CHALMERS
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# Power consumption modeling of an Arctic Trucks vehicle

Master's thesis in Automotive Engineering

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Department of Mechanics and Maritime Sciences
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2019

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# Abstract

In this thesis a model is created to simulate the power demand and fuel consumption of an Arctic Trucks vehicle. The model is created in Matlab Simulink using the QSS toolbox developed by ETH Zurich. A specific off-road driving cycle is created and investigated to verify the model as well as to investigate the power demand and fuel consumption. Arctic Trucks manufactures modified SUV and pickup trucks for off-road driving. Their vehicles are used for in extreme conditions where large tyres and associated modifications are needed. Arctic Trucks wanted to investigate the potential benefits of using hybrid vehicles and the model created in this thesis can be used to estimate the required hybrid powertrain size needed for such vehicles, that is, the battery and motor size. The model created was capable of estimating the power demand and fuel consumption with at least 95% accuracy for a typical off-road driving cycle in snowy conditions.

Key words: Hybrid, power demand, vehicle modeling, fuel consumption, hybrid modeling, Simulink, QSS Toolbox.

# Acknowledgment

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Aron Steinn Guðmundsson

# Contents

# Notations

**ICE** – Internal combustion engine
**DOF –** Degrees of freedom
**EM** – Electric motor
**QSS** – Quasi-Static Simulation
**ECU –** Engine Control Unit
**ABS –** Anti-Lock Brake System

# 1 Introduction

The aim of this thesis is to investigate and model the power demand and fuel consumptions of an Arctic Trucks vehicle. Arctic Trucks is a company that specializes in manufacturing and modifying cars and trucks that are used to drive in extreme conditions. Most changes are usually made to the vehicles' chassis and drivetrain to increase the ground clearance. This is done by mounting bigger tires, which requires major body work as well as changing some of the main parts in the powertrains e.g. final drive gear ratios.

## 1.1 Problem Description

As the human population increases and the demand for human transportation increases, one of the biggest goals of every car manufacture is to decrease pollution and design vehicles that are more efficient. This evolution has pushed the electric technology in the automotive industry extensively for the past years and almost all car manufacturers today offer a hybrid or full electric vehicle.
Arctic Trucks wants to investigate the possibility of using a hybrid vehicle for their projects as well as investigating the total power demand in a typical off-road driving scenario. At the same time, the company wants to investigate the fuel saving benefits of using a hybrid vehicle.

As these extreme driving conditions usually do not include driving on a regular asphalt, there are many unknown factors. Most of the driving is done in snowy conditions which introduces the additional problem of estimating the tyre grip and the rolling resistance of the tyres.

## 1.2 Objective

The objective of this thesis is to create a simulation model that can determine the actual power demand and fuel consumption for a typical Arctic Trucks vehicle, the model should be validated by comparing the model results with actual data. The valid model will then be used to estimate the total power demand of a conceptual Arctic Trucks hybrid vehicle.

## 1.3 Solution

By creating a quasi-static model in Matlab Simulink, using a QSS Toolbox developed by ETH Zurich based on a 2016 Toyota Hilux modified by Arctic Trucks, one can estimate the fuel consumption and the power demand for a given driving cycle. By recoding a driving cycle and monitoring the fuel consumption in the actual vehicle and then comparing the result with the model created with the QSS Toolbox, the model can be verified. These results can then be used to estimate the total power demand of a similar hybrid vehicle. Figure 1.1 shows a step-by-step plan for the research conducted.

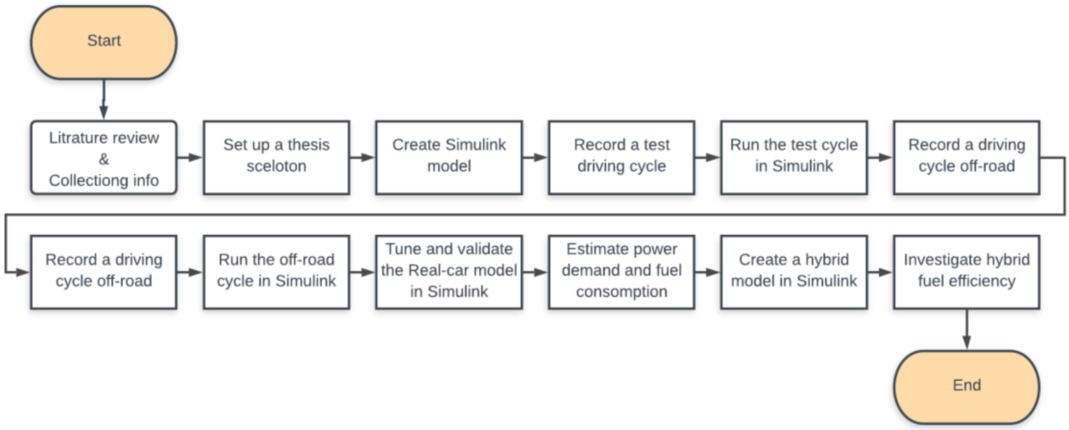


*Figure 1.1: A step-by-step plan of the research conducted*

## 1.4 Deliverables

The following deliverables are included in the thesis:

- A QSS Toolbox Simulink model of a Toyota Hilux AT38 validated with real data form a recorded driving cycle.
- Minimum power requirement estimation for an Arctic Trucks hybrid vehicle.
- A MSc thesis written in accordance to Chalmers requirements.

## 1.5 Limitations

The following is a list of the limitations that have been identified for this thesis:

- The QSS model uses the driving cycle road gradient and tyre rolling resistance to calculate the force required to propel the car. This can be hard to measure in real life when the vehicle is driving off-road. This is important as the driving scenarios for Arctic Trucks vehicle are usually not on a flat road.
- As the driving cycle will be off-road, the rolling resistant coefficient will be hard to determine, especially on long driving cycles where the road surface differs. Also, due to relevantly low speed driving cycles, the original aerodynamic drag coefficient for an original vehicle provided by Toyota will be used.
- The model verification will be performed on a vehicle equipped with a manual transmission. This will simplify the model validation as there will be no need for complex torque converter modelling.
- Accurate efficiency maps for the ICE and the EM are hard to find.

# 2 Theory

## 2.1 Arctic Trucks

Arctic Trucks is a company that specializes in modifying 4x4 off-road vehicles for driving in extreme conditions. The company is based in Iceland and has offices in Norway, UK and Dubai.

Arctic Trucks is a cutting-edge company when it comes to drastic modifications like getting trucks ready for the Antarctica and other extreme conditions. They have a subsidiary, Arctic Trucks Experience, that offers tours on the highlands of Iceland, as well as tours to the Antarctica on Arctic Trucks modified vehicles. They also engage in missions that are carried out in the Antarctica, such as refilling the airport's fuel supplies.

## 2.2 QSS Toolbox

Simulink is a graphical programing environment within Matlab, for simulating dynamic models. QSS Toolbox is a package within Simulink for quasi-stationary vehicle modeling. [1]

The QSS toolbox is an efficient tool that allows the user to design and simulate a powertrain and for example estimate fuel consumption as well as power demand quickly. This allows the user to evaluate various fuel saving strategies and create new powertrain concepts in a short time.

The basic principle of the QSS toolbox is that the dynamics of the system are eliminated and equilibrium is obtained at all states and time dependent equations can be neglected. This allows for a simplified calculation of the dynamic powertrain system.

## 2.3 Power demand

Power demand is the measure of the power required to propel a vehicle. The power demand can be calculated at the wheel. This excludes the powertrain from the equation and therefor the calculated power demand is the same for any powertrain (as long as the vehicle mass is constant). [2]

There are four things that affect power demand; air resistance, rolling resistance, road gradient and vehicle acceleration. All of those factors can be referred to as forces working either along or against the motion of the vehicle. Also, as power is defined as force applied over a certain speed, the vehicle velocity also affects the power demand.

Air resistance, often called drag, is a type of friction, in this case it is the friction created by the air that the vehicle is traveling through. This friction is highly dependent on the vehicle speed as the correlation is squared to the velocity. The friction is also dependent on the *drag coefficient* which is a constant that quantifies the drag of a component in a fluid environment, in this case air.
Rolling resistance is defined as the resistance of the tyres rolling on the surface. In a real vehicle there are more factors creating resistance such as wheel bearing and seals in the drivetrain.

Road gradient is defined as the rate altitude change with respect to distance traveled horizontally. This is usually measured in percentage.

Vehicle acceleration is defined as the rate of change in velocity with respect to time.

In reality, there are more factors that can affect power demand, like for example wind, nut those factors are neglected in this thesis.

## 2.4 Fuel consumption

Fuel consumption is the measurement of how much fuel a vehicle consumes to propel the vehicle. Fuel consumption is most often measured in the form of litres per 100km (l/100km), or miles per gallon (MPG).

A combustion engine requires fuel and oxygen to produce the power. The efficiency of the vehicle's engine then determines the amount of fuel needed to fulfil its power demand.

## 2.5 Vehicle speed estimation

Vehicle speed can be measured in many ways. Most manufactures use magnetic field sensors or hall effect sensors. Those sensors are usually a part of the vehicle's ABS system. The ABS sensors are placed on each wheel and can monitor the angular speed of each wheel independently. The car system can then use the measured angular speed to estimate the vehicle speed using equation . [3]

$$v = \omega R \tag{2.1}$$

Where v is the vehicle speed, ω is the angular speed and R is the radius of the tyre.

As the sensors only monitor the angular speed of the wheels the downside is that if the wheels are slipping or are locked, the vehicle speed estimation will not be correct. Another source of vehicle speed estimation is that the vehicle system assumes the tyre radius to be a predefined constant value. This predefined value is set to the measured tyre radius when the tyre air pressure is set to optimal for regular road driving. On vehicles with large tyres and high side wall the wheel radius is very sensitive to the tyre air pressure due to tyre deformation.

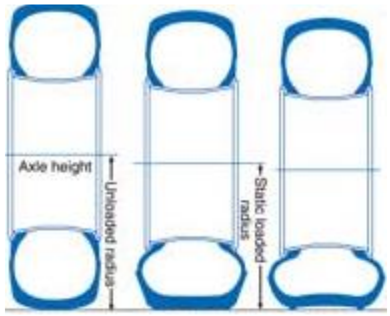


*Figure 2.1: Picture showing how axle load affects the tyre deformation.*

## 2.6 Tyre slip

Tyre slip is defined as the difference between the vehicle speed and the tyre surface speed. Tyre slip usually increases with driving force and is greatly affected by the tyre compound as well. In extreme cases when the wheel slip exceeds the tyre's slip limit the vehicle starts to lose grip which limits the amount of driving force.

