Design of an energy efficient high performance drive train

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Master’s Thesis

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Summary

In modern day transportation there is an increased demand for energy efficient vehicles. This trend is also noticeable in certain classes of motorsport, a good example is the Formula Student competition. Here, race cars are ranked on their dynamical performance and the fuel consumption that results from the racing. A special class is launched within the Formula Student framework, with the intention to let alternative drive trains compete with each other in a racing environment. The emphasis of this class is to build a race car that uses a minimal amount of energy, while still being as fast as possible on the track.

University Racing Eindhoven (URE) participates in this competition, however the requirements for such a vehicle are contradicting. The main question is, what kind of drive train suits the competition goals best. Furthermore, the implementation of the chosen drive train concept requires a different design approach than the previous URE cars.

In the first part of this report a method is presented, that evaluates different drive train topologies, in terms of performance and fuel economy. The method uses a mathematical model, that is able to predict lap times and the accompanying energy consumption. This is done with a simplified representation of the race car, namely a bicycle model, and an efficiency based description of the drive train components. Subsequently, the following drive train topologies are evaluated with this method:

- full electric
- series hybrid
- parallel hybrid

Besides, a conventional drive train, with an (petrol-powered) internal combustion engine is analysed as a reference. The presented method enables the user to perform quantitative research on aspects of interest, such as regenerative braking. Furthermore, drive train parameters can be optimized, like the final drive ratio and component sizes. As a result of the topology evaluation, and based on other practical arguments as well, a full electric drive train is chosen for further development.

In the second part the electrical drive train is implemented in an overall vehicle concept. The 2009 race car, the URE05, serves as the donor vehicle for this development. With the help of the competition rules, design criteria and the mathematical model, design specifications have been proposed. On basis of this, suitable drive train components have been selected. The concept has resulted in a rear-wheel-driven race car design with two independent permanent magnet direct current motors, of 35kW peak power each. Furthermore, a battery package has been selected, that consists of 78 high-quality Li-ion cells. All drive train components are located inside the rear frame.

In the third part of this report the design of the battery, that has been engineered in full detail, is presented. Different cooling techniques have been studied, and suitable materials that comply with the requirements have been selected. The main goal was to keep the battery construction as lightweight as possible, while still ensuring functionality and safety. The resulting design is subsequently illustrated with comprehensive figures and information. A thermal 2D finite element analysis is performed, with the intention of determining the highest expected temperature during the endurance. Although this analysis predicts that no cell temperature limits are exceeded, the battery must be cooled down for a sufficient period before recharging and starting another race.
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Chapter 1

Introduction

This chapter introduces the research objective, which focusses on the development of an alternative drive train for a Formula Student race car. Beginning with a brief introduction on trends in alternative drive train applications, the chapter proceeds with stating the problem definition and goals. Finally, the outline of the report is given in the last section.

1.1 Solutions for reduction of fuel consumption and emissions

Recently, governments are putting more stringent rules on car fuel consumption and emissions, and manufacturers try to meet these requirements by improving cars on a whole range of aspects [42]. These include reducing the vehicle’s losses, such as aerodynamic and rolling resistance, as well as internal drive train losses. The conventional petrol or diesel driven powertrain has undergone many improvements, of which recent examples are “down-sizing” and even more recent the application of the “MultiAir” valve mechanism [44]. The addition of bio-fuel to conventional fossil fuels is also expected to gain more and more interest and application [40,41]. Other, more radical, developments are for instance the introduction of hybrid electric vehicles (such as the Toyota Prius), hydrogen fuel cell cars (Honda FCX) and battery electric vehicles. The latter is expected to gain a great market-share and, therefore, an increasing amount of research is on the development of these vehicle types. However, debates are continuing, as it is still unclear what the best options are for future transportation.

1.2 Trends in motorsports

For many years high octane fuels, which are derived from fossil oil, have been the dominant fuel type in racing and motorsports. Petrol engines for instance are known for delivering power with excellent response, and, when designed for performance, can be build extremely lightweight. A good example of this are Formula One engines, which at some point in history produced up to 600kW, while weighing less than 100kg. However, the trend towards more energy efficient transport has not gone unnoticed in the world of motorsports. Competition committee’s have been asked to change their image, and to spend more time on making their race cars environmentally friendly. Furthermore, motorsports is seen as an opportunity to test new technologies under harsh conditions. The Kinetic Energy Recovery System (KERS) in Formula One is an example of this. Hence, one can see a clear trend in the latest years towards the use of alternative fuels and drive train technologies in motorsports. Although these new developments are slowly being accepted by teams and fans, there are already a few examples of how alternative drive trains beat the competition. Audi for instance opened the eyes of many, when they won the 24h of Le Mans in 2006 with a diesel powered car, the R10 TDI. Another spectacular victory was achieved by Toyota in 2007, when they won the Tokachi 24h endurance with a hybrid race car, the Supra HV-R. The Formula SAE competition, a worldwide racing challenge between universities, is also on the forefront of these developments; Dartmouth College started the Formula Hybrid competition in the United States, 2005. Also, in 2007 the class1A “low-
carbon” competition was launched in the United Kingdom, where the focus is placed on the application of energy efficient alternative drive trains. Soon, more Formula SAE competitions followed with setting up new classes of “green motorsports”.

1.3 Problem definition

In the end of 2008 University Racing Eindhoven (URE) decided to take part in the Formula Student class1A competition of 2010. Although there was great interest from several departments and students, it was not clear what kind of race car had to be designed and built. On the contrary, when the discussion evolved it soon became clear that the solution is not straight-forward and that many vehicle requirements are contradicting.

1.3.1 Problem statement

Clearly, low energy consumption and racing performance are difficult to combine. In literature, several attempts to fuel efficient race technologies are proposed [56,57,59,60,73–75] but these technologies often struggle with complexity, acceptance and other disadvantages. The abundance of requirements make the design process a challenging task. Moreover, a systematic approach for designing an energy efficient race car is lacking in available literature, as this is a fairly new development, and also due to the secretive nature of the racing world. Formula Student is an exception in this secrecy, as the goal of the competition is to give students a learning experience and allow new technologies to be applied and compared. As a consequence, several racing drive train designs have been published by student teams [51,52,54,55,58,61].

1.3.2 Goals of this research

The goals of the research described in this report are as follows:

- Investigate the various drive train lay-outs within the Formula Student class1A framework, and identify the most suitable drive train topology.
- Perform a detailed selection of the required drive train component sizes and characteristics, and propose an overall vehicle specification and design.
- Design and analyze key systems, by performing an in-depth engineering task.

1.3.3 Main contributions of this report

The main contributions of this report are the following:

- A detailed method of approach is introduced, in which different drive trains can be compared on several aspects, such as fuel economy, performance and component requirements. This method is not only suitable for URE, but can also be adopted by manufacturers of performance cars and other motorsports competitions.
- The method of approach is translated to a mathematical model, and 4 drive train topologies are compared with each other using the model;
  - conventional internal combustion engine (SI)
  - full electric
  - series hybrid
  - parallel hybrid
- The proposed model has proven to lend itself well for the further design of the race car drive train, as it is able to predict many important parameters. Hence the model can be used to optimize the chosen concept and design.
• A light-weight battery design is proposed, including a thermal analysis, which will form the basis of future URE electric race cars.

1.4 Outline

Due to the diversity of subjects discussed, the report is split up into three parts.

Part I covers the evaluation of the drive train types that are to be studied. Chapter 2 contains general information on the competition and goals of the model, as well as its structure. In Chapter 3 the build-up of the vehicle model is explained in detail. The analysis results for URE specific parameters and decisions are being treated in Chapter 4.

Part II covers the overall design of the chosen drive train solution. In Chapter 5 the desired vehicle specifications are being dealt with, while in Chapter 6 an extensive component selection procedure is summarized. Finally, in Chapter 7 the overall design of the race car is formed and presented.

In Part III the design of the battery system is treated (Chapter 8), and subsequently a thermal analysis has been carried out, of which the results are presented in Chapter 9.

In Chapter 10, conclusions and an outlook for future improvements are given. Additional information can be found in the appendices.
Part I

Evaluation of drive train types
Chapter 2

Method of approach

2.1 Introduction

Formula SAE [1,i2] is a world-wide competition between universities, in which student teams are challenged to design, build and race with a small single seater vehicle. The car is an open-wheel formula-style race car, with the focus on dynamic performance, in terms of acceleration and handling. Fig. 2.1 shows an example of such a car, the URE05. Teams are judged and ranked on a variety of aspects, which not only include the capabilities of the car, but also their business strategy, vehicle production cost and the overall design. To assess the rankings of teams on an event, a general scoring system is adopted, in which a maximum of 1000 points can be scored. Furthermore, the event is split into a static and dynamic part. The static part includes several presentations, while the dynamic part involves the actual racing.

In 2007, a new competition was launched at Formula Student UK; class1A. This event takes place together with the regular class1 event, but there are some major differences: the goal of class1A is to allow different types of fuels and drive trains to compete in a racing environment, and to compare them on two main aspects, namely dynamic performance and CO₂-production. A full overview of the scoring can be found in Appendix A. This appendix also shows the allowed drive train types and fuels, and presents more information on important class1A specific rules [i1].

Due to the great diversity of options and possible drive train combinations, the choice of a certain drive train type and fuel is non-trivial. Other important choices to be made are component technology and size and operation, etc. These factors will influence the outcome of the chosen solution. In Fig. 2.2 the main factors are depicted.

To evaluate all available options a first reduction of candidates has been carried out, which will be discussed briefly in Chapter 3. The remaining drive train types and power sources will be examined thoroughly in a vehicle model, which is able to predict the performance of each option on a quantitative level. In the next sections the motivation for building this vehicle model, and the overall structure, is explained.
2.2 Goal of the method

As indicated in Section 1.3.1 and 2.1, finding a suitable drive train for a class1A race car is difficult. One may decide to select a drive train on basis of practical reasons for instance, but in order to have a good chance of winning the competition, the selection of the drive train concept needs to be considered in detail. The question that arises, is on which grounds to evaluate the different options. A statistical analysis from past class1A results is not very helpful, as there are only a few results so far. Furthermore, it is not possible to evaluate a certain drive train by the amount of points the car can theoretically score, since that depends on the performance of the other participants as well. Calculation of fuel economy is also an option. In literature [79], several methods of approach are suggested, such as the quasistatic approach, covered in the QSS Toolbox Manual [47]. This method is not directly suitable since it assumes that the velocity profile is known in advance, which is not the case. However, energy consumption calculation is attractive due to the ease of implementation. In [46] a method for calculating lap times is proposed. Although this approach only considers the car as a point mass, with all tyres lumped, it provides a good basis for velocity profile estimation.

This research presents a method, that has the main goal of predicting the performance of a race car and the drive train, both in terms of lap times and energy consumption. Another goal of the method is to enable the user to evaluate the effect of energy-saving measures, such as regenerative braking. The proposed method contains a mathematical model, which is characterized by a short computation time. This enables a parametric study.

The following facts and assumptions are taken into account when defining the method:

- The drive trains are evaluated on the endurance race performance, since most of the points can be scored in this particular event (400 out 1000).
- In reality the number of variables that affect the results is vast, and it is necessary to exclude/limit certain effects from the evaluation:
  - There is no variation in driver performance, he/she is assumed to always race on the edge of the car’s capabilities.
  - Each lap in the endurance is driven exactly the same.
  - The chosen drive train operation strategies must result in a predictable dynamic behavior (from a driver point of view), that is similar to conventional racing.
  - Weather effects such as varying temperature and wind conditions are neglected.
  - Since the class1A race is held in Silverstone UK, two track conditions are taken into account: dry and wet.
Suspension behavior is excluded.

Other transient effects, such as corner entry / exit are neglected.

With these decisions the approach is simplified to an extent that it is possible to create a model that adequately calculates the parameters of interest, within a short amount of time.

2.3 Method structure

In Fig. 2.3 the overall method structure is depicted. MATLAB is used as the modelling environment. The method consists of function blocks, which interact with each other. This structure has the advantage that parts can be added or modified quite easily, in case the user wants to investigate other aspects and options.

The sequence of calculations starts at the top of Fig. 2.3, where one can specify an arbitrary track, vehicle properties and a drive train configuration. The track is divided into steps of constant length, and the problem is solved numerically; all variables are assumed to be constant within these steps.

The first step calculates the driving profile, on basis of the vehicle and track data. For this it makes use of several function scripts. Step 1 also computes other data such as vehicle losses, and average vehicle velocity. The second step comprises a calculation of the state of the drive train components, which are represented by more function scripts. Hence, the power profiles for all components are known, including efficiencies. From these data, the total energy consumption can be derived, and thus the equivalent CO$_2$-production.

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Chapter 3

Race car model

3.1 Introduction

This chapter presents a detailed description of the proposed analysis method, which has the objective of calculating lap times and CO$_2$ production. First a simplified model of a race car is provided in Section 3.2. Here, the relevant parameters and calculations required to model the race car dynamics are derived from the forces that act on the vehicle. Secondly, Section 3.3 deals with the construction of a preliminary (maximum) velocity profile from the car’s grip level and the lay-out of the racing circuit. In Section 3.4 the drive train topologies of interest are discussed. This section also gives information on why particular types of power sources and components are not considered in this report. The components of which the drive trains consist, and the modelling is subject of Section 3.5. After that the operation strategy of each drive train is discussed in Section 3.6. Once this is known, the final velocity profile of any of the drive trains can be calculated, and subsequently also the accompanying energy consumption. The main literature that is used for modelling and equations in this chapter is [77,79,82].

3.2 Vehicle dynamics

Race drivers try to minimize lap times, by driving their cars as fast as possible on the track. They are not only limited by the power the engine can deliver, but even more by the grip that the tyres are able to deliver. The lap velocity profile that results from a race car is consequently dictated by:

- the resistances that act on the vehicle
- the available tyre grip (longitudinal / lateral)
- the layout of the track
- the available power at the wheels (drive / brake)

To compute a velocity profile it is necessary to consider all four aspects.

3.2.1 Vehicle resistances

The elementary equation that describes the longitudinal dynamics of the vehicle is:

$$m \frac{dv}{dt} = F_f(t) - F_r(t).$$

(3.1)

In this equation the left-hand part represents the inertial force induced by the mass of the vehicle, where $v$ is the vehicle forward velocity and $m$, it’s mass. This inertial force is in equilibrium with the car’s net tractive force $F_t$ minus the total vehicle resistance force $F_r$. $F_t$ is generated by the prime movers minus the force that is used to accelerate the rotating parts inside the vehicle and minus all friction losses in the powertrain. Furthermore, the tractive force $F_t$ is limited by the amount of grip that the tyres are able to provide. The total vehicle resistance force $F_r$ is a summation of resistance forces and described as:

$$F_r(t) = (F_{air}(t) + F_{roll}(t) + F_{grade}(t) + F_{slop}(t)).$$

(3.2)
Here $F_{\text{air}}$ is the aerodynamic friction, $F_{\text{roll}}$ the rolling resistance, $F_{\text{grade}}$ is the force caused by gravity when driving on a non-horizontal road, and $F_{\text{slip}}$ is the drag force caused by the tyres, during cornering. The grade and rolling resistances can be computed with the following equations:

$$F_{\text{roll}} = c_r m_r g \cos(\varphi), \quad (3.3)$$
$$F_{\text{grade}} = m_r g \sin(\varphi), \quad (3.4)$$

in which $c_r$ is the rolling coefficient, $g$ is the gravity constant and $\varphi$ is the road inclination angle. Since the circuit is assumed to have no significant inclinations, the contribution of (3.4) to (3.2) is eliminated ($\sin(0)=0$) and (3.3) transforms into:

$$F_{\text{roll}} = c_r m_r g. \quad (3.5)$$

The rolling coefficient $c_r$ is dependant on several factors such as tyre pressure and road conditions, but is usually assumed to be constant. Typical values for normal passenger vehicles are in the order of 0.01 – 0.015. However, the used racing tyres have a much softer rubber compound and are also inflated to a lower pressure, $p_{\text{tyre}}$, than tyres for normal cars (Table 3.1):

<table>
<thead>
<tr>
<th>Tyre pressure $p_{\text{tyre}}$</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal road car</td>
<td>1.8 bar (±)</td>
</tr>
<tr>
<td>Formula SAE car</td>
<td>0.65 bar</td>
</tr>
</tbody>
</table>

The softer compound and lower pressure are applied to enhance tyre grip, however at the expense of increased rolling resistance and $c_r$ is therefore assumed to be 0.035.

The aerodynamic resistance force can be computed, using the following equation:

$$F_{\text{air}}(v) = \frac{1}{2} \rho_{\text{air}} A_f c_d v^2. \quad (3.6)$$

Here, $\rho_{\text{air}}$ is the air density, $A_f$ is the frontal area of the race car, and the aerodynamic drag coefficient $c_d$. The aerodynamic drag coefficient is often estimated by a comparison to similar vehicles with known coefficient. The best method is, however, to measure the aerodynamic behavior of a car in a windtunnel. From these data a corresponding drag coefficient can be calculated. Another method is to fit a second-order polynomial to the measurement data, relating drag force to wind speed. This last method is performed by the Monash Motorsport Team (Australia), and for an unwinged car they provided the polynomial coefficients in [13]. The following equation for aerodynamic drag force is formulated:

$$F_{\text{air}}(v) = (0.39v^2) + (0.18v) + 0.05 \quad (3.7)$$

This relation is visualized in Fig. 3.1.

![Fig. 3.1. Aerodynamic drag force as a function of vehicle velocity.](image)
Although in conventional considerations of the vehicle resistances the tyre cornering drag is usually neglected, this is certainly not the case for racing cars. In [84] it is shown that for high-speed corners the side slip angle of the tyres can cause an additional rolling resistance dissipation, that can easily be in the order of tens of kilowatts.

The origin of this resistance is illustrated in Fig. 3.2. In order to create a lateral force $F_y$, tyres need to have an angle with respect to the direction that they are moving at. Due to this side slip angle $\alpha$, the resulting lateral tyre force $F_y$ is not perpendicular to the direction of movement, and hence there is an additional force-component $F_{\text{slip}}$ that acts in the opposite direction of the car's longitudinal velocity.

In reality each of the 4 tyres are oriented differently with respect to the instantaneous centre of the corner, and moreover the instantaneous lateral forces of the tyres are also not the same. However, for the sake of simplicity, it is assumed that all tyres of the car are lumped, for determination of the cornering drag. Hence, $F_{\text{slip}}$ will be calculated by:

$$F_{\text{slip}}(t) = \sin(\alpha) F_{y,\text{car}}(t). \quad (3.8)$$

Taking into account that all four tyres are providing maximum lateral forces during cornering, the appropriate side slip angle for (3.8) can be derived from measurement data of the used Hoosier tyres. Fig. 3.3 shows the lateral coefficient of friction, as a function of side slip angle [48]. The maximum value is reached at $\alpha = \pm 6^\circ$.

Now that all vehicle resistances have been specified, (3.2) can be rewritten as:

$$F_r(t, v) = (0.39v^2 + 0.18v + 0.05) + (c_1, m, g) + (\sin(\alpha)F_{y,\text{car}}(t)). \quad (3.9)$$
Note that (3.9) depends on vehicle velocity, while this equation is part of the calculation sequence that determines the velocity profile. This problem is solved numerically; by taking the velocity of the previous step, only a small error is introduced in computing the aerodynamic resistance in (3.9).

The power that is needed to overcome the vehicle resistances is:

\[ P_r(t) = F_r(t)v(t). \]  

(3.10)

By integrating \( P_r(t) \) over one lap, the energy loss due to vehicle resistances can be computed:

\[ E_{r,\text{lap}} = \int_0^{t_{\text{lap}}} P_r(t)dt \]  

(3.11)

### 3.2.2 Moving the vehicle

The next step is to investigate the behavior of the tractive force \( F_t \) during racing.

As already mentioned, the velocity profile depends mainly on resistances, tyre grip, track layout and the available power. When examining acquisition data from past events, a repeating pattern can be discovered in the driver’s behavior. The driver constantly seems to be in either of one of the following situations:

1) Acceleration  
2) Braking  
3) Cornering  

This pattern repeats itself several times during a lap, according to the number of turns.

Obviously, the order is also the same every time (1 → 2 → 3 → 1 etc.).

While the car speeds up and down significantly on the straights, the cornering velocity can be assumed to be more or less constant, as the driver tries to keep the car on the edge of maximum lateral acceleration. This driving pattern can be used to approximate a velocity profile. Appendix B shows URE04 acquisition data of the endurance race in Italy, 2008, where the observed racing behavior is highlighted.
3.2.3 The track

Using this pattern as a guideline, a detailed determination of the track is performed. With the help of a ground picture of the Silverstone circuit and video footage of FS UK 2008, the track layout is reconstructed, as shown in Fig. 3.4. The derived track consists of a segmented line, which is divided into straights (red) and turns with constant radius (blue). The yellow lines indicate where barriers and cones had been placed in FS UK 2008. The whole track is assumed to have no significant road grades, and divided into small distance steps of 0.1 m. In Table 3.2 the main track properties are summarized.

![Fig. 3.4. Satellite image of a part of the Silverstone circuit with the reconstructed endurance track projected onto it.](image)

### Table 3.2 Track data

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>track length</td>
<td>891.9 m</td>
</tr>
<tr>
<td>step size</td>
<td>0.1 m</td>
</tr>
<tr>
<td>number of steps</td>
<td>8919</td>
</tr>
<tr>
<td>resulting endurance length (25 laps)</td>
<td>22.298 km</td>
</tr>
<tr>
<td>maximum length of a straight</td>
<td>71.8 m</td>
</tr>
<tr>
<td>number of turns</td>
<td>15</td>
</tr>
<tr>
<td>number of slaloms</td>
<td>4 (3 small, 1 large)</td>
</tr>
<tr>
<td>maximum / minimum turn radius</td>
<td>41.2 m / 4.96 m</td>
</tr>
</tbody>
</table>
3.2.4 Bicycle model

In the model the race car is represented as a bicycle [45,77], which means that the left and right tyres are lumped together. The advantage of this type of model, is that the dynamics remain quite simple, moreover it allows to incorporate an important aspect in the evaluation: longitudinal load-transfer. Every car that is subject to acceleration will experience a shift in wheel load, which is called load transfer. Because of this, more or less tyre traction force is available at each wheel. In general, less load transfer provides a higher level of grip.

Although lateral load-transfer is also very important for race car handling, it is assumed that the drive train topology will not have much effect on it. Therefore it is not taken into account. Longitudinal load transfer however, has an important influence since it limits the amount of tractive force that can be applied by the drive train and braking system. Moreover, Formula SAE cars have a fairly short wheelbase (needed for high maneuverability), which results in fairly large load transfer, when compared to other (race) vehicles.

