Institutionen för systemteknik
Department of Electrical Engineering

Examensarbete

Fuel Consumption Estimation for Vehicle Configuration Optimization

Examensarbete utfört i Fordonssystem
vid Tekniska högskolan vid Linköpings universitet
av

Fredrik Söderstedt

LiTH-ISY-EX--14/4775--SE

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Linköpings universitet
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Bränsleförbrukningssimuleringar för optimering av fordonsspecifikationer
Fuel Consumption Estimation for Vehicle Configuration Optimization

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Sammanfattning
Abstract

Fuel consumption is one of the factors that are considered when deciding a vehicle's optimal specification. In order to swiftly estimate the fuel consumed during real world driving scenarios, a simulation tool has been developed that is well suited for vehicle configuration exploration applications. The simulation method described in this paper differs from the static calculation method currently in use at Scania CV since the dynamic translation of the vehicle is considered, yet the simulation time is kept low. By adopting a more dynamic approach, the estimation accuracy is increased and simulation of fuel saving technology, e.g. intelligent driver support system, is enabled.

In this paper, the modeling and implementation process is described. Different approaches is discussed and the choices made during the development is presented. In order to achieve a low simulation time and obtain a good compatability with Scania's current software application, some of the influencial factors have been omitted from the model or described using simple relations. The validation of the fuel consumption estimation indicates an accuracy within three percent for motorway driving.

Utilizing the newly devised simulation tool, a look-ahead cruise controller has been implemented and simulated. Instead of continuously finding the optimal control signals during the driving scenario like most look-ahead controllers, a dynamic programming algorithm is used to find a fuel efficient speed profile for the entire route. The speed profile is used as the reference speed for a conventional cruise controller and comparison with another simulation tool indicates that this is a fast and accurate way to emulate a real look-ahead controller.

Nyckelord
Keywords

Fuel consumption, Computer simulation, Heavy duty vehicle, Vehicle optimization, Look-ahead control

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Fredrik Söderstedt

Contents

Notation

Abbreviation

Acronym	Meaning
ACEA	European Automobile manufacturers association
ASC	Ammonia slip catalyst
CC	Cruise controller
CO_2	Carbon dioxid
DOC	Diesel oxidation catalyst
DP	Dynamic programming
DPF	Diesel particulate filter
EGR	Exhaust gas recirculation
FSM	Fuel simulation model
HDV	Heavy duty vehicle
LACC	Look-ahead cruise controller
MPC	Model predictive controller
NO_x	Nitrogen oxide
OPC	OptiCruise
PID	Proportional, integral, differential (controller)
SCOP	Scania optimizing program
SCR	Selective catalytic reduction
STARS	Scania truck and road simulation
VECTO	Vehicle energy consumption calculation tool
VO	Vehicle Optimizer

1 Introduction

This report describes the masters thesis work, *Fuel Consumption Estimation for Vehicle Configuration Optimization*, with the goal of devising a fuel consumption estimation method that is well suited for Scania's vehicle configuration exploration application, VO. The work is conducted at Scania CV AB in Södertälje.

In this chapter, the background and purpose of the conducted work is described.

1.1 Background

Scania is one of the leading manufacturers of heavy duty trucks, busses and engines. The modular production system used by Scania CV enables a variety of customization options and each vehicle is tailored after customer specific demands [1]. According to Scania, consulting a distributer before deciding the specification can improve the fuel economy by up to ten percent [2]. Determining the vehicle configuration that gives the lowest fuel consumption is however, not an easy task. To facilitate this process, several tools are available.

Vehicle Optimizer (VO) is an software application used by Scania distributors and sales personnel. It calculates how the weight, dimensions, performance and fuel consumption of a truck or bus changes with different specifications. However, the method used to estimate fuel consumption in VO is only valid for a comperative purpose. The impact on fuel consumption when changing the vehicle specification is accurately depicted but it does not fully represent the actual fuel consumed during a real world driving scenario. Furthermore, the static nature of the estimation method makes it difficult to describe the influence of the driver or fuel saving technologies.

Since an improved fuel economy can be used as a powerful incentive when attracting customers, it is desirable to investigate if the accuracy of the estimations in VO can be improved.

1.2 Purpose and problem description

Vehicle technology is constantly progressing and becoming increasingly more advanced. In turn, the simulation tools used to describe vehicle behaviour needs more complex models to accurately depict the vehicle as a system. Unfortunately this often leads to a longer simulation time. Furthermore, large and complex simulation models require more effort to update and maintain, implementing new submodels and control algorithms becomes increasingly difficult.

VO is mostly used by distributers to find vehicle specifications suitable for their respective markets and by sellers to give customer recommendations. Therefore a complex simulation model with numerous inputs and an extensive simulation time is not desirable. The thesis work described in this report focuses on improving the fuel consumption estimations while preserving the beneficial attributes already present in VO, i.e. a short calculation time and the ability to describe a variety of different vehicle specifications without adjusting additional parameters.

In order to achieve this, the trade-off between simulation speed and estimation accuracy is considered when devising the fuel consumption estimation method. Additionally, since VO can be used to examine most of the vehicle specifications available at Scania CV, usage of already measured vehicle parameters in the simulation model is prefered. If a model were to be used, where new parameters needs to be measured or estimated, an extensive amount of effort need to be spent on remaking the component library in VO.

1.3 Method

To find a suitable method to estimate the fuel consumption, the causes of energy losses of different vehicles and driving tasks are examined in chapter 2. additionally, four different simulation tools that estimates the fuel consumption is studied in chapter 3. These simulation tools uses different approaches to simulate the longitudinal motion of vehicles and the resulting fuel consumption. The benefits and drawbacks of the different methods are presented and from the conclusions of the study, the approach used in this thesis is decided.

The complete vehicle model and implementation choices are tested and validated by creating a prototype of the simulation tool in MATLAB but is planned to be implemented in VO which is written in C#. Although validation of the fuel consumption estimation against measured data is performed, most of the validation is done by comparing the prototype with Scania's simulation tool STARS which is described in section 3.2.

One of the reasons to change the fuel consumption estimations in VO is to improve the ability to describe fuel saving technology e.g. hybridization and intelligent driver support systems. Therefore a look-ahead cruise controller is implemented and tested. Different approaches and implementation variations of look-ahead control is studied in order to find a suitable method for implementation in VO.

1.4 Delimitations

Simulating the entire vehicle in transient drive cycles is a difficult task and is therefore less prioritized. Instead, Focus lies on accurately estimating the fuel consumption during haulage operations which often is characterized by a high cruising speed and few amounts of stops, e.g. motorway driving.

Although the component library in VO contains bus specific components, no distinction between busses and trucks are made in the modeling process.

1.5 Related research

Constructing models of different components or the vehicle as a whole is thoroughly described in litterature, e.g. [3, 4]. In these books, well established models of the driving resistances a vehicle has to overcome are presented. Such models are often used when describing a vehicle's longitudinal motion and is also utilized in this thesis.

When performing simulations to estimate fuel consumption, the model of the vehicle is often reduced and only the most important factors for fuel consumption are included. Such models are described in [5, 6, 7]. The choices made during the modeling process in this thesis are mostly based on the conclusions from these papers. In all of them, modeling the engine is avoided by using engine fuel consumption maps to calculated the fueling based on the engine speed and torque. However, they all recognize the shortcomings of this method. Engine maps are measured under steady state conditions and lack the ability to describe the fueling during transient engine operation. Additionally, in order to uphold european emission legislations, EURO 6 engines uses different fueling strategies in order to ensure the functionality of the exhaust aftertreatment system.

The difference of steady state and transient fueling is studied in [5] by comparing a calculated steady state boost pressure with measured values. Although no quantification of the difference was made, it was concluded that the diesel engine of a heavy truck could be represented by an engine map during highway driving. In this thesis, simulations are used in an effort to actually quantify the importance of the transient fueling behaviour. Although simulations does not necessarily depict the actual difference, an assessment of the benefits of including the transient fueling behaviour in the model is obtained.

In order to simulate start and stop events without introducing chattering in vehicle speed, the friction model described in [20] is utilized. However, to increase the simulation speed it is slightly simplified by omitting the clutch from the simulation model.

Implementation and simulation of look-ahead controllers in heavy duty vehicles are described in e.g. [8, 9, 21]. A look-ahead controller uses information about the topology of the road to continuously find a fuel efficient way to control the vehicle speed. The fuel efficient control is most commonly found through an optimization method, e.g. dynamic programming. Since optimization algorithms often has a high computational complexity, a controller that continuously performs an optimization during the simulation is not well suited for VO.

Instead, the behaviour of a look-ahead controller is emulated by using the dynamic programming algorithm described in [10] to find a fuel efficient speed profile for the entire route which is used as the reference speed for a conventional cruise controller.

2 Fuel consumption

In this chapter, fuel consumption is discussed, the influence of the vehicle and the driver is studied. The chapter is wrapped up by discussing three approaches to estimate fuel consumption.

2.1 The vehicle's influence

There are numerous factors affecting the fuel consumption, both directly and indirectly. Examples of direcly influencing factors are the rolling resistance, air resistance and internal friction in the powertrain. These however, are influenced by ambient conditions such as the weather and the road quality. In figure 2.1 the distribution of driving resistances for different vehicles are illustrated. Clearly, there are no general rule to determine the largest energy consumer that is applicable for every individual vehicle. There are however several observable trends, in urban driving missions more energy is lost due to braking, which is expected of a driving scenario where the vehicle needs to start and stop often. Equally expected is the dominance of the air resistance during motorway driving. The high vehicle speed causes a large amount of air drag while the relatively static driving conditions minimizes the use of the brakes.

Studying the resistances of the entire driving scenario can be misleading if not put into the right context. When driving uphill, mechanical and kinetic energy is converted into potential energy. When driving downhill the process is reverted. The potential energy can however not always be reverted into kinetic energy, because this may cause the vehicle to exceed the speed limit. Instead the residual energy is dissipated by braking. This means that the energy used to climb hills will be represented by the engine and service brakes.

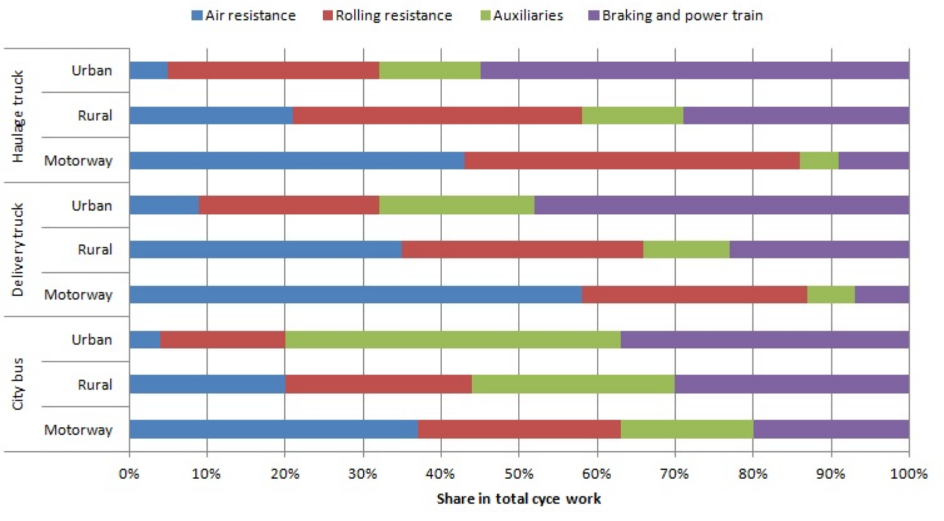

Figure 2.1: Distribution of the driving resistances for different vehicle types and driving scenarios [11].