## 2.7 Tyre deformation

Tyres that are used on road legal cars are made from complex rubber compound. Rubber is an elastic material that can be deformed. In order to minimize the tyre deformation, tyres are inflated with air. Depending on the load applied to the tyre the air pressure is adjusted to maintain as large contact patch between the road and the tyre as possible. Figure 2.3 shows how tyre air pressure affects the form of the shape of the tyre.

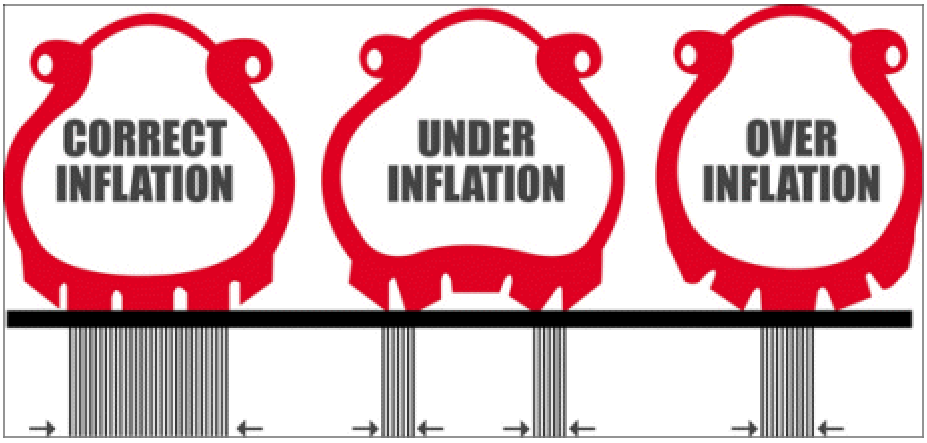


*Figure 2.2: Picture showing how tyre pressure affects the contact patch*

*Figure 2.3: Picture showing how deflating the tyre can minimize how much the tyre sinks into the ground. [4]*

When driving on gravel or in snow the surface that the vehicle is driving on is not as flat as asphalt. This means that the contact patch can be increased by deflating the tyre and allowing it to deform. This gives more traction and therefor higher driving force can be applied by the tyre before the tyre exceeds the tyre's slip limit. Another benefit of deflating the tyre is that as the contact patch size increases the tyre load is distributed over a larger area. This means that the tyre will not sink as deep in to the ground when driving on a loose surface (snow or sand). This can be seen in Figure 2.3.

# 3 Scope

## 3.1 Vehicle

The vehicle used in this thesis is a Toyota Hilux AT38, 2015 model. This vehicle has been used for the self-drive experience offered by Arctic Trucks Experience since it was manufactured and modified by Arctic Trucks. Self-drive is a driving tour that Arctic Trucks Experience offers where people can rent an Arctic Trucks vehicle and drive them self in the highlands of Iceland.

As mentioned before, the Arctic Trucks vehicles are modified for extreme conditions. The modifications that have been done to the vehicle used in this thesis are: The vehicle suspension has been raised by 90mm and the original 265/65R17 (30,5/10,5R17) tyres have been replaced with 38/12,5R15 tyres. This results in a total body lift of 230 mm and total a lift of 54 mm.[1] Also, the vehicle has been equipped with a differential lock in both front and rear differential to maximize the grip. The locks are pneumatic and can be activated from inside the car with electrical switches. The test vehicle can be seen in Figure 3.1. The full car specification can be found in appendix A. [5]

*Figure 3.2: Picture showing the test vehicle.*

To take advantage of the large tyres, the tyre pressure is often changed to increase traction and to give more comfort. The most common tyre pressure used for the test vehicle are presented in table 3.1.

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*Table 3.1: Optimal tyre pressure values for different driving conditions*

| Tyre pressure | Static load tyre radius | Conditions |
|---|---|---|
| 25 psi | 0.48 m | Asphalt road |
| 10 psi | 0.46 m | Gravel road |
| 4 psi | 0.4 m | Snow |

The tyre pressure used in all of the driving cycles in this thesis was set to 4 psi.

## 3.2 Driving cycle

The driving cycle is created using data recorded by driving the AT Hilux in various conditions in Iceland. The data is recorded using a data logger equipped with accelerometers and GPS. The logger then uses the sensor data to calculate accurate vehicle speed and to estimate the road inclination.
A driving cycle was created by driving a well known and commonly driven off-road track in the south of Iceland. For ease of data processing, the driving cycle was split up to multiple scenarios, as all of the data that was processed in the vehicle's ECU was logged with a sample rate around 10 Hz. This created large files of raw data that was hard to process.

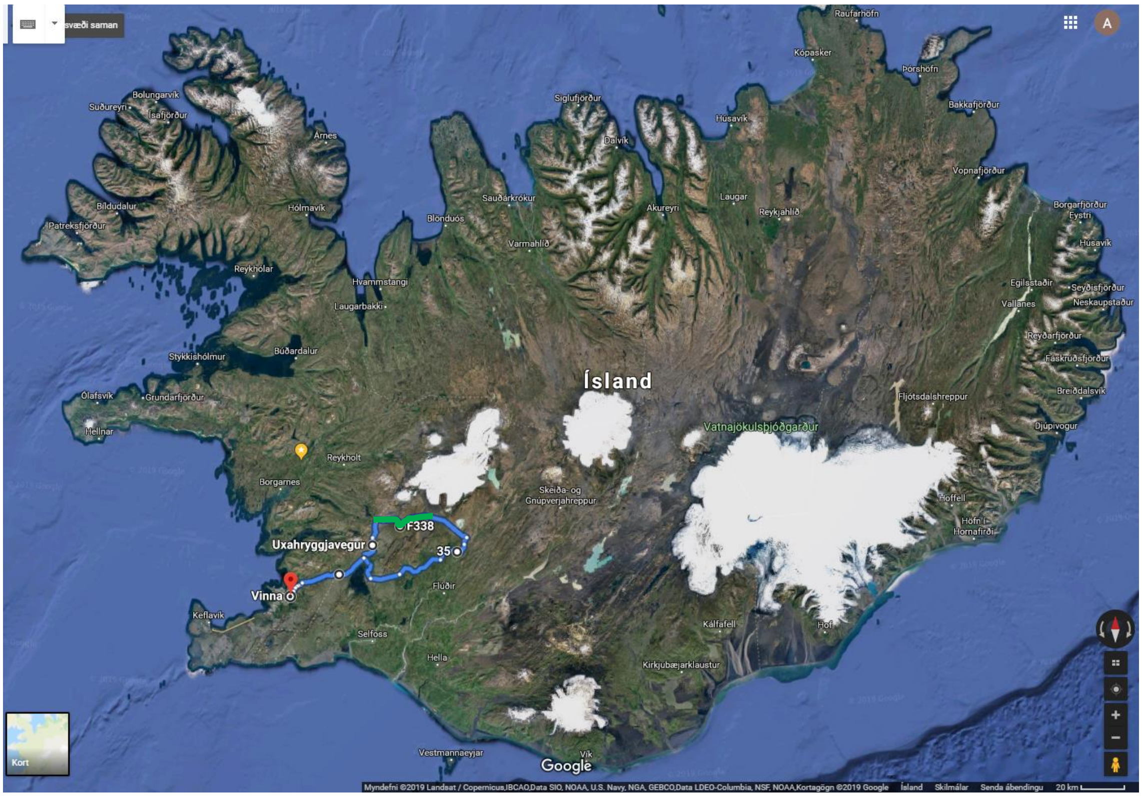


*Figure 3.2: The whole driving cycle can be seen in blue on the map, the cycle used in this thesis is marked in green.*

*Figure 4. 3: Road gradient messured with 6DOF gyroscope and GPS*

First part of the cycle was driving from Reykjavík towards Þingvellir on asphalt road. The second part was driven on a mountain road, partly on asphalt and partly on snow. The last part was then driving back to Reykjavík on asphalt.
For this thesis only the off-road part (marked green in Figure 3.2) was considered and split into 3 off-road parts; driving uphill, driving down hill and driving on a flat ground. All parts had similar road conditions, 100-300mm of soft uncompressed snow on top of a hard layer of frozen snow. In some spots the car managed to float on top of the hard layer and in other spots the vehicle sank deep into the hard layer. The parts are shown with different colors in Figure 3.3. The uphill cycle is in red, the down hill cycle is in blue and the flat cycle is in yellow. Figure 3.4 shows how the altitude changes during the cycles.

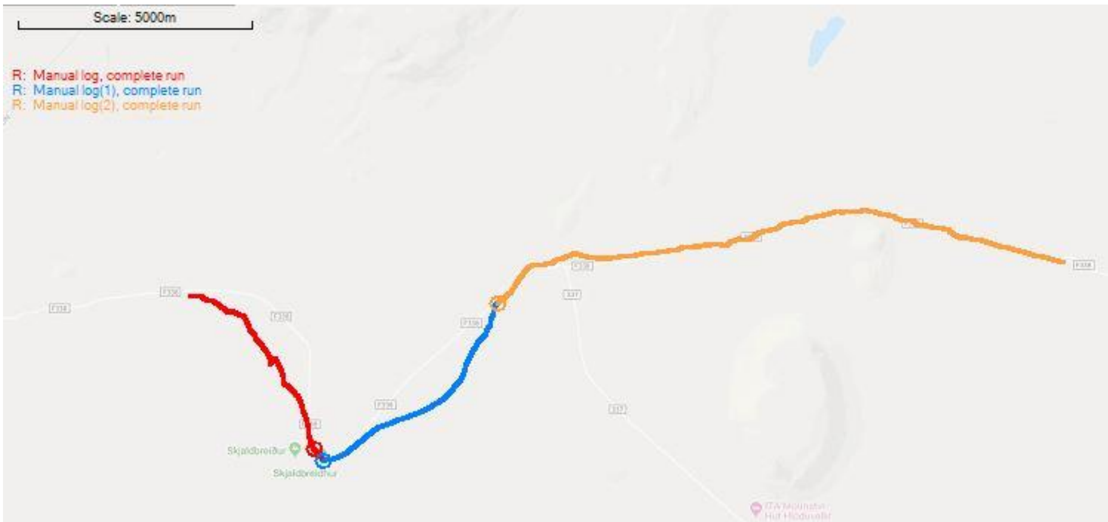


*Figure 3.3: Closer look on the three driving cycles create.*


*Figure 3.4: Figure showing the altitude of the three driving cycles.*

Figure 3.5 shows the test vehicle in the condition found in the driving cycles used in this thesis.