In Fig. 3.5 a side view of the intended bicycle model is depicted. It basically consists of two wheels and a point mass, the centre of gravity (COG). Important dimensions of the model are also shown here; \( L \) is the car’s wheelbase, whereas \( L_1 \) and \( L_2 \) are the distances of the COG to the front and rear axle respectively. The height of the COG with respect to the ground is denoted as \( h_{\text{COG}} \), while \( r_w \) is the wheel radius.

![Fig. 3.5. Side view representation of the bicycle model](image)

Fig. 3.6 shows the same bicycle model, now including the forces that act on the car during accelerating or braking. \( F_z \) and \( F_x \) is the total force on the COG, caused by gravity and accelerations. \( F_{z1} \) and \( F_{z2} \) are the front and rear wheel load respectively, while \( F_{x1} \) and \( F_{x2} \) are tractive wheel forces that originate from applying wheel torques \( T_1 \) and \( T_2 \).

The maximum tractive force that can be applied by the wheels is not only dependant on the amount of weight transfer but also on the tyre behavior. This will be discussed in the next section.

![Fig. 3.6. Forces and moments in the bicycle model.](image)
3.2.5 Tyre modeling

Tyres provide tractive force (either longitudinal or lateral, or a combination) through friction with the ground. The magnitude of the available tractive force is dependant on the friction coefficient $\mu$, which is defined as:

$$\mu_{x,y} = \frac{F_{x,y}}{F_z} \quad \text{(3.12)}$$

In turn, the friction coefficient is dependant on numerous variables, such as tyre temperature, wheel slip and load [77]. In this model the tyres are assumed to be at constant temperature. Furthermore, $\mu_x$ and $\mu_y$ are modeled to be operated at optimal slip conditions and only dependant on wheel load. From [48] the relationship between optimal friction coefficient and wheel load for the used tyre is taken and depicted in Fig. 3.7, where $\mu_x$ is represented by a dashed line, and $\mu_y$ by a gray line.

Clearly, $\mu_y$ of the Hoosier tyre is more sensitive to wheel load than $\mu_x$. From Fig. 3.7 it can also be concluded that a heavy car will have less grip than a light weight car with the same tyres. Using this graph, the following relationships are derived:

Longitudinal:

$$\mu_{x,i}(F_{z,i}) = 1.65 - 4 \cdot 10^{-5} \frac{F_{z,i}}{2} = c_1 - c_2 F_{z,i} \quad \text{(3.13)}$$

Lateral:

$$\mu_{y,i}(F_{z,i}) = 1.62 - 1 \cdot 10^{-4} \frac{F_{z,i}}{2} = c_3 - c_4 F_{z,i} \quad \text{(3.14)}$$

In (3.13) and (3.14) $F_{z,i}$ represents the total axle load and needs to be divided by two, in order to get the individual tyre load, needed for the determination of $\mu$. 

---

Fig. 3.7 Tyre friction coefficients as a function of tyre load.

Tyre data [48]:
Hoosier C2500
$p_{\text{tyre}} = 0.8$ bar
$T = \pm 35 \, ^\circ C$
### 3.2.6 Acceleration

The tractive force during acceleration can be calculated, by considering the situation in Fig. 3.6. All forces that act on the wheels and COG must be in equilibrium, and with known vehicle parameters and (3.13), the system can be solved for $F_{x,i}$. Since this research assumes that only rear-wheel driven cars are to be evaluated, $F_{x,1}$ consequently reduces to zero.

The acceleration force $F_{x,2}$ is given by:

$$F_{x,2} = \mu_{x,2} c_2 F_{z,2} = c_1 F_{x,2} - c_2 F_{z,2}^2,$$  \hspace{1cm} (3.15)

where the value of $F_{z,2}$ can be determined by the following equation:

$$F_{z,2} = \frac{-(L_1 + L_2 - c_1 h_{COG}) + \sqrt{(L_1 + L_2 - c_1 h_{COG})^2 - 4(c_2 h_{COG})(-L_1 mg)}}{2(c_2 h_{COG})},$$  \hspace{1cm} (3.16)

which is only dependent on vehicle and tyre parameters. In Appendix C.1 the derivation of (3.16) is presented.

### 3.2.7 Braking

Similarly, the tractive forces $F_{x,2}$ and $F_{x,2}$ during braking can be determined. Now both the front and rear wheels contribute in decelerating the vehicle:

$$F_{x,1} = \mu_{x,1} F_{z,1} = c_1 F_{x,1} - c_2 F_{z,1}^2;$$  \hspace{1cm} (3.17)

$$F_{x,2} = \mu_{x,2} F_{z,2} = c_1 F_{x,2} - c_2 F_{z,2}^2;$$  \hspace{1cm} (3.18)

The values of $F_{z,1}$ and $F_{z,2}$ are determined by:

$$F_{z,1} = \frac{-(2c_2 h_{COG} m g - L_1 - L_2) + \sqrt{(2c_2 h_{COG} m g - L_1 - L_2)^2 - 4(2c_2 h_{COG})(m g + m g L_2 - c_2 h_{COG} m^2 g^2)}}{2(-2c_2 h_{COG})},$$  \hspace{1cm} (3.19)

and:

$$F_{z,2} = F_z - F_{z,1} = (m g - F_{z,1}).$$  \hspace{1cm} (3.20)

In Appendix C.2. a derivation of (3.19) is given.

### 3.2.8 Cornering

For the calculation of the cornering characteristics, only the maximum attainable velocity of the vehicle needs to be computed, since it is assumed that the car is cornering steady-state. Calculation of this velocity is an approximation and based on the formula for centrifugal force:

$$F_y = \frac{m v^2}{R},$$  \hspace{1cm} (3.21)

where $F_y$ is the centrifugal force, and $R$ is the corner radius. $F_y$ is counteracted by the two (maximal) lateral tyre forces $F_{y,1}$ and $F_{y,2}$, as shown in Fig. 3.8.
The maximum velocity of the car in an arbitrary corner with constant radius is given by:

\[ v_{\text{max}} = \sqrt{\frac{R}{L} \left( \left( c_3 \frac{L g}{L} - c_4 \frac{m L g^2}{L^2} \right) + \left( c_3 \frac{L g}{L} - c_4 \frac{m L g^2}{L^2} \right) \right)} . \]  

(3.22)

Appendix C.3. shows how (3.22) is derived.

### 3.3 Preliminary velocity profile

Now all ingredients are available to calculate a preliminary velocity profile, that results from the bicycle model driving on the proposed track. This velocity profile assumes that the drive train and brakes have a surplus of power and that the tyres are the only limiting factor. The total vehicle mass is given by:

\[ m_v = m_{\text{empty}} + m_{\text{driver}} + m_{\text{drivetrain}} , \]  

(3.23)

where \( m_{\text{empty}} \) is the car mass without a drive train installed, and \( m_{\text{driver}} \) and \( m_{\text{drivetrain}} \) are the driver and drive train masses respectively. \( m_{\text{drivetrain}} \) is taken to be 80 kg for now. All necessary parameters are based on the URE05 and shown in Table 3.3.

According to (3.1) the car’s acceleration \( a \) can be determined at every distance step by evaluating:

\[ a_i = \frac{F_{i,i} - F_{i,i}}{m_v} , \]  

(3.24)

where \( i \) denotes the step number. The new velocity \( v_i \) of the car at the end of step \( i \) is calculated by rearranging the equation for uniform acceleration:

\[ v_i = \sqrt{v_{i-1}^2 - 2a_i s} , \]  

(3.25)

where \( s \) is the stepsize (0.1m). This method of approximating the velocity makes use of the trapezoidal rule for numerical interpolation. (3.25) needs the end velocity of the previous step, but obviously this value is not yet generated at the start of the simulation. To solve this, the lap end velocity is calculated first, where the car exits a corner with constant speed.

In Fig. 3.9 the build-up of the velocity profile, as a function of distance is shown. Firstly the constant corner velocities are determined (dashed lines). The next step is to calculate the braking velocity profiles backwards (thin lines), from the start of every corner. This will make sure that the car enters the corner with the appropriate speed. After that the acceleration profiles are added, starting at every end of a corner. By taking the minimum of these 3 profiles, the total \( v(x) \) profile is derived, which is indicated by the thick line.
Table 3.3. Parameter values for the simulation model.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Empty vehicle mass</td>
<td>$m_{\text{empty}}$</td>
<td>160</td>
<td>kg</td>
</tr>
<tr>
<td>Driver mass</td>
<td>$m_{\text{driver}}$</td>
<td>68</td>
<td>kg</td>
</tr>
<tr>
<td>Drive train mass</td>
<td>$m_{\text{drivetrain}}$</td>
<td>80</td>
<td>kg</td>
</tr>
<tr>
<td>Rolling coefficient</td>
<td>$c_r$</td>
<td>0.035</td>
<td></td>
</tr>
<tr>
<td>Wheel radius</td>
<td>$r_w$</td>
<td>0.25</td>
<td>m</td>
</tr>
<tr>
<td>Wheel base</td>
<td>$L$</td>
<td>1.6</td>
<td>m</td>
</tr>
<tr>
<td>Distance front axle to COG</td>
<td>$L_1$</td>
<td>0.8</td>
<td>m</td>
</tr>
<tr>
<td>Distance rear axle to COG</td>
<td>$L_2$</td>
<td>0.8</td>
<td>m</td>
</tr>
<tr>
<td>COG height</td>
<td>$h_{\text{COG}}$</td>
<td>0.28</td>
<td>m</td>
</tr>
<tr>
<td>Optimal tyre side slip angle</td>
<td>$\alpha_{\text{opt}}$</td>
<td>0.1</td>
<td>rad</td>
</tr>
<tr>
<td>First longitudinal coefficient</td>
<td>$c_1$</td>
<td>1.65</td>
<td></td>
</tr>
<tr>
<td>Second longitudinal coefficient</td>
<td>$c_2$</td>
<td>$2 \cdot 10^{-5}$</td>
<td></td>
</tr>
<tr>
<td>First lateral coefficient</td>
<td>$c_3$</td>
<td>1.62</td>
<td></td>
</tr>
<tr>
<td>Second lateral coefficient</td>
<td>$c_4$</td>
<td>$5 \cdot 10^{-5}$</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 3.9. Construction of the preliminary velocity profile.
The $v(x)$ profile can now be converted to a $v(t)$ profile, and other variables are derived. In this case also $a(t)$ and $P_{1,2}(t)$ are shown in Fig. 3.10.

This first simulation already shows some interesting facts. First of all, the lap time is 55.1 sec. By dividing the lap length by this time, we get an average velocity of 58.1 km/h.

It can also be seen that the positive accelerations are in the order of $10 \text{ m/s}^2$, while decelerations are around $15 \text{ m/s}^2$. The explanation is simple; the car can use both front and rear wheels for decelerating, while for acceleration it only has the rear wheels.

In Fig. 3.10 the last subfigure shows the net applied power at the front and the rear axle ($P_1$ and $P_2$ respectively). The maximum drive power has a value of 121.3 kW, which is not realistic for a class1 Formula SAE car; maximum drive powers usually are in the order of 40 – 75 kW. Another interesting value that can be derived from this first evaluation is the applied brake balance, resulting from the chosen vehicle parameters: 77 / 23 % (front / rear). This large front value is due to the short wheel base and the high coefficient of friction of the tyres, which causes a high amount of longitudinal load transfer.
3.4 Topologies

In this section the drive train topologies of interest are discussed. As already highlighted in Section 2.1, the number of allowed power sources, propulsion systems and drive train topologies in class1A is vast and it is out of the scope of this report to evaluate them all. Therefore a first reduction is made, which is based on practical arguments.

First of all, propulsion systems based on an Internal Combustion Engine (ICE); the main choice to be made is about the type of fuel that it uses. In Appendix A the options are summarized and furthermore the allowed amount of displacement is given. Since the engine is preferably an off-the-shelf part, the selection is restricted to commercially available ICE’s up to 250cc (with the exception of diesel: 310cc). Practically all engines with this displacement are powered by petrol [50], so running them on another fuel type will require moderate to extensive modifications. The displacement is also a concern; the allowed value of 250cc will severely limit the available power (estimated to be about 23.5 kW maximally). Combining the ICE with another propulsion system is therefore preferable, resulting in a hybrid drive train of some form. However, the ICE drive train, powered by petrol, is modeled in this Master’s thesis, since it will offer a benchmark for the other types, and can also be compared to results from past events.

Another propulsion system with a single prime mover is the battery electric vehicle (BEV), which is considered to be a good candidate for class1A. This type of vehicle is also in development by many companies, due to it’s good energy efficiency and zero-emission characteristics.

The other group of candidates for class1A is the hybrid vehicles. Possible combinations are for instance:

1. ICE + electric system
2. ICE + hydraulic system
3. Fuel cell + electric system
4. ICE/FC + hydraulic system + electric system

Option 1 will be evaluated in this report in the form of both a conventional parallel and series topology. More complex forms of option 1 are the mixed HEV’s [68,70], but are not taken into account.

Option 2 is discarded, since it usually associated with heavy-duty vehicles [71] and a rather poor efficiency for the hydraulic converters.

Option 3 is the subject of the Formula Zero competition [i8], but is deemed to be too complex to start with, since it involves two completely unknown technologies for URE, that need to be combined [58].

Option 4 indicates the group of hybrids with more than 2 combined propulsion systems

An example of such a system is presented in [51], but this group is also considered too complex and is very likely to result in poor reliability.

Apart from these main propulsion systems there are many other allowed subsystems and setup options in the class1A competition. These include:

- **Transmission systems:**
  - Rear / Front / All wheel drive
  - Fixed gear / Stepped gear / Continuous Variable
  - Final drive: Chain / Belt / Shaft with bevel gear
  - Differential: Open / Limited slip etc.
  - Clutch: dry / wet / torque converter

- **Internal Combustion Engine**
  - Naturally Aspirated
Turbo charged
Super charged

- Electric machine
  - Coupled via drive shaft to wheels
  - In-wheel

Since the intended car for class 1A will be a conversion of the URE05, it is chosen to place all drive components inside the existing rear frame. Furthermore, it is chosen to keep all off-the-shelf components as unmodified as possible. Final transmission systems are chosen to be conventional, from a Formula SAE point of view (i.e., RWD with chain final reduction). This will ensure reliability of components and the integrity of the used car. In the next sections, the drive train topologies are explained.

### 3.4.1 Conventional drive train (ICE)

As a benchmark, the standard ICE drive train is evaluated. In Fig. 3.11 the block diagram is depicted. Fuel is directed from the fuel tank into the ICE. The ICE converts the energy of the fuel into mechanical power. After that the mechanical power is transferred to the gearbox and subsequently to the final drive, which drives the rear wheels. ICE’s in Formula SAE are usually taken from motorcycles, due to their high specific power. These types of engines have the stepped transmission incorporated into the engine housing.

![Fig. 3.11. Block diagram of the conventional ICE drive train.](image)

### 3.4.2 Electric drive train

The electric drive train is shown in Fig. 3.12. The energy storage consists of a chemical battery. Next to that are the power electronics (PE), that regulate the amount of electrical power, exchanged between the battery and the electric machine (EM). The EM converts electrical power into mechanical power and vice versa; it can operate as a motor as well as a generator. This gives the opportunity to regenerate kinetic energy from the vehicle.

Several electric drive trains have been applied in motorsports so far. Examples of this can be found in [56,57,59].

![Fig. 3.12. Block diagram of the full electric drive train.](image)
3.4.3 Hybrid drive trains

The other two candidates are the Series and Parallel Hybrid Electric Vehicle (HEV). Both types combine an engine with an electric machine(s).

**Series hybrid**

The second candidate is the series hybrid drive train. In this topology the prime mover is an electric machine. Power for this machine is provided by an electrical generator, which in turn is powered by an ICE. This engine runs preferably at an efficient operating point. Furthermore, fluctuations in power demand are taken care of by a short-term energy storage, consisting of ultracapacitors. The choice for ultracapacitors for the hybrids will be discussed later in this chapter. The drive train is depicted schematically in Fig. 3.13. Information about series hybrids is given in [49,50,52,54,55,67,72].

![Fig. 3.13. Block diagram of the series hybrid drive train.](image)

**Parallel hybrid**

In Fig. 3.14 the block diagram of the parallel hybrid drive train is shown. The upper part is similar to the standard ICE drive train, but a parallel branch is added to the final drive. This branch consists of an electric machine, together with power electronics and a bank of ultracapacitors. The ultracapacitors are used as a short-term energy storage. The electric machine acts as a performance booster. Also, the electric branch makes it possible to store kinetic energy from the vehicle. Coupling the EM directly to the engine is also an option, but this would require significant modifications to engine, which decreases reliability. More information about parallel HEV’s can be found in [61–63,67,72].

![Fig. 3.14. Block diagram of the parallel hybrid drive train.](image)
3.5 Components

In this section all individual components are discussed, which are used in the drive trains of interest. Suitable models are presented for determining the component characteristics and efficiencies.

3.5.1 Internal Combustion Engine (ICE)

The ICE is represented by a maximum torque line and a non-linear static efficiency map that relates the mechanical output power to the enthalpic fuel input power:

\[ P_{\text{ICE}} = P_{\text{Fuel}}\eta_{\text{ICE}} \]  (3.26)

The efficiency \( \eta_{\text{ICE}} \) is a function of engine speed and engine torque. A problem is however, that Formula SAE engines have a mandatory intake restrictor, which alters their behavior significantly. Especially the maximum torque values at high RPM are lower (choked air flow), than those from engines without an intake restrictor. In [39] extensive data is given for an engine, specifically designed for use in Formula SAE. This engine is depicted in Fig. 3.15 and specifications are shown in Table 3.4.

In [39] engine efficiency is given as a function of manifold air pressure (MAP) and engine speed. Other data show the engine’s brake mean effective pressure (BMEP) as a function of engine MAP and speed. Since according to [85] the torque \( T_{\text{ICE}} \) for a 4-stroke engine is directly related to BMEP by:

\[ T_{\text{ICE}} = BMEP \left( \frac{V_d}{4\pi} \right), \]  (3.27)

the engine efficiency can be expressed as function of engine speed and torque, by combining the MAP and BMEP data given in [39].

<table>
<thead>
<tr>
<th>Name</th>
<th>UniMelb WATTARD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>Parallel twin, 4-stroke, liquid cooled</td>
</tr>
<tr>
<td>Displacement</td>
<td>433.8 cc</td>
</tr>
<tr>
<td>Bore x Stroke</td>
<td>69 x 58 mm</td>
</tr>
<tr>
<td>Air induction</td>
<td>Naturally Aspirated (NA)</td>
</tr>
<tr>
<td>Fuel induction</td>
<td>Pressure Fuel Injection (PFI)</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>13:1</td>
</tr>
<tr>
<td>Valve actuation</td>
<td>8 valve DOHC</td>
</tr>
<tr>
<td>Engine management</td>
<td>Motec M4 EMS</td>
</tr>
</tbody>
</table>

In the model, the maximum torque line (also referred to as Wide-Open-Throttle; WOT) can be scaled linearly proportional to the desired maximum output power. The efficiency map is then assigned to the obtained engine operating space.
In Fig. 3.16 the operating space is shown for a 250cc engine, with efficiency contours included. The power density of the modeled engine is taken as 1.25 kg/kW, from which the mass $m_{\text{ICE}}$ can be derived.

For selection of a suitable engine for use in hybrid FSAE race cars, the reader is referred to [53]. Extensive data about the development of the WATTARD engine can be found in [38]. Furthermore, information about FSAE engine emissions such as NO$_x$, CO and HC can be found in [43], although these emissions are not taken into account by the class1A scoring system.

### 3.5.2 Electric machine (EM)

The base EM that is used for the model is the UQM Technologies® PowerPhase75. The reason for this is the abundance of data that is supplied by the manufacturer, regarding the operation of the motor. Furthermore this motor is designed specifically for (H)EV’s. The motor is a brushless 3-phase PMDC motor, that can deliver a peak power of 75 kW. It makes use of liquid cooling. A picture of the motor is shown in Fig. 3.17. Specifications and detailed efficiency data can be found in [ds02]. Similarly to the modeling of the engine, the electric input power $P_{PE}$ coming from the power electronics, can be determined by dividing the requested mechanical power $P_{EM}$ by its operating point efficiency:

$$P_{PE} = \frac{P_{EM}}{\eta_{EM}} = \frac{T_{EM} \omega_{EM}}{\eta_{EM}(T, \omega)}.$$  \hspace{1cm} (3.29)

For negative power, the reciprocal of $\eta_{EM}$ is used. In the model the motor peak power can be scaled to any desired value, and the efficiency map is scaled linearly proportional to the new operation space. Fig. 3.18 shows the operation space with efficiency map for the UQM motor, scaled down to a peak power value of ± 50 kW.
Electric machines are usually characterized by a continuous and a peak power. The allowed duration of peak power operation is dictated by the effectiveness of the cooling system and the thermal properties of the machine. The continuous power of the PowerPhase75 is 45 kW, resulting in an overloading ratio of:

\[ r_{\text{overload}} = \frac{P_{\text{peak}}}{P_{\text{continuous}}} = \frac{75}{45} = 1.66. \]  

(3.29)

Since the electric machine is operated intermittently in the endurance race, it is assumed that during acceleration and braking peak power can be applied. The power density of the model EM is assumed to be 1.0 kg/kW. (based on peak power and additional mass of the cooling system and power electronics).

### 3.5.3 Energy storage

In the proposed topologies two main types of energy storage systems can be identified, namely a petrol fuel tank for the ICE, and electrical energy storage, either in the form of a battery or ultracapacitors. The maximum allowed capacity for electrical accumulators is 9.06 kWh, where it is assumed that only 80% of the total energy content can actually be used (7.25 kWh). Furthermore, the allowed nominal system voltage is 400 V maximum.

#### 3.5.3.1 Fuel tank

The fuel tank is simply modeled as a reservoir, that cannot deplete. The rate at which fuel leaves the tank is:

\[ \dot{m}_{\text{fuel}}(t) = \frac{P_{\text{fuel}}(t)}{h_f}, \]  

(3.30)

where \( h_f \) is the lower heating value of petrol (43 MJ/kg).

