first condition, it should be recalled what the results from the electric system simulations required a battery of at least 5.720,5 Wh. To protect the battery, a safety margin is required. For this project it was supposed that the battery can not be discharged below 20% of the state of charge, or in other words, the total energy stored should be 20% overdimensioned.

With battery cells capable of storing 55,5 Wh each, the battery pack requires 124 cells to be higher than the calculated 6.864,6 Wh. This results in a battery of 6882 Wh.

Following the second condition, and again using the results from the simulations, the current behaviour on the battery model was plotted. It can be seen in the following picture:



	Current
Maximum	189.1804 A
Minimum	-166.4191 A

FIGURE 46: *BATTERY CURRENT BEHAVIOUR DURING ENDURANCE RACE POWERTRAIN 4*

As it can be seen, the maximum discharge current is 190 A. This current is not continuously at this rate, so even though the results are higher than the max continuous discharge rate (150 A), what really matters is the peak discharge ratio. The cell can handle up to 225 A, hence the battery pack can be designed with a single branch of cells in parallel.

The last condition is the voltage from the motor and the electric system. The motor requires a voltage of 350 V. The data sheet indicates a voltage in the cell of 3,7 V. The number of cells allowed to be placed in series is 95. Then, at least two branches would be required.

However, this would imply a total of 190 cells (two branches of 95), which would mean an extra 66 cells, implying a higher cost and higher weight of the design.

The best solution to avoid a larger battery pack, would be to have a single row of 124 cells as firstly suggested, and make the motor work at a higher voltage of 458,8 V (124 cells ·3,7 V each). The manufacturer should adapt the wiring; in the case the motor cannot normally work at this voltage.

7 Benchmarking against LFS20

Some assumptions were made during the simulations of both the Vehicle Dynamics Model and Electric System Simulations. For instance, the total weight of the vehicle was estimated. It is believed that the proposed design should improve the results compared to the LFS20, the car designed by the Lund's Formula Student team for 2020's competition.

The first part of the benchmarking, compares the specifications of the car design. At this point, the components in both designs are known and therefore it is possible to estimate both the weight and the centre of gravity. The improvements from Powertrain 4 to the LFS20 on weight terms are shown below:

TABLE 18 *MASS COMPARISON BY CHANGING LFS POWERTRAIN'S TO POWERTRAIN 4*

Motor	Inverter	Battery	Gear reducer	Transmission system	TOTAL
- 0,3 kg	+3,9kg	-4,84 kg	+3,2 kg	+0,5 kg	**+2,46 kg**

The preliminary design presented in this project is pretty simple. Therefore, the positioning of each element shaped are unknown. The estimation was that the inverter and the motors are both at half distance from the body to the wheels, and at the same height as the LFS20 center of gravity. The reducers are placed in the centre of the wheels.

It is known that a lower gravity centre will provide better dynamics results. Hence, the battery pack which is the heaviest component, will be placed in the middle of the car, and in the lowest possible height.

In addition, it was considered that the car is symmetric. Thus, the race car suffers no centre of gravity variation on y-axis. In the next figure it is shown the estimated placement of the different components on the new design

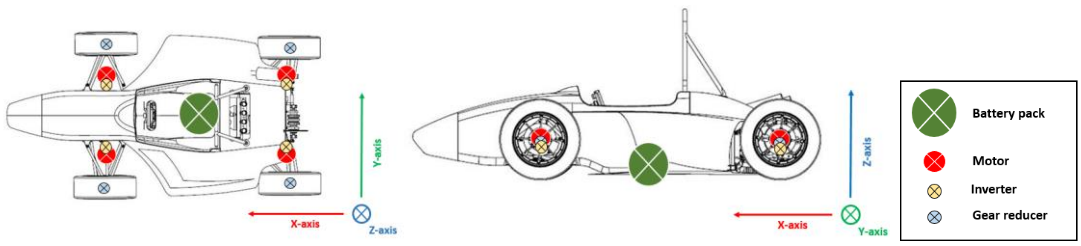


FIGURE 47: *DESIGN PLACING FOR CENTER OF GRAVITY CALCULATION*

How this placement alters the centre of gravity is shown in the following table:

TABLE 17: *CoG comparison by changing LFS Powertrain's to Powertrain 4*

Δ Centre of Gravity x-axis	Δ Centre of Gravity y-axis	Δ Centre of Gravity z-axis
+0,015 m	0 m	+ 0,003 m

It can be seen that the centre of gravity is moved to the front and the upper part of the vehicle. However, this change is only of a few centimetres, thus it should not have a significant impact. With these new calculations, it is time to check the performance of both vehicles.