*Figure 3.5: Figure showing the test vehicle when driving the uphill cycle. The vehicle had to be stopped to adjust the tyre pressure due to altitude change.*

### 3.2.1 Driving cycle creation

The required data for a driving cycle suitable for the Simulink model is:

- Time
- Vehicle speed
- Selected gear
- Road gradient

#### 3.2.1.1 Vehicle speed

The vehicle speed can be logged form the vehicle's ECU or with a GPS logger. Both options were evaluated and are shown in Figure 3.6. As can be seen in this figure, the two methods differed quite extensively in their measures.
This is due to wheel slip when driving in slippery snow conditions. Even though the car is equipped with a fully locked transfer case and locked differentials the traction is not high enough to ensure that all the four wheels will not start slipping. Another source of error is the tyre diameter which is not accurately adjusted in the vehicle's ECU. Due to this, it was decided to use the GPS speed as a reference when creating the driving cycles for the Simulink model.

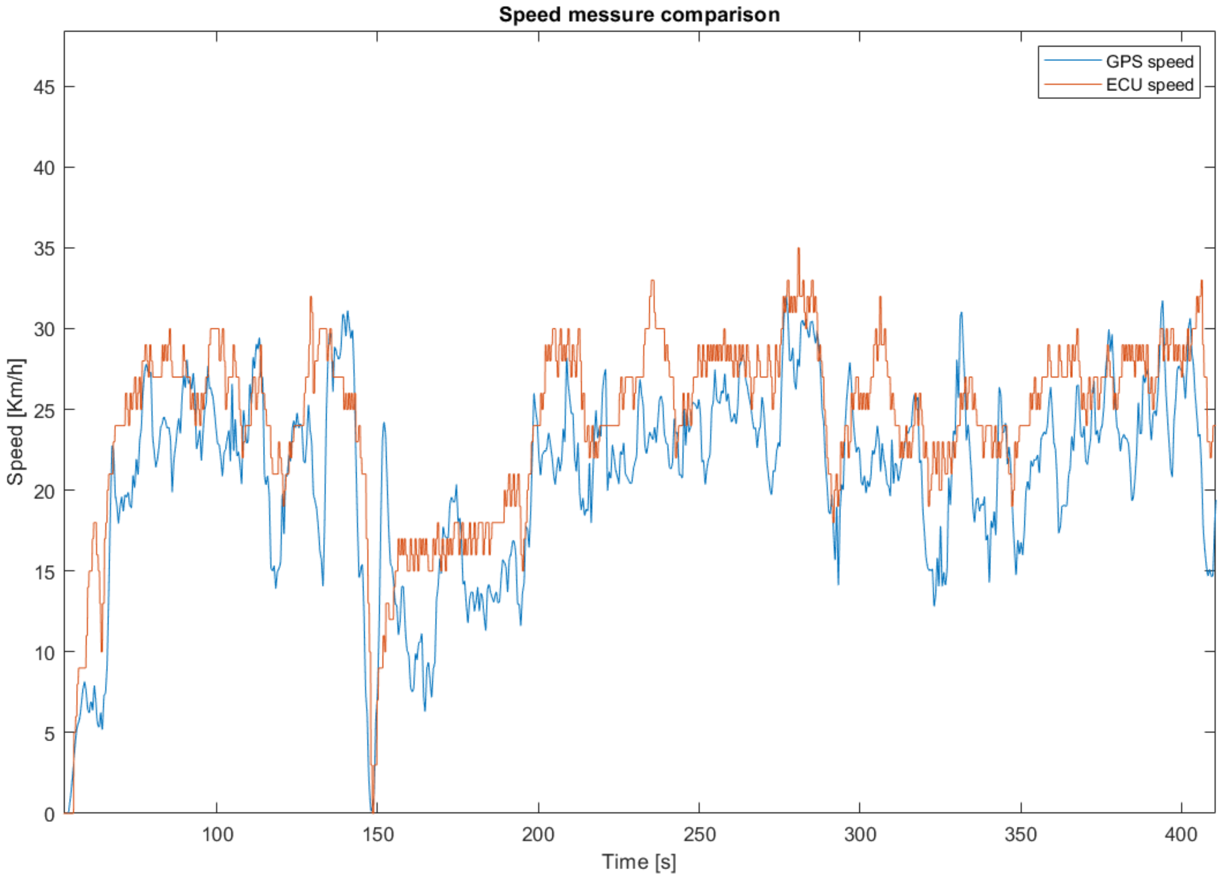


*Figure 3.6: Speed comparison between GPS logger and vehicle´s ECU.*

#### 3.2.1.2 Selected gear

In order to verify the Simulink model, the current gear selected had to be logged somehow. Depending on the vehicle, this can sometimes be logged through the vehicle's ECU, especially if the vehicle is equipped with an automatic transmission. Unfortunately, this option was not available in the test vehicle used in this thesis. The alternative was to log the vehicle speed and the engine speed and then by knowing the gear ratios in the gearbox, the selected gear could be calculated at each time. Another option would have been to add some sensors to the vehicle to monitor the selected gear, but that solution was complicated and time consuming.

#### 3.2.1.3 Road gradient

One of the problems that had to be solved during the thesis work was to figure out a way to measure the road gradient consistently while driving. The data logger used is equipped with a software that can estimate the road gradient based on GPS altitude change. To verify those measurements, a 6 DOF gyroscope equipped with 3 axis accelerometer and 3 axis gyroscope was used to record the road gradient as well. A test cycle was created on a road that had a total elevation change of 100 meters on a 3 km long path, with various road gradient. The results show minor variance and are presented in Figure 3.7. The gyroscope uses both the accelerometer and the gyroscope to measure the angle. The accelerometer is used to keep track of the 0-angle origin. This makes the gyroscope sensitive to vehicle acceleration, which explains the spikes in the graph. As there are more unknown variables in the system it was decided that this difference could be neglected for the sake of time constraints and therefor the GPS method was used.

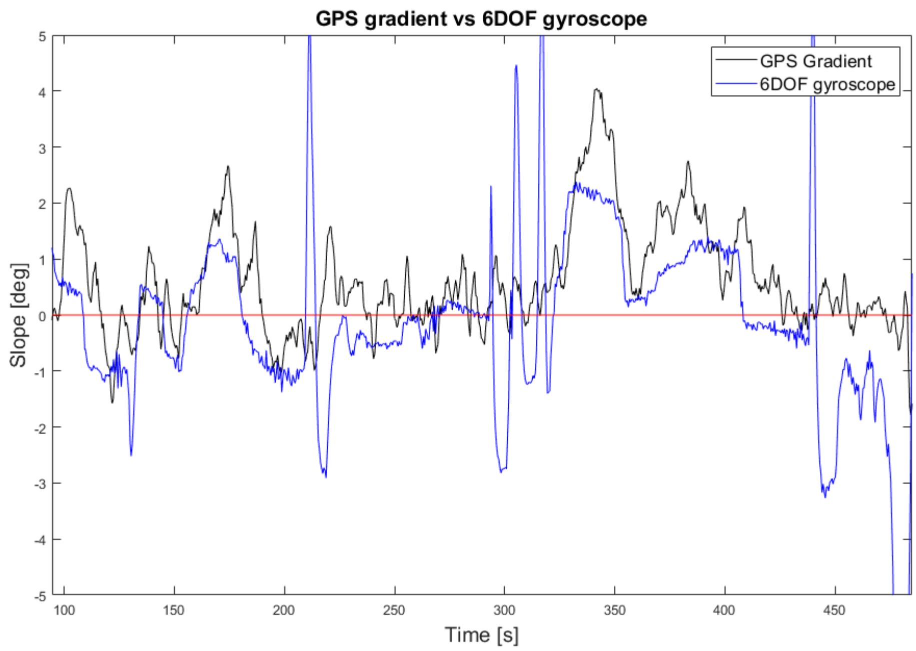


*Figure 3.7: Comparison of road gradient measurements with gyroscope and GPS.*

## 3.3 Real car model

The real car model was created in Simulink to create a base model that could be verified by comparing it with the data recorded from the actual test car.
Although Arctic Trucks has a good collaboration with Toyota Iceland it was hard for Toyota Iceland to provide us with all necessary data required for the Simulink model. Due to this, many variables had to be estimated and then tuned to get the real car model to match with the real data.
The unknown parameters that are estimated to have the largest influence on the results are listed below in table 3.2

*Table 3.2: Unknown variables in the real car model. [6]*

| Variable | Description |
|---|---|
| Engine BSFC | Estimated based on maximum efficiency |
| Rolling friction coefficient | Tuned so that the calculated torque matched the recorded torque |
| Vehicle drag coefficient | Kept the same as on an unmodified vehicle |
| Transmission efficiency | Estimated to be 90-95% |
| Tyre radius | Tuned so that the simulated engine speed matched the recorded engine speed |
| Road gradient gain | A multiplication factor for inaccurate measurements |
| Torque limit margin | Allowed margin of overshoot over estimated maximum torque |

The unknown parameters were tuned by comparing the simulation results to the recorded data. A Matlab script was created to iterate the simulation. This allowed for a simple optimization of the unknown variables to minimize the difference between the recoded results and the simulated results.
Two different approaches where used, minimizing the difference in power demand and minimizing the difference in fuel consumption.

# 4 Model description

The basic equations used in the model will be presented in this chapter.

The model is split into multiple blocks where each block represents different part of the system, starting with the driving cycle and ending with the fuel tank.
The model blocks are shown in Figure 4.1.