If instead, the distribution of the power produced by the engine is studied, a more intuitive description of the energy losses is obtained, see figure 2.2. Simulations of a 40 ton truck performed in [5] showed that 41 % of the energy produced by the engine was converted to potential energy, 29 % was used to overcome the rolling resistance, 23 % was lost due to air resistance. The remaining 7 % was used to overcome the internal friction in wheel bearings and transmissions, as well as powering the auxiliary units.

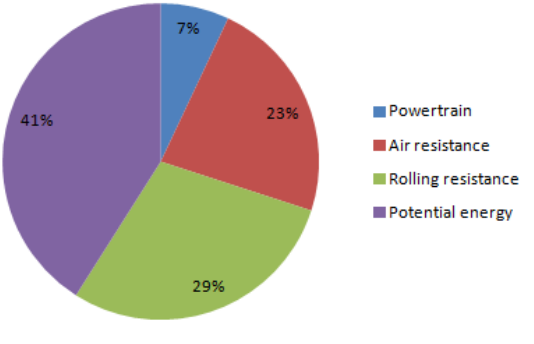

Figure 2.2: Illustration of how the energy produced by the engine is distributed. The values are obtained from a simulation of a 40 ton truck on a typical road performed in [5].

2.2 The driver's influence

As shown in figure 2.2, the energy consumption of a vehicle is highly dependant on how it is driven. Different driving missions results in different fuel consumption, but depending on the driver, the fuel consumption of the same vehicle driving the same route may also differ. A common way to decrease the fuel consumption in light vehicles is to maintain a constant cruising speed. This is however not always possible for trucks and busses. Due to their weight, a speed reduction in steep uphills is sometimes inevitable. Furthermore, maintaining a constant speed in steep downhills requires braking which dissipates some of the energy produced by the engine.

In [12] analytical solutions are used to find optimal speed profiles for heavy trucks. Coincidal with light vehicles, a constant speed was shown to be optimal for level road and in small road slopes. For steeper slopes, the fuel consumption can be reduced by efficiently utilizing the kinetic of the vehicle to accelerate in downhill slopes. If the slope is steep enough to cause the vehicle to accelerate even if the engine is not producing any torque it is beneficial to reduce the speed before the slope in order to minimize braking. These fuel optimal driving strategies are consistant with how an experienced driver operates in order to save fuel. It is also the behaviour expected of the look-ahead cruise controller presented in chapter 7.

Although the driver's influence on fuel consumption is substantial, human behaviour is hard to predict. The reactions of the driver are influenced by many unkown factors such as their training and experience. Therefore no elaborate driver model is constructed in this thesis work. Instead, the simulation model will be tested and validated by using predifined speed profiles as input.

2.3 Estimation methods

There are several approaches to estimate the fuel consumption of vehicles. In [3] three different methods are presented and they are described below.

Average Operating Point Method is used to estimate the energy required in a drive cycle. Using the energy consumption and the average speed of the drive cycle, an average engine operating point is calculated. The fuel consumption is then calculated with the assumption that the engine constantly operates in this point. This method is very fast and can therefore be used to give a rough estimate.

Backwards Modeling, also called inverse modeling, is a method to describe a vehicle's energy consumption based on the instantaneous speed and acceleration. Backwards models are often simulated using a quasi-static approach, where the drive cycle is divided into small parts where both the speed and acceleration are considered constant. In each of these segments, the required engine speed and torque output is calculated which in turn are used to estimate the fuel consumption. Since the fuel consumption is calculated from the prescribed speed, tracking of the drive cycle is exact. The total fuel consumed is obtained as the sum of the fuel consumption in every segment of the drive cycle. Pure quasi-static backwards simulation models does not have a driver model that controls the vehicle. The models thereby lacks the ability to compensate for the fact that the speed and acceleration of the drive cycle may exceed the vehicle performance.

Forward Modeling is a more intuitive way of describing the vehicle. In forward looking models, the engine produces a torque which is transfered through the rest of the powertrain to the wheels. This results in a tractive force that accelerates the vehicle. Forward models have the ability to describe important dynamics of the vehicle but generally have a longer simulation time than the other two methods [13]. Since acceleration of the vehicle is derived from the torque produced by the engine, the vehicle performance is naturally limited if the engine torque is restricted.

3
Simulation tools

To get a better understanding of how fuel consumption is estimated in different applications, four simulation tools are studied and compared in this chapter. Interesting and relevent solutions are enlightened to explain the modeling approach chosen in this thesis.

3.1 Vehicle Optimizer

Vehicle Optimizer (VO) is a simulation tool used to examine and compare different vehicle configurations. It is used to help optimize vehicles by providing information about their weight, performance, fuel consumption etc. VO is primarily used as a sales support tool for distributors, but also for internal estimations for research and development.

To estimate the fuel consumption, the road is divided into smaller segments based on the slope, see figure 3.1. The segments with similar slope is then lumped together and a method similar to the average operating point method is applied to estimate the fuel consumption.

In addition to the road topology, VO also take the transport task into consideration. One of the required inputs to the simulation is an estimation of the traffic density which are represented by a number of stops along the route. The number of stops has two major effects on fuel consumption. The average speed during the estimation is lowered and the fuel required to accelerate the vehicle is calculated and added to the total fuel consumption.

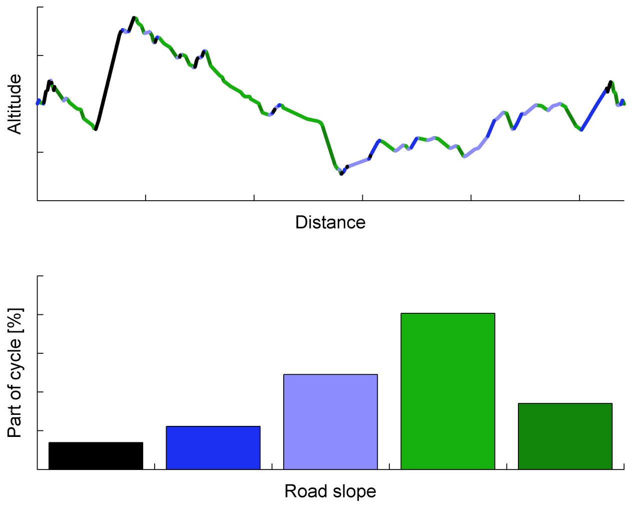

Figure 3.1: *In VO, parts of the road with similar slope are lumped together. The fuel consumption is then estimated using a method similar to the average operating point method described in section 2.3*

The absolute accuracy of the fuel consumption estimations in VO is rather low. The primary purpose is to depict the relative accuracy, e.g. how much the fuel economy is improved by changes made in the vehicle specification. Since simple models are used for the calculations, the time needed to compute the fuel consumption is very low. A single drive cycle of 500 km is simulated in approximately 0.1 seconds.

In a study of VO's predecessor SCOP it was concluded that the main reason to why the accuracy of this estimation method is low, is because it does not consider the dynamic behaviour of the vehicle during acceleration and retardation [6].

3.2 Scania Truck And Road Simulation

Scania Truck And Road Simulation (STARS) is a more powerful simulation tool. It is mostly used internally to develop and examine engines and their effect on the powertrain. STARS has a GUI programmed in Matlab where simulation parameters are configured. The actual models are written in Modelica and implemented in Dymola. In STARS a forward modeling approach is used and several dynamic processes, such as the heat transfer in the engine and catalyst, are taken into account. Furthermore, the CAN-bus is modeled which allows for a realistic simulation where the vehicle components are controlled the same way as in a real vehicle.

Although the primary purpose of STARS is to evaluate engines, the accuracy of the estimated fuel consumption has been verified to be within approximately six percent compared to measured values for long haulage routes [14]. The major drawback is the long simulation time. Since the simulations are time dependent the time required for a simulation varies, however a simulation of a drive cycle of 500 km can take up towards one hour to perform.

3.3 Vehicle Energy Consumption calculation Tool

The Vehicle Energy Consumption calculation Tool (VECTO) was developed to lay a foundation for monitoring and certification of CO_2 from heavy duty vehicles. The first version was developed by the Graz University of Technology and the Joint Research Centre of the European Commission [7].

In VECTO the fuel consumption is estimated using the backwards modeling approach where vehicle speed is the input. In order to limit the speed and acceleration according to the vehicle performance during the simulation, VECTO uses a forward looking driver module that converts the drive cycle into a cycle with realistic driving operation. The driver module also enables the simulation of a more realistic driver behaviour aswell as driver support system. Figure 3.2 shows how the actually simulated speed differs from the target speed. When driving uphill, the speed is reduced due to the increased driving resistance and when driving downhill, overspeed is allowed.

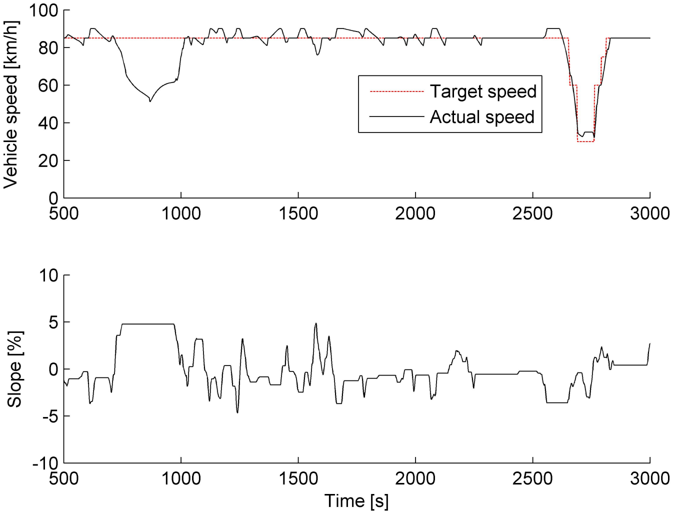

Figure 3.2: *Example of how a drive cycle in VECTO are converted to a cycle with realistic driving operation.*

Simulations in VECTO are relatively fast, a simulation takes a few minutes to perform, and has been proven accurate when validated and benchmarked against similar applications. A drawback with the backwards modeling approach is the inability to describe some vehicle technologies. Due to this limitation, VECTO may shift to the forward modeling approach in the future.

3.4 Fuel Simulation Model

The Fuel Simulation Model (FSM) was the result of a masters thesis work with the goal of finding a more accurate method to estimate fuel consumption than the one described in section 3.1 without inreasing the simulation time [6]. In FSM, the energy consumption of the vehicle was described by simple models like the one used in VO. FSM however, departed from the static simulation methods and instead adopted a more dynamic approach.

