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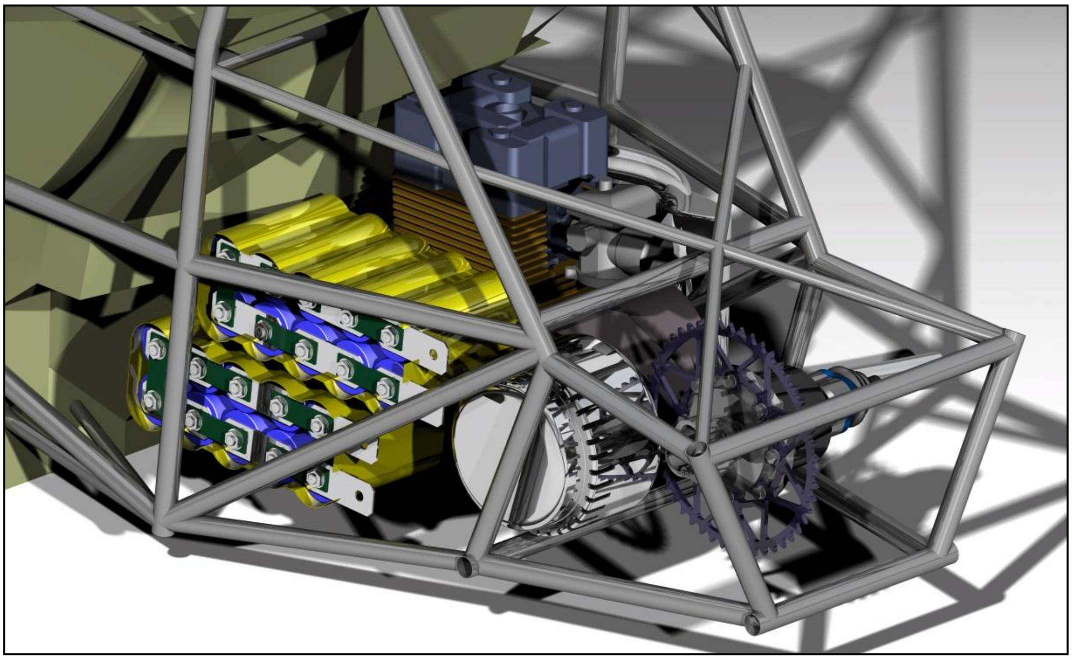

Alternative powertrain

Hybridization of Formula Student car

Bachelor Thesis in Applied Mechanics

Fredrik Hermansson
Martin Holder
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Håkan Winqvist

Department of Applied Mechanics
Division of Combustion
CHALMERS UNIVERSITY OF TECHNOLOGY
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Fredrik Hermansson	900514-1891
Martin Holder	900118-C370
Carl Petersson	890614-5076
David Stenman	900130-1671
Martin Strängberg	900530-0711
Håkan Winqvist	880603-5930

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FREDRIK HERMANSSON

MARTIN HOLDER (EXCHANGE STUDENT FROM TU DARMSTADT, GERMANY)

CARL PETERSSON

DAVID STENMAN

MARTIN STRÄNGBERG

HÅKAN WINQVIST

Kandidatarbete 2012:04
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Institutionen för Tillämpad mekanik
Avdelningen för Förbränning
Chalmers tekniska högskola
SE-412 96 Göteborg
Sverige
Telefon: + 46 (0)31-772 1000

Front cover:
The picture shows the conceptual packaging of the hybrid system of the Formula Student vehicle.

Tryckeri/Institutionen för Tillämpad mekanik
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Preface

This Bachelor Thesis was written during spring 2012 by six third-year-engineering students from different programs.

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Fredrik Hermansson
Martin Holder
Carl Petersson
David Stenman
Martin Strängberg
Håkan Winqvist

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Abstract

This report investigates the hybridization of a formula student powertrain. It starts by studying different existing solutions and technologies and continues with describing the result of this project. All decisions are driven by data and well-grounded in calculations, experiences in former Chalmers Formula Student (CFS) cars, simulations and the Formula student rules (Rules 2012).

The powertrain will be a parallel hybrid with a single cylinder engine of 600 cc since its weight is much lower than the current four cylinder engine, a Yamaha Fazer. An electric machine with its sprocket on the final drive chain will be used. It will also act as a chain tensioner. The electric energy will be stored in super capacitors because of their high power per weight relation. For this application high power has higher priority than high energy content.

A GT Power model of the engine has been created and delivers data for making assumptions about the predicted power of the vehicle with a single cylinder engine. Regenerative braking will be applied in order to use the braking energy for acceleration when required. All components of the powertrain will be controlled by components by National Instruments and Nira using CAN communication.

The components have been packaged virtually in the CFS-10 frame, since the frame of the actual car where this powertrain is intended to be mounted is not yet built.

Contents

List of figures and tables

Abbreviations

ABS	Anti-lock Braking System
AC	Alternating current
AoW	Angle of Wrap
ATV	All-Terrain Vehicle
BLDC	Brushless Direct Current Machine
BMEP	Brake Mean Effective Pressure
CAD	Computer Aided Design
CAN	Controller Area Network
cc	cubic centimeters [cm^3]
CFS	Chalmers Formula Student
CVT	Continuously Variable Transmission
DC	Direct current
ECU	Engine Control Unit
EMF	Electromotive Force
FIA	Fédération Internationale de l'Automobile
HEV	Hybrid Electric Vehicle
HP	Horsepower (1 [hp] ≈ 746 [W])
ICE	Internal Combustion Engine
IMD	Insultation Monitoring Device
KERS	Kinetic Energy Recovery System
MEP	Mean Effective Pressure
NVH	Noise Vibration and Harshness
OBD	On-Board Diagnostics
PMSM	Permanent Magnetic Synchronous Machine
RMS	Root Mean Square
SI	Spark Ignition
SOC	State of charge
ssp	specific swept volume power ratio
FPGA	Field-programmable gate array

1 Introduction

This report concerns the proceeding of the project of developing an alternative powertrain for future Chalmers Formula Student (CFS) projects. Due to the comprehensiveness of the project only electric hybrid systems have been investigated. The project has been a bachelor project at Chalmers University of Technology and may be seen as a pre-study for the future CFS projects.

1.1 Background

The CFS racing vehicles have always used conventional four-stroke internal combustion engines (ICE's). Research has recently emphasized the importance of reducing the large negative impact on the environment that ICE's have today. There is currently debate about how to solve these problems which has led major car manufacturers to invest heavily in developing more sustainable technologies.

The rules of the Formula Student contest have been changed to promote alternative powertrains and allowing hybrid solutions. This is an important step in the process toward more energy efficient vehicles and a sustainable society. Chalmers has therefore decided to develop this technology as a transitional phase towards a complete electric powertrain.

To optimize the torque and power for different ranges of powertrain speeds (rotations per minute, rpm) without increasing fuel consumption, a hybrid system is considered ideal. This is because the electric motor in general terms has high torque at lower rpm and the combustion engine has high torque at higher rpm. While decelerating, the kinetic energy can be stored as electric energy and used at later stage by the electric motor. This increases the overall efficiency of the system.

1.1.1 Hybrid systems

The word “hybrid” has its origin in the Greek word “Hybrid” and means “mixed” or “in two ways”. A hybrid system is a system which consists of at least two different power systems which provides the same energy. In automotive applications hybrid means that both a traditional combustion engine and, for example, an electric drive provides the power which is needed to drive the car. In general, the electric machine is used as a motor when the car accelerates and as generator when the car decelerates since here the kinetic energy from the car can be recovered and stored as electric energy. Read more about this in 2.7 Regenerative braking. A vehicle with a partly electric powertrain is called Hybrid electric vehicle (HEV).

There are different types of hybrid systems, which are described below and the series and parallel hybrid are illustrated in Appendix A Hybrid systems.

1.1.1.1 Series

In a series hybrid, the combustion engine runs at constant speed in order to drive a generator which loads up an energy storage. This storage is used by an electric machine which drives the car. Series hybrid cars are considered to be fully electric vehicles (Guzzella and Sciarretta 2007). An advantage is that the combustion engine can be operated at the engine speed at which it is the most effective and the specific fuel consumption and the emissions are at their lowest level compared to power output. A disadvantage is that the electric machine must be dimensioned so that it can provide all energy which is needed to drive the car. That means that it will be a large and comparably heavy machine. Due to many energy transformations, from fuel in the combustion engine to mechanical energy and further to electric energy from generator to electric storage and back to mechanical energy in the electric motor there is a risk of low efficiency.

1.1.1.2 Parallel

In this concept, both the combustion engine and the electric machine deliver the power needed to drive the car. Both systems are connected through a clutching system on the powertrain. Advantages are that there is the potential to use a smaller combustion engine, which gives lower fuel consumption and emissions. In addition the working point, the loading case where the electric system works on the powertrain, can be flexibly assigned. It is also possible to run the car on the combustion engine only. By using the electric machine during acceleration, some benefits in the vehicle dynamics could be observed. Disadvantages are lesser flexibility of the powertrain since many components must be placed together. The rate of energy generation is limited due to a comparably small electric machine.

1.1.1.3 Split

Combinations between these two concepts are called a “combined” HEV or “split” HEV. Based on the parallel hybrid configuration, there are two electric machines where one is used as prime a mover or for regenerative braking, working the same way as in the parallel hybrid, and the other one acts like a generator in a series hybrid system (Guzzella and Sciarretta 2007), which is used to charge the battery. This system combines the advantages from both the parallel and the series powertrain, but likewise it also combines the disadvantages of both configurations.

1.1.2 Hybrid concepts in race cars

The idea of a hybrid car is not new. The first hybrid car was developed by Lohner-Porsche in 1899. It had two in-wheel electric drives at the front, a maximum velocity of 50 km/h and range of 50 km (Berman 2007). The lead-acid battery had a weight of 410 kg. Since this vehicle, hybrid technology has developed significantly.

The changed environmental awareness in the car industry is reflected also in their investigations in race vehicles which are eligible for higher environmental compatibility which results in lower CO_2 emissions due with is a consequence of lower fuel consumption.

For instance in 2009, the Fédération Internationale de l'Automobile (FIA) allowed in the FIA Formula One World championship, also known as Formula 1, series the usage of a system for recovering kinetic energy, called "kinetic energy recovery system" – KERS. It was abandoned in 2010 but re-introduced in 2011 and allowed in 2012 (2012 Formula one Technical regulations 2012).

The leading thinking behind introducing the KERS system in Formula 1 by the FIA was lowering the environmental impact of Formula 1 (Teams Comment on F1's Environmental Future 2008). Figure 1 (Bayona 2012) shows a Formula 1 car equipped with the KERS system. In 2009, FIA regulates the power of the KERS system to 60 kW (82 hp) which can be used over a time span of 6.6 seconds per lap. This means the total energy release in one lap is 400 kJ (2012 Formula one Technical regulations 2012). In 2014, the power of the KERS system will be doubled to 120 kW (163 hp) and the time span will be extended to 8.3 seconds (2014 Formula one Technical regulations 2011).

Figure 1: 2012 Formula 1 car with KERS

Formula 1 cars are sensitive to weight and the approximate 25-30 kg[1] of additional mass of a KERS can lead to balance problems of the car as it raises the car's center of gravity. Note that the overall car weight of a Formula 1 car is approximately 600 to 700 kg, including the driver. The KERS system is criticized from environmentalists because some of the used components have to be treated as hazardous waste while disposing.

As the regulations allows both flywheels and super capacitors, the teams made intensive tests in 2008. One method is using a dynamic brake like it is used in hybrid cars. During a braking maneuver, the kinetic energy is converted by a generator to electric energy and saved in an electric storage like

[1] Exact weights of the entire system are not published by any team.

a super capacitors or accumulator. This method is used by most teams. Another approach is accelerating a fly wheel in an evacuated cylinder up to 64000 rpm during braking. The stored energy in the fly wheel can be used at a later time. Williams-F1 Team uses a Continuous Variable Transmission (CVT) which delivers this additional energy to the powertrain. Porsche uses this technique in the 911 GT3 R Hybrid which is used in FIA-GT3 series.

Porsche developed a hybrid version of their 911 for the GT3 series, the Porsche 911 GT3 R Hybrid (Figure 2 (911 Magazine 2010)), which was shown to the public at the 2010 Geneva Motor Show and took place as experimental green race car at the Nurburgring 24hrs in May 2010. The car uses the traditional Porsche 911 4 liter, 480 hp opposed six-cylinder engine which drives the rear wheels. However, the front wheels are electrically driven. These electric motors are connected to a 40 000 rpm so called "electric flywheel power generated". This system is fully controlled by the driver who can decide when to use it to provide extra power or increase fuel efficiency. (Porsche 911 GT3 R Hybrid Race Car Will Make Its Debut at the Geneva Motor Show 2010)

Figure 2: Porsche 911 GT3 R Hybrid

Toyota and Audi developed a parallel hybrid powertrain for the 2012 LeMans series. Exact information about their powertrain has not been published, but it is known that Toyota utilizes super capacitors while Audi utilizes a flywheel instead. Both Toyota and Audi motivates their choice of their energy storage with the lower weight and the quick reaction to the extreme deceleration forces and the quick discharge of the energy storage to the electric machine which supports the combustion engine. Both teams mount the energy storage abreast of co-driver seat. (von Thomas Lang 2012)

Recently, as the collective consciousness of the society attitude of the car industry to the environmental awareness of racing car manufacturers changed, at least due to the public debate.

1.2 Purpose

As a part of a technology plan to reach a fully electric vehicle, CFS initiated this project to develop a hybrid powertrain with the possibility to run on combustion engine only. The purpose of this project was to evaluate alternative hybrid solutions for the application of Formula Student. The hybrid powertrain that will propel the car of future CFS teams will then be designed with the aim of enhancing the dynamic performance and lowering the fuel consumption. The purpose of the project can be summarized by maximizing the torque for all speeds, and since the electric machine has its maximum torque at low rpm and the combustion engine has higher torque at higher rpm a combination of these should increase the overall torque, and simultaneously keep the weight and the fuel consumption as low as possible. Appendix G How GT-Power works shows the principal torque-speed curves for the different kind of engines/motors.

The purpose can be divided into sub goals. The first goal is that CO_2-production and fuel consumption must be reduced as they are measured and rated at the competition, with more points being awarded, the lower the figures attained. The ultimate goal is to achieve as many points in the competition as possible.

The second goal is that the driving performance of the car should be equal to that of the most recent CFS vehicle, but ideally have a higher performance resulting in lower recorded times for all dynamic events.

The third goal is that the overall weight of the car should be reduced compared to the most recent CFS vehicle, which will have the added effect of improved dynamic performance.

The fourth goal of this project was to investigate the possibilities of recuperating the kinetic energy available while braking and using it while the car is accelerating.

The subsystems being investigated in this project were: combustion engine, electric generation and motor, electric storage, physical interaction between the two drivetrains and the possibility to control these. All components must be packaged into the frame. Whether it is possible or not and how it should be done were also examined. One of the most important purposes is to find a reliable solution with possibility to run the car even without the new electric system of which the knowledge is limited.

The summarizing question became “How should the hybrid system be designed and how does this solution point toward a fully electric system?” The aim of this report is to answer that question.

In the end of the project one solution was chosen and motivated. This was the intention from the beginning and the next CFS team will be able to start implementing this solution in the competing vehicle.

1.3 Goal statement

In the beginning of the project the project group stated a common goal:

“During the spring of 2012 we will develop and test a hybrid powertrain using proven engineering methodology in order to provide an increase of performance to future CFS teams.”

1.4 Methodology

In order to achieve the goals it was necessary to research hybrids and racing design to understand the problems posed and how best to solve them. The project was completed in conjunction with members of the 2012 team where necessary, but has primarily been an independent study.

An engineering approach was used and the work began with making a functional model which is shown in 2.2 Functional analysis and experts in the field of hybrid technology were consulted. The project group consists of six students from different programs and nationalities. The study was divided into subprojects for following the individual interests and experiences of the students. The students with mechanical background were responsible for the combustion engine, physical interaction and packaging of the components while the students with electrical background investigates on the electrical components and the student from the automation and mechatronic program made studies on the control system.

Before anything could be put into practice benchmarking was undertaken to investigate how the problem posed had been solved previously and what type of components would suit the application best. Calculations and simulations were made and target values were set up to have values to work toward. After this the concepts were evaluated and developed and one was chosen as the best for our application. Finally the choice of components and how they should be packed and cooperate with each other were made.

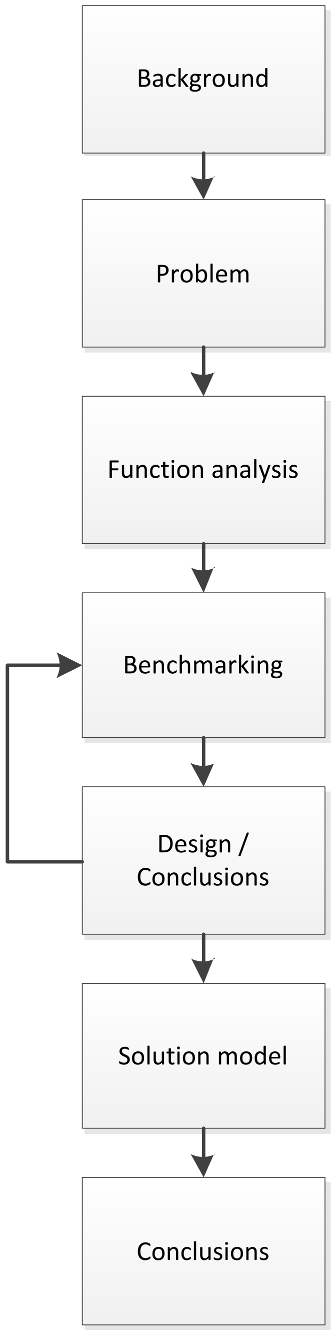


Figure 3: Work process

Figure 3 describes the work process and also the structure of this report. First of all there is a background that has led to a problem which may be solved. To focus the project correctly a solution analysis was made which then shows which parts of the project being in need of being investigated. The functional model also makes it easy to divide the project into smaller ones and set up responsibilities. There are often several already existing solutions of the problems. During the benchmarking phase these are studied and boundaries and e g the regulations by the competition (Rules 2012), are also considered. The design phase includes calculations and simulations in order to get values for choosing suitable components. It also includes the choosing of components and the packaging of them. When this is done a solution model can be made. The benchmarking, design and conclusions are in some ways made simultaneously. Conclusions are made through the entire project but in the end, which is shown by Figure 3, final conclusions which are more general are made.

As stated above the structure of this report follows the work process and thence differs slightly from the standard form of a Chalmers Bachelor Thesis. The theory and the benchmarking chapters have been merged into one chapter “Theory and benchmarking” since they cannot really be separated. This chapter considers a small section of theory of each subject and different alternatives that have solved similar problems earlier. It also includes comments on the rules of the competition regarding each subject and some conclusions that might be made after the benchmarking. The system is more or less defined in this chapter and then the “Design” chapter discusses the sizing and configuration of these systems. The design targets are stated in the end of the benchmarking phase and used as the basis for the decisions made in the design phase.

1.5 Limitations

Limitations were established since the project could not investigate all possible solutions. Some were given by the supervisor and some were set up by the project group.

- The report only includes aspects of the alternative powertrain and systems directly related to that. Other systems may be discussed briefly if necessary.
- Due to the complexity of the project, it was not possible to investigate all subsystems within the powertrain. Therefore the most important and new ones were analyzed and the ones not that important specifically for the hybrid system were not further studied. Those systems may be developed by the main team of the car in which the hybrid system implements.
- The following subsystems: starter and electric generation for the low voltage system, fuel system, air/fuel mixer, lubrication system, ignition system, and cooling system were not studied nor developed in any detail. The electronic control system is not ready for driving but still investigated.
- The report does not consider internal combustion engines with more than two cylinders and 610 cc. Two stroke engines, Sterling engines, gas turbines and other types of engines are also excluded. This is on the one hand due to the specifications given in the FSAE rules (Rules 2012) and on the other hand due to the size of the project.
- The energy storage method that was investigated was batteries and super capacitors since the supervisor limited system to be electrical. For example is it possible to store energy in flywheels but these methods are not discussed. These limitations are given to shorten the investigation time and get the project going in a reasonable time.
- Since the main purpose of the project is a theoretical concept and time limited, a physical model was not possible within this project but it is intended that the results of this project will be used in future CFS projects.
- No efforts in marketing of the project and/or the alternative powertrain occurred.
- Since the car will participate in Formula Student competitions the design of the hybrid system was bounded by the rules of the competition.
- This project does not cover the evaluation of different strategies for increasing the engine power by tuning, but the general notes about the impact of the air restrictor on the engine performance will be evaluated with this model which gives general tendencies.
- This project does not cover the development of an FSAE rule compliant air intake system and exhaust system due to the lack of time and the overall limitations.
- The environmental impacts of the material choices will not be considered.

2 Benchmarking and theory

Before starting to develop the concept, benchmarking was undertaken whereby subgroups were formed and their first mission was to investigate the market – what is available, and how other teams had done. There is no reason to reinvent things and therefore this phase is important as other teams may have experienced difficulties and by studying them, these can be avoided.

2.1 Existing Formula Student hybrid cars in general

One part of the benchmarking was to investigate existing solutions of the hybrid system of race cars and current formula student cars. There are many differences between these and it is neither possible nor relevant within the scope of this project to present them all, but contained in this section are general properties and experiences.

There is a mix of parallel and series hybrids. The choice of hybrid system is probably not the main issue, both perform well. There are pros and cons with both kinds of systems. The important thing is to build the system well and give the car the right conditions. However, this project was quickly focused on a parallel hybrid in order to increase the reliability. One example of a car with big issues was Dartmouth's Tina 2007 (DFR History 2007). It was a series hybrid and got capacitor problems just before the competition so it was not able to start. With a parallel hybrid it is still possible to drive, even if the electric system does not work.

Brushed and brushless motors are both common. They have both their own pros and cons. The Lund team (Osvaldsson and Skoog 2012) was convinced that brushless are the best because there is a smaller risk of sparks which can be dangerous close to the fuel.

The most common engine size is 250 cc. This is probably because bigger engines were prohibited. Newer cars usually have twice as big engine since the rules now allows up to 610 cc. A small comparison is given from Carleton's university (Revving it Up with Powertrain 2011) a 510 cc engine performs approximately the same as a 250 cc plus 2 PMDC motors (26 kW, 53 Nm).

Batteries and super capacitors both occur frequently. The choice of respective depends on the choice of hybrid system and the application.

It is extremely important to keep the weight low. A heavy car needs more power and gets worse road properties. Illinois's series hybrid Hammer Hawk (The cars 2012) weighted as much as 350 kg and Lund's LUR4 (LUR4 Hybrid 2012) weighted over 300 kg as well. This may be compared with Chalmers Formula Student with around 100 kg less. (All weights are excluding driver.)

2.2 Functional analysis

The functional analysis, summarized by the functional model of the powertrain, was made to visualize the functions and how they interact. The functional model is shown by (Figure 4). From this different workgroups are created which are responsible for the different areas of the project. The functional model is then divided so that all subprojects know the responsibilities and boundaries of theirs. These smaller functional models also show the connections to the other subgroups and thus defining clear communication paths.

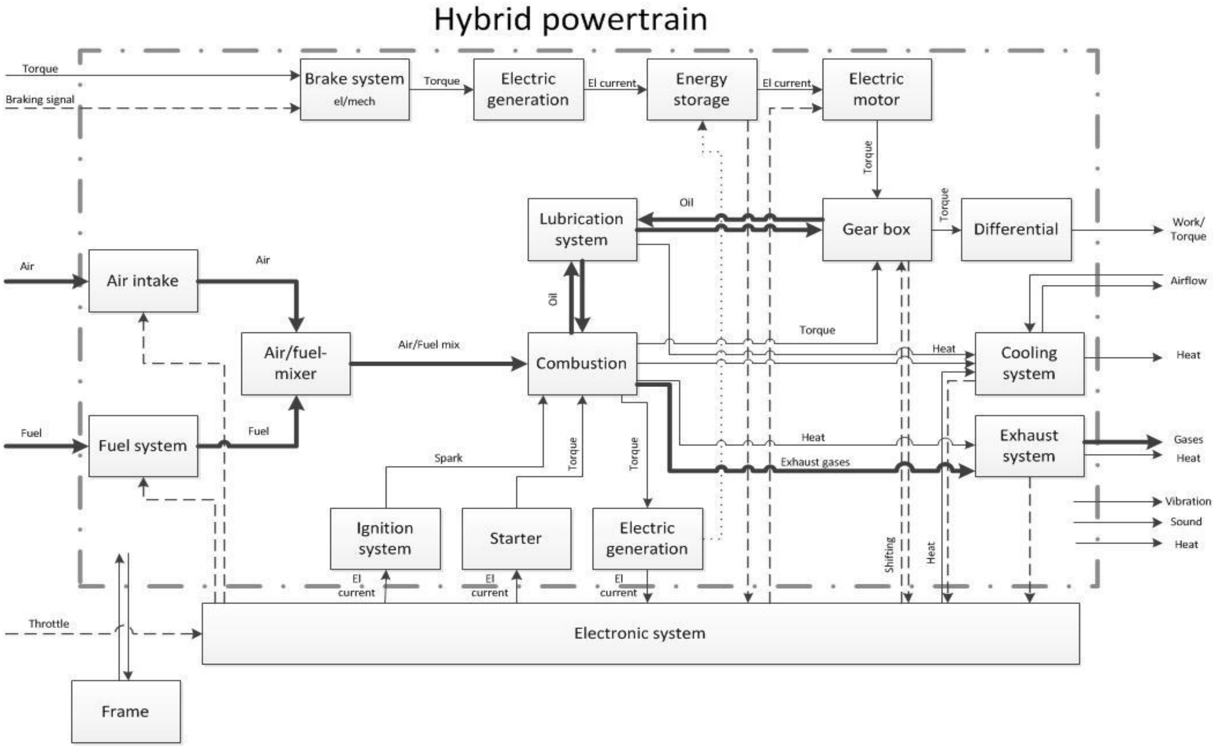


Figure 4: Functional model of the powertrain

Thin arrow = Energy transportation
Thick arrow = Fluid transportation
Crosshatched arrow = Information transportation

The ICE needs a premixed air/fuel mixture in order to operate. Combustion releases heat and emissions and needs to be lubricated. The mechanical output is torque. The torque drives the gearbox which drives the differential, which drives the wheels.

The electronic system includes the ignition system, the starter, electric generation and control system. The ignition system and starter are required in order to run and start the internal combustion engine respectively.

Heat is generated by the internal combustion engine, ICE, the electronic system and the electric components of the powertrain.

Everything in the system has a relationship to the frame since it all will be mounted.

2.2.1 Internal Combustion Engine

The combustion engine includes the combustion system and the lubrication system. The air intake and the fuel system are included in the combustion engine since they have a significant influence over the combustion. The functional model of the combustion engine is shown in Figure 5.

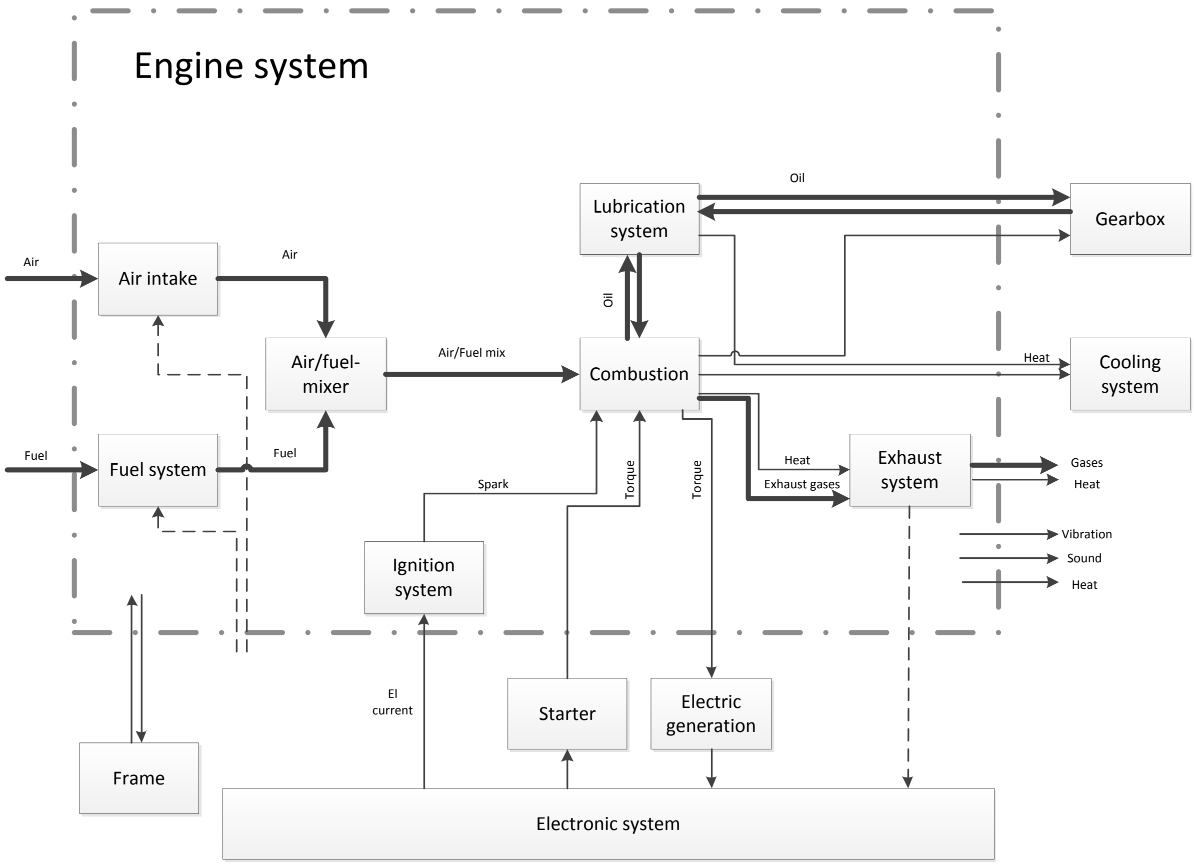


Figure 5: Functional model of the combustion engine

2.2.2 High voltage electric system

The electric powertrain has the purpose of delivering and receiving energy to and from the car respectively. When the car brakes the electric machine will act as a generator, converting the cars kinetic energy to electric energy which will be stored. At a later point the stored energy will be delivered to the electric machine and used to propel the car forward. A power control unit will be used to control the current through the electric machine. The power control unit needs a control signal so it knows how it should direct the current. All components in this system have heat losses. This is shown in Figure 6.

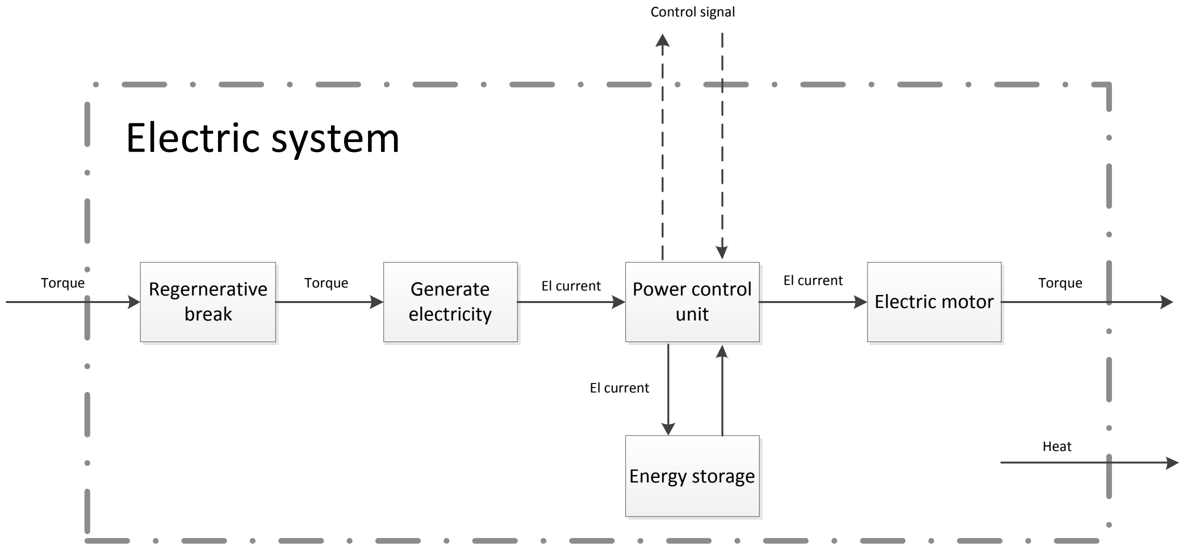


Figure 6: Functional model of the high voltage electric system

2.2.3 Cooling system

The main purpose of the cooling system is to maintain the temperature of the engine system and the electronic system within a range where no damage occurs due to overheating. The combustion generates elevated temperatures higher than the surrounding materials could withstand without cooling. In addition to this, the electrical components are heat-sensitive and the electrical devices of the hybrid system generate heat and therefore require a cooling system. The functions are shown in Figure 7.

The existing formula student car is water cooled, that means all the generated heat inside the subsystems releases their heat to the cooling liquid (in this case: water) which heats up and this is then cooled down by airflow though the heat exchanger, which occurs during driving. The electronic system of the car is able to control and regulate the car's cooling system.

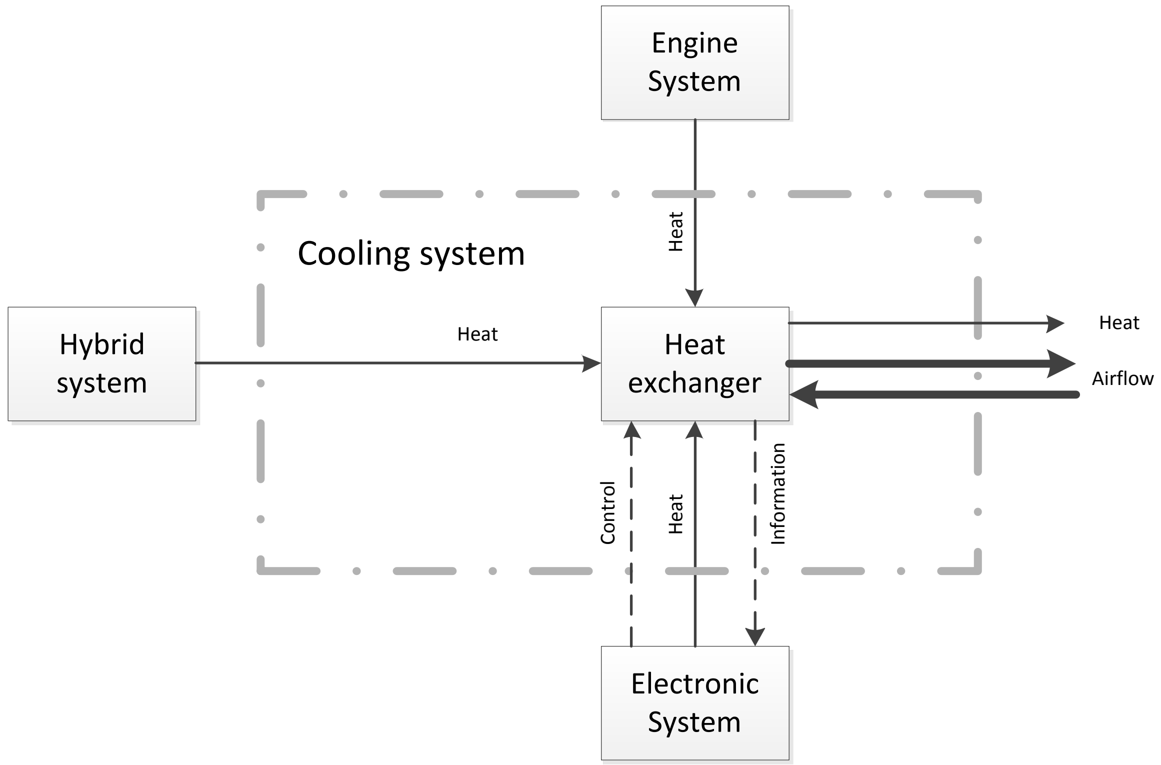


Figure 7: Functional model of the cooling system

2.2.4 Transmission system

The main purpose of the transmission system (Figure 8) is to transmit torque from the power sources and send it to the wheels. The gearbox itself is connected to a differential which is linked to the rear wheels and needed to let the wheels roll at different speeds, for example in curves. Lubrication is needed for the internal rotating parts.

Since it is not given how the system will be configured this is the simple way to include both the powering systems. The electric motor could deliver the torque directly to the wheels for example.

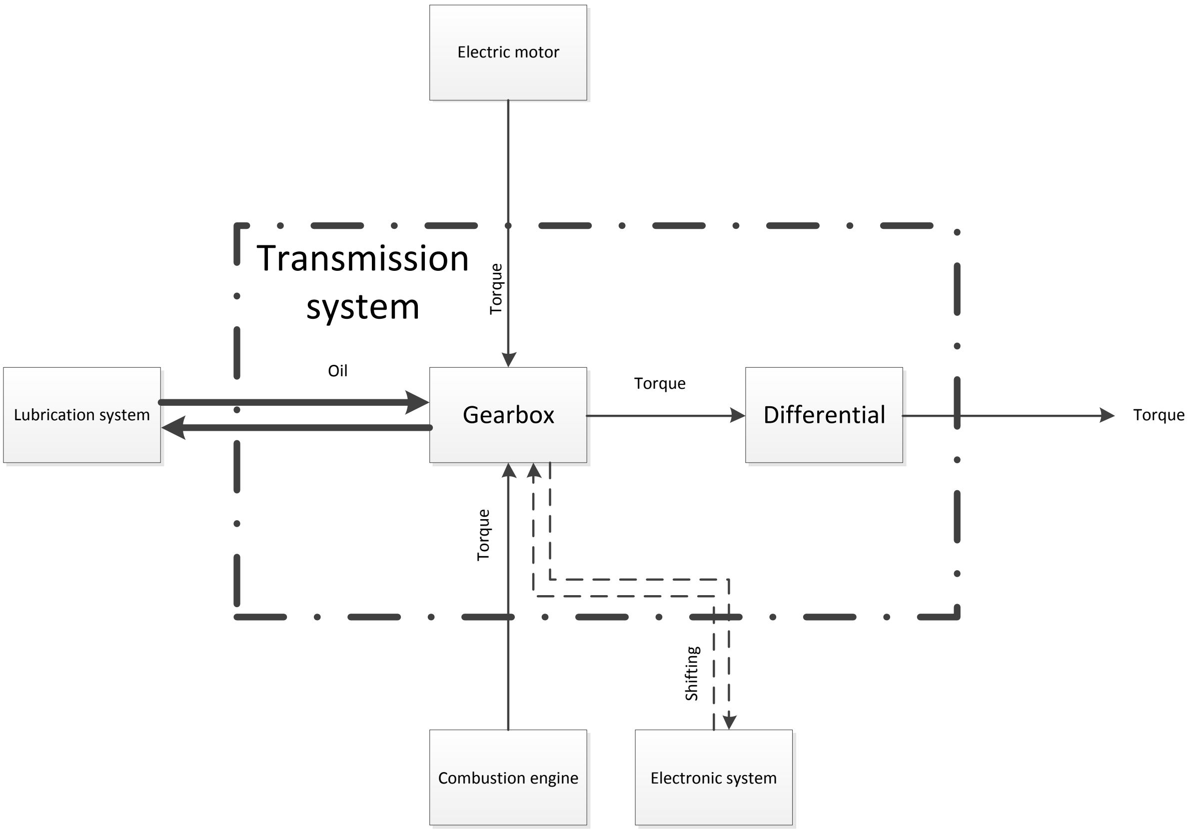


Figure 8: Functional model of the transmission system

2.3 Combustion engine

This section describes the grounds on which the combustion engine is chosen. The main principle behind a combustion engine will be introduced. The derivations are assumed from Fundamentals of Internal Combustion Engines (Heywood 1988).

2.3.1 Theory

The centerpiece of a car is its internal combustion engine. Considering the ICE as a block, it has the following form, shown in Figure 9.

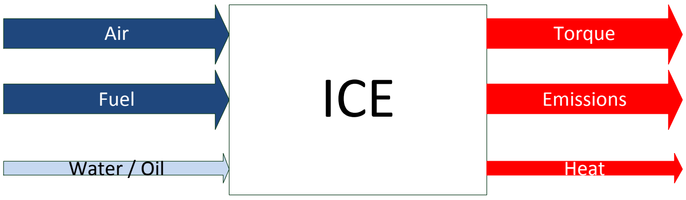


Figure 9: ICE as block-scheme

It is a heat engine, which converts the chemical energy of the fuel to mechanical work by burning the fuel with air. The thermal expansion of the ignited air-fuel mixture is used to move the piston. Common examples for combustion engine are Otto and Diesel. The Otto-cycle which is named after Nikolaus Otto, the inventor of the first combustion engine of this type, is also called "constant volume cycle" and due to the fact that the air-fuel-mixture is ignited by a spark, these engine are called "spark ignition" engines. In contrast, diesel engines, invented by Rudolf Diesel, are described in the "constant pressure" cycle and since the ignition is triggered by compressing the gas, they are also known as "compression-ignition" engines.

Even though the combustion progress is significantly different, the main theory behind converting the chemical energy of the fuel to mechanical torque is the same: the burning gas expands, releases heat and presses down the piston which is connected through the connecting rod to the crankshaft. This mechanism converts the horizontal movement of the piston to a rotational movement of the crankshaft. Losses which decreases the overall efficiency of a combustion engine to about 30 %, appears both in mechanical friction and heat- and flow loss during the entire cycle (Heywood 1988).

2.3.1.1 Derivation of the produced torque

One of the changes of states during the thermodynamic cycle of an ICE is called "power stroke", where the piston is pressed down and work can be utilized from the piston or rather from the crankshaft. This cyclic work can be expressed as:

$$Work/cycle = \frac{P\,n_r}{N}$$

Where P is the emerging engine power, n_r the number of crank revolutions for each power stroke (2 for a four stroke engine) and N is the engine speed (Heywood 1988). Normalizing this work to the displaced volume V_d is called mean effective pressure (MEP):

$$MEP = \frac{P\,n_r}{V_d N}\ \text{[bar]}$$

Its unit is like that from a pressure, but it is not a pressure in a physical context. The engine parameter brake mean effective pressure (BMEP) is a useful parameter since it is the scale of measurement for an engine's potential to produce torque. A formula expression is:

$$BMEP = \frac{P_b\, n_r}{V_d N} \text{ [bar]}$$

where P_b is the brake power:

$$P_b = 2\pi NT \text{ [W]}$$

where T is the torque of the engine, which can be described as the engine power useable to load a brake.

The efficiency of the engine (also called "fuel efficiency") is expressed by:

$$\eta_f = \frac{P}{\dot{m}_f Q_{HV}}$$

where $\dot{m}_f$is the mass-flow indicated each cycle and Q_{HV} the heating value of the fuel, measures the efficiency of converting the chemical energy of the fuel to power. Typical values for the heating value are 44 MJ/kg and 42.6 MJ/kg (Heywood 1988) for gasoline and diesel respectively.

Another important parameter is the volumetric efficiency, which measures the effectiveness of the engines induction progress. It is defined as the volume flow rate of air into the intake system divided by the rate at which volume is displaced by the piston.

$$\eta_v = \frac{m_a}{\rho_{a,i} V_d} = \frac{2\dot{m}_a}{\rho_{a,i} V_d N}$$

$\rho_{a,i}$ is the density of the intake air, m_a and $\dot{m}_a$ are the mass of air and its flow rate respectively. Heywood suggests typical values in a range of 80 % to 90 %.

As shown in Figure 9, both the mass flow rate of air and fuel can modified independently but within a certain range. The rate of those two mass flows is an important operating condition and is called air-fuel ratio (A/F -ratio):

$$\frac{A}{F} = \frac{\dot{m}_a}{\dot{m}_f}$$

Typical values suggested by Heywood are between 12 and 18 for gasoline engines and 18 to 70 for diesel engines. An A/F-ratio equal to 14.7 means that the combustion is stoichiometric and the fuel is burned completely (Heywood 1988).

These parameters can be combined for showing the relations between each parameter in a four stroke engine:

$$P = \frac{\eta_f m_a N Q_{HV}}{2\,(A/F)} \text{ [W]}$$

With consideration of the introduced volumetric efficiency, the final equation for the engines power and torque reads:

$$P = \frac{\eta_f \eta_v N V_d Q_{HV} \rho_{a,i}}{2(A/F)} \text{ [W]}$$

Or with taking the relation between brake power, BMEP and torque into account:

$$T = \frac{\eta_f \eta_v V_d Q_{HV} \rho_{a,i}}{4\pi(A/F)} \text{ [Nm]}$$

2.3.2 Rules

Following the Formula SAE UK rules (Rules 2012) the engine has to match the following rules:
- four stroke
- maximum of 610 cc
- car must be equipped with a carburetor / throttle body of any size or design
- drive by wire is allowed for hybrid and electric vehicles according to Article B 8.5.2

Further it is dictated that all air which supplies the engine must pass through a restrictor of 20 mm diameter[2] which reduces the amount of air flowing into the cylinder. The rules specify also the position of the throttle body. It is written in the FSAE rules (Rules 2012): *"In order to limit the power capability from the engine, a single circular restrictor must be placed in the intake system between the throttle and the engine and all engine airflow must pass through the restrictor."*

A reduction of air, without any further modifications on the engine as shown in section 2.3.1 Theory, leads inevitably to a loss of maximum power and torque. As engines in the 600 cc class can produce up to 120 hp without the restrictor, the limitation makes the engine and the entire vehicle safer to drive for the non-professional drivers. At the same time, this restrictor represents a challenge for the teams to overcome the performance losses.