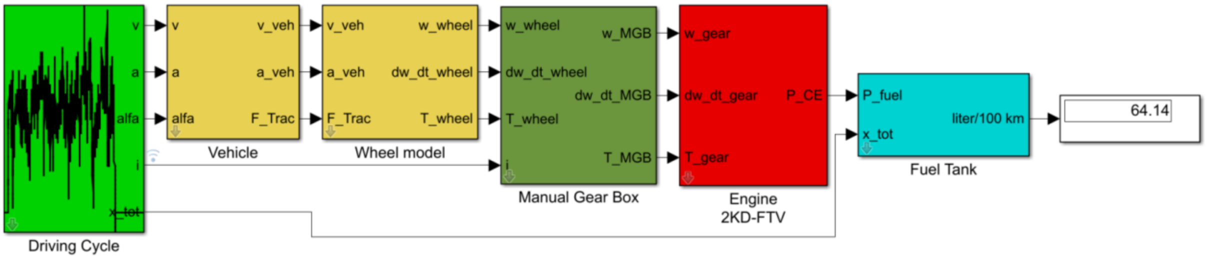


*Figure 4.1: The real car model in Simulink*

## 4.1 Driving cycle block

During this thesis multiple driving cycles were created from real driving situations. The data required to create a cycle in the QSS model are equally long vectors containing time **T_z**, speed **V_z**, acceleration **D_z,** gear selected **G_z** and road gradient **alfa**. When recording the driving cycle, **T_z**, **V_z**, **G_z** and **alfa** were measured and logged. However, it is difficult to measure acceleration accurately in a moving vehicle. Therefor the vehicle acceleration calculated from the speed difference between each step using equation (4.2).

$$\mathrm{D\_z}(k*h) = \frac{V\_z(k*h+h) - V\_z(k*h)}{h}, \quad k = 1, \ldots k_{1-max}, \quad k_{max} = 0 \tag{4.2}$$

Where k is the time step and h is the time step size. The step size used in this thesis is 1 second.

Figure 4.2 shows the driving cycle block. As already mentioned, the input to the driving cycle block are the vectors **T_z**, **V_z**, **D_z**, **G_z** and **alfa.** The outputs are indicated with red boxes and are either calculated or passed through. The covered distance (x_tot) is calculated by integrating the speed (V_z), and the acceleration (D_z) is calculated by differentiating the speed (V_z). The gradient is passed through a gain to convert from degrees to radians as required by Matlab.

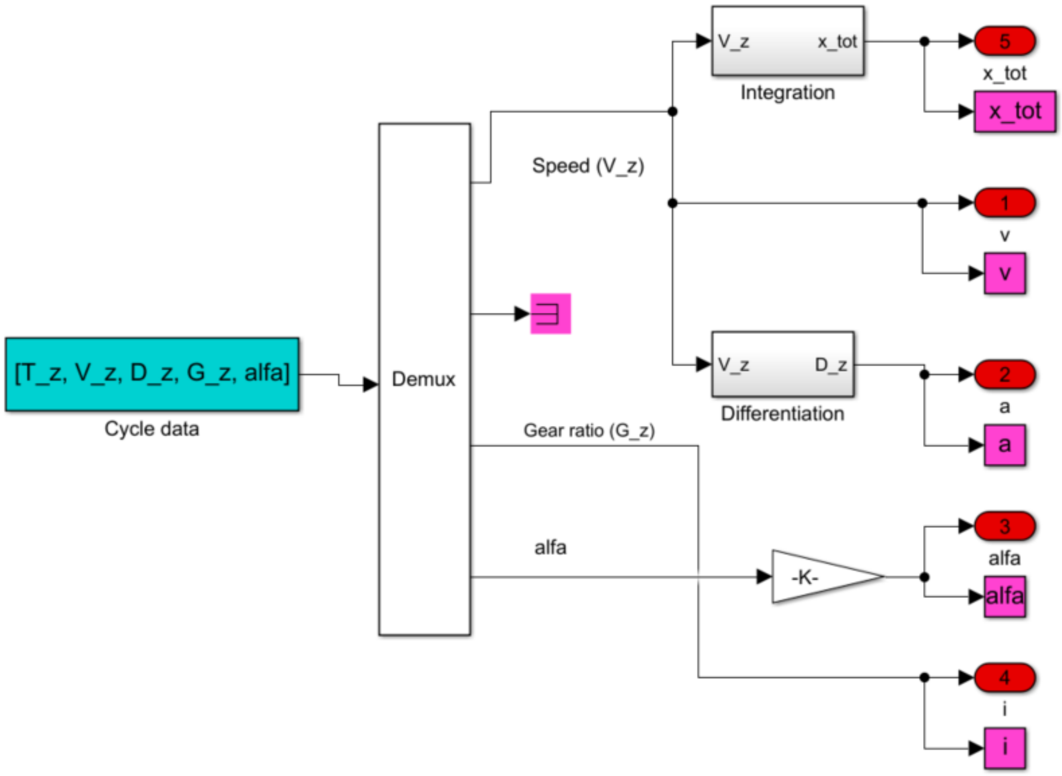


*Figure 4.2: The driving cycle block.*

## 4.2 Vehicle block

The vehicle block calculates the force required to propel the vehicle at each velocity and acceleration defined by the driving cycle (F_trac). The vehicle block also compensates for the road gradient. Equations (4.3), (4.4), (4.5) and (4.6) are used to calculate the total required driving force (F_trac).

$$F_{roll} = mg\mu \cos\alpha \tag{4.3}$$

$$F_{aero} = \frac{1}{2}\rho\, c_d\, A_f\, v^2 \tag{4.4}$$

$$F_{net} = ma\left(1 + \frac{k_{inertia}}{100}\right) \tag{4.5}$$

$$F_{grad} = mg \sin\alpha \tag{4.6}$$

$$F_{trac} = F_{roll} + F_{aero} + F_{net} + F_{grad} \tag{4.7}$$

To create a more smooth and realistic simulation the vehicle block also converts the vehicle speed at each timestep to average vehicle speed between two steps using equation (4.8).

$$v_a = \frac{v_i + v_{i-1}}{2} \tag{4.8}$$

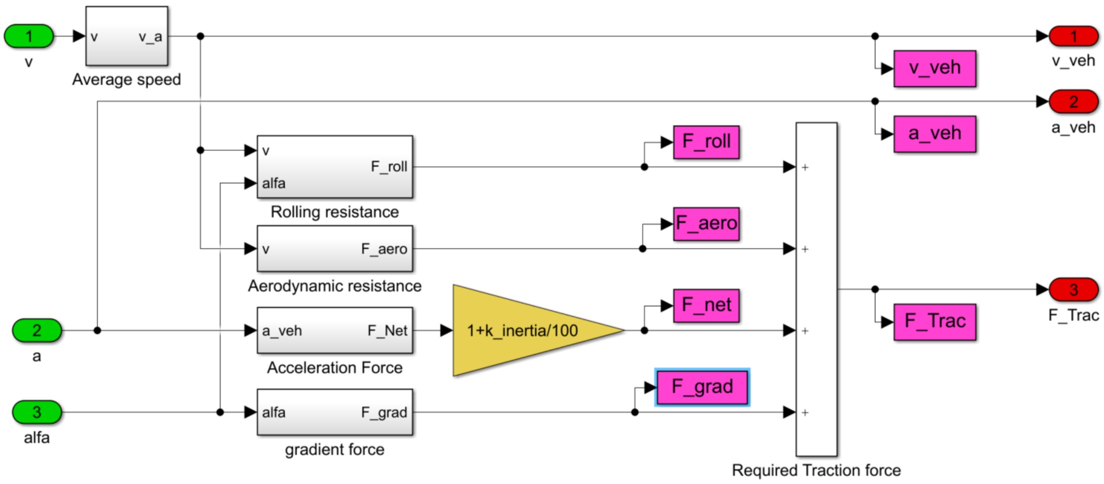


*Figure 4.3: The first vehicle model block.*

The vehicle block is presented in Figure 4.33. This figure shows how the block uses the inputs from the driving cycle block to calculate F_Trac which is passed through to the wheel block, along with the averaged speed (v_veh) and acceleration (a_veh).

## 4.3 Wheel block

The wheel block uses the results from the vehicle block to calculate the wheel angular speed and acceleration, as well as the required torque at the wheel, using equations (4.9), (4.10) and (4.11).

$$\omega_{wheel} = \frac{2v_{veh}}{d_{wheel}} \tag{4.9}$$

$$\frac{d\omega_{wheel}}{dt} = \frac{2a_{veh}}{d_{wheel}} \tag{4.10}$$

$$T_{wheel} = F_{trac} \cdot \frac{d_{wheel}}{2} \tag{4.11}$$

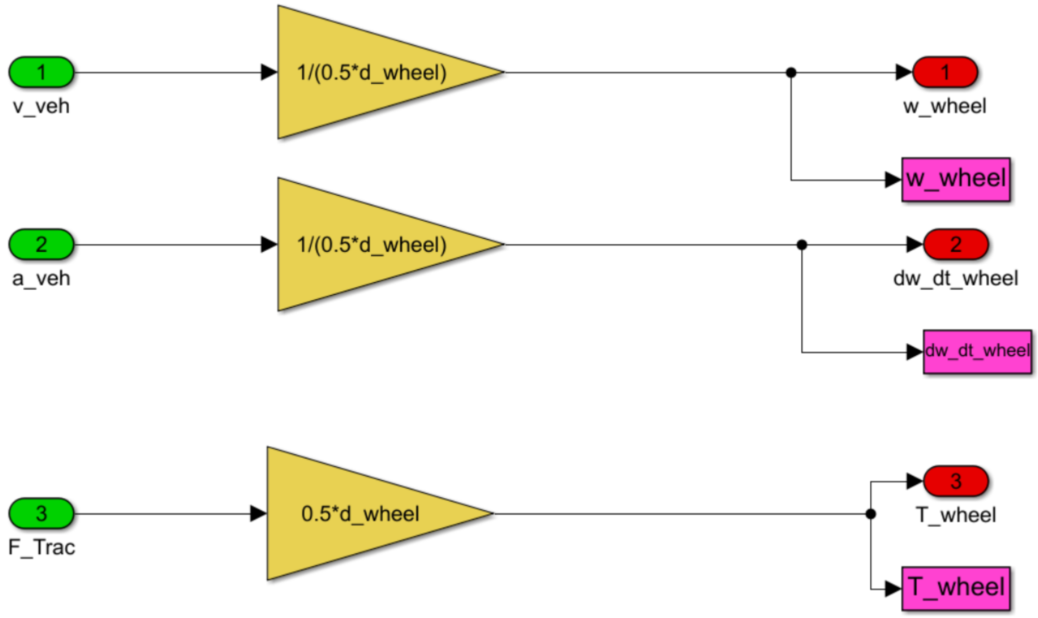


*Figure 4.4: The wheel block.*

The wheel block is presented in Figure 4.34. This figure shows how the block uses the inputs from the vehicle block to calculate the angular wheel speed ($\omega_{wheel}$) and angular wheel acceleration ($\frac{d}{dt}\omega_{wheel}$) of the wheel as well as the torque ($T_{wheel}$) at the wheel using equations (4.9), (4.10) and (4.11). Those values are then passed through to the gearbox block.