In an attempt to speed up the simulations while maintaining a good accuracy the models in FSM were discretized and made distance based. This enabled the implementation of an adaptive step length Δs. The idea was to use large steps when the vehicle velocity was constant and reduce the step size when the velocity was varying. The length of the steps was decided by

$$n = c_1 \cdot \frac{v_{ref} - v}{v_{ref}} + c_2 \cdot \frac{\alpha - \alpha_{max}}{\alpha_{max}} \tag{3.1}$$

where n is the scaling factor for the step length, v is the current velocity, v_{ref} is the desired speed, α is the road slope for the current distance step and α_{max} is the maximum slope the simulated vehicle can handle at the highest gear at cruising speed. c_1 and c_2 are constant used to adjust the impact of these conditions.

The use of distance based models has several drawbacks. When transforming a time based equation to distance based using the chain rule according to

$$\frac{df}{ds} = \frac{df}{dt} \cdot \frac{dt}{ds} = \frac{1}{v} \cdot \frac{df}{dt} \tag{3.2}$$

the models become invalid when the vehicle is standing still, due to division of the velocity. In FSM start and stops can't be simulated and has to be calculated using static methods.

The implementation of the variable step length also prevented the use of a PID-controller as the driver since this caused the system to become unstable [6]. Instead, a feed forward controller was implemented to adjust the speed. The resulting model behaviour is not consistant with how a vehicle is actally driven and resulted in an overestimation of the fuel consumption.

3.5 Conclusion

The simulation tools described in this chapter uses different methods to estimate fuel consumption, each application utilizes an approach that is well suited for their respective purpose. In this thesis work, the goal is to improve the accuracy of VO's fuel consumption estimations and therefore a suitable method has to be chosen.

3.5.1 Static vs dynamic simulations

The dominant source of error in the estimation method used in VO is the static nature of the simulations. Approximating dynamic processes as static will almost always result in some kind of discrepancy.

In section 2.1 it was concluded that a large amount of the engine's output power is used to climb up hills, i.e. is converted into potential energy. When driving downhill, the potential energy is converted into kinetic energy by allowing the vehicle's speed to increase. In static calculations, the vehicle is assumed to hold a constant speed and therefore lacks the ability to describe how the potential energy is utilized to decrease fuel consumption.

Using dynamic simulations doesn't ensure that the potential energy is handled as in a real driving scenario, it depends on how the model is implemented. For instance, the forward dynamic simulations in FSM lacks a driver model. Instead a feed forward cruise controller is used that tries to reach the reference speed in one integration step. Overspeeding is thereby not possible and the vehicle dynamic behaviour is only described during acceleration and retardation, not when driving downhill.

To improve VO's estimation of fuel consumption, the simulations needs to be changed from static to dynamic. This will however increase the simulation time if the same models are used due to the increased amount of calculations that needs to be performed.

Increasing the estimation accuracy is however not the only reason to use dynamic simulation, it also enables the implementation of fuel saving technology into the model. In this thesis a look-ahead cruise controller is implemented, see chapter 7. The simulation models used, are however easily modified and allows for more fuel saving technologies, e.g. hybridization, to be implemented.

3.5.2 Forward vs backwards modeling

One of the reasons that backwards models are faster to simulate, is that the time step used in the simulations can be large without creating an unstable system. In VECTO for instance, the time step is one second. Forward models are often expressed as a system of ordinary differential equations. This means that the time step used in forward simulations needs to be smaller than for backwards simulations, which in turn increases the simulation time.

Although VECTO manages to overcome many of the shortcomings of the backwards modeling approach by using forward looking control modules, there are still vehicle technology, such as certain control systems, that can't be fully depicted by backwards models [7] and therefore a forward modeling approach is used in this thesis.

4

Vehicle model

To get a fast and accurate estimation of the fuel consumption, an appropriate vehicle model needs to be implemented. It is important to point out that the model chosen in this thesis is constructed with consideration to VO and how the simulations could be used in VO. Three primary attributes, and the trade-off between them, are especially considered: simulation speed, estimation accuracy and flexibility.

Simulation speed is obtained by only modeling processes that are important for fuel consumption. Occurrences with fast dynamics, such as gear shifting, are modeled as instantaneous. By only modeling slow dynamics, such as the acceleration of a heavy vehicle, a relatively large time step can be used in the simulations and thus a lower simulation time can be obtained.

Estimation accuracy is achieved by including the most relevant factors for fuel consumption in the model. Some functionalities that are important for fuel consumption do however require complex models to be fully and accurately depicted. In such cases, simplified methods are used to describe their effect on the fuel consumption.

Flexibility is an attribute that is essential for a model if it is to be used in VO. The model has to be easily modified to describe different vehicles. In VO this is achieved by estimating the energy losses in the powertrain with look-up tables. Since the look-up tables are already implemented in VO, it is natural to use the same method in this thesis. The interpolation performed when using look-up tables can however increase the simulation time compared to estimating the energy losses with a polynomial. Using look-up tables has the benefit of increasing the flexibility without having to adjust any parameters, e.g. changing the gearbox only means changing the tables used for the interpolation and the gearbox ratios.

4.1 Engine

In this thesis, the inner workings of the engine are treated as less important since using a detailed model of the engine, including all control systems, would increase the simulation time. Instead, an affine relation between fueling and engine torque is assumed and the fuel consumption is estimated using look-up tables.

The engine output torque T_e is modeled as a function of the control input u_g and the allowed torque at the current engine speed. u_g is a fictive gas pedal, controlled by the driver model.

$$T_e = u_g \cdot (T_{max}(\omega_e) - T_{drag}(\omega_e)) + T_{drag}(\omega_e),\ u_g \in [0, 1] \tag{4.1}$$

The maximum output torque $T_{max}(\omega_e)$ and the drag torque $T_{drag}(\omega_e)$ are obtained from one dimensional look-up tables based on the current engine speed. In figure 4.1 the maximum torque and drag torque curves for an engine are plotted along with the operating points where the fuel consumption is measured.

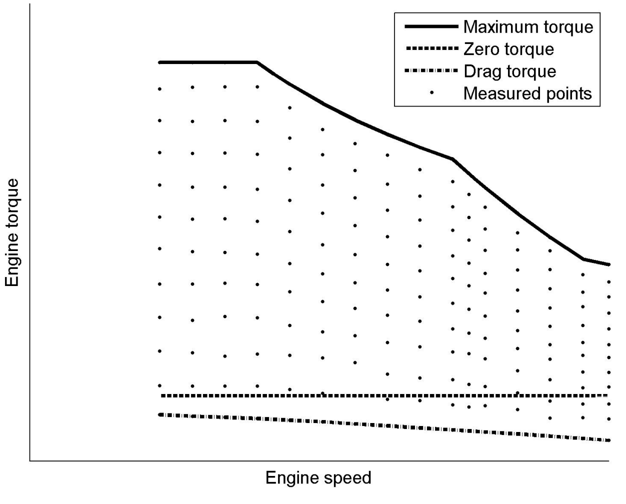

Figure 4.1: *Example of an engine map (values below a certain speed are hidden).*

Using Newton's second law of motion, the dynamics of the engine crankshaft is modeled as

$$J_e\dot{\omega}_e = T_e - T_c - T_{aux} \tag{4.2}$$

Where J_e is the engine inertia, ω_e the rotational speed, T_c the external load from the clutch and T_{aux} the load caused by the auxiliary units, se section 4.1.2.

4.1.1 Fuel consumption

The instantaneous fuel consumption $\dot{m}_{fuel}$ in each time step is obtain from engine fueling maps, measured under steady state conditions.

$$\dot{m}_{fuel} = f(\omega_e, T_e) \tag{4.3}$$

The total fuel consumed during the simulation is obtained by integrating the fuel consumption over the duration of the simulation

$$m_{fuel} = \int_{t_0}^{t_{end}} \dot{m}_{fuel}\, dt \tag{4.4}$$

Engine maps are often expressed in specific fuel consumption, using the unit g/kWh. Using such an engine map causes numerical errors when the output power goes towards zero. In VO, STARS and FSM this problem is avoided by using engine maps expressed in the unit mg/stroke. The solution chosen for this model is the same as the one used in VECTO where the engine maps are expressed in fuel mass consumed per time unit.

The interpolation method used to estimate the fuel consumption is based on Delaunay Triangulation where a network of triangles are created from the measured points, see figure 4.2. When the fuel consumption is to be calculated for an operating point, the triangle enclosing the point is located. If there is no triangle matching the critera, the algorithm doesn't return anything. If a valid triangle is found, the fueling value is calculated as a point on the plane of the triangle.

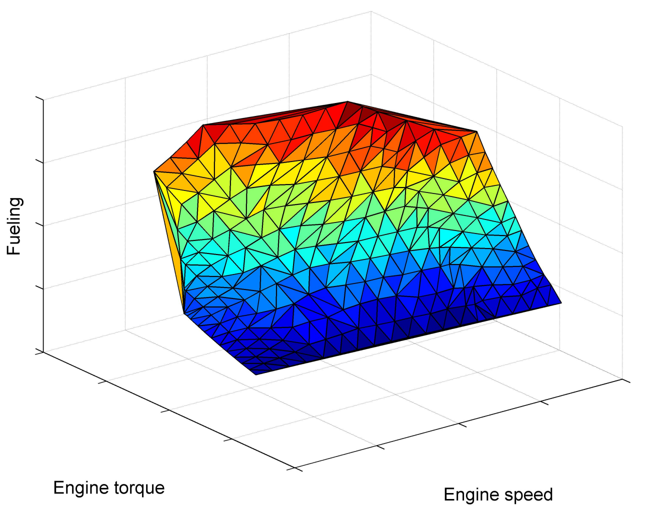

Figure 4.2: *The surface of triangles generated by the delauney triangulation.*

Although using a single engine map gives satisfactory estimation of the engine fueling during steady-state operation, this method lacks the capability to accurately describe the fueling during transient engine operation. There are several methods to increase the accuracy during these conditions.

For older generations of engines, i.e those that doesn't uphold the EURO 6 emission legislation, VO, STARS and FSM uses similar methods. They use two different engine maps, a steady state map and a dynamic map. In STARS and FSM the engine speed and fuel derivative determines when the dynamic engine map is activated. In VO the steady state and dynamic maps are combined into a single map where the ratio depends on the engine type. For actual EURO 6 engines, STARS uses several steady state engine maps due to the different heating strategies for the exhaust aftertreatment, see section 4.1.4. The fuel consumption derived from the steady state maps is then corrected to accurately describe the fueling during transient engine operation.

VECTO does not yet have method to deal with the shortcomings of using a steady state map. However, a correction factor is planned to be implemented in the future. The correction factor is planned to be determined by comparing the measured fuel consumption in a transient drive cycle with simulations of the same cycle performed with a steady state map [7].

To estimate the importance of depicting the transient fueling behaviuor, a series of simulations are performed in STARS using a EURO 6 engine. The fuel consumption calculated from the steady state maps and the actual fuel consumption presented by STARS are compared. Figure 4.3 shows the difference in fuel consumption over time for one of the simulations.