2.3.3 Alternatives

The CFS team uses a 600 cc Yamaha FZ6 engine since CFS-06 and motivates the use of this engine in CFS-12 by the considerable amount of knowledge and experience with the different subsystems, dyno-testing and tuning. Further they observe that this engine delivers high torque and horse power at lower engine speed at lower compression ratio. Inside the CFS team there is a large inventory of spare parts available. Some data of this engine is presented in Table 1. It is a heavy motorcycle engine with four cylinders but often used in Formula Student contexts and the reasons for this are the high reliability of the engine, the good availability of spare parts and the sufficient power.

Table 1: Data of CFS 11 car with Yamaha FZ6 Engine

Engine model	**2008 Yamaha FZ6 Fazer**
Cylinders / Valves	4 / 16
Maximum power [kW]	63
Maximum torque [Nm]	63
Engine weight [kg]	85
Car weight without driver [kg]	217
Power-to-weight ratio [kW/kg]	0.29

[2] If E85-fuel is used, the prescribed diameter is 19 mm.

This enables the calculation of the vehicles power-to-weight ratio which is the quotient of the peak power of the vehicle to its dry mass. Hence it can be described as energy density of the entire car. Formula 1 cars have a power-to-weight ratio of 1 kW / kg (Williams FW27 n.d.), Sport cars about 0.2 - 0.3 kW/kg (1997 McLaren F1 GT 2009) and common street cars about 0.1 – 0.2 kW/kg (Subaru R2 S technical specifications 2008). This states two facts clear:

1) The higher this ratio, the higher is its assumed acceleration due to inertia effects.
2) It allows a more general comparison of formula cars attending the same competition.

One aim of a hybrid powertrain is to gain the same power as a conventional powertrain but with lower fuel consumption and hence lower emissions. Therefore the combustion engine and its role as source of emission is one of the first which are studied in detail. In this case, four major strategies can be followed and will be reviewed in the following. All four strategies include electric parts for the hybrid powertrain and of course optimizing the driveline.

- Keeping the current Yamaha Fazer 600 cc engine
- Taking a small diesel engine with turbocharger
- Taking a smaller and lighter combustion engine
- Downsize an > 610 cc engine to < 610 cc

2.3.3.1 Keeping the current Yamaha Fazer 600 cc engine

Keeping the current engine would be the easiest method both from the technological and the economic point of view. As written above one can benefit from the in-house knowledge and no additional costs for the powertrain would occur. Looking at competitions shows that most teams continue to use the same type of 600 cc motorcycle engine over a period of several years. This approach is very reasonable since the combustion engine is a huge cost factor and by keeping the engine over a number of competing years rises a unique in-house experience with the engine.

The main weakness is the loss of dynamic performance with the additional weight of the electrical powertrain. The worst case scenario could be that the additional power of the electric machine is only used to carry the whole package and no improvement in the vehicle dynamics or in fuel consumption can be observed. Further, the limited space and packaging regulations by the rules does make this an impractical alternative.

2.3.3.2 Taking a small diesel engine with turbocharger

An alternative solution is the development of a one cylinder turbocharged diesel engine which combines both the advantages of using a diesel engine and an electric machine. A Chalmers development would need to be undertaken as there are no suitable engines on the market.

The advantages of using a diesel engine in a hybrid system are based on the typical Diesel advantages: Comparing a diesel and a gasoline engine with equal power, the diesel engine has a higher torque since diesel fuel has a higher heating value, a higher efficiency based on the higher possible compression ratio and lower emissions of HC and CO. This makes diesel engines a more suitable choice in hybrid powertrains.

On the downside there are the higher emissions of NO_x. The weight of a diesel engine is higher since it contains more robust parts used which can resist the higher occurring forces.

The idea displayed below is based on a one cylinder diesel engine which provides sufficient power both for driving without electric system and in the hybrid case. According to the FSAE rules, the air restrictor is not required for a diesel engine (Rules 2012).

Consider a standard turbocharger as shown in Figure 10 (Heywood 1988) with a compressor (C) and turbine (T) connected by a mechanical rod and engine (E).

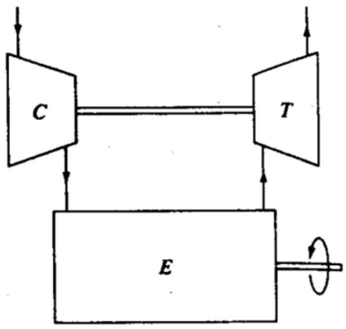


Figure 10: Turbocharger

The mechanical connection between the compressor and the turbine can be replaced by using the rotation of the turbine for generating energy which is stored and used for driving the compressor. The advantage of this decoupling is the higher flexibility of the turbocharger because the working point can be changed. In ideal case, this is done by the engine control unit, depending on the demanded load power. A more advanced step is to store this energy in the energy storage of the hybrid system. Even a solution without a turbine can be discussed. That means to use a supercharged system with a compressor which obtains its electrical power from the electrical storage.

As this goes beyond the scope of this work it cannot be discussed in detail. Some competitors are also using custom engines which are designed in close partnership with partner companies, the majority of which are engine manufacturers.

Using a turbo/super-charger might open gains in engine performance but will increase the fuel consumption of the engine. It should be noted that as the selected vehicle is driven around a narrow, twisty autocross style course, the engine has to provide a responsive and smooth application of power. The turbo charger lack can make the car unpredictable during cornering and requires therefore special driver training as well. This underlines why the natural aspirated engine is the most popular choice by most competitors in the FSAE competition.

2.3.3.3 Taking a smaller and lighter combustion engine

The 600 cc engines like those which are commonly used in FSAE have their origin in motorbikes in the 600 cc superbike class. Typical examples for these motorbikes are the Honda CBR600, the Yamaha YFZ-R6 and the Suzuki GSX 600, with engine power of approximately 98 kW in production guise. This power output is not achievable with the air-restrictor and natural aspiration. In 2006, QFR-Racing managed to increase the power output of their restricted Yamaha YZF-R6 FSAE Engine to 65 kW and reached the second place in the acceleration event. (Corrigan, McCullough and Cunningham 2006). In 2011, CFS-11 won the acceleration contest, using a similar engine and a slightly lower power of 63 kW. That implies the assumption that such a restricted 600 cc four-cylinder engine works under lower stress than in their original state and are therefore oversized and overweight for this application (Corrigan, McCullough and Cunningham 2006).

The current Yamaha Fazer engine weights approximately 85 kg and therefore a lighter layout is desirable. Further benefits from a smaller and lighter engine might be the lower center of gravity-point and decreased polar moment of inertia. These values both have an influence on the vehicles dynamics and can lead to improvements in the vehicles handling, acceleration and braking behavior. Hence it is reasonable to assume, that a lower center of gravity (COG) can improve car handling and negate minor accelerative by lowering the roll center of the vehicle. Further, it contributes to the CFS lightweight concept which will be more discussed in the future by the CFS team. The difference in size between a single cylinder engine and the Yamaha Fazer four cylinder engine is illustrated in Figure 11.

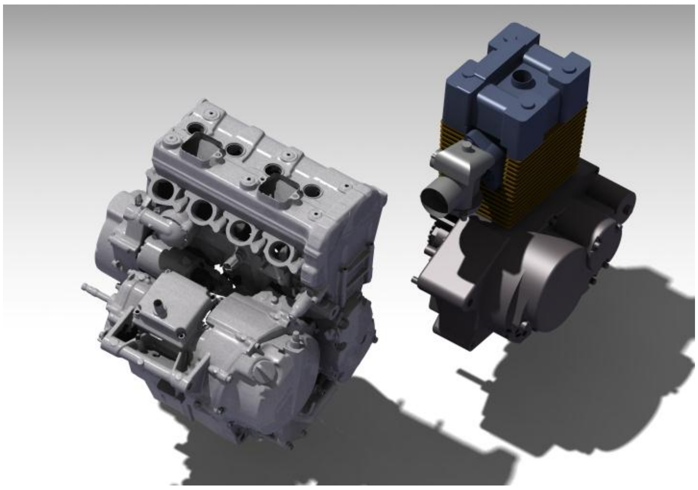


Figure 11: Yamaha Fazer vs. conceptual single cylinder engine

There is a large number of one cylinder four-stroke SI engines on the market which are commonly used in motocross motorbikes and in all-terrain vehicle (ATV) as well as stationary work machines such as power generators or high-pressure cleaners. Following the scope of this project, this work discusses only engines with a displaced volume between 450 cc and 610 cc.

In order to make suggestions about the engine power, another value will be introduced which allows comparisons between different engines over a wide range of displaced volume and making objective data driven decisions: The specific swept volume power ratio (ssp-ratio) indicates the power which the engine would have at a displaced volume of 1 liter (1000 cc)[3] and has got the unit kW per liter. A formula expression is:

$$ratio_{ssp} = \frac{P}{V_d} \text{ [kW/l]}$$

where P is the engine power in kW, V_d the displaced volume in liters. The significance of this value is limited as it indicates rather a rate of taping the potential of the specific engine to produce power. With regard to suitability and performance of the car one has to mention that parameters affecting the choice of the engine such as reliability and availability of the engine are not covered in any way

[3] The normalization to one cylinder is not reasonable since it makes more sense to evaluate the entire engine as one package.

by this value. The introduced power-to-weight ratio is much more suitable for making statements about vehicle performance

Appendix D Short list of engines presents the engine that has been shortlisted for this work. However it was not possible to obtain all information from the manufacturers about the engines by the end of this study.

It becomes obvious, that engines with lower displaced volume than this provide consequently less power and torque, too less to be considered to deliver sufficient power for driving the car with reasonable performance. Since the engines of interest have only one cylinder their weight is lower compared to the current engine due to the smaller and simpler engine case and a reduction in rotating mass such as the piston and the con rod. This allows the assumption of an improved throttle response of a one cylinder engine.

However the provided information allows data driven decisions and shows that even 600 cc one-cylinder engines do not achieve the ssp-ratio of the restricted four cylinder 600 cc engine. The ssp ratio is noticeable: The current Yamaha 600 engine is given as reference and has an ssp-ratio of 105 kW/l. Only the Yamaha YZ450F with the set-up from (Corrigan, McCullough and Cunningham 2006) with an ssp-ratio of 91 kW/l is similar. Other engines such as the Husqvarna engines reaches ssp-ratios in the range about 65 to 70 kW/l. The value of the Aprilla SXV 550 engine with 26.2 kW/l is outstandingly low. In contrast both the BMW G450X and the KTM 450 EXC have equal values of around 83 kW/l. The presumption seems reasonable that these two motorbikes are using the same engine.

Comparing these values leads to the following assumption:

- A high value (> 70 kW/l) indicates well-tuned engines and a high stress working condition for the engine
- A medium value (50 to 70 kW/l) indicates a potential for gaining higher power by adequate tuning
- Low values (< 50 kW/l) indicates comparably poor power output from the engine

With regard to the construction and packaging of the powertrain, a one cylinder provides benefits due to the possibility of installing it in the car both longitudinally and laterally. This is an important advantage which gains the constructor freedoms of finding possible solutions for the overall packaging of both the combustion engine and the electric powertrain.

Motorbikes such as the KTM 450 EXC, BMW G450X and Yamaha YZ450F are used for motocross applications and, as the ssp-ratio showed, are often tuned for high performance. This affects both sensitive parts like the valves and cylinder heads, which usually have a lightweight design, but the pledged higher performance through quicker throttle response (lower inertia) is at the expense of the durability of the parts. This is reflected in short and extensive service intervals which includes not only a regular oil change but also the change of valves, piston, camshaft and crankshaft after 15 - 25 operating hours. Further, full throttle driving is not recommended as the lubrication system of these engines is engineered for a typical motocross application which does not provide the case of a full throttle ride. The weak point is the lubrication system which cannot assure the oiling of the piston in full throttle case. With regard to the 75 m acceleration event where the car will be driven in full throttle, this is a disadvantage which disqualifies these engines from being considered in the proposed application.

The Husqvarna SMR 630 engine which is also mounted in the Husqvarna TE 630 is notable[4]. It combines both high power and torque and long service intervals. Husqvarna dealers and drivers of the named motorcycles report concurrently the high reliability of this engine. The full specifications and excerpts from the driver's manual about the service interval are attached in Appendix J Husqvarna TE 630 service interval data.

Looking at competitors shows that in 2012 no team is utilizing this engine apart from the German team “KüHN-Racing” utilizing a similar engine, the Husqvarna SMR 511, a one-cylinder four stroke SI-engine with a displaced volume of 477,.5 cc (KüHN Racing e.V. n.d.).

Some teams are using the Yamaha WR450 engine. It has been said, that Yamaha is potentially willing to allocate this engine for Formula Student teams for free. This could not be confirmed. The FSAE team from Belfast studied the possibility to tune the Yamaha 450 engine up to 41 kW with natural aspiration but this power is not sufficient for being competitive (Corrigan, McCullough and Cunningham 2006).

At the FSAE competition some teams utilize the Aprilla SXV550 engine. However, there is unanimous agreement inside the teams, that this engine is not suitable for the FSAE application.

It can be observed that sometimes teams are changing back to the more common four cylinder engines after one or two years. One can assume that this is both due to the comparable insufficient power and to the poor reliability of some of these engine types (Corrigan, McCullough and Cunningham 2006). After 2006 the rules have been changed so that the fuel economy is worth more in the competition now. There is a number of teams which utilize a high performance motocross engine which is adapted for a use in FSAE but these teams have mostly partnerships with engine manufacturers which provide both the knowledge and the materials for adapting the engines.

There is an independent formula student series which deals exclusively with hybrid powertrains. This competition is called Formula Hybrid with 39 teams attending (Formula Hybrid International Competition 2012). These teams use mostly the Honda CRF250R Engine with 34 hp or the Yamaha YZ250F with 32 hp. The biggest difference is the limiting of the displaced volume of the combustion engine to 250 cc and no dictation of an inlet restrictor. Using these engines for the alternative powertrain application is not reasonable since the differences in performance compared to existing formula student powertrains are too high. As motivated above it is more reasonable to use the full potential of engines with higher displaced volume.

2.3.3.4 Downsize an > 610 cc engine to < 610 cc

There is also the alternative to downsize an engine in order to make it compatible to the Formula Student rules. Downsizing is an efficient concept for both SI and CI engines for lowering the fuel consumptions and the pollutants emissions. The concept is to produce a higher power and torque from less displaced volumes. Downsizing requires powerful fuel injection and super-charging systems. It makes high demands on the engine's mechanic (Golloch 2005).

[4] The Husqvarna SMR series and the TE series use the same engine.

Such an engine could be the BMW F650GS engine, a one cylinder four stroke SI engine with fuel injection which is manufactured by Rotax. The Formula Student Team from Fachhochschule Joanneum is the only team which uses this engine. They both downsize it to make it capable with the limitation to 610 cc and add a compressor. The original BMW F650GS has a maximum power of approximately 35 kW, a comparably low value for this class of motorbikes and an ssp-ratio of 53.7 kW/l which is also low. Thus, it can be assumed that the engine set up such as the cam timing, is moderate. Therefore the required air restrictor might not have a high influence on the performance of the engine and a high reliability of the engine can be assumed.

Lowering the displaced volume can only be achieved by a downsizing of the combustion chamber. The original engine has a bore of 100 mm and stroke of 83 mm which result in a displaced volume of 652 cc. To achieve the maximum 610 cc allowed by the rules (Rules 2012), it has to be reduced by 6.5 %. The displaced volume of the engine with the assumption of a cylindrical form defined as:

$$V_d = 10^{-3}\pi\left(\frac{b}{2}\right)^2 s \text{ [m}^3\text{]}$$

where b is the bore of the cylinder and s the stroke, both measured in millimeter. To keep the units consistent the correction factor 10^{-3} is introduced which converts cubic millimeters to cubic centimeters. As result the unit of V_d is cubic centimeters (cc). Since the bore has a quadratic influence it is more reasonable to focus on lowering this[5]. To end up with 610 cc, calculations show that a bore of 96 mm is required which means a reduction of 4 %.

In practice this can be realized by adding a bush into the combustion chamber which results in reducing the bore, requiring the mounting of an adapted piston. Difficulties might appear as the valves are at the extremities of the combustion chamber and this could cause difficulties with a view to the valve seat pocket. Another strategy is to develop a crankshaft which results together with a new piston in a lower stroke.

All these modifications are not trivial. Modifying the bore would come along with modifications of the cylinder block such as inserting a bush into the combustion chamber which would require significant effort. The surface finish of this bush which can be realized with common manufacturing techniques in the same way as for cylinder block might not be the problem. Instead the more complex problem is the heat transfer between the sleeve and the engine cylinder block. A sufficient interference fit between the sleeve and cylinder block must be maintained in order to assure heat transfer by conduction between the two parts. The reliability of such a modified engine cannot be guarantee and extensive testing is required.

Saving weight can be achieved by lowering the rotating masses with the help of lightweight components. Parts like the crank shaft, piston rod and piston have the potential to deliver this. Lower weight means lowering the inertia of the parts which improves the throttle response of the engine.

The development of a piston is not trivial either since the geometry of the piston to the flame development and thus to the power characteristics. The team from Fachhochschule Joanneum did this by buying an adapted piston from a manufacturer specialized on high performance racing parts.

[5] Modifying the stroke would lead to a new stroke length of 77 mm or a reduction of approximately 7 %.

The Austrian team, TUW Racing, uses a KTM LC4 690 Evo 1 cylinder engine which is downsized to 609 cc. The engine produces 46 kW (63 hp) and 65 Nm (KTM 690 LC4 Enduro 2012) as it comes off the shelf. Their 2011 FSAE car provides 82 hp and a torque of 78 Nm (TUW Racing Vienna Formula Team u.d.). The power is achieved by using ethanol as fuel and a compressor (Der Standard 2008).

As a conclusion of studying the possibility of downsizing an engine it has to be mentioned that the modifications which have to be performed in the comparably short time of this research are not covered by the boundaries of the project. A self-developed downsizing solution without extensive testing cannot accommodate with the teams purpose to achieve a high final result at the competition of next year.

2.3.4 Conclusion

Major aspects were figured out under which the engines are compared. The engines are compared regarding the points of engine power and torque as the most significant performance indicator. This specification has got the highest influence on a competitive drive power of the car, even without the electrical part of the powertrain the car must perform well. In particular, as the behavior of the hybrid powertrain in the competition is not known yet, it is desirable to choose an engine which provides sufficient power to drive the vehicle without the powertrain and has the ability to deliver a competitive performance.

The weight and size of the engine is important as mentioned. The current powertrain has got an overall weight of 85 kg. To fulfill the design target of having an overall weight of less than 85 kg (Appendix E Design targets) for the hybrid powertrain, that means both the combustion engine and the electrical drive and storage, the weight is of particular significance.

As it is planned to develop a full drive-by-wire control system of the car and as inherent of its functional principle, an engine with fuel injection is preferred. The literature ((Golloch 2005) and (Heywood 1988)) states consistently, that a fuel injection system can provide a higher overall efficiency with lower emissions and fuel consumption of the combustion engine.

To lower the running costs and in order to deliver an overall high reliability of the engine and the entire powertrain package, the service intervals which come along with the maintenance costs and the availability of spare parts are of particular interest. With regard to the reliability short service intervals pose an unjustifiable particular challenge.

The Pugh matrix in Table 2 compares the engines from the short list which seems most suitable under the aspects of performance, weight, service intervals, costs and parts availability. The Yamaha and KTM engine is not rated in terms of costs, since no information was received. Only the Subaru engine came across poorly both in costs and parts availability. (Thater 2012).

Table 2: Pugh matrix comparing one cylinder engines

	Weight	Baseline	Husqvarna TE/SMR 630	Husqvarna TE/SMR 530	Yamaha YZ450F	Subaru Polaris 500	KTM 450 EXC
Performance	3	0	2	1	2	2	2
Weight	2	0	1	1	2	-1	2
Service Intervals	2	0	1	1	- 1	1	-1
Costs	1	0	1	1	0	-1	0
Parts availability	1		1	1	1	-1	1
Total		**Sum**	**12**	**9**	**9**	**3**	**9**
	# of positive values		5	5	4	2	4
	# of negative values		0	0	1	3	1

The Husqvarna TE/SMR 630 engine gets most points. The engine has high power, long service intervals even for sensible parts such as the piston and valves and since it is introduced in 2011, the availability of spare parts is secured. The engine has the biggest displaced volume and the highest power and torque of all compared engine. Following the main aim of the study it is decided to take this engine as it is assumed that it will deliver a higher performance than any of the other engines would, even when the electric powertrain is not considered. It is likely, that this engine can produce a higher power than the mentioned Yamaha 450 engine does due to the higher displaced volume. The influence of the required air restrictor will be further discussed in chapter.

Appendix I Husqvarna TE 630 engine data and Appendix J Husqvarna TE 630 service interval data gives an overview about its new specifications.

As the engine is smaller in size and lighter in weight, no significant packaging problems can be observed. No exact geometric properties of the Husqvarna engine were available by the end of the project, but the final solution is indicated in 3 Design.

2.4 Electric machine

This section describes the background of the choice of electric machine.

2.4.1 Theory

The purpose of the electric machine is to improve fuel efficiency and performance, it will do this by acting as a generator while the car is braking and converting mechanical energy to electrical energy that will be stored. When the car accelerates the electric machine will use the stored electrical energy and provide additional acceleration. For this application it is preferred to have a machine which is constructed to work with a high rpm, as shown below, the more torque the machine is required to deliver the larger rotor is required and therefore the entire machine will be larger. For this application as high power to weight ratio as possible is required to achieve the best acceleration.

$$T = BA\,(\pi DL)\frac{D}{2}\ \text{[Nm]}$$

$$P = T\omega\ \text{[W]}$$

$$P = BA\,(\pi DL)\frac{D}{2}\omega\ \text{[W]}$$

where D is the rotor radius, L is the length of the rotor, B is the magnetic field, A is a uniformly distributed current in the rotor and ω is the angular velocity. If a machine volume and rotational speed are specified, the machine with the highest torque per unit volume will deliver the highest power.

The electric machine will have a particular specified power output, this power output can sometimes be exceeded with few problems. The amount of overload and for how long the machine can be overloaded depends on its construction and cooling. If a machine is continually loaded below its specified power and have sufficient cooling it can be overloaded for a short period of time. How much the machine can be overloaded and for how long depends on its construction. As machines can be dimensioned very differently depending on their purpose, no general correlation exists regarding how much overload a machine can withstand. However, the heat produced in the current conductors in the machine is proportional to the square of the current:

$$P_{cu} = I^2R\ \text{[W]}$$

If the current is doubled, the dissipated heat will increase fourfold. The electric machine works with magnetic forces, and there is a magnetic circuit in the machine. The magnetic flux is generated by the current that flows through the windings in the machine, which is then conducted by the iron and through the air gap in the machine. The iron cannot conduct an infinite amount of magnetic flux. If the flux density (measured in Tesla [T]) increases to approximately 1.6-1.8 T the reluctance (Magnetic resistance) of the iron will increase significantly, however this number is material dependent. If the saturation starts to affect the reluctance in the magnetic circuit even though the magneto motive force (number of turns in the windings times the current it conducts) is increased, the flux density will not increase. The information of this section was sourced by (Hughes 2006).

To be able to utilize any overload capacity of the electric machine, the electric motor drive needs to be able to handle the peak power when the machine is overloaded. This is different from the machine that only needs to be able handle the average power over the working cycle. If the machine is overloaded and run at higher average temperature than it is constructed for, its lifetime will be shortened.

2.4.2 Rules

There are many rules concerning the electric system and below the most important are listed. For the full specification, see the FS rules (Rules 2012). These regard both the electric machine and the electric storage.

- Total accumulator voltage may not exceed 600 V
- The accumulator must not store more than 7250 Wh
- The accumulator must be contained in a container which must meet the specified requirements
- There must not pass more than 85 kW though the HV cables
- The high voltage system must contain an insulation monitoring device (IMD)
- The IMD must trip when the resistance between the HV and LV system is below 500 Ω/V

Figure 12 shows how the grounding must be wired. The HV system may not be grounded by the frame. The figure is copied from the rules.

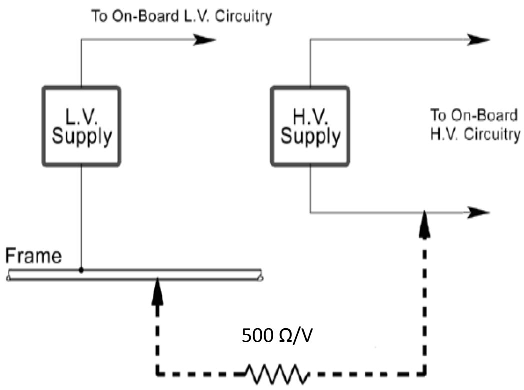


Figure 12: Low/High voltage grounding

2.4.3 Alternatives

There are different types of electric machines. They are built up differently and their characteristics differ. Below the machines are introduced.

2.4.3.1 Induction machine

The induction machine is a robust and cheap machine, it is convenient to use where an alternating current (AC) grid is available because it works by simply plugging it into an AC source. The induction machine must receive an AC current to work. When an AC current runs through the stator, a rotating magnetic field will be generated and currents will be induced in the rotor. This will produce a force between the stator and the rotor and the rotor will start rotating.

2.4.3.2 PMSM / BLDC

The permanent magnetic synchronous machine (PMSM) and the brushless direct current machine (BLDC) are more or less the same machines, the difference is that the PMSM has an electromotive force (EMF) which is similar to a smooth sine wave and the BLDC has a trapezoid EMF. These machines have permanent magnets in their rotors and the stator generates a rotating magnetic field, the forces between these fields will cause the rotor to turn. The permanent magnetic machines do not need brushes to magnetize the rotor which increase their efficiency and reliability. Due to the permanent magnets the permanent magnetic machine will produce voltage when it is rotated because the magnets always produce their magnetic field.

2.4.3.3 Ordinary DC machines

The ordinary DC machine uses brushes to magnetize and alternate the magnetic field in the rotor, the brushes will cause mechanical friction which will cause lower efficiency and give off dust which must be vented away. The DC machine comes in different configuration. Separately magnetized, universal machine and shunt magnetization.

2.4.3.4 Reluctance machine

The reluctance machine is a mix between the induction machine and a synchronous machine, when it starts it behaves in a similar way to an induction machine, when it is up to speed it will run at the synchronous speed. If it is loaded too much it will fall out of the synchronous speed. The rotor in the reluctance machine is constructed so that it will try to align itself so that the magnetic flow can pass as easily as possible, the stator of the machine will apply the magnetic field that will turn the rotor (Alfredsson, et al. 2002).

2.4.3.5 Comments

Table 3 shows guideline values for torque per weight of different type of machines.

Table 3: Guideline torque per weight for different machines

Machine type	Permanent magnet	Induction machine	Switched reluctance
Nm/m^3	28 860	4170	6780

(Ehsani, Gao and Gay 2004)

A machine with high efficiency is preferred. Compare the two extremes, a machine with zero and one with 100% efficiency. In the case of regenerative braking the machine with zero efficiency will act as a disk brake, aside from the mechanical wear depending on its construction. The machine with 100 % efficiency will recover all available energy so it can be stored and then used later while accelerating. Table 4 shows some guideline values for efficiencies of different types of machines.

Table 4: Guideline efficiencies for different machines

Machine type	Permanent magnet	Switched reluctance	Induction machine	Direct current
Efficiency (%) (Machine only)	97	94	90	80
Efficiency (%) (Machine with electronics)	90	85	84	78

(West 1993)

Some types of electric machines use brushes. Brushes will wear and produce dust, which must be vented out or it can build up inside the machine and later cause a fault. Debris or water can come in through the venting device and damage or short circuit the commutator (Hughes 2006).

The permanent magnets suffer from some problems; the magnets can be demagnetized if they are not handled properly. If the magnets in the motor are heated above a certain temperature they will start losing their magnetization (Cheng 1989). If the magnets are demagnetized the machine will have a poor efficiency and not operate as expected. Any metal that comes in contact with the permanent magnets sticks to them, if metal dust comes in contact with the permanent magnet it will stick to it and can be hard to remove. Machines utilizing permanent magnets do not suffer from this problem.

2.4.4 Conclusion

A high rpm brushless permanently magnetized motor is preferred. It does not have any mechanical commutation which gives better reliability. It has the highest efficiency and the highest torque per volume of the compared motor types. The chosen electric machine will define the motor control system.

2.5 Electric energy storage

This section describes the background of the choice of energy storage.

2.5.1 Theory

In 'Introduction to Electric Circuits, 7th Ed.' (Dorf and Svoboda 2006) a capacitor is described as two conducting plates and a dielectric material between them. Each conducting plate has a terminal to the rest of the circuit. The electric charge is stored on the conducting plates which creates an electric field between the plates where the energy is stored.

The book 'Vehicle Propulsion Systems' (Guzzella and Sciarretta 2007) explains the super capacitor (See Figure 13). It uses different materials and there are other physical processes involved than in a conventional capacitor. The dielectric material of a super capacitor is an ion-conducting electrolyte and because of the physical characteristic of the electrolyte the voltage over the capacitor is limited to two to three Volts. To have a high storage capacity with low voltage the capacitance is increased by increasing the surface area and decreasing the thickness of the electrolyte.

The surface area increases by using electrodes of a material with high specific area. These electrodes are connected to the conducting plates and the electrodes are separated by a separator which is an insulating, ion-conducting membrane.

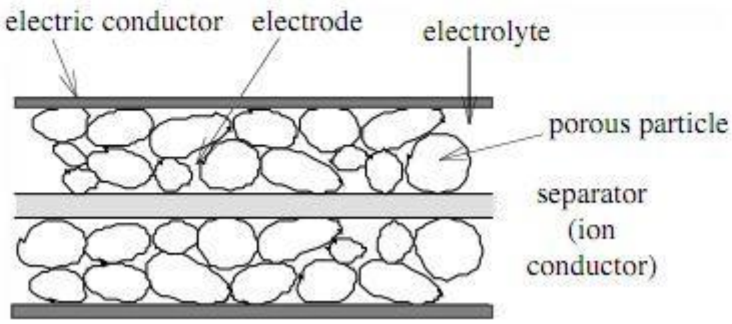


Figure 13: Super capacitor

The unit capacitance is Coulomb per Volt [C/V] and is called Farad [F]. The capacitance measures the ability to store energy in the form of separated charge or an electric field.

The capacitance C is obtained from the formula:

$$C = \frac{\epsilon A}{d} \text{ [F]}$$

where ϵ is the dielectric constant, A is the area of the conducting plates and d is the distance between the plates.

The formula for the energy W stored in a capacitor is:

$$W = \frac{CV^2}{2} \text{ [J]}$$

where C is the capacitance and V is the voltage (Dorf and Svoboda 2006).

2.5.2 Rules

The rules concerning the electric machine are valid also for the energy storage (see 2.4.2 Rules).

The rules (Rules 2012) determine the energy storage to be covered and shielded from fire and damage. The cover may be of several layers and one layer must be UL94-V0 classified. The cover must also be able to keep the energy cover in position at a retardation of 20 g. All rules can be found in section B19.13 in (Rules 2012).

2.5.3 Alternatives

This section describes different solutions for storing electrical energy and the information source is Vehicle Propulsion Systems (Guzzella and Sciarretta 2007).

There are two ways of storing electrical energy in a vehicle, batteries and super capacitors. The energy storage can be charged in different ways; with the vehicle either being plugged into a wall socket or the electric machine can regenerate energy when braking. In a series hybrid the generator charge the energy storage with energy from the combustion engine. This is also possible in a parallel hybrid but then the procedure does not provide maximum efficiency. The energy storage is then discharged and the electric machine uses this energy to provide torque to the vehicle. The most important parameters for energy storage are specific energy, specific power, lifetime and cost.

shows a short comparison of some important properties with guideline values of super capacitors and batteries. For the application of this project, charge and discharge time must be short, because the acceleration and deceleration will be higher than for a standard road vehicle.

Table 5: Comparison of some important properties of super capacitors and batteries

	Super capacitors	Batteries
Specific energy (Wh/kg)	4	113
Specific power (W/h)	1500	174
Charge during 5s (W/h)	1550	330
Discharge during 5s (W/h)	1417	2100
Lifetime (year)	10-12	2,5-5

(Khaligh and Li 2010) (Ceraolo, et al. 2009)

The batteries are electrochemical and transform electrical energy to chemical energy when stored and then back to electrical energy. In cars, all batteries are rechargeable and are an amount of individual cells that are connected. The main parts in a battery are the two electrodes and the electrolyte which is a medium for the transportation of the ion between the electrodes. There are some alternative types of batteries, for example lead-acid, nickel-metal hydride and lithium ion. These batteries have different electrolyte and their electrodes are made of different material. Lead-acid is the most common battery for the low voltage systems in cars today, but for most hybrids the lead-acid batteries have too low specific energy.

The super capacitors can also be made by different materials which give the super capacitor different power and energy characteristic. The specific power is usually higher for super capacitors than batteries but the specific energy is lower. Since the specific energy is low, super capacitors are not suitable for pure electric vehicles. They are more used as an extra source of power when needed, for example during acceleration.

A combination of super capacitors and batteries is possible, but it requires more advanced controls than having just one of the technologies. Having both technologies means that the advantage of both batteries and super capacitors can be used, but it also brings disadvantages, increased weight being just one example.

2.5.4 Conclusion

A combination of super capacitors and batteries is interesting but as the controls of the energy storage would initially be more complex, a combination is not a suitable solution for the first CFS hybrid car. In the future, when the controls of the energy storage have been proven to work well, this option will become more relevant. Between super capacitor and battery, the super capacitor is the better choice for the CFS application, as the power is much higher for the super capacitor; the extra energy in batteries is not relevant because the super capacitor is able to make use of a greater amount of the regenerated energy. Using batteries would result in a few extra kilos compared with super capacitors for the same amount of saved energy.

2.6 Physical interaction

This section describes the background of the choice of transmission.

2.6.1 Theory

The transmission shall deliver the torque from the power source to the wheels in an efficient way and optimize the relations between the torque and revolutions per minute. Torque cannot be created by the transmission but it is possible to increase it by lowering the revolutions per minute.

Gear ratio is determined by the relation between the teeth on sprockets or gears. They are therefore determined by shape. For non-teeth transmissions the gear ratio is determined by the relation between the sizes of the pulleys.

Efficiency, low weight, reliability and low increase of rotating inertia are important factors to have in mind when designing the transmission.

2.6.2 Rules

The FSUK and FSAE rules (Rules 2012) regarding the transmission do not include any details, but there are still things to consider. All of the components must lie within the surface defined by the top of the roll bar and the outside edge of the four tires, due to B 8.14.1.

First of all it is stated that any transmissions and drivetrain may be used.

All exposed high-speed final drivetrain equipment must have scatter shields to avoid bits and pieces from causing damage in case of failure. The shield must cover the chain or belt from the drive sprocket to the driven sprocket. It must end parallel to the lowest point of the final sprocket/pulley.

The materials and dimensions of the scatter shields are also determined in the rules. They differ a bit whether a chain or a belt is used.

There should also be Finger Guards to protect the parts that spins while the car is standing still. The material may be light ant withstand finger forces. It must prevent the passage of a 12 mm diameter object from passing through.

Adjusting of belts, chains and clutches are allowed after the car has passed the technical inspection.

2.6.3 Alternatives

The physical interaction is divided into different functions. These are described in the following subchapters.

2.6.3.1 Configuration

A key part of the project concerns how the components are integrated together. Since the decision to have a parallel hybrid powertrain was previously made, the main focus is how to integrate the electric motor in the powertrain. In a parallel hybrid the engine is driving the wheels directly which constrains the engine mounting. It is also desirable to be able to run the car without the electric drive in case of an electrical failure and this configuration is known as a mild hybrid.

The basic options for the electric machine position are listed below:

- Before the gearbox
- After the gearbox
- Directly on the differential
- Directly on the rear axle
- At the front wheels

In the following text, advantages and disadvantages of the different mounting ideas are listed and discussed to reach a conclusion on the most suitable solution for the CFS application.

Electric machine before the gearbox

If the machine is mounted before the gearbox there are two ways to achieve this setup. Either the machine is placed in between the ICE and the gearbox or it can be located at the opposite side of the engine, mounted directly on the crankshaft. A benefit is that proven technology can be used in the rest of the drivetrain with this configuration. If a permanent magnetic machine is used, this mounting strategy has an advantage since it is possible to move the car in neutral or with the clutch depressed when the car is switched off.

Alternatively, since the engine and gearbox come in a package, it can be difficult to mount an electric motor in between them. Mounting on the crankshaft is another option but can be difficult if the crankshaft does not extend to present a sufficient mounting surface.

The gearbox itself is a potential weak point in this setup, since the torque will increase which may cause failure to the gearbox. Since regenerative braking is integrated this could lead to an increased risk of fatigue related problems due to the quick changes of the force direction.

Electric machine after the gearbox

The machine can be mounted after the gearbox directly on the output shaft. It is still possible to get a gear ratio if the connection is made with chain or belt and different sized sprockets or pulleys. This method was employed successfully by the Lund University Formula Student team, LURacing (LUR4 Hybrid 2012) on their hybrid car in 2010.

This way of mounting can be adjusted a lot more than in the method suggested in section ‘Electric machine before the gearbox’ to fit in the frame. It also leaves the differential and rear axle unmodified.

Electric machine directly on the differential

Mounting the electric machine directly on the differential will have roughly the same characteristics as the method discussed in ‘Electric machine after the gearbox’. The most significant difference is the difference in gear ratio, which will demand a lower rpm, therefore requiring a larger machine.

Electric machine directly on the rear axle

Another way of mounting the electric machine is to mount it directly on the rear axle. This could be done by using two smaller machines on each side of the differential. This could also give the ability to have electronic traction control for each wheel. In hub wheel motors could be used, but they have the disadvantage of increasing the unsprung mass.

Electric machine on the front wheels

All of the other mounting strategies involve the rear axle which has benefits in terms of mounting and the acceleration performance, but a disadvantage is the ability to recover brake energy. On most vehicles the majority of the braking force is applied to the front wheels. This would give a higher amount of recovered energy to use in the acceleration phase. The four wheel drive would also result in increased traction.

In hub wheel motors could be used, but would increase the unsprung mass and affect the road properties negatively. But on the other hand it could, with good control systems, be possible to control the separate wheels and instead get better road properties. There is also a possibility to use two smaller motors mounded in the frame and connected via drive shafts to the wheels, just like a front wheel drive car. This could also provide traction control when cornering.

This technology is used on race cars today, such as the Toyota TS030 LMP1 and the Porsche 911 GT3 R hybrid (Lindner 2011).

The disadvantage of this configuration for the CFS vehicle is the increased complexity since the front of the frame and the front suspension would require a new configuration.

2.6.3.2 Couplings

There are not only different locations to mount the electric machine in the driveline; there are also a number of alternative technologies available in order to do perform physical connection. In some ways the chosen coupling depends on where the electric machine shall be mounted in the driveline.

Technologies that will be considered are listed below:

- Planetary gear
- Sprockets and chain
- Belt and pulleys
- Prop shaft and differential

Planetary gear

A planetary gear is commonly used in hybrid vehicles and is often known as the “Power Split Device”. It has benefits when it comes to allowing the ICE to turn independently to the wheel speed. It will act as a continuously variable transmission (CVT) which is preferred for a road car.

The losses are approximately 3 % which is a good efficiency, but a disadvantage is the complexity and ability to find suitable a component. The weight might also be a disadvantage. This application is however more common road going vehicles, as opposed to hybrid race vehicles.

Sprockets and chain

On the current CFS vehicle, sprockets and a chain are used to transfer the torque from the gearbox to the rear axle. The use of sprockets and a chain could be adapted when the electric machine is mounted in the drivetrain.

It is a well-proven technology that is already used in today’s CFS team for connecting the output shaft to the differential.

Figure 14 Double sprocket

One way of interact both the ICE and the electric machine to the same differential is the use of a double sprocket (Figure 14).

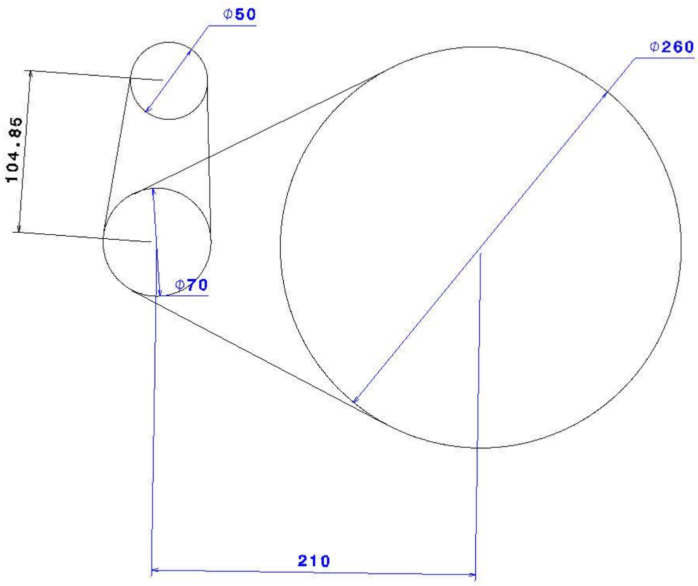


Figure 15 Double sprocket layout

The layout could then look like in Figure 15 where the double sprocket is the one connecting the two chains. A benefit with this system is the possibility to disconnect the electric machine and still be able to run the car without any further modifications. On the downside is the increased inertia. The numbers in the picture refers to the CFS10-car.

Belt and pulleys

A belt and pulleys could be used in a similar configuration as the sprockets and chain. Either a V-belt or a toothed belt is suitable in this application. The V-belt will always have minor losses due to micro slip and that may cause problems when accelerating and using the regenerative braking system.

That is not a problem with a toothed belt, which is commonly used as timing belt and also instead of a driving chain on some motorcycles. A pulley is shown by Figure 16.

Figure 16: Pulley

A benefit of the belt-driven transmission is that it gives a slightly more smooth power transfer than a chain. The belt is also lighter than a chain.

Shafts

This is mainly thought of as a way to mount motors for the front wheel in the frame, rather than in the rear. The setup would look roughly as on a normal front wheel drive car with constant velocity joints to allow suspension travel and steering.

For the front of the car it would give the benefits of not increasing the unsprung mass in the same way as in-wheel motors would. It could even reduce the unsprung mass by making it possible to mount the brakes in the frame.

As for the rear drivetrain it would be feasible to use shafts to connect the electric motor.

2.6.3.3 Electric machine as a chain tensioner

A solution for the integrating of the electric machine in the drivetrain would be to place it in line with the gearbox output shaft and the differential. They would then be connected to the same chain and drive the differential together. This kind of layout is similar to that of the conventional fan- and timing belt arrangement used in car engines.

It would have the benefit of not adding additional chains and sprockets. The only extra sprocket would be the one required on the electric machine. This would reduce the rotating inertia and also minimize the added weight. Another advantage is the fact that the differential mount can be changed to a fixed position rather than adjustable since the tensioning of the chain can be carried out using the electric machine. The electric machine mount could be carried out with slots to move the machine, like an alternator is used to tension the fan belt on many car engines. An additional method is to use springs that are dimensioned to hold the motor in place and make sure that the chain tension is correct.

The disadvantage of this configuration is that the transmission depends on the electric motor being present. Since it is desired to be able to run the car with the electric motor removed, this is a disadvantage. A potential solution for this is to make an adaptor that consists of only a sprocket and holder that fits in the motor mount. If the electric motor then needs to be removed, the adaptor is installed. It is also crucial that the forces involved are calculated carefully, since the part of the sprocket that is engaged is limited and there is a risk of premature wear and damage of the sprocket which could occur.

2.6.3.4 Release clutch

In order to allow the electric motor, the engine or both to free wheel when not delivering torque, a clutching device is required. This could have advantages in reducing the fuel consumption by reducing the rotating inertia from parts that are not producing power.

In this application the largest advantage of a one-way clutch would be to release the engine during deceleration in order to let the electric motor do all the braking. This could be achieved either by using a separate clutch mechanism or using the built-in clutch.

A solution for this is the use of a sprag clutch that transfers torque only one way. This is used in a number of different applications such as helicopters, in order to allow for auto-rotation in case of an engine failure and on starter motors on bikes instead of a bendix gear. Chalmers have used clutches from Suter racing in previous years. The basic function of the sprag clutch is shown in Figure 17 (Vogel n.d.).