## 4.4 Gearbox block

The gearbox is used to transfer power and to alternate the ratio of speed and torque between the input and the output shaft of the transmission.
There are three main types of transmission used in automotive manufacturing nowadays:

- Manual transmission
- Automatic transmission
- Continually variable transmission

The QSS toolbox allows for any of these transmissions to be modelled. For this thesis a manual transmission was used. The manual transmission model included in the QSS Toolbox was modified to be able to simulate a 6-speed manual gearbox. Also, a torque overload detection system was implemented to ensure that the correct gear was selected.

The vehicle used to create the driving cycles was equipped with a manual gearbox to avoid complicated torque converter modelling. Therefor there was no option to log the selected gear through the car's ECU. The solution was to create a Matlab script that was able to calculate the current gear based on vehicle speed and engine speed. The script was able to handle all gear positions in most cases. Although, where the vehicle speed estimations were wrong the script could calculate the wrong gear position. The speed estimation becomes wrong in case of a tyre slip or tyre deformation.

The overload detection system was implemented to handle those cases. This system ensures that the current gear selected at each timestep would not request more torque form the engine than the engine could produce.

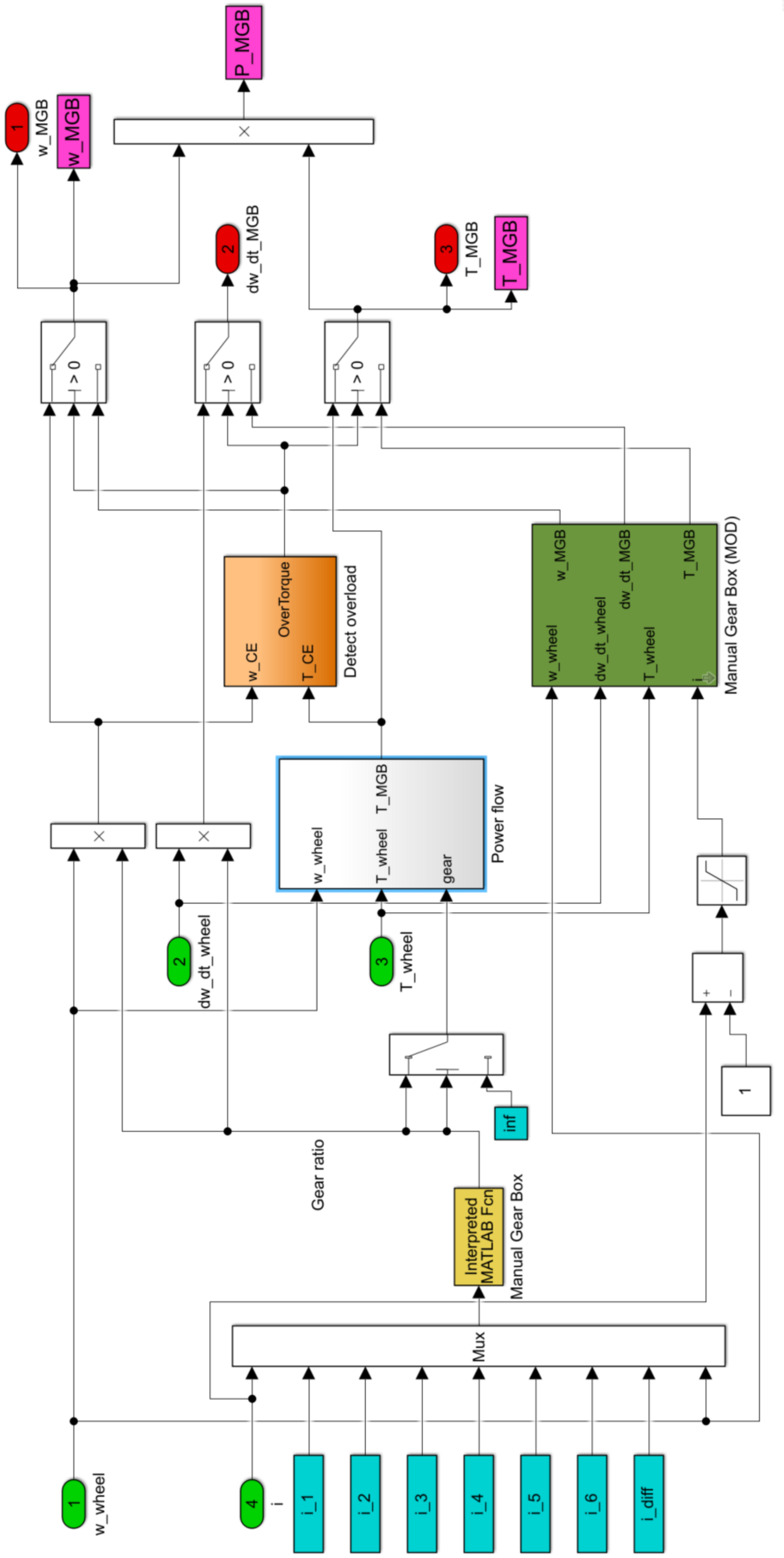


*Figure 4.5: The gearbox block.*

The gearbox block is presented in Figure 4.3. This figure shows how the block uses the inputs from the wheel block to calculate the angular transmission speed ($\omega_{wheel}$) and angular transmission acceleration ($\frac{d}{dt}\omega_{wheel}$) at the input of the transmission, as well as the torque ($T_{mgb}$) at the input of the transmission using equations (4.9), (4.10) and (4.11). Those values are then passed through to the gearbox block.

$$T_{mgb} = \frac{T_{wheel}}{Gear\ ratio} \tag{4.11}$$

$$\omega_{MGB} = \omega_{wheel} * Gear\ ratio \tag{4.12}$$

$$\frac{d\omega_{MGB}}{dt} = \frac{d\omega_{wheel}}{dt} Gear\ ratio \tag{4.13}$$

The power transferred through the transmission is also calculated using equation (4.14)

$$P_{MGB} = \omega_{MGB} T_{MGB} \tag{4.14}$$

A simple Matlab script is used to look up the correct gear ratio for the selected gear. Before passing the values through to the engine block the *over torque* detection system checks if the available torque of the engine has been exceeded. In that case the *over torque* detection system forces the model to reiterate the current time step with the selected gear reduced by one. The power flow block is used to detect in which direction the power is flowing through the transmission by simply detecting if the applied torque is positive or negative. This is necessary in order to compensate for transmission efficiency correctly.

## 4.5 Engine block

The engine block calculates the amount of fuel required to generate the required torque from the engine. The block is based on a torque curve and volumetric efficiency map. The efficiency map is a function of engine speed and engine torque, it is different for each engine type and is a critical part of the fuel consumption calculations. Only the maximum efficiency was known and therefor the whole map had to be estimated based on that point. An efficiency map from a 2-litre diesel engine was used as a reference and the map was then fine-tuned when the model was validated.

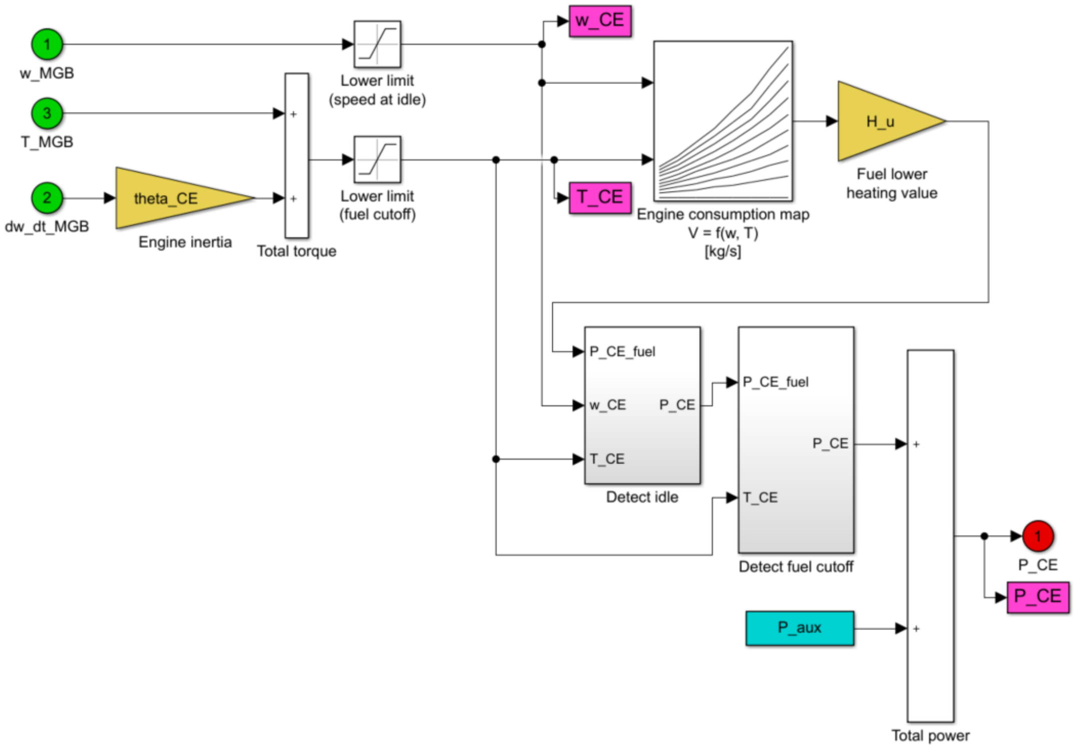


*Figure 4.6: The engine block.*

The engine block is presented in Figure 4.36. This figure shows how the block uses the inputs from the gearbox block to calculate the total power required by the engine. The total torque required from the engine is calculated using equation (4.15). A lower limit is added to the engine speed and torque to compensate engine idling.

$$T_{CE} = T_{MGB} + \frac{d\omega_{MGB}}{dt}\,\theta_{CE} \tag{4.15}$$

Where $\theta_{CE}$ is the engines' rotating assembly inertia.

The engine torque ($T_{CE}$) and the engine speed ($\omega_{CE}$) are then fed into the volumetric efficiency map and the idle detector. A fuel cutoff block is also fed with the engine torque to detect if fuel should be cut to the engine in case of negative torque request. Finally, the total fuel power ($P_{CE}$) is passed through to the last block, the fuel tank block.