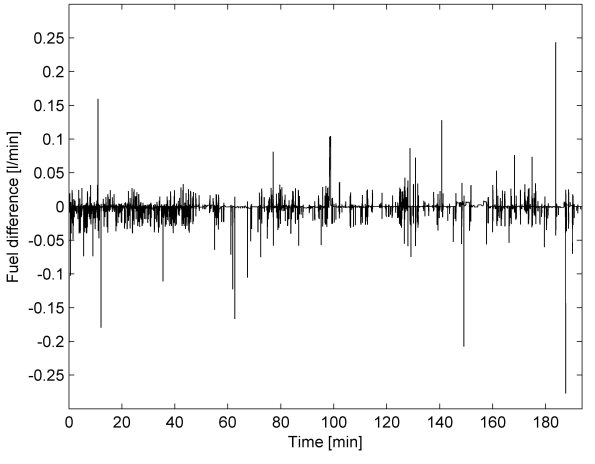

Figure 4.3: *The difference in instantaneous fuel consumption using two different estimation methods in STARS, using steady state engine maps and compensating for the transient fueling behaviour.*

In figure 4.4 the difference in instantaneous fuel consumption is plotted along with the engine torque and engaged gear for a smaller part of the simulation. The figure illustrates the major difference between the steady state and transient fueling behaviour. During gear shifting, spikes in the fuel difference can be observed. When the gear is changed, the engine operating point changes almost instantaneous. Estimating the fuel consumption with an affine relation such as a steady state engine map will result in an almost instantaneous change in fueling. In a real truck however, the fueling is controlled by a PID-controller [5].

Additionally, using a steady state engine map to estimate the fuel consumption lacks the ability to describe the engine dynamics such as the air transport into the intake manifold. The effect of this can be seen in figure 4.4 at approximately 5520 seconds into the simulation. When the torque is increasing rapidly, the difference in fuel consumption increases.

Even in steady state operation some discrepencies can be seen. Between the time 5560 and 5580 the engine is operating steady at full torque, yet a small fuel difference can be observed.

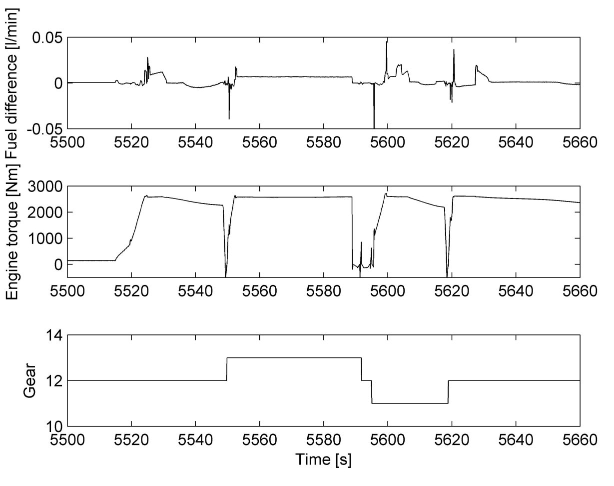

Figure 4.4: *Engine torque and difference in fuel consumption.*

Although the difference over time is apparent, the impact on the total fuel consumption is of more interest. Therefore, the total fuel consumed during the simulation is calculated for the comparison. STARS saves the results of the simulations in arrays with values sampled in 10 hz and the actual fuel consumption is presented both as a cumulative function and an instantaneous. The fuel consumption calculated from the steady state maps is however not presented as a cumulative function. So to avoid truncation errors influencing the comparison, both the steady state and dynamic fuel consumption is calculated by numerically integrating the instantaneous values using the same method. The results are presented in table 4.1. The difference of the two methods are relatively small, at most 0.2 percent.

		Fuel consumption	
Drive cycle	**Vehicle weight**	**Steady state**	**Transient**
ACEA coach	10 t	83.65 l	83.64 l
ACEA coach	30 t	102.93 l	102.97 l
ACEA inter-urban bus	10 t	32.40 l	32.30 l
ACEA inter-urban bus	30 t	60.98 l	60.96 l
ACEA haulage	10 t	23.21 l	23.17 l
ACEA haulage	30 t	34.62 l	34.61 l

Table 4.1: *Fuel consumption in different drive cycles, estimated with steady state engine maps as well as considering the fueling behaviour during transient engine operation. The values are obtained from simulations performed in STARS.*

Simulations in STARS doesn't necessarily reflect the impact of the transient fueling behaviour during real world driving scenarios. It is however not feasible to describe the transient behaviour better than STARS using a simplified model as the one constructed in this thesis. The fuel consumption is therefore estimated using steady state maps.

4.1.2 Auxiliary units

Modern vehicles are fitted with several auxiliary units such as electrical generators, cooling fans and air compressors. These are often mechanicaly powered by the engine which results in an increased load. According to [6], the power consumed by each auxiliary device can accurately be estimated with e.g. look-up tables or polynomials. The difficulty in describing the effects of the auxiliary devices is to determine when they are active and when they are not. The truck examined in [5] showed that the load to the engine when all auxiliaries was turned of was 24 Nm and the maximum load from the auxiliaries reached 270 Nm. When simulating the longitudinal motion of the vehicle, it is relevant to know when the devices are active. For instance, if they are active when driving uphill, the increased load to the engine can cause the vehicle to slow down and force the driver to shift gears. On the other hand, if the devices are active during a descent they can be powered by the vehicle's kinetic energy instead of the engine.

Due to the difficulty in determining which auxiliary devices are active at any given moment, the torque required to power the auxiliaries are modeled with the assumption that they require a constant power P_{aux}.

$$T_{aux} = \frac{P_{aux}}{\omega_e} \tag{4.5}$$

4.1.3 Exhaust emission reduction

The acceptable limits for exhaust emissions of new vehicles sold in the EU are regulated by the European emission standards [15]. One of the systems used to lower the amount of emissions in diesel engines is the exhaust gas recirculation (EGR). The primary function of the EGR system is to reduce the amount of nitrogen oxide (NO_x) formed during combustion by lowering the temperature inside the cylinder [6]. A portion of the exhaust gases produced by the engine are cooled and recirculated back into the engine cylinders along with fresh air. The decreased concentration of oxygen inside the cylinder lowers the peak temperature of the combustion flame. Furthermore, the exhaust gases are better at absorbing heat than air, which enables more heat to be expelled with the exhaust gases. The EGR system lowers the engine's efficiency and thus, increase the fuel consumption. The effects are included in the engine maps, so a model of the system is redundant when only considering fuel consumption.

4.1.4 Exhaust aftertreatment

In addition to preventing the formation of NO_x inside the engine cylinders, the emissions are further reduced by the exhaust aftertreatment system. The primary components of the aftertreatment system are fitted inside the muffler and consists of a diesel oxidation catalyst (DOC), a diesel particulate filter (DPF), selective catalytic reduction- (SCR-) catalysts mounted in parallel, followed by ammonia slip catalysts (ASC), see figure 4.5.

The DOC uses the excess air present in diesel exhaust gases to oxidize carbon monoxide and hydrocarbons into carbon dioxide and water. The amount of NO_x is not affected by the DOC but the NO is oxidized into NO_2 which reacts with the particles gathered in the DPF [16].

The DPF prevents emission of particulate matter, or soot, by physicaly stopping the particles. As the filter fills up, the efficiency decreases and the pressure drop over the DPF increases. The accumulated soot are then combusted in a process called regeneration.

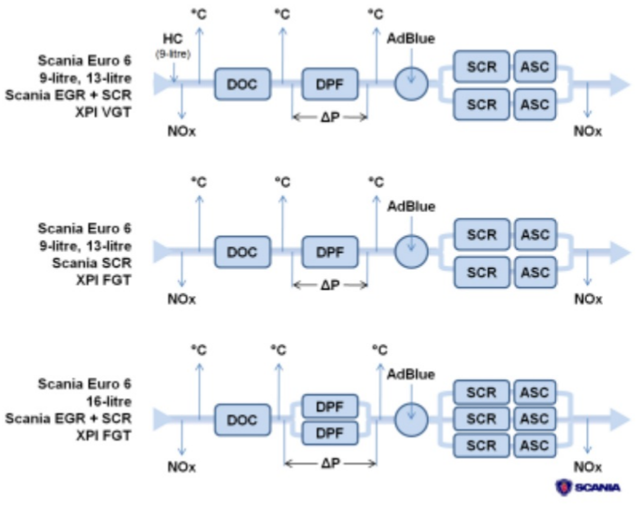

Figure 4.5: The exhaust aftertreatment system components that are fitted inside the muffler. NO_X-sensors are mounted at the inlet and outlet. The temperature is measured in every step and the pressure drop over the particle filter is measured to determine it's condition. [17]

The reduction of NO_x in the aftertreatment system is achieved in the SCR-catalyst. The oxidation process needs an additive called urea, also known as Adblue, to work. Urea is basically a solution of ammonia which reacts with the NO_x to form diatomic nitrogen (N_2), water and carbon dioxid. Any excess ammonia is removed by the ASC. A high temperature is important for the functionality of the SCR-catalyst [17], the system isn't active when the temperature drops below 200 °C.

The SCR system is a fuel efficient way of reducing emissions since it doesn't directly affect the fuel consumption. However, when the engine operates at low loads, the exhaust gases isn't enough to keep the system hot enough. If this happens, the engine control unit signals to increase the fuel injection rate. By burning more fuel in the engine, the exhaust gases gain more heat and the temperature in the SCR-catalyst is raised. However, the energy of the consumed fuel is not used to propell the vehicle.

Since the relation between fuel consumption and engine operating point changes when different fueling strategies, or modes, are applied, an engine map for each mode is implemented. The mode changing algorithm used in the simulation is described in section 5.4. The temperature of the exhaust gases are used to determine what mode is active. The temperature is obtained from look-up tables based on the engine's operating point.

4.2 Gearbox

The gearbox modelled in this thesis is a manual transmission that uses the automatic gear shifting system OptiCruise. The gearbox is assumed to be of mechanical type with no slipping parts. The energy losses is represented by an internal torque loss T_{Loss} obtained from measured values. The full gearbox model can then be expressed as

$$T_{out} = i \cdot T_{in} - T_{Loss} \tag{4.6}$$

$$\omega_{in} = i \cdot \omega_{out} \tag{4.7}$$

$$T_{Loss} = f(T_{in}, \omega_{in}, i) \tag{4.8}$$

Where in indicates the engine side of the gearbox, out indicates the wheel side and i is the ratio of the current gear.

The gear shifting sequence of OptiCruise is described in [18] as

1. OptiCruise signals that is time to change gear.
2. The engine is controlled in such a way that there is no torque on the propeller shaft.
3. The gear is disengaged.
4. The engine speed is adjusted for the new gear.
5. The new gear is engaged.
6. The engine increases the torque to match the original torque demand.
7. Control is returned to the driver.

However, in order to avoid increasing the simulation time, the changing of gear is modelled as instantaneous and without any energy losses.