The amount of \( \text{CO}_2 \) released into the atmosphere by burning this fuel is determined by:

\[ m_{\text{CO}_2} = \int_{0}^{t_{\text{end}}} \dot{m}_{\text{fuel}}(t) dt \left( \frac{1000}{\rho_{\text{petrol}}} \right) E_{\text{Q petrol}} \]

(3.31)

Here, \( \rho_{\text{petrol}} \) is fuel density and \( E_{\text{Q petrol}} \) the equivalent \( \text{CO}_2 \)-production of one litre petrol.
3.5.3.2 Batteries

Batteries are electrochemical energy storage systems and are a key component in electric and hybrid drive trains. Secondary batteries are rechargeable; chemical energy can be transformed into electrical energy and vice versa. Chemistries that are applied most for secondary batteries are lead-acid, nickel-metal hydride and lithium-ion. The latter has the highest energy density of these three and is therefore regarded as the most suitable chemistry for a race car.

Batteries are characterized by their capacity \( Q_0 \), which is the integral of the current that can be delivered under certain conditions. The charge level \( Q_c \) of a battery is given by:

\[
Q_c = Q_0 + \int_{0}^{t} I_{\text{batt}}(t) \, dt.
\]  

(3.32)

The state-of-charge \( SOC \) is a dimensionless parameter, representing the relative charge in the battery:

\[
SOC = \frac{Q_c}{Q_0} \times 100\%.
\]  

(3.33)

Since the component models are power based, the battery characteristic is modeled as:

\[
P_{\text{batt,eff}} = P_{\text{batt,int}} - P_{\text{batt,loss}}.
\]  

(3.34)

\( P_{\text{batt,eff}} \) is the effective power measured at the terminals, \( P_{\text{batt,int}} \) the actual power stored in the battery and \( P_{\text{batt,loss}} \) is the internal loss power, generated as heat.

In this respect it is more appropriate to consider the energy stored in the battery by the state-of-energy \( SOE \), which is defined as:

\[
SOE = \frac{E_s}{E_0} \times 100\%.
\]  

(3.35)

where \( E_s \) is the remaining amount of battery energy and \( E_0 \) the initial amount, defined as:

\[
E_0 = (Q_0 V_0) \cdot 3600.
\]  

(3.36)

Here, \( V_0 \) is the nominal battery voltage. In order to compute the internal losses of the battery, use is made of the so-called \( R_{\text{eq}} \)-model. In [17] this model is verified with experiments and proven to be accurate within 10%. The \( R_{\text{eq}} \)-model assumes that the battery is in series with an equivalent resistance \( R_{\text{eq,batt}} \), which is defined as:

\[
R_{\text{eq,batt}} = R_{\text{cell}} \frac{n_s}{n_p}.
\]  

(3.37)

where \( R_{\text{cell}} \) is the individual cell resistance and \( n_s \) and \( n_p \) are the number of cells in series and parallel respectively. In Fig. 3.19 the \( R_{\text{eq}} \)-model is schematically depicted.

Fig. 3.19. Schematic overview of \( R_{\text{eq}} \)-model.
The battery internal power $P_{batt,int}$ is:

$$P_{batt,int} = V_{OC}I_{batt}$$

(3.38)

where $V_{OC}$ is the open-circuit voltage and $I_{batt}$ the battery current. $V_{OC}$ is determined by:

$$V_{OC} = V_{cell}(SOE)n_s$$

(3.39)

Furthermore, the internal loss power is given by:

$$P_{batt,loss} = I_{batt}^2R_{eq.batt}$$

(3.40)

Combining (3.34), (3.38) and (3.40) yields:

$$P_{batt,eff} = (V_{OC}I_{batt}) - (I_{batt}^2R_{batt})$$

(3.41)

Here $I_{batt}$ can be computed by rearranging (3.41) to a quadratic equation and finding the root:

$$I_{batt} = \frac{V_{OC} - \sqrt{V_{OC}^2 - 4R_{eq.batt}P_{batt,eff}}}{2R_{eq.batt}}$$

(3.42)

Once $I_{batt}$ is known, (3.34) can be solved. The battery efficiency for discharging is consequently given by:

$$\eta_{batt,dc} = \frac{P_{batt,eff}}{P_{batt,int}}$$

(3.43)

and for charging:

$$\eta_{batt,c} = \frac{P_{batt,int}}{P_{batt,eff}}$$

(3.44)

The total energy taken from the battery can be determined by integrating the internal power over one lap:

$$E_{batt,int} = \int_{t_0}^{t_{no}} P_{batt,int}dt$$

(3.45)

Finally, the amount of CO$_2$ that is generated in a power plant for supplying this amount of battery energy is given by:

$$m_{CO_2} = \frac{E_{batt,int}}{3.6 \cdot 10^6}EQ_{electric}$$

(3.46)

As an example, the discharging efficiency of a battery is given as a function of output power in Fig. 3.20. For this example, the battery configuration shown in Table 3.5 is used [i7]. A picture of this particular cell can be seen in Fig. 3.21.
Table 3.5 Battery configuration

<table>
<thead>
<tr>
<th>Cell type</th>
<th>LiFeBatt 40138 10Ah</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cell chemistry</td>
<td>LiFePO₄</td>
</tr>
<tr>
<td>Cell resistance</td>
<td>6 mΩ</td>
</tr>
<tr>
<td>Battery config.</td>
<td>102 series / 2 parallel</td>
</tr>
<tr>
<td>Nominal voltage</td>
<td>400 Volt</td>
</tr>
<tr>
<td>Total capacity</td>
<td>8 kWh</td>
</tr>
<tr>
<td>State-of-Energy</td>
<td>50 %</td>
</tr>
</tbody>
</table>

This cell type is used as a base cell for the model and battery properties, such as total mass, are determined from this unit cell.

3.5.3.3 Ultracapacitors

Ultracapacitors store energy in the electric field of an electrochemical double layer. When compared to batteries, ultracapacitors have a higher specific power density, but much lower specific energy. In automotive applications they are mostly used as short-term energy storage devices, such as power assist and regenerative braking in hybrids [8]. Furthermore they are potentially useful when applied in so-called dual storage systems, where they assist a battery in delivering high currents. This may allow for downsizing of the battery [8]. An extensive overview of available modern ultracapacitors and their performance is given in [7]. Information on modelling can be found in [9–12], and [6] presents an application of supercapacitors in motorsports.

As mentioned in Section 3.4.3, supercapacitors are a better choice for the hybrid drive trains in this evaluation, than a battery. This is because the main part of energy is supplied by the ICE, whereas the electrical energy storage is only needed for short pulses of power, both positive and negative. Based on a consideration of power capability, a pack of ultracapacitors is the most light-weight option, as shown in Appendix D.

In contrast to batteries, the voltage of ultracapacitors varies significantly during use. The stored energy $E_{UC}$ can be expressed as:

$$E_{UC} = \frac{1}{2} CV_{UC}^2,$$  \hspace{1cm} (3.47)

where $C$ is the capacitance and $V_{UC}$ the voltage measured over the terminals.

Similarly to batteries, the SOE is defined as in (3.35). The voltage $V_{UC}$ can be determined by:

$$V_{UC}(t) = \sqrt{SOE \cdot V_{acc}^2}.$$ \hspace{1cm} (3.48)

Here, $V_{0,UC}$ is the ultracapacitor voltage when fully charged, which is 2.7 Volt. The total pack resistance and voltage can be determined in a similar way as (3.37) and (3.39) respectively. Since the voltage decreases during discharge, the drawn current for a constant power output will increase accordingly. A consequence is that the losses are significant at low state-of-energy, so it is beneficial to avoid the low SOE region of an ultracapacitor. Moreover, power electronics usually have a minimum voltage threshold at which they still function properly.

The losses within the ultracapacitors are calculated using the same $R_{int}$ model as with the battery. The mass addition of the ultracapacitor pack to the vehicle is computed on basis of the average specific power of Maxwell BoostCAP cells, which is about 16 kJ/kg.
3.5.4 Transmission

Transmission systems adapt the tractive power provided by the power converter(s). Transmission components mostly used in Formula SAE are stepped gear transmissions, chain reductions and differentials. Some teams have used continuous variable transmissions, of the passive type (also known as variators). These have been adopted from snow scooters, due to their light-weight design and automatic control.

The modeling of reductions and the transmission in this report is kept fairly simple. The output torque is given by:

\[ T_{\text{out}} = \eta_{TR} \frac{T_{\text{in}}}{r}, \]  

where \( r \) is the ratio and the efficiency \( \eta_{TR} \) is assumed to have a constant value of 0.99 for each reduction.

Because a standard motorcycle engine will be used for the ICE and parallel hybrid topology, a 6-speed sequential transmission is already present. The respective ratio’s are taken from Kawasaki’s 250cc Ninja engine, and listed in Table 3.6. The distribution of the gear steps is progressive. The transmission has the selectable gears in series with a primary gear, so the efficiency of the transmission is 0.99\(^2\). Furthermore there is a final drive reduction, which can be varied in the model.

<table>
<thead>
<tr>
<th>( r_{\text{primary}} )</th>
<th>( r_1 )</th>
<th>( r_2 )</th>
<th>( r_3 )</th>
<th>( r_4 )</th>
<th>( r_5 )</th>
<th>( r_6 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.309</td>
<td>0.384</td>
<td>0.558</td>
<td>0.709</td>
<td>0.862</td>
<td>1.000</td>
<td>1.120</td>
</tr>
</tbody>
</table>

Shifting between gears is modeled to be without any time delay. This is a fairly safe assumption, since the gear boxes have a sequential selection mechanism and the actuator is modified to a pneumatic system. Shifting times in the order of 20ms can be achieved using this setup.

The selected gear is determined by assigning a maximum allowed angular velocity for the ICE, and then taking the gear that produces the highest ICE speed under this value. This method allows for avoiding high rpm regions (also referred to as ‘early-upshifting’), where the ICE efficiency is slightly worse, but at the cost of reduced maximum power.

3.5.5 Power electronics

The power electronics that regulate the power to the electric machine is simply modeled with a constant efficiency \( \eta_{PE} \) of 96\%. The output power is given consequently by:

\[ P_{\text{out}} = \eta_{PE} P_{\text{in}}, \]  

where \( P_{\text{out}} \) and \( P_{\text{in}} \) are determined by the direction of the power flow.

3.6 Drive train control strategies

In this section the control strategies for operating the different drive trains are discussed. In literature [79], several types of control strategies are presented. These can be split up into three main categories:

- Heuristic
- Sub-optimal
- Optimal
In this research the heuristic approach is adopted, in the form of a rule-based operation of the drive train components. The main advantage of heuristic controllers is that they are intuitive to conceive and rather easy to implement. Drawbacks are the fact that they need to be tuned to get the best results, and that the results are not optimal. More information about specific hybrid controllers can be found in [64,67,69,72,76].

### 3.6.1 ICE drive train

The operation of the ICE drive train is straightforward. Fig. 3.23 shows the applied ICE and brake power as a function of the desired power. The driver indicates the desired power by actuating the accelerator and brake pedal. In the case of braking, the driver always brakes maximally (-100%), which corresponds with the optimal braking force, determined in Section 3.2.7.

![Fig. 3.23. ICE operation: drive and brake power as a function of pedal position](image)

### 3.6.2 Full electric drive train

The full electric drive train operation has the same characteristic as the ICE strategy, since there is only a single drive unit. A difference is however, that for negative power request the EM also helps with slowing down the vehicle, thereby storing kinetic energy in the accumulator. Depending on the size of the EM and the available traction at the rear wheels, the EM may be able to provide all rear wheel braking power. If not, the EM power is augmented with the power from the rear service brakes. This last situation is shown in Fig. 3.24.

![Fig. 3.24. Full electric operation: drive and brake power as a function of pedal position](image)
3.6.3 Series hybrid drive train

Fig. 3.25 shows the control strategy for the series hybrid drive train. The operation characteristics of main EM drive unit is the same as for the full electric drive train. Furthermore a constant flow of electrical power is provided by the ICE-Generator combination (yellow line), which runs at it’s most efficient operating point. The difference in power from the generator and the main EM, is accommodated by the short-term accumulator (green line); When power demand is high, both the generator and the accumulator deliver power to the EM. For low power demands, the excess of generator power is temporarily stored in the accumulator. During braking the accumulator is charged by the generator as well as the main EM. The series hybrid’s accumulator is chosen to be charge-sustaining. By choosing the right size of the generator and the accumulator, it is assured that this requirement is met and that the accumulator stays within it’s allowed state-of-charge boundaries.

The following rule-based equations can be formulated for the stored or extracted power $P_{UC,int}$ at the ultracapacitor energy reservoir:

High rear wheel power demand : $P_{PE} > P_{EG}$

$$P_{UC,int} = \left( \frac{P_2}{\eta_{TR}\eta_{PE}\eta_{EM}} - P_{EG,\text{out}} \right) \frac{1}{\eta_{UC}}$$  \hspace{1cm} (3.51)

Low rear wheel power demand : $P_{PE} < P_{EG}$

$$P_{UC,int} = \left( \frac{P_2}{\eta_{TR}\eta_{EM}\eta_{PE}} - P_{EG} \right) \eta_{UC}$$  \hspace{1cm} (3.52)

Negative rear wheel power demand : $P_{PE} < 0$

$$P_{UC,int} = \left[ (P_{EM}\eta_{EM}\eta_{PE}) - P_{EG} \right] \eta_{UC}$$  \hspace{1cm} (3.53)

The required fuel power is constant during the race and determined by:

$$P_{fuel} = \frac{P_{EG}}{\eta_{EG}\eta_{ICE}}$$  \hspace{1cm} (3.54)
3.6.4 Parallel hybrid drive train

The strategy for the parallel hybrid topology makes use of three basic operation modes. These modes have the intention of making full use of the available EM. First of all, the EM captures kinetic energy during braking. Secondly, the EM augments the ICE power during high power demand, thereby boosting acceleration performance. The third mode makes use of the fact that at low power demand, the available ICE power may not be fully exploited. By putting the EM in generator mode, and raising the ICE power, additional electrical energy is produced, which can be used later for acceleration. An advantage is that the amount of part-load operation of the ICE is reduced this way, which enhances overall ICE efficiency. The proposed 3-mode operation is illustrated in Fig. 3.26. An important parameter [64,66] is the ratio between EM power and total power, often referred to as the “hybridization factor” (HF):

\[ HF = \frac{P_{EM}}{P_{total}} = \frac{P_{EM}}{P_{EM} + P_{ICE}}. \]

Furthermore, the dashed line in Fig. 3.26 can be shifted left and right, which influences the intensity of the electrical system charging behavior, during periods of positive power demand. All simulation input parameters will be optimized by hand, with the focus on:

- charge sustainability of the electric accumulator.
- keeping the accumulator state-of-energy within the allowed boundaries.
- minimizing lap time and fuel consumption.

![Fig. 3.26. Parallel hybrid operation: Component power as a function of pedal position](image)
Chapter 4

Simulation

4.1 Introduction

In this chapter the 4 drive train topologies are simulated using the modelling method of Chapter 2 and 3. The 2009 race car, the URE05, will be used for the vehicle parameters, as stated in Table 3.3. The car is simulated on the endurance track of the Silverstone circuit. The main parameter that will be varied is the maximum available drive power, \( P_{\text{max,rear}} \). This will be discussed in the next section. After that, simulation results for each topology are presented. At the end of this chapter, the drive train topologies are being evaluated both on the quantitative results of the simulations, as well as on qualitative characteristics. Finally, a drive train topology and preferred configuration is selected and motivated.

4.2 Simulation parameters

A race car’s lap time and energy consumption is to a large extent determined by the available drive power. Obviously, a very powerful car goes faster, but will also have an increased energy usage. Therefore, each topology is evaluated for a range of drive powers:

\[
P_{\text{max,rear}} = [7, 5 \ 10 \ 12.5 \ 15 \ 20 \ 25 \ 30 \ 35 \ 40 \ 45 \ 50] \ \text{kW}.
\] (4.1)

Other parameters, for instance the final drive ratio \( r_{\text{final}} \), are of less influence. For each value of \( P_{\text{max,rear}} \), other input parameters are optimized, with the focus on minimization of lap time and energy consumption, subject to constraints, such as charge-sustainability. Due to the short computation time of the model, this can be done on basis of trial and error. This leads to 4 sets of 11 optimized “drive train configurations”, which can then be analyzed with respect to each other, and to results from past events. To assess the effect of a slippery track, the simulation is also done for wet conditions. By halving the tyre \( \mu \)-values, this track condition can be implemented. Regenerative braking is disabled for wet track conditions, since in [49] it is suggested that this is likely to lead to vehicle instability.

In contrast to the preliminary simulation of Section 3.3., the available tractive force of each drive train is taken into account as well, which results in slower acceleration phases.

\[
F_{x2}(t) = \min \left( F_{x2,\text{tyre}}, F_{x2,\text{drivetrain}}(t) \right)
\] (4.2)

Deceleration phases remain the same, since it is assumed that the hydraulic brakes have overcapacity. Furthermore, component sizes are now specified, so the drive train mass can be determined accurately and added to the empty vehicle mass.
4.3 Results

First the individual topology results will be discussed in Section 4.3.1, and after that all results will be compared with each other in Section 4.3.2.

4.3.1 Individual topology results

ICE drive train

The standard ICE topology is simulated for the power range as stated in Section 4.2. There are two other parameters that are optimized. The first is the angular velocity at which the engine shifts up. This parameter is empirically determined at 10,000 RPM for all ICE simulations, since this results in the best compromise between power and efficiency. Secondly, the final drive ratio needs to be adjusted, according to the available power. This value varies gradually from 0.2 – 0.32, for low to high power respectively. In Fig. 4.1 a special case is shown, namely the resulting trajectories for a 250cc ICE vehicle on a dry track. This displacement corresponds to approximately 23.5 kW maximum output power. The lap time is 59.47 seconds. In the subfigures of Fig. 4.1 one can see the velocity, selected gear, rear wheel drive torque, the engine power and the resulting fuel consumption. From the consumed amount of fuel the equivalent CO₂-production can be calculated. Although ICE drive trains with more power are simulated, this ICE drive train configuration is the most powerfull one, that is still allowed by the rules.

Fig. 4.1. Results for a 250cc ICE drive train topology
The fuel consumption / laptime trade-off of the ICE drive train topology is depicted in Fig. 4.2. In the left figure simulation results for a dry track are illustrated, where maximum power on the rear wheels is varied from 7.5 to 50 kW. As expected, lap times decrease and fuel consumption increases for increasing drive power. The part where more than 250cc for propulsion is needed, is indicated in gray. In the right figure results are depicted for the ICE drive train on a wet track. Lap times are about 20 seconds higher than for a dry track. This is quite much, so halving the tyre $\mu$-values is probably too drastic. Nevertheless, it gives a good indication of the effect that the tyres have. It is remarkable that at some point (23.5 kW) the lap times start increasing again, for increasing drive power. From this point on, fuel consumption also increases rapidly. The explanation is that the tyres are the limiting factor for the whole length of the track, so additional power does not improve performance anymore. Meanwhile, the vehicle weight increases and the $\mu$-values decrease, which has a negative impact on the lap times. Due to the fact that the ICE is being operated in part-load conditions more and more (for increasing power), the overall efficiency also decreases, resulting in even higher fuel consumption.

**Full electric drive train**

The electric drive train is mainly characterized by the EM and the battery size. A larger EM is able to consume more energy and, consequently, a larger battery is needed to complete the full endurance. The battery is sized such that the state-of-energy decreases from 100% to a minimum of approximately 15%. The final drive ratio is adapted, in order to keep the EM speed under it’s limit and in the constant-power region, where efficiency is good. In Fig. 4.3 results are shown for a drive train configuration, which uses a 35 kW EM. This value is interesting, because for completing the endurance a 9 kWh battery is needed, which is the maximum allowed capacity. Again, the trajectories for velocity, rear wheel torque and power are depicted. The lower subfigure of Fig. 4.3 shows the total efficiency of the drive train. The average efficiency value is about 84%.

![Fig. 4.2. Overview of the ICE drive train topology on a dry (left) and wet track (right).](image-url)
The state-of-energy trajectory is calculated by repeating the simulation for 25 laps, and using the previous SOE end-value for each next lap simulation. In Fig. 4.4 the state-of-energy is shown for the 35 kW electric drive train, as a function of endurance time.

In Fig. 4.5 the results for the range of drive powers are shown. Again, the gray part indicates the situation where the drive train power does not comply with the competition rules anymore. For the wet track, the same behavior is seen as with the ICE drive train. However, at high drive power, the increase in energy consumption is not as large as for the ICE drive train. This is due to the fact that the part-load efficiencies of an electric machine are not as bad (relatively), as those of an internal combustion engine.
Series hybrid drive train

For the series hybrid topology, several parameters have to be optimized. The final drive ratio is adjusted to the size of the EM. Furthermore, the size of the ICE/Generator combination and the ultracapacitor pack is adjusted to ensure that there is always enough energy for the main EM. The ultracapacitor pack is charge-sustaining over one lap and the state-of-energy is kept with 100% and 40%. This will ensure efficient operation of the pack. The ICE is also operated at its most efficient point (sweet-spot), which is 33.6%. A consequence is that the ICE then delivers only 68% of the maximum power. Furthermore, the required ICE may not be larger than 250cc (23.5 kW max). If more power is needed, then the only option is to shift the engine operating point of the 250cc engine to a higher RPM, where more power is available, but efficiency is worse. In Fig. 4.6 a special case is shown again. These are the simulation results of a drive train with a 24.3 kW main electric motor. At this EM size, the ICE is just able to supply the requested average power at the sweet-spot. The velocity trajectory is shown in the upper subfigure, and below that the power of the EM, the generator and the ultracapacitor is shown. The last subfigure shows the state-of-energy of the ultracapacitor pack. The lap time is 59.95 seconds.
Fig. 4.6 clearly shows the control strategy of the series hybrid. Power from the ultracapacitor is positive during high power demand, whereas it is negative for low power demand (i.e. cornering) and braking periods.

The series hybrid drive train results for the range of drive powers are shown in Fig. 4.7. On the left one can see the dry track performance results. More than 40 kW on the rear wheels is useless, since it will not decrease laptime anymore. Furthermore, the 250cc ICE / Generator combination has reached maximum power output, which is 23.5 kW at 11,000 RPM. The region where the ICE can still be operated at the sweet-spot is indicated between 7.5 and 24.3 kW. From this point on, ICE efficiency drops, which has the effect that fuel consumption increases even faster. The characteristics on a wet track (right subfigure) are quite similar to those of the full electric drive train. This is to be expected, since both drive trains use an electric machine as means of propulsion.

The parallel hybrid drive train has the most parameters that need to be optimized. Again, the shift-up speed of the ICE is set to 10,000 RPM. Also, the final drive needs to be adjusted, to make sure that the ICE stays within it’s speed boundaries at lowest and highest gear. Furthermore there is an extra reduction that couples the EM to the final drive. Tuning of this reduction ensures that the EM stays within the efficient area of operation. The size of the ultracapacitor pack ranges from 30 kJ to 300 kJ energy capacity, for increasing rear wheel power. Again, SOE upper and lower limits are set to 100% and 40% respectively. The hybridization factor $HF$ is ranging from 0.06 to 0.5 for low to high rear wheel power respectively.