## 4.6 Fuel tank block

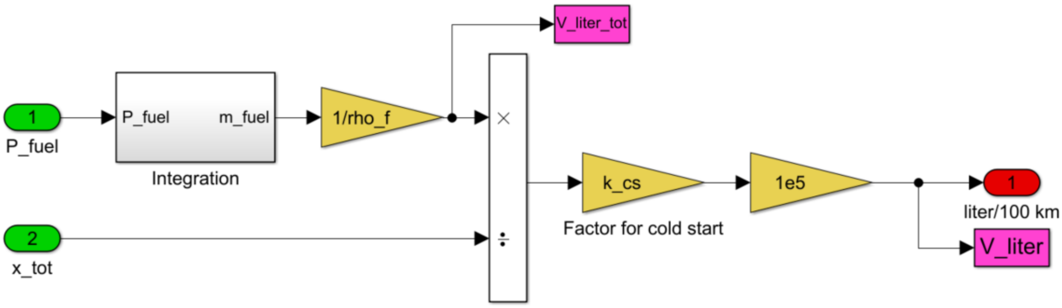


*Figure 4.7: The fuel block.*

The fuel tank block is presented in Figure 4.7. The figure shows how the block uses the power input from the engine block ($P_{fuel}$) to calculate the total fuel mass used during the cycle by intergrading the power. Inside the integration the power is also multiplied with the lower heating value of the fuel to convert the power to mass of fuel. Last the average fuel consumption in L/100km is calculated using equation (4.16).

$$\frac{L}{100km} = \frac{\frac{m_f}{\rho_f} * 10^4}{x_{tot}} \tag{4.16}$$

The model also includes an optional cold start factor to compensate for richer fuel mixture during cold starts. The cold start factor is not considered in this thesis.

# 5 Model validation

The real car model was validated by comparing the simulation results to the recorded data from the three parts of the driving cycle created. These three parts represent three different driving scenarios with similar road surface; driving uphill, driving downhill, and driving flat ground. More information about the parts can be found in table 5.1.

*Table 5.1: Driving cycle parts information.*

| Driving cycle type | Flat road | Uphill | Downhill |
|---|---|---|---|
| Cycle time [mm:ss] | 36:16 | 27:02 | 17:19 |
| Distance [Km] | 13,11 | 5,10 | 5,32 |
| Average speed [Km/h] | 21,7 | 11.3 | 18.4 |
| Altitude change [m] | 0 | 564 | 564 |
| Slope [%] | 0 | 10,5 | 10,4 |

The vehicle's ECU calculates the torque output from the engine in percentage. To convert that into SI unit the vehicles torque curve must be known throughout the RPM range of the engine. Since Toyota was not able to supply the actual torque map of the engine, it had to be estimated from the recorded data. These values are not fully accurate.

The real torque was therefor calculated using the same torque map as used in the real car model, using equation (5.1).

$$T_{est}(\omega, T_{ecu}) = T_{ecu} \cdot T_{max}(\omega) \tag{5.1}$$

This can then be used to compare and evaluate the simulated torque, to see if the simulated wheel speed is close to the actual wheel speed.

When comparing the calculated and recorded engine speed there is a noticeable difference between the data throughout the cycle. As mentioned before, this is because the model does not include any wheel slip. A time step of 1 second might be too large but due to time constraints there was no work performed to reduce this size. A visual comparison of the data can be seen in Figure 5.1.

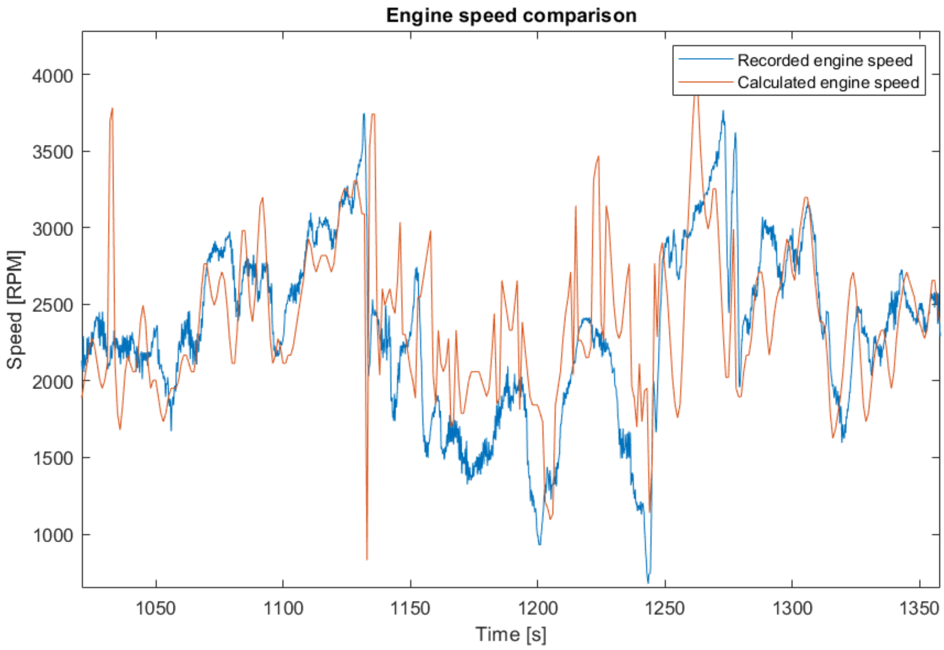


*Figure 5.1: Engine speed difference compared.*

For this reason, it was concluded that comparing the power consumption and the total energy consumption might be more relevant in this case. The output power was calculated using equation (5.2) and (5.3) The total energy consumption was calculated by integrating the power over time, see equation (5.4).

$$P_{real} = \omega_{eng} * T_{est} \tag{5.2}$$

$$P_{sim} = \omega_{sim} \cdot T_{sim} \tag{5.3}$$

$$E = \int P \ dt \tag{5.4}$$

# 6 Results

## 6.1 Real car model validation

### 6.1.1 Power demand optimized

The model was first validated by comparing the difference in power demand. The results from this model validation are shown in table 6.1.

*Table 6.1: Power demand optimized model verification.*

| Driving cycle type | Uphill | Downhill | Flat road |
|---|---|---|---|
| Average recorded power [kW] | 53,0 | 35,7 | 62,1 |
| Average calculated power [kW] | 38,7 | 17,1 | 61,5 |
| Total recorded energy used [kWh] | 23,8 | 10,3 | 37,6 |
| Total calculated energy used [kWh] | 23,6 | 10,5 | 37,6 |
| Recorded fuel consumption [L/100km] | 91,3 | 33,3 | 62,2 |
| Calculated fuel consumption [L/100km] | 116,5 | 49,7 | 71,8 |
| Recorded energy consumption [kWh/100km] | 446,4 | 188,9 | 282,9 |
| Calculated energy consumption [kWh/100km] | 443,6 | 191,9 | 283,6 |

Vehicle parameters used in this model are presented in table 6.2.

*Table 6.2: Power demand optimized vehicle parameters.*

| | |
|---|---|
| **Frontal area** | 3.13 $m^2$ |
| **Wheel diameter** | 0.75 $m$ |
| **Drag coefficient** | 0.394 |
| **Rolling friction coefficient** | 0.42 |
| **Drivetrain efficiency** | 77.5 % |
| **Road gradient gain** | 2.2 |
| **Torque limit margin** | 28 Nm |

Figure 6.1 to 6.3 show how the correlation between the recorded data and the results from the simulation model. Both the energy consumed correlation as well as the fuel consumed correlation are shown.

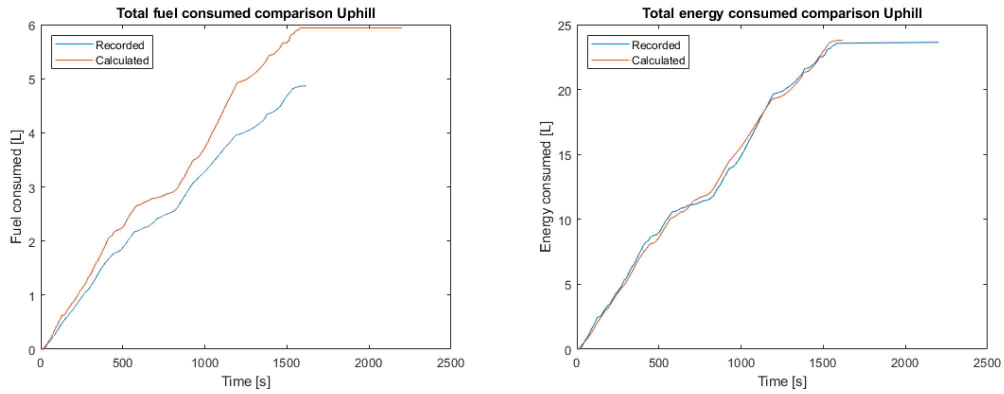


*Figure 6.1: Fuel and energy consumption comparison – Uphill cycle.*

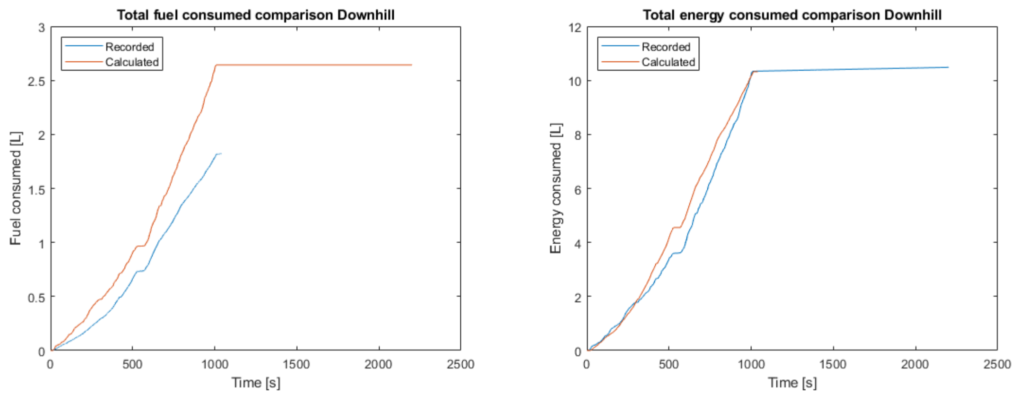


*Figure 6.2: Fuel and energy consumption comparison – Downhill cycle.*

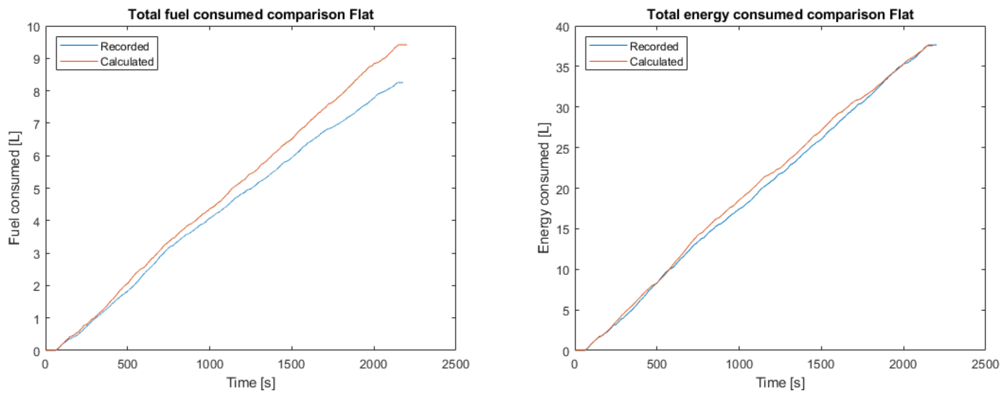


*Figure 6.3: Fuel and energy consumption comparison –Flat cycle.*

### 6.1.2 Fuel consumption optimized

The model was also validated by comparing the difference in fuel consumption. The results from this model validation are shown in table 6.3.