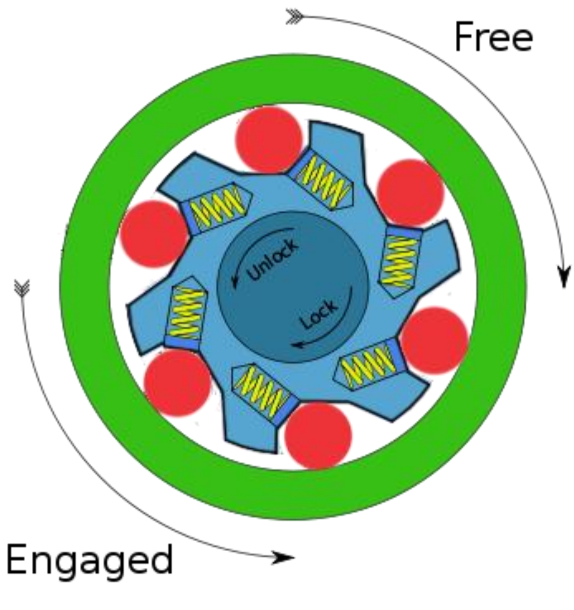


Figure 17: A clutch that allows torque to be transferred in one direction

The advantage of using a separate sprag clutch is that it will act automatically just the way it is. No programming or activating by the driver is needed. A downside with adding extra components to the system is that the reliability of the system will decrease. If the sprag clutch would fail the drive from the engine would disappear and that is not desirable.

If instead the built-in clutch is used to release the engine there are a both benefits and disadvantages. Firstly the clutch has to be activated either by the driver or automatically. To make the system stable and consequent it would be necessary to exclude the driver and make the system controlled by the electronic system. If the clutch mechanism is powered with computer controlled pneumatics the solution is mainly software orientated. This means that the system has the benefit of not having to add additional components. There is also a possibility of engaging the ICE to allow for engine braking if for example the super capacitors are fully charged. This can be done to avoid that the behavior of the car gets unpredictable.

Losses will occur since the gearbox will turn and the risk of additional wear on the components of the clutch is present.

2.6.6 Conclusion

The choice of the components is a dynamic process where all decisions depends on each other, and picking individually good parts will most likely not give the best solution. It will in that case be a sub-optimized drivetrain.

The most important factor is of course performance, but factors such as reliability (the car needs to complete the race) and manufacturability (the car must be built) are also important to finish a race.

Newton’s second law is important to have in mind during this work since racing is all about acceleration.

$$F = ma \text{ [N]} \;=>\; a = \frac{F}{m} \text{ [m/s}^2\text{]}$$

The basic ways of increasing the acceleration is to keep the force, F, high and the mass, m, low. Therefore a light transmission with small losses and low rotating mass is desired in this application.

For this application the benefits of having the electric motor connected to the output shaft of the gearbox is greatest. This allows for an electric motor with relatively high rpm and therefore low weight since the final drive reduction can be used. Also, it will not cause any extra stress on the gearbox. Having the engine in standard form is good in terms of spare parts and reducing sources of failure.

Sprockets and chain will be used to connect the electric motor to the output shaft from the gearbox, since they are strong, lightweight and allows for an extra reduction in the gear ratio. On optimal gear ratio will be calculated and the sprockets will be chosen after that.

It will still be possible to run the car without the electric motor which is desirable.

2.7 Regenerative braking

One of the advantages with an electric machine is that it can act both as motor and generator. During acceleration energy is taken out of the energy storage and used to spin the electric machine and contribute to the propulsion. Additional to this, during deceleration the electric machine can be used for braking and act as a generator which convert the kinetic energy to electric energy and store in the energy storage.

This allows the car to use the braking energy for acceleration later on. Regular brakes reject the rotation energy into heat which can be seen as losses.

There is a maximum torque that can be put onto the wheels, positive for acceleration and negative for deceleration. Due to this it is not possible to use all of the moving energy of the car. The wheels will start slipping at some torque. Calculations of this are made in 3.2.2 Maximum torque on rear wheels while braking.

2.8 Control system

This section describes the background of the choice of control system for the hybrid powertrain.

2.8.1 Theory

There is no simple step by step method which ends up with ”the only reasonable design” of a hybrid electric vehicle. Compromises in performance will have to be made, and depending on where and how the vehicle will be used. However, the basics of the common control strategies are quite general:

- Use regenerative breaking to store as much kinetic energy as possible
- Use the electric motor as much as possible instead of the internal combustion engine during acceleration
- Operate the ICE at higher efficiencies
- Disable the ICE when it is optimal to use only the electric motor
- Downsize the ICE

The design can though primarily focus on either to obtain the lowest fuel consumption possible or to obtain the best possible performance. In road going vehicles, the aim is always a combination of these targets to achieve both higher performance and lower fuel consumption. Likewise, the aim for a formula student vehicle is to get the lowest possible fuel consumption, without compromising the performance. The power to weight ratio has to be equal or higher than conventional formula student cars. Dynamic performance of the vehicle has to be equal or improved.

The complicity of controlling the different elements of a hybrid powertrain causes that the control system has to include several control units. The functions of the control system are illustrated in Figure 18.

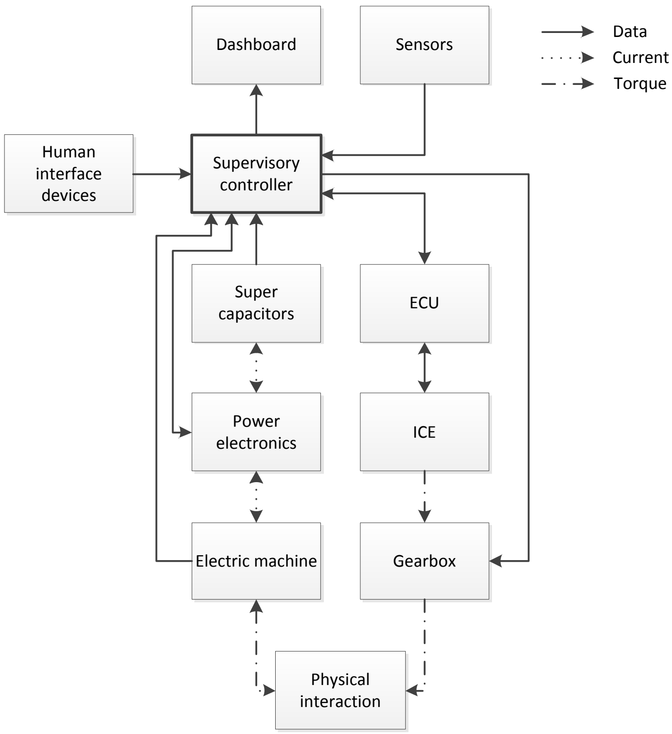


Figure 18: Overview of the control system functions

The different parts of the control system must communicate with a central control unit, called a supervisory controller. The control system of this project is mainly focused on the supervisory controller, the ECU, the communication between all elements and control strategies. Electrical safety takes priority in all decisions in the project. Unfortunately, the safety of the control system is mainly managed by the programming of the components which will not be processed.

2.8.2 Rules

The rules concerning the electric and control system are numerous however below the most important are listed. For the full specification, see the FS rules (Rules 2012).

- Drive-by-wire systems which control the power delivered to the wheels are allowed for alternative powertrains
- There must be at least two completely independent systems to shut off power due to the throttle pedal being released or the brake pedal pushed, see the FS rules (Rules 2012)
- A tractive system master switch which disables the tractive system must be fitted to allow work to be done on other systems on the vehicle
- Any components (e.g. Electronic throttle or regenerative controls) carrying high voltage must be mounted outside the driver's cell area unless separated from the driver by a firewall

2.8.3 Alternatives

The communication between the control units can typically be undertaken via serial communication or via Controller Area Network (CAN) bus communication, but all previous CFS cars are using serial communication. The reasons are that it is easy to implement and that serial communication is preferred if the systems consists of few subsystems.

2.8.3.1 Serial communication

The car is divided into different nodes and the components can only communicate directly with the node they are connected to. If a component called component *A(1)* is connected to node *A* wants to communicate with component *C(1)* connected to node *C*, the communication must go via both node *A, B* and node *C* to reach component *C(1)*. This is illustrated in Figure 19. Therefore, all the nodes must contain a microprocessor to handle the data flow.

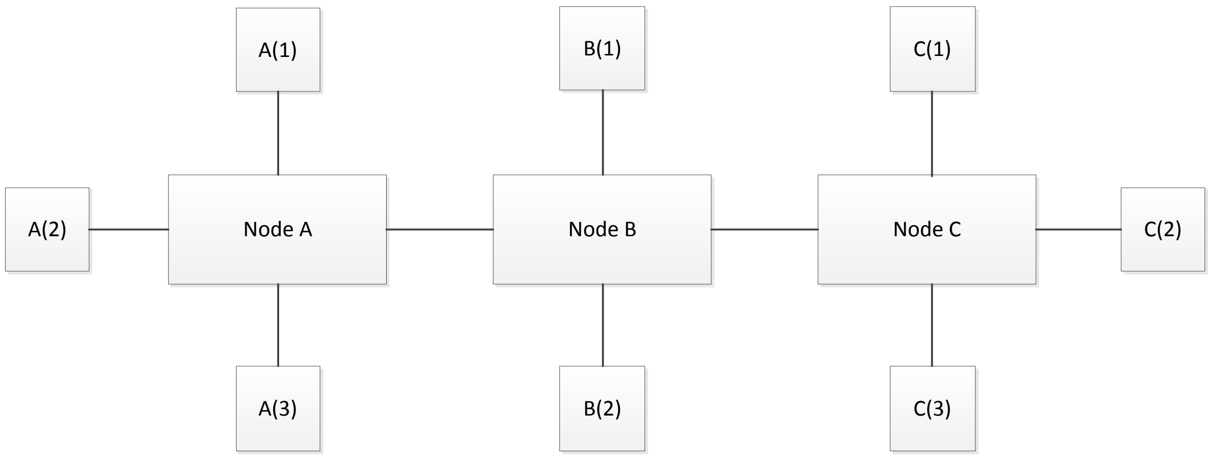


Figure 19: Serial communication

2.8.3.2 CAN communication

CAN is a vehicle bus standard which has been used by the automotive industry for almost thirty years. The purpose with the CAN bus is to allow microcontrollers and other devices to communicate with each other without the requirement of a host computer. Aside from the automotive industry, it is also being used in other areas such as industrial automation and medical equipment.

A modern automobile uses several electric control units for various subsystems such as powertrain, transmission, Anti-lock Braking System (ABS), cruise control, electric power steering, airbags, audio system, mirror adjustment, windows, door lock, battery systems, climate control, seat control etc.

Each node connected to the bus is able both to send and receive messages, but not simultaneously. The messages consist of an identifier, which represents the priority of the message. The data is transmitted serially onto the bus.

The devices that are connected to a CAN bus are typically actuators, sensors and other control devices. The devices are connected through a host processor and a CAN controller. All the devices can communicate with each other through the bus, as illustrated in Figure 20Figure 20.

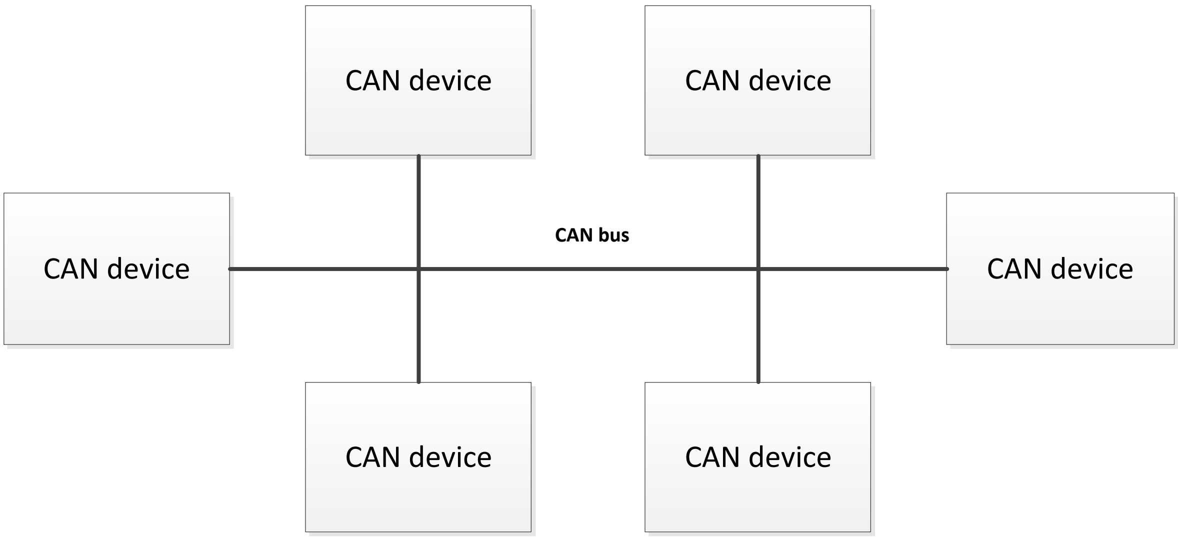


Figure 20: CAN communication

Any node may begin to transmit if the bus is free of use. If more than one node begins sending messages at the same time, the message with the more dominant identification will overwrite the other nodes less important messages. The most dominant message remains and is received by all nodes. This function is referred as priority based bus arbitration.

Each node requires a host processor, CAN controller and a transceiver. The host processor translates received and sent messages. Control devices, sensors and actuators can be connected to the host processor. The CAN controller stores the data serially from the bus until an entire massage is available. This is both for sending and receiving. The transceiver converts and adapts the signals between the CAN controller and the bus.

2.8.3.3 ECU

There is a wide range of manufacturers who can deliver a reliable ECU. However, there is one manufacturer who can deliver a system which seems to be the optimal system for CFS. CFS used NIRA ECU's designed for four cylinder engines since 2009. However, NIRA also have an ECU for one- and two cylinder engines, called NIRA i2x. There are many advantages of using the NIRA i2x:

- In house experience from previous formula student teams
- Integrated CAN communication
- Drive by wire
- Low size and weight

2.8.3.4 Supervisory controller

In a hybrid vehicle, a supervisory controller must determine how to distribute the power and torque between the ICE and the electric motor. The purpose is to satisfy the power demand of the driveline in the most convenient way. The main objective of the optimization is the reduction of the overall energy use. The supervisory controller must meet certain requirements. It must have

- Satisfactory I/O options in order to gather relevant information from sensors and actuators
- Support for CAN communication
- Sufficient processing power for data processing
- Reliability
- Low weight
- Low size
- Low power consumption
- Good data logging possibilities
- Affordable price

Two different manufacturers products can be used for our application.

dSPACE MicroAutoBox

The product is used for automotive applications such as chassis control, powertrain, body control, electric drives control etc. It is widely used by the industry for applications similar to CFS. It is a very powerful real-time system that operates without user intervention, just like an ECU. The special strengths of the hardware are for example high performance, extensive I/O options, robust design, reliability, advanced software etc.

Table 6: Requirements dSpace

Requirement	Value
I/O requirements	Satisfied
CAN	Satisfied
Performance	Satisfied
Reliability	Satisfied
Weight	Not completely satisfied
Size	Not completely satisfied
Power consumption	Satisfied
Data logging	Satisfied
Affordable price	Not completely satisfied

National Instruments cRIO

The tool is an integrated system that combines a real-time processor and a reconfigurable field-programmable FPGA within the chassis. It is mainly used as control and monitoring system within the industry. It is thus not primarily intended for automotive applications. The product consists of a chassis with an integrated processor and FPGA and up to eight I/O modules. There are many advantages of choosing exactly which I/O to use. The key strengths of the hardware are for example high performance, excellent I/O options through modular design, robust design, reliability, advanced software, FPGA etc.

Table 7: Requirements National Instruments

Requirement	Value
I/O requirements	Satisfied
CAN	Satisfied
Performance	Satisfied
Reliability	Satisfied
Weight	Satisfied
Size	Satisfied
Power consumption	Satisfied
Data logging	Satisfied
Affordable price	Satisfied

2.8.3.5 Torque controller for the electric machine

The torque control for the electric machine can be managed in two ways. Either, the power electronics can contain the regulator, or the supervisory controller will include the regulator and will send all the control signals to the power electronics. The choice will be based upon the specifications of the chosen power electronics.

2.8.4 Conclusion

In this section conclusions regarding the specifications for the control system will be described.

2.8.4.1 Communication

As a number of devices will be utilized which have to communicate with each, the use of CAN bus communication is preferred instead of the conventional serial communication. The increase of data rate also affects the choice of communication type. An additional consideration is the products available on the market for this application. Many components which are suited for the application are built for CAN communication. The automotive industry is widely using CAN communication so it is a natural step to change communication method. Finally, the knowledge about CAN communication is significant at Chalmers which can ease the development phase. These factors combined suggest that CAN bus communication is the appropriate method of communication for this application.

2.8.4.2 ECU

Due to the earlier mentioned reasons, Nira i2x is chosen as ECU. CFS has in cooperation with a company gained a good relationship to Nira so the choice of ECU manufacturer is natural.

2.8.4.3 Supervisory controller

The choice of supervisory controller was not as straightforward as the ECU choice. Both the dSPACE and the NI alternatives are suitable for our application. The softwares are also quite similar however, there are some key differences between the products. The NI products are physically smaller and have lower weight than the dSPACE MicroAutoBox. The modular I/O is also an advantage, as is the FPGA. A partnership with NI is also preferred. They can provide us with components in a more advantageous manner than the dSPACE retailer. Likewise, NI can give us better educational and product support. Due to earlier mentioned reasons, the National Instruments cRIO is chosen as supervisory controller.

2.9 Challenges

This subchapter concludes the benchmarking and presents the greatest challenges while going into the developing of the hybrid powertrain.

A single-cylinder engine was chosen which replaces the current four cylinder Yamaha Fazer engine. Since the behavior of the engine with the mandatory 20 mm air-restrictor is not known and the engine is not in house, a model for simulating the performance of the engine has to be made. The results will used for calculations on the performance of the powertrain.

To work out the best compromise between power, weight and efficiency for the electric system was the greatest challenge. Determine the most suitable energy capacity and voltage for the super capacitors. To design the system so that it has highest possible efficiency and is lowest possible weight. Do calculations on the system to get a hint of how the electrical system will work. To design an electrical system that maintains the performance of the vehicle and lowers its CO_2 emissions.

The transmission and physical interaction cover areas such as getting the both the power sources to transmit torque to the wheels in an efficient way. During braking and deceleration the electric machine should regenerate as much energy as possible and therefore should the ICE be released. The added inertia should be kept low. There was also an intention to be able to run the car on the ICE only, so it should be possible to disconnect the electric machine in just three minutes and remove it in one hour.

The challenges with the packaging were basically to arrange the positioning of the components in such a way that they all are able to cooperate and have enough space. Just as an example the differential could not be positioned arbitrary since the drive shafts must be able to connect it to the wheels. All components would also have to be placed into the frame and be positioned and safe there in scenarios of both driving and collision. Finally the rules define limitations and following these became a challenge as well.

The most important challenge regarding the control system was the optimization of the driving cycle. It was very hard do determine how to use the electric machine in the most convenient way as possible and to minimize the energy demand based only upon the current energy in the super capacitors, the input from the driver and the measurements from the other sensors in the car. To get all components to work well together in a satisfactory way is also a very complex task.

3 Design

Early in the project there had to be decided whether the hybrid powertrain should be a parallel, series or split hybrid. (These are described under 1.1.1 Hybrid systems)

The powertrains where analyzed under different aspects. These were performance, ICE rpm, weight, flexibility, reliability, simplicity and efficiency. These had different importance. Performance and weight were seen as the most important and got a high weight factor. Also simplicity got a high factor because of the size of this project. Reliability was also seen as important because if the hybrid system collapses the car still has to be able to run. Efficiency was important due to the part of the competition where fuel consumption is measured. ICE rpm is directly related to the emissions and fuel consumption but was not weighted that high since it does not have that much influence on the powertrain. Also flexibility was seen as not that important because the car is a race application and does not need that much flexibility.

The hybrid solutions were compared with the pure ICE and obtained a value between -3 and 3 to show how much poorer or better the system is. To get the comparable result the grade and the weight factor were multiplied and finally all values were summarized. An overview of the result is shown by the Pugh matrix in Table 8.

Table 8: Pugh matrix

		Parallel		Parallel (front wheels mounted el. motor)		Serial		Split		ICE
Performance	**3**	3	**9**	3	**9**	1	**3**	3	**9**	0
ICE rpm	**1**	0	**0**	0	**0**	1	**1**	1	**1**	0
Weight	**3**	-1	**-3**	-1	**-3**	-2	**-6**	-2	**-6**	0
Flexibility	**1**	1	**1**	1	**1**	0	**0**	2	**2**	0
Reliability	**2**	0	**0**	0	**0**	-3	**-6**	0	**0**	0
Simplicity	**3**	-2	**-6**	-1	**-3**	-1	**-3**	-3	**-9**	0
Efficiency	**2**	1	**2**	1	**2**	1	**2**	1	**2**	0
		2	**3**	3	**6**	-3	**-9**	2	**-1**	0

The result showed that the split was too complicated and did not give much more than the unneeded flexibility. The series hybrid system was not suitable for the application since it weighted too much because of bigger and heavier electrical actuators. It also has a large dependence of the electrical parts and would not be able to drive without them. The positive aspect of the series hybrid is that it is possible to run the combustion engine on optimum speed and lower the fuel consumption, but the negative arguments poised, see Table 8. The heaviest was that the vehicle would very much depend on the electric system (about which the knowledge is limited). Therefore the best hybrid system would be a parallel hybrid. It is able to use the braking energy for boosting at high accelerations and the car is able to run without the electrical parts of the hybrid system. The weight may be higher but alternatively the engine size can be reduced. This benefits both fuel consumption and weight. Whether the hybrid system would be separate for front and rear wheels (e g combustion engine rear and electrical motors in front) was not decided at first. Later it was realized that using the front wheels for driving would be difficult and the weight distribution would not be very good. Therefore it was decided to go for a parallel hybrid system with the electrical motor close to the combustion engine.

3.1 Solution model

After the benchmarking was completed the functional model had to be remodeled so that the functions could be switched to components. Some functions could be accomplished by the same component. This model would be called the solution model (Figure 74).

The solution model shows that for example the electrical part of the hybrid system performs functions for both generating torque from electric energy and vice versa. There are, as described in 2.6 Physical interaction, many ways of connecting the electrical motor within system. The same chapter also described why the motor was placed after the gearbox.

Why there are arrows in both directions between some boxes is simply because the energy or fluid is flowing in both directions – at some places simultaneously and at other places at different times depending on whether the car is accelerating or decelerating.

The control system is part of the electronic system but this project concerns this anyway as mentioned before.

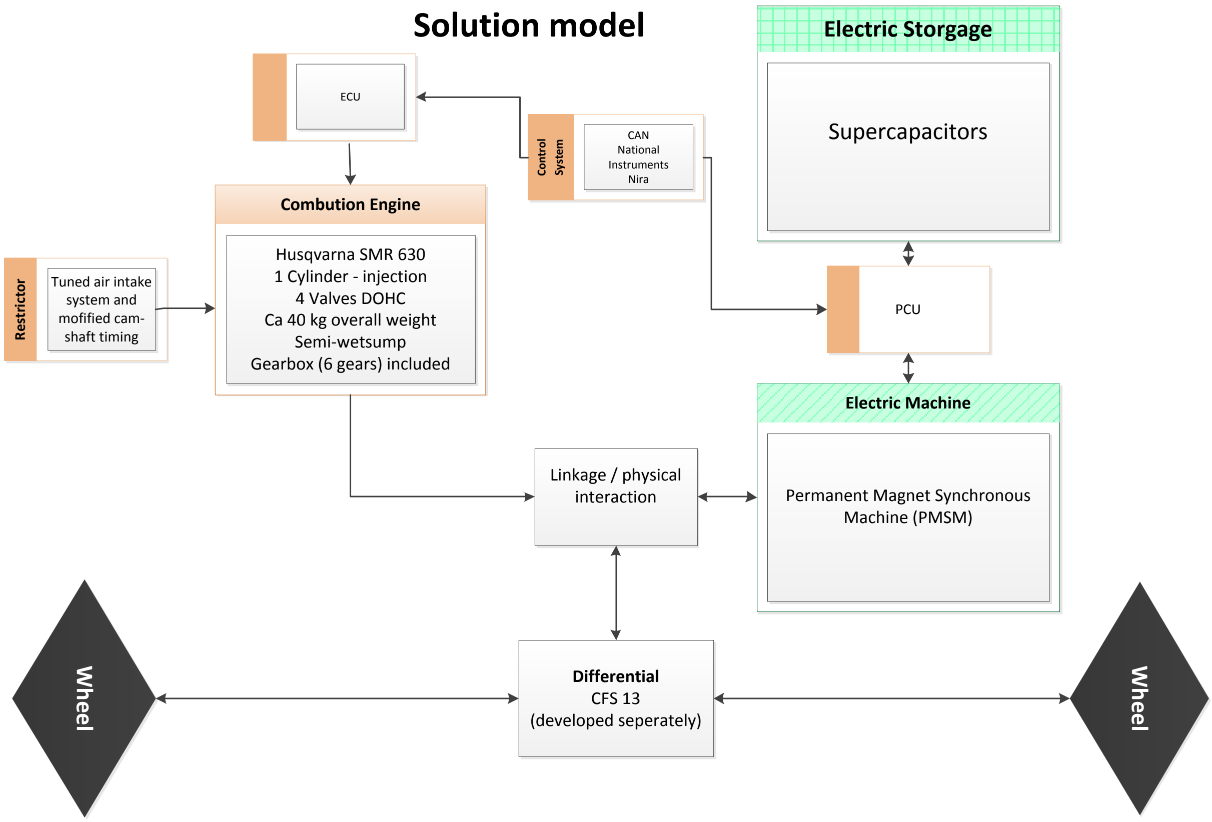


Figure 21: Solution model

Chapter 3 Design describes the solutions for the different subsystems in detail.

Figure 21 shows an overview of the whole chapter. It is a solution model with more specific components than Appendix K Solution model, which shows the functions.

3.2 Design targets

To have something to work toward design targets were established. These were not taken out of nothing but grounded in calculations, experiences from former cars, measured data and last but not least the rules of the competition (Rules 2012). The design targets are then used to dimension and design the system. The values should not be seen as absolute but minimum or maximum values so if the real value is better than the target value of course that is good. This has to be in parity with cost and reasonability. The design targets are attached in Appendix E Design targets.

3.2.1 Maximum torque on rear wheels while accelerating

This section concerns the calculations about the maximal torque on the rear wheels during accelerating and decelerating.

There is a maximum torque possible to put on the rear wheels. The friction does not allow the torque to be infinite because the wheels will spin.

The mass of the car was assumed to be 220 kg and the driver's weight 80 kg. The weight distribution is rounded to 45 % on the front wheels and 55 % on the rear wheels. Gravitation acceleration is 9.82 m/s^2.

Björn Pålsson[6] gave the coefficient of friction 1.5 which was derived through given by testing of former formula student cars. Race tires have very good grip characteristics, especially when the friction heats them up during the race.

The radius of the wheel is 0.25 m.

The friction force is given by:

$$F_{fr} = \mu F_N \text{ [N]}$$

The torque is then calculated by:

$$T = F_{fr} r_{wheel} \text{ [Nm]}$$

This torque is the torque out of the system, after gearing and inertia losses etc. as shown below.

It should be mentioned that all values are approximate and therefore give approximate answers.

$$m = 220 + 80 = 300 \text{ [kg]}$$
$$g = 9.82 \text{ [m/s}^2\text{]}$$
$$R = 0.267 \text{ [m]}$$
$$\mu = 1.5$$
$$L = 1.62 \text{ [m]}$$
$$L_f = 0.55\, L$$
$$L_r = 0.45\, L$$

[6] Björn Pålsson, Institution of Applied Mechanics, Chalmers University of Technology

$$N_r = \frac{L_f}{L} mg = 1620 \text{ [N]}$$

$$T = \mu N_r R = 649 \text{ [Nm]}$$

This value, 649 Nm, is valid for the car standing still. That is not the case because when the wheels start rotating the car will start moving and the center of gravity will move. Foad Mohammadi[7] was consulted and he introduced the braking calculations from their car. The same height of the center of gravity, h_{cg}, as in 2011's CFS car is used in the following calculation.

$$h_{cg} = 0.32 \text{ [m]}$$

The acceleration, a, is given by logged data from previous year's car. It was then maximal 5,8 m/s^2 but is here increased to 6 m/s^2.

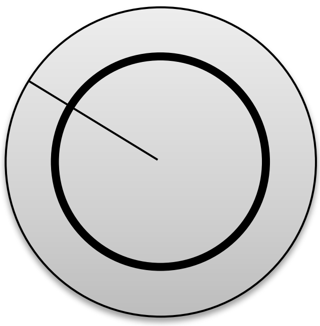
$$a = 6 \text{ [m/s}^2\text{]}$$

The maximal torque that is possible to put on the rear wheels while accelerating is

$$T_{ar} = \mu R m \frac{L_f g + h_{cg} a}{L} = 790 \text{ [Nm]}$$

Hence slightly more than without respect to the movement of the center of gravity.

Inertia should also be considered, for example there are losses in the wheels and shafts while they start rotating. The following calculation shows briefly the losses in the wheels, where the largest contributing factor to losses are. Shafts have a much shorter radius and therefore do not affect as much as the wheels. Consider the mass of the wheel to be placed 2/3 out from the wheel center. Figure 22 illustrates this by the inner black circle. The mass of one wheel is 4 kg and there are 4 wheels.



,

Figure 22: Mass of the wheel

$$m_{wheel} = 4 \text{ [kg]}$$

[7] Foad Mohammadi, Chalmers Formula Student Team 2011

$$J_{wheel} = m_{wheel} \left(\frac{2}{3}R\right)^2 \text{[kg m}^2\text{]}$$

$$\dot{\omega} = \frac{a}{R} \text{[rad/s}^2\text{]}$$

$$T_{loss,wheels} = 4\, J_{wheel}\, \dot{\omega} = 4\, m_{wheel} \left(\frac{2}{3}R\right)^2 \frac{a}{R} = 11.4 \text{ [Nm]}$$

Hence, a very small number compared with 790 Nm and for the shafts it will have less influence and as these calculations are an approximation it was decided to use the values as guidelines and maximize the torque instead of finding an exact value.

A short notation about acceleration calculations should also be made. There is actually no need of calculations about how much torque or energy that is needed to accelerate the car to a certain speed within a certain time. Since the calculations above were made the mission was to maximize the torque without over-dimension and get a larger torque than needed. Though, the effect is shown by the measured data from CFS-11 (CFS 2011).

3.2.2 Maximum torque on rear wheels while braking

To see how much torque that is available during braking some calculations have been undertaken. This torque can be used to charge the super capacitors (see section 2.7 Regenerative braking).

These calculations are very similar to those in section 3.2.1 Maximum torque on rear wheels while accelerating – only the sign within the parenthesis is changed. T_{br} and T_{bf} is the braking torque on the rear and front wheels respectively. The braking acceleration is set to the maximal measured value:

$$a_{1.5} = 1.5\, g \text{ [m/s}^2\text{]}$$

$$T_{br1.5} = \mu R m \frac{L_f g - h_{cg} a_{1.5}}{L} = 299 \text{ [Nm]}$$

$$T_{bf1.5} = \mu R m \frac{L_r g - h_{cg} a_{1.5}}{L} = 880 \text{ [Nm]}$$

For the softer braking acceleration a_{b1} = 1 g the torques are:

$$a_1 = 1\, g \text{ [m/s}^2\text{]}$$

$$T_{br1} = \mu R m \frac{L_f g - h_{cg} a_1}{L} = 415 \text{ [Nm]}$$

$$T_{bf1} = \mu R m \frac{L_r g - h_{cg} a_1}{L} = 763 \text{ [Nm]}$$

The most interesting values are 299 Nm and eventually 880 Nm in case the car would have had front wheel mounted motors. Unfortunately this is beyond this project so the number that is used is 299 Nm for the rear axle. This shows the difference between the front and rear wheels. There is a large amount of torque available on the front axle which is wasted without regenerative braking there.

299 Nm is the available torque on maximum braking that can be used for charging the super capacitors. Hence this is the value used in the MatLab simulation for the entire electric system which is described in 3.4.1 MatLab simulation. The electric machine would not have to deliver nor brake this torque since there is a gear which changes the torque-rpm- ratio.

3.2.3 Powertrain performance

This section summarizes chapter 3 Design in the meaning of performance of the new drivetrain. More detailed information is found under respective section.

Section 2.3.3 Alternatives introduced the power-to-weight ratio to which will be referred here again. The CFS-11 car has a power to weight ratio of 0.29 kW/kg considering a car weight of 217 kg without driver. The Husqvarna SMR 511 Engine has an approximated overall weight of 40 kg and the electric system (both PMSM motor and super capacitors) has an approximate weight of 15 kg. That means, the hybrid powertrain will lower the car weight about 30 kg to 187 kg. Consequently, the powertrain has to produce about 55 kW for an equal power to weight ratio. The reduced overall mass of the one-cylinder car lowers the center-of-mass of the car significantly. The lower center of gravity height and the additional power from the electric machine can improve the car handling. Thus a minor overall power deficit can be neglected and a slightly lower power-to-weight ratio is acceptable. An engine power of 40 kW is set as target value.

With the knowledge about the power of the combustion engine the gear ratios and CFS-11 car specifications such as wheel ratio, the traction force from the powertrain to the road can be demonstrated by means of acceleration as function of the speed of the vehicle. This allows making reasonable and data driven statements about vehicle power and acceleration capabilities.

From Newton’s law of motion

$$F_{net} = m_{car} a \text{ [N]}$$

The current acceleration a can be obtained by simply dividing the net force of the powertrain by the mass m_{car}. This mass is set to the sum of vehicle mass, engine mass, driver mass and mass of all electric parts.

The net force is defined as the amount of power which is actually available for driving. That means it is the brake power of the engine[8] less the power needed to overcome rolling resistances and aerodynamic drag, which are considered to be “small” for a race car.

$$F_{net} = F_{brake} - F_{resistances} - F_{drag} \text{ [N]}$$

All losses and resistances can be expressed as a lower net force which can act on the car.

The acceleration is limited by the traction of the wheels. Slip occurs as produced torque during acceleration is higher than the tires can manage. For this study, the acceleration value of CFS-11 was set as reference. In 2011, CFS won the acceleration event by clearing the 75 m acceleration run in 4.661 seconds, or in other words: the CFS-11 car had a maximum acceleration of 6.9 m/s^2.

The relation between angular speeds from the engine and the actual car velocity can be computed with the definition of gear ratios and the physical relation of in line sprockets, The vehicle speed v_{car} is limited both by the performance of the engine and in this case by the reference value which is again taken from CFS-11. Logged data shows that during the endurance run a maximum velocity of 95 km/h occurred.

[8] The term ‘gross power’ is avoided in order to avoid any misunderstandings as this term usually appears in connection with engine power characteristics.

The Husqvarna workshop manual (Husqvarna Motorcycles S.R.L. 2010) shows the arrangement of the gearbox and provides information about the specific gear ratios. Figure 23 shows schematically the introduced arrangement of the powertrain.

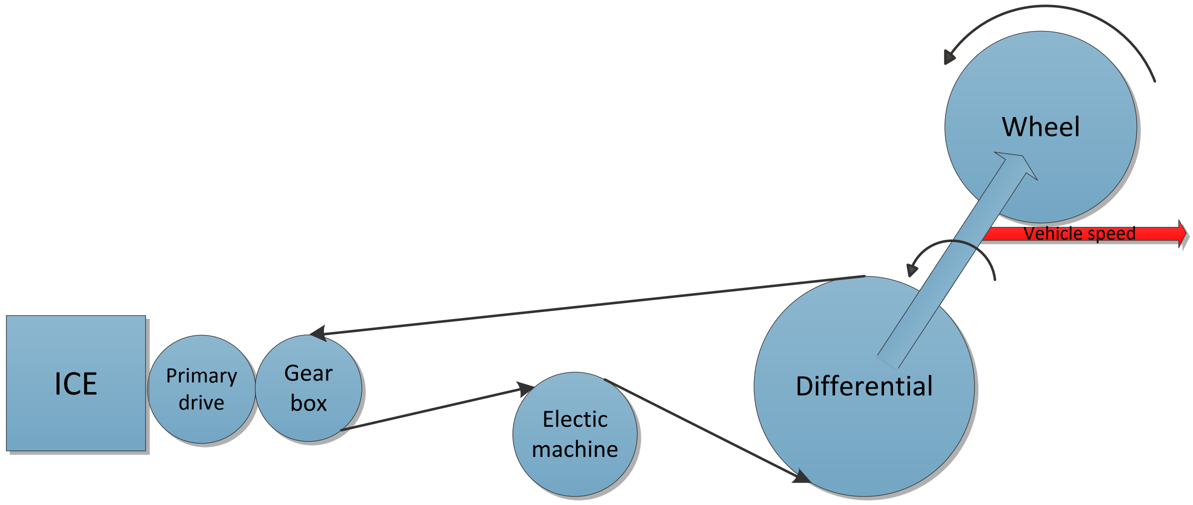


Figure 23: Gear arrangement

The speed of the chain is exactly determined through the engine speed and thus, the speed of the electric machine can be obtained.

The available power of the electric machine is a function of its speed and the SOC of the super capacitors

$$P_{el\,machine} = f(N_{el\,machine}, SOC) \text{ [W]}$$

In this model it is assumed that the SOC is sufficient high at any time so that the electric machine is able to provide its maximum power. The overall gear ratio between electric machine and wheel is assumed to 10. The power of the electric machine was set to 15 kW and thus slightly lower than it actually is. This assumption contributes to a small extend to the consideration of overall losses in the powertrain.

This model is only a very rough first approach which may only shows tendencies. It represents only the case of positive vehicle speed and acceleration with WOT and gives no information about braking maneuver. The real improvement might be smaller. The start-up procedure is not taken into account. This depends to a large extend on the driver as he decides the intensity and time for using the clutch. During clutch slipping the available engine power is less.

The result of these calculations is given in section 3.5.3 Traction force and Appendix M MatLab code.

3.3 Combustion engine

This section concerns the design and simulation of the combustion engine.

3.3.1 Design of components

The comparison of the chosen Husqvarna SMR 630 engine to the functional model in Figure 5 shows which parts have been adapted for the assembly of the engine into the hybrid powertrain. Figure 24 shows an adapted functional model.

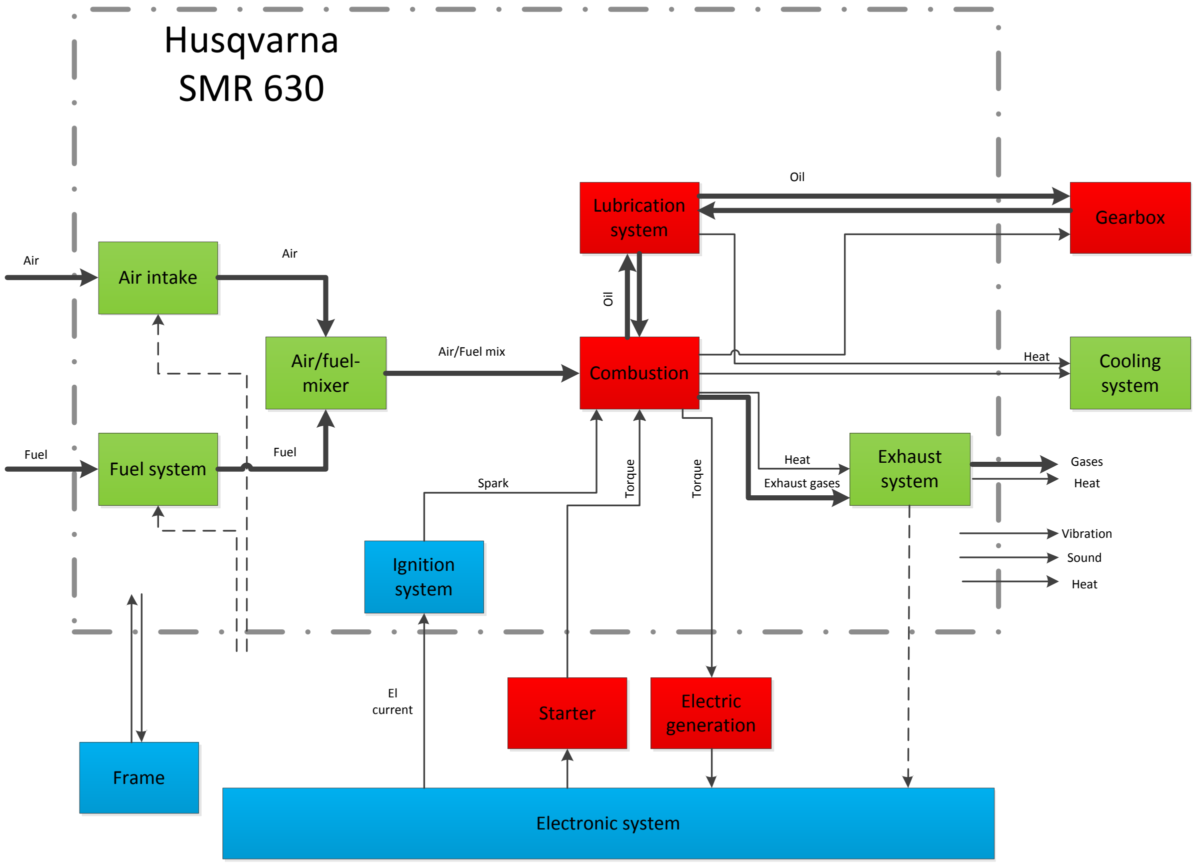


Figure 24: Adapted function model Husqvarna SMR 630

- The red boxes show parts or systems of the engine which are either not modifiable such as the gearbox or no modification is needed since the original equipment can be taken.
- All parts which needs a modification either for establishing rule compliance as introduced in 2.3.2 Rules or for packaging issues are represented in green color. A typical example is the air intake system which has to be equipped with the air restrictor or cooling and exhaust system. Probably both will need to be adapted so that the engine can be mounted in the frame.
- The blue parts represent all systems which are not part of the design face of the combustion engine. This is the frame, the electronic system of the hybrid powertrain which also controls the ignition system.

3.3.1.1 Intake System

The rules dictate the sequence of the components in the air intake system to throttle body – air restrictor – intake runner – engine. This means that the stock engine equipped injector throttle body shown in Figure 25 (Husqvarna Motorcycles S.R.L. 2010) is not conform to the rules.

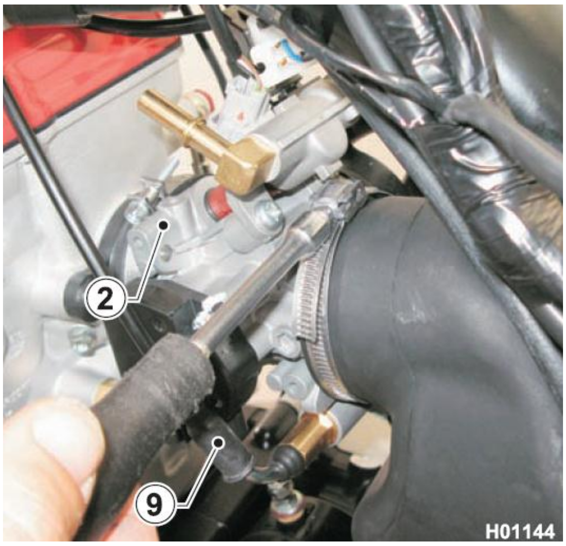


Figure 25: Original injector throttle body

From a fluid mechanical point of view, the restrictor acts like a partially closed throttle which generates a vacuum in the inlet plenum (McClintock, et al. 2008). Considering standard engine operating conditions, a vacuum is generated while the piston travels from top dead center down to bottom dead center. Hence air is pulled from the intake manifold into the combustion chamber. Without restrictor the air has the possibility to flow freely through the intake manifold to the cylinder. However the restrictor limits the cross section within the manifold and due to the narrow section, the air speed increases until it reaches sonic speed which is the limiting factor of the air mass flow rate. This phenomenon is called Venturi effect and the limitation of the volumetric flow rate is called chocking. In a chocked restrictor an increase in pressure difference does not result in an increase in mass flow (Meekhof 2011). Thus an intake system which inducts the maximum amount of air must be developed which improves the amount of charge within these regulations. The leading strategy is to reduce the pressure loss as the air flows through the intake runners. Under WOT conditions; turbulence phenomena can be observed which may impair the engines throttle response. Further parameters characterizing an intake system are discussed later on in this section.

Due to the cyclic nature and thermodynamic phenomena during the combustion progress in a combustion engine, pressure waves are generated. It is more likely that these pulsating pressure waves, both on the intake and on the exhaust side, interfere. In case of a destructive interference, the amplitudes of the pressure waves reduce themself which would lead to a more steady flow of the fluid[9]. Steady flow is preferred but more laborious to produce in a single-cylinder engine.

Therefore it is common to use a plenum as storage for air in the intake system which acts similar to a capacitor in an electric circuit. This may have the availability of holding the pressure of the air at

[9] The term fluid is chosen here as a more general description of the flows both at the intake and exhaust side.

constant level and helping to overcome the pulsing effect of the cyclic pressure waves. This leads to a more steady flow.