Fig. 4.8 shows results for a parallel drive train with a 250cc engine. The combined power of the ICE and the EM is 40 kW. First, the velocity profile is shown, and below that power trajectories for the rear wheels, the ICE and the EM. The state-of-energy is shown in the last graph. The lap time is 56.88 seconds. It can be seen that the control strategy of the parallel hybrid leads to the same behavior as the series hybrid, since both drive trains are set to store ICE power during low power demands and to capture kinetic energy during braking. The result is a very similar SOE trajectory.
The results for the parallel hybrid simulation are summarized in Fig. 4.9. The rear wheel power is varied from 7.5 kW up to 50 kW for the dry track. No competition rules are violated for any of the configurations. At wet track conditions the simulation is done up to 40 kW, since no improvements are expected in lap time. The parallel hybrid shows the same behavior for wet track conditions as the ICE drive train; fuel consumption increases rapidly, as the engine is operated at part-load more and more.

Fig. 4.9. Overview of the parallel hybrid drive train topology on a dry (left) and wet track (right).
4.3.2 Comparison of topologies

In this section the final results for all topologies are presented in two figures, that plot all drive train configurations as a function of lap time and equivalent CO₂-production. The results for a dry track are shown in Fig. 4.10.

![Graph showing comparison of topologies](image)

Fig. 4.10. Results overview of all drive train simulations on a dry track.

The following conclusions can be drawn from this graph:

- The series hybrid topology does not lead to any substantial fuel reduction, when compared to a conventional ICE topology. It is able to deliver faster lap times than the ICE (which is limited to 250cc), but at a high efficiency penalty.
- Both the electric and the parallel drive train are more energy efficient than the standard ICE drive train. Savings range from about 20% for low power drive trains to approximately 40% for high power drive trains.
- The electric drive train is limited by the allowed battery capacity, which results in this topology not being able to reach very fast lap times ( < 58 s).
- The parallel topology is able to reach the fastest lap times, while still providing good fuel consumption values.
In Fig. 4.11 all simulation results for wet track conditions are shown. The differences in performance are now smaller, due to all drive train topologies being very limited by their tyres. However, the following remarks are made:

- Again, the series hybrid shows no reduction in fuel consumption, when compared to the standard ICE drive train.
- The electric drive train has the lowest CO\(_2\) production for low to medium power.
- The parallel drive train is able to produce faster lap times than the electric and the series hybrid.
- Both the electric and the series hybrid drive train are less affected (in terms of energy efficiency) by being “over-powered”.

![Graph showing CO\(_2\)-production vs. Laptime for different drive train types](image)

Fig. 4.11. Results overview of all drive train simulations on a wet track.

4.4 Comparison with past events

In this section, results from the simulation will be compared with past event results, to validate the model accuracy. For this purpose, the class1 results from Formula Student UK 2008 are taken, since these cars have raced on the same track as proposed in Section 3.2.3. In class1 events not only the lap times, but also the fuel consumption is measured and published, so these figures can be converted to CO\(_2\)-production, using the class1A rules for CO\(_2\)-equivalency.

The cars of class1 make use of engines with displacements ranging from 400 to 600cc. There are also a number of cars that use E85 as fuel, which is a blend of 85% ethanol and 15% petrol. Since lap times are awarded with substantially more points than fuel economy in class1 (350/50 vs. 200/200 points), the cars are tuned mostly towards performance.
In Fig. 4.12 the ICE simulation (7.5 – 56.5 kW) is plotted together with results from the class1 2008 event. The track conditions were dry. It can be seen that the distribution of the class1 results is quite large. However, they are in the proximity of the high power ICE simulation (35 – 56.5 kW). Differences between the model outcome and reality can be explained by variations in:

- Driver behavior; not all drivers drive the car on the limit of the car’s capabilities.
- Weather and track conditions, temperature.
- Vehicle setup (suspension etc.).
- Engine tuning; some car’s have a badly tuned engine, with rather poor efficiency.

In Fig. 4.12 two interesting car’s are highlighted. First, team Renstall Uni Stuttgart, the winner of the 2008 event. This car was incredibly fast, with average lap times more than 4 seconds lower than the rest. The other car is from Delft University of Technology Racing. This car is characterized by it’s lightweight design (± 150kg total) and efficient drive train, running on E85. With this car, they won the award for most fuel-efficient car, and the second place overall.

### 4.5 Discussion and topology choice

Now that all drive trains have been evaluated quantitively, the final choice can be made. In order to do that, it is sensible to consider other, more practical, aspects and issues of the topologies as well. Therefore, the main advantages and drawbacks of each topology are summarized below:

**Full Electric**

+ Very energy-efficient, resulting mainly from the characteristics of the used components itself.
+ A gearbox is not required, which makes the car more easy to race with.
+ With a full electric race car, URE can also participate in the new Formula Student Electric competition (in Germany).
+ Straight-forward operation of the drive train, no complex control strategies required.

− Li-Ion batteries are associated with safety issues, which may be a concern.
− Monitoring the battery state-of-charge is critical, for completing the endurance.
− Charging the battery takes a long time (> 1h), which may cause difficulties and inconvenience on testing days. Building spare batteries will raise project costs significantly.
Series hybrid

+ The control strategy is less complex to implement than the parallel hybrid strategy.
+ A series hybrid has a larger EM than a parallel hybrid with the same total power, which results in an increased potential of regenerative braking.

− Hybrids are essentially more complex systems to be developed by a FSAE team, than a full electric drive train. This is due to the fact that there are two completely different systems that need to function correctly, and moreover need to interact with each other.

− Energy in a series hybrid is converted from mechanical to electrical, and back to mechanical again. This is a fundamental drawback of the series hybrid topology. It is also considered as the main reason why this topology does not offer a fuel reduction in the simulation results.

− The selected ICE needs to be examined and tuned thoroughly, in order to operate at the efficient sweet-spot.

− A series hybrid has two EM’s which increases weight and cost.

Parallel hybrid

+ The parallel hybrid is an efficient and fast option.
+ If the high voltage electrical system fails, the parallel hybrid is still able to continue the endurance, by driving solely on the engine.
+ The parallel topology is, given a certain total power output, very likely to be the most low-cost option.

− See first drawback of series hybrid.

− The potential of regenerative braking may be reduced by the size of the EM and the additional friction losses of the engine (when not decoupled).

− The control strategy is difficult to realize, since both propulsion systems are coupled to the wheels. Charging and boosting may prove to be disturbing to the driver, if not controlled properly. In this respect, producing electrical energy is more difficult than with the series hybrid.

Taking all advantages and drawbacks into consideration, the final choice is made by the complete team of URE (including previous board members). The drive train topology that will be used for the 2010 race car, is Full Electric.

The following facts have been decisive for this choice:

• The option to participate in more than one race.
• The class1A competition in 2009 has been won by a race car with a full electric drive train, proving the capabilities of this concept.
• The full electric drive train is a futuristic system, that is gaining a lot of interest. Much publicity is to be expected, which is in favour of sponsorships, a critical factor for a race team.
Part II

Vehicle design
Chapter 5
Vehicle specification

5.1 Introduction

In part I of this report, a full electric drive train is selected for the URE race car of 2010. In this chapter an overview of the vehicle requirements is given, which is mainly dictated by the competition rules and the intended purpose of the vehicle.
The car will be fielded in two competitions - class1A in the UK and at Formula Student Electric - which complicates things. There are differences in the event rules, so it needs to be taken into account that the car has to comply with both sets of rules.
After stating the requirements in Section 5.2, the model of part I is used to formulate a preliminary vehicle specification, by providing key values for the the intended vehicle design in Section 5.3. Furthermore, electric drive trains offer the possibility of incorporating some innovative techniques that can be beneficial to racing. These are discussed in Section 5.4.
Since the new car will be an electric conversion of the URE05, it is consequently called the “URE05e”.

5.2 Competition rules and requirements

First of all the most important vehicle design objectives, stated by the competition rules, are:
• The vehicle should adopt alternative powertrain technologies such that it is high performance, whilst the amount of energy it consumes is as low as feasibly possible and the CO$_2$ and energy embedded in the vehicle is minimized.
• The vehicle should have high performance in terms of acceleration, braking and handling and be sufficiently durable to successfully complete all the events described in the FSAE rules.
• Additional design factors to be considered include: aesthetics, cost, ergonomics, reliability, maintainability, manufacturability and recyclability.

The most important system constraints and mandatory / allowed features are:

1) Maximum allowed electrical power is 75 kW.
2) Maximum allowed voltage of the High Voltage system (HV) is 400 Volt DC or AC rms.
3) Maximum allowed battery capacity is 7.25 kWh (at 80% depth-of-discharge).
4) The chassis must fulfil at least the 2007 FSAE Rules.
5) All power train system components must lie within the surface defined by the top of the roll bar and the outside edge of the four tires.
6) HV systems and HV wiring must be contained within the primary structure of the frame and when located less than 350mm from the ground must be protected from side or rear impacts.
7) Charging is allowed in the car.
8) An extensive safety system must be present and consist of at least the following:
   a. Floating HV system (HV completely isolated from the car).
   b. Battery contactor relays that disconnect the battery in case of any error.
   c. Ground fault detector (GFD)
   d. Battery management system (BMS)
5.3 Vehicle specification and design optimization

For reliable operation and competitive racing, two critical factors are identified for the design of the new electric race car.

I. Total mass of the car.
II. The available energy in the battery

The mass of the car is restricted by the current design of the URE05’s chassis and suspension. The URE05 weighs approximately 235 kg, so adding too much mass can bring the integrity of the vehicle in danger during racing.

Secondly, the full electric simulation results (Ch.4) point out that the available battery capacity imposes constraints to the maximum drive power; if the motor has too much power, and state-of-charged is not monitored cautiously, the car runs out of energy prematurely.

This has led to the decision to focus on minimization of component weight, and to select the maximum allowed battery capacity. Furthermore, an acceleration time and top speed target are set as drive train performance indicators. In Table 5.1 the main vehicle design specifications are stated.

<table>
<thead>
<tr>
<th>Table 5.1. Main vehicle specifications and targets</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design target weight</td>
</tr>
<tr>
<td>Weight distribution (F/R)</td>
</tr>
<tr>
<td>Top speed</td>
</tr>
<tr>
<td>Acceleration 0-75m</td>
</tr>
<tr>
<td>Endurance lap time</td>
</tr>
<tr>
<td>Battery capacity</td>
</tr>
</tbody>
</table>

The ideal weight distribution is determined in [90].

Since there is only one fixed reduction, a good compromise is required between top speed and rear wheel torque. Therefore, the top speed is set to a moderate 100 km/h, the 35 kW simulation of Chapter 4 indicates that this top speed is encountered once. A top speed of 100km/h corresponds to a rear wheel angular velocity of 1060 RPM.

Another important design parameter is the maximum EM power. In Chapter 4 it was shown that a 9 kWh battery restricts the EM power to 35 kW. Of course, it is possible to select a higher power EM, and to limit the power during the endurance. This will be in favour of the 75m acceleration event, where the size of the battery is only dictated by the amount of requested power. From past class1 results, an acceptable time of 4.5 seconds for the 75m acceleration event is chosen. The simulation model is used to determine the minimum power, needed for achieving an acceleration time of less than 4.5 seconds. The endurance track is replaced by a single straight of 100m, and the start velocity is set to zero. The stepsize has also been reduced.
In Fig. 5.1 the simulation results are shown for a 260kg heavy race car with a 40kW electric motor. The time needed to complete 75m is 4.31 seconds. A final drive ratio of 0.14 is used, to enable a top speed of 100 km/h. However, the model does not take rear wheel slip into account. In [49] it is shown that this will have a negative effect on the acceleration times. Therefore, a 40 kW motor (peak power) is the minimum power requirement, but a bit more is preferred. The model also predicts that the longitudinal tyre grip limits the useful rear wheel torque to about 900 Nm.

5.4 Additional drive train features

Formula SAE rules are based for a large part on safety, and the result is that the typical Formula SAE racing environment is unusual, when compared to many other motorsports classes. The short length of straights and tight corners limit the maximum velocities significantly, which makes the application of aerodynamic features very difficult to be effective. The net effect is a tremendous emphasis on low speed handling. Mechanical grip is very important and any technique that can improve low speed handling is of great interest. In this respect, application of drive train control systems, such as traction control and anti-lock braking systems have always been popular themes in Formula SAE. Equipping a race car with an electrical drive system opens completely new possibilities for improved vehicle dynamics. Electric motors generally have a high gravimetric and volumetric power density. This enables a large degree of design freedom, in terms of motor location and packaging. The fact that electric motors use flexible wiring for their energy supply, enhances this design freedom even more. As a result, electrical vehicles are often associated with independent wheel drive systems; multiple motors are installed, sometimes directly mounted inside the wheel hub. The application of more than one electric motor in the design of the URE05e is a design choice that is certainly evaluated, provided that suitable electric motors can be found for this purpose. An independent wheel drive system makes it possible to apply “torque vectoring” to the vehicle. This means that the direction of the vehicle can be controlled by applying more or less torque to the left or right wheels, in addition to steering the vehicle. This makes it possible to:
• improve vehicle stability (by counteracting over- and understeer)
• allow for higher manueverability (especially at low speeds)
• enable higher levels of combined grip, by exploiting the full grip potential of each individual tyre. This is beneficial to corner entry and exit situations.

A downside of such a system is that it needs additional motor controllers, as well as a supervisory controller that determines the set points for each motor controller. Furthermore, smaller motors usually have a slightly worse efficiency than a large one. The increased weight due to having more controllers and wiring is partially offset by the fact that a mechanical differential is not needed anymore.

If a drive system with multiple motors is selected, the supervisory controller can also be programmed to incorporate:

• launch control
• torque vectoring using both positive and negative torques.

For the URE05 the most viable solution for torque vectoring is to apply two motors; one at each rear wheel. In order to keep the unsprung mass as low as possible, it is best to mount the motors inside the rear frame, and to couple them via conventional drive shafts to the wheels. Additional front wheel drives are assumed to be too drastic for the URE05 conversion; it would require extensive modifications to the carbon fibre monocoque and involves even more control complexity, which is not preferable at this moment. All-Wheel-Drive (AWD) is however a very interesting option for future electric race cars of University Racing Eindhoven.

For more information about torque vectoring in (hybrid) electric vehicles, the reader is refered to [86,87]. Applications in motorsports applications can be found in [88,89], although these systems make use of mechanical differentials to realize torque vectoring.
Chapter 6
Component selection

6.1 Introduction

This chapter presents the component selection procedure and the results of this. On basis of the proposed system specifications of Chapter 5, components are being selected, and compared on several aspects, such as weight, compliance with the rules, cost and availability.

The selection procedure is as follows: first a suitable electric motor is chosen. After that, a final drive ratio and construction concept can be selected. The chosen motor directly influences the type of controller, so this component can also be selected. The system voltage is determined by the motor / controller combination and consequently a battery can be configured with cells that meet the specifications. Once these main components are known, other necessary parts can be selected and added, in order to complete the drive train system.

6.2 Electric motor

The selection of the electric motor is a challenging task. Drive motors for electric vehicles (often called traction motors) form a unique group, that has very different requirements than standard industrial motors. The major requirements are summarized in [2,5] and are the following:

- high instant power and high power density
- high torque at low speed for starting and hill climbing
- very wide speed range, including constant-torque and constant-power regions.
- high efficiency over wide speed and torque ranges.
- high efficiency for regenerative braking.
- high reliability and robustness for various environmental conditions.
- reasonable cost

Fig. 6.1 illustrates the standard characteristics of a traction motor used in electric vehicles. This characteristic corresponds to the profile of the tractive effort versus speed of the drive wheels.

Although the number of motor working principles and construction is very diverse, it is suggested in [5] that three main types of traction motors are considered as the current main candidates for application in an electric vehicle. These are:

- DC motor (brushed)
- AC induction motor
- Permanent magnet brushless motor

Fig. 6.1. Typical torque and power curve for an electric motor
DC motors

DC motors is a term that refers to the group of motors, where the rotor windings are supplied of electricity by a mechanical commutator. The commutator makes use of sliding brushes, that need maintenance. Due to the presence of the mechanical commutator, these motors usually have a limited voltage and speed range. DC motors use simple power electronics, when compared to the other two motor types. Moreover, the technology is mature and well available at low cost. DC motors that have windings on both the stator and the rotor are categorized as series and shunt DC motors. Another group is the PMDC motors, where the stator windings are replaced by permanent magnets (PM). PMDC’s are characterized by a higher power density and are mainly available in small to medium motor sizes (< 30 kW).

AC induction motors

This type of motor is widely accepted as one of the most potential candidates for electric vehicles. It does not make use of a mechanical commutator, instead it supplies the rotor with power by means of electromagnetic induction. Induction motors are often used for industrial motors, but have been obsolete in electric vehicles in the past, due to a number of drawbacks, such as a relatively bad starting torque. With the advancement of semi-conductor electronics, many obstacles for induction motor application in EV’s have been solved. However, the control electronics are still expensive and therefore this motor type is mainly available in medium and large sizes. (> 30 kW).

Permanent magnet brushless motors

By virtually inverting the stator and rotor of the PMDC motor, the brushless permanent magnet motor is created. The stator windings are fed by a rectangular AC waveform. The main advantages of this machine are the high power density and efficiency, as well as very low levels of maintenance. Also, generated heat can easily be removed from the stator, which is in favor of a high continuous power density. Drawbacks are a relatively high cost for the electronics and the motor itself. Also, the technology is quite new and still in development.

More information about motor types and how they work can be found in [3,78,83].

For the selection of the URE05 electric motor, the following requirements are set:

1. The power density must be as high as possible, to meet the 260kg vehicle mass target.
2. High zero-speed torque for good starting acceleration.
3. The maximum motor speed is preferably lower than 6000 RPM, which makes it possible to use a single stage chain reduction (low weight and high efficiency)
4. Liquid cooling generally adds extra weight and increases complexity. This is not preferable.
5. The cost of the motor and power electronics must fit in the team budget.

Due to the fact that the motor will be used in a racing application, the following aspects are considered less important:

1. Maintenance. The service life of the URE05e is significantly less than for a passenger vehicle.

A comprehensive overview has been made of available traction motors, in the range of 10 to 80 kW. Table 6.1 shows the motors and their respective performance figures.
### Table 6.1. Overview of available traction motors

<table>
<thead>
<tr>
<th>Brand / Type</th>
<th>Mass [kg]</th>
<th>( P_{\text{max}} ) [kW]</th>
<th>( T_{\text{max}} ) [Nm]</th>
<th>Top speed [RPM]</th>
<th>Cooling</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>PMDC brushed</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Perm PMG132</td>
<td>10</td>
<td>11.2</td>
<td>38</td>
<td>4000</td>
<td>Air</td>
</tr>
<tr>
<td>LEMCO 170</td>
<td>8.5</td>
<td>21</td>
<td>52</td>
<td>4000</td>
<td>Air</td>
</tr>
<tr>
<td>LEMCO 200</td>
<td>10.6</td>
<td>34.3</td>
<td>81.3</td>
<td>4000</td>
<td>Air</td>
</tr>
<tr>
<td><strong>AC Induction</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Siemens 1LH</td>
<td>41.5</td>
<td>80</td>
<td>123</td>
<td>10000</td>
<td>Liquid</td>
</tr>
<tr>
<td>Brusa ASM6</td>
<td>49</td>
<td>54</td>
<td>192</td>
<td>11000</td>
<td>Liquid</td>
</tr>
<tr>
<td>Brusa HSM6</td>
<td>53</td>
<td>82</td>
<td>223</td>
<td>11000</td>
<td>Liquid</td>
</tr>
<tr>
<td>Symetron P-42</td>
<td>41.5</td>
<td>42</td>
<td>200</td>
<td>6000</td>
<td>Liquid</td>
</tr>
<tr>
<td>GMS AC M1</td>
<td>15.5</td>
<td>27.5</td>
<td>72</td>
<td>4000</td>
<td>Liquid</td>
</tr>
<tr>
<td>Solectria AC24</td>
<td>38</td>
<td>47</td>
<td>82</td>
<td>12000</td>
<td>Air</td>
</tr>
<tr>
<td>Solectria AC24LS</td>
<td>40</td>
<td>47</td>
<td>92</td>
<td>12000</td>
<td>Air</td>
</tr>
<tr>
<td><strong>PM brushless</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>PERM PMS120</td>
<td>13</td>
<td>14</td>
<td>40</td>
<td>6500</td>
<td>Air</td>
</tr>
<tr>
<td>PERM PMS150</td>
<td>22.3</td>
<td>27</td>
<td>80</td>
<td>6500</td>
<td>Air</td>
</tr>
<tr>
<td>PERM PMS156</td>
<td>28.5</td>
<td>34</td>
<td>80</td>
<td>6500</td>
<td>Air</td>
</tr>
<tr>
<td>UQM HiTor</td>
<td>41</td>
<td>50</td>
<td>440</td>
<td>6500</td>
<td>Liquid</td>
</tr>
<tr>
<td>UQM PowerPhase75</td>
<td>41</td>
<td>75</td>
<td>240</td>
<td>8000</td>
<td>Liquid</td>
</tr>
<tr>
<td>Siemens ACWS 80</td>
<td>22</td>
<td>50</td>
<td>60</td>
<td>12500</td>
<td>Liquid</td>
</tr>
<tr>
<td>PMAC Mars</td>
<td>11</td>
<td>13.1</td>
<td>41</td>
<td>3500</td>
<td>Air</td>
</tr>
</tbody>
</table>

![Graph showing specific power and specific rear wheel torque for motors of table 6.1.](image)

**Fig. 6.2.** Specific power and specific rear wheel torque for motors of table 6.1.
Although some motors in Table 6.1 do not comply with the 40 kW requirement, they can be used in a double arrangement, as indicated in Section 5.4.

To compare the motors on performance, they are plotted in Fig. 6.2, that shows the specific power and torque of each motor. A number of motors directly stands out when looking at this figure. First of all the UQM HiTor motor (41kg) delivers exceptional starting torque. However, when adding a suitable final drive of about 1:6, the resulting rear wheel torque would be about 2700 Nm, which is too high. All motors from Siemens, Brusa and UQM deliver good performance, but their prices are very high (all above 10.000 euro including controller). Perm has some well priced motors, but specific power is rather low.