The results are summarized in the following table

TABLE 19: Vehicle dynamics simulation *results comparison between LFS20 and Powertrain 4*

	Endurance	Autocross	Skidpad	Acceleration
LFS20	80,65 s	79,44 s	5,415 s	3,05 s
Powertrain 4	78,91 s	78,43 s	5,423 s	2,91 s
Total Difference	**1,74 s**	**1,01 s**	**0,008 s**	**0,14 s**

It is clear that the new design performs better in the competition. It is faster in every race than LFS20. On the first two races which really show the dynamically better prepared vehicle, the proposed design runs faster. The Skidpad's winner race is Powertrain 4, but the advantage on this race is not really relevant. On the acceleration race Powertrain 4 wins again, with an advantage that shows a design with a higher tractive force than LFS20.

These numbers show better results than those presented in the simulation and results chapter. This is due to an overestimation of the weight from the vehicle. After choosing every component of the powertrain, the final design becomes lighter then estimated. Additionally, the results show that designing a lighter vehicle is also relevant for obtaining the best performance.

The last part of the benchmarking comparison is on the Electric system simulations. The results are shown below:

TABLE 20: Electric System simulation *results comparison between LFS20 and Powertrain 4*

	Energy recovered in Endurance race	Energy used in Endurance race	Battery pack designed
LFS20	5,79 kWh	11,85 kWh	7,77 kWh
Powertrain 4	5,89 kWh	11,55 kWh	6,88 kWh
Total difference	**-0,1 kWh**	**- 0,3 kWh**	**- 0,89 kWh**

The results show higher energy recovering capacity for the LFS20 car. This is mainly caused by a higher efficiency from this powertrain.

However, the total energy used to complete the race by the designed Powertrain 4 is less than the corresponding for LFS20. This is mainly due to a faster race finish. Although the design has a lower motor efficiency, the fact that it completes the endurance race about 24 seconds faster (on the 14 laps considered) means that it requires a smaller battery pack.

There is a large difference from the battery pack proposed by the team for LFS20 compared to the one proposed in this thesis. The reason behind it is that the margin stablished by the team on the first is wider, while for Powertrain 4 the design made is based on the hypothesis that a 20% over dimensioned battery pack is enough.

8 Simplifications and future work

During the development of this project, many assumptions were made. In order to assess the impact of these, further studies are required, which were out of the scope of the project. Some of them are:

- **Linear G-G diagram**: As it can be seen in **Figure 19** of section **3.2** the g-g diagram surface developed is made of rectangles, that come from two 14x14 matrices. However, the diagram is curve shaped, and as so for a smoother diagram, polynomial interpolation could be used between the points instead of linear. This should be implemented in future versions of the vehicle dynamics model.

- **Torque and power motor results**: During the development of the electric system simulation, it was required that, as an input of it, the torque and power of the motors were calculated. As seen in **Table 5** in section **3.4**, the torque was calculated with the formula:

  $$\tau = \frac{\left(\mathrm{a}_{long} \cdot \mathrm{m} + \mathrm{F}_{drag}\right) \cdot r}{i \cdot numb\ of\ motors}$$

  For configurations with multiple motors, as it can be seen in the equation, it is considered that the total torque delivered by the motors is equally distributed among them. However, with the load transfer implemented on the model, the torque delivered by each motor should be different. The electric system model does not include that, meaning that all motors work at the same operating point, and thus providing inaccurate efficiency results.

  For future reference, the load transfer proportion at each wheel should be calculated for each longitudinal and lateral acceleration, in a similar way as it was done for the g-g diagram. If the load transfer ratio for each wheel is known, it would be easy to identify the torque produced by each motor, and so the electric system model results would be more accurate.

- **Powertrain 3 consideration**: Throughout the project, it was considered that the configuration with three motors, had the rear motor working a full load, and the motors in the front part were only used when necessary.

  It was seen that this powertrain placed low when analyzing the results, especially in efficiency terms, which was probably caused by the mentioned consideration. For future works, the load should be distributed proportionally to the size of the motors, so that they would work on a more efficient area.

- **Lack of driver skills**: In the models used in this project, it is considered that during the lap simulations all the track points of the race are run at the optimal velocity. It should be noted that in the real competition, there is a driver that controls the car, which of course would be impossible to drive in a perfectly accurate way.