The intake runners are a very sensitive part of the intake system. According to (Heywood 1988) and (Bell 1998) longer pipes provide higher volumetric performance and therefore more torque at the low-mid range engine speed and short pipes at higher engine speeds. This may be regarded as a rule of thumb. When the intake valve opens (closes) pulsating pressure waves are formed and travel through the intake runner. Depending on engine speeds which is determining the duration for valve opening and closing (but not the cam duration) and length of the intake runner, these pulses can be timed so that they arrive just as the valve closes and thus, increasing the volumetric efficient which has got a positive influence on the torque. When the pulsating wave travels from the cylinder to the plenum and then bounces back, it has gone from a low pressure pule to a high pressure pulse and as it arrives just as the valve closes it can fill the cylinder with extra needed charge. This effect is sometime described as “Helmholtz Resonant Oscillation Manifold”. If this is not tuned correctly, the pressure pulse does not have enough time to enter and exit the cylinder before the valve closes. Drawing air out which then decreases the volumetric efficiency which leads consequently to a bad impact on the produced torque.

Similar observations can be made for a second variable, the diameter of the intake runner: Based on Bernoulli's and Venturi’s principle for internal flows, the velocity of the fluid increases (decreases) with a smaller (bigger) diameter of the pipe. This increases inertia and turbulence as well as it supports the mixing of air and fuel which is injected into the intake runner. The books (Heywood 1988) and (Bell 1998) describe how these changed properties can be used to gain benefits in engine performance. The effect is called “Ram-effect” and means basically that the volumetric efficiency can be increased since unburned air-fuel mixture continues to flow into the cylinder when the piston is after TDC and is traveling up. The improved volumetric efficiency increases consequently the torque. Also the benefit of turbulence helps to increase the combustion flame front and thus decreases combustion duration as a turbulent flame spreads more rapidly than a laminar one. The downside with having intake runners with small diameter is that at higher engine speeds the smaller diameter inhibits the airflow which leads to a decreased volumetric efficiency.

This project does not cover the development of an air intake system and exhaust system due to the lack of time and the overall boundaries. However, to correspond to the goal statement of this study, a suggested intake system is taken and adapted to the Husqvarna engine. Coincidental suggestions of intake systems within the FASE rules for one cylinder engines are given by (Meekhof 2011) and (Corrigan, McCullough and Cunningham 2006). This might not gain the maximum theoretically reachable power. It is suggested to be more suitable for a one cylinder application than for the use of a CFS air intake which is developed and extensively tested for a four cylinder engine.

The throttle body is simply modified by a round pipe. The simulation will only consider the case of WOT. The intake system is adopted from (Corrigan, McCullough and Cunningham 2006) as well as Figure 26 which illustrates this system.

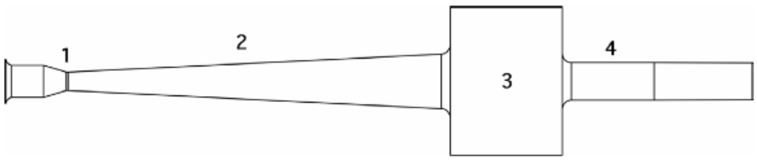


Figure 26: Illustration of the inlet system

Table 9: Numeric geometric values for the inlet system

Section	Length [mm]	Entrance Diameter [mm]	Exit Diameter [mm]	Volume [l]
1 (Restrictor)	30	40	20	
2 (Diffuser)	400	20	60	
3 (Plenum)				4.5
4 (Inlet Runner)	220	41	41	

A plenum volume of 4.5 l has been chosen. CFS-11 uses a 4 l plenum. In agreement with several studies on intake systems for FSAE cars ((Corrigan, McCullough and Cunningham 2006), (Hong, Huang and Bai 2012) and (Gilani 2012)) it can be observed that a little benefit can be gained by using plenum volumes of larger than 4 l. However the packaging regulation limits the volume for the plenum.

3.3.1.2 Exhaust

Figure 27 shows what the off-the-shelf exhaust system looks like.

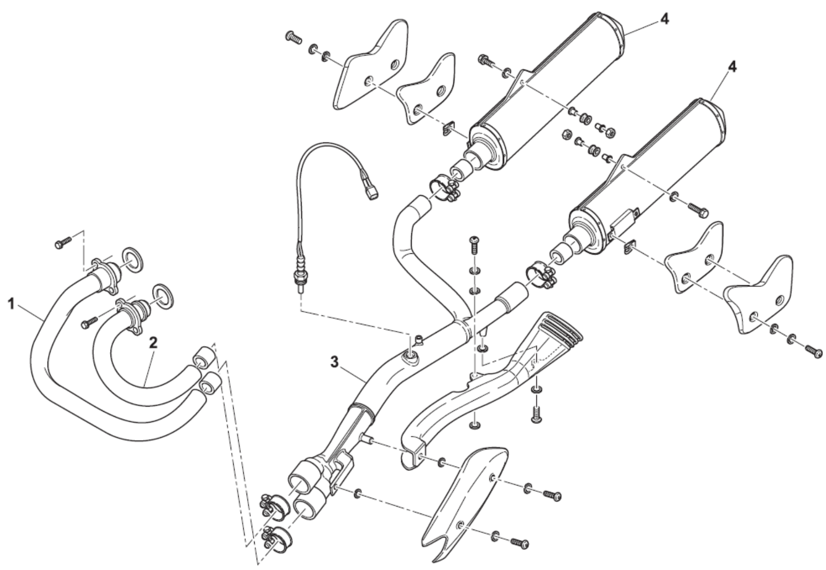


Figure 27: Original exhaust in exploded view

This reveals that this system cannot be used without any modifications. However it might be possible to adapt this system by adequate adaptions of the geometry by bending or adapting the length of the pipes. Exact geometric properties of the exhaust system are not known. A detailed research has to be carried out when the engine is in house.

The main task of the exhaust is to expel as much of the combustion products out of the cylinder as possible and preserving the amount of inducted intake charge. This is basically determined by the exhaust valve timing. The exhaust pipe geometry determines back pressure. Research studies showed that the use of a diffuser pipe shape as an air restrictor can increase maximum power and torque (Peng, Yunqing and Jincheng 2008).

In a trapped exhausts continuous wave reflection from a diffuser results in a reduced gain of entropy in comparison to a similar sudden change of area (Corrigan, McCullough and Cunningham 2006). This also results in the tuning effect occurring over a wider engine speed range.

The overall engine noise can be reduced as the flow of the exhaust gas is flowing though cavities which acts sound-absorbing.

3.3.1.3 Cooling

The original equipped liquid cooling system is shown in exploded view in Figure 28.

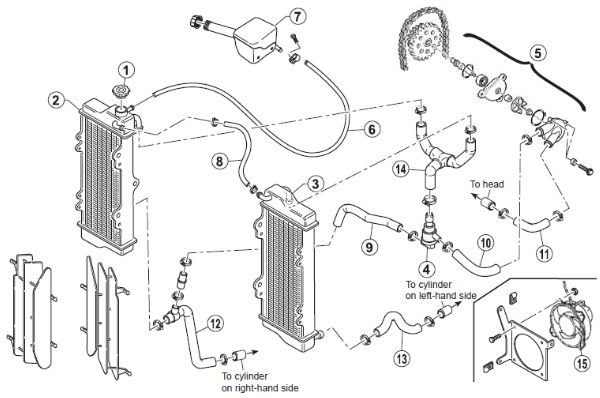


Figure 28: Original equipped cooling system in exploded view

Considering a FSAE car it becomes obvious, that this system cannot be in use anymore in the hybrid powertrain: The FSAE cars usually have a heat exchanger with radiator in one of the side boxes. Since the engine is water-cooled, it will be possible to develop an adequate system.

3.3.1.3 Engine Lubrication

A dry-sump system would be preferred but experience from the FSAE Team Kühn Racing which uses the Husqvarna TE/SMR 530 engine shows that the engine’s wet-sump system does not cause any significant competitive disadvantages (KüHN Racing e.V. n.d.).

To sum it up, the parts needed for a rule compliant air intake system are: Throttle Body, restrictor, plenum and intake runners with the injection system

3.3.1.4 NVH behavior

The engine is a main source for Noise Vibration and Harshness. The changed behavior with the new engine is taken into account. NVH influences the quality and the reliability of the entire car and the rules dictate a maximum noise of 110 dB (Rules 2012). Noise is an increasing factor to the vehicle user and environmentalists while harshness is related to the quality and transient nature of noise and vibration. In general, good NVH properties are linked to the vehicle sophistication and make it more comfortable to drive.

The internal combustion engine is manifested to be the “most dominant on-board source of vibration” (Hall 2002) In general, all unbalanced rotating parts are potential sources of vibrations. Looking at a cylinder with its moving masses, their fluctuating inertia force causes reciprocating unbalance. These forces act along the cylinder axis. In multiple cylinder engines, the cumulating forces leading to shaking forces and moments are acting on the motor block and this causes vibrations. In particular in multi cylinder engine it is possible to compensate these effects by a sensible arranging the crankshaft positions but it is not possible to remove these effects completely (Hall 2002).

Two dynamic forces appear in a single cylinder. These are on the one hand the reaction torque at the crankshaft and on the other hand a shaking force acting along the line of stroke. It is in the nature of dynamic forces, that they are both cyclic and proportional to engine speed. The mentioned torque is caused by the gas force due to its change in thermo-dynamical properties during the combustion cycle and a inertia force, based on the acceleration of the connecting rod and the piston. This causes a certain shaking as well.

In a four cylinder engine such as the Yamaha Fazer engine some of the mentioned shaking forces cancel out each other. However this is not the case in a one cylinder engine. Hence higher vibrations are expected when using a one cylinder engine and thus one has to put more effort in methods to reduce those.

A simple strategy to reduce the vibrations is trying to perform a vibration-damping to constrain the transmission of vibration to the vehicle body. Further all cavities in the intake and exhaust system acts like a sound absorber. The detailed mechanism and possibilities of these sound absorbing procedures goes beyond the scope of this study and are to be discussed separately.

3.3.2 Engine Simulation

Because the engine is not in house yet, a simulation model of the engine is required. Such a simulation is the most convenient and cost effective way of evaluating the combustion engine. Virtual Engine Simulation Tools such as GT-Power®[10], AVL Boost® or VirtualEngines® and are widespread both in industry and in research and are able to compute reasonable results. The use of an accurate and trustworthy model offers the potential of making more data driven decisions on any engine modification without costly test runs on the dynometer. Appendix G How GT-Power works describes briefly how GT-Power works.

[10] GT-Power is a part of GT-SUITE which is a product of Gamma Technologies (GTI), a software company which is solely focused on the Engine and Vehicle Industry and is commonly used for computer aided engineering of combustion engines and related phenomena.

3.3.2.1 Simulation method

For developing a reasonable and reliable model, Figure 29 shows the suggested strategy for the experimental design.

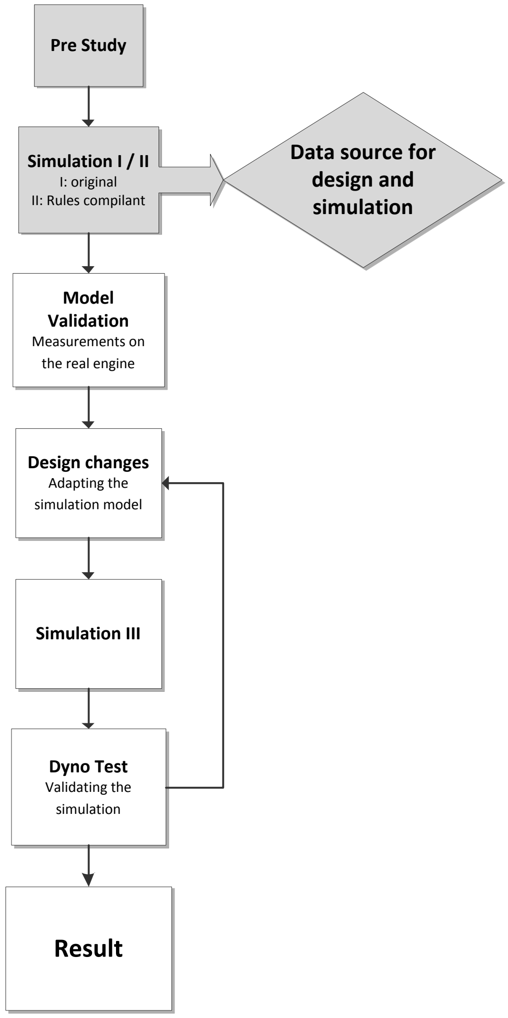


Figure 29: Experimental design of engine simulation

The pre-study face covers a systematical work with data and information about the engine given from pictures, the workshop manual and technical specifications as well as a pre-study on different strategies for increasing the performance of the engine.

The workshop manual contains various pictures which motivate the following strategy:

> As the number of parameters which characterizes an engine and influences its performance is high, a method for an effective testing is needed. The Orthogonal Array Testing Strategy (OATS) is used for designing the experiments. (Bauer 2009) describes OATS as "*It is a systematical, statistical method for testing pair-wise interactions by deriving suitable small set of test cases from a large number of scenarios. The testing strategy can be used to reduce the number of test combinations and provide maximum coverage with a minimum number of test cases. OATS utilizes an array of values representing variable factors that are combined pair-wise rather than representing all combinations of factors and levels*".

This strategy can take into account potential synergy effects between geometric parameters to a small extend. The spacing between settings is determined to adequate step-sizes; this means for example 5 – 10 crank angle degrees (CAD) for valve timing or other crank angle dependent parameters and 5 – 10 mm for geometric properties. It is also used by other FSAE Teams (Hong, Huang and Bai 2012).

The power and torque curves will be evaluated for all set ups. The set-up which delivers the "best" values will be chosen. If any obviously unrealistic result appears, for example negative power and torque or unrealistically high or low efficiencies, the values are considered to be "wrong" and are out of range.

In many cases, for example the initial valve timing, literature values suggested by (Heywood 1988) or (Bell 1998) are used and the results are evaluated. This procedure can be realized in praxis by simply relocating the cam shaft to one or two teeth on the sprocket or changing the length of the different parts.

Initial simulations are made which are eligible to indicate the engines performance with and without restrictor and other rules compliance. Following the discussion above, these simulations are the source for all data about engine power and performance for this study.

Section 3.3.2.5 discusses the necessary simulation model verification as soon as the engine is in house which delivers the required information in form of measured geometric properties.

The reliability and the quality of the adapted simulation are verified by an engine test on a dynamometer (dyno). To overcome the discussed losses through the dictate use of the restrictor, the model should now be suitable enough to evaluate different strategies for increasing the engine power.

A final dynamometer testing verifies the final result of engine performance.

The tones indicates that this study does not cover the simulation model validation and the steps following that because the engine is not in house and therefore not available for further testing by the end of the project.

As figured out in the overall boundaries the scope of the project does not cover the evaluation of different strategies for increasing the engine power by tuning. The general notes about the impact of the air restrictor on the engine performance will be evaluated with this model which gives general tendencies.

3.3.2.2 GT-Power implementation

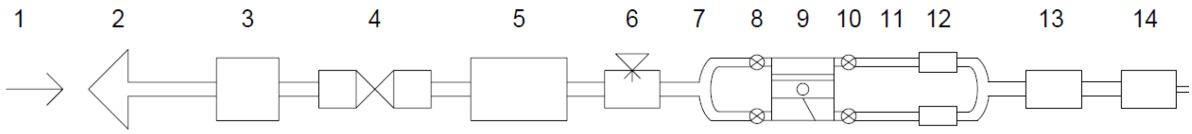
Engine simulation tools are able to compute reasonable results even if only a small amount of data is known. GT Power is a suitable software-tool for modeling an engine and the gas exchange in particular. This is of particular interest as the behavior of the engine with the restrictor is not known yet. Moreover it can show the impact of a number of different variables on engine performance and pollutant emissions.

The engine simulation focuses on the gas exchange and the produced performance. Many parameters are not known and idealizations are made. The model cannot make any suggestions about comfort issues (NVH behavior) of the engine neither can it give statements about temperatures and heat distribution for designing the cooling system.

A first model should represent the real engine without restrictor. After that the model of the engine with restrictor is added. This approach reflects also the procedure which will be done in praxis as all the modifications will be made starting from the real engine in its original state.

In the following it is described how the stock engine basically looks like and which modifications have to be done for the compliance with the FSAE rules.

Figure 30 shows the schematically arrangement of the engine corresponding to the rules which dictates the position of the throttle body.



1 Air Intake
2 Air Filter
3 Throttle body
4 Restrictor
5 Plenum
6 Intake Runner with injector
7 Intake Port
8 Intake Valves
9 Combustion
10 Exhaust Valves
11 Exhaust Port
12 Exhaust Runner
13 Muffler
14 End Environment

Figure 30: Schematically arrangement of the engine

Subsystems were formed consisting of Air intake, Intake port, cylinder, exhaust port and exhaust. The four valve DOHC[11] cylinder head is highlighted by the two pipes from the intake port block to the cylinder and from the cylinder to the exhaust port consequently.

Table 10 gives all parameters which are known about the engine and considered to be important for modeling it by using GT-Power.

Table 10: Known engine parameters

Parameter	Value
Number of Cylinders / Valves	1 / 4
Camshaft setup	D.O.H.C.
Bore [mm]	100
Stroke [mm]	76.4
Compression ratio	12.4 : 1
Valve clearance	Intake: 0.1 – 0.15 mm Exhaust: 0.15 – 0.2 mm
Cooling	Liquid

The data is taken either from the datasheet (Appendix I Husqvarna TE 630 engine data) or calculated according to these values.

[11] DOHC is the short form for Dual Overhead Camshaft and signalizes that two camshafts are placed in the cylinder head, one for each the intake and the exhaust side.

Implementation intake and exhaust port

As both ports are located in the cylinder head, the following section will deal with both the intake and the exhaust port. All relevant data has been transmitted by Husqvarna and made it possible to model the cylinder head and the valve properties as realistic as possible in GT-Power. Figure 31 shows a view of the cylinder head both from the intake and from the exhaust side.

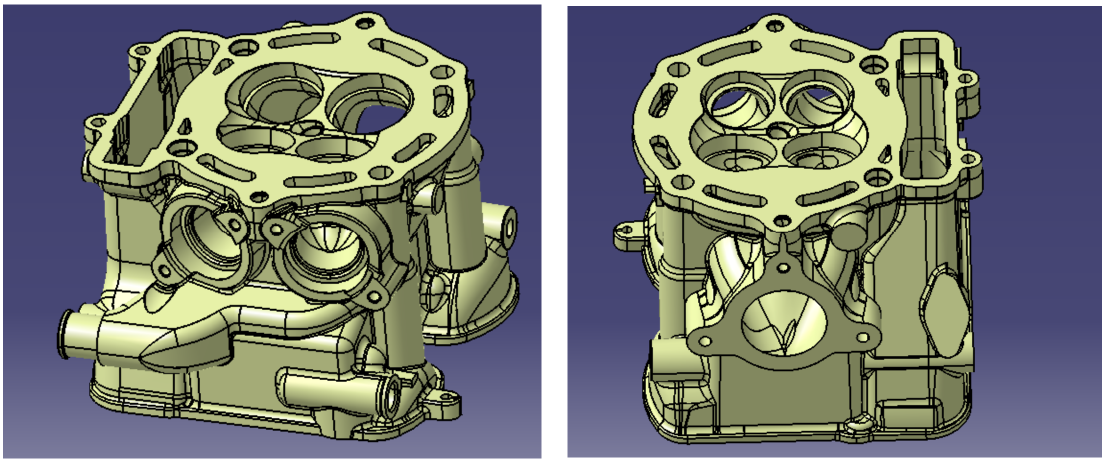


Figure 31: CAD model of the cylinder head. Left: exhaust side Right intake side

This enables one to measure relevant geometric data, given in Table 11:

Table 11: Measured relevant geometric properties

	Intake Port	Exhaust Port
Valve Diameter [mm]	36,3	30
Port length [mm]	80	50
Inlet Diameter [mm]	41	29

The valve lift profile which is implemented in GT-Power is shown in Figure 32.

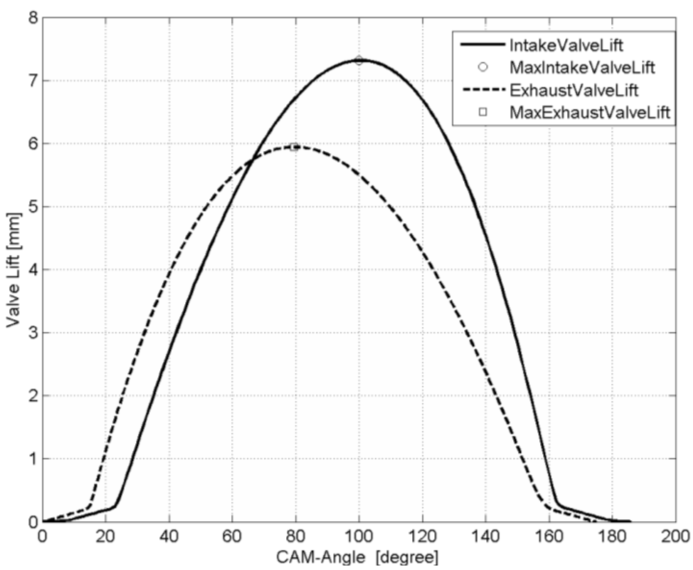


Figure 32: Husqvarna TE / SMR 630 valve lift profile as function of Camshaft Angle

It shows the typical shape as it was expected. The intake lift is higher and reaches its maximum at 100 CAM-Angle degrees with 7.3139 mm. The exhaust lift is slightly smaller and has a maximum valve lift of 5.9418 mm which appears at 79.5 CAM-Angle degrees. This data is transmitted by Husqvarna what motivates the usage of these high-precision values. From this, the valve opening duration, both in crank-angle-degrees and CAM-angle degrees is known[12] and given in Table 12.

Table 12: Measured total valve opening duration

Valve	Total duration	
	CA [degree]	CAM [degree]
Intake	186	372
Exhaust	175	350

These numbers represents only a measurement of the duration and does not contain any information about the exact valve timing which is discussed later.

GT-Power offers a number of methods for implementing the valve timing. The method chosen for this study is to refer the Cam Timing Angle to the point of maximum lift in the implemented angle array. Husqvarna did not transmit any further information concerning the valve timing.

Implementation Cylinder

The properties for the flow- and heat properties as well as the combustion object were set do default. It is not worth trying to model these properties more exactly and even the CFS team sticks to standard values and ideal models in this case.

The engine block specifies the geometric attributes of the cylinder. GT-Power models the kinematics and rigid dynamics of the crank train configuration. All important parameters are known or can be easily estimated.

Implementation Exhaust

The exhaust is simply modeled as a long, straight pipe without bending. It contains of exhaust runner, muffler and end pipe. The muffler is necessary for limiting the noise of the engine which is determined by the rules. The geometric properties of the assumed exhaust geometry in the simulation are given in Table 13.

Table 13: Geometric properties of exhaust in simulation

Part	Diameter In/Out [mm]	Length [mm]
Exhaust runner	30 / 50	250
Muffler	60 / 60	150
Endpipe	60 / 60	150

[12] The relationship between cam angle and Crank Angle (CA) is: CA = 2 * Cam Angle Degree.

Final GT-Power Model

Figure 33 shows the final implementation in GT-Power.

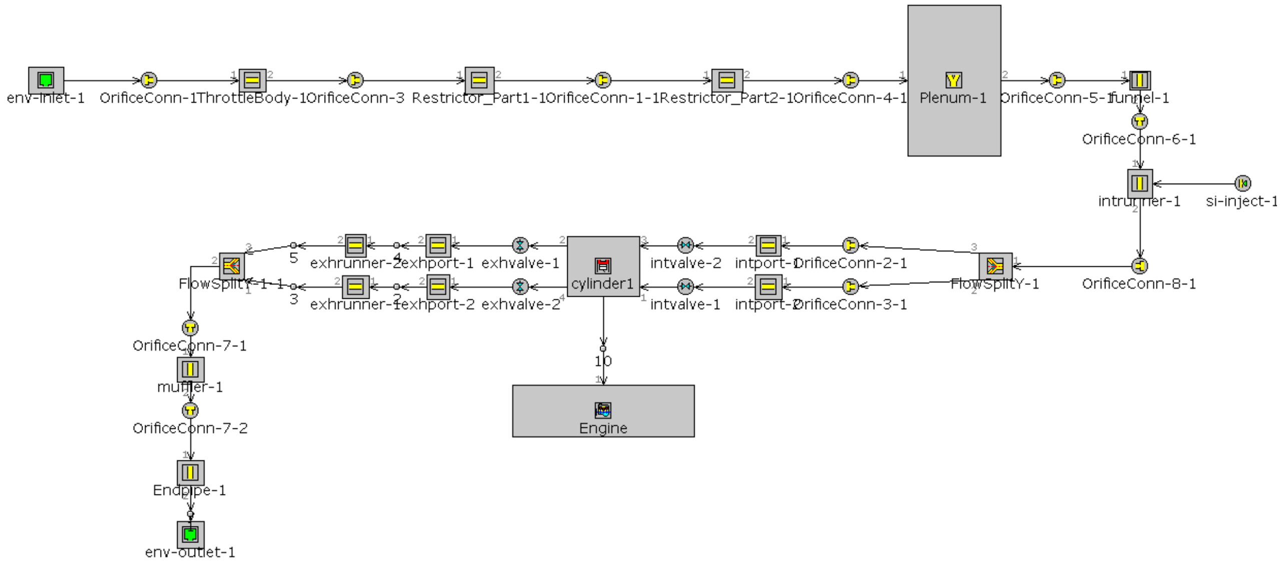


Figure 33: GT-Power set-up of intake restricted one four valve one cylinder engine

The orifices expresses that there might be no additional losses between two flow components and these forward and backward discharge coefficients are set to 1.

Referring to the introduced subsystem level, Figure 34 shows this arrangement in GT-Power:

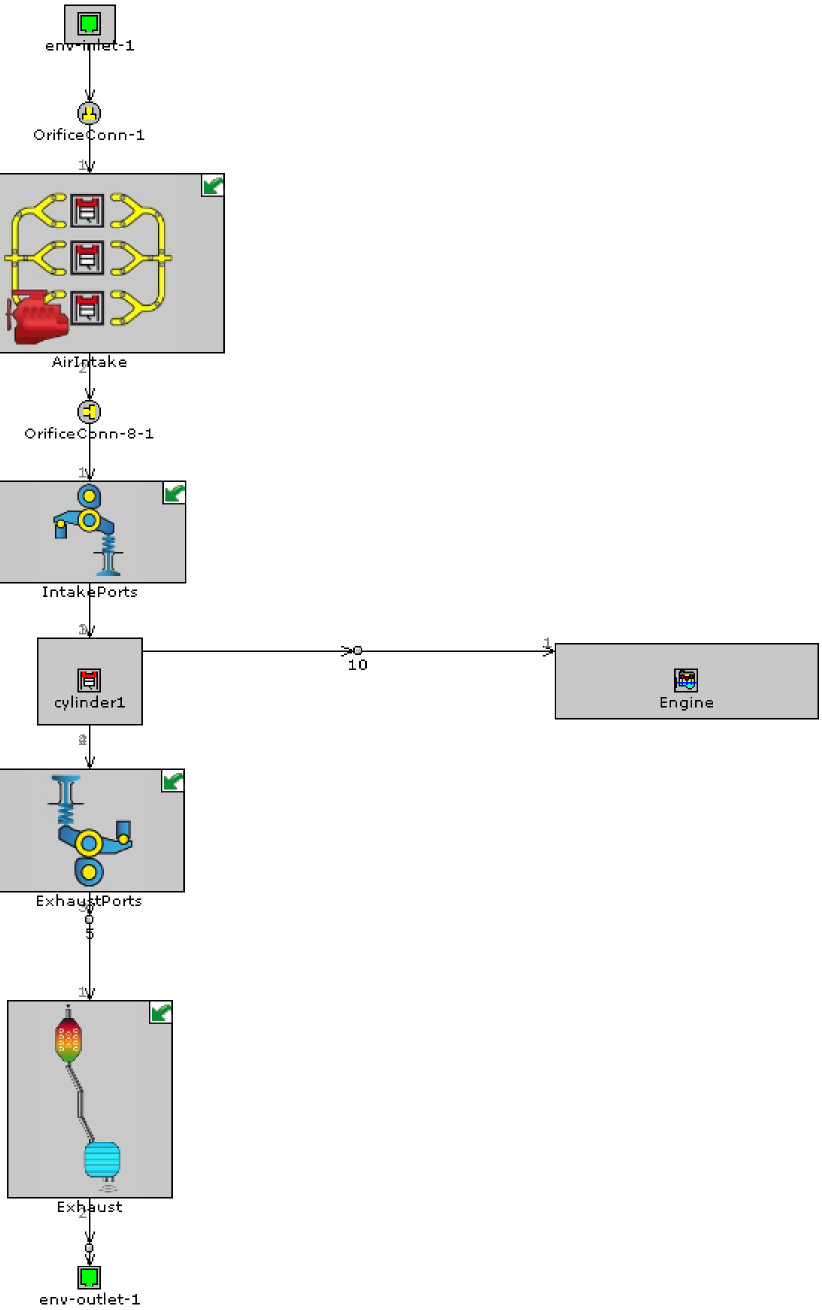


Figure 34: GT-Power Model on subsystem level

3.3.2.3 Simulation Results

The following section presents the results of the simulation, using the values which gave the best compromise in result. The guiding thinking when making compromises between the distribution of power and torque over the engine speed was to have a high level of torque even at lower engine speed and trying to generate regions with “constant” torque. This makes the car handling easier and more predictable for the driver. Furthermore it is aspired to have a certain constant stress-level on the driving chain. The complete, detailed set-up of the simulation is given in the appendix.

An engine speed range of 1500 rpm (idle speed) to 8500 rpm (maximum engine speed) is assumed as this is a common speed range for one cylinder engines due to the author’s knowledge and previous experience. The maximum engine speed might be approximately in the same range as the one from the stock engine. Reason for this is to avoid any mechanical overstraining due to higher engine speeds.

Figure 35 shows the brake torque in Nm against the engine speed.

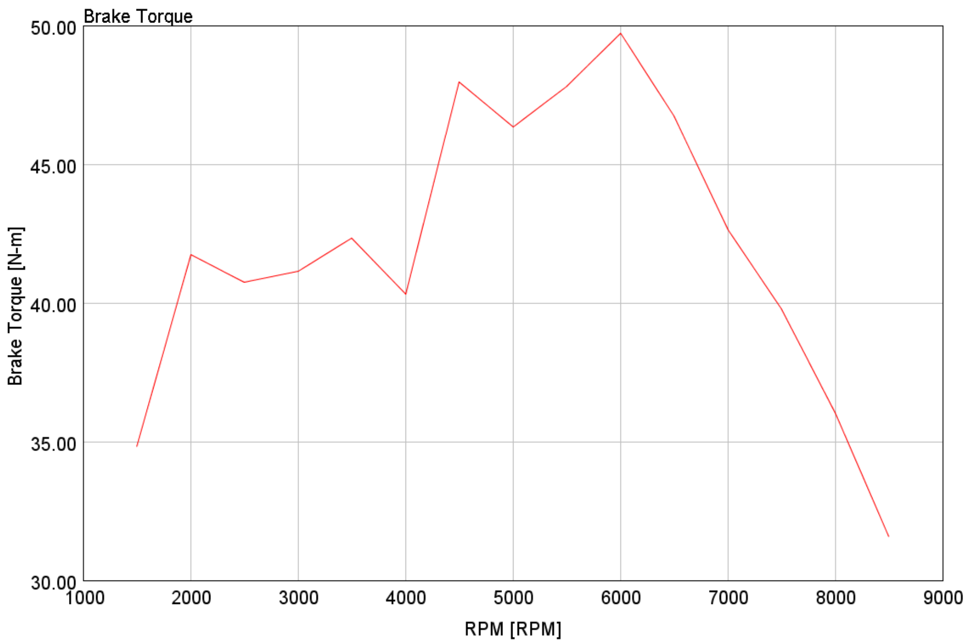


Figure 35: Brake Torque [Nm] vs. engine speed [RPM]

The simulated engine can produce a maximum torque of 49.7 Nm at 6000 rpm. A second, slightly lower peak at 48 Nm is observed at 4500 rpm. After 6000 rpm, the torque decreases rapidly.

The computed brake power based on the introduced relationship is shown in Figure 36. The engine has a maximum power of 31.8 kW which appears in the simulation at 6500 rpm. The power development from idle speed (1500 rpm) to 6000 rpm is approximately steadily ascending.

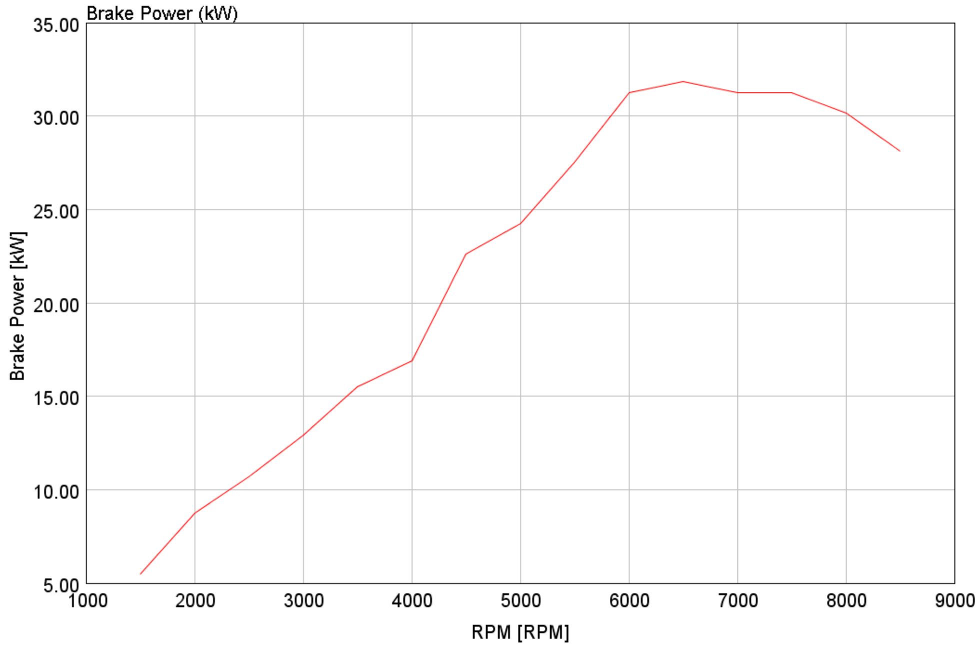


Figure 36: Brake Power [kW] vs. engine speed [RPM]

Based on the relationship between BMEP and brake torque introduced in Section 2.3.1.1, Figure 37 shows a plot of the BMEP vs. engine speed. It is in a range of 8 to 9 bar at lower rpms (1500 rpm to 4000 rpm) 10 bar for mid-range rpm (4500 rpm to 6000 rpm) and drops down to 6,5 bar at higher engine speed (6500 to 8500 rpm).

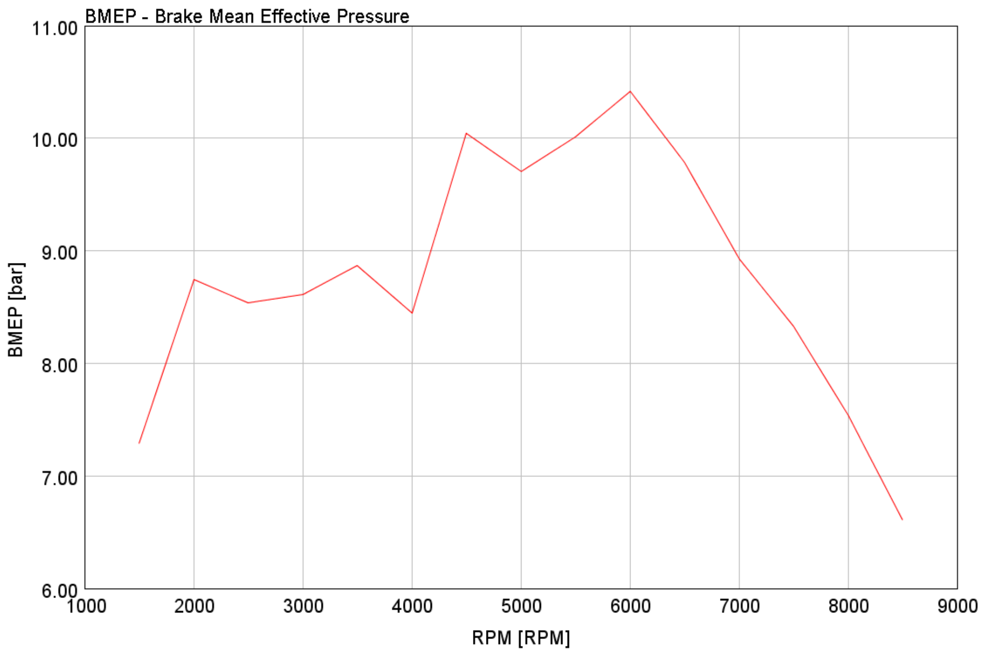


Figure 37: BMEP [bar] vs .engine speed [RPM]

Figure 38 shows the Valve Lift in mm against the Crank Angle in CAD. The exhaust valve (left) opens at 145 CAD (35 CAD BBC) and the intake valve (right) opens at 300 CAD (60 CAD BDC)

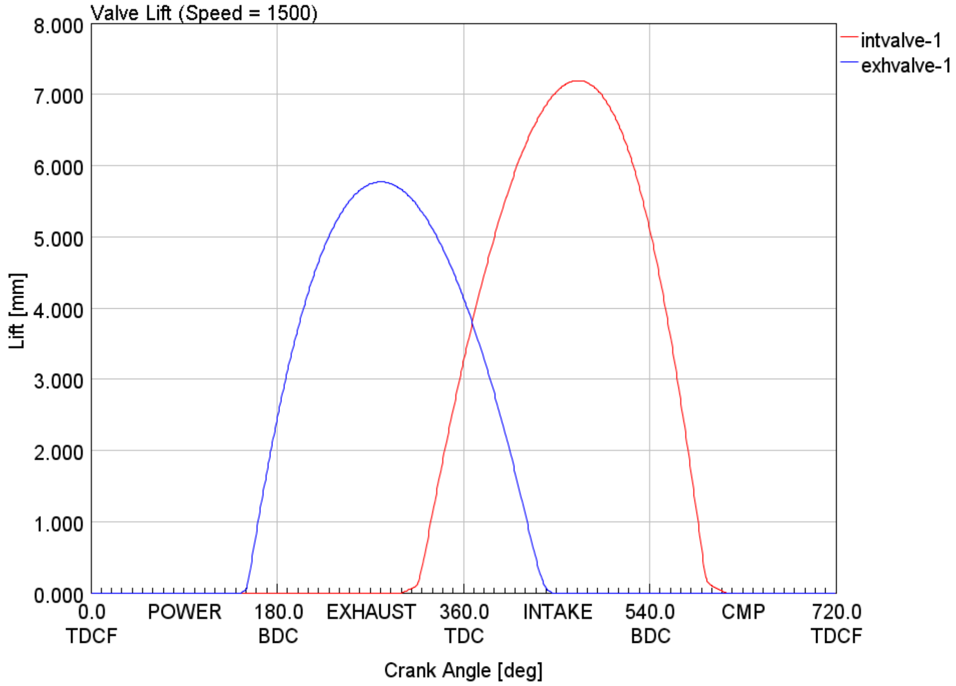


Figure 38: Valve Lift [mm] vs. Crank Angle [deg]

Figure 39 shows the volumetric efficient against engine speed. A step at 4000 rpm is apparent. The simulation computes a maximum volumetric efficiency of 94.5 % at 6000 rpm.

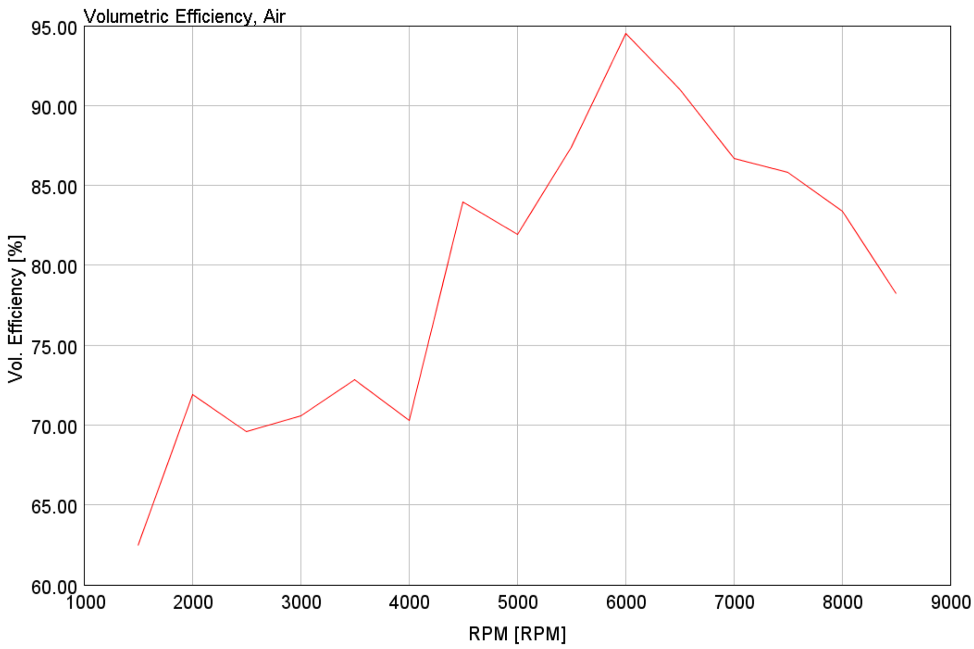


Figure 39: Volumetric efficiency against engine speed [RPM]

3.3.2.4 Discussion of engine simulation results

The representativeness of the model is limited. A lot of characteristic data about the engine is either not known or not published by the manufacturer. Literature values or ideal models of the combustion or knock phenomena were used and considered to be suitable for this model. Simplifications as for example modeling the exhaust system as only one pipe while the original has two end pipes and adopting reference values suggested by the software, can contribute to meanderings between the simulation results and the measurements on the dyno. In addition simplified flow-models of GT-Powers CFD code do contribute to meanderings.

However, the driver's and workshop manual allows making fairly assumptions of characteristic geometrics like intake and exhaust geometry and gives the engines maximal torque and performance. Further information like the engine load map, a torque curve or even a valve lift profile is not known. Once the engine is in house, the model has to be adapted as described above. Therefore, all numerical values of the simulation have to be handled with care. The thermal efficiency indicate to a small extend the plausibility of the model, as it is for usual around 25 – 30% for combustion engines (Heywood 1988).

A weakness of the simulation which might have effects on the results is the assumption, that all parts are mounted straight in-line, which means that all bending is neglected. Bending in pipes changes the cross-sectional velocity distribution of the flow inside the pipe: The velocity is higher at the outer edge than at the inner edge of a bending and thus, the pressure distribution differs. Bending will also cause friction losses in the pipes. This does not reflect the reality as the limited available space in the frame and safety requirements determined by the rules do not allow this.

As expected the plot of the brake torque (Figure 35) and the BMEP (Figure 37) have got the same shape. After a moderate increasing at lower engine speed, a step appears "suddenly" in both plots at 4000 rpm. Then it drops slightly down at 4500 rpm and rises again to a peak value before it decreases at higher engine speed (> 6000 rpm). This is due to the intake system which is not well balanced as it is not explicitly developed for this case. The significant decrease of both BMEP and torque is both due to the lack of air resulting from the impact of the restrictor and the increasing engine friction at higher engine speed. If the engine in the dyno would show a similar behavior at higher RPMs it is warmly recommended to limit the maximum engine speed to about 7000 to 7500 rpm. The BMEP is in the range of 6 bar to 10 bar and thus within a range which is typical for naturally aspirated SI engines (Heywood 1988).

The volumetric efficiency in Figure 39 supports this assumption: It shows, that the rate of charge which is moved into the engine is fairly low (about 72 %) at lower engine speeds (1500 rpm to 4000 rpm). At mid-rage engine speeds, 4000 rpm, it increases suddenly from 70% at 4000 rpm to 84 % at 4500 rpm. The presumption seems reasonable that a Helmholtz resonance phenomenon of the fluid in the intake system appears at this speed range. The efficiency drops slightly to 82% at 5000 rpm but increases again to its peak at 94.5 % at 6000 rpm. This can be due to the comparably high valve overlapping duration shown in Figure 38. The literature (Heywood 1988) and (Bell 1998) explains the impact of modifying the valve opening and closing timing events. In general, the impact of cam timing on the gas oscillations in the inlet manifold (gas dynamic) is determined by:

- Time instance and intensity of the stimulation of gas oscillations
- Time instance of valve closing which results in inclosing of the gas in the combustion chamber
- Time dependent behavior of the flow characteristics and its resulting change of inertia.

A long overlapping period can cause the flowing of a certain part of the unburned air-fuel-mixture through the cylinder. That means, it passes from the intake port to the exhaust port without taking part in the combustion. The volumetric efficiency is satisfying high at higher engine speeds. Due to the exhaust valve opening before BDC (during the power stroke), a compression wave is generated which travels along the exhaust pipe. When this wave meets an increase in area at the collector, an expansion wave is generated which travels back towards the exhaust port. If the intake valve is open when this expansion wave arrives at the port then the waves draws particles from the intake system into the cylinder, die to a lower pressure in the cylinder than in the intake plenum. Due to the mandatory restrictor, the pressure in the intake plenum is lower than in the exhaust system and so, during valve overlap period, there is a reserve flow of exhaust gas into the intake system (Meekhof 2011).