*Table 6.3: Power demand optimized model verification.*

| **Driving cycle type** | **Uphill** | **Downhill** | **Flat road** |
|---|---|---|---|
| Average recorded power [kW] | 53,0 | 10,3 | 62,2 |
| Average calculated power [kW] | 30,9 | 6,8 | 49,4 |

| | | | |
|---|---|---|---|
| Total recorded energy used [kWh] | 23,8 | 35,7 | 37,6 |
| Total calculated energy used [kWh] | 18,9 | 11,1 | 30,2 |
| Recorded fuel consumption [L/100km] | 91,3 | 33,3 | 62,2 |
| Calculated fuel consumption [L/100km] | 95,9 | 33,9 | 60,0 |
| Recorded energy consumption [kWh/100km] | 446,4 | 188,9 | 282,9 |
| Calculated energy consumption [kWh/100km] | 354,9 | 123,8 | 227,6 |

Vehicle parameters used in this model are presented in table 6.4.

*Table 6.4: Fuel consumption optimized vehicle parameters.*

| | |
|---|---|
| **Wheel diameter** | 0.65 $m$ |
| **Drag coefficient** | 0.394 |
| **Rolling friction coefficient** | 0.30 |
| **Drivetrain efficiency** | 80 % |
| **Road gradient gain** | 1.8 |
| **Torque limit margin** | 0 Nm |

Figure 6.4 to 6.6 show how the correlation between the recorded data and the results from the simulation model. Both the energy consumed correlation as well as the fuel consumed correlation are shown.

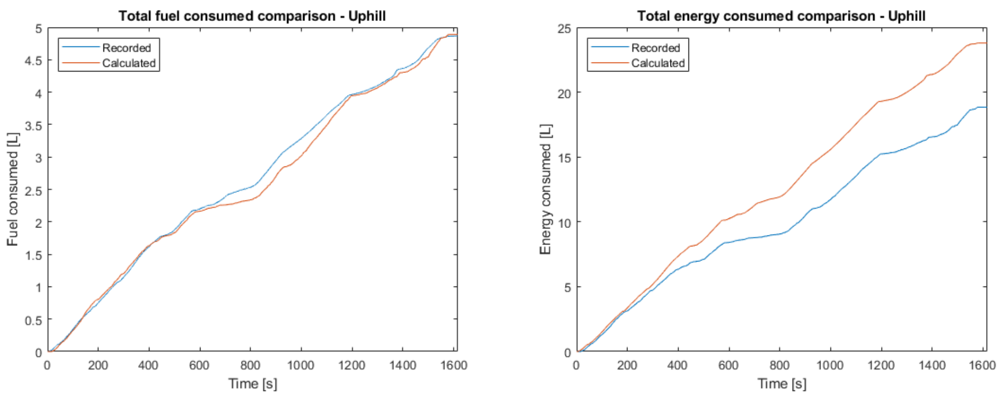


*Figure 6.4: Fuel and energy consumption comparison – Uphill cycle.*

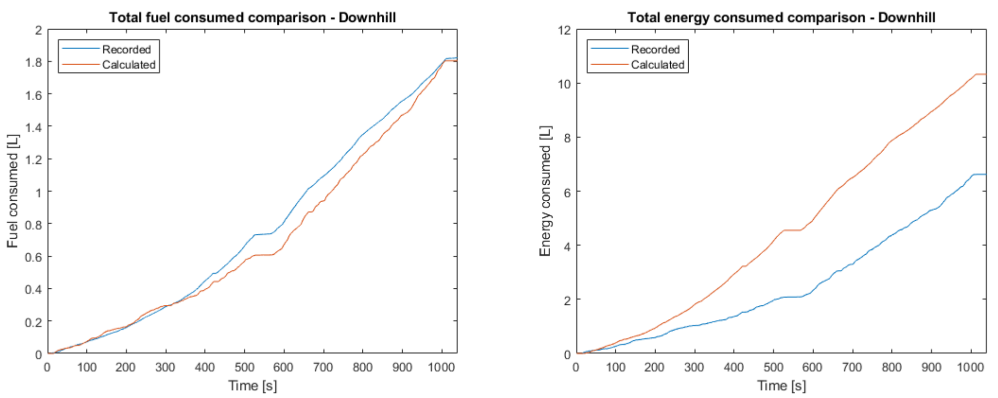

*Figure 6.5: Fuel and energy consumption comparison – Downhill cycle.*

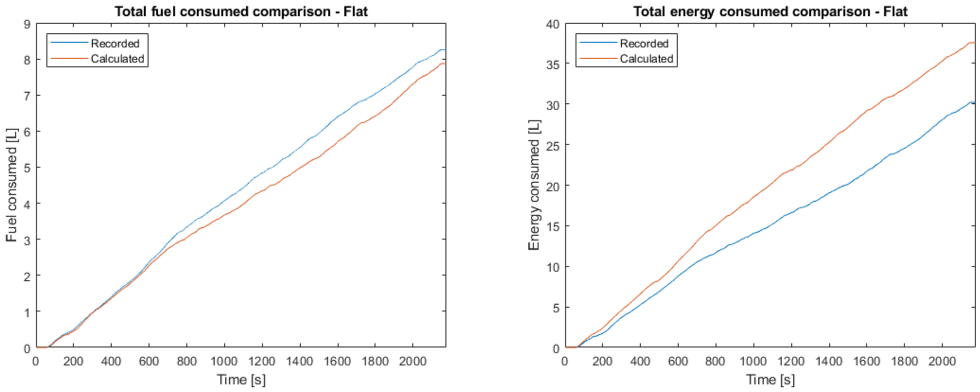


*Figure 6.6: Fuel and energy consumption comparison – Flat cycle.*

## 6.2 Hybrid concept

The second research question was to estimate how much power a hybrid powertrain in an Arctic Trucks vehicle should be able to produce.

The answer to this question can be divided into two sections, power requirements in a long period and power requirements in a short period.

The data recorded with the test vehicle was used to estimate a suitable battery size as well as electric motor size for a hybrid vehicle that would be based on the test vehicle used in this thesis.

A visual presentation of the power demand and the engine load is presented in the histograms in Figure 6.7, 6.8 and 6.9. The bars in the histograms are based on counts.

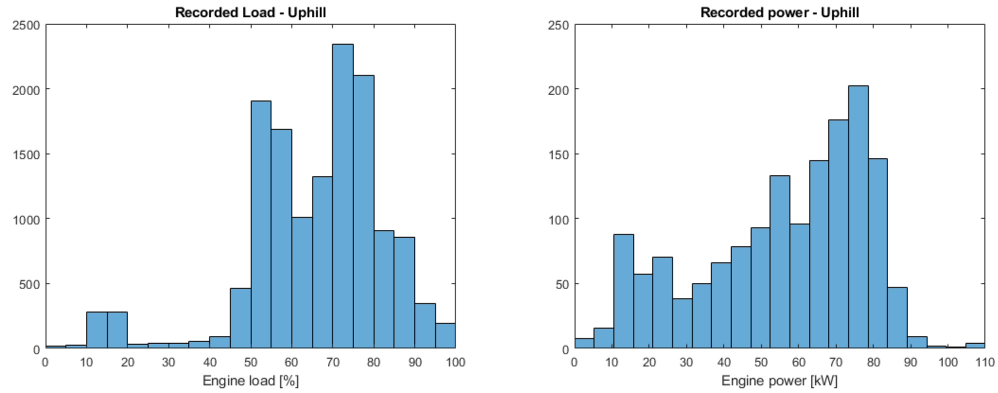


*Figure 6.7: Histograms showing the power demand and engine load distribution - Uphill cycle.*

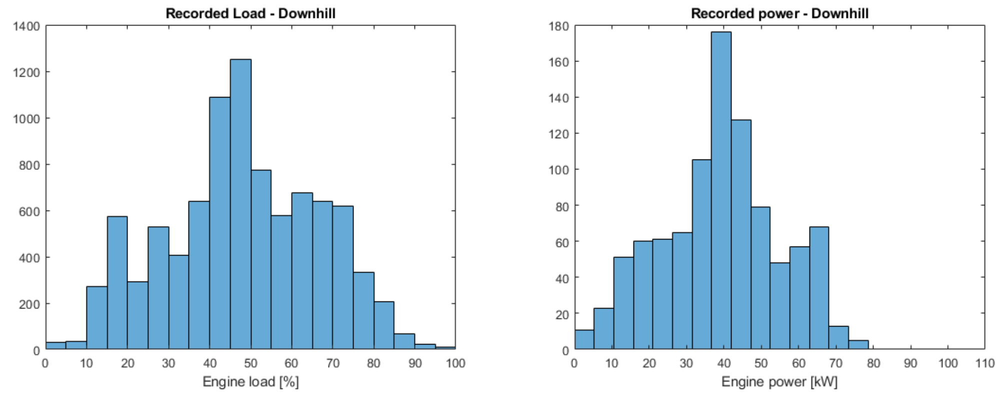


*Figure 6.8: Histograms showing the power demand and engine load distribution - Downhill cycle.*

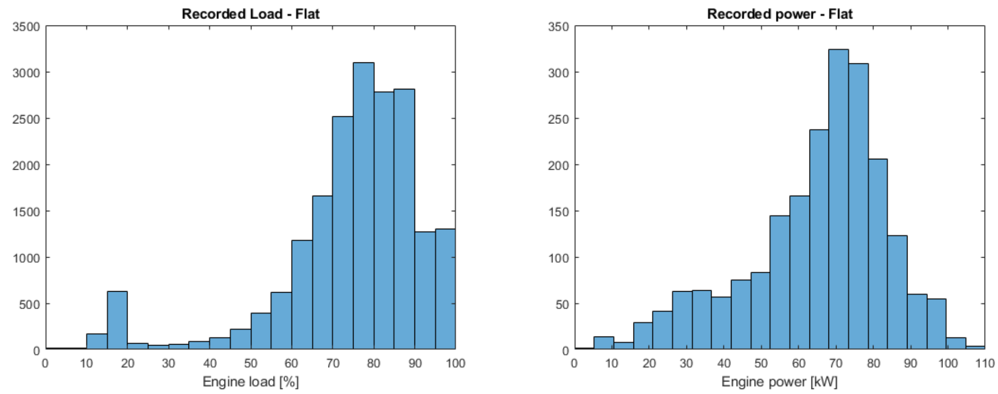


*Figure 6.9: Histograms showing the power demand and engine load distribution - Flat cycle.*

The power demand distribution throughout the cycles was investigated to estimate the long- and short-term power demand. In Figure 6.10 to 6.12 you can see both the actual power demand as well as the moving averaged power demand of 10 seconds. The same figure includes the 60% - 90% percentile power demand.