In the future, the model should include a level of uncertainty, so that the laps are not always run with the perfect trajectory. With it, a statistical model should also be included in order to analyze how reliable the results are.

- **Control system limitations:** The control system was out of the scope of the project. It was never studied if the torque distribution system would be achievable with the control system. This should also be included in future works.

9 Conclusion

In this project it was stated that the configuration that gets the best results is the one with an actuating motor on each wheel. The four motors configuration has the best results mainly for the advantage that AWD torque vectoring provides the ability of distributing the torque among the four wheels in a way that the dynamics of the car are improved.

The results were proven in this thesis, however the powertrain configuration with four motors is the state of the art. Most of the FS teams participating in the electric car competition propose this design for their own powertrain. This trend was really confirmed on 2012 and 2013, where teams like Stuttgart and TU Delft teams stablished new records on the competition using AMK servomotors. **[38]**

This project proposes an innovative approach to test the vehicle's dynamics with different powertrain configurations. The code developed by the team together with the changes proposed on this thesis becomes a good tool to evaluate the dynamics of different designs. The results shown on this project cannot be directly applied to the reality. However, all the considerations done are based on expert research articles, most of them belonging to SAE Mobilus collection. This source belongs to the Society of Automotive Engineers which of course have a wide expertise on automotive theories.

The electric system simulation is also based on similar sources. The models used for the battery and the motor have been used on related studies of automotive engineering. Thus these models are verified by experienced engineers.

The part of the project related to the preliminary design; requires a deeper study. Particularly, the design of the gear reducer and the transmission system should be made deeper. The aim of the thesis was to compare different alternatives and selected the best powertrain, and the design was really brief compared to the performance analysis. A proper design of the powertrain would require a wider scope, but it should be done before designing the whole vehicle for the competition.

Another remark that should be made is that the powertrain should also include the control system. This system includes the electronics which connect the inverters and the battery pack. The control part was out of the scope of this project.

As a personal conclusion, this project has provided the author with new knowledge on vehicle dynamics. The study of the Matlab code provided by the Lund's team required to understand many vehicle dynamics concepts. Learning those was satisfying since this was an unknown field of study for the author.

Also, being able to use software to model a real case study was gratifying. Not only because of the acquired knowledge of software modelling, but also because the results are intended to modify the design of a real and complex whole system. Being able to apply the results to a real case scenario feels like a meaningful project for me.

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Lunds Universitet
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A. Appendix

A.1 Motor technical data [39]

DT 7 liquid-cooled servo motors

BENEFITS

- Excellent standstill torques
- Excellent torque and power density
- High power at low speeds
- IP54 degree of protection

Applications

- Positioning and actuating drive for drive tasks with exceptional requirements on the torque and power density
- Variable-speed drive for continuous running
- For applications with difficult cooling situations due to a high integration density, contamination or ambient temperature conditions

Connection cable:

Rated cross-section of copper conductor: 10 mm^2, Power connector size 1.5

Equipment	Standard	Option
Brake	–	18 Nm
Encoder	Resolver	E-, F-, P-, Q-encoder
Shaft	Keyless shaft	Shaft key DIN6885 A10x8x36

Technical data

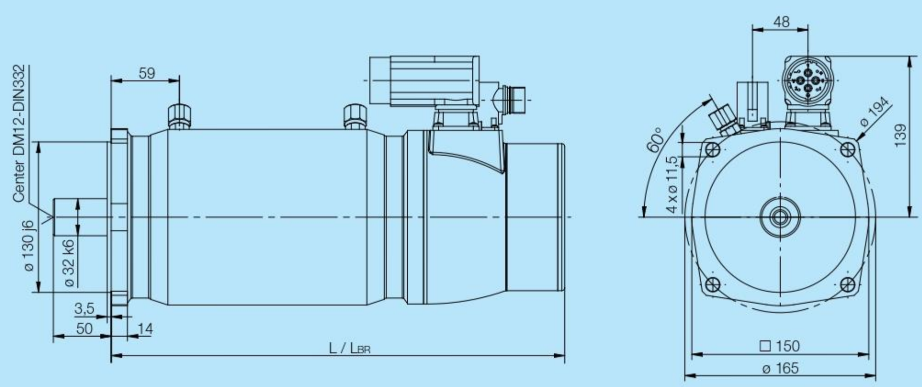
Motor type	Standstill data		Rating data					Maximum data		Electrical data		Mechanical data				
	M_0 [Nm]	I_0 [A]	M_N [Nm]	P_N [kW]	I_N [A]	n_N [rpm]	k_T [Nm/A]	M_{max} [Nm]	I_{max} [A]	L_{tt} [mH]	R_{tt} [Ω]	n_{max} [rpm]	J [kg cm^2]	L [mm]	L_{BR} [mm]	m [kg]
DT7-75-20-xxW-3500	75	51	66	21	48	3 000	1.48	120	99	1.25	0.294	3 400	55	298	342	3
DT7-110-20-xxW-3700	110	74	90	28.1	64	3 000	1.55	156	116	0.78	0.153	3 700	81	348	392	28.5
DT7-145-20-xxW-4000	145	96	114	35.9	82	3 000	1.51	220	200	0.5	0.122	3 600	107	408	452	35.7