This phenomenon is called circulation losses and is desired in race car engine since it can flushes the residual gases, due to its inertia, out of the cylinder which increases the volumetric efficiency as a higher amount of unburned gas can now flow into the cylinder. This leads to a higher performance in race vehicles.

Modifying the cam timing is affected by two opposed compromises: Power and Torque versus fuel consumption, emissions and driving comfort. This discussion shows the importance of implementing the original valve timing.

Figure 40 and Figure 41 are showing the average burned gas fraction in the intake port and exhaust port respectively.

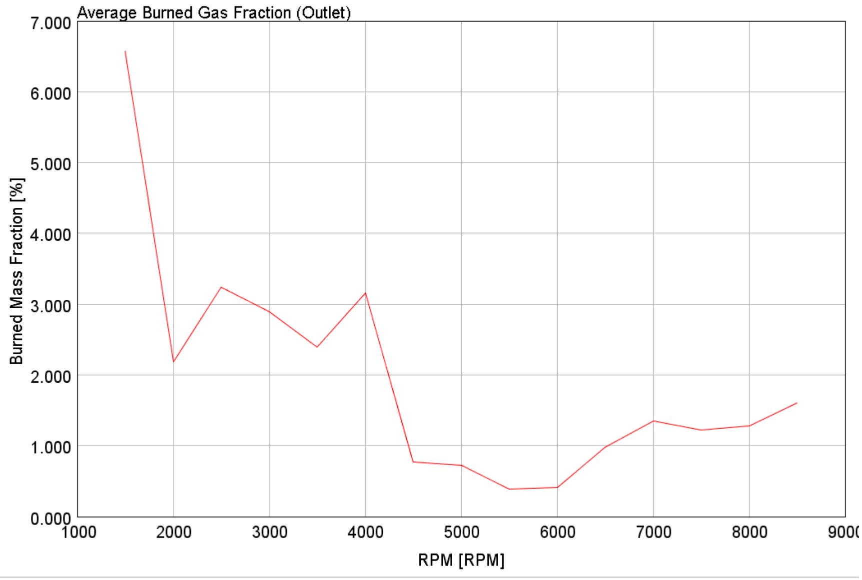


Figure 40: Average burned gas fraction in the intake port

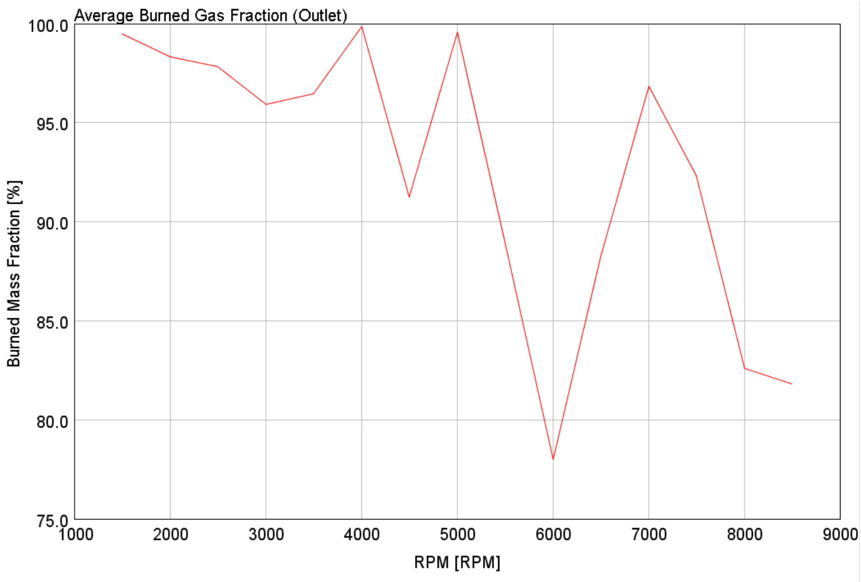


Figure 41: Average burned gas fraction in the exhaust port

At the exhaust port the burned fraction is high at lower engine speed (1500 – 4000 rpm) and starts oscillating at 4500 rpm. This is due to the large valve overlap and contributes to the loss in torque at higher engine speed. At the intake port the average burned gas fraction occurs at between 2 and 3 % at lower engine speeds and fairly low at medium engine speed (ca 1 %) and increases slightly to 2% at higher engine speeds.

Two typical examples in this simulation the combustion model as an example for an idealization and the fixed air-fuel ratio as an example for a simplification. The combustion is simulated using the Wiebe approximation implemented in GT-Power. This Model is an idealization since it might not reflect the real combustion in this particular engine but it is still able to deliver reasonable values. The air-fuel ratio is set to $\lambda = 1$. That means the combustion is stoichiometric at every time and the air-fuel mixture contains sufficient air to burn completely. Modern engines uses a lambda sensor which assures this, but usually the real lambda-value oscillates slightly around 1. A possible impact of higher lambda values $(\lambda > 1)$, that means a lean combustion, or lower lambda values $(\lambda < 1)$ – rich combustion –on fuel consumption and engine power which is not explicitly studied here, should be taken into account while designing the ECU. However this should be designed to ensure MBT (maximum brake torque) ignition timing.

Improving the overall engine performance is only possible by either rising the torque or the engine speed.

The fuel-injection rate is set to a fixed value and a certain position of the fuel-injector on the inlet runner is assumed. The fixed fuel-injection rate represents WOT operating condition. These parameters have been found following the overall simulation method, by testing different values and taking values which deliver the highest engine performance. This approach disregards any fuel vaporization phenomena, which can also affected by adapting the injection pressure. It might be possible to gain higher engine performance by having a well-designed fuel injection system. Such a system has to be developed separately.

Comparing the performance of the simulated engine (49.7 Nm at 6000 rpm and 31.8 kW at 6500 rpm) and looking at the consequential power to weight ratio in this case which is computed to 0.17 kW/kg shows, that the design target of 0.21 kW/kg was not met when the vehicle is operated with the combustion engine in this set up only.

This discussion showed the eligibility of natural aspirated engine tuning for increasing the engine performance. Many FSAE team are using supercharged engines. Results suggested by (Corrigan, McCullough and Cunningham 2006) and (Granqvist 2005) show the possibilities of increasing the engine performance with supercharging and turbocharging respectively within the FSAE rules.

In summary, all presented plots and data are within typical ranges and follow reasonable explanations. This verifies the reliability and the trustworthiness of the model and fleshes out the approach of using the values of this simulation for further calculations.

3.3.2.5 Validating the simulation model

According to the simulation method the following section will describe the methodology for the necessary validation of the model which has to be done as soon as the engine is in house. In particular the following geometric properties have to be evaluated as they are either not clearly shown in the manuals or not visible.

Air Intake

The prototype will use an adapted but already developed air intake system. Some parts of the CFS stock can be used and can be easily implemented in GT-Power.

Intake Port

As a CAD Model of the Intake Port is known, it was possible to implement these geometrics as well as possible. No further measurements are necessary. It can be recommended to implement the bending of the pipes into GT-Power. As mentioned above, the model neglects all bending, but since the software claims a bending angle, an angle of 8 degree was implemented.

The cam-lift profile is already implemented. However, the exact valve timing for the intake valve opening and exhaust valve opening event is not known and only roughly guessed based on data driven decisions discussed above. The discussion about the impact of valve timing in section

3.3.2.4 Discussion of engine simulation results makes clear that the original valve timing should be known. This measurement might be complex but in order to gain a valuable simulation, this is without alternative. It can be realized by using special software or evaluating the pictures from the workshop manual: Software named "Cam Doctor"[13] is used by other FSAE Teams (Corrigan, McCullough and Cunningham 2006). A contact to Husqvarna has been established but they did not provide all data by the end of this study.

The fuel injection should be implemented more properly as the final intake system will be developed.

Cylinder / Engine

As discussed the simulation uses basic and typical models for the combustion and heat release. This is a common approach in engine simulation and does not need to be modified. Even if it results in a more realistic simulation, a try for mapping the exact Husqvarna combustion process would be very complex. With regard to the investment of time even more detailed figures are not worth the effort.

Instead more effort should be put on geometric and engine characteristic measurements inside the combustion chamber. The engine needs to be totally disassembled for measuring the connecting rod length. These measurements have to be done only once. They are due to their characteristics fundamental for the simulation. If it is possible, the crank angle at IVC which defines in GT-Power the Start of Cycle CA should be measured.

Exhaust Port

The same approach as for the intake port is to be applied. Figure 31 shows the assembly of the exhaust and the difference from the intake port is apparent. Thus the exhaust runners are assumed to have a certain length before they are joined together.

Exhaust

The original exhaust system may not fit into the CFS11 frame, at least because of its alignment. The adequate adapted or developed exhaust system has to be measured separately. It is to mention that the rules dictate the maximum noise level of the exhaust system. This might be neglected for a prototype vehicle but it is necessary for the certification for the race car.

[13] http://www.powermechanics.com/camdoctor.html opened on 2012-04-24.

3.3.3 Design advices for mounting the ICE

The following section will describe and motivate which parts of the ICE have to be either added or adapted for a rule compliant operation. The advices do not have the pretense to gain maximum performance but they are suitable to realize a functioning system in the role of the combustion engine in the powertrain on a prototype level.

Subsystem Air Intake

Air filter and throttle body do not have any special requirements. In-house parts can be taken. The throttle body should be a “butterfly” type.

The restrictor can be built with the presented geometry. Still a deeper investigation is necessary.

The impact of plenum volume has been shown. A circular shape is preferred in order to minimize all type of flow losses at the boundary shape. Both spherical and cylindrical shapes are commonly used by other teams ((William , Watson and Konidaris 2007) and (Gilani 2012)). A spherical shape might be preferred due to the more advantageous relation between volume and surface for a sphere.

With regard to the in house knowledge and years of experience of the CFS teams, intake runners made of plastic can easily be manufactured. Following the discussion above an elegant solution is to design an intake system with variable intake diameter and length which can benefit from the descried Helmholtz Resonance effect and can yield to several “peaks” in performance at certain engine speeds. An injector compatible with the ECU is to be fitted into the Intake runner at its adequate position which has to be determined separately.

Subsystem Intake and Exhaust Ports

No modifications at the intake ports are necessary but due to its impact, the correctness of the model in the simulation is to emphasize.

The special FSAE competition requirements for the engine can be met by a variable cam timing system or the designing of an adapted cam phasing. The possibility of this is shown in (McClintock, et al. 2008).

Subsystem Exhaust

Figure 27 shows how the original equipped exhaust system looks. A rough, cheap and easy to realize method could be the adaption of the length and bending of the pipe.

Engine Cooling

Since the engine is water-cooled, all cooling hoses can be piped corresponding to the package concept. An existing air cooler with radiator can certainly be adapted.

3.4 Electric system

This section describes the simulations, optimizations and choice of the electric components.

3.4.1 MatLab simulation

The electric system was simulated in MatLab to get an estimation of the size of the electric components and how these cooperate and be optimized during a race. This also made it possible to calculate the fuel saving for a similar car as the one measured.

Figure 42 illustrates the forces acting on the vehicle.

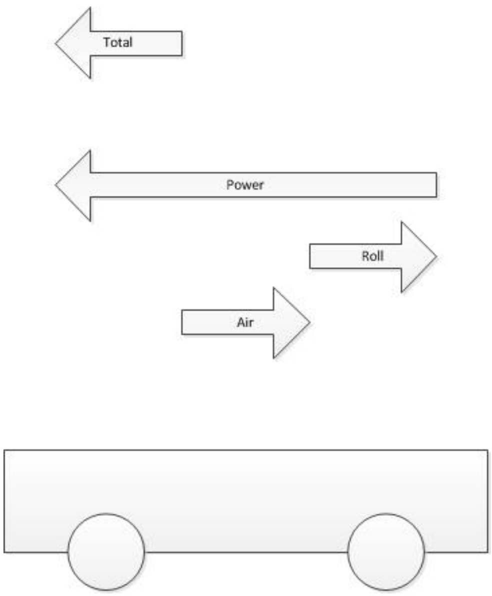


Figure 42: Forces acting on the vehicle

$$F_t = ma + \frac{1}{2}\rho A C_d v^2 + mgf_r$$

- F_t is the force on the wheels
- m is the mass of the vehicle
- ρ is the density of air
- A is the frontal area
- g is gravity
- Cd is the drag coefficient
- v is the velocity
- f_r is rolling resistance

(Guzzella and Sciarretta 2007)

To calculate the force required on the wheels to drive an available speed profile from CFS was used. From this the accelerating and braking power could be calculated. Drivetrain losses were not considered. Also the rotational speed of the wheels and the torque are calculated from the speed profile.

The constants from a super capacitor and the numbers of the super capacitors are used in the model. This makes it possible to calculate the resistance, capacitance and voltage of the super capacitors.

The rated current, the magnetic flux and resistance of the electrical machine is assumed from real electrical machines in the range of study. The maximum speed before the field weakening should start, the minimum voltage of the super capacitors and the maximum current in the electrical machine are set. From these actual and assumed values we calculate the rated power, maximum torque and maximum EMF from the electrical machine.

In the Figure 43 is a flowchart over the MatLab-code can be seen. A loop is used for calculating the different quantities for each discrete time interval of the speed profile. First a continuing generation of the electrical machine is set, this is for braking with the electrical machine and charging the super capacitors when the ICE is not in full load. After this the torque that the electrical machine is requested to deliver is calculated, this is used to calculate the current in the electrical machine. If the current needed to deliver the torque is higher than the maximum current allowed the current is lowered. Depending on what the current in the electrical machine is, the duty cycle is calculated in different ways. If the duty cycle is greater than one, which is physically impossible, the field weakening region has been entered. The magnetic flux and the current are recalculated. If the duty cycle is less than zero the duty cycle is set to zero and the current is determent of the magnetic flux, resistance and speed. When these values have been calculated and checked to be valid the back-EMF, voltage over the electrical machine, current in the super capacitors, voltage over the super capacitors are calculated from the previous calculated quantities. The torque that is actually delivered from the electrical machine is also calculated. When the quantities are restricted it is possible that the torque deliver is not equal to the torque that is requested.

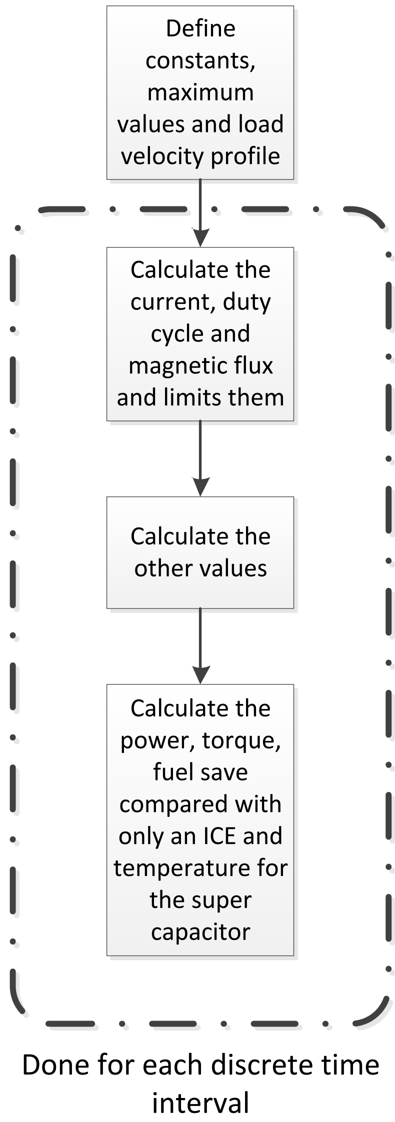


Figure 43: A flowchart over the MatLab-code

When it is know how much torque is needed but cannot be delivered from the electrical machine the rest of the torque needed to reach the torque from the speed profile comes from the ICE when accelerating and the brakes when decelerating can be calculated. The power and energy from the ICE and brakes are calculated from this torque. This will give information on how much of the energy comes from the ICE and how much from the electrical machine. The energy required for following the speed profile is also calculated without the electrical machine, this gives an approximation of the energy saved with the hybrid system.

The voltage over the super capacitor for the next discrete time interval is calculated, this can be done because the current in the super capacitor, the capacitance and the time interval are known. If the voltage in the next time interval would be greater or less than the maximum/minimum allowed voltage, the current for next value is set to not charge/discharge. If the current is restricted to not discharge more, it could still charge. When the statement is no longer true the restriction for the current is reset to the original.

A simple temperature of the super capacitor is included to give an approximation of how warm the super capacitors will be during an event.

This program has some shortcomings, the model is following the input model and it is possible that the new model could accelerate faster than the previous thanks to the electrical machine that gives maximum torque directly. The MatLab simulation follows the speed profile. The dynamic when driving cannot be seen in the model, therefore it is suspected that lower weight is better than the result shows. The temperature calculations for the super capacitors are based on data of the thermal resistance and the thermal capacitance for one super capacitor in free space. In the vehicle more capacitors will be used, they will be stored in one or a few boxes with some capacitors in each box, this can result in a higher temperature than has been calculated. Further simulations using more advanced software is required to achieve more exact results.

3.4.2 Electric machine

The electric machine will propel the vehicle during acceleration and brake it during deceleration. The electrical machine should be as light as possible and deliver the required power. There is no use in having a machine which can deliver or recuperate more torque then the tires can cope with. The tires can be a limiting factor during heavy breaking due to the weight shift in the vehicle. If the combustion engine can deliver a high torque and the electric machine is very powerful the tires may start slipping during acceleration.

One time per lap the vehicle experiences a significant deceleration, see Figure 44, optimizing the system for this event could increase the systems weight, such a system would be over dimensioned for the rest of the lap. The electric machine is coupled to the rear wheels, this significantly reduces the possibility to recuperate energy during braking (see 3.2.2 Maximum torque on rear wheels while braking), this is also a factor that needs to be accounted for when dimensioning the electrical machine size.

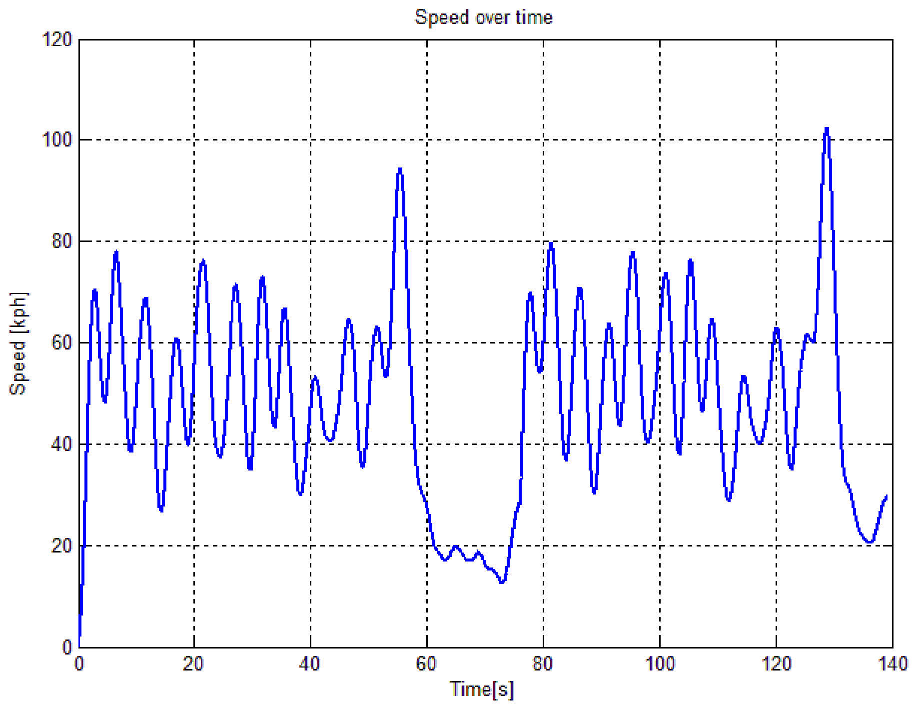


Figure 44: Speed curve

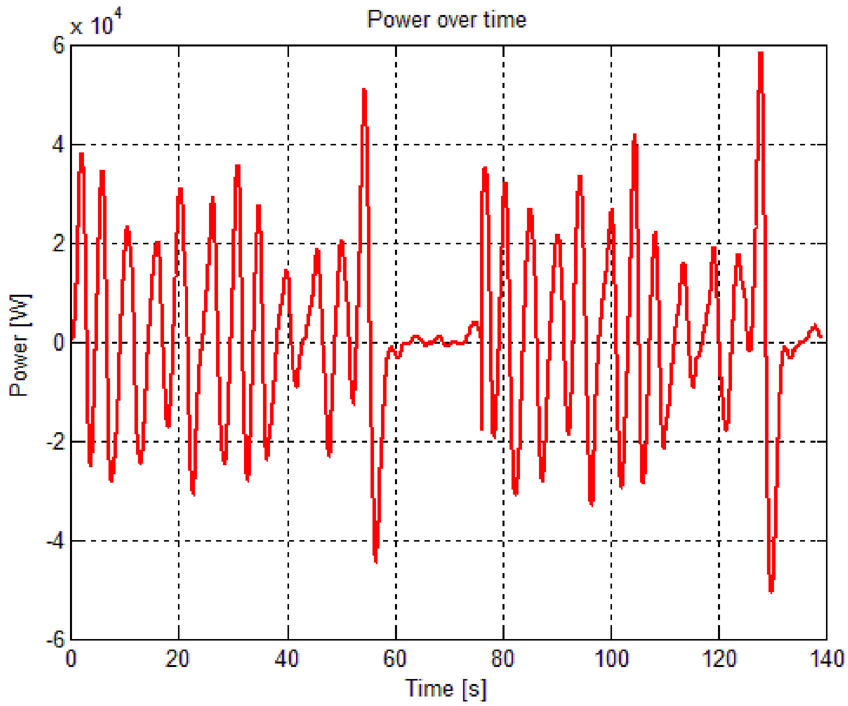


Figure 45: Power curve

Using a machine that has the same rated power as the maximal power needed could give better efficiency but would probably weight more and have a larger inertia. A small machine which needs to be overloaded would probably have lower efficiency than the larger machine but lower weight and lower inertia, lower weight gives better performance in every situation on the track, reducing the weight of the machine can make it possible to dimension down the frame, brakes etc. A compromise between efficiency, power and weight is required.

Asymmetric overload; run the machine at a predetermined current when accelerating and run an even larger current though it when using the electric machine as brake.

Using asymmetric machine loads could be a good approach. The energy that can be recuperated during braking is “free”, even if the system is very inefficient (due to overload) during the regenerative brake, the energy contribution to the capacitors may be greater than with lower load and higher efficiency.

The electrical machine must be able to handle the root mean square (RMS) current during the driving cycle, else it may become overheated and break down after time. The super capacitors and the electrical machine should be chosen so that the super capacitors can deliver the required power to the machine, there is no use in having an electric machine which can deliver much more power to the driveshaft than the super capacitors can deliver to the machine over a driving cycle.

Absolute maximum power that can be available during heavy breaking is: Available torque * the angular velocity of the driveshaft at maximum speed ≈ 30 kW

According to (Lukic and Emadi 2004) the optimum hybridization factor is between 0.3 and 0.5 but the advantage of a hybrid system shows between 0.15 and 0.5. If the combustion engine has an output power of 30-35 kW and we aim for a hybridization factor of 0.15-0.3 the required power of the electric machine will be 5.3-15 kW. A hybridization factor above 0.3 would require an even more powerful machine, it can be difficult to find a machine with so high power and a good power to weight ratio. Also the utilization of such a machine would probably be low (see Figure 45).

$$Hybridization\ factor = \frac{P_{EL}}{P_{EL} + P_{ICE}}$$

The super capacitors can deliver some specific RMS current, the electric machine does not need to be able to deliver more RMS power than what the super capacitors can deliver to the electrical machine.

MATLAB was used to calculate of how much power that is the best choice for this application, depending on the resistance in the system, the suitable power varies between 8-12 kW (The MATLAB code can only be seen as a guidance and can be found in (Appendix M MatLab code). The graph shows the fuel consumption reduction quota between the hybrid system and the conventional system for different powers and resistances.

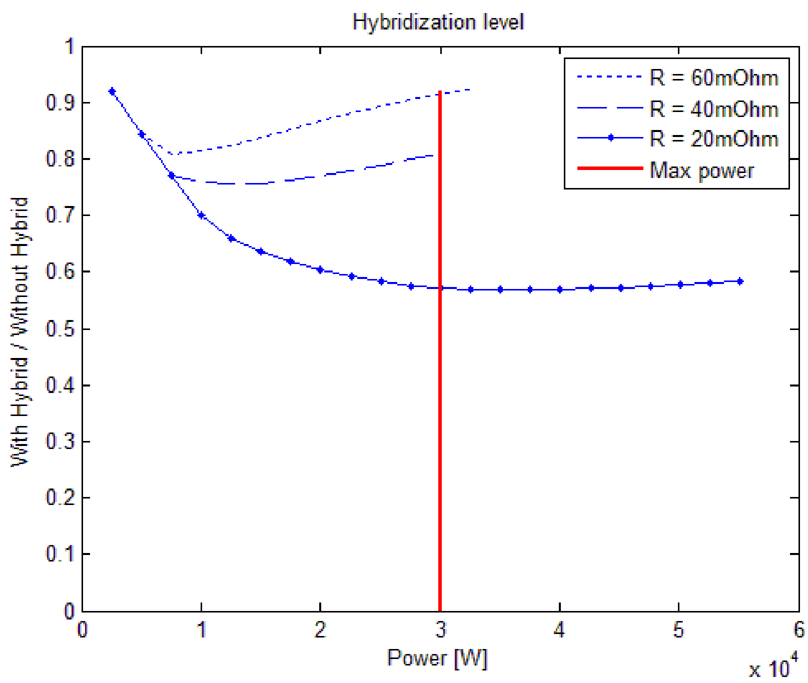


Figure 46: Power vs improvement by hybridization for different resistances

All parameters in the system where constant except for the allowed current in the system

$$P = V\ I\ [\mathrm{W}]$$

According to our own measurements (Appendix L Test and calculations on model aircraft machine) and specifications from retailers ((Hobbyking 2011) (Electricmotorsport 2010)) the resistance of the electric machine can vary between 10 and 50 mΩ, the capacitor configurations that have been considered had an internal resistance which varies between 10 and 25 mΩ when 29 are wired in series.

Table 14: Different electrical machines

Name	Weight [kg]	Power [kW] (Peak value)
Turnigy CA 120-70	2.5 kg	17.5
Turnigy RotoMax 100cc	2.1 kg	8
Motenergy ME0907	10	4.4 (11)
Motenergy ME0201014201	10	4.4 (14)
Motenergy ME1012	15.9	10 (24)

(Hobbyking 2011) (Electricmotorsport 2010)

Machine suggestion: Turnigy CA 120-70

The Motenergy machines are much heavier than the Turnigy machines, therefor they are undesired. The Turnigy rotomax 100cc cannot deliver enough power without overload.

This type of machines is not intended to be used to drive a vehicle, but with appropriate cooling it should be possible to use it in the hybrid system (Appendix L Test and calculations on model aircraft machine). The machine has no own cooling, it will need forced air cooling. This machine is relatively low voltage. It can probably deliver the power that is needed, the machine is very light compared to the alternatives, the efficiency is probably satisfying (Appendix L Test and calculations on model aircraft machine). The eventual heat problems may be a more easy to overcome if the machine has more surface from which it can be cooled compared to a smaller machine. It is of the type "outrunner" (Appendix L Test and calculations on model aircraft machine) so it should have a cover in case the rotor breaks so the fragments do not injure someone.

3.4.3 Energy storage

The market for super capacitors in automotive application is quite small so there are not many developers in the market. Maxwell Technologies is the largest company and there are also a few smaller companies. Most of the smaller companies have super capacitors with lower capacitance. Another possibility is to develop an own super capacitor but that would take time and that is outside the scope of this project.

The choice is between complete modules or single super capacitor assembles. Multiple modules can be connected to each other to create higher voltage and would make it possible to store more energy. The major drawback for using modules is the weight, which can be seen in Table 15. A BCAP0650 weights 160 g, this result in a weight of 4.8 kg if thirty of them are used, compared with the modules in Table 16, where the lightest weights 5.51 kg and is only 16 V, it can clearly be seen that the modules is a heavier solution than assemble single super capacitors.

Table 15: Single cells from Maxwell Technologies K2 series

Units/Models	BCAP0650	BCAP1200	BCAP1500	BCAP2000	BCAP3000
Capacitance [F]	650	1200	1500	2000	3000
Voltage [V]	2.7	2.7	2.7	2.7	2.7
Maximum current to get 40°C temperature difference [A]	88	110	140	170	210
Weight [g]	160	260	280	360	510
Specific power [W/kg]	6800	5800	6600	6900	5900
Specific energy [Wh/kg]	4.1	4.7	5.4	5.6	6
Volume [l]	0.149	0.214	0.246	0.295	0.399

(Datasheet K2 Series Ultracapacitors u.d.)

Table 16: Modules from Maxwell Technologies

Units/Models	BMOD0500	BMOD0083
Capacitance [F]	500	83
Voltage [V]	16	48
Current [A]	160	100
Weight [g]	5510	10300
Specific Power [W/kg]	5500	5600
Specific Energy [Wh/kg]	3.2	2.6
Volume [l]	5.088	8.463

(Datasheet 16V Modules u.d.) (Datasheet 48 V Modules u.d.)

All of the single super capacitors have been tested in the MatLab-simulation, in packs of 29. The reason for using 29 super capacitors are that it gives a maximum voltage under 80 V, there exists fuses of 80 V with very low weight, it could be difficult to find fuses with higher voltage and low weight. A high voltage is wanted because it means lower current can be used for the same power and lower current means lower losses. Using 29 super capacitors gives the closest highest maximum voltage that still is under 80 V.

The results showed that there are no large differences between the different super capacitors, the voltage varies more for the super capacitors with lower capacitance but that is not an issue because the maximum voltage level is not obtained during any braking event, which means no regenerating energy is lost. A larger difficulty was that the result also showed that the root mean square (RMS) - current is very near the maximum for having a temperature increase of 40°C compared with the surrounding. To be certain that the temperature of the super capacitors do not reach 65°C, the maximum operating temperature, the temperature development during the race was included in the simulation, both with and without any cooling. The result from the temperature-part of the simulation was positive, the simulation shows that the temperature does not increase more than 53°C for super capacitors, total number of super capacitors were 29, with lowest capacitance even without cooling.

In Table 17 it can be seen that a higher number of super capacitors is good for the fuel consumption, even if the differences are small. The best number of super capacitors from an efficiency perspective would be 29 or more if fuses of more than 80 V are used, but there is one significant challenge with using 29 super capacitors. The problem is that from a weight/fuel consumption perspective it would be better to have fewer super capacitors, approximately 26 or even less. The issue with only weight/fuel consumption in mind is the temperature of the super capacitors, if 25 super capacitors are used the temperature would reach over 61°C without cooling, this temperature is under the maximum operating temperature but this does not create a large enough safety margin to the maximum operating temperature when using this number super capacitors.

Table 17: Table of different numbers and minimum discharge of super capacitors

Number of super capacitors	Discharge	With/without hybrid
29	0.78	0.8092
28	0.78	0.8096
27	0.78	0.8100
26	0.78	0.8104
25	0.78	0.8110
29	0.8	0.8088
28	0.8	0.8092
27	0.8	0.8099
26	0.8	0.8109
25	0.8	0.8126
29	0.82	0.8092
28	0.82	0.8105
27	0.82	0.8120
26	0.82	0.8136
25	0.82	0.8152

The optimal minimum permitted discharge of the super capacitors is approximately 80 % of maximum voltage, because at this minimum discharge the super capacitors are not completely filled during the braking that generates most energy, which happens when using 82 % as minimum discharge and some of the possible regenerated energy is lost. At 80 % minimum discharge the average voltage is higher than at 78 % minimum discharge, a higher average voltage makes it possible to have a lower current and obtains the same power. The lower current therefore results in lower energy losses.

How the temperature for a super capacitor develops over a complete race can be seen in Figure 47. In this figure a time of cooling after the race is included to get a view of the time it takes to cool down the super capacitor. It is important to note that the temperature is for a case where the one super capacitor is standing with free air around itself. Therefore the temperature increase is not representative for the super capacitors when placed in the vehicle. A simulation was made without any cooling at all, this simulation showed that the temperature would increase to about 53°C with 29 super capacitors is connected in series, the maximum temperature for the super capacitor is 65°C. With fewer super capacitors the temperature increases more, since the power is divided over less super capacitors.

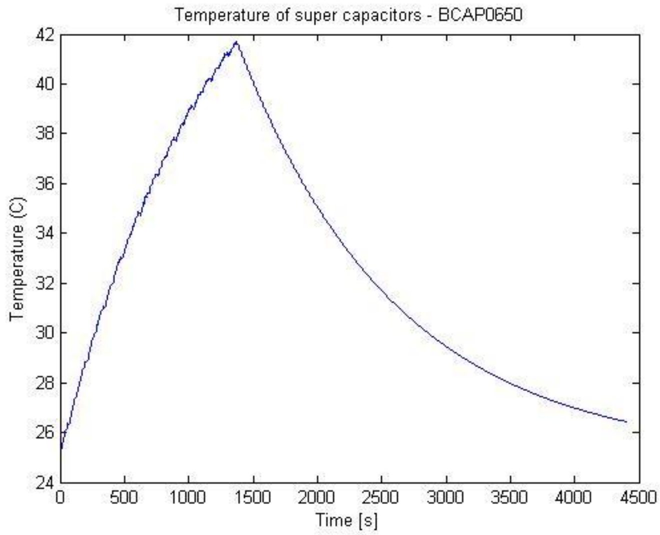


Figure 47: Temperature of a super capacitor

So the simulation results indicates that BCAP0650 would be the best super capacitor to use, it does not reach maximum voltage when braking and it have the lowest weight. 29 super capacitors will be connected in series because the temperature is lowest where the maximum voltage is still under 80 V.

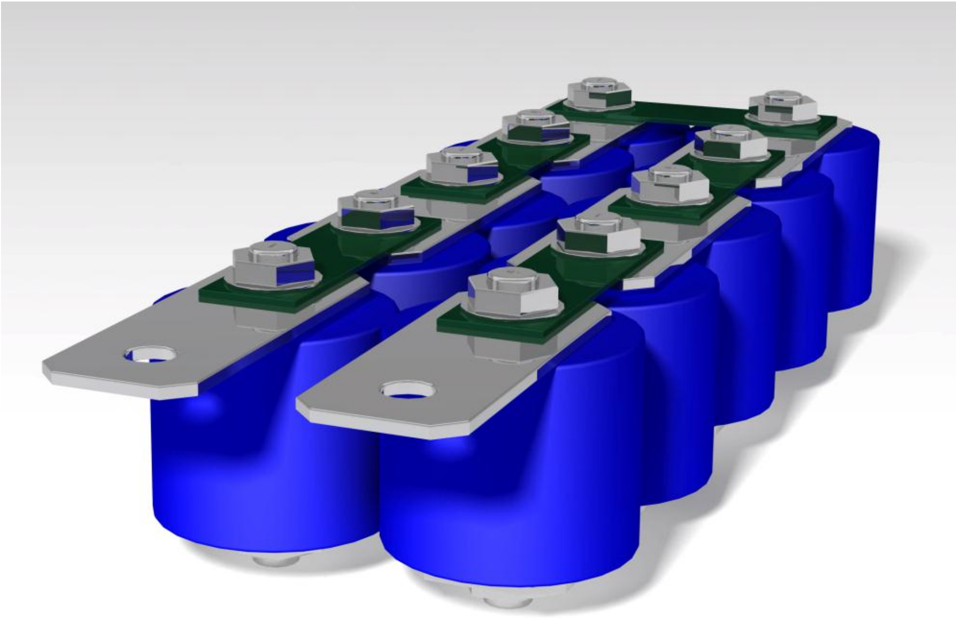


Figure 48: Super capacitors

3.4.4 Energy storage cover

Due to the FSAE Rules 2012 B 19.13 (Rules 2012) the cover of the energy storage must be able to protect the energy cover from unwanted electrical connection and fire and keep the energy storage in position even at decelerations of 20 g.

To meet all targets of the rules and maintain low weight the best solution was to use several layers of different materials. The fire proof material had to be UL94-V0 classified (UL94 V0, V1 and V2 - Plastmaterial, SP n.d.). One example of approved material is polycarbonate (PC) (PC Transparent FR V0 ML849 (UL94 V0), Alibaba n.d.). Due to the rules (Rules 2012) the cover should preferably be transparent to ease the inspection. PC is possible to get transparent. Also polypropylene (PP) is approved and used in batteries (FR Polypropylene pellet PP UL94 V0, Alibaba n.d.) – probably a suitable material. The same company also sells for example UL94-V0 classified polyamide (PA66) and acrylonitrile butadiene styrene (ABS).

There are two main ways of getting these covers; CAD one and manufacture or buy one of the shelf. The first alternative allows the cover to be perfectly fitting to the super capacitors, but it is not a reasonable cost. Using injection molding would require an injection molding tool that costs over 100 000 SEK (Boldizar 2012). The second alternative is much cheaper and the cost makes it the better choice, even if the performance is lower.

Enersys Reserve Power is one example of company that sells this type of equipment. EnerSys uses the material ABS (Battery Range Summary 2010), precisely as in one of the examples from Alibaba. The main disadvantage of buying off the shelf is that the cover becomes bulky and the frame does not allow the size to be too large.

One way of meeting all targets is to use different materials in different layers. For example one layer of fire proof material and another with higher strength. This is likely to be the best way to minimize the weight and still match all targets.

Since the super capacitors were decided very late in the project, there was not much time for designing these covers. A conceptual cover developed, within this project, for the Maxwell BCAP650 is shown by Figure 49.

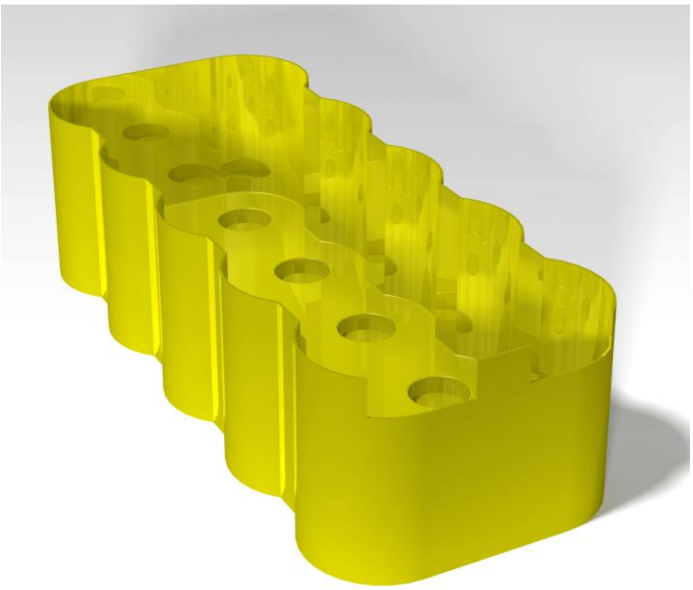


Figure 49: Conceptual energy storage cover for Maxwell BCAP650

3.5 Physical interaction

The transmission design that is chosen is the one where the electric motor and the ICE operate on the same chain directly related to the diff. The design phase of the transmission includes things as calculating the Angle of Wrap (Alpha in Figure 50), the possibilities of using a model airplane motor from the mechanical point of view, finding space for all the components and give estimations of desired gear ratios.

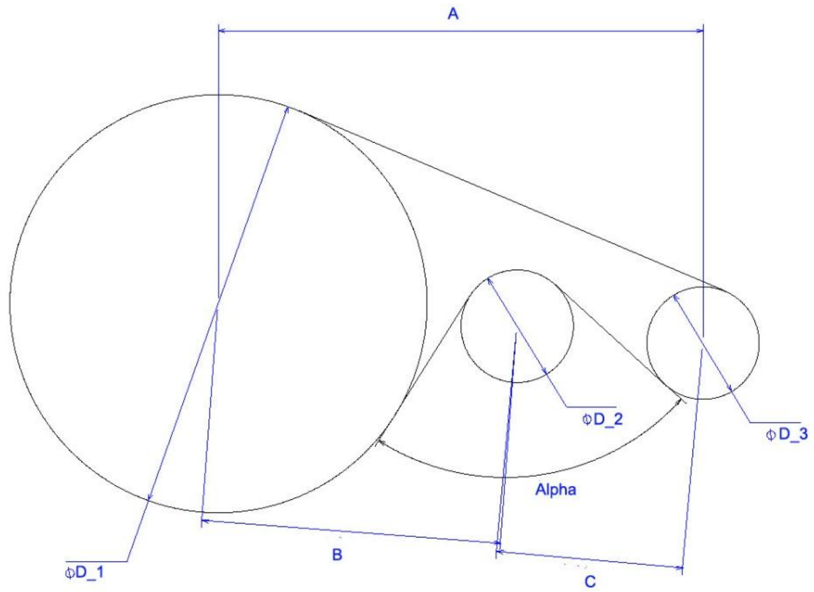


Figure 50: Layout of sprockets

3.5.1 Engagement of the sprockets

With the layout determined the important Angle of Wrap for the electric machine that runs on the outside of the chain can be calculated. The number of engaged teeth must be taken into consideration. A basic rule of thumb is that at least six teeth should be operating at the same time. With small sprockets is this rule not suitable, since the angle of wrap increases. If, for example, the small sprocket have ten teeth the angle become 216°. This example shows that the rule of thumb is not suitable.

A calculation of the forces involved is therefore necessary to determine if a certain layout will be possible. (Mägi and Melkersson 2010) recommend that the relation κ between the forces from the chain on each side of the sprocket is calculated. The recommended relation is κ = 10-50 and this is needed to calculate the preload of the chain. Preload is necessary to avoid the chain from slipping over the teeth and cause failure.

The maximum relation κ_{max} is calculated in the following way:

$$\kappa_{max} = \left[\frac{\sin(\theta + \gamma)}{\sin(\theta)}\right]^n$$

Where n is the number of engaged teeth, θ and γ are the angles as shown in Figure 51 (Mägi and Melkersson 2010).

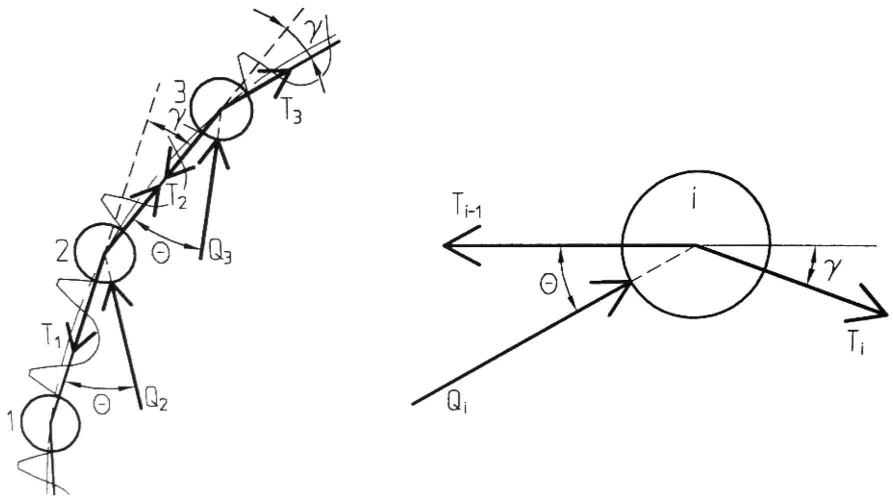


Figure 51: Forces on a single roller in a chain

This formula is then iterated to get a value for n. When the number of teeth that needs to be engaged and the total number of teeth on the sprocket is known, the Angle of Wrap can be calculated as follows:

$$\mathrm{AoW} = \frac{\mathrm{n}}{\mathrm{n_{tot}}} * 360°$$

With κ_{max} = 10 the number of engaged teeth will be 4.0405 and with a total of 12 teeth the AoW will be 121.215°. That is a reasonable value that is possible to achieve in this configuration. Important to notice is that this is the minimum required numbers. A safety margin is recommended. In this case an increase of the AoW would be advised.

3.5.2 Gear ratios

The definite gear ratios that will be used in the powertrain depend on the electric machine and the ICE. It is recommended to use standardized sprockets as it can be a good idea to carry out some testing with different sprockets. For calculations and estimations a final ratio of 5:1 has been used.

Internal gear ratios in the bike gearbox are not considered to be modified in this application. Further testing will show if it is possible to reduce the number of gears by removing them from the gearbox.

It is possible to have slightly different gear ratios for the ICE and the electric machine in this configuration, since the sprockets do not need to be identical.

3.5.3 Traction force

The traction force diagram for running on combustion engine only is shown in Figure 52.