An excellent candidate is the LEMCO 200, which has the highest specific power of all motors, and weighs only 10.6kg. Furthermore, the price is very reasonable (<2000 euro), and DC controllers are also available at low cost. Another advantage of this motor is the top speed, which is 4000 RPM. This enables the application of a single stage reduction. To achieve good acceleration performance, it is best to have two of these motors, which results in an effective rear wheel torque of about 640 Nm.

Concluding, the LEMCO 200 motor is chosen as the best option for the URE05e. Appendix E.1. shows a list of the motor specifications. A drawback is that manufacturers of DC motors usually offer little data about the complete motor operation area. This is also the case for the LEMCO motor. It is however fairly simple to construct the characteristics from the given data. Since torque is linear proportional to motor current, torque can be determined. The LEMCO motor is able to handle a peak value of 400 A over the complete speed range, for a duration of about 10 seconds. This results in a flat torque line of 81.3 Nm. Peak power therefore increases linearly with speed. The continuous current is dependant on motor speed, since the motor is cooled by the movement of the rotor. Data is also given for the continuous current rating as a function of motor speed. The resulting performance of the LEMCO 200 motor is shown in Fig. 6.3. It can be concluded that this motor only has a constant torque region and that high power is not yet available at low vehicle speed. It is also confirmed by several users that this motor can be overloaded to about 500 A, albeit for only a very short time.

In order to gather more knowledge about the efficiency and thermal limitations of this motor, a test rig has been built. More information about this test rig can be found in Appendix E.2. For literature information about overloading of motors, the reader is referred to [4].
6.3 Final drive

The chain reduction ratio is dictated by the maximum speed of the motors (4032 RPM) and the desired vehicle velocity (100 km/h):

$$r_{\text{final}} = \frac{\omega_{\text{max, wheel}}}{\omega_{\text{max, motor}}} = \frac{1060 \text{ RPM}}{4032 \text{ RPM}} = 0.263$$ (6.1)

This can be realized by taking a 12teeth motor sprocket and a 45teeth wheel sprocket. It is chosen to leave some room for adjustment on the track so the following sprocket sets will be ordered:

<table>
<thead>
<tr>
<th>motor sprocket</th>
<th>wheel sprocket</th>
<th>$r_{\text{final}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>10 t</td>
<td>44 t</td>
<td>0.227</td>
</tr>
<tr>
<td>11 t</td>
<td>44 t</td>
<td>0.250</td>
</tr>
<tr>
<td>12 t</td>
<td>45 t</td>
<td>0.266</td>
</tr>
<tr>
<td>12 t</td>
<td>44 t</td>
<td>0.278</td>
</tr>
</tbody>
</table>

The design of the final drive box must allow for easy replacement of sprockets and chains. The URE05 made use of a single 520-chain. It is expected that a smaller chain type can be used. Therefore, a peak load estimation is done which is described in Appendix F. As a result a 428-chain type is chosen for the final drive.

6.4 Motor controller

Next, a suitable controller for the LEMCO 200 motors is selected. Since two motors are used, a single controller is not sufficient: the motors are able to run at different speeds, and consequently different input voltages are needed then. Ref. [2] gives extensive information on the various types of motor controllers that are used nowadays.

A PMDC motor controller is effectively a DC-DC converter, which regulates the voltage applied to the motor. The current $I_{\text{EM}}$ that flows through the motor is determined by:

$$I_{\text{EM}} = \frac{V_{\text{controller}} - V_{\text{EMF}}(\omega_{\text{EM}})}{R_{\text{armature}}},$$ (6.2)

where $R_{\text{armature}}$ is the motor ohmic resistance and $V_{\text{EMF}}$ is the (speed dependent) voltage generated by the motor itself. The controller regulates the output voltage by switching the battery input voltage on and off, at high frequency. The inductance of the motor has the effect that the current stays non-zero when the controller switches off, although there is a small current ripple. The duty-cycle (the percentage of time the controller applies battery voltage to the motor) determines the effective voltage that is felt at the motor terminals. This method of voltage regulation is called Pulse-Width-Modulation (PWM) and controllers that make use of this technique are often called “choppers”. There are several manufacturers that produce PMDC controllers: Curtis, Kelly, Alltrax, SigmaDrive and Sevcon are all well-known brands.

The requirements for the controller are:

- **Controller type:** Permanent Magnet Direct Current
- **Control parameters:** Voltage (speed) / Current (torque)
- **Output voltage:** At least 0 – 96 Volt
- **Maximum current:** 400 A
- **Continuous current:** 200 A
Furthermore, the selection criteria for the controller are: lightweight, low cost and regenerative braking capabilities. A controller that meets these requirements best is the Kelly KDHB controller. Of all DC controllers, it is capable of supplying more than 72V, and furthermore it features torque control, by regulating the output current. Most DC controllers only control motor speed. A downside is that Kelly controllers seem to be overrated in terms of stated peak current. Many users recommend to take at least 50% extra peak current, in order to achieve the desired peak value. Therefore, a 650A version is selected. Fortunately, the controller can be limited with a configuration menu, so testing will reveal the actual performance and the limiter can be set for optimal output. A full list of controller specifications is given in Appendix E.2.

6.5 Battery cells

Lithium-Ion battery cells are complex products, that are available in a wide variety of specific chemistries, cell formats, sizes and other attributes. The technology is still expensive, and often associated with safety issues; Lithium-Ion cells are prone to “thermal-runaway”, where the cell initiates an exothermic reaction above a certain temperature-level, with catastrophic results. Keeping a cell within specific voltage and temperature boundaries is therefore absolutely critical. Large capacity batteries, consisting of multiple cells, must therefore be equipped with a Battery Management System (BMS), which must fulfill the following main tasks:

- balancing the voltages of all cells
- monitoring / protecting:
  - voltage level of each cell
  - temperature of each cell
  - cell state-of-charge
  - charge and discharge current

In Fig. 6.4 the main 3 cell formats are depicted; cylindrical, prismatic and pouch. The latter is superior in terms of energy density, due to the absence of a reinforced shell. Proper fixation of this cell format may prove difficult however.

Although each manufacturer has it’s own (secret) recipe for cell chemistry, the main cathode material of most lithium-ion cells is made up of: LiCoO$_2$, LiMn$_2$O$_4$ or LiMnNiCoO$_2$ (often a combination of these). A fairly new cathode material is LiFePO$_4$, which is quite popular now, due to increased thermal stability. Specific energy is approximately 35% lower however.

Important cell attributes are:

- nominal voltage [V]
- nominal capacity [Ah]
- specific energy and power [Wh/kg and W/kg]
- volumetric energy and power [Wh/l and W/l]
- continuous and peak (dis)charge current. [C-rates]
A determination of the required peak and continuous current output during racing is done with help of the vehicle model. An advantage is that this parameter is already calculated for determination of the battery efficiency, so little modification is required. The EM model is adapted to the data from the LEM200 motor and the battery voltage is set to 96V. The autocross event (see Appendix A for explanation) is taken as the worst case scenario. Due to the short duration of this event (± 60 sec.) it is assumed that both motors can be operated at peak current during acceleration. The computed absolute battery current for an autocross lap is shown in Fig. 6.5. The solid line indicates the actual current trajectory, while the gray line represents the average current drawn. Short peaks of up to 1000 A are observed, while most peaks are in the order of 700 – 800 A. The average battery current is 260 A. These values will be used as a guideline for selecting a suitable cell type.

URE05e battery requirements

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal system capacity</td>
<td>9 kWh</td>
</tr>
<tr>
<td>Nominal system voltage</td>
<td>± 96V</td>
</tr>
<tr>
<td>Ideal cell capacity</td>
<td>93.75 Ah</td>
</tr>
<tr>
<td>Continuous / peak current</td>
<td>260 / 1000 A</td>
</tr>
<tr>
<td>Mass</td>
<td>&lt; 100 kg</td>
</tr>
<tr>
<td>Number of cells</td>
<td>&lt; 300 cells</td>
</tr>
<tr>
<td>Availability</td>
<td>Delivery in less than 2 months</td>
</tr>
</tbody>
</table>

The number of cells in series is determined by dividing the nominal system voltage by the nominal cell voltage and rounding:

\[ n_s = \text{round} \left( \frac{V_{0,\text{system}}}{V_{0,\text{cell}}} \right) \]  

Of course it is possible to take smaller cells (< 93.75 Ah), and to connect them in parallel. For instance, 2 cells of 45Ah in parallel will result in an effective capacity of 90Ah. In order to have a clear overview of all available cells and their properties, a sheet has been made with over 100 cell types from 13 different manufacturers (see “EV_battcells.xls” in digital appendix). In this sheet a battery is composed for each cell type, which is as close to 96V and 9 kWh as possible. This makes it possible to compare the cell types on the resulting battery properties, such as total mass and required number of cells. After discarding the cell types that did not meet the demands stated above, the manufacturers of the remaining cell types were contacted and asked if they could supply the cells, and for what price. A number

![Fig. 6.5. Battery current (solid line) and average current (gray) during an autocross lap.](image-url)
of them showed no interest in cooperating with us, due to safety reasons. Finally, three cell options have been considered, which are listed in Table 6.3.

Table 6.3. Comparison of 3 Li-Ion cell types for application in the URE05e battery

<table>
<thead>
<tr>
<th>Option</th>
<th>1</th>
<th>2</th>
<th>3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manufacturer:</td>
<td>LiFeBatt</td>
<td>Thundersky</td>
<td>Kokam</td>
</tr>
<tr>
<td>Cell type:</td>
<td>40138</td>
<td>LFP90AHA</td>
<td>78216216H</td>
</tr>
<tr>
<td>Cell chemistry</td>
<td>LiFePO₄</td>
<td>LiFePO₄</td>
<td>LiMnNiCoO₂</td>
</tr>
<tr>
<td>Cell format</td>
<td>Cylindrical</td>
<td>Prismatic</td>
<td>Pouch</td>
</tr>
<tr>
<td>Cell nom. voltage</td>
<td>3.3 V</td>
<td>3.2 V</td>
<td>3.7 V</td>
</tr>
<tr>
<td>Cell nom. capacity</td>
<td>10 Ah</td>
<td>90 Ah</td>
<td>31 Ah</td>
</tr>
<tr>
<td>Cycle life</td>
<td>3000</td>
<td>3000</td>
<td>800</td>
</tr>
<tr>
<td>Number of cells</td>
<td>261</td>
<td>31</td>
<td>78</td>
</tr>
<tr>
<td>Configuration</td>
<td>29S9P</td>
<td>31S1P</td>
<td>26S3P</td>
</tr>
<tr>
<td>Nominal capacity</td>
<td>8613 Wh</td>
<td>8928 Wh</td>
<td>8947 Wh</td>
</tr>
<tr>
<td>Nominal voltage</td>
<td>95.7 V</td>
<td>99.2 V</td>
<td>96.2 V</td>
</tr>
<tr>
<td>Discharge current (cont.)</td>
<td>1080 A</td>
<td>270 A</td>
<td>465 A</td>
</tr>
<tr>
<td>Discharge current (peak)</td>
<td>1260 A</td>
<td>1800 A</td>
<td>930 A</td>
</tr>
<tr>
<td>Charge current</td>
<td>270 A</td>
<td>270 A</td>
<td>186 A</td>
</tr>
<tr>
<td>Total mass</td>
<td>94 kg</td>
<td>93 kg</td>
<td>60 kg</td>
</tr>
<tr>
<td>Total volume</td>
<td>45.5 litre</td>
<td>67 litre</td>
<td>29.5 litre</td>
</tr>
<tr>
<td>Price</td>
<td>€ 15000*</td>
<td>€ 3700</td>
<td>€ 8600</td>
</tr>
</tbody>
</table>

Option 1 is characterized by a high number of cells, high weight and price. The cells are however very well capable of delivering high currents. Option 2 is the most low-cost option, but the weight is still high and the required volume is very large. Option 3 scores best in terms of volume and weight. The price is fairly high, so sponsoring is inevitable. The chemistry of these cells leads to a shorter lifespan than the other two, but 800 cycles is still more than enough for this application. Also, this cell chemistry is regarded more dangerous than option 1 and 2. However, the team is willing to take this risk, due to the great number of mandatory safety systems and precautions. The material safety data sheet (MSDS) [ds04] of the Kokam cells provides all information on proper handling of the cells. It is likely that the cells will be used in the next race car, and this is one of the main reasons to start with high quality cells right away from the beginning. The configuration of option 3, shown in Table 6.3, will be the basis of the URE05e battery. The cells measure 215x220x8mm. This size allows the cells to be stacked to each other in the battery design.

*) LiFeBatt only sells battery cells including a BMS and other integration equipment, which results in an increased price.
6.6 Electronics

In this section a number of auxiliary components are discussed. These are essential for an electric vehicle to keep working properly and safe.

Battery Management System

The BMS is selected from Elithion for its high flexibility, low cost and extensive documentation, which is supplied on their website [i4]. The Elithion BMS makes use of a main board (depicted in Fig. 6.6) and multiple small cell boards, that monitor individual cells. The Elition BMS requires little wiring, thereby reducing the “spaghetti” in the battery, when compared to other systems. The Elithion BMS can be connected to a PC for monitoring and configuration, and has also CAN capabilities.

Ground Fault Detector

The GFD (also called insulation monitoring device) is a mandatory system that checks if there are any insulation problems with the HV system. If the device senses a leakage to low-voltage ground it immediately shuts down the high voltage system by opening the battery contactor relays. The GFD for the URE05e will be a Bender IR155-2, which is depicted in Fig. 6.7. A full overview of this device is shown in [ds08].

Charger

Of course a charger is needed for recharging the Lithium-Ion battery. This charger will not be installed in the car. Li-Ion cells must be charged with a so-called CCCV profile (constant current, constant voltage). This means that the main amount of energy is charged using a constant current supply, but when the cells are nearing 100% state-of-charge (i.e. 4.2V), the charger switches to a constant voltage supply. This will ensure that the last part of charging is done more gradually, which is necessary for safety. After this the BMS will balance the cells.

The chosen charger is a Zivan NG9, and is shown in Fig. 6.8. This charger is able to supply about 8 kW (1C) to the battery. The estimated charging time will be about 2 hours with this charger. More information about the NG9 charger can be found in [ds14].

Fig. 6.6. Elithion main BMS board

Fig. 6.7. Grand Fault Detector

Fig. 6.8. Zivan NG9 battery charger
Chapter 7
Overall design

7.1 Introduction

This chapter presents and motivates the design of the URE05e, based upon the selected components. The design has been made in the 3D modeller Unigraphics NX5. One of the major design choices was the location of the battery. The result of this decision affected the design of the complete drive train system. Another important aspect is that the chassis and suspension of the car are already finished, since the old chassis of the URE05 will be used. This posed strict constraints to the available space and arrangement of the drive train.

Firstly, the electrical scheme of the electric car will be presented in Section 7.2. In Section 7.3 the packaging of components is treated, where 2 main concepts are compared to each other. This section presents a global overview of all the systems in the car, and their placement. Furthermore, a detailed weight analysis is presented in Section 7.4, together with a determination of the centre of gravity’s location. Section 7.5 deals with the final drive design. Of course, modifications have to be carried out, mainly on the URE05 rear frame. This is the subject of Section 7.6. Conclusions are given in Section 7.7.

7.2 Electrical scheme

The electrical scheme is determined for a large part by the competition rules [i1,i9]. Fig. 7.1 shows the scheme with the main systems in it. The build up is as follows; in the center one can see the HV battery. The HV supply wiring exits the battery containment and splits up to the motor controllers. Each motor controller is connected to a motor. The pedalbox has potmeters for both the throttle and brake pedal, which send signals to the motor controllers, via the differential controller. Furthermore there is a safety loop which is indicated in blue. This wiring energizes the two battery contactor relays. There are several devices placed in the safety loop, that are able to interrupt the energy supply to the contactor relays. These are the master switch, the emergency buttons, the ground fault detector and the battery management system. When the loop is interrupted, power to the motors is shut down, and no high voltage will be present outside the battery containment. The charger is also shown in the scheme. This charger can be connected to the battery by temporarily disconnecting the motor controllers from the battery and inserting the charger connectors in the battery. Of course there are many smaller subsystemes and circuitry needed to complete the whole wiring of the URE05e. It is however out of the scope of this report to discuss this.
Fig. 7.1. Schematic overview of the URE05e electrical system.
7.3 Location of drive train components

With the main components known, a packaging design can be established. Two concepts are compared and the most suitable one is selected and further worked out:

A. 2 battery containments in side pods
B. 1 battery containment in rear frame

Fig. 7.2 shows the concepts.

<table>
<thead>
<tr>
<th>A</th>
<th>B</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image1" alt="Diagram A" /></td>
<td><img src="image2" alt="Diagram B" /></td>
</tr>
</tbody>
</table>

Fig. 7.2. Two packaging concepts illustrated: Battery in side pods (left) and battery in rear frame (right).

In this figure one can see the battery (blue), controllers (yellow), the motors (red) and the final drive construction (green). In both concepts the battery consists of 2 stacks, where each stack has 39 cells placed face to face. The difference is that in concept B both stacks are located in a single containment. Furthermore, the cells in concept A are mounted vertically, whereas in concept B they must be placed horizontally, due to the available space in the rear frame. The final drive construction is the same for both concepts, with the exception of the reduction box dimensions.

The concepts are compared by their qualitative characteristics. This is done by summarizing the advantages and drawbacks of both. Table 7.1 presents an overview.
From Table 7.1 it can be concluded that concept A has quite a large number of drawbacks. In fact, the 260kg weight target might not be achieved with concept A. The packaging of concept B fits the existing design of the URE05 very well. Therefore it is chosen to adopt concept B for the further design of the drive train.

### 7.4 Mass analysis

The weight distribution of a race car is an important aspect, that has a large effect on the overall handling. The definite placement of the main masses can now be determined accurately, and from this the weight distribution can also be estimated.

The front load percentage $FP$ of the longitudinal weight distribution is calculated by:

$$FP = \frac{\sum (m_i x_i)}{\sum m_i L} \times 100\% ,$$

(7.1)

where the index $i$ denotes a component or assembly, $m$ it’s mass and $x$ the location with respect to the origin of the coordinate system (= center of rear wheels). $L$ is the wheel base.

The rear percentage $RP$ is simply given by:

$$RP = 100\% - FP .$$

(7.2)

Before assigning component locations it is necessary to determine the longitudinal weight distribution of the URE05, without any drive train components. Of course a measurement would be the best solution for this, but the car has so far not been weighed without the conventional ICE drive train installed. Therefore a detailed examination is carried out on the car components and the driver. A complete overview of this is given in appendix G. Individual components have been weighed and their respective center of masses has been determined with the help of the URE05 3D model.
In Fig. 7.4 a top view of the URE05 is shown with the main dimensions along it. The COG of the car without a drive train is located 984mm from the rear axis, which results in a load distribution of 61/39 % (front/rear). The mass/COG of the driver is very dominant in a FSAE car, and deviations from the standard person, that is used for calculations, can alter the distribution by a few percent. The ideal placement of the drive train COG (blue) is also indicated in Fig. 7.4, which results in a 53/47% total weight distribution. Table 7.2 shows the estimated drive train component masses and desired locations. The locations are also based on the packaging of concept B from Section 7.3.

Table 7.2. Desired location and mass of main drive train components

<table>
<thead>
<tr>
<th></th>
<th>m [kg]</th>
<th>x_{COG} [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motors</td>
<td>21.2</td>
<td>140</td>
</tr>
<tr>
<td>Final drive</td>
<td>10</td>
<td>25</td>
</tr>
<tr>
<td>Controllers</td>
<td>6.6</td>
<td>330</td>
</tr>
<tr>
<td>Battery</td>
<td>80</td>
<td>525</td>
</tr>
</tbody>
</table>

Apart form these main components a number of additionally required systems will add weight to the car, and their location and mass is not yet known. A full overview of all component masses and locations will therefore be given in Section 7.7.
7.5 Final drive design

With two small motors it is possible to incorporate an electronic differential by connecting each motor to a separate rear wheel. This allows the motors to run at different speeds and torques. Two identical chain reductions are necessary and the whole final drive construction can be made symmetrical with respect to the center plane of the vehicle. This is favourable, since many final drive parts can be designed and ordered the same for left and right. Also, less spare parts are needed. In Fig. 7.5 a schematic concept overview is depicted of the proposed final drive construction. The chain reductions (blue) are housed in a structural box. The mandatory chain guard, a 3mm thick steel plate around the chains, can be used to form part of this box, thereby enhancing the box stiffness.

![Fig. 7.5. Schematic overview of final drive construction](image)

Next, the definite design of the final drive is presented. The exact engineering has been performed by other team members and FEA calculations have been carried out to ensure sufficient strength and stiffness. Fig. 7.6 shows the design. In the left image a top view is depicted, with the chain guard removed. The right image is a section view of the final drive, through the plane of symmetry. The motors are placed as low to the ground as possible. Replacement of parts is simple; by removing the chain guard sprockets and chains can be changed. The motor is attached with 8 M6 bolts. The side walls of the final drive box consist of 2 lightweight aluminium plates of 0.8 kg each. The rear sprockets are mounted in these plates with hub pods and each pod has 2 bearings for supporting the chain forces. The pods also enable the chain to be tensioned; the inner and outer diameter of the pods are eccentric, so by rotating the pod, the centre distance between the front and rear sprocket can be adjusted. The complete final drive construction (motors excluded) weighs 9.0 kg.

![Fig. 7.6. The final drive construction.](image)
Fig. 7.7. Rear frame modifications: The existing rear frame (black) has been augmented with a tube structure in the middle (purple). This assembly forms the intermediate support for the final drive, the controllers and the battery. The controllers are mounted on an aluminium plate (yellow) that is fixed at the tubes. Small brackets (blue) enclose the battery containment. Furthermore, the supports for the wheel connection rods consist of welded structures (green).
7.6 Rear frame modifications

In this section the modifications of the rear frame are treated. All structures that are added have the main task of supporting other components. The necessary structures are:

- final drive suspension
- controller suspension
- battery suspension structure
- wheel suspension attachment structure

Although several concepts have been compared, it is out of the scope of this report to show them all. Only the final design is presented. This final design is shown in Fig. 7.7 and the build-up is as follows:

A “middle” structure had to be added to the existing rear frame of the URE05, in order to support the new components. This is achieved by designing several steel tubes (purple), that will be welded to the frame. Because this middle frame is not supported at frame nodes, a cross member tube is added below the battery. Within the middle tubes, a plate is mounted (yellow) that holds the two controllers.

The battery is not fixed at the frame. Instead, it is enclosed by small brackets (blue), similar to the construction of a flightcase. The front and upper brackets are demountable, so that the battery can be positioned into the rear frame from the direction of the monocoque. The battery contact surfaces of the brackets are covered with rubber sheets, which provides some dampening of vibrations to the battery.