Motor data for 350V motor voltage · Measuring data at winding overtemperature $\Delta T < 80\,K$

28

Legend on Page 43. All dimensions are given in mm.

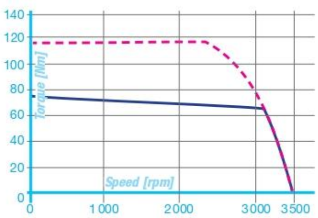
Dimensions



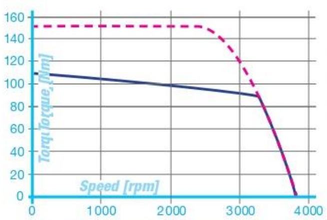
Cooling

- Minimum coolant flow rate: 2.0 L/min; max. inlet temperature: 40°C, example cooling connection shown

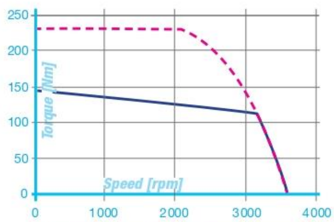
Characteristics



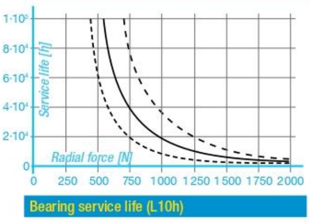
DT7-75-20-xxW-3500



DT7-110-20-xxW-3700



DT7-145-20-xxW-4000



Bearing service life (L10h)

Maximum torque — Continuous thermal torque — Bearing service life: 2 x n_N — n_N — 0.5 x n_N

A.2 Inverter technical data [40]

HVLP
Inverter

High Voltage Low Power Motor Controller

A new range of low power AC motor controllers/ inverters designed primarily for ancillary control of pumps and fans on hybrid and electric buses, agricultural implements, tractors and other high voltage vehicles. There are two models in the range, which cover both liquid cooled and air cooled designs.

A compact, rugged and cost effective design, Sevcon's HVLP Inverter range is well suited for OEMs, hybrid/ electric bus & truck conversions, Agricultural OEMs and EV system integrators. The high voltage range is fully operational from 200V up to 800VDC, matched to the needs of the automotive/commercial transport markets as well as the AEF led Agricultural Electrification. The same hardware platform handles both AC Induction and Permanent Magnet AC synchronous motor technologies.

Air Cooled HVLP-10

Features

- 200V to 800VDC battery voltage
- Up to 53A peak current (HVLP-20)
- Up to 33A continuous current (HVLP-20)
- Advanced flux vector control
- 12V or 24V nominal supply
- High Voltage Interlock (HVIL)
- CAN J1939 compatible
- AC Permanent Magnet synchronous motor
- AC Induction motor
- Resolver and Sin/Cos position sensors
- Induction motor control without position sensor

Liquid Cooled HVLP-20

Key Parameters

HVLP-20 (liquid cooled)

- Operating voltage range at full current 200V to 800V
- Output motor phase current:
  - o 53A rms (1 min)
  - o 33A rms (Continuous)
- Water/Glycol coolant
  - o Full operation up to 70oC inlet temperature and at 6/l min flow
- Weight: Liquid Cooled = 2.3kg (dry)

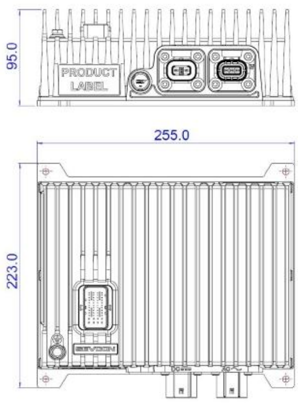
HVLP-10 (air cooled)