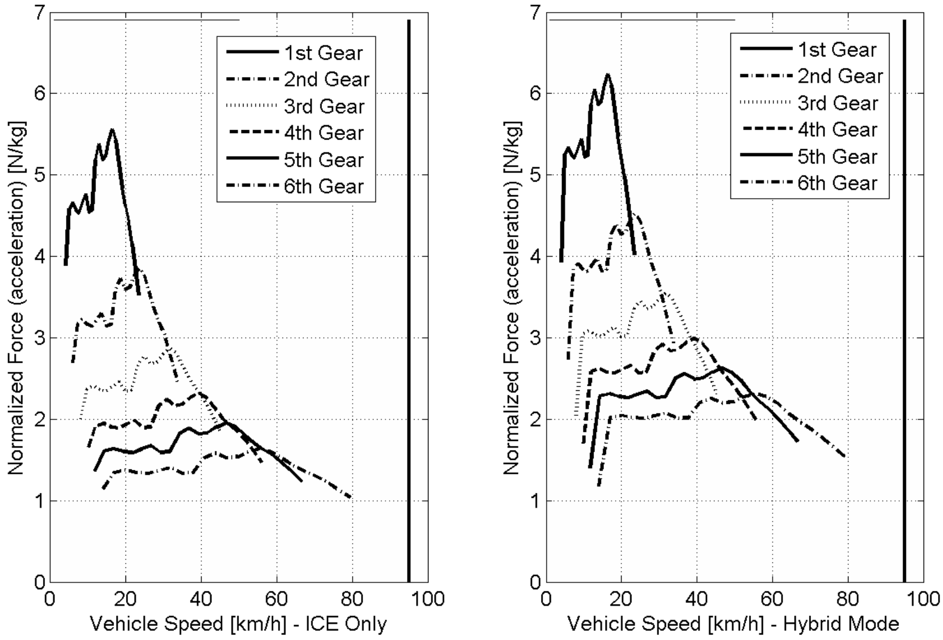


Figure 52: Traction force from powertrain: 30 kW ICE performance

The shape of the curves follows basically the shape of the engine torque. The normalized traction force is higher in lower gears but high vehicle speeds can only achieved in higher gears. It has a peak value of 5.5 N/kg and 6.2 N/kg in the first gear running on ICE mode and hybrid mode respectively.

The engine does not provide enough power for having the same acceleration as CFS-11 and top speed. Using both ICE and electric machine leads to a higher acceleration force but does not raise the vehicle speed. The maximum speed of the electric drive was the limit to the design of the gear unit between the electric drive and the wheels. The traction force peak is increased by 0.7 N/kg by the electric machine and thus, 13% higher. The model shows an incensement of 0.5 N/kg for highest vehicle speed. This value can be questioned since the model disregards both increasing fraction of the aerodynamic drag at higher speeds. However, the incensement will be lower at higher vehicle speeds as the torque of the electric drive decreases.

The third and the fifth gear might not be used since the entire vehicle speed range is sufficiently covered by the first, second, fourth and sixth gear. These gears can be removed from the gearbox which can lead to a slightly reduction in the inertia by lowering the rotating mass. Further, the typical phenomena are apparent: Driving in a lower gear makes higher acceleration possible but high speeds can only be reached with driving in higher gears.

An about 30% higher engine power is necessary for achieving CFS11 performance. In order to confirm this assumption, the simulation is performed with a 30% higher maximum engine power (40 kW). The result is shown in Figure 53.

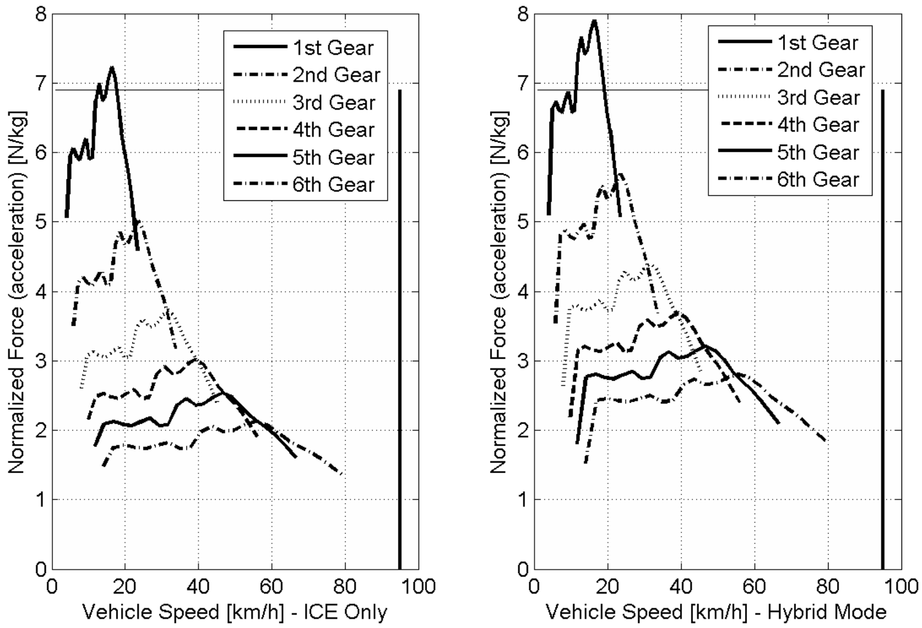


Figure 53: Traction force from powertrain: 40 kW ICE performance

With that performance, it might be possible to archive CFS-11 performance in acceleration and the overall acceleration performance is increased. The CFS-11 acceleration contest performance is reached and exceeded at a certain engine speed interval. In the case of running in ICE only, the peak value for the normalized traction force is 7.2 N/kg. This shows the general possibility of a higher performance with the hybrid powertrain. The traction force in the right graph has a peak value of 7.9 N/kg and thus 14% higher than the CFS-11 reference. The values might be lower due to the described assumptions for the simulation. As mentioned above, it is limited by the tire friction on the road.

The maximum vehicle speed is in this model (79 km/h) limited by the engine speed. This is usually not the case in reality since the aerodynamic drag and road friction are more likely to lower remaining engine power available to maintain the needed drive power. The Husqvarna SMR 630 motorbike is an enduro / off-road Motorbike and therefore the gearbox is not designed for high vehicle speeds. This has to be taken into account while designing the vehicle.

The shape of the course shows a strong decrease, similar to the decrease of torque at higher engine speeds. As this engine behavior changes, these traction force curves will change adequately. In particular the gear discussion above has to be adapted appropriately.

3.6 Packaging

All components must be mounted into the frame in such way that they all have enough space and at the same time are able to interact with each other. This section describes the packaging of the components into the frame.

3.6.1 Frame

Since the frame of the car, in which the hybrid system will be implemented, is not designed yet the frame of the CFS-10 car was used. This was because of the box in the back of the vehicle (Figure 55) which creates more space than for example the CFS-11 car which did not have this box (Figure 54). It is preferred to have more space because of the components added for the hybrid system.

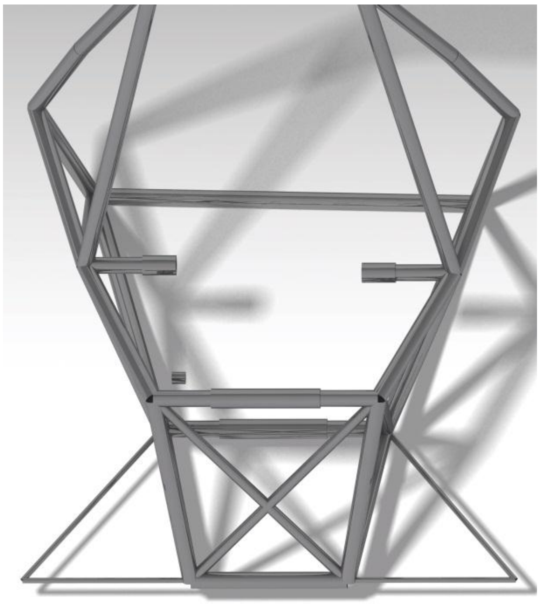


Figure 54: CFS-11 frame, no box in the back

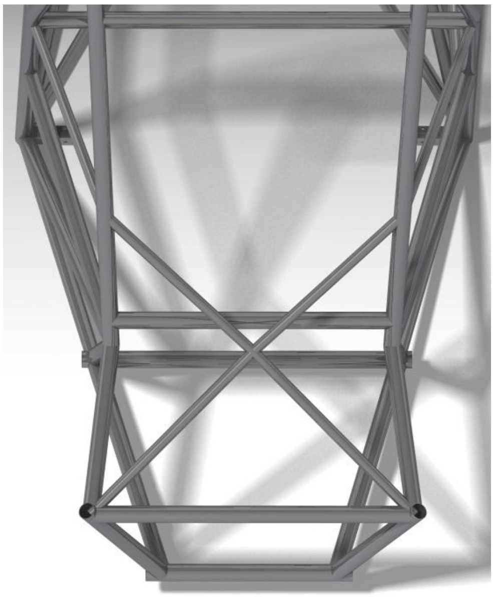


Figure 55: CFS-10 reworked frame, box in the back

The CFS-10 frame had a design suited specifically for the components of the vehicle. This frame had to be slightly redesigned to allow mounting of some components and some compromises were reset. The cross in the back was redesigned so that it is symmetric. Compare Figure 55 and Figure 56.

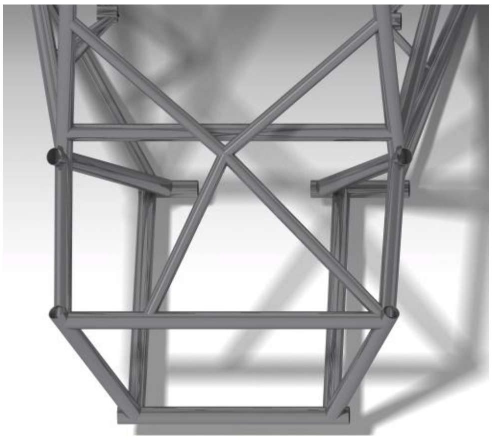


Figure 56: CFS-10 frame, not reworked

The brace in the lower region of the box was stretched to create mounting points underneath the components. Inserting the differential with its largest sprocket into this box should not be a problem since the opening in the front is still big enough.

The junction points in the lower front of the box are moved a little away from each other to create space for the electric machine and the engine mounts for the previous Yamaha Fazer were removed.

Figure 57 and Figure 58 shows the redesign from another angle.

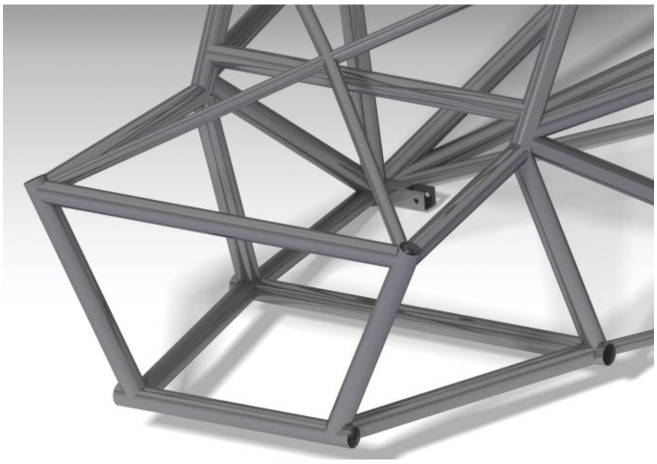


Figure 57: CFS-10 frame reworked

Figure 58: CFS-10 frame, not reworked

3.6.2 Hybrid components

The ideal case would have been to make a final design with all the right components, but since the aim of the project was to decide these things the time did not allow the CAD to be finished. Therefore the CAD is conceptual and does not include exact measures. The mountings have not been defined because the components are conceptual as well.

The components that have been considered while packaging are; combustion engine, fuel tank, exhaust system, electric motor, super capacitors, differential and sprockets. These are the components that require the most space. For example cables and pipes and the control unit have not been considered.

3.6.2.1 Internal combustion engine

Starting off with the combustion engine there must be said that the used CAD model does not show the Husqvarna SMR 630 which will be used in the real car. The model has not been found so the only way of getting a CAD model was to create it. Only the presented model of the cylinder head was transmitted from Husqvarna by the end of this project. Since this was a large project in itself and no measures were found either and furthermore the engine was not bought there was no possibility to get a Husqvarna model of relevance. For this reason the air cooled one cylinder CAD model (shown by Figure 59) was cut out of a race vehicle of the same size as a formula student car and used. Its size and shape is similar to the Husqvarna engine.

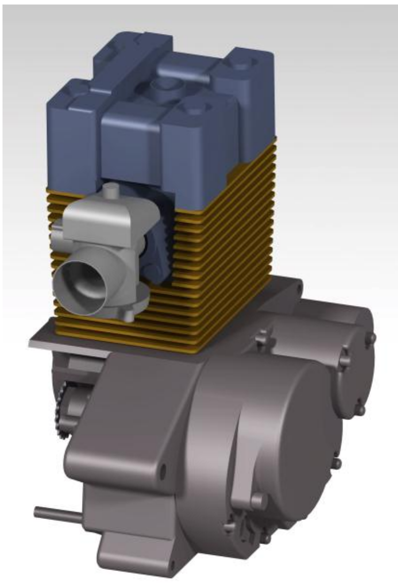


Figure 59: Single cylinder, conceptual, engine

3.6.2.2 Fuel tank

The fuel tank had to be moved and rebuilt totally. In the new position the shape of the CFS-10 fuel tank was not suited. The volume of the CFS-10 fuel tank was 5 liters. One of the main purposes of this project was to decrease the fuel consumption and with a single cylinder engine instead of four and a hybrid system added the fuel tank should not require the same volume. Though, the new fuel tank has a volume of 5 liters so that it is possible to complete the endurance race even without the electric motor. The new fuel tank is shown by Figure 60. The shape allows the fuel tank to be placed close to the fire wall, directly behind the driver, on the right side of the vehicle.

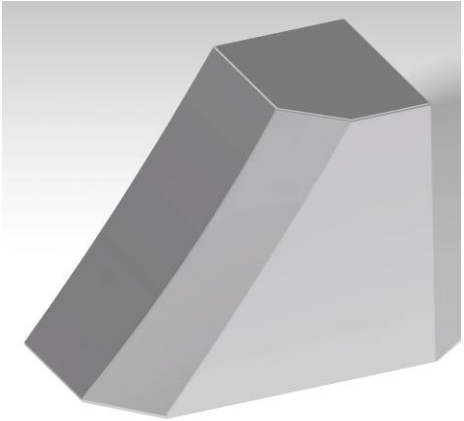


Figure 60: Fuel tank

3.6.2.3 Exhaust system

The exhaust system is as the ICE very conceptual and conformed to the ICE CAD model and the CFS-10 frame. For the Husqvarna engine it has to be rebuilt and furthermore the muffler is not investigated in aspects of sound reducing which of course is relevant. The exhaust system is shown by Figure 61.

The exhaust system must be designed so that the driver does not breathe the fumes at any speed. The end of the exhaust pipe may not extend more than 45 cm behind the rear axle and it shall not be above 60 cm the ground (Rules 2012). Therefore the exhaust pipe was designed so that the outlet is just above the ground and in the front of the rear axle.

It had to be formed in such a way that it can be connected correctly to the engine and does not intersect with the frame. Further it will be very hot and should not come too close to components that are sensitive of heat. The regulations (Rules 2012) also determine that it should not be possible to be hurt by the exhaust system and therefore it should be well packaged and metal surfaces should not be exposed where persons driving or approaching the vehicle might be hurt.

The shape shown by Figure 61 is a concept and could without too much problems be changed.

Figure 61: Exhaust system

3.6.2.4 Electric machine

The electric machine was not decided during the CAD modeling. More about the choice of motor can be found under 3.4.2 Electric machine. Since this was not decided when the conceptual packaging was done a principle motor was developed. This is shown by Figure 62.

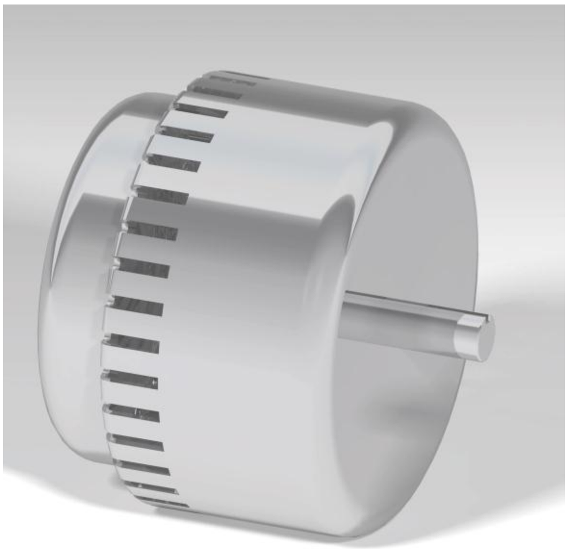


Figure 62: Conceptual electric machine

The size of this is comparably large so the real packaging should be easier and not a big problem.

3.6.2.5 Super capacitors

The super capacitors were not defined during the CAD modeling. Therefore the Maxwell BCap3000F was modeled since the supervisor of the project had a physical model of them and the project aimed for using that model for a long time. Section

3.4.3 Energy storage describes that this was not the final choice though. The final choice had the same dimensions except lower height so the packaging problem became easier to solve. Figure 63 shows the CAD model of the Maxwell BCap3000F used in the CAD model.

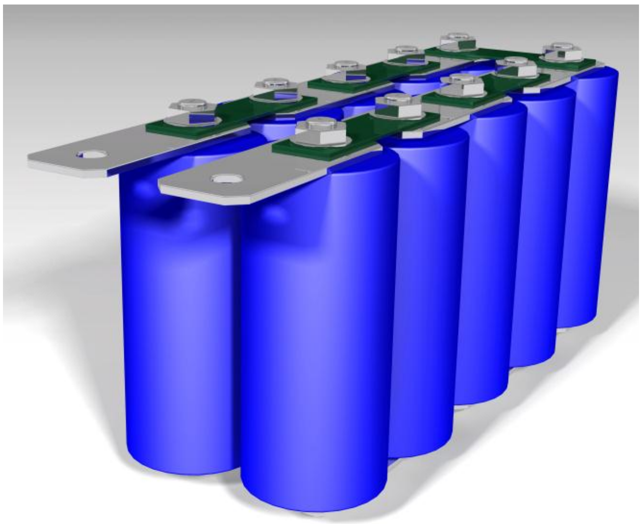


Figure 63: Super capacitors Maxwell BCap3000F

3.6.2.6 Differential

The differential from the CFS-10 car was used in the CAD model. This differential was the one that was originally used in the frame. The supervisor of the project also gave the advice to use this one and since it was only used as conceptual model no further studies on the differential was made for packaging. Figure 64 shows the CFS-10 differential.

Figure 64: Differential of the CFS-10 vehicle

3.6.3 Composition

The goal with the packaging was certainly to place the components in such a way that everything had sufficient space and prevent intersections, which was a challenge. Moreover the components had to be able to cooperate and interact which made the packaging more difficult. Figure 65 shows the result of the packaging and below it is commented and motivated. More pictures can be found in Appendix F Pictures of the packaging.

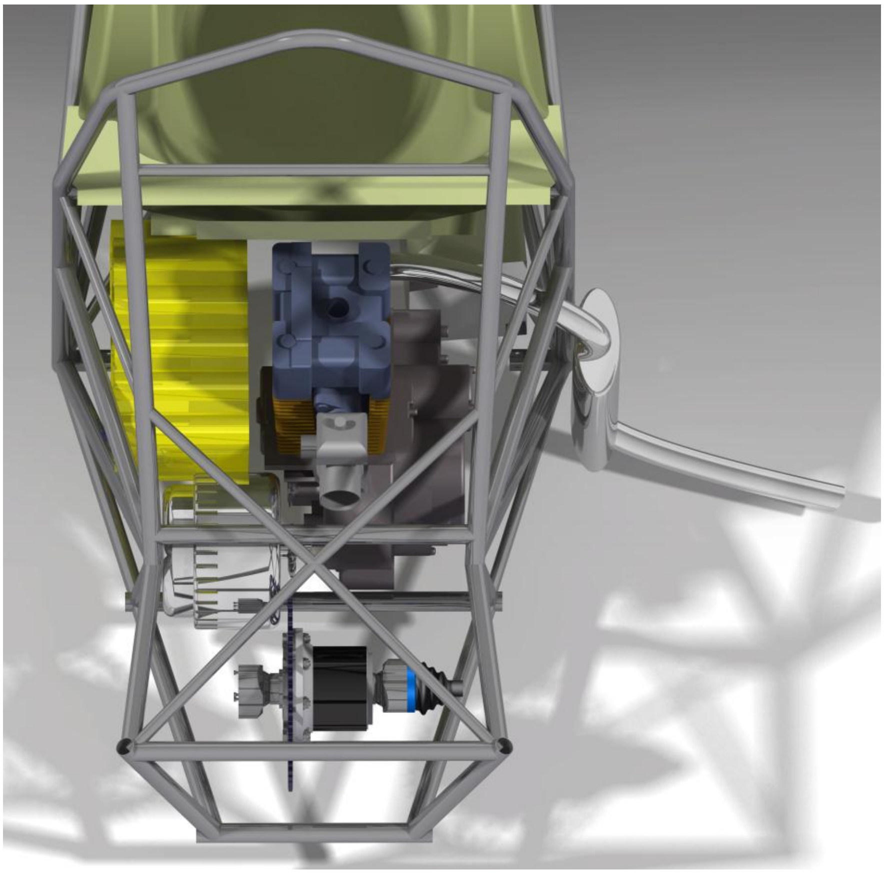


Figure 65: Result of the packaging

The gearbox of the ICE is on the right side of the ICE and should preferably be positioned in the right side of the frame so that the engine center and output sprocket is not offeset too much to the left. One of the problems in that case would be that the weight is shifted to the left and this may influence the dynamic performance negatively. This would also lead to that the cross in the back must be redesigned, as in the CFS-10 car (see Figure 56), so the differential does not intersect with the frame.

The output shaft is approximately in the middle and the differential must be in line with the output shaft. It is, while using a chain, not possible to have an offset between the sprockets. 2.6.3.3 Electric machine as a chain tensioner describes that the electric motor will act as a chain tensioner and as a consequence it has to be placed in line with the output shaft and the differential. That section also describes one problem that might occur – the sprocket of the electric machine comes too close to the returning chain. This became very clear while making the CAD model.

The exhaust pipe is formable and left its positioning with high degree of freedom. It could not be formed totally free nor placed independently, but it was one of the simpler parts to position and shape.

The fuel tank and the super capacitors do not interact physically with the other components more than being attached to the frame. They are connected via cables or pipes respectively and could therefore be placed where there is space left. Paragraph B 9.5.1 of the FASE rules (Rules 2012) says that “All parts of the fuel storage and supply system must lie within the surface defined by the top of the roll bar and the outside edge of the four tires”. See Figure 66. Paragraph B 9.5.2 adds the requirement shielding all fuel tanks from side and rear impact collisions. The easiest way to do this is to place the fuel tank within the frame, but the paragraph permit the fuel tank to be placed outside the main frame as long as it is still protected from impact collisions and within the surface envelope shown by Figure 66.

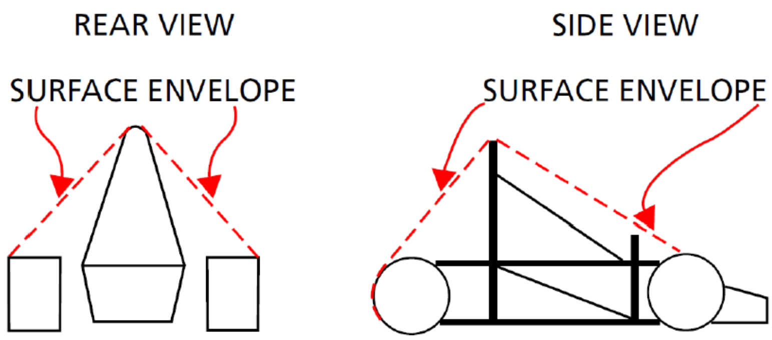


Figure 66: Envelope within the fuel system must be kept

Sections 3.2.1 Maximum torque on rear wheels while accelerating & 3.2.2 Maximum torque on rear wheels while braking show that the height of the center of gravity is important for how the weight distribution changes during acceleration and retardation. The lower the center of mass is positioned, the less change in weight distribution occurs. This is a preferred situation, since all wheels have approximately equal friction and the total friction increases. Hence another important aspect of the packaging was to place the heaviest components as low as possible. This is of course important for all components of the vehicle but since the components within this project, especially the engine, constitute a higher proportion of weight it becomes more important to place them as close to the ground as possible.

3.6.4 Mountings

When the positioning of the components was completed, mountings had to be designed. These are to be attached to the frame. Since the components and consequently their positioning did not become finished this section does not describe the final design of the mountings. The task of the mountings is to keep all components in position during driving and also in the event of a crash. Below some brief calculations are presented.

3.6.4.1 Combustion engine

The largest component, which also generates a large torque, is the ICE. These mountings become the most important. As described before the CAD-model of the ICE could not be found and the mounting points were undefined. Thus it became difficult and complex to design and make calculations on these mountings. The torque generated by the engine is known and an approximate value of the torque of the whole engine must be the same value but in opposite direction. The maximal torque of the engine is around T = 50 [Nm] as discussed in 3.3 Combustion engine so the mountings together must be able to resist this. The mounting points are not known yet but if they are supposed to be s = 100 [mm] from each other, which is an extreme case, and there are only two points the maximal force on the mountings are:

$$F = \frac{T}{s} = 500 \text{ [N]}$$

In case of a collision the retardation, and hence the forces, are the worst case to be considered. The rules (Rules 2012) do not state anything about the retardation but as described under 3.4.4 Energy storage cover the retardation is approximately 20 g. The mass of the engine is around 40 [kg]. This generates a force of:

$$F = ma = 7856 \text{ [N]}$$

This is the most critical force that the mountings must be dimensioned for and this force is divided between the different mountings. The forces generated by the engine itself are smaller.

3.6.4.2 Fuel tank

The fuel tank has a volume of five liters and will weigh approximately 6 kg. The force in case of a crash is:

$$F = ma = 1178 \text{ [N]}$$

Paragraph B 9.4.2 of the FSAE rules (Rules 2012) describes that the mountings must allow some flexibility of the fuel tank. This is because the fuel tank should not crack at high stresses – the mountings should absorb these stresses instead.

There may not be any mountings of other components attached to the fuel tank and this is because high stresses should be avoided.

3.6.4.3 Exhaust system

The exhaust system is not designed in detailed, therefore calculations of the mountings is not necessary.

3.6.4.4 Electric machine

The electric machine is smaller than the combustion engine and since it is smaller the mounting points are placed closer to each other and the resisted torque may generate large forces.

The mountings are supposed to be s = 50 [mm] from the center and the electric machine must be able to absorb and generate T = 60 [Nm] as supposed earlier (300 [Nm] after gearing of 1:5). This creates a force of:

$$F = \frac{T}{s} = 1200 \text{ [N]}$$

Also this force may be divided into different mounting points.

The approximate mass of the electric machine is 3 kg. In case of a collision the force may be:

$$F = ma = 589 \text{ [N]}$$

So in this case the torque generates the critical force.

3.6.4.5 Super capacitors

For a long time the Maxwell BCap3000F was thought to be used. Later in the project it was realized that these may have to low a voltage and smaller ones, but more, should be used instead, more about this in

3.4.3 Energy storage. The final decision was not made and therefore it becomes irrelevant to make calculations on the mountings. The principle is the same as before.

3.6.4.6 Differential

The differential may have a weight of 2.6 kg and the force in case of a collision may not be the problem.

$$F = ma = 511 \text{ [N]}$$

The differential creates a reaction force while the chain is pulling and this may create the highest torque. The maximum torque the wheels can make use of was calculated to T = 790 [Nm] in 3.2.1 Maximum torque on rear wheels while accelerating. That is also the maximum torque that the differential will be imposed by. With a sprocket of radius r = 100 [mm] the reaction force becomes:

$$F = \frac{T}{r} = 7900 \text{ [N]}$$

This is an approximation and it must be redone when all, or at least more, input parameters are known.

The conclusion is however that the differential is exposed for high forces and must have strong mountings.

3.6.5 Safety

In case of a collision all components must have strong attachments to the frame as no mounting or component should fail. The specifications given by the competition (Rules 2012) focus on the fuel system, batteries and other energy storages. All these components must be securely attached to the frame. They also establish (in paragraph B 8.9.2 of FASE) that the fuel rail has to be “securely attached to the engine cylinder block, cylinder head or intake manifold with brackets and mechanical fasteners” and in B 8.8.3 “Fuel lines must be securely attached to the vehicle and/or engine”. B 8.8.4 adds the requirement “All fuel lines must be shielded from possible rotating equipment failure or collision damage.”

In B 14.2 they also mention the securing fasteners of steering, braking, driver’s harness, and suspension but that is outside the scope of this project.

Regarding the rest of the components they all have to be securely fastened, both in case of a collision and regular use. These might be attached to the frame and the sub chapter right above includes some brief calculations on the forces.

In case of a fire the driver must be protected. All components of fuel supply, liquid cooling engine oil and energy storage must be separated away from the driver with a firewall. This is also regulated by the FSAE and FSUK rules (Rules 2012). Section B 4.5 in the FSUK rules states the details. The firewall must be UL94-V0 classified (UL94 V0, V1 and V2 - Plastmaterial, SP n.d.) and have a shear strength equivalent to a 1.6 mm thick aluminum plate. It may not have holes for seatbelts but small holes for wiring etc. may be accepted if grommets are used for sealing the pass-throughs. The last paragraph, B 4.5.5, should also be commented. Since the vehicle will have a HV system the firewall must also be electrically insulating and scratch and puncture resistant.

3.7 Control system

As mentioned in section 2.8.1 Theory, the aim for the hybrid powertrain is to obtain the lowest possible fuel consumption, without compromising the performance. The power to weight ratio has be equal or higher than conventional formula student cars.

The goal is to regenerate as much kinetic energy as possible and then use the stored energy at the correct time when accelerating. The use of an electric motor also makes it possible to operate the ICE at more efficient operating points.

The friction torque compared to the desired engine torque at constant rpm in the ICE is illustrated in Figure 67. As seen in the graph, the friction loss compared to the engine torque gets lower when the torque increases.

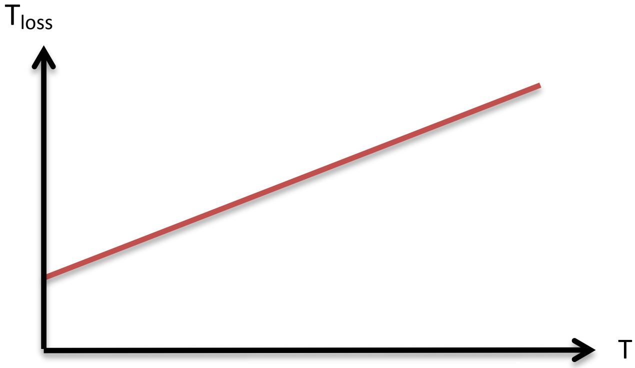


Figure 67: Engine friction losses

The same characteristics can be observed in Figure 68 (Grauers 2012). As seen in the figure, the torque and rpm span with the highest efficiency is very small.

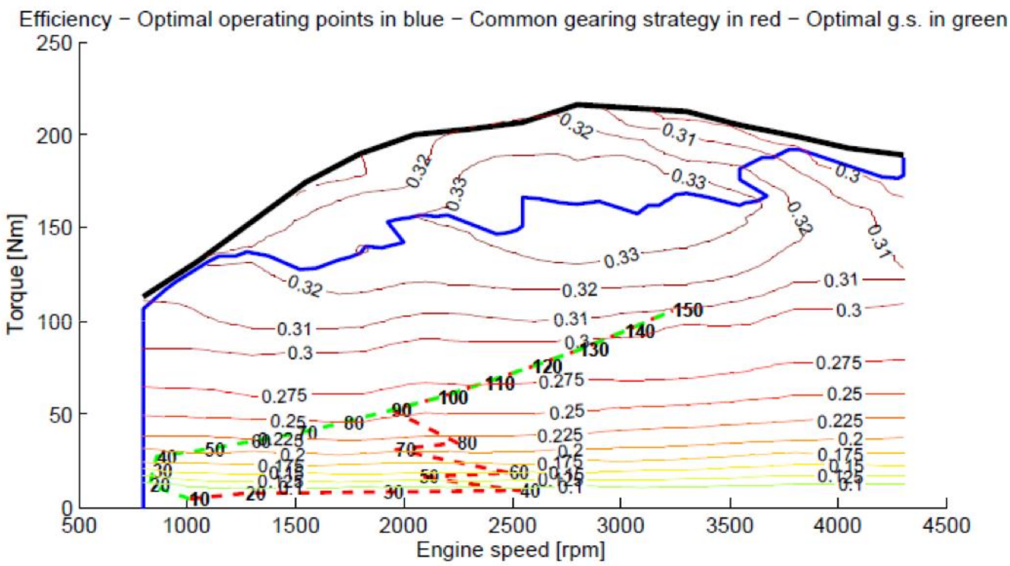


Figure 68: Engine efficiency

This correlation indicates that the highest efficiency is obtained at high torque levels, which results in a low rotation speed. The relationship between torque, rotation speed and power is described by the equation

$$P = T\omega \text{ [W]}$$

This equation shows that a low rotational speed, which causes high friction torque compared to engine torque, results in high power losses.

Figure 69 illustrates how the ICE and the electric machine affect the vehicle. For more information see Appendix B Principal Torque-speed curves. As described earlier, the efficiency increases sufficiently when the rotational speed is lowered and the torque increases. Therefore, it is preferred to operate in high gears to acquire maximum efficiency. Though, high gear ratio limits the acceleration as seen in Figure 69. The torque from the electric machine makes it possible to achieve the requested total torque in order maintain performance.

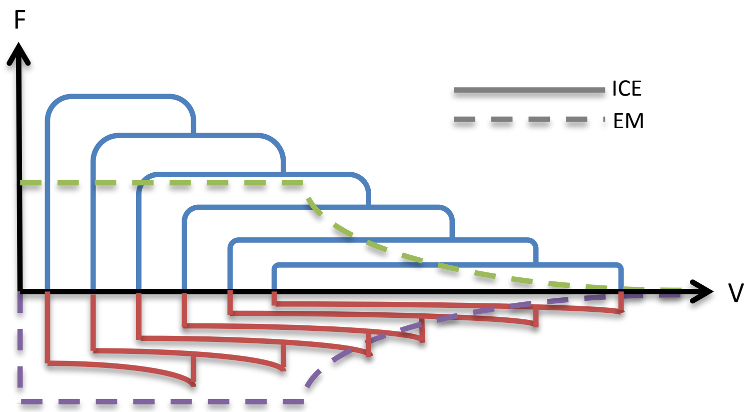


Figure 69: Traction force from the powertrain

More information regarding the chosen engine can be read in 3.3 Combustion engine.

One possibility with the use of an electric machine is to compensate for the torque loss during gear shift. Thereby, the acceleration can be smoother and a greater acceleration performance can be achieved.

The significant torque the electric machine can provide makes it possible to downsize the ICE. The lower weight of the vehicle may contribute to a major fuel consumption decrease.

4 Conclusions and discussion

In this chapter the regular conclusion and discussion chapters have been merged into one chapter because it has a discursive and advisory structure where there is no reason to divide the information to different chapters. The chapter summarizes and comments the results of the project and tries to give some advice for implementing them.

Using a single cylinder engine comes along with a loss in engine power due to the lower power of a one-cylinder engine compared to a four cylinder engine. The simulated engine has a power of about 32 kW at 6500 rpm and a maximum torque of about 50 Nm at 6000 rpm. This results in a power-to-weight ratio of 0.17 kW/kg and thus only 58% of the CFS-11 value. The performance of the vehicle when using the one-cylinder ICE only might not be competitive. An about 30% higher ICE performance is necessary. 3.5.3 Traction force showed the potential of the hybrid powertrain for the same or slightly higher performance than CFS-11 with 40 kW ICE power.

Further investigations in natural aspirated tuning have to be done in order to increase the engine performance. As shown engine tuning deals a lot with making compromises. But using more advanced techniques offers the opportunity to avoid compromises. Examples for that are a variable valve timing system and variable intake runner length. Both systems offer the potential to increase the engine performance both at lower and higher engine speeds and are nowadays already used in car industry. One can assume that the development and implementation of such advanced systems and their application is realizable due to the simpler layout of a one-cylinder engine.

Exact statements about to what extend the fuel consumption and CO_2 emissions can be lowered with the hybrid powertrain cannot be made based on the scope of this study. However both fuel consumption and CO2 emissions are to a certain extent lower since a smaller combustion engine is used.

The use of a high speed permanent magnetic electrical machine will give low weight and satisfying efficiency. The electric machine will recuperate energy during braking, the energy will be stored and then the electric machine will give an extra push out of the corners. The characteristics of the electrical machine give good opportunities to complement the combustion engine where it has limited performance. The suggested machine will produce a lot of heat on a small volume, which will require good forced air cooling. If the electrical machine has good cooling the machine can be overloaded to produce more torque than it is rated for.

For energy storage super capacitors enables a lower weight for the same power as batteries. Since the super capacitors have equal charge and discharge current it is better suited for situations where there is a high frequency of acceleration and deceleration, this means that the super capacitors can regenerate more energy from braking the vehicle. The super capacitors should be light-weighted and they could have relatively low RMS-current. It is recommended that they are connected in series and have the maximum voltage just below 80 V, the high voltage means that lower current can be used for the same power. The lower current means lower losses and less temperature increase in the super capacitors. If fewer capacitors had been used the temperature could be too high. With a greater number of super capacitors it would also possible to increase the power if it turns out that the temperature is not a problem.

The solution of having the electric motor and ICE operating at the same chain reduces the number of needed components and also the number of unique components. Similar types of differentials as used today can be used without modifications. It is important that the forces involved are studied in order to dimension the components.

Due to the cyclic nature of the combustion process, pulsations of engine speed and inertia, which alternates between a certain mean value, occurs at the crankshaft. Following the discussion in 3.3.1.4 NVH behavior, it happens that in a single-cylinder machine the rotational non-uniformity leads to torsional vibrations in the powertrain. The mass inertias of both the ICE and the electric machine are mechanically coupled to the chain. This can lead to special challenges for designing the chain. The use of torsional vibration damper and the resulting requirements for the chain has to be discussed separately.

The packaging of all components is possible and there is enough space in the rear of the frame for implementing a hybrid system. The weight of the concept of this project compared with previous solutions may with high probability be able to make lower. From this aspect it is plausible to implement the hybrid system. At least it is possible. Since many components were too complicated to model and there was no information to find about measures the CAD model could not be more than conceptual. On the other hand it turned out that both the electrical machine and the super capacitors were modeled larger than their actual size so the packaging would be easier. The CAD model of this project was made to show a concept and a possible way of position the components. Since the frame is not yet designed it could be conformed to the powertrain so the limitations are not definitive.

The control theories must be based upon the behavior and specification of especially the ICE and the electrical motor. The uses of good control strategies are very important in order to achieve a low overall energy use. Electrical safety must be very highly prioritized. The core of the control system must be a supervisory controller which must determine how to distribute the power and torque between the ICE and the electric motor.

A very important conclusion from this project is how complex a hybrid system is. The interaction between all components requires that they are matched to each other well. A small change in one area might have a large effect somewhere else in the powertrain.

An area that should be investigated is the possibility of using in-wheel motors. With one motor on each wheel and a clever control strategy the performance and handling could be improved. Motors added to the front wheel would also increase the amount of energy that could be recuperated when braking and lower the risk of changing the brake balance between the front and rear. This idea was supported by the project group and many people with knowledge of hybrid systems but it turned out to be too complicated in this early stage of CFS hybridization. But as mentioned, it would really open up new possibilities in terms of regenerating energy and enhance the performance. This project may be seen as a step in this direction.

The project was aimed to develop a hybrid powertrain for CFS-13 in order to implement a partly electric drive and go fully electric within a few years. Since the knowledge about combustion engines is greater than the knowledge about electric systems in context of CFS the thought was to implement the electric system and still have the combustion engine to back up the drivetrain. In the end of the project new apprehensions was won. It was realized that a hybrid powertrain is more complicated than a pure electric vehicle and a simpler way to go fully electric is probably not to go hybrid first.

It is recommended to continue the work with hybrid vehicles to teach the students to cooperate between different areas of knowledge.

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6 Appendix

Appendix A Hybrid systems

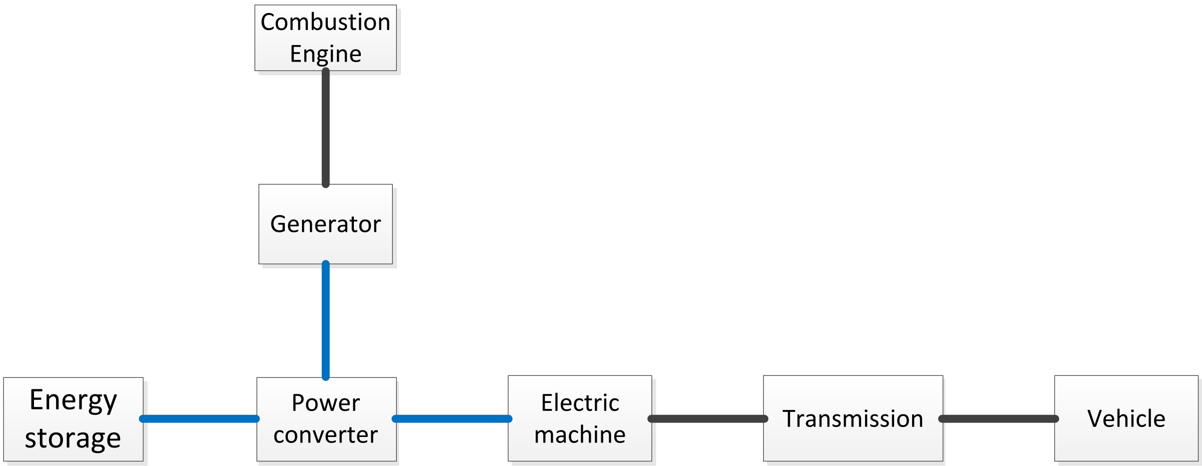


Figure 70: Flowchart over a series hybrid

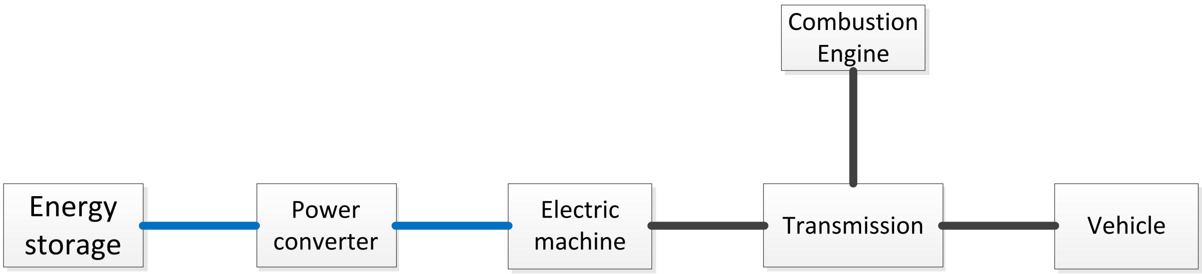


Figure 71: Flowchart over a parallel hybrid

Appendix B Principal Torque-speed curves

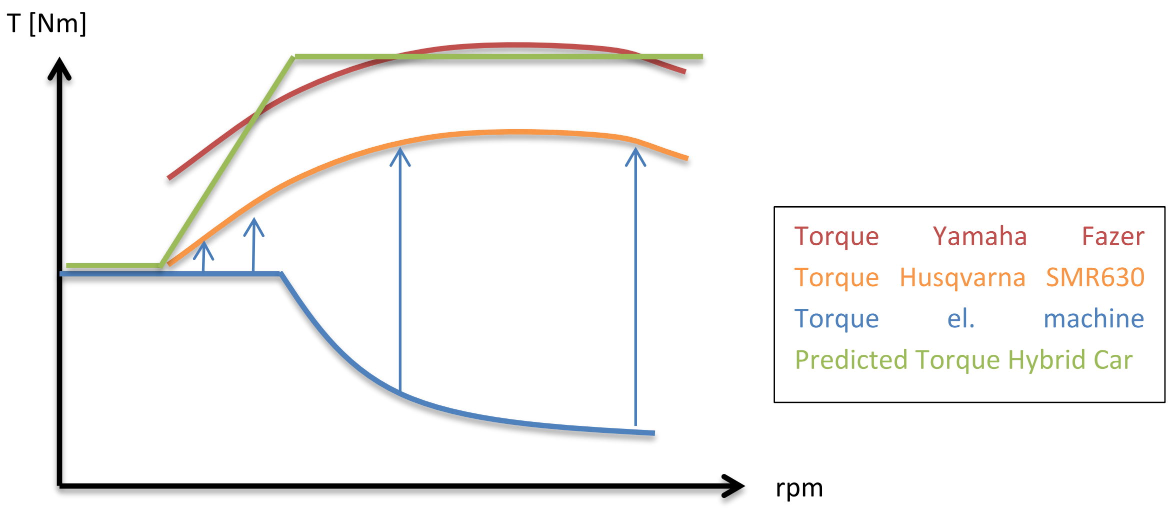


Figure 72: Principal sketch over torque curves

Appendix C Timeplan

- The group had a weekly meeting with Jonathan Rice All group members sent a weekly report to Jonathan Rice and Sven Andersson every Sunday.
- Meeting with Jonathan Rice (2012-01-25) to discuss goal statement and functional model.
- Design of the powertrain (2012-03-25).
- Meeting with former formula student team from Lund, who have built a hybrid version of a formula student car (2012-01-31), by then we had formulated a goal statement, functional model and given the members areas of responsibilities.
- Contact this or last year's formula student team to receive data regarding how the car is driven on the racetrack.
- Project time plan due 6th - 10th feb (was done 3th Feb (23:59)).
- Visit HRM Ritline (2012-02-10) and e-AAM.
- Benchmarking (2012-02-17)
- Solution model complete (2012-03-11)
- Expert panel night (2012-03-28)
- Aims regarding the hybrid systems performance.
- Finish the report 2012-05-16.
- Send report to opposing team 2012-05-16.
- Written individual opposition 2012-05-16 to 2012-05-24.
- Presentation and opposition from 2012-05-28 to 2012-05-29.