Furthermore, a new construction had to be designed for attaching the front connection rods of the rear wheel suspension, to the frame. In the old design these were connected via tubes to the engine. For the new design there was little space, due to the battery being very close. Meanwhile the construction needed to be very stiff and lightweight at the same time. Another function of the construction is to form a small side impact structure for the battery. The resulting design (green) is shown in Fig. 7.7. All rear frame modifications have been engineered by team member Mark Versteegde, including FEA analysis and impact calculations. The combined weight of the whole assembly is 7.2kg.

7.7 Conclusions

In this section the final result of the mass and COG analysis for the complete car is given. Therefore, the definite designs are taken into account, including the final weight of the battery. For a full overview of this analysis the reader is referred to “COG_analysis.xls” in the digital appendix.

<table>
<thead>
<tr>
<th></th>
<th>(m ) [kg]</th>
<th>(x_{COG} ) [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Empty car</td>
<td>109.75</td>
<td>880</td>
</tr>
<tr>
<td>Driver</td>
<td>71.82</td>
<td>1143</td>
</tr>
<tr>
<td>Drive train</td>
<td>141.63</td>
<td>420</td>
</tr>
<tr>
<td>Grand total of the car:</td>
<td>251.4 kg</td>
<td></td>
</tr>
<tr>
<td>Weight distribution including driver:</td>
<td>46/54 % (front/rear)</td>
<td></td>
</tr>
</tbody>
</table>
Part III

Battery design
Chapter 8

Battery concept and design

8.1 Introduction

This chapter describes the design of the URE05e battery. The battery is a key-element in any electric vehicle, and has a large influence on the performance of the vehicle. The quality of a battery is not only determined by the cell characteristics itself, but also by the integration of the cells into a complete system. To minimize the volume and weight of the URE05e battery, a dedicated cell packaging is presented that offers enough cooling capabilities to cover the endurance race. Besides, design constraints due to mechanical and electrical safety are accounted for. In [37] a comprehensive overview is presented of batteries in electric vehicles. In the next section the requirements for the battery are summed up. After that the overall design is presented in Section 8.3. and more detailed engineering is covered in Section 8.4. Finally, the complete battery design is shown in Section 8.5.

8.2 Competition rules and design specification

From the class1A and the FS Electric competition the requirements are combined and as follows:

Containment:
- Mechanically robust and fireproof material
- Rain proof (> IP45 [13])
- Rugged fixation of cells within containment
- Insulation of all electrical live parts(*)
- Removable (transparent) cover

Electrical:
- BMS monitoring: voltage, current and temperature
- Galvanic separation of low and high voltage
- Contactor relays (normally-open) at positive and negative end.
- Fuse
- HV Connectors with inter-lock
- Strain-relief for all electric wiring

For the design the next specification targets are set:

- Dimensions (LxWxH): 255x455x390 mm (Packaging, Section 7.3)
- Weight: < 80kg (Mass analysis, Section 7.4)
- Temperature: < 60°C (MSDS [ds04])
- Maximum current: 1000 A (Cell specification, Section 6.5)
- Continuous current: 250 A

Furthermore the design is preferably easy to manufacture, (dis)assemble and as low-cost as possible.

(*) Electrical live parts are electrical conductors, with the intention of carrying electricity, i.e. wiring and busbars.
8.3 Overall concept

On basis of the requirements and specifications stated above a battery concept will be established. There are several design proposals given in literature [24,25,i5,i10], which give examples of realized batteries with Kokam cells. Regarding the battery concept, [i5] proved to be most helpful, due to the extensive information provided. An important design feature is the removal of heat, generated within the cells during use. This will be discussed next.

8.3.1 Thermal management

The Kokam cells have a pouch format, giving them a large surface for transferring heat to its surroundings. In [23] it is proven that pouch cells show superior thermal management properties in comparison with cylindrical and prismatic cells. For the URE05e battery, the heat generation is in the order of 1 kW, which will be shown later. In Fig. 8.1 four different cooling concepts are depicted. Concept I & II make use of internal cooling by air (I) or water (II). For these concepts the cells need to be spaced, to allow the fluid to flow along the cells. This is not preferable, since this will increase the volume of the battery, while the rear frame offers little space. Moreover, concept II has a risk of leakage inside the battery, which could lead to a short-circuit.

The other two concepts use highly conductive plates to transfer heat to the outside of the battery, where it is removed by the vehicle air flow. With this principle the containment structure itself is used as a heat transferring medium. Concept IV is based on the use of a Phase Change Material (PCM) inside the containment. This material (for example paraffin) is able to absorb an amount of heat by changing from solid to liquid. During this transition, the temperature stays constant at the PCM melting point.

![Fig. 8.1. Four different concepts for cooling the battery.](image)

Although this is a very interesting cooling concept it is not regarded as applicable: since the cells will be mounted with the terminals sideways, the liquid PCM may enter the terminal area.

Concluding, concept III is taken for further development of the battery structure. The main advantages are the compact layout and the combination of functions into 1 system (i.e. fixture
of cells and transfer of heat). The core of the battery will consist of a cell-frame which clamps the cells tightly together. This cell-frame is then placed in an outer casing, which is also thermally well conductive. Finned heat sinks will be mounted on the sidewalls of the casing to increase the surface area with the air. For this cooling concept to work properly, it is essential that all conducting components have good thermal contact with each other. This will be achieved by adding materials that reduce thermal contact resistance and is discussed in the next section.

8.3.2 Materials

The main material used for the battery containment must be robust, fire proof and thermally well conductive. A lightweight material that complies with these demands is aluminium. Table 8.1 shows an overview of important properties of a general purpose aluminium alloy [i11].

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\rho$</td>
<td>Density</td>
<td>kg/m$^3$</td>
<td>2720</td>
</tr>
<tr>
<td>$k$</td>
<td>Thermal Conductivity</td>
<td>W/(m·K)</td>
<td>175</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Specific Heat Capacity</td>
<td>J/(kg·K)</td>
<td>900</td>
</tr>
<tr>
<td>$E$</td>
<td>Youngs Modulus</td>
<td>GPa</td>
<td>70</td>
</tr>
<tr>
<td>$\sigma_{UTS}$</td>
<td>Ultimate tensile strength</td>
<td>MPa</td>
<td>290</td>
</tr>
<tr>
<td>$\sigma_y$</td>
<td>Yield strength</td>
<td>MPa</td>
<td>250</td>
</tr>
<tr>
<td>$\tau$</td>
<td>Shear strength</td>
<td>MPa</td>
<td>200</td>
</tr>
</tbody>
</table>

A disadvantage of aluminium is the high electrical conductivity. This means that electrical live parts must be insulated with an additional material. Moreover, this material must be flame-retardant and preferably transparent for inspection of the electrical circuitry. A suitable candidate has been found: Lexan F2000 [ds10] is a special-grade polycarbonate, intended for electrical equipment. This tough plastic is delivered in clear sheets and can be machined and thermo-bend into the desired shape. Electrical insulation assemblies can be made with this plastic by using a heat-resistant glue; Araldite 2031 [ds17]. Furthermore, a material is needed for connecting the cell terminals together and to form the electrical path in the battery. In case of wiring this will be copper, but another option is to use “busbars”. These are custom strips of conductor and have the advantage that more diverse shapes can be used, where needed. A material for busbars could be aluminium, but this is often associated with loosening connections, due to galvanic corrosion and thermic cycling. Therefore, standard copper 2.0060 is used for the busbars, which does not suffer from these drawbacks.

For enhanced thermal contact between aluminium parts a paste will be used. Several thermal compounds are available, and for this application a silicone-based compound is selected [ds16], which has a thermal conductivity of 3 W/(m·K). For these pastes to work properly, it is necessary that the aluminium contact surfaces are clamped together (appr. 200 - 400 kPa), which results in a thin layer of interface paste. However, if the surfaces have significant irregularities, the paste will not be able to fill the resulting air gaps. This is the case with the cells, which have some surface irregularities in the order of 0.5mm. To solve this, compressible thermal pads are applied. eGRAF HiTherm 1220-AA [ds13] is chosen as a suitable padding material. Not only does this material provide good thermal contact, it also ensures proper fixation of the cells in the cellframe, due to the adhesive coating on both sides. Additionally it is flame-retardant.
8.3.3 Electrical configuration

As already mentioned in Section 6.4, the cells will be connected in a 26S3P configuration. To avoid the need for 3 separate parallel strings, the cells are connected as indicated in Fig. 8.2.

By binding the terminals of 3 cells together an “equivalent” cell of 93 Ah is created. Because of this interconnection the 3 cells have the same voltage, so balancing within a pack of 3 is not required. This method of cell connection is often encountered in laptop batteries. Each pack of 3 cells will have a BMS slave board mounted over the plus and minus pole, to monitor the pack voltage. In Fig. 8.3 the physical layout of the cell interconnection is shown. Two stacks of 39 (13x3) cells are placed next to each other and the blue line indicates how the electrical path is connected to the cell terminal side of the stacks.

8.4 Detailed design

This section deals with the design of the battery in detail. There are 5 main assemblies to be discussed, starting with the cell suspension frame.
8.4.1 Cell suspension

For mechanical protection and suspension of the cells an aluminium frame is needed that clamps the stacks together. By incorporating plates into this frame, heat can transported from the middle of the battery to the outside. Furthermore the frame will act as medium that reduces temperature differences within the cell stack. The following parts are designed for the frame:

A. Rectangular tubes (3x13)
B. Conduction plates (12x)
C. Endplates (2x)
D. Steel studbolts (9x)

In Fig 8.4 views of the cell frame are shown. The outer dimensions of the rectangular tubes sections measure 8x25.4mm, which enables 3 cells to be placed within each compartment. These tubes also have the studbolts running through.

The studbolts consist of 9 steel rods, with M5 thread on each end. These bolts clamp the endplates towards the rectangular tubes, until both the cells and the tubes are fixed with enough pressure. The compressible thermal pads assist in creating a uniform pressure distribution between the cells and the conduction plates. The endplates are milled components that have reinforcement ribs, for increased bending stiffness. Furthermore, the thickness of the conduction plates influences the heat transfer capabilities of the battery and is set at 1mm. The whole cell frame weighs 7kg.
8.4.2 Electrical connection

The cell terminals will be connected to each other by making use of copper busbars. As a rule of thumb [112] the current density within the busbars may not exceed 5A/mm² (based on average current). This means that the minimum cross-section must be at least 50 mm².

Another important aspect is the connection of electrical live parts to each other [13]. In the case of cable lugs and screw connections a very firm fixture is required for low electrical resistance. Contact surfaces must be abraded prior to assembly and loctite can be used to prevent loosening of threaded connections. Special attention has also gone to the clamping of the cell terminals. In Fig. 8.5 a view on the busbar interconnection assembly is shown. Rectangular busbar plates are placed over the cell terminals and transfer electricity to subsequent cell packs. In Fig. 8.6 a cross-section is depicted for such a cell pack interconnection. The cell terminals (A) are clamped between a base busbar (B) and a top busbar (C) which extends to the next cell pack. 8 M4 bolts are used to per top busbar for compression of the cell terminals.

![Fig. 8.5. Copper busbar plates connect the cell packs together.](image1)

![Fig. 8.6. Cross-section of cell pack interconnection.](image2)

8.4.3 Electrical insulation

The inner walls of the aluminium housing must be insulated from any live electrical parts of the battery. The material used for the insulation is 3mm Lexan F2000 sheet. A main insulation assembly (transparent blue) is designed for the busbar area, as indicated in Fig. 8.5 and 8.7. This assembly has a base plate which rests on the conduction plates of the cell frame. This base plate has slots for the cell terminals and small nodges (A in Fig. 8.7) at the side, which fit into the cell frame tubes and endplates. This ensures fixation of the isolation assembly in all directions. The electrical busbars are glued to the base plate with high-strength adhesive (see red lines in Fig. 8.6). A window of Lexan strips is glued onto the base plate, as well as a mid strip. The top plate (B in Fig. 8.7) of the insulation compartment is incorporated in the removable cover of the battery. Furthermore, small insulation boxes (C) are created around the output terminals of the busbar assembly.
8.4.4 Battery casing

The cell frame and the insulated busbar assembly are protected from the environment by an aluminium case of 1.5mm thick welded sheets. The case has a cover, that seals the containment and allows for a visual inspection of the interior. As can be seen in Fig. 8.8, an additional topbox is attached to the main case, that will accomodate the auxiliary electronics of the battery. The inner walls of this topbox are insulated with Lexan sheets as well, and furthermore, a small cover is designed for this topbox, similar to the main cover. The two covers are pulled against the containments by means of springs, as indicated in Fig. 8.8. The topbox is divided into 3 area’s, by Lexan partitions. One area is reserved for low voltage electronics, such as the ground fault detector. The other two area’s are for the positive and negative high voltage parts. Furthermore, two finned heat sinks (aluminium) are located at the side walls of the main case, that measure 320x200x10mm each. 12 bolts are used to mount the heat sinks and these bolts protrude into threaded bushes in the cell frame tubes (see Fig. 8.9). The compressive force of the bolts will ensure proper thermal contact between the cell frame, the casing and the external heat sink. In Chapter 9 the convective transfer properties of the heat sinks will be discussed.
8.4.5 Auxiliary electrical components

The battery is equipped with several electronic devices and sensors, that are needed for proper functioning. First of all the BMS slave boards, that are mounted in parallel with each cell pack, at the busbars. These boards have an additional temperature sensor [ds18], that is wired through the insulation main plate, and glued on the middle cell, where temperature is expected to be the highest. In Fig. 8.10 a section view is depicted that shows how this is done.

The rest of the electronics is located in the topbox and the following components can be identified:

A. High current fuse [ds06]
B. 2 contactor relays [ds07]
C. Current sensor [ds12]
D. Ground fault detector [ds08]
E. HV connectors [ds19]
F. LV connectors

In Fig. 8.11 the components are indicated. Furthermore the electrical path is shown with the blue lines, where the dot indicates the point where it enters the topbox area, from the main case below. The dashed boxes indicate where some free space is reserved for additional wiring and small electronics, such as LV relays, a HV LED and measuring points [i1,i9].

8.5 Overview of the final battery design

In the previous sections all subassemblies are discussed and the final design can now be presented. In Fig. 8.12 the complete battery and it’s location inside the rear frame is shown. It is notable that the topbox is located at an easy-to-reach position, without having to remove bodywork. In Appendix H a full bill-of-material (BOM) of the battery is listed, with component materials and masses. The total assembly weighs just over 85kg, which is close to the 80kg design target. However, an improvement in mass can be made by applying aluminium busbars. Research and testing of this material application is recommended. In the digital appendix all technical drawings of self-designed parts are listed [80].
Chapter 9

Battery thermal analysis

9.1 Introduction

Lithium-ion batteries are known for their high energy density, but are also sensitive to misuse. Especially the Lithium-ion polymer battery (LIPB), which makes use of traditional LiCoO$_2$ chemistry, is prone to several failure modes. Mild failures can lead to disfunctioning of the battery, but more severe cell failures can lead to dangerous short-circuits and fire hazards. An extensive analysis on these conditions is the subject of [14,33,35]. In [i6] a comprehensive overview is presented on Lithium-ion safety and possible failure modes and effects. It can be concluded that all individual Kokam cells must be kept within allowed limits [ds03,ds04] regarding:

- Voltage (2.7 to 4.2 V)
- Temperature (max 60°C)
- Current (-62 to 310 A)

Furthermore, damage to the cells will, most likely, lead to leakage of chemical electrolyte, and possible short-circuiting of the internal electrodes. The protection structure of the battery is designed to prevent physical damage to the cells and the BMS will take care of the voltage, temperature and current limits; if one of the limits is reached the BMS will shut down the HV system. It is therefore of great interest how the battery will behave during the endurance race, regarding temperature. In this chapter a method is presented, where the proposed design is analyzed on the transient temperature behavior and distribution within the cells. First, the problem and goals are defined in Section 9.2 and after that the type of modelling is discussed in Section 9.3. Subsequently it is explained in Section 9.4 how the proposed design will be analyzed, using this modeling method. Determination of important parameter values is the subject of Section 9.5, where parameters are derived from both experimental and analytical techniques, as well as literature research. Finally, the simulation results are presented and discussed in Section 9.6.

9.2 Goal of the analysis

Battery usage results in heat generation in the cells; the cell temperature will rise from the initial temperature to either a steady-state temperature where the battery is in equilibrium with the environment, or a final temperature when the battery is depleted. The main problem that can be defined here is:

- Is the cell temperature limit exceeded during the endurance race, when making full use of the available battery capacity?

The goal of this chapter is to answer this question, and therefore, it is necessary to analyse the battery system from a thermal perspective. Ref. [81] is used for providing insight to the governing thermal equations and parameters of a heat transfer system. An analysis method must be adopted to calculate the temperature distribution within the battery as a function of time. This is the subject of the next section. Another subquestion is how long it will take after the endurance race to cool the battery to a temperature that allows recharging.
9.3 Modelling approach

The thermal system of the battery can be described by a combination of bodies that exchange heat. The cells generate heat and the result is a heat flux in the direction of the coolest objects, which are the heat sinks. The heat is then removed from the heat sinks by air flow.

![Fig. 9.1. Schematic overview of heat transfer system](image)

Many studies have been conducted on the thermal management of batteries, and different methods of evaluating the temperature behavior are presented. In [15] an extensive analysis is performed of a battery cell, that takes electro-chemical behavior into account, as well as thermal characteristics. In [30,31] systematic approaches are presented for designing a battery thermal management system (BTMS). Some studies [16,20] calculate the temperature of the battery on basis of analytical equations, while others adopt numerical methods. Often the results are compared with experimental results [26] for validation of the proposed analysis. A common method for examining the temperature behavior numerically, is to use finite-element-analysis (FEA). 1D FEA for battery heat transfer problems is found in [14,34], while [18,21,24,27,29] present detailed 2D/3D FEA studies.

Due to the repetitive nature of the battery cell lay-out, the analysis domain can be reduced to a small section of the battery. It is assumed that every section behaves in the same way. It is chosen to use two-dimensional FEA for analysis of a battery section, which enables a detailed evaluation to be performed. The PC program COMSOL MultiPhysics® is used for computation of the system.

9.4 2D Heat transfer model

The elementary equation that describes the transient temperature distribution within a two-dimensional heat transfer system is:

\[
\rho C_p \frac{\partial T}{\partial t} = k_x \frac{\partial^2 T}{\partial x^2} + k_y \frac{\partial^2 T}{\partial y^2} + Q. \quad (9.1)
\]

At the left-hand side of this equation \( \rho \) is the material density, \( C_p \) is the specific heat, \( T \) is temperature and \( t \) represents time. At the right-hand side \( k_x \) and \( k_y \) are the material thermal conductivities in the \( x \)- and \( y \)-direction respectively, while \( Q \) is the volumetric heat source.

First it is necessary to define assumptions and to specify the domain, which allows the system to be computed using the FEA technique:

**Assumptions**

- Due to battery pack symmetry, the model can be reduced to 1/52\(^{\text{th}}\) of the total stack (= 1½ cell)
- End effects of the battery are neglected (at the stack end for instance)
- Material properties such as specific heat are independent of temperature and cell SOC.
• Within the battery pack only conduction is assumed to be present, at the outside the heat is carried away by forced air convection. Radiation is assumed to be absent.
• Heat generation is constant and uniformly distributed within cell
• The air flow is constant and uniformly distributed over the finned heatsink

In Fig. 9.2 a 3D view of the battery section is depicted, where component shapes have been simplified. The red surfaces indicate where the plane of symmetry is located. The blue surfaces show the location of convective heat transfer and the green surfaces indicate the thermal contact area’s (not all visible).

![Fig. 9.2. 3-dimensional view of the section](image)

In Fig. 9.3 a more detailed cross-section is shown for the domain of interest. Here the body interfaces and thermal pads are depicted as well.

![Fig. 9.3. Schematic view of the 2-dimensional domain.](image)

**Boundary conditions**

All boundary conditions in the model are defined using the so-called *Neumann* conditions, which specify the outward heat flux. For the dashed lines of symmetry (see Fig. 9.3) the boundary conditions are defined as:

\[ n \cdot (k\nabla T) = 0, \] (9.2)

where \( n \) is the outward surface normal and \( k \) is the material thermal conductivity. At the outer surfaces of the heat sink (blue lines in Fig. 9.3) the heat flux is set equal to the convective heat flux:

\[ n \cdot (k\nabla T) = h_{\text{conv}} (T - T_{\text{amb}}) \] (9.3)
where \( h_{\text{conv}} \) denotes the convective heat transfer coefficient and \( T_{\text{amb}} \) is the ambient air temperature. The value of \( h_{\text{conv}} \) is determined in Section 9.5.4.

**Body interfaces and heat source**

The interfaces between aluminium parts are modeled as thin layers of thermal compound material, with a thickness of 25µm [22], whereas the thermal pads are system bodies with a thickness of 0.5mm. Conductivities of all materials will be stated in Section 9.5.3. The volumetric heat source \( Q \) [W/m³] can be derived from the total heat generation rate divided by the total cell volume, and this value will be assigned to both cell bodies in the domain.

**Domain geometry and mesh**

In Fig 9.4 the exact geometry and generated mesh in COMSOL is illustrated.