- Operating voltage range at full current 200V to 800V
- Output motor phase current:
  - o 24A rms (1 min)
  - o 19A rms (Continuous)
- Ambient -40°C up to +45°C full operation
- Weight: Air Cooled = 3.7kg

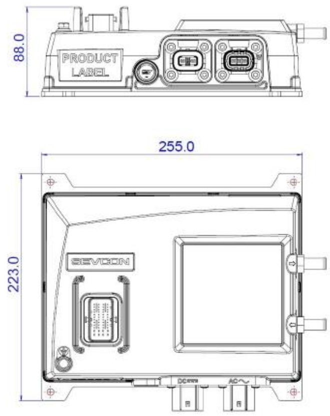
Common Parameters

- Safety:
  - o HVIL (High Voltage Interlock H/W & S/W)
  - o Designed to meet the electrical isolation of electrically propelled vehicles ISO 6469
- Environmental:
  - o IP6k9k and IP67 protection
- 12V or 24V nominal supply

Air Cooled HVLP-10



Liquid Cooled HVLP-20



All dimensions in mm.

Integrated I/O

HVLP includes a fully-integrated set of inputs and outputs (I/O) designed to handle a wide range of requirements. This reduces the need for additional external I/O modules or vehicle controllers and connectors.

- All I/O protected to 40V
- 4 analogue inputs 0-10V
- 4 digital inputs
- 2 power supplies 5-10V (100mA & 200mA)
- 2 digital outputs

For more information visit sevcon.com

A.3 Battery cells technical data [41]

3.7V 15000mAh (55.5Wh) 10C super high capacity lithium ion polymer battery

Product Details:

Place of Origin:	**China**
Brand Name:	**Melasta**
Certification:	**CE, RoHS**
Model Number:	**SLPBA875175**

Payment & Shipping Terms:

Minimum Order Quantity:	**100pcs**
Price:	**negotiation**
Packaging Details:	**Standard package**
Delivery Time:	**45 working days**
Payment Terms:	**T/T in advance, L/C**
Supply Ability:	**5000pcs/week**

Nominal Capacity:	15Ah	**Type:**	Li-polymer
Nominal Voltage:	3.7V	**Size:**	Prismatic
Max Continuous Charge Current:	15A	**Peak Charge Current:**	30A
Max Continous Discharge Current:	150A	**Peak Discharge Current:**	225A

3.7V 15000mAh 10C (55.5Wh, 150A rate) high power high capacity LiPo battery cell

Item Specification:

Model Lithium Polymer battery SLPBA875175

Nominal Capacity: 15000mAh

Nominal Voltage: 3.7V

Cycle Life >100 times

Charge Voltage: 4.2V

Max Continuous Charge Rate: 1C (15A)

Peak Charge Rate(C): 2C (30A)

Max Continous Discharge Rate(C): 10C (150A)

Max Plus Dicharge Rate(C): 15C (225A)

AC Impedance(mOHM): <1.5

Discharging End Voltage: 2.75V

Size: 10.8 x 75.5 x 175mm

Weight Approx (g):302.5±5g

Energy Density: 180.19 Wh/kg

Power density: 1801.95W/kg

Note:

Melasta's development of high drain type of Li-polymer batteries ($LiCoO_2$) can bear max continuous charge rate up to 2C(Peak current 4C) and continuous discharge rate ranging from 10C to 40C. One of key features of this Li-PO is that they will have a good plateau (Mid- point –Voltage) compared with regular battery in discharge, obviously turns up for much more higher one, it's voltage keeps stable. The same principles apply to comparison between two high drain of cells with different rating.

Tabs:

Positive tab: nickel-plated copper (Thickness: 0.2mm, width: 25mm, length: max 30mm)

Negative tab: nickel-plated copper (Thickness: 0.2mm, width: 25mm, length: max 30mm)

(We can provide pure alumium tabs if required)

Application:

High drain Lithium polymer batteries can be applied to RC Model, Formula Student Racing car, Hobby Racing car, UPS, E-bicycle, E-scooter, Electric Wheelchair, Power

tools and other special products which need high drain design.
(PS: Melasta high power rate single battery cells already adopted by several Formula Student team.)

Warrenty: 6 months

Certification and Test Report:
CE(EMC directives), RoHS, KC(Korea Certificate), UN, 1.2m Drop test, Cargo Transportaion Test Report