*Figure 6.10: The figure shows the total power demand for uphill cycle, both raw data and moving average over 10 seconds, along with the 60-90 percentile lines.*

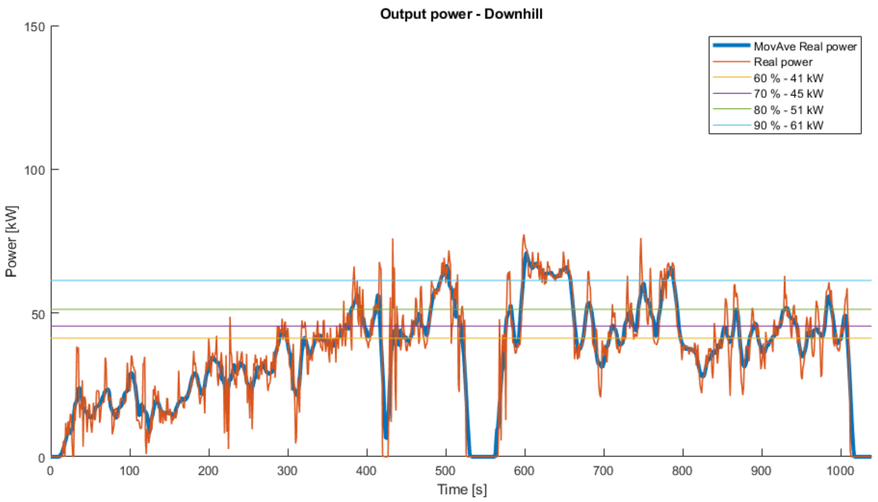


*Figure 6.11: The figure shows the total power demand for downhill cycle, both raw data and moving average over 10 seconds, along with the 60-90 percentile lines.*

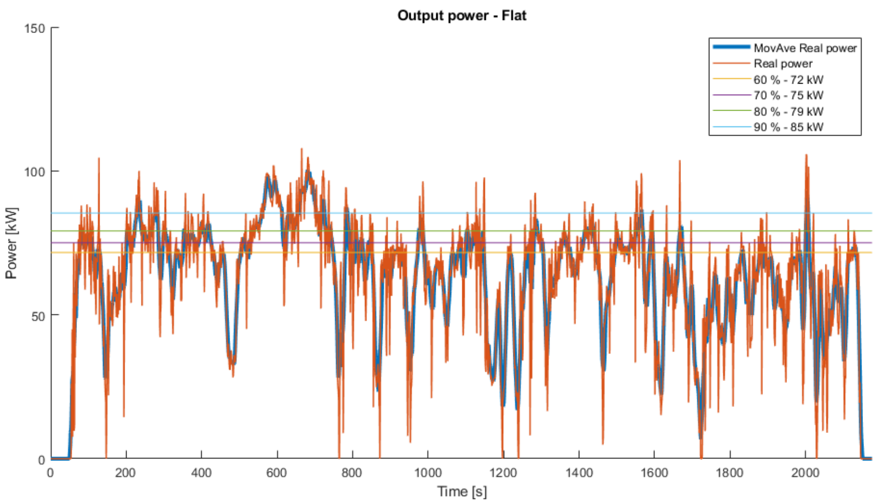


*Figure 6.12: The figure shows the total power demand for flat cycle, both raw data and moving average over 10 seconds, along with the 60-90 percentile lines.*

Based on this data in Figure 6.12, the largest power demand is when driving on a flat surface. This is mostly because the average speed was significantly higher during this cycle compared to the other cycles. The vehicle was usually driven as fast as the conditions allowed. During the uphill cycle the largest speed limitation was the amount of traction force as the vehicle struggled for grip. The speed limitation in the other cycles was mostly constrained to the surface roughness that the vehicle was driven on.

# 7 Discussions

The result show that the power demand was below 85 kW 90% of the time and below 75kw 70% of the time when driving on a flat road with an average speed of 21 km/h. This shows that in a hybrid vehicle the ICE could potentially be downsized to 85 kW while maintaining that same drivability as long as the electric motor added would be able to support the power demand above 85 kW. The added weight the comes with the hybrid powertrain would also increase the power demand which would also need to be covered by the electric motor.

The model is very flexible and can estimate the power demand and fuel consumption for any vehicle and any recorded driving cycle with recorded vehicle speed, road gradient and selected gear. The model is also independent of the vehicle used to record the cycle. This is very useful for Arctic Trucks as they have multiple vehicles that are all used for driving in similar conditions.

The estimated power demand can then be used to size the conceptual hybrid powertrain for an Arctic Trucks vehicle. Arctic Trucks idea is to make an parallel hybrid vehicle where the main power is produce by the ICE and then an electric motor would support the exceeding power demand. By using the model Arctic Trucks could estimate the electric motor size as well as the battery size needed to propel vehicles through different driving cycles.

## 7.1 Model validation

It can be seen from the results in chapter 6.1 that the model created during the thesis work can be concluded as a valid model with under 5% deviation for both models. The model created by using the power demand optimization is more accurate an was therefor used for the hybrid concept work in this thesis.

In a vehicle model like the one presented in this thesis there are multiple sources of errors that can affect the results. In this thesis many variables had to be estimated due to time constrains, limited access to measurement equipment or lack of resources.

The reason why two models were needs was because the engine volumetric efficiency map was unknown and therefor had to be estimated. This made tuning of the unknown parameters hard and therefor it was not possible to get the model to estimate the fuel consumed and energy consumed correctly at the same time.

## 7.2 Possibilities for hybrid Arctic Trucks vehicles

During this thesis work it became clear that regenerative braking would not be beneficial in heavy snow conditions, as the brake pedal was barely used. Even when driving down a hill with 10% road gradient the brake pedal was never used. In case the driver had to slow down the car only the throttle request had to be reduced.

This is mostly due to very large rolling resistant coefficient which can both be related to the snow it as well as the tyres. When driving with low air pressure in the tyres significant amount of power is absorbed by the tyres as they deform.

Hybrid vehicles are known for there great fuel saving capabilities compared to “regular” vehicles only powered by an internal combustion engine. In most hybrid vehicles 20-30% of that fuel saving is gained by regenerative braking.

This means that for a typical off-road cycle Arctic Trucks would not be able to fully make use of the benefits of hybrid vehicles. On the other hand there is another unusual situation where Arctic Trucks could potentially use the benefits of hybrid vehicles. During Arctic Trucks missions on the Antarctica their cars are kept running 24/7 as shutting of the engine in the extreme conditions found on the Antarctica can create huge problems when it comes to turning the cars on again. The cold conditions with up to -40°c lead to water contained in the fuel freezes and clogs up the fuel filters. Also, starting up the engine with the engine oil temperature that low can cause damage to the engine. Arctic Trucks could therefor make use of a hybrid vehicle that would be able to charge the battery overnight while keeping the engine running.

## 7.3 Further research

### 7.3.1 Rolling resistance and road condition correlation

The vehicle model was verified based on data collected in one day where the road conditions were roughly the same at all time. Further work should be performed in different road conditions, specifically in deeper snow as well as less snow. Also, the resistance in the snow varies a lot with the density of the snow, which is highly dependent on the time since last snowfall as well as the air temperature. The model created in this thesis could be used to carry out this research by simply recording data from the same path on different days in different conditions and then estimating the rolling resistance of the tyres for different conditions.

### 7.3.2 Torque dependent efficiency

In this thesis the powertrain efficiency is always assumed to be constant for simplification. This is not the actual case in reality, as research have shown that transmission efficiency highly depends on the torque applied to the transmission.

As the vehicle investigated in this thesis is a 4x4 driven car, it even has an additional set of gears in the transfer case which accumulates the torque dependent efficiency even more. Introducing a hybrid power train would add more sets of gears to the system again increasing this effect.

The model could be modified to encounter for this effect. Further research on the gearbox block used in the model would then be required.

# 8 References


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[2] National Academies of Sciences, Review of the 21st Century Truck Partnership: Third Report, 2015.

[3] L. Hella KGaA Hueck & Co., Wheel speed sensors in motor vehicles.

[4] Daimler, “Unimog U 4023 and U 5023: Setting standards for off-road mobility,” Daimler, [Online]. Available: https://media.daimler.com/marsMediaSite/ko/en/9903771. [Accessed 23 May 2019].

[5] H. Arngrímsson, “Technical specification Toyota Hilux AT38,” Arctic Trucks, 2012.

[6] N. R. Council, Cost, Effectiveness, and Deployment of Fuel Economy Technologies for Light-Duty Vehicles, The National Academies Press, 2015.

# 9 Appendix A

| | | |
|---|---|---|
| **Tyre size** | 265/65R17 | 38x12.5R15 |
| **Wheel size** | 17xJ8 | 15xJ12.5 |
| **Front suspension lift** | - | 90 mm |
| **Rear suspension lift** | - | 90 mm |
| **Suspension modification** | - | Upgraded suspension, stiffer springs and larger dampers, Wheel alignment adjusted. |
| **Height** | 1820 mm | 2050 mm |
| **Width** | 1835 mm | 2160 mm |
| **Wheelbase** | 3085 mm | 3130 mm |
| **Track length** | 3130 mm | 3130 mm |
| **Bodywork** | - | Fender trimming for clearance. Fibre glass fender flares. Side steps. |
| **Final gear ratio** | 3,909:1 | 4,88:1 |
| **Curb weight** | 1870 kg | 2060 kg |