4.3 Final drive

The final drive is used to divide the torque to the driving wheels. It also allows the wheels to rotate at different speeds when the vehicle is turning. since the model is restricted to longitudinal motion, the final drive can be approximated as a gearbox with a constant gear ratio and is therefore model with equations 4.6-4.8.

4.4 Wheels

The wheels are the vehicle component that transforms the driveline torque into a traction force that accelerates the vehicle. It is assumed that the vehicle is driven in such a way that there is no substantial loss of traction, so that the wheels can be modeled as

$$F_{trac} = T_{fd} \cdot r_e - F_{brake} \tag{4.9}$$
$$v = \omega_{fd} \cdot r_e \tag{4.10}$$

Where F_{trac} is the tractive force,T_{fd} and ω_{fd} is the final drive output torque and speed respectively, v is the vehicle speed, r_e the effective rolling radius and F_{brake} is the braking force from the service brakes.

4.4.1 Braking system

Excessive use of the service brakes may cause them to overheat and lose their effectiveness, this is called fading. The truck primarely use other braking systems such as the retarder or the exhaust brake to reduce the speed, if they provide enough braking power. For the purpose of estimating the fuel consumption, modeling the different braking systems are redundant. Instead it is assumed that the braking force requested by the driver is obtained at the wheels. Only the energy losses caused by the retarder are included in the model which are calculated from a one dimensional look-up table. The losses are expressed as a torque, counteracting the rotational motion of the propeller shaft.

4.5 Vehicle dynamics

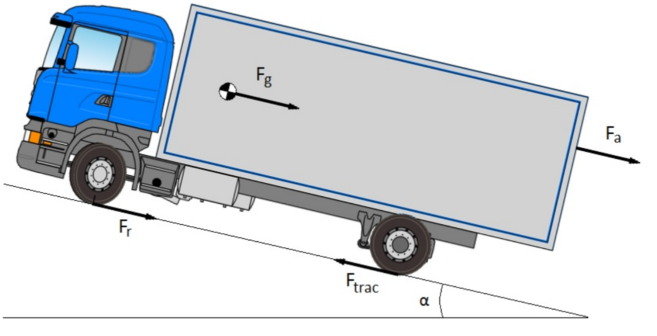

Figure 4.6: *An illustration of a truck and the forces that affects the translational motion. F_{trac} is the tractive force, F_a is the aerodynamic forces, F_r is the rolling resistance force and F_g is the gravitational force acting on the vehicle due to road slope.*

The translational motion of the vehicle is modeled using Newton's second law of motion. Figure 4.6 illustrates the forces acting on the vehicle and the corresponding equation governing the vehicle dynamics becomes

$$m_r \dot{v} = F_{trac} - (F_a + F_r + F_g) \tag{4.11}$$

Where m_r is the vehicle inertia, v is the vehicle speed, F_{trac} is the tractive force from the wheels, F_a is the aerodynamic forces, F_r is the rolling resistance force and F_g is the gravitational force acting on the vehicle due to the road slope.

4.5.1 Vehicle inertia

In the model, the entire powertrain is considered rigid and thus the powertrain component's inertia can be transformed into a equivalent mass which is added to the vehicle mass. The sum of the vehicle mass and the powertrain equivalent mass then becomes the vehicle's total inertia [3] i.e. the resistance to changes in velocity. Taking the gearbox and final drive ratio into consideration, the inertia can be calculated as

$$m_r = m_v + \frac{1}{r_e^2} \cdot J_w + \frac{\gamma^2}{r_e^2} \cdot J_e \tag{4.12}$$

In this equation, m_v is the total vehicle mass, J_w is the wheel inertia, γ is the total gear ratio, J_e is the engine inertia and r_e is the wheel effective rolling radius.

4.5.2 Aerodynamic resistance

The aerodynamic resistance force acting an a vehicle is hard to predict due to the many uncertain factors that influences it, e.g. weather, wind and temperature. If ideal conditions are assumed with calm weather and no wind, the aerodynamic force can be expressed as [4]

$$F_a = \frac{1}{2} \cdot \rho_a \cdot C_D \cdot A_f \cdot v^2 \tag{4.13}$$

where ρ_a is the density of the air, C_D is the air drag coefficient and A_f is the frontal area of the vehicle.

4.5.3 Rolling resistance

The rolling resistance is an energy loss, caused mainly by hysteresis due to deformation of the tires while rolling [4]. For simplicity it is often modeled as a force, acting on the center of the wheel, that counteracts the vehicle's motion. It is common to express the rolling resistance force as a function of the vehicle's mass m_v, road slope α and a tire specific rolling resistance coefficient C_{rr}.

$$F_r = C_{rr} \cdot m_v \cdot g \cdot \cos(\alpha) \tag{4.14}$$

4.5.4 Gravity

The gravitational force contributing to the longitudinal motion of the vehicle is expressed as

$$F_g = m_v \cdot g \cdot \sin(\alpha) \tag{4.15}$$

5

Implementation

In addition to the models used, the accuracy and simulation time are influenced by how the models are implemented into the simulation environment. In this thesis, the simulations are performed using MATLAB.

5.1 Driver models

Two types of driver models are implemented. The first one is a PI-controller that uses a time based reference signal. It is used to follow a predefined speed profile. The other one is a cruise controller (CC) that follows a distance based reference signal.

5.1.1 PI-driver

As shown in section 2.2, a vehicle's fuel consumption depends on how it is driven. However, The behaviour of a driver is complex and not easily described by conventional control theory [19]. An estimation of the fuel consumption during real driving conditions can still be obtained by implementing a PI-controller that regulates the vehicle speed according to a predifined speed profile. Since the simulation models in this thesis is compared to STARS, the speed profiles used for the PI-controller is created using STARS. The implemented controller, equation 5.1, ensures that the vehicle follows the referance speed v_{ref}. Equations 5.2 and 5.3 ensures that negative control signals are interpreted as a braking signals u_b and positive control signals are interpreted as a signal from the gas pedal u_g. This implementation also makes sure that the gas pedal and brake pedal are not used simultaneously.

$$u = K_p(v_{ref} - v) + K_i \int_{t_0}^{t} (v_{ref} - v), \quad u \in [-1, 1] \tag{5.1}$$

$$u_g = \begin{cases} u, & u > 0 \\ 0, & u \leq 0 \end{cases} \tag{5.2}$$

$$u_b = \begin{cases} u, & u \leq 0 \\ 0, & u > 0 \end{cases} \tag{5.3}$$

5.1.2 Cruise controller

A CC that follows a distance based reference signal is implemented as a PI-controller. The CC tries to follow the reference speed by only using the gas pedal, see equation 5.4. When driving downhill, heavy vehicles such as trucks can gain enough speed to exceed the speed limit even if the gas pedal isn't used. To prevent this a new control signal is introduced, v_{limit} which is the upper speed limit that the vehicle is allowed to reach. A separate PI-controller is implemented to control the brake pedal according to equation 5.5.

$$u_g = K_{p,g}(v_{ref} - v) + K_{i,g} \int_{t_0}^{t} (v_{ref} - v), \quad u_g \in [0, 1] \tag{5.4}$$

$$u_b = K_{p,b}(v_{limit} - v) + K_{i,b} \int_{t_0}^{t} (v_{limit} - v), \quad u_b \in [-1, 0] \tag{5.5}$$

Figure 5.1 illustrates the functionality of the CC. When traveling on level road, the gas pedal is used to maintain the reference speed of 85 km/h. When the downhill slope is reached the vehicle starts to accelerate, the gas pedal is then released and the engine goes to fuel cut-off mode. When the vehicle reaches the upper speed limit, the brakes are applied and the acceleration is halted. The gas pedal is engaged again when the vehicle speed has decreased to the referance speed.

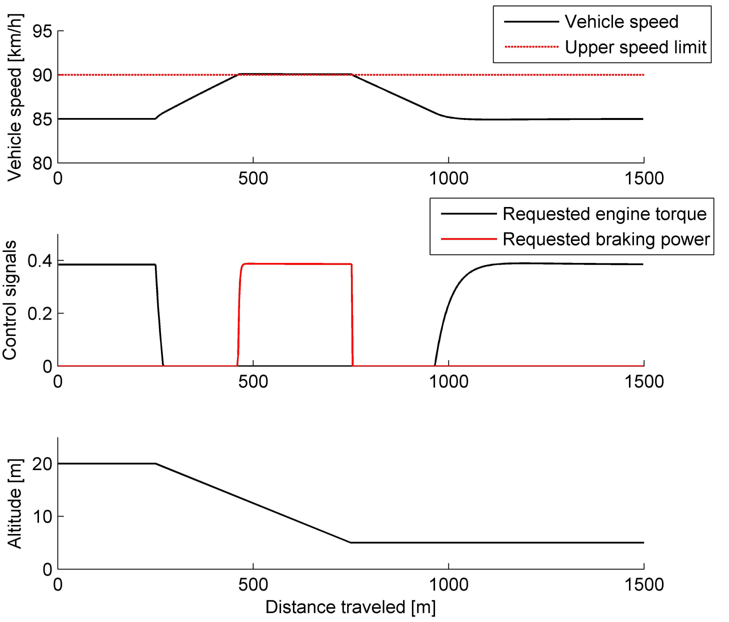

Figure 5.1: *Ilustration of how the cruise controller works. When the vehicle drives downhill and the speed increases, the fueling to the engine stops. Braking only occurs if the upper speed limit is reached.*

5.2 Gear shifting

The gear shifting logic used in this thesis is based on Scania's automatic gear shifting system, OptiCruise (OPC). In this section, the simplification and implementation of the OPC system is described.

Just as in the real OPC, the implemented gear shifting strategy is based on a target engine speed. The target speed is mapped as a function of the gas pedal angle, see figure 5.2. The more torque requested by the driver, the higher engine speed is allowed. In each timestep of the simulation the algorithm calculates the resulting engine speed of a gear shift one, two and three steps up or down. The gear is always changed to the one that gives the engine speed closest to the target speed. To avoid unnecessary gear shifts a damping algorithm is also implemented. It prohibits gear shifting in rapid succession by introducing a minimum time interval between gear shifts. During fast acceleration and deceleration the engine speed may spike, this is avoided by overriding the damping algorithm if the engine speed reaches a predifined limit.

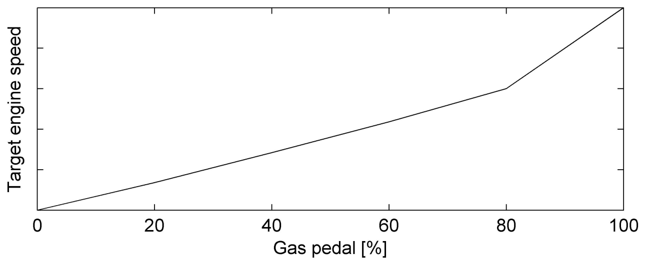

Figure 5.2: *The gear shifting strategy chooses the gear that gives the engine speed closest to the target speed. The target speed is decided by the gas pedal*

One of the larger differences between the implemented strategy and OPC is how power deficit is handled. If the current gear is unable to provide enough tractive force OPC calculates a force equilibrium gear and predicts how to reach it, using the best gear shift sequence. The strategy used in this thesis shifts down when the speed has been reduced due to the power deficit.