![Fig. 9.4. Image of the domain geometry and mesh.](image)

**9.5 Thermal properties and parameters**

To produce realistic results it is necessary to specify accurate values for the thermal properties of the system. Summarizing, the following parameters must be determined:

- Battery heat generation
- Material specific heat capacities
- Material thermal conductivities
- Convective heat transfer coefficient

**9.5.1 Battery heat generation**

The heat generation can be derived from the vehicle model of part I, where the battery losses are calculated according to the efficiency. These losses result in the release of heat in the battery. The model makes use of the proposed method of [17], namely the \( R_{\text{int}} \)-model. Therefore, the vehicle model for an electric drive train will be used again, but now with cell properties of the Kokam cells. The volumetric heat source \( Q \) will be derived from the model using:

\[
Q = \frac{\int_{t=0}^{t_{\text{final}}} P_{\text{batt,loss}}(t)dt}{t_{\text{final}} V_{\text{cells}}}.
\]  

(9.4)

Here, \( t_{\text{final}} \) is the total endurance time and \( V_{\text{cells}} \) the total cell volume. It is necessary to compute the complete endurance with the model, since the open-circuit voltage \( V_{\text{OC}} \) slowly decreases during depletion of the battery. This means that the required battery current (which influences the efficiency) increases proportional, in order to deliver the requested battery output power (\( P=VI \)). The \( V_{\text{OC}} \) trajectory as a function of state-of-charge for the Kokam cells [ds20] is inserted in the model. Another important cell attribute for calculation of the losses is the internal cell resistance. For a given cell, the internal resistance behaves non-linear and depends mainly on the following factors:
• temperature (in general: $T \uparrow$, $R \downarrow$)
• state-of-charge
• discharge / charge current
• cell age (in general: age $\uparrow$, $R \uparrow$

Unfortunately little data is available for the selected 31Ah cell. The inspection sheet shows an average impedance value of 0.53m$\Omega$, however, this is measured using a 1kHz AC test. The true DC resistance is higher, due to the AC test not capturing the ionic cell resistance that appears after a delay. See [15,36] for more information on cell resistance. In this research the cell resistance $R_{int,cell}$ is taken as constant for both charging and discharging. Ref. [17] shows that this is a safe assumption. The value of $R_{int,cell}$ will be estimated in two ways:

1. Comparison with another, similar cell.
2. Verification of the resistance using the kokam discharge curves.

In [ds21] the DC resistance for a GAIA cylindrical 27Ah cell with the same chemistry is given, as well as the AC impedance value. These are 2.0 and 0.5 m$\Omega$ respectively. Since the AC impedance of this cell corresponds very well with the Kokam cells, it is assumed that the DC resistance for both cells must be in the same order as well. Therefore a value of 2.0 m$\Omega$ is assigned to $R_{int,cell}$ in the model. For extra verification of this value the discharge curves of the Kokam cell are used to derive $R_{int,cell}$ values, as a function of state-of-charge and discharge current. This derivation is shown in Appendix I, and reveals that 2.0 m$\Omega$ is a good average value for the resistance.

![Fig. 9.5. Battery losses as a function of endurance time.](image)

With the cell parameters and motor characteristics updated, the heat losses of the battery can be computed with the model from part I. The motors have been restricted to deliver 40kW maximally, which results in a final battery SOE of 20%. The heat loss trajectory (black) is depicted in Fig. 9.5 and furthermore, the average value of the heat loss is shown (blue). The average heat loss, predicted by the model, is 1070 W. The total (active) cell volume is determined from the 3D battery model and is $78 \times 3.2 \times 10^{-4} \text{ m}^3 = 0.025 \text{ m}^3$.

From these values $Q$ can be derived and is:

$$Q = \frac{P_{\text{batt,loss,avg}}}{V_{\text{cell,total}}} = 42.8 \left[\frac{\text{kW}}{\text{m}^3}\right]$$

9.5.2 Material specific heat capacities

The thermal system mainly consists of 3 materials, namely the cells, aluminium and thermal padding. The applied thermal compound is present in very small quantities so this is neglected. The specific heat for the used aluminium is given in Table 8.1 and is 900 J/(kg·K). Furthermore, the $C_p$ value of the thermal padding is listed in [ds13] and is 711 J/(kg·K). A complication arises when considering the specific heat of the Kokam cells. The cells consist of a combination of materials, such as copper, aluminium, hydrocarbon electrolyte and polymer separator foils. In most studies of battery thermal properties, the overall specific heat of a Lithium-ion cell is determined either experimentally or through an analysis of the cell ingredients, taking mass percentages into account. However, the presented values
[28,30,32,35] differ significantly and range from ± 800 to 1350 J/(kg·K). Due to these differences it is chosen to determine the overall specific heat of the Kokam cells, by means of an experiment. This will be discussed next.

### C. Experiment

For determination of the cell specific heat an experimental setup is composed, that makes use of simple measuring techniques. The following devices are needed for the experiment:

- Cell chamber, consisting of isolation material
- Thermocouple (accuracy 0.1°C)
- Scale (accuracy 0.1g)

In Fig. 9.6 a picture is shown of the experimental setup. The basic idea behind the experiment is that when water of approximately 50°C is poured into the cell chamber (room temperature), the water/cell combination will quickly\(^a\) reach an equilibrium temperature. From the final temperature the specific heat of the cell can be derived. The following parameters must be measured during the experiment:

- Initial cell and water temperature
- Inserted water mass
- Final water temperature

The equation used to calculate the overall specific heat of the cell is:

\[
C_{p,\text{cell}} = \frac{(T_{\text{initial,water}} - T_{\text{final}}) m_{\text{water}} C_{p,\text{water}}}{m_{\text{cell}} (T_{\text{final}} - T_{\text{initial,cell}})}. \tag{9.5}
\]

During the experiment it turned out that small correction had to be applied to the initial water temperature. When pouring the water in the cell chamber the temperature decreases by approximately 0.5°C and it is concluded that this is the result of intense contact between water and air when pouring.

\(^a\) Literature research shows that the thermal conductivity of Li-ion cells is fairly high due to the presence of aluminium and copper foils. This means that the heat exchange between the water and the cells is very dominant in the experiment, and that little heat is absorbed by the cell chamber material, which has a low density and poor conductivity.
Experiment verification:
In order to check the validity of the experiment, a slab of aluminium (same mass as cell) is tested in the same way as the cell. The calculated value of \( C_p \) can then be compared with the literature value of aluminium alloy. The experiment is repeated 3 times for the validation and the results are:

Calculated \( C_p \) values of aluminium slab:

\[
\begin{align*}
\text{Test1:} & \quad 934.4 \text{ J/(kg·K)} \\
\text{Test2:} & \quad 978.2 \quad ,
\text{Test3:} & \quad 923.0 \quad ,
\end{align*}
\]

Average: \( C_p = 945 \text{ J/(kg·K)} \)

The average value has an error of about +5% when compared to the literature value. This is acceptable for the purpose of the experiment and it is decided to round the measured \( C_p \) value of the cells down by about 5% as well, to a value of 100 J/(kg·K) accurately. (i.e. 800, 1200 etc.)

Calculated \( C_p \) values of the Kokam cell:

\[
\begin{align*}
\text{Test1:} & \quad 1098.4 \text{ J/(kg·K)} \\
\text{Test2:} & \quad 1167.5 \quad ,
\text{Test3:} & \quad 1185.0 \quad ,
\text{Test4:} & \quad 1140.6 \quad ,
\text{Test5:} & \quad 1202.1 \quad ,
\text{Test6:} & \quad 1147.8 \quad ,
\end{align*}
\]

Average: \( C_p = 1157 \text{ J/(kg·K)} \)

On basis of these results, a \( C_p \) value of \textbf{1100 J/(kg·K)} is assigned to the Kokam cell material in the 2D FEA analysis. A full overview of the measurements is presented in “Cp_test.xls” in the digital appendix.

9.5.3 Material thermal conductivities

The thermal conductivities for both aluminium and thermal compound paste are already stated in Section 8.3.2. The thermal properties for these are isotropic, but this is not the case for the cells and the cell padding. From the available literature it can be concluded that the conductivity values perpendicular to the thickness of the material is significantly lower than the in-plane values (anisotropic behavior). The reason for this is the layered build-up of both battery components. From [ds13,19] the thermal conductivities are taken. Table 9.1 shows the respective values of the cell and padding material.

<table>
<thead>
<tr>
<th>Material</th>
<th>( k_x )</th>
<th>( k_y )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal padding</td>
<td>10</td>
<td>150</td>
</tr>
<tr>
<td>Kokam cell</td>
<td>2</td>
<td>40</td>
</tr>
</tbody>
</table>

9.5.4 Convective heat transfer coefficient

The battery is cooled by the flow of air along the heat sinks and this flow is generated by the movement of the car. Hence, this cooling mechanism can be categorized as heat transfer by forced air convection. The amount of heat that is removed at the finned heat sink is dependant
on the air flow, the shape of the heat sink and the temperature difference between the air and
the heat sink. To compute the rate of cooling, the heat transfer model makes use of a
convective heat transfer coefficient $h_{\text{conv}}$, as was shown in (9.3). A common method for
estimating the value of $h_{\text{conv}}$ in a turbulent flow is the Dittus-Boelter correlation [81], which
states that:

$$ h_{\text{conv}} = \frac{k_{\text{air}}}{D_H} N_u. \quad (9.5) $$

All parameter descriptions are given in Table 9.2.

<table>
<thead>
<tr>
<th>Table 9.2. Overview of forced air convection parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
</tr>
<tr>
<td>Density</td>
</tr>
<tr>
<td>Heat capacity</td>
</tr>
<tr>
<td>Thermal conductivity</td>
</tr>
<tr>
<td>Kinematic viscosity</td>
</tr>
<tr>
<td>Thermal diffusivity</td>
</tr>
<tr>
<td>Fluid viscosity</td>
</tr>
<tr>
<td>Nusselt Number</td>
</tr>
<tr>
<td>Prandtl number</td>
</tr>
<tr>
<td>Reynolds number</td>
</tr>
<tr>
<td>Coefficient</td>
</tr>
<tr>
<td>Fin Height</td>
</tr>
<tr>
<td>Fin-to-fin distance</td>
</tr>
<tr>
<td>Hydraulic diameter</td>
</tr>
<tr>
<td>Cross-sectional area of flow</td>
</tr>
</tbody>
</table>

The hydraulic diameter $D_H$ in (9.5) is given by:

$$ D_H = \frac{4HW}{(2H + W)}, \quad (9.6) $$

where $H$ and $W$ are depicted in Fig. 9.7. The heat sink has 8 mm high fins that are spaced
4mm. The height is constrained by the available space, and the width results from a
compromise between the heat sink cooling capabilities and the CNC-manufacturing time.

![Fig. 9.7. Fin dimensions](image)

The Nusselt number is derived by:

$$ Nu = 0.023 \text{Re}^{0.8} \text{Pr}^n, \quad (9.7) $$

where the Reynolds and the Prandtl number are determined with:

$$ \text{Re} = \frac{mD_H}{\mu A}, \quad (9.8) $$

and

$$ \text{Pr} = \frac{C_p \mu}{k_{\text{air}}}, \quad (9.9) $$

respectively. The coefficient $n$ in (9.7) has a value of 0.4 for a heating fluid. The accuracy of
the Dittus-Boelter correlation is anticipated to be ±15%. 

85
In Fig. 9.8 the convective heat transfer coefficient of the heat sink is depicted as a function of air velocity. Since the Dittus-Boelter correlation predicts a fairly linear relationship between $h_{\text{conv}}$ and the air velocity, it can be concluded that using the average velocity of the vehicle for determination of $h_{\text{conv}}$ is a safe assumption. Therefore, this velocity is retrieved from the model simulation in Section 9.5.1. and has a value of 53.8 km/h. However, the effect of the vehicle velocity on true air velocity along the heat sink is unclear, due to the lack of aerodynamic analysis of the packaging concept. It is assumed that due to aerodynamic deviations the real value of the air velocity is somewhat lower: 40 km/h. The resulting convective heat transfer coefficient that will be used in the thermal model is $60 \text{ W/(m}^2\text{ K)}$.

### 9.6 Simulation results and discussion

In this section the results of the heat transfer analysis are presented, with used simulation parameters as listed in the previous sections and Table 9.3. As a worst-case scenario the endurance is presumed to take place at a hot summer day. The allowed temperature limits are stated in Table 9.3 as well. First of all, the temperature distribution within the analysis domain at $t = 1500$ sec. is shown in Fig. 9.9. The maximum temperature is located in the middle cell, at the opposite side of the cooled battery case. This is as expected.

![Fig. 9.9. Temperature distribution within analysis domain [°C].](image)

The largest ‘in-cell’ temperature difference is approximately 4.5°C. In Fig. 9.10 the temperature gradient is shown, again at $t = 1500$ sec. The highest value occurs at the transition from cells to conduction plate, at the right side. This reveals that a large part of the heat transport to the heat sink takes place within the cells themselves, due to the high in-plane conductivity value of the cells ($k_x$).

![Fig. 9.10. Temperature gradient within analysis domain [°C/m].](image)
In Fig. 9.11 the temperature trajectory of the $T_{max}$ spot is shown as a function of time. The temperature rises from 30 to 50°C (point A) during use of the battery, and after the endurance the same amount of cooling is applied, which results in an additional waiting time of at least 30mins. to recharge (B).

![Figure 9.11](image.png)

**Fig. 9.11.** Temperature trajectory of $T_{max}$ spot during and after endurance

The results presented above indicate that the thermal mass of the battery has a large impact on the final temperatures. The cooling system works slower than expected, but nevertheless no temperature limits are exceeded in this thermal analysis. It is recommended to always start an endurance with a cool battery (<30°C). Furthermore, rapid cooling of the battery after an endurance or in between test runs can be achieved by applying ice on the heat sinks.
Chapter 10

Closing

Inspired by the decision of University Racing Eindhoven to take part in the class1A Formula Student competition, this report presents an extensive study on the development of an energy efficient high performance drive train. The conclusions of this study are presented in this chapter, and recommendations for future research and development are given.

10.1 Conclusions

The report is split up into three parts, and hence conclusions will be given accordingly:

Part I

Three alternative drive train topologies have been evaluated as possible candidates for the new race car and are:

- Full electric
- Parallel hybrid
- Series hybrid

Furthermore, a standard drive train based on a petrol-powered internal combustion engine, is used as a benchmark. For this purpose a method is developed that compares the drive trains, by calculation of both lap time and energy consumption. Because of the absence of a prescribed driving cycle, the method incorporates computation of the velocity profile on a predetermined track, by describing the race car as a bicycle model and calculating the vehicle dynamics numerically. The energy usage is determined through evaluation of drive train component efficiencies. By varying the maximum available drive power, an overview is created of the capabilities of each drive train, which allows the user to compare the candidates of interest effectively.

In this research, the full electric and parallel hybrid drive train scored best, with estimated fuel savings in the order of 20 – 40%, in comparison with the benchmark ICE drive train. On basis of the evaluation results and practical arguments the decision is made to develop a rear-wheel-driven electric race car.

Besides being used as a tool for the comparison of the drive trains stated above, the modeling method can also be used to evaluate for instance:

- different component technologies
- energy saving methods and strategies

Furthermore optimization studies can be done for different circuits, as well simulating acceleration times. These features make the proposed method not only helpful for use within the URE team, but also for other parties that conduct research on performance vehicles, where energy consumption plays a role.
Part II
The chosen drive train concept is translated in Part II to a complete vehicle concept, where the URE05 race car chassis serves as a starting point. Firstly, important design specification have been defined, and are:

- vehicle mass: $< 260\text{kg}$
- load distribution: 47/53% (front/rear)
- top speed: $100\text{km/h}$
- acceleration 0-75m: $< 4.5\text{ seconds}$.

From these specifications and the competition rules, suitable components have been searched for, that are able to realize the desired concept. The selected main components that form the electric drive train are:

- 2 independant 35 kW PMDC drive motors
- 2 matching controllers, with torque control capabilities
- A 9kWh battery package, consisting of 78 Lithium-ion polymer cells, with a pouch format.

With these components, a packaging lay-out has been designed, in which all components are located in the steel tube rear frame. The result is a compact and lightweight drive train that weighs 140 kg, including rear frame modifications. The new race car is expected to have a mass of approximately 250 kg and is called the URE05e. Furthermore, calculations show that the final weight distribution is very close ($\pm 2\%$) to the design target.

Part III
Finally, a battery is designed and thermally analysed in detail, where again the model of Part I served as a tool for parameter extraction. The main guidelines for design were the competition rules and selected design targets. The battery design is characterized by an aluminium containment, in combination with polycarbonate insulation parts and vital electronics. Cooling is taken care of by internal conduction through aluminium plates and external removal of heat by air cooled heatsinks. It is expected that the battery will have a mass of 85kg.

Subsequently, the battery design has been analyzed thermally, with the intent of determining the highest temperature during the endurance. For this purpose a two-dimensional heat transfer model has been composed, that is able to compute the battery design’s time-dependant thermal behavior. It can be concluded that the battery will stay within the allowed temperature limits (60°C), although the cooling system was anticipated to be more effective than the simulation results predicted. The expected maximum temperature rise in the battery is approximately 20°C.

10.2. Recommendations

Some open issues and other suggestions for improvements are the following.

Part I
The model can be adapted in several ways to produce more accurate results. These are:

- Include longitudinal wheel slip, which results in an additional loss factor.
- The bicycle model may be replaced by a two-track model (i.e. 4 wheels). This will enable to incorporate lateral load transfer, which has a reducing effect on the cornering velocity.
• Addition of corners with a variable curvature; calculate tyre forces with combined slip conditions at each step. This results in a more realistic velocity profile and drive train operation points.
• The ICE internal losses during braking are not taken into account. This can be added and will have a reducing effect on the amount of regenerative braking for the parallel hybrid topology.
• The HV electrical wiring also introduces drive train losses, but are not modeled in this report. It is advised to perform research on a suitable method to implement this.
• The modeling in Part I uses an empty URE05 mass, that was estimated too high. A more accurate determination is necessary before starting to simulate results.
• The drive train inertia’s have not been modeled. It is recommended to implement this, either by an additional vehicle mass factor, or direct calculation of the induced inertia torque.

In general, a method for model validation must be searched for. With this validation, aspects that absolutely must be taken into account, can be identified. Meanwhile, other aspects may be excluded. In my opinion, the goal is to have an uncomplicated model, that is still able to provide the results of interest accurately.

Part II
• A difficult issue during the design stage was the following; the URE05e makes use of a rear frame that was originally intended for an internal combustion engine. Furthermore, this particular rear frame design was not fully matched with the rear suspension design, which originated from the URE04. These circumstances have lead to some design compromises. It is therefore recommended to start an extensive research on a new design concept, that fully focusses on an electric drive train. This concept can be applied to successors of the URE05e. For instance, an improvement can be a battery suspension, that allows for quick replacement of the battery. All-Wheel-Drive is also a very interesting option to investigate.
• It appeared that an enhanced version of the LEM200 motor is available; the AGNI 95R. Although this motor has the same maximum power output, it employs a maximum speed of 6000 RPM, which allows for a higher continuous current, due to increased air cooling.
• An advantage of drive trains is the ability to test them in advance on a test bench, without having to wait for a fully finished vehicle. It is therefore advised to further optimize the test bench at the EPE lab, by incorporating a high current supply and a fully automated operation of the setup.

Part III
• The cooling of the current battery design can be enhanced by applying air ventilation in the busbar area. However, waterproofness must be guaranteed.
• It is recommended to build a small test setup for battery cells, where experiments on the thermal and electrical behavior can be conducted. Simulation of an endurance in a cell test setup is an interesting option in this regard.
• For a new chassis design more space must be reserved for the battery. This allows for internal air cooling, which is expected to be the most effective (and still lightweight) cooling method.
• To reduce the battery weight, the following materials are advised to utilize in the next design:
  o Carbon Fibre Reinforced Plastic (CFRP) containment, with flame-retardant resin.
  o Aluminium busbar material, using safe and robust interconnection methods.
Appendix A

Formula Student class 1A competition scoring and main rules

**Static** 325
- Sustainability 100  Embedded energy and CO$_2$ in the production of the vehicle
- Presentation 75  A presentation of the business plan
- Design 150  Evaluation of the engineering effort that went into the car’s design

**Dynamic** 675
- Acceleration 75  Evaluation of the car’s acceleration capability on a straight line (75m)
- Skid pad 50  Evaluation of the car’s cornering ability (constant radius turn, R=8m)
- Autocross 150  Evaluation of the car’s maneuverability and handling qualities. A single lap of various straight’s, turns and slaloms.
- Endurance 200  Test of overall performance and durability. 25 laps, total 22km. Driver change at the middle.
- Fuel economy 200  After the endurance the energy consumption is measured and converted to a figure representing CO$_2$ released into the atmosphere.

**Total** 1000 pts.

*Allowed power sources:*

<table>
<thead>
<tr>
<th>Source</th>
<th>Max displ.</th>
<th>Max. Ø inlet restrictor</th>
<th>Equivalent CO$_2$ prod.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Petrol</td>
<td>250cc</td>
<td>12.9mm</td>
<td>2.30 kg/litre</td>
</tr>
<tr>
<td>Diesel</td>
<td>310cc</td>
<td>no limitation</td>
<td>2.63 kg/litre</td>
</tr>
<tr>
<td>E85</td>
<td>250cc</td>
<td>12.3mm</td>
<td>1.64 kg/litre</td>
</tr>
<tr>
<td>LPG</td>
<td>250cc</td>
<td>13.4mm</td>
<td>1.60 kg/litre</td>
</tr>
<tr>
<td>CNG</td>
<td>250cc</td>
<td>13.8mm</td>
<td>2.90 kg/kg</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>250cc</td>
<td>no limitation</td>
<td>7.90 kg/kg</td>
</tr>
<tr>
<td>Hydrogen fuel cell</td>
<td>-</td>
<td>-</td>
<td>7.90 kg/kg</td>
</tr>
<tr>
<td>Electric</td>
<td>-</td>
<td>-</td>
<td>0.54 kg/kWh</td>
</tr>
</tbody>
</table>

All ICE’s must be four-stroke type.
Max. electrical power is restricted to 75 kW.
Combination of any of the above is allowed (Hybrid).
Kinetic flywheels are not allowed.
Appendix B

How does a Formula SAE car drive on an endurance circuit?

Formula student endurance tracks typically have a length of about 1 km, and are made up of a number of turns, straights and slaloms. Formula SAE has a standard specification on the size and number of the turns, and also on the length of the straights. By providing these lengths, the maximum and average velocity that the cars can attain is limited to around 105 and 55 km/h respectively. This is fairly moderate compared to most other classes of motorsports, and is chosen from a safety point of view. (The students are also the drivers and are assumed to be non-professionals)

The track is outlined with rows of cones on either sides of the track, which is about 3m wide. Hitting a cone will result in a time penalty. This track setup requires the cars to have excellent handling and mechanical grip, in order to achieve good results.

![Sample of logged wheelspeed data from URE04 during endurance.](image)

In Fig. B.1 a sample of logged velocity data from the URE04 is shown. By looking at the profile one can determine quite accurately what the car is doing. The green lines indicate the points where the driver applies full throttle on a straight, after exiting a corner. At a certain moment the driver needs to brake as hard as he can, in order to enter the next corner with the appropriate speed. The start of braking is indicated by the red lines.

Then the driver enters the corner (gray line) and completes it with fairly constant speed.