Table 18: Gantt scheme

	1	2	3	4	5	6	7	8	1	2	3	4	5	6	7	8	9	(10)
Goal statement	■																	
Functional model	■	■	■															
Aim project			■	■														
Projectplan	■	■	■	10-feb														
Investigate hybrid technology and existing systems	■	■	■	■	■	■												
Choose hybrid system		■	■															
Dimension system			■	■	■	■	■											
Choose components					■	■	■	■	■									
Simulate system						■	■	■	■	■	■	■						
Develop control logic								■	■	■	■	■	■					
Design system								■	■	■	■	■	■	■				
Presentation Expert panel											28-mar							
Prepare presentation																■	■	28-maj
Write report			■	■	■	■	■	■	■	■	■	■	■	■	■	16-maj		
Object on other group's report																■	24-maj	

Appendix D Short list of engines

Table 19: List of different engines

Manu-facturer	Modell name	Dis-placed volume [cc]	Number of cylinders / valves	Cooling	Power [kW] / rpm	Torque [Nm] / rpm	Weight [kg]	Ssp-ratio [kW / l]	Lubrication	Fuel-mixture generation and throttle body diameter [mm]	Source
Yamaha	FZ6 2008	600	4 / 16	Liquid	63	63	85	105			(CFS-12 team)
Subaru	Polaris 500	448	1 / 4	Liquid	33.6 / 8250	41.2 / 7500	48	75	Dry sump	Carburetor – 42	(Thater 2012)
Yamaha	2012 WR450	449	1 / 5	Liquid	-	-	-	-	Dry sump	Carburetor – 39	(2012 WR450F 2012)
Yamaha	2012 YZ450F	449	1 / 4	Liquid	37 (41)[14]	-	27[15]	82 / 91	-	Fuel Injection	(Corrigan, McCullough and Cunningham 2006)
Husqvarna	TE / SMR 449 2011	449.6	1 / 4	Liquid	30 / 7000	43 / 6500	~35	66.8	Wet sump	Keihin Electronic Injection Feed - 46	(TE 449 Technical Data 2012)
Husqvarna	2011 SMR511	477.5	1 / 4	Liquid	31 / 7000	44 / 6500	~35	65	Wet sump	Electronic Injection Feed - 46	(TE 511 Technical Data 2012)
Husqvarna	2011 SMR 630	600	1 / 4	Liquid	42 / 7500	57 / 6500	~40	70	Wet sump	Mikuni electronic injection feed - 45	(Datenblatt TE630 '11 n.d.)
Honda	2011 CBR250R	249.4	1 / 4	Liquid	19.4 / 8500	23.8 / 7000	-	77.8	-	PGM-FI - 38	(CBR250R '11 Teknisk info n.d.)
BMW	G450X	449	1 / 4	Liquid	37 / 7000	43.4 / 6500	-	82.4	Wet sump	Electronic Fuel Injection	(Specifications BMW G 450X n.d.)
BMW	F650	652	1 / 4	Liquid	35 / 6500	57 / 5200	-	53.7	-	-	(BMW F650 Specifications u.d.)
KTM	450 EXC	449.3	1 / 4	Liquid	37.3	-	-	83	-	-	(KTM 450 EXC (2010-current) Motorcycle Review 2012)

[14] with FSAE compliant air restrictor
[15] without intake and exhaust system

Appendix E Design targets

Sorted by type. The green fields are targets that are not included in Table 21, sorted by component.

Table 20: Design targets sorted by type

Design target	Target value		Unit		
Weight <			**kg**		
ICE	50				
Intake + Fuelsystem	5				
Exhaust system	2				
Electrical machine	11				
Energy storage	12			*Total	
Cables	2				
Power electronics	3	**85**			
Drivetrain	7				1,737
Sprocket	1,7	**93,7**			
Volume <			**mm**		
ICE	400x500x500			lxwxh	
Exhaust	20x1000			rxl	
Energy storage	400x150x200 each			lxwxh	
Electric machine	200x150			rxl	
Power electronics	200x200x100			lxwxh	
Torque >			**Nm**		
ICE	60				
Electric Machine	300*			* After gearing	
End of powertrain	790				
Effect >			**kW**		
ICE	60				
Electric Machine	15				
Supercap	15				
Energy consumption <			**l/22 km**		
Total equivalent	3				
Efficiency >			**%**		
"Wheel-to-wheel" Electric	60				
ICE	25				
Transmission	95				
Time <			**min**		
Disconnect Electric Motor	3				
Remove Electric System	60				

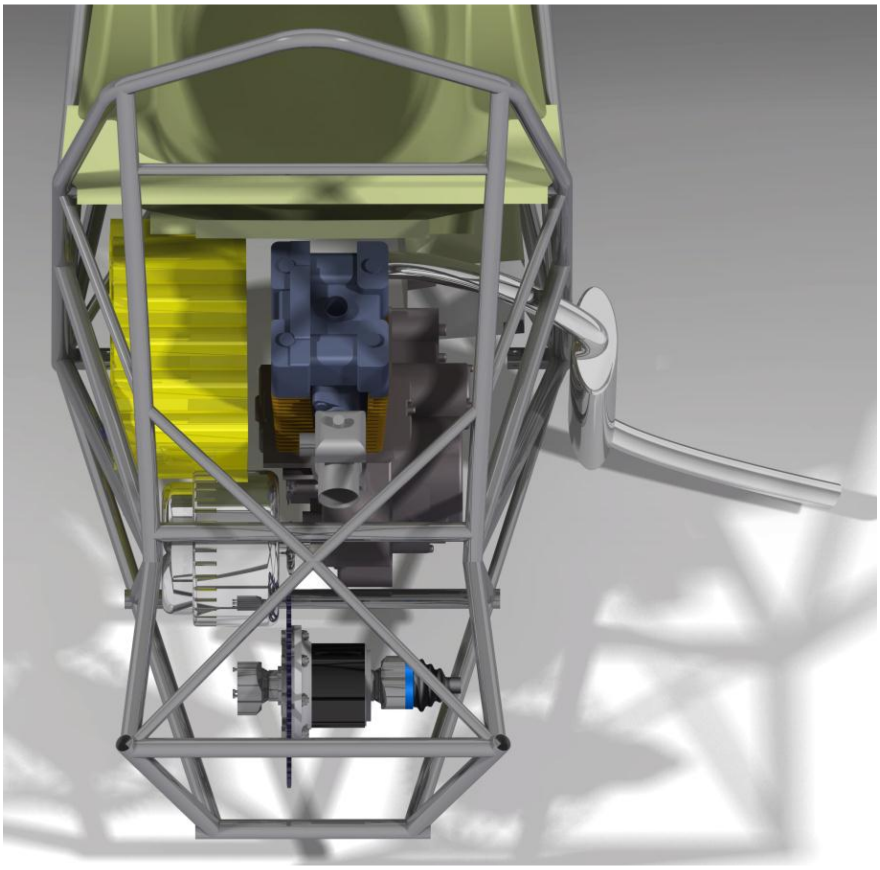
Remove Engine	160		
Other			
Noise overall <	110		dB
Noice ICE <	100		dB
Damp resistance	Yes to rain		
Possibility to fasten all components to the frame	Yes, with enough force to prevent slackening		

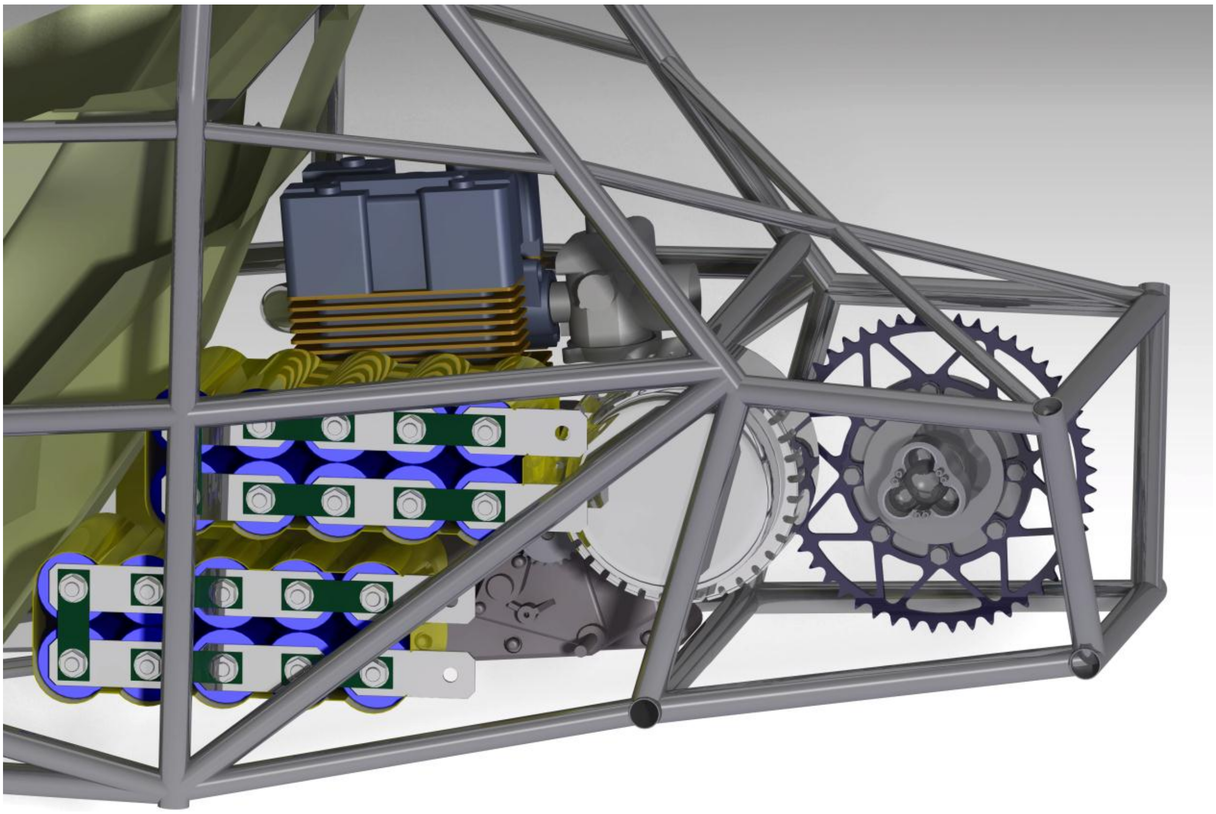
Table 21: Design targets sorted by component

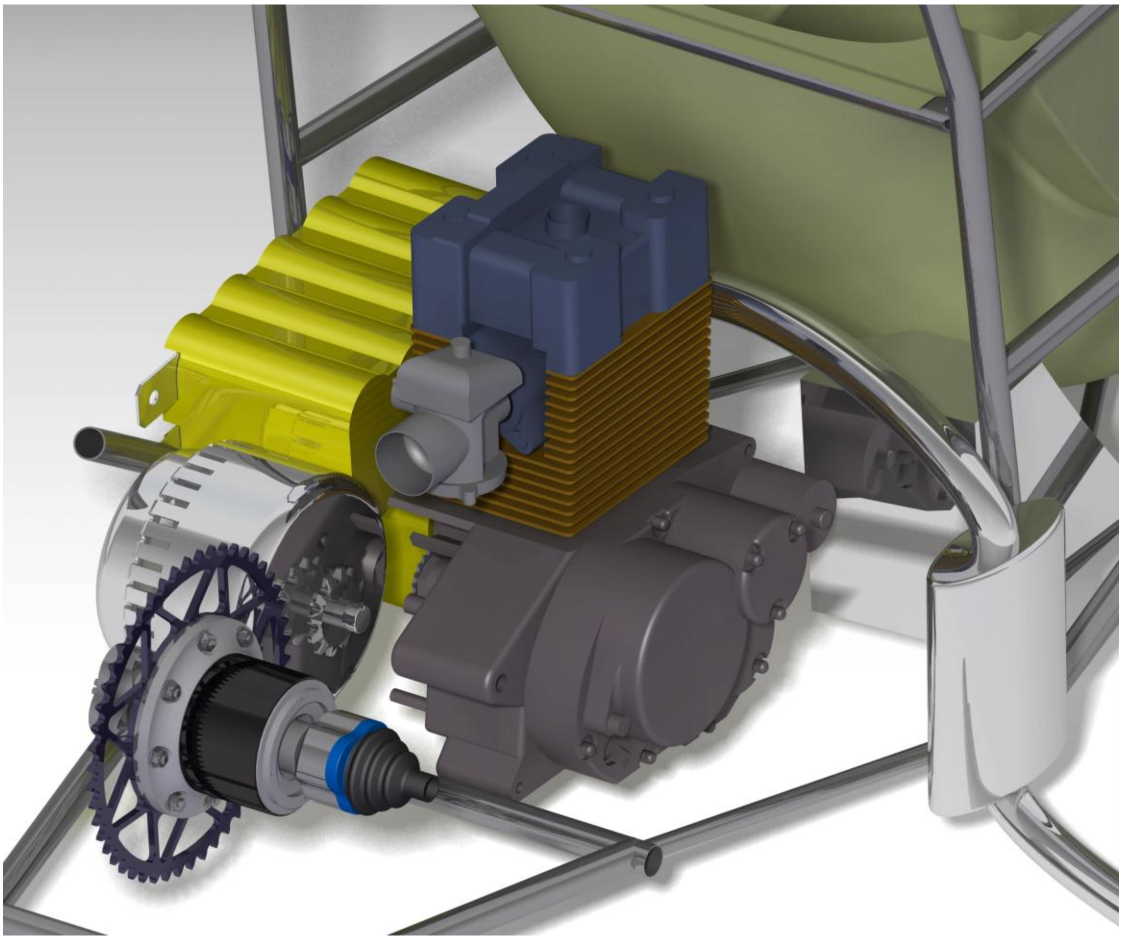
Component	**</>**	**Target value**	**Unit**	
ICE				
Weight	<	50	kg	
Volume	<	400x500x500	mm	
Torque	>	60	Nm	
Effect	>	60	kW	
Efficiency	>	25	%	
Time to remove	<	160	min	
Damp resistance		Yes to rain		
Intake and fuel system				
Weight	<	5	kg	
Restrictor	<	20	mm	
Octane		99	RON	
Damp resistance		Yes to rain		
Exhaust system				
Weight	<	2	kg	
Volume	<	20x1000		
Noise	<	100	dB	
Damp resistance		Yes to rain		
Electrical machine				
Weight	<	11	kg	
Volume	<	200x150	mm	rxl
Torque	>	300*	Nm	* After gearing
Effect	>	15	kW	
Voltage	>	48	V	
Time to remove	<	20	min	
Time to disconnect	<	3	min	
Damp resistance		Yes to rain		

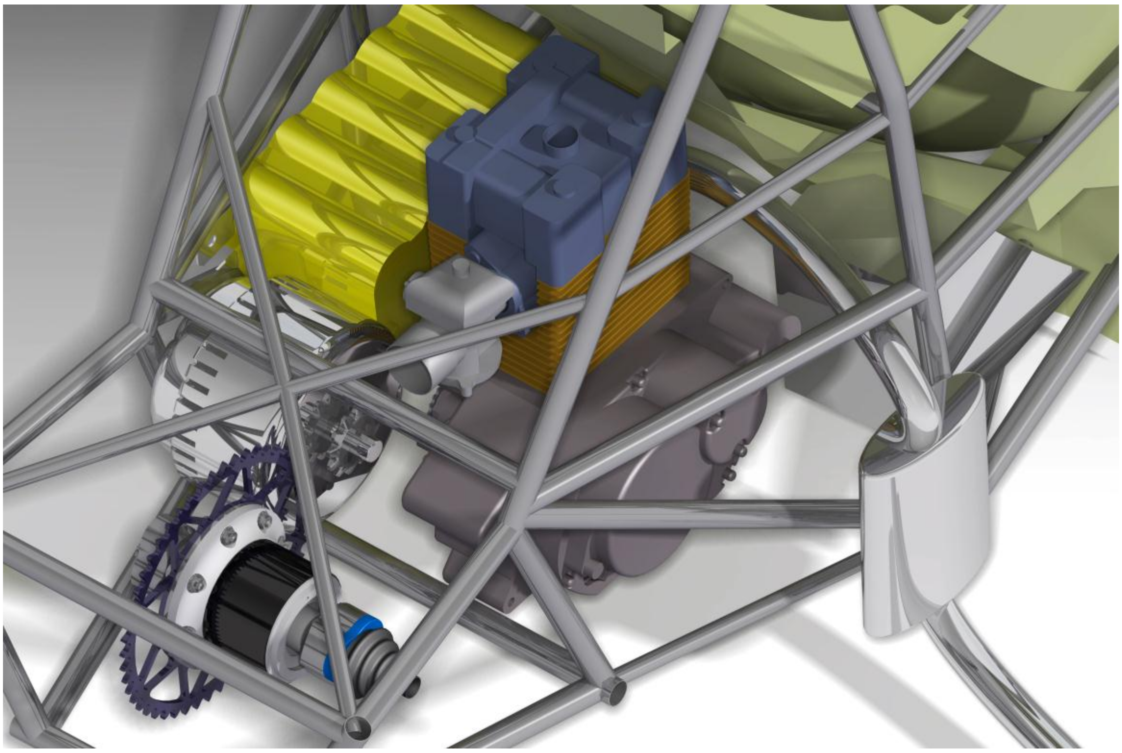
Energy storage				
Weight	<	12	kg	*Total
Volume	<	400x150x200 each	mm	lxwxh
Effect	>	15	kW	
Amount of energy	>	30	Wh	
Voltage	>	48	V	
Current	<	400	A	
Time to remove	<	15	min	
Damp resistance		Yes to rain		
Cables				
Weight	<	2	kg	
Voltage	>	48	V	
Current	>	400	A	
Time to remove	<	10	min	
Damp resistance		Yes to rain		
Power electronics				
Weight	<	3	kg	
Volume	<	200x200x100	mm	lxwxh
Voltage	>	48	V	
Current	>	400	A	
Time to remove	<	15	min	
Damp resistance		Yes to rain		
Drivetrain				
Weight	<	7	kg	
Torque	>	790	Nm	
Efficiency	>	95	%	
Damp resistance		Yes to rain		
Sprocket				
Weight	<	1,7	kg	
Torque	>	790	Nm	
Damp resistance		Yes to rain		

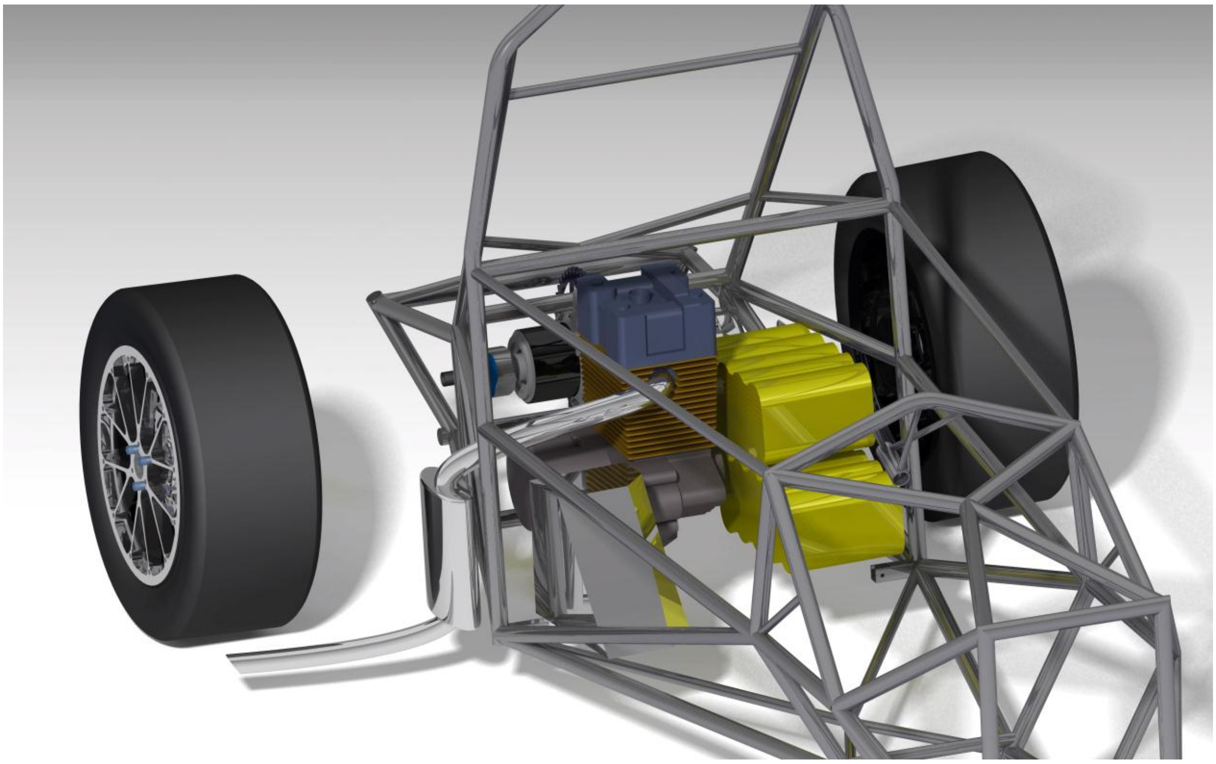
Appendix F Pictures of the packaging











Appendix G How GT-Power works

GT-Power is a one-dimensional computational fluid dynamic (CFD) program, which calculates fluid flows and heat transfer based on the corresponding gas dynamics laws. This is commonly done in such CFD Programs by discretizing components in many smaller components which have very small volumes and the fluid's scalar properties. As they are assumed to be "small" enough, they are considered to be constant. The scalar properties of a fluid are density, internal energy, pressure and temperature. Each volume also has vector properties that can be transferred across its boundaries. These properties include mass flux and fluid velocity. Figure 73 from (Gamma Technologies 2011) explains the dissimilarity between the vector and scalar properties.

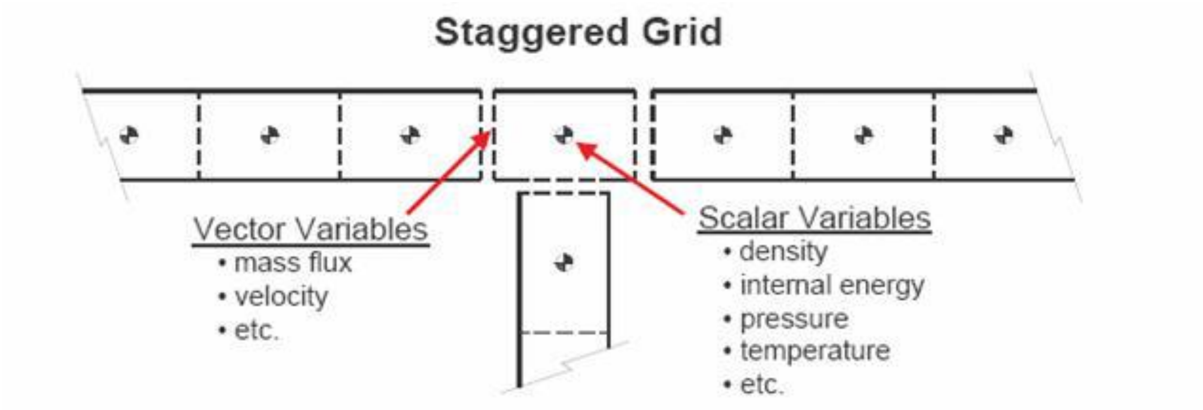


Figure 73: GT-Power Flow Manual Gamma Technologies Inc.

GT-Power determines the change in the scalar properties by computing simultaneous one-dimensional equations which is done by solving the Navier-Stokes equations, namely the energy, conservation of continuity and momentum equations. For a holistic understanding of the principle of operation of GT Power, the governing fluid mechanic equations are given in the following:

The conservation of continuity equation

$$\frac{dm}{dt} = \sum_{boundaries} \dot{m}$$

ensures the conservation of mass.

The conservation of momentum is directly related to the mass continuity equation as seen in the following equation:

$$\frac{d\dot{m}}{dt} = \frac{dpA + \sum_{boundaries}(\dot{m}u) - 4C_f \frac{\rho u|u|}{2}\frac{dxA}{D} - C_p\left(\frac{1}{2}u|u|\right)A}{dx}$$

By solving these two equations, the trapped air and residuals masses can be found

GT-Power also uses the energy equation (explicit solved):

$$\frac{d(me)}{dt} = -p\frac{dV}{dt} + \sum_{boundaries}(\dot{m}H) - hA_s(T_{fluid} - T_{wall})$$

which ensures that energy is conserved.

Further, with the knowledge of the enthalpy equation (implicit solved)

$$\frac{d(pHV)}{dt} = \sum_{boundaries} (\dot{m}H) + V\frac{dp}{dt} - hA_s(T_{fluid} - T_{wall})$$

GT-Power is able to determine the fuel dynamics as well as mass flow rates and heat transfer from a volume to the walls or other adjoining volume.

Table 22 defines the variables which are used in the presented equations.

Table 22: Declaration of variables

$\dot{m}$	boundary mass flux into volume, $\dot{m} = \rho A u$	**H**	Total enthalpy, H=e+ p / ρ
M	mass of the volume	**h**	heat transfer coefficient
V	volume	T_{Fluid}	fluid temperature
P	pressure	T_{Wall}	wall temperature
ρ	density	C_f	skin friction coefficient
A	flow area (cross-sectional)	C_p	pressure loss coefficient
A_S	heat transfer surface area	**D**	equivalent diameter
e	total internal energy (internal energy plus kinetic energy) per unit mass	**dx**	length of mass element in the flow direction (discretization length)
u	velocity at the boundary	**dp**	pressure differential acting across *dx*

Jason Meyer describes in his work 'Engine Modeling of an Internal Combustion Engine' (Meyer 2007) Page 35 the handling of GT Power as: *"GT-Power is an object oriented program with a logical user interface. To create a model, components are placed on a worksheet. Components are connected using lines to mirror the fluid paths. This process is similar to creating a block diagram or Simulink model. Several parameters must be entered into each component to specifically reflect the physical engine. To define these values, a user must double click on the object and enter the required values in a graphic user interface window."*

Appendix H Husqvarna TE 630 data

Husqvarna

HUSQVARNA TE 630 i.e. / 2011

ALLGEMEINE CHARAKTERISTIKA / GENERAL FEATURES

Radstand / wheelbase	1485 mm
Länge ges. / overall lenght	2260 mm
Breite ges. / overall width	820 mm
Höhe ges. / overall height	1240 mm
Sitzhöhe / saddle height	945 mm
Bodenfreiheit / min. ground clearance	300 mm
Gewicht (trocken) / dry weight	150 kg
Tankinhalt / fuel tank capacity	12 Liter

MOTOR / ENGINE

Art / type	Einzylinder, Viertakt / single cylinder, four stroke
Bohrung / bore	100 mm
Hub / stroke	76,4 mm
Hubraum / capacity	600 ccm
Verdichtung / compression ratio	12,4 : 1
Leistung / horse power	42 kW bei 7.500 U/min
Drehmoment / torque	57 Nm bei 6.500 U/min
Starter / starter	E-Starter / electric start
Kühlung / cooling	Flüssigkeit / liquid
Kühler / radiator	2
Ventile / valve	4 Ventile D.O.H.C.
Zündung / ignition	digitale Zündverstellung / electronic variable advance
Zündkerze / spark plug	NGK CR8EB
Elektrodenabstand / spark plug gap	0,7 - 0,8 mm / 0.027 – 0.031
Schmierung / lubrication	Druckumlaufschmierung mit 2 Ölpumpen / forced with 2 oil-pump
Gemischaufbereitung / carburation	Einspritzung Mikuni / electronic injection feed
Durchmesser / diameter	45 mm

ANTRIEB / TRANSMISSION

Kupplung / clutch	Hydraulische Nasskupplung / wet multiplate hydraulic control
Primärübersetzung / primary drive	32 : 75
Übersetzungsverhältnis / transmission ratio	2,343
Übersetzungsverhältnis 1. Gang (Zähne)	2,615 (34/13)
Internal ratio 2. Gang (Zähne)	1,812 (29/16)
3. Gang (Zähne)	1,350 (27/20)
4. Gang (Zähne)	1,091 (24/22)
5. Gang (Zähne)	0,916 (22/24)
6. Gang (Zähne)	0,769 (20/26)
Sekundärübersetzung / secondary drive	15 : 42
Übersetzungsverhältnis / transmission ratio	2,800
Kette / drive chain	5/8"x1/4" DID 520 V6

FAHRWERK / SUSPENSION

Gabel / front fork	Marzocchi 45 mm
Federweg / wheel travel front	270 mm
Federbein / shock absorber	Sachs
Federweg / wheel travel rear	320 mm

Husqvarna

Appendix I Husqvarna TE 630 engine data

TECHNICAL DATA

ENGINE
Type .... single cylinder, 4 stroke
Cooling .... Liquid with double radiator and heater fan
Bore ....3.94 in.
Stroke...............................3.01 in.
Displacement .... 36.6 cu.in.
Compression ratio 12,4:1
Starting .... electric

TIMING SYSTEM
Type double overhead camshaft chain operated; 4 valve
Valve clearance (with engine cold)
Intake.......................0.004 ÷ 0.006 in.
Exhaust......................0.006 ÷ 0.008 in.

LUBRICATION
Type ....wet sump with cartridge lobes and filters

IGNITION
Type ....Electronic, inductive with adjustable advance (digital control)
Spark plug type ....NGK CR8EB
Spark plug gap................0.027 ÷ 0.031 in.

FUEL SYSTEM
Type Electronic injection feed

PRIMARY DRIVE
Drive pinion gear .... Z 32
Clutch ring gear .... Z 75
Transmission ratio 2,343

CLUTCH
Type .... oil bath multiple disc clutch, hydraulic control

TRANSMISSION
Type.................... constant mesh gear type
Transmission ratio
(TE)
1st gear ....2.615 (z 34/13)
2nd gear.......................1.812 (z 29/16)
3rd gear ....1.350 (z 27/20)
4th gear ....1.091 (z 24/22)
5th gear ....0.916 (z 22/24)
6th gear ....0.769 (z 20/26)
(SMS)
1st gear ....2.615 (z 34/13)
2nd gear.......................1.812 (z 29/16)
3rd gear ....1.350 (z 27/20)
4th gear ....1.091 (z 24/22)
5th gear ....0.957 (z 22/23)
6th gear ....0.880 (z 22/25)

SECONDARY DRIVE
Transmission sprocket ....Z 15
Rear wheel sprocket (TE) .... Z 42
Rear wheel sprocket (SMS).................. Z 38
Transmission ratio (TE) 2.800
Transmission ratio (SMS).................. 2.533
Transmission chain dimensions ....5/8"x1/4"

FINAL RATIOS
(TE)
1st gear 17.163
2nd gear.............................. 11.894
3rd gear 8.859
4th gear 7.159
5th gear 6.016
6th gear 5.048
(SMS)
1st gear 15.529
2nd gear.............................. 10.762
3rd gear 8.016
4th gear 6.477
5th gear 5.679
6th gear 5.225

CHASSIS
Type . single frame, in circular sectioned tubes, in steel; rear chassis in squared sectioned tubes in light alloy.

FRONT SUSPENSION
Type ...."Upside-down" telescopic hydraulic front fork with advanced axle (adjustable in rebound stroke); tubes ø 45 mm
Leg axis stroke (TE).....................10.63 in.
Leg axis stroke (SMS) ....9.84 in.

REAR SUSPENSION
Type progressive with hydraulic single shock absorber (preload regulation of spring and hydraulic brake in compression and extension)
Wheel stroke (TE) ....12.6 in.
Wheel stroke (SMS)......................11.4 in.

FRONT BRAKE
Type (TE) .... fixed disc Ø 10.24 in. with hydraulic control and floating calliper
Type (SMS) ....floating disk ø12.6 in. with hydraulic command and radial fixed calliper

REAR BRAKE
Type... fixed disc Ø 8.66 in.with hydraulic control and floating calliper

RIMS
Front (TE) ....in light alloy: 1,6"x21"
Front (SMR) ....in light alloy: 3,50"x17"
Rear (TE) ....in light alloy: 2,15"x18"
Rear (SMS)...............in light alloy: 4,25"x17"

EN

TIRES
Front
(TE) ...."Metzeler" MCE Karoo or "Pirelli" MT 21; 90/90x21"
(SMS)........ "Pirelli" 58HT Diablo or "Dunlop" ZR; 120/70x17"

Rear
(TE) ...."Metzeler" MCE Karoo or "Pirelli" MT 21; 140/80x18"
(SMS)......... "Pirelli" 66H Diablo or "Dunlop" ZR; 150/60x17"

Cold tyre pressure (TE)
Front
Rider only.............................. 17 psi
Rider and passenger .... 21,5 psi
Rear
Rider only........................... 21,5 psi
Rider and passenger .... 25,5 psi

Cold tyre pressure (SMS)
Front
Rider only........................... 25,5 psi
Rider and passenger .... 28,5 psi
Rear
Rider only........................... 28,5 psi
Rider and passenger .... 31,5 psi

DIMENSION, WEIGHT, CAPACITY
Wheelbase
(TE) ....58.46 in.
(SMS)...............................58.86 in.
Overall length
(TE) ....88.98 in.
(SMS)...............................85.43 in.
Overall width
(TE) ....32.28 in.
(SMS)...............................33.07 in.
Overall height
(TE) ....48.82 in.
(SMS)...............................46.85 in.
Saddle height
(TE) ....37.20 in.
(SMS)...............................35.83 in.
Minimum ground clearance
(TE) ....11.81 in.
(SMS)................................9.45 in.

Kerb weight, without fuel.
(TE) ....328.5 lb.
(SMS)...............................332.9 lb.

Fuel tank capacity
reserve included ..... 2.63 Imp. Gall. / 3.17 U.S. Gall.
Reserve fuel (warning light goes on). 2.2 Imp Qt./2.64 U.S. Qt.
Coolant capacity .....2.18 Imp. Pints / 2.53 U.S. Pints
Transmission oil
Oil and oil filter replacement 1.58 Imp. Qt. / 1.90 U.S. Qt.
Oil replacement.........1.40 Imp. Qt. / 1.68 U.S. Qt.

TABLE FOR LUBRICATION, SUPPLIES
Engine, gearbox and primary drive lubricating oil....... ....CASTROL POWER 1 RACING 10W-50
Engine coolant ....CASTROL MOTORCYCLE COOLANT
Brake system fluid.......CASTROL RESPONSE SUPER DOT 4
Clutch fluid.................. CASTROL FORK OIL 10W
Grease lubrication ....CASTROL LM GREASE 2
Final drive chain lubrication .CASTROL CHAIN LUBE RACING
Front fork oil ....CASTROL SYNTHETIC FORK OIL 5W
Oil for rear shock absorber ....CASTROL SYNTHETIC FORK OIL 5W
Electric contact protection . CASTROL METAL PARTS CLEANER
Fillers for radiator ....AREXONS TURAFALLE LIQUIDO

Appendix J Husqvarna TE 630 service interval data

TE - SMS 630	SCHEMA DI MANUTENZIONE PERIODICA (DA EFFETTUARE PRESSO IL CONCESSIONARIO HUSQVARNA) SCHEDULED MAINTENANCE CHART (TO BE CARRIED OUT AT THE HUSQVARNA DEALER) SCHÉMA D'ENTRETIEN PÉRIODIQUE (À EFFECTUER AUPRÈS D'UN CONCESSIONNAIRE HUSQVARNA)				
	MOTORE - ENGINE - MOTEUR				
	DOPO I PRIMI 1000 Km AFTER THE FIRST 1000 Km APRÈS LES 1000 PREMIERS km	OGNI 5000 Km EVERY 5000 Km TOUS LES 5000 Km	OGNI 10000 Km EVERY 10000 Km TOUS LES 10000 Km	OGNI 15000 Km EVERY 15000 Km TOUS LES 15000 Km	OGNI 20000 Km EVERY 20000 Km TOUS LES 20 000 Km
VALVOLE - VALVES - SOUPAPES	C (*)	C (*)	C (*)	C (*)	C (*)
BILANCIERI DI ASPIRAZIONE/SCARICO - INTAKE/EXHAUST ROCKER ARMS CULBUTEURS D'ADMISSION ET D'ÉCHAPPEMENT			C		C
CATENA DI DISTRIBUZIONE - TIMING CHAIN - CHAÎNE DE DISTRIBUTION					S
PATTINO CATENA DI DISTRIBUZIONE - TIMING CHAIN SLIDER PATIN DE LA CHAÎNE DE DISTRIBUTION					S
INGRANAGGIO CONDOTTO DISTRIBUZIONE - TIMING DRIVEN GEAR ENGRENAGE CONDUIT DE DISTRIBUTION					S
CANDELA ACCENSIONE - SPARK PLUG - BOUGIE D'ALLUMAGE		P	S	P	S
PIPETTA CANDELA - SPARK PLUG CAP - PIPETTE DE BOUGIE		C	C	C	C
OLIO MOTORE - ENGINE OIL - HUILE MOTEUR	S	S	S	S	S
FILTRO ASPIRAZONE OLIO MOTORE - ENGINE OIL INTAKE FILTER FILTRE D'ASPIRATION DE L'HUILE MOTEUR	P		P		P
CARTUCCIA FILTRO OLIO MOTORE - ENGINE OIL FILTER CARTRIDGE CARTOUCHE DU FILTRE DE L'HUILE MOTEUR	S		S		S

TE - SMS 630	SCHEMA DI MANUTENZIONE PERIODICA (DA EFFETTUARE PRESSO IL CONCESSIONARIO HUSQVARNA) SCHEDULED MAINTENANCE CHART (TO BE CARRIED OUT AT THE HUSQVARNA DEALER) SCHÉMA D'ENTRETIEN PÉRIODIQUE (À EFFECTUER AUPRÈS D'UN CONCESSIONNAIRE HUSQVARNA)				
	MOTORE - ENGINE - MOTEUR				
	DOPO I PRIMI 1000 Km AFTER THE FIRST 1000 Km APRÈS LES 1000 PREMIERS km	OGNI 5000 Km EVERY 5000 Km TOUS LES 5000 Km	OGNI 10000 Km EVERY 10000 Km TOUS LES 10000 Km	OGNI 15000 Km EVERY 15000 Km TOUS LES 15000 Km	OGNI 20000 Km EVERY 20000 Km TOUS LES 20 000 Km
PARASTRAPPI FRIZIONE - CLUTCH CUSH DRIVE DAMPER SILENTBLOC DE L'EMBRAYAGE			C		C
FILTRO ARIA - AIR FILTER - FILTRE À AIR	P	P/S (**)	S	P/S (**)	S

A

C: CONTROLLARE - CHECK - CONTRÔLER
C (*): CONTROLLARE GIOCO - CHECK CLEARANCE - CONTRÔLER LE JEU
P: PULIRE - CLEAN - NETTOYER
P/S (**): PULIRE O SOSTITUIRE (in funzione alle condizioni di impiego del motociclo)
CLEAN OR CHANGE (depending on the conditions of use of the motorcycle)
NETTOYER OU REMPLACER (en fonction des conditions d'utilisation de la moto)
S: SOSTITUIRE - CHANGE - REMPLACER

TE - SMS 630	WARTUNGSPLAN (DIESE ARBEITEN MÜSSEN BEI EINEM HUSQVARNA-VERTRAGSHÄNDLER AUSGEFÜHRT WERDEN) ESQUEMA DE MANTENIMIENTO PERIÓDICO (PARA REALIZAR EN UN CONCESIONARIO HUSQVARNA)				
MOTOR - MOTOR					
	NACH DEN ERSTEN 1000 km DESPUÉS DE LOS PRIMEROS 1000 Km	ALLE 5000 km CADA 5000 Km	ALLE 10000 Km CADA 10000 Km	ALLE 15000 Km CADA 15000 Km	ALLE 20000 Km CADA 20000 Km
VENTILE - VÁLVULAS	C (*)	C (*)	C (*)	C (*)	C (*)
SCHWINGHEBEL AM EINLASS/ AUSLASS - BALANCINES DE ASPIRACIÓN/ESCAPE			C		C
VENTILSTEUERKETTE - CADENA DE DISTRIBUCIÓN					S
KETTENSPANNERAUFLAGE VENTILSTEUERKETTE - PATÍN CADENA DE DISTRIBUCIÓN					S
ANGETRIEBENES ZAHNRAD VENTILSTEUERUNG ENGRANAJE CONDUCIDO DE DISTRIBUCIÓN					S
ZÜNDKERZE - BUJÍA DE ENCENDIDO		P	S	P	S
ZÜNDKERZENSTECKER - PIPETA BUJÍA		C	C	C	C
MOTORÖL - ACEITE MOTOR	S	S	S	S	S
MOTORÖL-ANSAUGFILTER - FILTRO DE ASPIRACIÓN ACEITE MOTOR	P		P		P
MOTORÖL-FILTERPATRONE - CARTUCHO DEL FILTRO ACEITE MOTOR	S		S		S

TE - SMS 630	WARTUNGSPLAN (DIESE ARBEITEN MÜSSEN BEI EINEM HUSQVARNA-VERTRAGSHÄNDLER AUSGEFÜHRT WERDEN) ESQUEMA DE MANTENIMIENTO PERIÓDICO (PARA REALIZAR EN UN CONCESIONARIO HUSQVARNA)				
MOTOR - MOTOR					
	NACH DEN ERSTEN 1000 km DESPUÉS DE LOS PRIMEROS 1000 Km	ALLE 5000 km CADA 5000 Km	ALLE 10000 Km CADA 10000 Km	ALLE 15000 Km CADA 15000 Km	ALLE 20000 Km CADA 20000 Km
REISSSCHUTZ KUPPLUNGE - DISPOSITIVO ANTIVIBRACIÓN EMBRAGUE			C		C
LUFTFILTER - FILTRO AIRE	P	P/S (**)	S	P/S (**)	S

A

C: KONTROLLIEREN - COMPROBAR
C (*): SPIEL KONTROLLIEREN - COMPROBAR EL JUEGO
P: REINIGEN - LIMPIAR
P/S (**): REINIGEN ODER AUSWECHSELN (abhängig von den Einsatzbedingungen des Motorrads)
LIMPIAR O SUSTITUIR (según las condiciones de uso de la motocicleta)

Appendix K Solution model

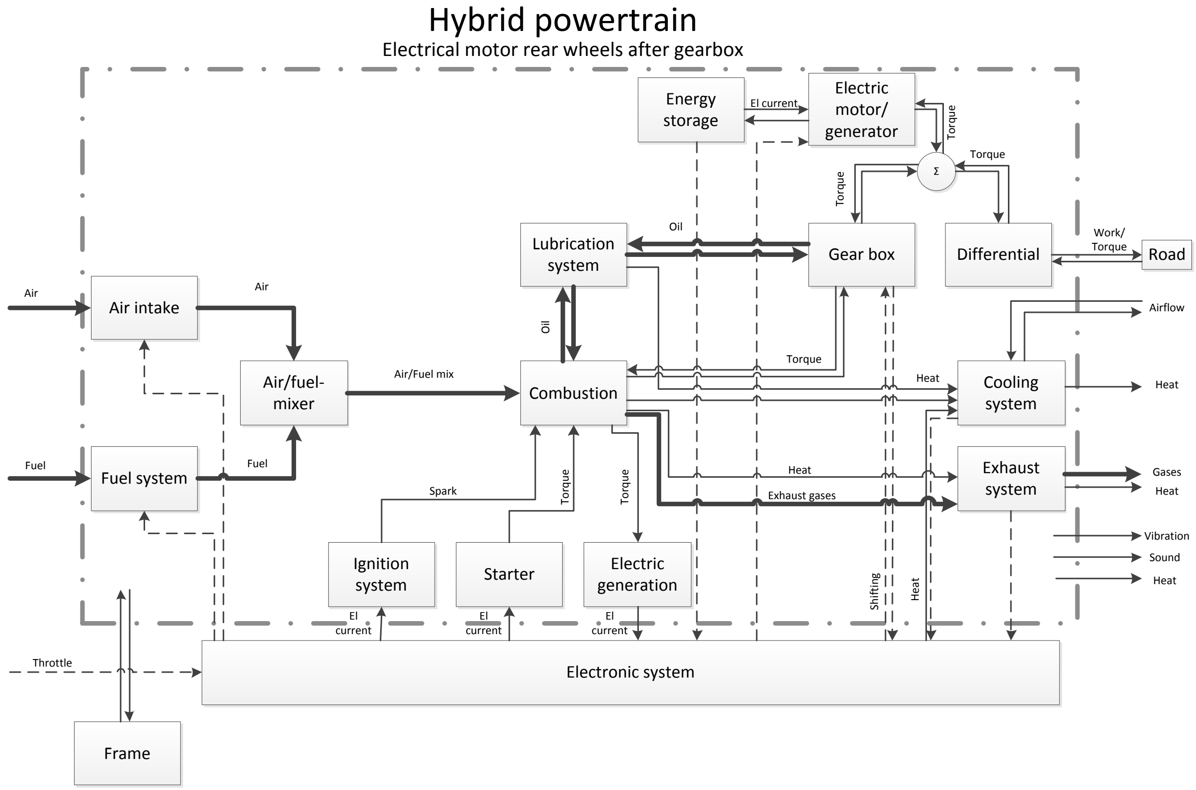


Figure 74: Solution model

Appendix L Test and calculations on model aircraft machine

What we want to know

- Current/voltage in => Torque /rpm out
- Efficiency and heat
- Overload
- Cooling required
- Back EMF
- Torque needed to rotate it when it is turned off.
- Physical dimensions, weight (complete and inertia), position of bearing etc.