OPC is a complex system with several submodels used to predict how to optimize the gear shifting, it considers both drivability and fuel consumption. The simplified strategy implemented in this thesis does however capture the general behaviour of OPC without increasing the simulation time.

To examine the effects of using the simplified gear shifting strategy, several simulations are performed in STARS. The same engine and gearbox is used in all the simulations but the vehicle weight and driving mission are varied. The velocity profile obtained from STARS are then used as input to the simulation tool developed in this thesis. Each combination of weight and driving mission are then simulated twice, once using the OPC gear shifting sequence obtained from STARS and once where the gear shifting is decided by the simplified strategy. The difference in fuel consumption between the simulations are presented in table 5.1. The velocity profile aswell as the roadslope of the different simulation can be found in appendix A

	Vehicle weight [t]			
Drive cycle	10	20	30	40
ACEA coach	+0.9 %	+0.4 %	-0.2 %	+0.5 %
ACEA inter-urban bus	+1.7 %	+0.8 %	+0.0 %	-
ACEA haulage	+0.1 %	+0.1 %	-0.0 %	-0.0 %

Table 5.1: *Table showing the impact on fuel consumption when using the simplified gear shifting strategy.*

The simplified strategy works well in most cases, especially for haulage mission where rapid velocity changes doesn't happen often. For urban and similar driving missions where start and stops are common, the fuel consumption can be affected by over 1.5 percent. This is mostly caused by how the starting gear is chosen and which gears are chosen for the acceleration phase. OPC considers how the vehicle will accelerate after a gear shift and how long it will take to reach the target engine speed. The simplified strategy just tries to reach the target speed with the least amount of gear shifts.

5.3 Simulating start and stop

It is important for the simulations to handle start and stops well, even in steep hills. The physically correct way to model this process is to implement a clutch in the powertrain and create a driver model that utilizes it. However, this would require a more complex driver model and implementation of the clutch dynamics can cause stiffnes in the system [18]. Instead, all external forces acting on the vehicle is disregarded when the vehicle has stopped. Using this simplification, the driver doesn't have to keep pressing the brake pedal when the vehicle is standing still and no smooth transition from braking to accelerating has to be performed by the driver when starting again.

When the vehicle starts to accelerate after a stop, the external forces starts affecting the vehicle again. To avoid chattering in the simulation caused by this discrete event, acceleration is not allowed before the traction force exceeds the static forces that should be acting on the vehicle [20]. This is done by modifying equation 4.11 according to

$$\dot{v} = \begin{cases} 0 & if\ F_{trac} < F_r + F_g, v = 0 \\ \frac{1}{m_r}(F_{trac} - (F_a + F_r + F_g)) & else \end{cases} \tag{5.6}$$

To avoid numerical errors, the integrator that calculates the velocity is reset to zero when the velocity drops close to or below zero.

In addition to modifying the vehicle's longitudinal motion, a clear distinction has to be made between engine idling and engine braking. Since engine braking is only possible when the vehicle is moving, the engine is put in idling mode when the vehicle speed is zero. When the engine is idling, the engine speed is controlled to it's idling speed and the only torque produced is used to power the auxiliary units.

These changes in the simulation models is a robust method that enables simulation of the vehicle dynamics during start and stops without a complex driver model.

5.4 Engine mode changing

A real Scania EURO 6 engine uses several different modes to control the fueling in different situations and the mode changing algorithm is based on more information than just the temperature of the exhaust aftertreatment system. However, to avoid increasing the complexity of the simulations, the same approximation used in VO is implemented. When the temperature of the exhaust gases are above a certain value, the engine map from mode X is used to estimate the fuel consumption. For temperatures below another value, the engine map from mode Y is used. For temperatures in between, a linear interpolation of the values obtained from the two maps is used to estimate the fuel consumption.

Mode X is a favorable mode with low fuel consumption that is used when the amount of NO_x is succesfully reduced in the aftertreatment system. This mode is often used when the engine operates with a high power output, e.g. when driving on a motorway, since the heat from the exhaust gases keeps the temperature in the aftertreatment system high. Mode Y is used to lower the amount of NO_X and causes the engine to consume more fuel.

This simplified mode changing algorithm is devised through empirical testing. It is based on the general mode distribution of different vehicles and gives a good estimation of the total fuel consumed during a drive cycle. The difference in number of modes considered and when they are activated compared to the control alghoritm used in an actual vehicle makes it hard to validate the simplified alghoritm. A known problem with this simplification is that the mode used to warm up a cold aftertreatment system is disregarded. This mode has significantly higher fuel consumption than mode X and Y. This is however not a problem when simulating haulage mission since it is rarely used when driving on a motorway.

5.5 Discretization

Since the number of states in the model are reduced and only slow dynamics are considered, it is interesting to see how large the time step can be while still retaining stability and estimation accuracy. The standard time step length in the simulation is 0.1 second. Simulations are performed with increasing time step and the fuel consumption estimation is compared with the simulation using the standard time step. The relative error caused by the increased time step, expressed as the quotient of the fuel consumption during the simulation with increased time step and the one with the standard time step is presented in figure 5.3. The comparison is performed for two different vehicle weights and two different driving mission, haulage and delivery. The engine used for the simulation has a maximum output power of 490 hp and a maximum torque of 2500 Nm.

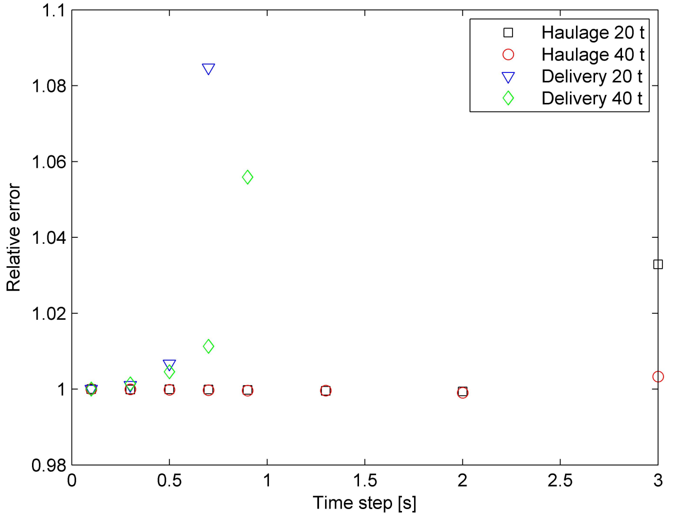

Figure 5.3: *Plot of the relative error in fuel consumption caused by increasing the time step. Simulations are performed with two different vehicle weights and for two different driving missions, haulage and delivery.*

For haulage tasks, the time step can be increased surprisingly much, no large discrepancies are introduced for a time step that is shorter than 2 seconds. For the transient delivery cycle, where fast accelerations and intense braking are common, the system becomes oscillative when the time step is increased. This results in an increased estimate of the fuel consumption.

The impact on the simulation time when increasing the time step is significant. With the standard time step of 0.1 second approximately one hour of driving is simulated in one second. Increasing the time step to 0.3 seconds reduces the simulation time with more than 50 percent. Figure 5.4 shows how many hours of driving is simulated in one second for different time steps [1].

[1] The simulation are performed on a computer with the following specification:
Processor: Intel Core i5-3470 CPU @ 3.20GHz
RAM: 6.00 GB
System: 64-bit operating system

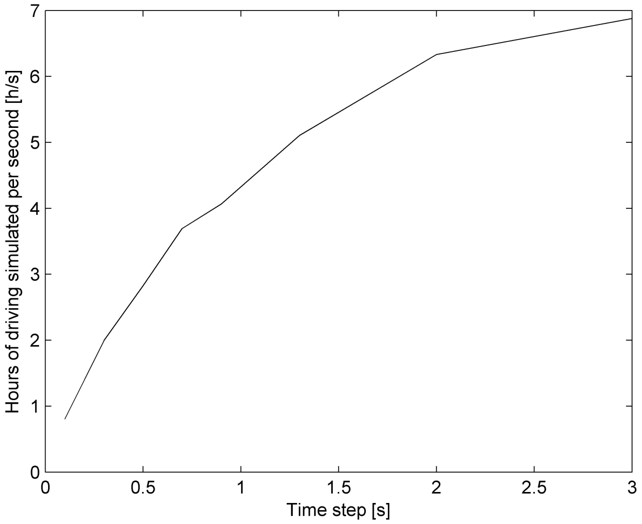

Figure 5.4: *Plot of how many hours of driving that is simulated in one second for different time step. The values are obtained as the average from the simulations described in figure 5.3.*

The length of the time step used in the simulations for the implementation in VO depends on how the models are integrated into VO which is not yet decided. The simulation results presented in this report is obtained using the standard time step of 0.1 second if nothing else is stated.

6

Fuel consumption estimation

In this chapter, the accuracy of the fuel consumption estimation method is validated. In section 6.1 and 6.2 comparisons between the developed method and Scania's simulation tool STARS are presented. A validation against real measurements is also performed and presented in section 6.3.

6.1 Haulage

In order to compare the developed simulation model to STARS, a series of simulations are performed with STARS. A 40 tonne truck driving with the cruise controller, using the reference speed 85 km/h, is simulated using the topology from four different roads. The velocity profile obtained from STARS and the same topology are then used as the inputs to the simulation model developed in this thesis. The result from the fuel consumption estimations are presented in table 6.1.

	Fuel consumption [l/100 km]		
Route	STARS	Estimation	Difference
Södertälje - Norrköping	34.47	34.75	+0.8 %
Koblenz - Trier	41.21	41.16	-0.1 %
Zwolle - Apeldoorn	31.25	31.73	+1.5 %
ACEA haulage	36.46	36.23	-0.6 %

Table 6.1: *Comparison of fuel consumption estimation of the method devised in this thesis and STARS.*

Although the simulation in STARS is considerably slower, the results does not differ that much. For haulage driving missions the difference in estimated fuel consumption is at most 1.5 percent. The main reason for the difference in the results are illustrated in figure 6.1, where the differences in engine torque, speed and fuel consumption over time are plotted. In the figure, the difference is expressed so that a positive value is an overestimate, compared to STARS.

At steady conditions, when the engine speed of the two simulations are identical there is a difference in engine torque which causes the fuel estimate to differ. This is due to the fact that the driving resistances are calculated differently in the two models. Furthermore, the method to estimate fuel consumption is different. The effect of this is clearly visible in figure 6.1 at the time 3280 where both engine speed and torque is overestimated while the fuel consumption is underestimated compared to STARS. Also, at approximately 3126 seconds into the simulation the effect of the engine changing mode in the STARS simulation can be seen.

Although the difference in fuel consumption during steady state operations is clear in the plot, notice that the fuel consumption is expressed in g/min and the difference is smaller than 6 g/min which is approximately equal to 0.43 l/h, or 0.36 l/100 km at 85 km/h. Considering the simulation and fuel consumption estimations are executed in approximately two seconds while the same simulation in STARS takes almost ten minutes, the accuracy is considered good enough.