Concluding, there are 3 phases to be identified, which are “cycled” a number of times per lap, depending on the number of turns:

1) Acceleration (full-throttle)
2) Braking (maximally)
3) Cornering (approximated as steady-state)

In Fig. B.1 the observed driving behavior is shown idealised by placing the blue lines over the real logging data.
Appendix C

Calculation of acceleration, braking and cornering forces

C.1 Acceleration

First the front wheel load is described as a function of the rear wheel load by rewriting:

\[ F_z = mg = F_{z1} + F_{z2}, \quad (C.1.1) \]

to:

\[ F_{z1} = F_z - F_{z2} = (mg - F_{z2}). \quad (C.1.2) \]

The determination of acceleration forces is based on considering the summation of moments around the COG, which must be equal to zero:

\[ \Sigma M_{COG} = 0 \quad (C.1.3) \]

Taking all forces that apply a moment around the COG results in:

\[ F_{z2} L_2 - F_{z1} L_1 - F_{z2} h_{COG} = 0 \quad (C.1.4) \]

Substituting (C.1.2) in (C.1.4):

\[ F_{z2} L_2 - (mg - F_{z2}) L_1 - F_{z2} h_{COG} = 0 \quad (C.1.5) \]

Rewriting (C.1.5) and inserting (3.15) yields:

\[ F_{z2} L_2 - L_1 mg + L_1 F_{z2} - c_1 h_{COG} F_{z2} + c_2 h_{COG} F_{z2}^2 = 0 \quad (C.1.6) \]

This equation can be rearranged into the quadratic function:

\[ (c_2 h_{COG}) F_{z2}^2 + (L_1 + L_2 - c_1 h_{COG}) F_{z2} + (-L_1 mg) = 0, \quad (C.1.7) \]

A   B   C

\[
F_{z2} = \frac{-(L_1 + L_2 - c_1 h_{COG}) + \sqrt{(L_1 + L_2 - c_1 h_{COG})^2 - 4(c_2 h_{COG})(-L_1 mg)}}{2(c_2 h_{COG})} \quad (C.1.8)
\]
C.2 Braking

The determination of the braking forces is similar to the procedure of acceleration forces:

\[ \sum M_{\text{COG}} = 0 \]  \hfill (C.2.3)

Taking all forces that contribute to (C.2.3):

\[ F_{z2} L_2 - F_{z1} L_1 - F_{z1} h_{\text{COG}} - F_{z2} h_{\text{COG}} = 0. \]  \hfill (C.2.4)

Substituting \( F_{z1} \) for all \( F_{z2} \) and inserting (3.17) and (3.18) yields:

\[ (m_g - F_{z1}) L_2 - F_{z1} L_1 - (c_1 F_{z1} - c_2 F_{z1}^2) h_{\text{COG}} - (c_1 F_{z2} - c_2 F_{z2}^2) h_{\text{COG}} = 0 \]  \hfill (C.2.5)

and:

\[ (m_g - F_{z1}) L_2 - F_{z1} L_1 - (c_1 F_{z1} - c_2 F_{z1}^2) h_{\text{COG}} - (c_1 (m_g - F_{z1}) - c_2 (m_g - F_{z1})^2) h_{\text{COG}} = 0 \]  \hfill (C.2.6)

Rewriting (C.2.6) to a quadratic function:

\[ \left( -2 c_2 h_{\text{COG}} \right) F_{z1}^2 + \left( 2 c_2 h_{\text{COG}} m_g - L_1 - L_2 \right) F_{z1} + \left( c_1 h_{\text{COG}} m_g + m_g L_2 - c_2 h_{\text{COG}} m_g^2 \right) = 0 \]  \hfill (C.2.7)

A \quad B \quad C

Which can be solved by:

\[ F_{z1} = \frac{-2 c_2 h_{\text{COG}} m_g - L_1 - L_2 + \sqrt{(2 c_2 h_{\text{COG}} m_g - L_1 - L_2)^2 - 4(-2 c_2 h_{\text{COG}})(c_1 h_{\text{COG}} m_g + m_g L_2 - c_2 h_{\text{COG}} m_g^2)}}{2(-2 c_2 h_{\text{COG}})} \]  \hfill (C.2.8)

Once \( F_{z1} \) is calculated, the other forces can be derived as well.
C.3 Cornering

Determination of the cornering velocity is based on the following relationships:

\[ F_{z1} = \left( \frac{L_1}{L} \right) mg, \quad F_{z2} = \left( \frac{L_2}{L} \right) mg, \]  
\[ (C.3.1) \]

and the equation for centrifugal force:

\[ F_y = \frac{mv^2}{R}. \]  
\[ (C.3.2) \]

Both lateral tyre forces support the centrifugal force:

\[ F_{y1} + F_{y2} = \frac{mv^2}{R}, \]  
\[ (C.3.3) \]

where \( F_{y1} \) and \( F_{y2} \) can be rewritten in terms of wheel loads by:

\[ \mu_1 F_{y1} + \mu_2 F_{y2} = m\nu_{max}^2 R^{-1}. \]  
\[ (C.3.4) \]

Substitution of (C.3.1) into (C.3.2) yields:

\[ \mu_1 \left( \frac{L_1}{L} \right) mg + \mu_2 \left( \frac{L_2}{L} \right) mg = m\nu_{max}^2 R^{-1}. \]  
\[ (C.3.5) \]

Rearranging for \( \nu_{max} \):

\[ \sqrt{R \left( \mu_1 \left( \frac{L_1}{L} \right) g + \mu_2 \left( \frac{L_2}{L} \right) g \right)} = \nu_{max}. \]  
\[ (C.3.6) \]

Combination of (C.3.6) and (3.14) yields:

\[ \nu_{max} = \sqrt{R \left( \left( c_3 - c_4 F_{z1} \right) \left( \frac{L_1}{L} \right) g + \left( c_3 - c_4 F_{z2} \right) \left( \frac{L_2}{L} \right) g \right)}. \]  
\[ (C.3.7) \]

This can be rewritten to:

\[ \nu_{max} = \sqrt{R \left( \left( c_3 \frac{L_1 g}{L} - c_4 \frac{m_1 L_1^2 g}{L^2} \right) + \left( c_3 \frac{L_2 g}{L} - c_4 \frac{m_2 L_2^2 g}{L^2} \right) \right)}. \]  
\[ (C.3.8) \]
Appendix D

Comparison of Li-Ion and ultracapacitors for use in a hybrid race car.

An improvement in energy efficiency is expected to come from the recuperation of kinetic energy of the race car. Especially for the hybrid topologies, the electrical system will be used as a short-term energy storage, used for this purpose. However, the extra weight that this recuperation system adds to the total mass of the car, needs to be kept to a minimum. A brief examination is therefore done, where two technologies are compared, namely a storage system consisting of a Li-Ion battery and one of ultracapacitors.

From the power profile presented in Fig. 3.10 the average brake power at the rear wheels is determined. This is 21.3 kW. Taking an average efficiency of 90% for the EM generator mode, the power applied to the short-term energy storage is about 19.2 kW. The maximum braking duration is 1.65 seconds. Therefore the maximum energy send to the storage during a braking event is about 32 kJ. In Table D.1 these values are used to estimate the minimum mass of the two types of energy storage, in order to recuperate a substantial amount of kinetic energy. The values for average specific energy and power are taken from [7,i7].

<table>
<thead>
<tr>
<th>Specific energy [kJ/kg]</th>
<th>Li-Ion</th>
<th>Ultracapacitor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific power [kW/kg]</td>
<td>0.6</td>
<td>2.5**</td>
</tr>
</tbody>
</table>

From Table D.1 it can be seen that for both storage systems the specific power is the most critical parameter for regenerating kinetic energy of the vehicle. The ultracapacitor system however has a minimum mass that is more than 4 times lower than the Li-Ion system. Ultracapacitors are therefore chosen as the best option for the hybrid race car topologies.

Other ultracapacitor advantages:

- Lifespan > 500,000 cycles
- Cheaper than Li-Ion batteries.
- Management system less complex than one for Li-Ion batteries

*) It is assumed that only the upper half of the ultracapacitor voltage range is used, thereby covering 75% of the total energy content.

**) Higher values are possible, but 2.5 kW/kg ensures efficiencies above 90%.

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Appendix E

E.1 Motor & Controller specifications

Motor

Brand: The Lynch Motor Company
Version: 2 x LEM200 D135RAG
Type: Brushed PMDC pancake motor
Max. speed: 4032 RPM
Max. voltage: 96 V
Max/Cont. power: 35 / 17 kW
Max/Cont. torque: 80 / 40 Nm
Max/Cont. current: 400 / 200 A
Max/Avg. efficiency: 91 / 85 %
Cooling: Air
Mass: 10.6 kg
Torque constant: 0.207 Nm/A
Speed constant: 42 RPM/V
No load current: 7.36 A
Armature resistance: 16 mΩ
Armature inductance: 15 mH
Rotor inertia: 0.0236 kgm²
Price: € 1100

Motor controller

Brand: Kelly Controls, LLC.
Type: 2 x KDH14651B PMDC
Nominal voltage range: 24 – 144 V
Max. operating voltage: 180 V
Switching frequency: 16.6 kHz
Max. current (10s.): 650 A
Continuous current: 260 A
Regenerative br.: Yes
Control input: 0-5 V Analog Brake & Throttle
Mass: 3.3 kg
Price: € 1000

Fig. E.1. Picture of LEM200 motor.
Fig. E.2. Picture of Kelly controller
### E.2 Motor test rig

A motor test rig has been designed and built in the EPE lab at the TU/e campus. In Fig. E.2.1 the 3D design is depicted. The LEM200 motor is mounted on a construction, that consists of welded and bolted 8mm steel plates. The motor is connected via a flexible coupling (red) to a large generator (green). The finished test rig can be seen in Fig. E.2.2. The generator is able to provide a load for the motor, and the produced electricity is fed back into the power net. At first, the standard power supply of the EPE lab is used for conducting experiments, but this supply is limited to 100A at 100V (= 10 kW). Once the battery is finished, it will be tested on this test rig, which will enable higher powers to be applied to the motor.

The test rig is equipped with several sensors:

- Voltage at the motor
- Voltage at the controller input
- Current at the motor
- Current at the controller input
- Temperature (2x) of the motor terminals
- Shaft speed (laser sensor)
- Generator torque
- Thermal camera (IR)

All data can be viewed and logged on a computer. For now the test rig is operated by hand, which gives the opportunity to measure steady-state efficiency for instance. Also the temperature behavior of the motor and controller can be examined, with the temperature equipment. An example of this is given in Fig. E.2.3, where the infra red camera shows the temperature of the whole test rig. The motor housing is approximately 50°C in this picture. By the time of writing this report, the test rig is being converted to a full dynamic test bench, where load and drive parameters are controlled by a computer. The purpose of this setup is to simulate a full endurance with the motor and to check what the thermal limits are.
Appendix F

Chain load cases

In this appendix three load cases are calculated, in order to estimate the maximum peak load that may occur in the final drive. These load cases are:

1. Maximum torque applied by the motor (overloaded to 500 A current)
2. Spinning wheel hitting the ground
3. Locking a wheel (at full speed) with the hydraulic brakes in 0.1 sec.

The load case that results in the highest chain load will be normative for the chain type selection. Dynamic effects that occur from component flexibility and clearance are neglected in this examination. Fig. F.1 shows a schematic overview of the chain reduction.

1. The chain tension force $F_{\text{chain}}$ at a motor current $I_{\text{EM}}$ of 500A is:

$$F_{\text{chain}} = \frac{I_{\text{EM}} \cdot K_t}{r_{10}} = \frac{500 \cdot 0.207}{0.02} \approx 5175 \, N,$$

where $K_t$ is motor torque constant and $r_{10}$ the radius of a 10teeth sprocket.

2. In this load case, the wheel is assumed to hit the ground (after being lifted) with a different roll speed, so wheel slip will occur. The load on the wheel is assumed to be 150kg. The resulting chain force is determined by:

$$F_{\text{chain}} = \frac{T_{\text{slip}}}{r_{44}} = \frac{150 \cdot g \cdot \mu_s \cdot r_w}{r_{44}} = \frac{588.6}{0.089} \approx 6600 \, N,$$

where $T_{\text{slip}}$ is the wheel torque resulting from the friction with the ground, $\mu_s$ is the tyre friction coefficient, $r_w$ the wheel radius and $r_{44}$ the radius of the large sprocket.

3. By locking the wheel with the brakes at full speed, the chain is subjected to the inertial load of the decelerating motor. It is assumed that the brakes lock the wheel in 0.1 seconds. The deceleration (absolute) value is given by:

$$\dot{\omega} = \frac{\omega_{\text{EM, max}}}{t_{\text{lock}}} = \frac{419}{0.1} = 4190 \, \text{rad} / \text{s}^2.$$
The tension force in the chain is determined by:

\[
F_{\text{chain}} = \frac{T}{r_{10}} = \frac{J_{EM} \cdot \dot{\omega}}{r_{10}} = \frac{0.0236 \cdot 4190}{0.02} \approx 4950N,
\]

where \(J_{EM}\) is the motor inertia.

The highest chain force in this examination is calculated at the second load case and is approximately 6600N. In Table F.1 three different chain types from manufacturer Regina are listed.

<table>
<thead>
<tr>
<th>Chain type</th>
<th>Tensile strength [kN]</th>
</tr>
</thead>
<tbody>
<tr>
<td>420</td>
<td>15.7</td>
</tr>
<tr>
<td>428</td>
<td>18.8</td>
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<tr>
<td>520</td>
<td>27.0</td>
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</table>

From the tensile strength figures it appears that a 420-type chain is sufficient for application in the final drive. This chain type however is not very common, and the number of manufacturers that sell sprockets for this chain size is limited. Therefore the more common 428-type chain is chosen for the final drive. This results in an estimated chain strength reserve of about 12 kN.
Appendix G

Determination of weight distribution of URE05 without drive train

<table>
<thead>
<tr>
<th>m [kg]</th>
<th>$x_{\text{COG}}$ [mm]</th>
<th>Description</th>
</tr>
</thead>
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<tr>
<td>m1</td>
<td>17.55</td>
<td>1364.3</td>
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<td>m2</td>
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<td>m32</td>
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<td>637</td>
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</table>

| Wheelbase | L                | 1601.1 |
| Empty mass | $m_{\text{empty}}$ | 109.8  |
| Total mass  | $m$              | 181.6  |
| Front load | $m_1$            | 111.6  |
| Rear load  | $m_2$            | 70.0   |
| Weight Distribution | Front | 61.5 | % |
|             | Rear        | 38.5  | % |
| COG Location | $x_{\text{COG}}$ | 984    | mm |
## Appendix H

### Bill of Material - Battery

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<thead>
<tr>
<th>Part description</th>
<th>Part code</th>
<th>Prod./Purchase</th>
<th>Producer/Supplier</th>
<th>Quantity</th>
<th>Material</th>
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Nuts and bolts are excluded from this bill-of-material.
Appendix I

Calculation of the Kokam internal cell resistance

The internal cell resistance can be retrieved from the voltage-drop that occurs after applying a load to a cell. This is explained with the help of Fig. I.1 [36]. After a small initial load, in this case 5mA, a heavy drain load is applied of 505mA. Firstly, the voltage of the cell exhibits an instantaneous decrease, which is associated with the pure electric resistance of the cell. After that, the cell voltage drops even further, and this decrease is attributed to the ionic cell resistance, resulting from various electrochemical factors. The total voltage drop is consequently given by:

\[
\Delta V = \Delta V_{\text{electrical}} + \Delta V_{\text{ionic}}
\]  

(I.1)

The total internal cell resistance at the conditions of Fig. I.1 is calculated by:

\[
R_{\text{int.cell}} = \frac{\Delta V}{\Delta A} = \frac{1.485 - 1.38}{0.505 - 0.005} = \frac{0.105}{0.5} = 0.21 \Omega
\]

(I.2)

![Fig. I.1. Example of cell resistance determination, by means of voltage drop measurement.](image)

In [ds20] the discharge curves of the Kokam high power cells are given as a function of Depth-of-Discharge (D.O.D) and are shown in Fig. I.2.

![Fig. I.2. Discharge curves of the Kokam high power cells, as a function D.O.D.](image)

The temperature is kept constant at 23°C.
This plot allows for a determination of the internal cell resistance, as a function of discharge current and state-of-charge. The relationship between state-of-charge and depth-of-discharge is:

\[ SOC = 1 - D.O.D. \]  

(1.3)

By using (1.2) for all discharge curves in Fig. I.2 and at steps of 5% D.O.D. (taking 0.33C as the initial base load), the relationship depicted in Fig. I.3 is obtained:

![Graph showing cell internal resistance as a function of state-of-charge and discharge current](image)

Fig. I.3. Cell internal resistance as a function of state-of-charge and discharge current

\( R_{int} \) at low currents (<62A) is quite distributed, due to the detail of the discharge curve image, provided by Kokam. However, \( R_{int} \) at high loads is most important, since the main portion of heat is generated at high currents (\( P_{loss} = I^2 \times R_{int} \)). From 93A (1C) and above the values of \( R_{int} \) are all below 2 mΩ, which indicates that this is an acceptable value for determination of the Kokam cell heat generation.
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[ds09] HV Wiring
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[ds11] L.M.C. Datasheet
[ds12] LEM Current sensor
[ds13] E-Graf Thermal pad (HT-1220-AA)
[ds14] Zivan NG9 Battery Charger
[ds15] Aluminium 6082
[ds16] Heat transfer compound
[ds17] Araldite 2031 Adhesive
[ds18] BMS temperature sensor
[ds19] Deutsch HV connector
[ds20] Kokam Catalog
[ds21] GAIA 27Ah cell datasheet
Nomenclature

Acronyms

<table>
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<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tr>
<td>AWD</td>
<td>All Wheel Drive</td>
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<tr>
<td>AC</td>
<td>Alternating Current</td>
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<td>BEV</td>
<td>Battery Electric Vehicle</td>
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<tr>
<td>BMEP</td>
<td>Brake Mean Effective Pressure</td>
</tr>
<tr>
<td>BMS</td>
<td>Battery Management System</td>
</tr>
<tr>
<td>BTMS</td>
<td>Battery Thermal Management System</td>
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<tr>
<td>CFRP</td>
<td>Carbon Fibre Reinforced Plastic</td>
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<td>CI</td>
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<td>Centre of Gravity</td>
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<td>DOHC</td>
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<td>WOT</td>
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## Symbols

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Samenvatting

In de hedendaagse mobiliteit wordt steeds meer nadruk gelegd op het ontwikkelen van energie zuinige voertuigen. Deze trend zet zich inmiddels ook door naar takken van de motorsport en een goed voorbeeld daarvan is de Formula Student competitie. Hier worden race auto’s niet alleen gewaardeerd op hun snelheid, maar tevens op het brandstof verbruik dat daarmee gepaard gaat. Een nieuwe ontwikkeling binnen Formula Student is de oprichting van een speciale klasse, waarbij het doel is om verschillende alternatieve aandrijvingen op het circuit te laten wedijveren met elkaar. De nadruk ligt hierbij vooral op het ontwikkelen van een race auto, die een minimale hoeveelheid energie nodig heeft, maar toch zeer snelle rondetijden kan neerzetten.

University Racing Eindhoven heeft besloten om hieraan deel te nemen en al snel is gebleken dat de vereisten voor een dergelijke wagen tegenstrijdig zijn. De hoofdvraag is (zie) wat voor type aandrijving het beste voldoet aan de gestelde competitie doelen. Ook de uiteindelijke uitvoering van het aandrijvings concept vergt een goede aanpak om succesvol te zijn.

In het eerste deel van dit proefschrift is een methode gepresenteerd, waarmee diverse aandrijf topologieen te vergelijken zijn, in termen van prestaties en zuinigheid. De methode maakt gebruik van een rekenmodel, waarin met behulp van een vereenvoudigd voertuig model ronde tijden bepaald kunnen worden, alsmede het bijbehorende energie verbruik. Vervolgens zijn de volgende aandrijf topologieen met deze methode bestudeerd:

- volledig elektrisch
- serieel hybride
- parallel hybride

Tevens is een conventionele aandrijving met verbrandingsmotor als referentie doorgerekend. De gepresenteerde methode maakt het ook mogelijk om bepaalde aspecten quantitatief te onderzoeken, en parameters te optimaliseren. Als resultaat van simulaties met dit model en andere argumenten is gebleken dat een volledig elektrisch aandrijving een goede kandidaat is om verder uit te werken.

In het tweede deel is de elektrische aandrijving uitgewerkt in een voertuig concept, waarbij de race auto van 2009, de URE05, als uitgangspunt diende. Met behulp van de competitie regels, ontwerp criteria en met gebruik van het rekenmodel zijn voertuig specificaties opgesteld. Aan de hand hiervan zijn geschikte aandrijflijn componenten geselecteerd. Het concept is daarbij uitgekomen op een achterwiel-aangedreven race auto met twee onafhankelijke PMDC motoren, met een piekvermogen van 35 kW ieder. Tevens is er voor een batterij pakket gekozen, bestaande uit 78 hoogwaardige Li-Ion cellen. Alle aandrijflijn componenten bevinden zich in het achter frame.

In het derde deel van dit proefschrift is het ontwerp van de batterij, dat tot in detail is uitgewerkt, gepresenteerd. Verschillende koel technieken zijn bestudeerd, alsmede materialen gezocht die aan de specificaties voldoen. Het gekozen ontwerp is vervolgens uitgebreid toegelicht in woord en beeld. Doel is geweest om het batterij ontwerp zo licht mogelijk te houden, terwijl deze toch functioneel en veilig is. Hiertoe is ook een thermische analyse uitgevoerd, waarmee bepaald is wat de maximaal te verwachten temperatuur is tijdens de ‘endurance’ wedstrijd. Hoewel de berekende temperatuur stijging geen problemen voorspelt, is wel aangetoond dat de batterij eerst voldoende moet worden afgekoeld alvorens hem opnieuw te gebruiken.
Nawoord en dankbetuiging

Dit proefschrift is het hoogtepunt van drie en een half jaar studie, waarvan ik twee jaar heb doorgebracht bij het studenten race team van de Technische Universiteit Eindhoven. Het is dan ook tot stand gekomen met de hulp van velen om mij heen. Niet alleen heb ik enorm veel kennis en ervaring opgedaan in deze periode, maar ook een uiterst spannende en leuke tijd gehad samen met alle team leden en andere mensen om mij heen. Het werk bij URE heeft mij altijd enorm geboeid, mede door de intense gang van zaken, met alle ups and downs van dien. Hoogtepunten waren het behalen van de 1e prijs voor de ontwerp wedstrijd in Engeland en bij Audi in Duitsland. Ook de gezellige avonden, waarbij wij allen aan de auto werkten of juist uitgelaten feest vierden, zal ik nooit vergeten. Diepepunten zijn er ook geweest; het verliezen van team lid en vriend Sven was een zware klap voor mij. Maar ook het uitvallen van de auto tijdens een endurance is een onthutsende gebeurtenis waarbij de moed diep in de schoenen wegzakt. Ondanks de moeilijke momenten kijk ik met een zeer positief gevoel terug op mijn tijd bij de faculteit in eindhoven. Ik hoop dan ook velen die ik heb leren kennen later nog tegen te komen.

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