Initial difficulties

Due to the limitations in the available equipment, the electrical machine cannot be overloaded. The brake that the machine is connected to, the power converter and the power supplies are limiting factors. The power that goes in to the power converter will be measured, not the power that goes in to the electrical machine. This is because it is difficult to measure the pulse width modulated voltage and current in a suitable way. By measuring in this way the losses in the controller will also be included. The power electronics used is not constructed to run the machine at low speed and high torque.

The main purpose of the test; “Can the electrical machine be cooled with a cooling system that fits in the car when it is running at its rated load” can still be fulfilled, even though some of our desired test cannot be conducted.

Testing

The torque and rpm measuring device was tested using a tachometer and a known weight to see that it gives correct measurements. The power supply was tested using an ampere meter to make sure that the power supply showed the correct current. An oscilloscope was used to measure back EMF.

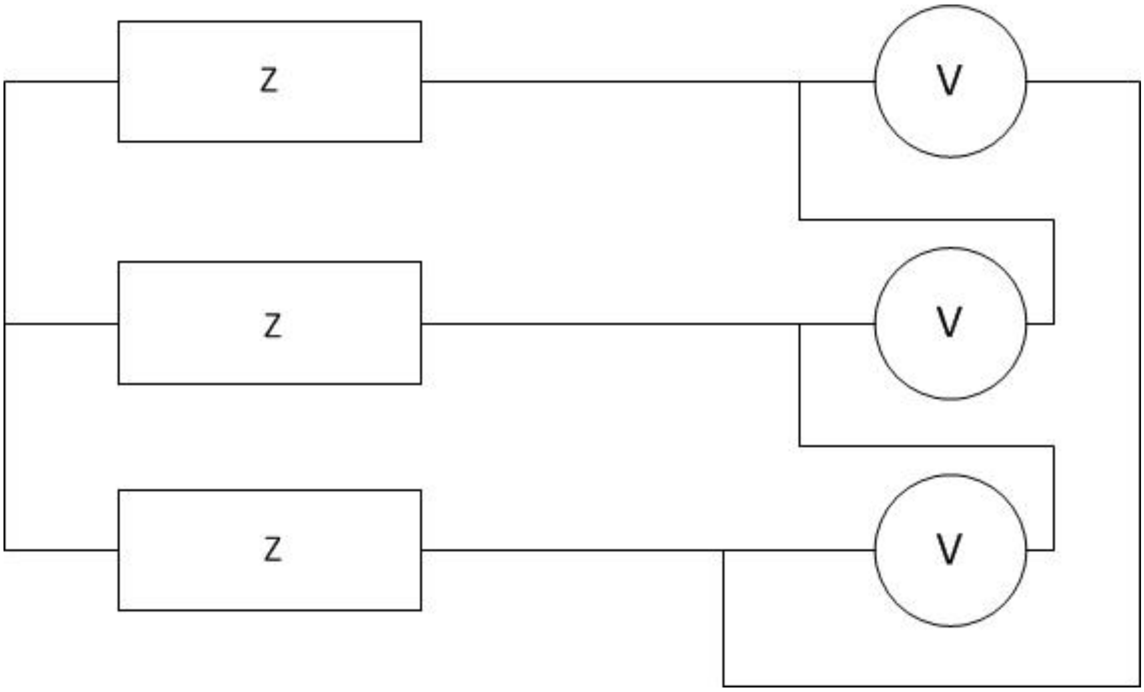


Figure 75: Voltage measurement

To measure the back EMF the machine was rotated with an electric screwdriver and differential probes was used to measure the voltage output.

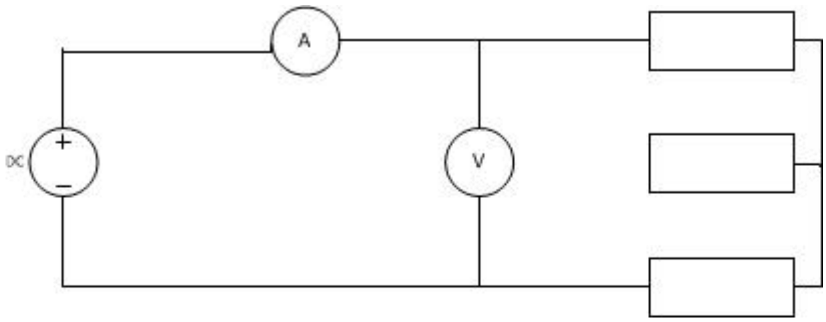


Figure 76: Resistance measurement

The electrical machine ran at two different voltages (30 V and 50 V). The machine got up to speed and the load was applied via the brake. The power electronics was set to deliver as high speed and torque as possible, the machine could be regulated using the power electronics. The temperature was measured during the maximum load that could be achieved at 30 V. The temperature was measured every 15 s until the machine was considered too hot to continue the test.

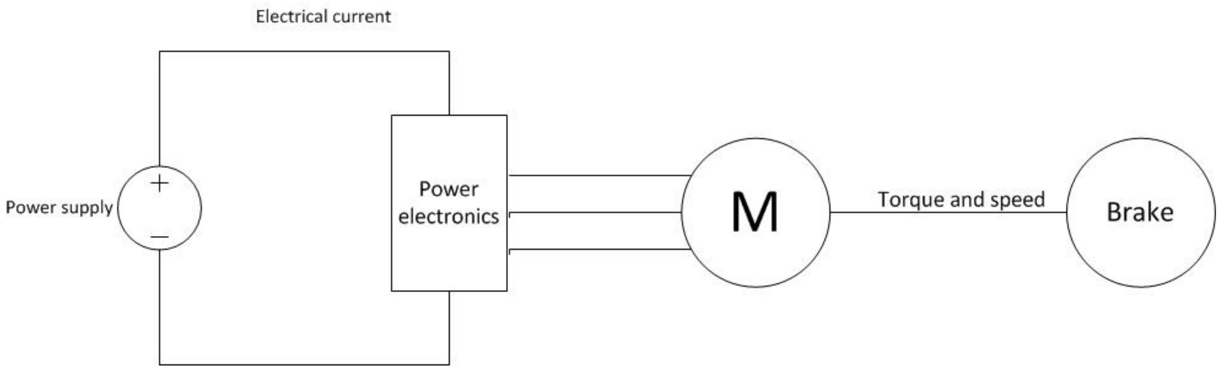


Figure 77: Machine setup

Probably due to a voltage spike from the power supply a transistor in the control circuit broke, “lesson learned” the electronics is very sensitive to overvoltage.

Due to lack of time and power supply availability, it was not possible to conduct all of the desired tests. How much torque required to rotate the machine and the disassembly was not conducted.

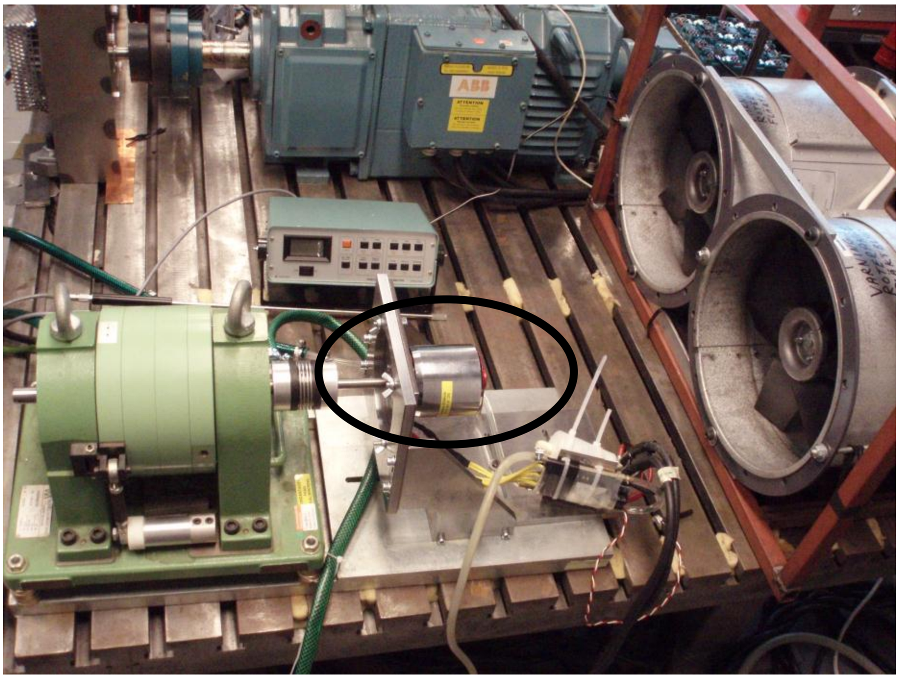


Figure 78: Electric machine testing

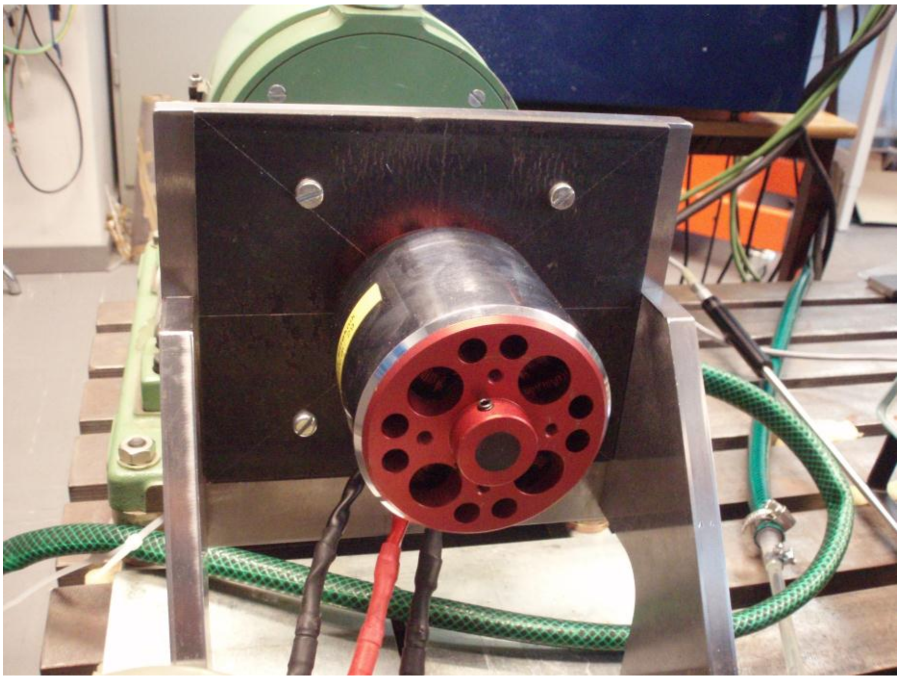


Figure 79: Electric machine testing

Results

The results from the resistance measurement showed that the resistance in this electrical machine is about 46 mΩ.

The Figure 80 shows the primary voltage for the electrical machine when it is run in 856 rpm. That gives approximately 140 rpm/V.

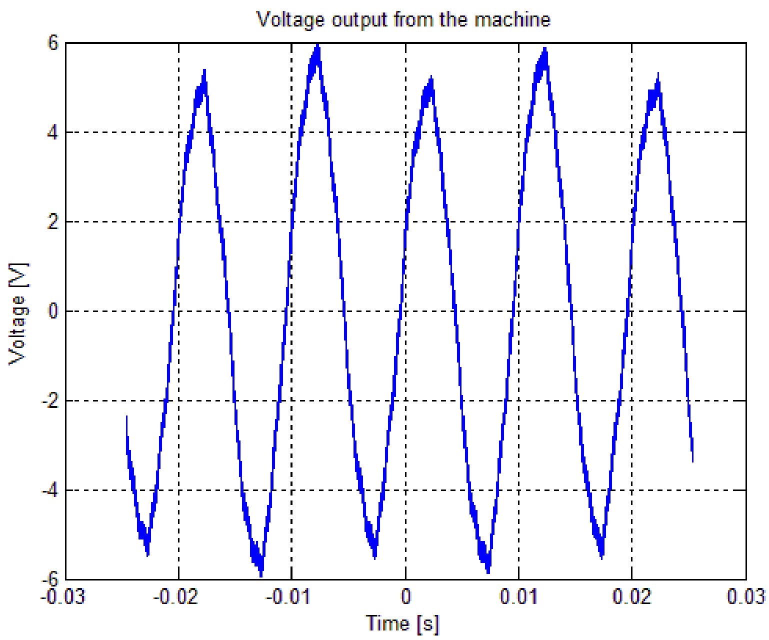


Figure 80: Primary voltage from back EMF-test

As it can be seen in the Figure 81, the relation between the current and the torque of the electrical machine is linear. The brake could not deliver more than 7.2 Nm at the rotational speed.

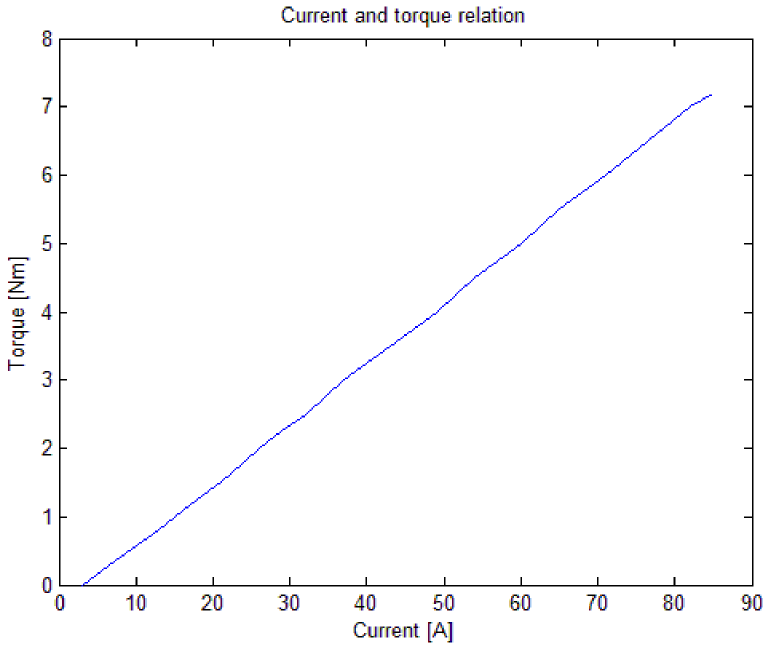


Figure 81: Relation between current and torque

The temperature of the electrical machine did not increase as quickly as thought. It should be noted that the temperature was 38°C at the beginning of the test and that the electrical machine was run continues, in the car it will be stopped during short intervals. The cooling used during this test a fan at approximately 0.5 meters distance, which had a wind speed of 20 m/s at its output. Due to the limits of the brake and voltage supply, the electrical machine was not loaded at its maximal torque. The load was: 85 A, 30.6V and 2680 rpm and 7.2Nm

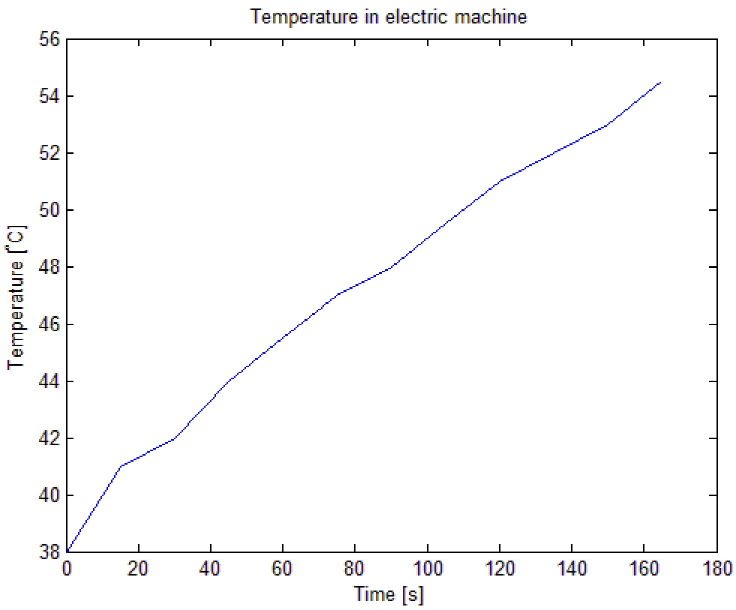


Figure 82: Temperature in the electric machine during heat test

The efficiency of the electrical machine was high, over 80 % in most cases. The low efficiency under low load is because the idle losses and friction is relatively higher as they do not increase so much when the load is increased. When the electrical machine is run at 50 V it seems to have higher efficiency at high load and 30 V is more efficiency during low load.

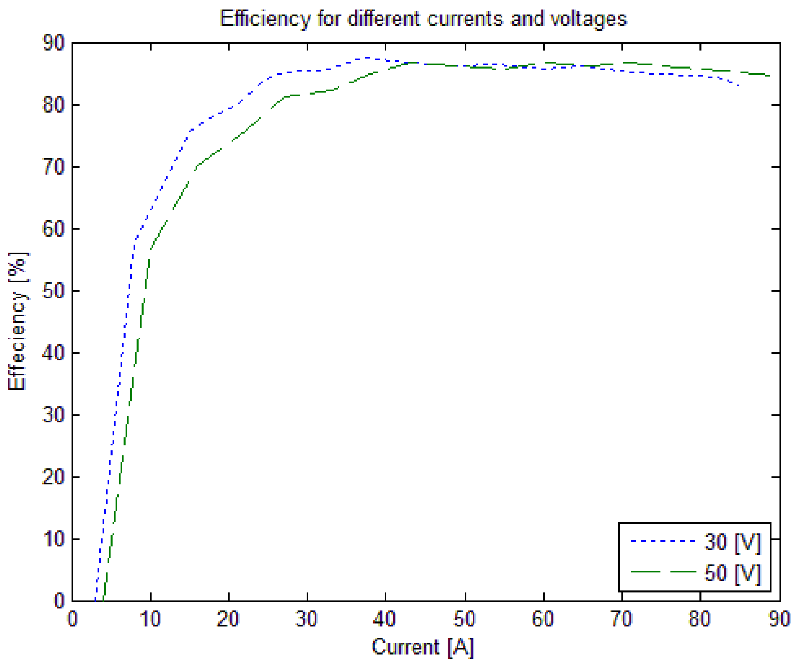


Figure 83: Efficiency for the electric machine and power electronics

Mechanical calculations on model aircraft motor

An area where high performance electric machines are used is in radio controlled model aircrafts. These motors have a very high power-to-weight ratio and can replace engines at over 100cc in those applications.

From a mechanical point of view is it important to calculate the forces on the motor, since they would be different in a race car application compared to driving the propeller in an aircraft. The force from a propeller acts in the direction of the axle and transmits the force through the mounting points.

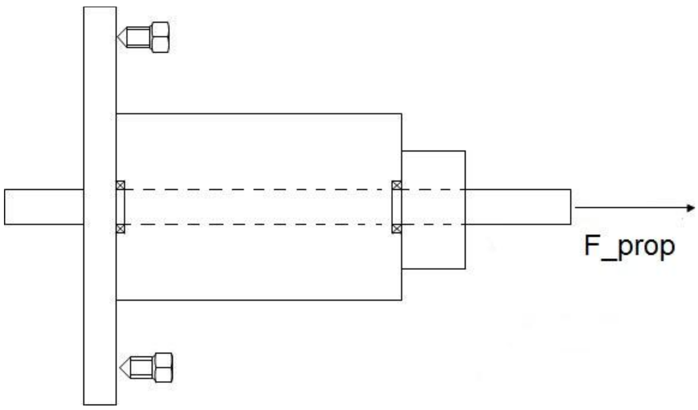


Figure 84: Machine mounted in aircraft

When connected directly in the driveline the forces would be different. The forces in the chain would cause the axle to bend and a risk of fatigue occurs.

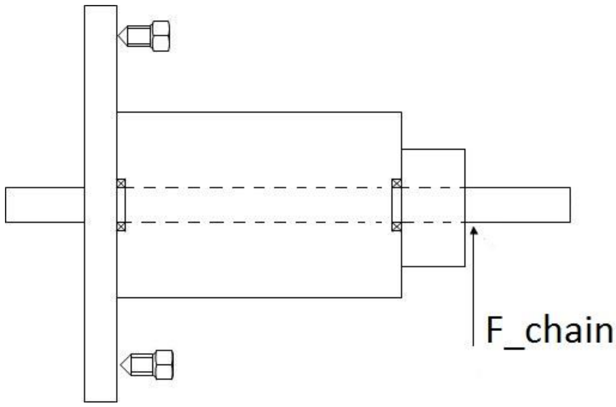


Figure 85: Machine mounted in car

The motor has a solid steel axle in the middle and an outer aluminum “axle” that will support.

With the assumption of solid aluminum, the risk of fatigue can be calculated as follows:

$$F_{brake} = \frac{M_{brake}}{r_{diff}} = \frac{300}{0.13} \approx 2300\ [N]$$

$$W_b = \frac{\pi d^3}{32} \qquad [KTH]$$

$$\sigma_b = \frac{M_b}{W_b} = \frac{F(B-D)}{W_b} = 22\ [MPa] \quad [E.M\ (8)]$$

$$K_f = 1 + q(K_t - 1)$$

With ρ = 0.1mm and aluminium as material, q can be determined with KTH (25.4) to:

$$A_1 = 0.5\ [mm]$$

$$q = \frac{1}{1+\sqrt{\frac{A_1}{\rho}}} = 0.31$$

$$\frac{D}{d} = \frac{118}{40} = 2.95 \qquad \frac{\rho}{d} = \frac{0.1}{40} = 0.025$$

KTH Table 33.4:

$$K_t = 2.9$$

E.M [51] states that:

$$\sigma_{dim} = K_f \sigma_{nom} = K_f \sigma_b = 35\ [MPa]$$

Evertsson [51] also states that:

$$\sigma_{N,red} = k_s k_d \sigma_N$$

From Evertsson [55], [56], the fact that the axle is polished and σ_B = 200MPa and d = A = 40 [mm]

$$k_s = 0.95$$

$$k_d = 0.85$$

The limit for fatigue due to Evertson 5.1-3 is estimated to 43 % of σ_B

$$\sigma_u = 0.43\sigma_B$$

This yields

$$\sigma_{N,red} = k_s k_d \sigma_u = 69.4\ [MPa]$$

$$\sigma_{N,red} = 69.4\ [MPa] > \sigma_{dim} = 35\ [MPa]$$

These numbers indicates that the axle will not brake due to fatigue. Aluminum is not behaving the same way as steel, but since the axle have a solid steel center this will help strengthen it.

The heat generated from the motor is also important to take into consideration. When mounted directly behind a propeller and flying, it will have all the air blowing over it and is therefore kept cool. In a racecar it will most likely be mounted in an area with high temperatures from the engine and exhaust system. The amount of passing air to cool it will also be restricted.

Conclusion

The electric machine and its controller had higher efficiency (83-85%) than what was expected.

The machine did not heat up as fast as we thought, but it still needs a powerful forced air cooling system. The machine ran at 2.5 kW and blew air at 20 m/s over it. The machine got to about 55 degrees Celsius after 2 min and 45 sec (Datasheet), after it was turned off and stopped cooling it the temperature rose to about 68 degrees Celsius at the windings.

This machine and similar machine are significantly lighter than the other machines that was considered.

Appendix M MatLab code

Electric system

```
close all;
clear all;
clc;

% Super capacitor data, assumes supercaps are serial
M_cap = 0.160;  % Mass
R_cap = 0.8e-3;  % Resistance
C_cap = 650;  % Capacitance
V_cap = 2.7;  % Voltage
C_th = 190; %Thermal capacitance
R_th = 6.5; %Thermal resistance
T_amb = 25; %Ambient temperature

% Number of supercaps in system
num_cap = 29;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% All parameters that we can change

% Gearing, from the el-machine to driveshaft
% To not use the gearing, set it to 1 and the gear ratio between the
% machine and the driveshaft will be 1:1
G_elm = 3;

% Efficiency of the electrical machine
eff = 0.74;

Discharge = 0.8; % level of discharge where we still want to be
% able to reach maximum torque at rated speed
% If the discharge is too low then the internal resictance of the supercaps
% will be higer then the disered R and Ra will be negative

% Speed where field weakening should start (in kph)
Speed = 56;

% Overload of machine
overload = 1;

% Negative overload of machine
underload = 1;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%General vehicle data
% Gravitaional constant [m/s^2]
g = 9.82;
% Mass
m=273+M_cap*num_cap; %kg
% Roling resistance of the wheel on road
f=0.01;
% Air density
rho=1.275; %[kg/m^3]
% Drag coefficient
C_d=0.35;
% Frontal Area
A=1.25; %[m^2]
```

```
% Radius of the wheel
r = (21*2.54*10^(-2))/(2); %[m]

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Load velocity profile, assumes input is filtered
load v_race.mat
vprofile = v_complete_race;
% Sampling interval
Ts = 0.01;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Compute total force, torque and speed
% Compute air resistance
F_air=vprofile.^2.*(0.5*rho*A*C_d);
% Rolling resistance
F_r = m*g*f+vprofile*0;
for i = 1:length(vprofile)
    if vprofile(i) < 0.01
        F_r(i) = 0;
    end
end

% The acceleration force F = ma
F_acc = m.*diff(vprofile)./0.01;
F_acc(length(vprofile)) = 0;
% Total force
F_tot = F_acc+F_air+F_r;

% Torque on the driveshaft
T_ds = F_tot.*r;
% Speed on the driveshaft
w_ds = 2*pi*vprofile./(2*pi*r); % 2*pi*V/O


%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Data of the electrical system
% Given data
w_elm = w_ds.*G_elm;

% Internal resistance of the supercap
Rc = R_cap*num_cap;
% Capacitance of the supercap
C = C_cap/num_cap;
% Initial and peak voltage of the supercap
V_cap = V_cap*num_cap;

% Estimated data

% Lowest voltage of the supercap that should put the system on the border
% of field weakening at w_ds=w_max (D=1 at this point)
Vc_min_fw = V_cap*Discharge;

%Requested torque before gearing
T = 130/G_elm;

% Maximum speed of the machine before field weakeing should start
w_max = Speed*G_elm/(r*3.6);

% Electrical power that is needed to drive the machine at max power
```

```
P_elm = (w_max*T)/eff;

% Current through the machine to achieve maximum torque (rated current)
Ia_rated = P_elm/Vc_min_fw;

% Maximum current for the converter
Ia_max = overload*Ia_rated;

% Minimum current for the converter
Ia_min = -underload*Ia_rated;

% Internal resicanace of el-machine
R = (P_elm.*(1-eff))/(Ia_rated^2);
Ra = R-Rc;
if Ra < 0
    disp('WARNING Ra < 0')
end

% Flux in the electrical machine
psi_max = (Vc_min_fw)/(2*w_max)+sqrt(((Vc_min_fw)/(2*w_max)).^2-
T*((Rc+Ra)/(w_max)));

%Power on the machines output shaft
P = w_max*psi_max*Ia_rated;

% Maximum EMF,torque and power delivered from the electrical machine
T_max_elm = psi_max*Ia_max;
E_max = psi_max*w_max;
P_rated = psi_max*Ia_rated*w_max;

disp(' ')
disp(' ')
disp(['Gearing = ',num2str(G_elm),' Gearing between the electric machine
and the driveshaft'])
disp(['V_cap = ',num2str(V_cap),' [V]'])
disp(['Ia_rated = ',num2str(Ia_rated),' [A]'])
disp(['Ia_max = ',num2str(Ia_max),' [A]'])
disp(['w_max = ',num2str(w_max),' [rad/s]'])
disp(['Vc_min_fw = ',num2str(Vc_min_fw),' [V], minimum voltage of the cap
that is on the border for field weakening'])
disp(['psi_max = ',num2str(psi_max),' [Wb]'])
disp(['Ra = ',num2str(Ra),' [ohm]'])
disp(['T_max_elm = ',num2str(T_max_elm),' [Nm], Rated torque of the
electrical machine'])
disp(['T_max = ',num2str(T_max_elm*G_elm),' [Nm], Rated torque from the
electrical machine on the driveshaft (Gearing between the machine and
driveshaft)'])
disp(['E_max = ',num2str(E_max),' [V], rated back-emf voltage'])
disp(['Vdc_min_fc = ',num2str(Vc_min_fw-Rc*Ia_max),' [V]'])
disp(['Va_max_fc = ',num2str(E_max+Ra*Ia_max),' [V], rated voltage of the
machine'])
disp(['P_rated = ',num2str(P_rated),' [W], Rated power of the machine'])
disp(['eff = ',num2str(eff),'=',num2str(P_rated/(P_rated+R*(Ia_rated)^2))])


%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Initial values for the simulation

% Defining the result vectors
```

```
Ea = zeros(1,length(T_ds));
Ia = zeros(1,length(T_ds));
Va = zeros(1,length(T_ds));
D = zeros(1,length(T_ds));
Ic = zeros(1,length(T_ds));
Vc = zeros(1,length(T_ds));
Vdc = zeros(1,length(T_ds));
psi = ones(1,length(T_ds)).*psi_max;
T_elm = zeros(1,length(T_ds));
P_brake = zeros(1,length(T_ds));
P_ICE = zeros(1,length(T_ds));
P_cap = zeros(1,length(T_ds));
Th_cap = zeros(1,length(T_ds));

Th_cap(1) = T_amb;
% Initial voltage of supercap
Vc(1) = V_cap;

% Maximum and minimum values for the control system
Ia_max_c = Ia_max;
Ia_min_c = Ia_min;

Vc_min_c = Discharge*V_cap;

%Time vector
t = linspace(0,length(T_ds)/100,length(T_ds));

E_ICE = 0;
E_brake = 0;
E_ICE_no = 0;
E_brake_no = 0;

for i = 1:length(T_ds)
    % Calculate the current in the motor, assuming no field weakening
    Te = (T_ds(i))/(G_elm);

    Ia(i) = Te/(psi(i));
    % Check the current in the motor and limit it if it is to large
    if Ia(i) < Ia_min_c
        Ia(i) = Ia_min_c;
    elseif Ia(i) > Ia_max_c
        Ia(i) = Ia_max_c;
    end

    if Ia(i) == 0 % Calculate the requiered duty cycle to get Ia = 0
        D(i) = (w_elm(i)*psi(i))/Vc(i);
    elseif ((Vc(i)/(2*Rc*Ia(i)))^2-Ra/Rc-w_elm(i)*psi(i)/(Rc*Ia(i)))<0
        if (Ra*Ia(i)+w_elm(i)*psi(i))==0
            D(i)=0;
        elseif (Ra*Ia(i)+w_elm(i)*psi(i))<0
            D(i) = -1;
        else
            D(i) = 2;
        end
    else
        D(i) = Vc(i)/(2*Rc*Ia(i))-sign(Ia(i))*sqrt((Vc(i)/(2*Rc*Ia(i)))^2-
Ra/Rc-w_elm(i)*psi(i)/(Rc*Ia(i)));
    end

    % Check if D > 1 -> Field weakening
```

```
    if D(i)>1
        D(i) = 1;
        % Calculate the flux that gives D=1
        if ((Vc(i)/(2*w_elm(i)))^2-(Rc+Ra)*Te/w_elm(i))>=0
            psi(i)=Vc(i)/(2*w_elm(i))+sqrt((Vc(i)/(2*w_elm(i)))^2-
(Rc+Ra)*Te/w_elm(i));
            % Calculate the current for this flux
            Ia(i)=(Vc(i)-psi(i)*w_elm(i))/(Rc+Ra);
            % Checking the current limits, if limit then recalculate psi
            if Ia(i) < Ia_min_c
                Ia(i) = Ia_min_c;
                psi(i) = (Vc(i)-(Rc+Ra)*Ia(i))/w_elm(i);
            elseif Ia(i) > Ia_max_c
                Ia(i) = Ia_max_c;
                psi(i) = (Vc(i)-(Rc+Ra)*Ia(i))/w_elm(i);
            end
        else
            if Ia(i) < 0
                Ia(i) = Ia_min_c;
                psi(i) = (Vc(i)-(Rc+Ra)*Ia(i))/w_elm(i);
            elseif Ia(i) >= 0
                Ia(i) = Ia_max_c;
                psi(i) = (Vc(i)-(Rc+Ra)*Ia(i))/w_elm(i);
            end
        end

    % Check if D<0
    elseif D(i)<0
        D(i) = 0;
        Ia(i) = -psi(i)*w_elm(i)/Ra;
    end


    Ea(i) = psi(i)*w_elm(i);
    Va(i) = Ra*Ia(i)+psi(i)*w_elm(i);
    Ic(i) = Ia(i)*D(i);
    Vdc(i) = Vc(i)-Rc*Ic(i);
    T_elm(i) = psi(i)*Ia(i);

    % Calculating power and energy from brakes and ICE
    T_rest = T_ds(i)-T_elm(i)*G_elm;
    if T_rest < 0 % Brake torque, power and energy
        P_brake(i) = T_rest*w_ds(i);
        E_brake = P_brake(i)*Ts+E_brake;
    else % Torque,power and energy for the ICE
        P_ICE(i) = T_rest*w_ds(i);
        E_ICE = P_ICE(i)*Ts+E_ICE;
    end

    % Calculating energy from brakes and ICE without hybrid system
    if T_ds(i) < 0 % Brake torque, power and energy
        E_brake_no = T_ds(i)*w_ds(i)*Ts+E_brake_no;
    else % Torque,power and energy for the ICE
        E_ICE_no = T_ds(i)*w_ds(i)*Ts+E_ICE_no;
    end

    if (i+1)<=length(T_ds)
        % Calculates the next capacitor voltage
        Vc(i+1) = -Ts*Ic(i)/C+Vc(i);
```

```
        if Vc(i+1) <= Vc_min_c %If the system tries to discharge the
capacitor when it has reached lowest allowed voltage
            Ia_max_c = 0;
            Ia_min_c = Ia_min;
        elseif Vc(i+1) >= V_cap %If the system tries to charge the
capacitor when it is fully charged
            Ia_max_c = Ia_max;
            Ia_min_c = 0;
        else
            Ia_max_c = Ia_max;
            Ia_min_c = Ia_min;
        end
    end


    P_cap(i) = Ic(i)^2*R_cap;
    if (i+1) <= length(T_ds)
        Th_cap(i+1) = Th_cap(i)+(Ts/C_th)*(P_cap(i)-(Th_cap(i)-
T_amb)/R_th);
    end
end

disp(['RMS Power electric machine =
',num2str(sqrt((sum((psi.*Ia.*w_elm).^2))/(length(w_elm)))),' [W]'])
disp(['RMS Current electric machine =
',num2str(sqrt((sum(Ia.^2))/(length(Ia)))),' [A]'])
disp(['RMS Current super capacitor =
',num2str(sqrt((sum(Ic.^2))/(length(Ic)))),' [A]'])
disp(' ')
disp(' ')
disp(['E_ICE = ',num2str(E_ICE/1e3),' [kJ]'])
disp(['E_ICE_no = ',num2str(E_ICE_no/1e3),' [kJ]'])
disp(' ')
disp(['E_brake = ',num2str(E_brake/1e3),' [kJ]'])
disp(['E_brake_no = ',num2str(E_brake_no/1e3),' [kJ]'])
disp(' ')
disp(['With hybrid / Without hybrid= ',num2str(E_ICE/E_ICE_no)])

%%
close all


figure
subplot(2,2,1)
plot(t,vprofile,'b',t,vprofile*3600/1000,'r')
xlabel('Time [s]')
ylabel('Speed [m/s]')
legend('Speed [m/s]','Speed [km/h]')
grid on
subplot(2,2,2)
plot(t,F_tot,'b',t,F_acc,'r',t,F_r,'g',t,F_air,'m')
xlabel('Time [s]')
ylabel('Force [N]')
legend('F_t_o_t','F_a_c_c','F_r','F_a_i_r')
grid on
subplot(2,2,3)
plot(t,w_ds,'b')
xlabel('Time [s]')
ylabel('Shaft speed [rad/s]')
grid on
```

```
subplot(2,2,4)
plot(t,T_ds,'b')
xlabel('Time [s]')
ylabel('Shaft torque [Nm]')
grid on


figure
subplot(2,2,1)
plot(t,Ea,t,Va,t,Vdc,t,Vc)
legend('Ea','Va','Vdc','Vc')
title('Voltages')
xlabel('Time [s]')
ylabel('Voltage [V]')
grid on
subplot(2,2,2)
plot(t,Ia,t,Ic)
legend('I_a','I_c')
title('Currents')
xlabel('Time [s]')
ylabel('Current [A]')
grid on
subplot(2,2,3)
plot(t,D)
title('Duty cycle')
xlabel('Time [s]')
ylabel('Duty cycle')
legend('D (Duty cycle)')
grid on
subplot(2,2,4)
plot(t,psi)
title('Flux linkage')
xlabel('Time [s]')
ylabel('Flux linkage [Wb]')
grid on

figure
subplot(2,1,1)
plot(t,T_ds,t,T_elm.*G_elm)
title('Torque')
xlabel('Time [s]')
ylabel('Torque [Nm]')
legend('T_d_s','T_e_l_m (With gearing)')
grid on
subplot(2,1,2)
plot(t,P_ICE,t,P_brake,t,T_elm.*w_elm,t,Vc.*Ic)
title('Power')
xlabel('Time [s]')
ylabel('Power [W]')
legend('P_I_C_E','P_B_r_a_k_e','P_e_l_m','P_c')
grid on

figure
plot(t,Rc.*(abs(Ic)).^2,t,Ra.*(abs(Ia)).^2,t,Ra.*(abs(Ia)).^2+Rc.*(abs(Ic))
.^2)
legend('P_R_c','P_R_a','P_t_o_t_a_l_ _l_o_s_s')
title('Losses in the system')
xlabel('Time [s]')
ylabel('Power [W]')
```

```
figure
plot(t,Th_cap)
title('Temperature of super capacitors')
xlabel('Time [s]')
ylabel('Temperature (C)')
```

Traction force from powertrain

```
% This script calculates the traction force from the powertrain based on
% "traction force from the powertrain"-graph from Anders Grauers (SHC)
% written by Martin Holder - may 2012
%% Initialize
close all
clear all
clc

% transmission
% dividing all ratios with 2.343 (2nd drive) to end up with the gear ratio
% the engine has at its output shaft.
i_1 = 17.163/2.343;
i_2 = 11.894/2.343;
i_3 = 8.859/2.343;
i_4 = 7.159/2.343;
i_5 = 6.016/2.343;
i_6 = 5.048/2.343;
i_diff = 5;

%converting rpm to rad
rad = 2*pi/60;

%vehilce definitions
m=290; %130+70+40+15
a_max = 6.9; %from CFS11
v_max = 95;  %from CFS11
enginespeed = (1500:500:8500); %RPM
%enginespeed_rad=enginespeed.*rad;
r = 0.2667; %radius of wheel

%Brake engine power importet from GT-Power
P=[5.46669
8.74445
10.6672
12.9205
15.5181
16.8864
22.6066
24.2648
27.5218
31.243
31.8238
31.2503
31.2493
30.1554
28.1038
];

P=P';

%multiplying factor for "increasing" engine performance:
%1.3 for 40kW simulation

P=P*1.3;

P=P*0.85; %assuming 15% overall losses
P=P*10^3; %converting from kW to W
```

```
%interpolating from 15 to 23 values
xii=linspace(1500,8500,23);
P_e = spline(enginespeed,P,xii);
enginespeed_rad = xii.*rad;

%% x-axis calculating: vehicle speed
%driving in different gears and go through engine rpm = 1500 to 8500
w_1 = enginespeed_rad.*(1/i_1);
w_2 = enginespeed_rad.*(1/i_2);
w_3 = enginespeed_rad.*(1/i_3);
w_4 = enginespeed_rad.*(1/i_4);
w_5 = enginespeed_rad.*(1/i_5);
w_6 = enginespeed_rad.*(1/i_6);

%gearbox output := a omega one for each gear
w = [w_1;w_2;w_3;w_4;w_5;w_6];
W_1 = w_1.*(1/i_diff);
W_2 = w_2.*(1/i_diff);
W_3 = w_3.*(1/i_diff);
W_4 = w_4.*(1/i_diff);
W_5 = w_5.*(1/i_diff);
W_6 = w_6.*(1/i_diff);

%transform from [w]=rad to km/h via respecting wheel radius r
v_1 = W_1.*r*3.6;
v_2 = W_2.*r*3.6;
v_3 = W_3.*r*3.6;
v_4 = W_4.*r*3.6;
v_5 = W_5.*r*3.6;
v_6 = W_6.*r*3.6;

%% y-axis calculation: driving force

%from definition of Power P = Tw = Fv follows: F = Tw/v = P/v
%calculatuing from engine power which delivers the same values
f_1 = P_e./v_1;
f_2 = P_e./v_2;
f_3 = P_e./v_3;
f_4 = P_e./v_4;
f_5 = P_e./v_5;
f_6 = P_e./v_6;

%normalizing the force by dividing with the vehicle mass.
nf1 = f_1./m;
nf2 = f_2./m;
nf3 = f_3./m;
nf4 = f_4./m;
nf5 = f_5./m;
nf6 = f_6./m;


%% EL MOTOR
w_el=[
1
50
100
150
200
250
300
```

```
350
400
450
500
550
600
650
700
750
800
850
900
950
1000
1050
1100
];

P_el=[
1
940
1880
2820
3760
4700
5640
6580
7520
8460
9400
10340
11280
12220
13160
14100
15040
15040
15040
15040
15040
15040
15040
];

%relevant gear ratios for el motor
i_diff = 1;
i_el = 10;

%gearing with an assumed gear ratio 10:1 between el motor output and chain
w_el_1 = w_el.*(1/i_el);

%assuming el motor work directly on the diff
W_el = w_el_1.*(1/i_diff);

%velocity "flows" through the wheels on the road
v_el = W_el.*r*3.6;

%traction force calculation
F_el = P_el./v_el;
nF_el = F_el./m;
```

```
%plotting el-machine power characteristics:
figure (1)
subplot(1,2,1)
plot (w_el,P_el); %engine speed vs engine power

subplot(1,2,2)
plot(v_el,nF_el); %traction force contribution to the powertrain


%% combining both

hf1 = nf1+nF_el';
hf2 = nf2+nF_el';
hf3 = nf3+nF_el';
hf4 = nf4+nF_el';
hf5 = nf5+nF_el';
hf6 = nf6+nF_el';
v_1_h=v_1+v_el';

%% ploting

figure (2)
subplot(1,2,1)

plot(v_1,nf1,'b-','LineWidth',2)
hold on
plot(v_2,nf2,'g-.','LineWidth',2)
hold on
plot(v_3,nf3,'k:','LineWidth',2)
hold on
plot(v_4,nf4,'m--','LineWidth',2)
hold on
plot(v_5,nf5,'c-','LineWidth',2)
hold on
plot(v_6,nf6,'r-.','LineWidth',2)
grid on
xlabel('Vehicle Speed [km/h] - ICE Only')
ylabel('Normalized Force (acceleration) [N/kg]')
legend ('1st Gear','2nd Gear','3rd Gear','4th Gear','5th Gear','6th Gear')
hold on
plot(1:0.01:50,a_max,[v_max,v_max],[0,a_max],'k','LineWidth',2) %plotting
constants
hold off

subplot(1,2,2)

plot(v_1,hf1,'b-','LineWidth',2)
hold on
plot(v_2,hf2,'g-.','LineWidth',2)
hold on
plot(v_3,hf3,'k:','LineWidth',2)
hold on
plot(v_4,hf4,'m--','LineWidth',2)
hold on
plot(v_5,hf5,'c-','LineWidth',2)
hold on
plot(v_6,hf6,'r-.','LineWidth',2)
```

```
xlabel('Vehicle Speed [km/h] - Hybrid Mode')
ylabel('Normalized Force (acceleration) [N/kg]')
legend ('1st Gear','2nd Gear','3rd Gear','4th Gear','5th Gear','6th Gear')
grid on
hold on
plot(1:0.01:50,a_max,[v_max,v_max],[0,a_max],'k','LineWidth',2)
hold off
```