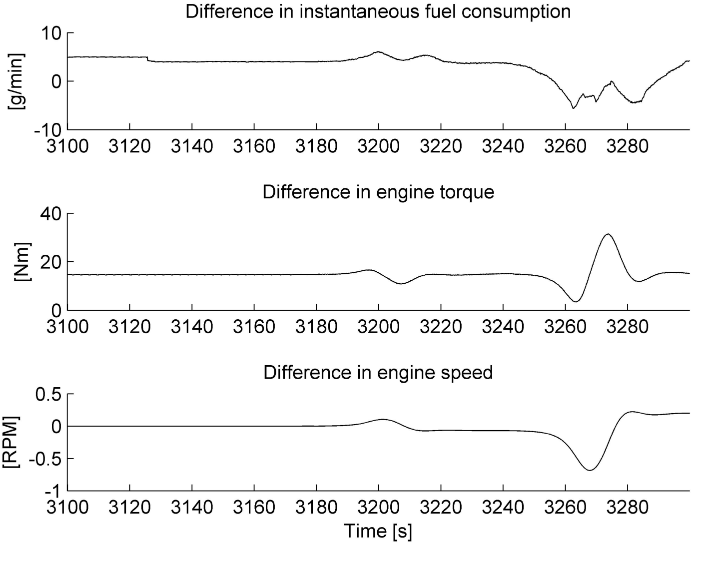

Figure 6.1: *Difference in engine speed, engine torque and fuel consumption compared to STARS. Positive values is an overestimation compared to STARS.*

6.2 Transient cycles

Although VO is mostly used to estimate fuel consumption for haulage driving mission, it is also of interest to investigate the accuracy of the fuel consumption estimations when simulating more transient driving cycles. Therefore, another comparison is done with STARS using the drive cycle illustrated in figure 6.2.

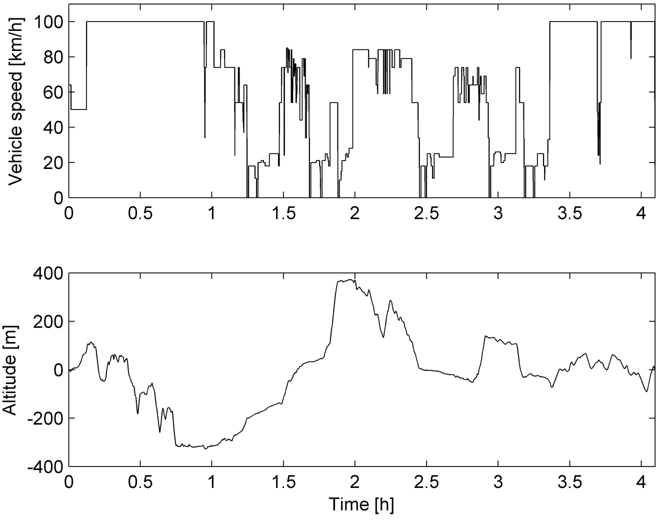

Figure 6.2: *The transient drive cycle used for the comparison.*

Trucks with different vehicle weights are simulated in STARS and the velocity profile obtained is used as the reference speed for the PI-controller. The result of the fuel consumption estimations are presented in table 6.2.

	Fuel consumption [l/100 km]		
Vehicle weight	STARS	Estimation	Difference
40 t	44.46	43.97	-1.1 %
30 t	37.41	36.92	-1.3 %
20 t	30.39	30.08	-1.0 %
10 t	24.03	23.35	-2.8 %

Table 6.2: *Comparison of fuel consumption estimation of the method devised in this thesis and STARS.*

Although the difference between the two simulation models are larger than for haulage missions, a difference lower than three percent is a reasonable trade-off considering the simulations are performed in under one percent of the time required for a STARS simulation. Furthermore, the fuel consumption estimation in STARS has not been validated for transient driving missions, i.e. the estimations are not three percent wrong, they differ from STARS by three percent at most.

6.3 Validation against real measurements

In addition to comparing the simulations against STARS, a validation against measured data is also performed. No measurements are performed during this thesis work. The data used for the validation were recorded for another project and therefore no information about the ambient conditions are available. The measured data were obtained from a truck, driving from Södertälje to Jönköping and back. The data is divided into four parts.

6.3.1 Reference speed

During the simulation, the vehicle speed is controlled by the PI-driver described in section 5.1.1. The measured speed is used as the input for the controller. In order to reduce the measurement noise, a 5th order lowpass digital Butterworth filter is used to smooth the signal. The proportional and integrator constants of the controller are adjusted manually so that a good respons is obtained.

6.3.2 Topography

No recording of the topography or the road slope were performed during the measurement. Instead, the road slope from a different measurement had to be used for the simulations. From the road slope measurements, the slope of the entire route was obtained as a function of distance traveled. During the simulations, the measured position of the vehicle was used find the correlating slope. The topography of the road is presented in figure 6.3.

The tachometer used to measure the traveled distance, and position, is not precise enough for this method. Therefore the positions correlating to the slope data had to be adjusted manually. The position of the slopes was tuned so that the peaks in the simulated engine torque matched the measured engine torque.

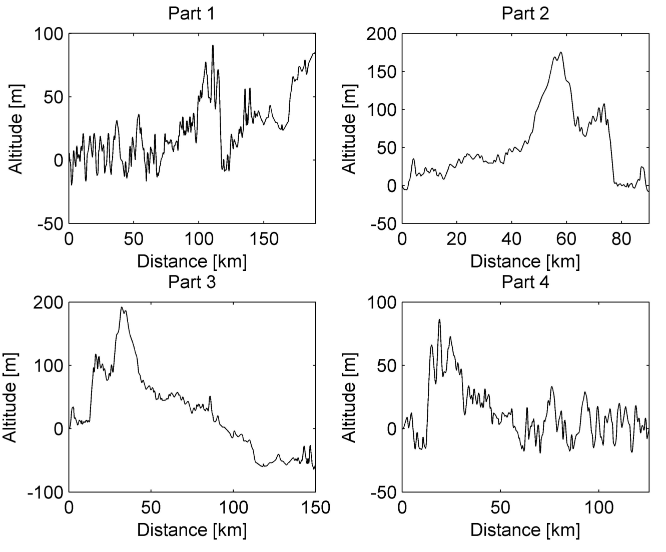

Figure 6.3: *Topography of the road used for the validation.*

6.3.3 Vehicle specification

The measurements were performed, using a truck with the specification presented in table 6.3.

Component	Type
Engine	DC13 125
Gearbox	GRS905R
Final drive	R780
Retarder	3500
Drive wheel	315/70

Table 6.3: *Specification of the truck used for the validation.*

The parameters used in the simulation, that are not obtained from look-up tables, are presented in table 6.4.

Notation	Parameter	Value
C_D	Air drag coefficient	0.607
A_f	Vehicle front area	10.2 m^2
m_v	Vehicle total weight	14 000 kg
r_e	Wheel effective radius	0.492 m
C_{rr}	Rolling resistance coefficient	0.006

Table 6.4: *Parameters used for the simulation.*

Initial simulation with these parameters indicated a discrepancy between the simulated and actual powertrain ratio. When the velocity of the simulated and the actual vehicle was the same, there was a small difference in their engine speed. Since this would effect the fuel consumption, the effective wheel radius was increased with one percent in the validating simulations.

6.3.4 Results and evaluation

In table 6.5 the results of the fuel consumption estimations are compared to the measured values. Although the simulations are fast, the accuracy is high. The largest difference in estimated and measured fuel consumption of the validation is less than three percent.

	Measured	Estimated	Estimation error
Part 1	44.05 l	44.14 l	+0.2 %
Part 2	20.82 l	20.37 l	-2.1 %
Part 3	34.25 l	34.08 l	-0.5 %
Part 4	27.63 l	28.44 l	+2.9 %

Table 6.5: *Measured and estimated fuel consumption.*

In addition to estimating the total fuel consumption with a relative high accuracy, the fueling during the entire simulated route matches the measured fuel consumption. In figure 6.4 the cumulative fuel consumption during the simulation is presented and compared to the measured.

In the plots of part two and four, where the estimation accuracy is lowest, a trend can be seen. The majority of the estimation error is caused in a small part of the simulation. In part two, the estimated value starts to deviate from the measured value in the beginning of the simulation. Thereafter the estimation error is more or less constant for the rest of the simulation. In part four, the estimated fuel consumption is almost identical to the measured up until the end of the simulation where it starts to deviate. There are several plausible explanations to why the estimations are accurate for the most part of the simulations but deviates in small parts.

When fitting the road slope data to the correlating positions for the simulation, the measurement error of the tachometer is assumed to be represented by a constant bias error. If this is an erroneous assumption, some of the slopes can be positioned wrong which leads to the driving resistance being over- or underestimated. Another reason to why the estimation deviates from the measured values is the unknown ambient conditions. For example, the simulation model does not consider how the wind affects the vehicle.

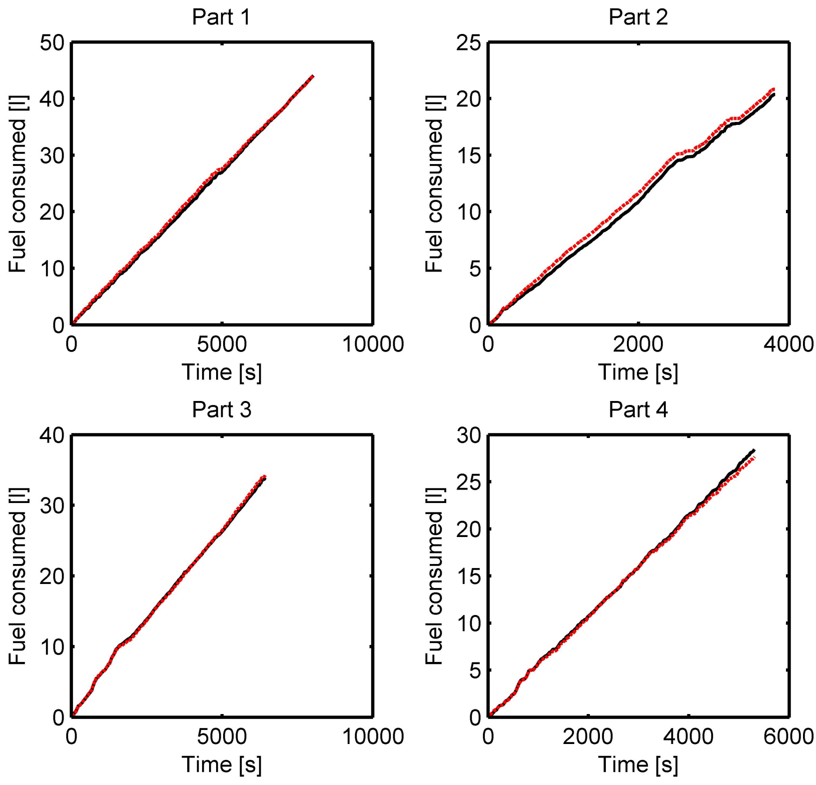

Figure 6.4: *Cumulative fuel consumption of the different parts of the road between Södertälje and Jönköping. The estimated value (solid line) follows the behaviuor of the measured value (dotted line) well.*

However, considering the entire measured route, the estimated fuel consumption only deviates 0.22 percent from the measured. In this case, measured fuel consumption refers to the values obtained from the engine control unit. These values are known to deviate from the actual fuel consumed [6, 5]. At the end of the day when the measurements were performed, the amount of fuel measured by refueling the truck was 2.57 percent lower than the values obtained from the engine control unit.

7

Look-ahead control

An important part of the simulation tool, is the ability to describe the fuel saving capabilites of driver support systems. This chapter describes a look-ahead cruise controller that uses information about topography of the road ahead to minimize the fuel consumption. Several types of look-ahead controllers have been developed and three types are discussed in this thesis.

Logic based control is a control strategy that relies on logical reasoning. In [9] such a controller, where the logical rules are represented by flowcharts, is described. The computational load of this method is low but the rules can be hard to define. In [9], interesting points, e.g. slope starts and ends, are found along the road and simple strategies are formulated to get from one point to the next. The placement of the points are decided using the results of previous simulations and educated guesses.

Model Predictive control (MPC) uses internal models to find the optimal control signals. the performance of the controller is mostly dependant on the quality of the models. A sensitivity analysis of a look-ahead MPC is performed in [21] which shows that a small divergency in the model parameters can create a oscillating velocity profile.

Dynamic programming (DP) is an optimization method that divides a complex problem into a sequence of simpler ones. DP is also dependant on the model but is considered to give a solution very close to the optimal. A drawback is that the computational load of the optimization grows exponentially with the size of the problem.

Creating an optimal look-ahead controller for real world driving is not a trivial problem. This is however, not the goal in this thesis. The purpose is to find a method that emulates Scania's look-ahead controller while having a low computational load.

If only the simulation speed was considered, a logic based controller would be optimal since no optimization has to be performed. Such an heuristic method is however, not very suited in this application. The logical rules determining the controller behaviour would have to be adjusted each time the vehicle specification changed. A method to optimize the control signals based on the vehicle is more appropriate for an application such as VO.

Look-ahead cruise controllers (LACC) such as the Scania Active Prediction uses GPS to predict the topography of the road ahead. By continuously adjusting the vehicle speed accordingly, the fuel consumption is reduced in hilly terrain compared to a regular CC. Simulating the LACC means performing predictions and finding the fuel optimal control continouosly during the simulation. Since this would increase the simulation time considerably, a different method was chosen. Before the simulation starts, a fuel efficient velocity profile for the entire route is found using dynamic programming. The velocity profile is then used as the referance speed to the conventional cruise controller.

7.1 Dynamic programming

The optimization method used to find the optimal speed profile is based on [10] and [8]. A discrete model is used to compute the fuel consumption which is used along with the velocity to construct a cost function. The optimal speed profile is the one that minimizes the accumulated cost to travel the entire route.

7.1.1 DP model

A backwards looking model is used to calculate the fuel consumption in the DP-implementation. Vehicle speed v, acceleration a and road slope α are input signals and the output is fuel consumption. The power at the wheels P_{wheel} is calculated as

$$P_{wheel} = (F_a + F_r + F_g + m_v \cdot a) \cdot v \tag{7.1}$$

where the same notations as in chapter 4 are used. The engine speed and torque output is calculated as

$$\omega_e = \frac{i \cdot v}{r_w} \tag{7.2}$$

$$T_e = \begin{cases} \frac{P_w \cdot \eta}{\omega_e} & T_e < 0 \\ \frac{P_w}{\omega_e \cdot \eta} & T_e > 0 \end{cases} \tag{7.3}$$

where η is the powertrain efficiency, which is assumed constant during freeway driving, and i is the total powertrain gear ratio. The fuel consumption is estimated using the method described in section 4.1.1.

7.1.2 Discretization and cost function

The variable to be optimized is the vehicle speed over the distance. Therefore the DP-model is discretized and made distance based by dividing the distance into steps with a constant length h. During each step, the velocity and slope is assumed constant. By denoting the step number with k and the total number of steps N, the speed and acceleration is obtained through

$$x_k = k \cdot h \tag{7.4}$$

$$v = \frac{v_k + v_{k+1}}{2} \tag{7.5}$$

$$t = \frac{x_{k+1} - x_k}{v} \tag{7.6}$$

$$a = \frac{v_{k+1} - v_k}{t} \tag{7.7}$$

The road slope α is set to the mean value over the interval $[\alpha_k, \alpha_{k+1}]$.

Reducing the fuel consumption is easily accomplished by lowering the average speed. The goal is to reduce fuel consumption with a very small, or no reduction in trip time. Therefore the cost function to be minimized in the optimization has to weigh the velocity against fuel if the controller is to perform as desired.

The cost to travel the total distance can be expressed as

$$J = \sum_{k=0}^{N-1} \zeta_k(x_k, x_{k+1}, v_k, v_{k+1}, \alpha_k, \alpha_{k+1}) \tag{7.8}$$

where the cost of each step is

$$\zeta_k = \dot{m}_{fuel} + \beta \cdot v_{low} + \gamma \cdot v_{high} \tag{7.9}$$

v_{low} and v_{high} are deviation under, respectively over a set referance speed. β

and γ are weighing parameters that decides how much the different factors are penalized.

The constraint on the optimization is given by the discretization of the speed which is allowed to vary between v_{min} and v_{max} with the discrete step δ. If no feasible solution to the problem is found within the constraints the algorithm returns v_{min} as the optimal speed for the current step.

It is not feasible to assume that the vehicle will be able to drive the entire route with a single gear ratio, neither is it suitable to include the gear shifting as a part of the optimization. The additional state in the state-space grid would increase the simulation time to much even if the grid were to be reduced by e.g preprocessing. Instead, the gear used when calculating the cost of each step is expressed as a function of the vehicle speed. The selected gear is the one that results in the engine speed that is closest to a prefered engine speed at cruising speed.

7.1.3 Optimization

If S_k is defined as the possible set of states in step k and $\zeta_k^{i,j}$ is the cost to travel from $x^i \in S_k$ to $x^j \in S_{k+1}$ [8], The DP-algorithm can be described as

1. Assign a final cost $J_N(x_N)$.
2. Let $k = N - 1$.
3. Let $J_k(x^i) = \min\limits_{x^j \in S_{k+1}} \left\{ \zeta_k^{i,j} + J_{k+1}(x^j) \right\}$, $x^i \in S_k$.
4. Repeat 3. for $k = N - 2, N - 3, ..., 0$.
5. The optimal speed is obtained by following the optimal path from the initial state.

7.2 Simulation results

To validate the behaviour of the LACC, comperative simulations are performed in STARS. A 40 ton truck traveling between Södertälje and Norrköping is simulated in both applications and compared. The discretization grid used in the DP-optimization is presented in table 7.1.

Notation	Parameter	Value
h	Distance step	50 m
δ	Velocity step	0.2 km/h
v_{ref}	Reference velocity	85 km/h
v_{min}	Minimum velocity	79 km/h
v_{max}	Maximum velocity	89 km/h

Table 7.1: *State space discretization for the DP-algorithm. The parameter values are chosen to give give a controller behaviuor similar to STARS*

The velocity profiles on part of the simulated road are presented in figure 7.1. The simplified LACC used in this thesis shows the same behaviuor as the one in STARS, accelerating before steep uphills and decelerating before downhills where the maximum velocity would be exceeded without use of the brakes.

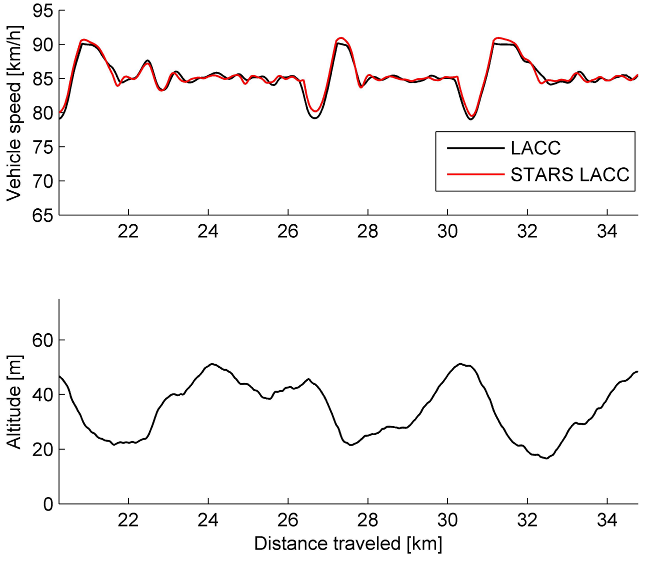

Figure 7.1: *Comparison of the look-ahead cruise controller (LACC) constructed in this thesis and the one implemented in STARS. The results are obtained from simulations of the road between Södertälje and Norrköping.*

To validate the functionality of the LACC and to ensure that it actually improves the fuel economy, several simulations are performed with both the LACC and the conventional CC. The result of the fuel consumption estimation is presented in table 7.2. In the simulations the upper speed limit is always set 4 km/h above the reference speed and the lower speed limit is set 6 km/h below. The brakes are applied when the speed reaches 5 km/h over the reference speed.

	Fuel consumption [l/100 km]			Trip time [s]		
v_{ref}	CC	LACC	Differance	CC	LACC	Differance
85 km/h	35.02	33.95	-3.06 %	5028	5043	+0.3 %
75 km/h	32.52	31.27	-3.84 %	5665	5699	+0.6 %
55 km/h	28.96	28.12	-2.90 %	7664	7710	+0.6 %

Table 7.2: *A comparison of fuel consumption when using the conventional CC and the LACC. The values are obtained from simulation of the road between Södertälje and Norrköping*

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7.3 Conclusion and discussion

A look-ahead cruise controller (LACC) has been constructed and tested. The simplified model used for the optimization algorithm results in a short simulation time. With the discretization used, the optimization of the reference speed takes approximately two seconds per ten kilometers. The optimization speed can be increased further by using a polynomial, instead of an engine map, to estimate the fuel consumption. Although the interpolation method described in section 4.1.1 is relative fast, using a polynomial approximation, e.g. a Willans line, reduces the time needed for an optimization by nearly 50 percent. However, since the engine maps for each engine are already implemented in VO, these will probably be used when this LACC is initially implemented in VO.

The cost function aswell as the discretization parameters were chosen so that the controller performed similiar to the LACC implemented in STARS. No validation against a real controller were performed. Depending on the inaccuracy of STARS' controller, the cost function may have to be modified in order to accurately depict the fuel saving capabilities of Scanias LACC, Scania Active Prediction.

In similar implementations where DP is used to find the optimal speed, the cost function is based on trip time instead of deviation from the reference speed [8, 10, 9]. Since the objective of a LACC is to minimize fuel consumption and trip time, a cost function based on these factors should solution closer to the optimum. However, as previously mentioned, constructing the best LACC is not the goal in this thesis.

Simulations where the trip time was penalized instead of the speed were performed. Although these simulation indicated an improvement in fuel saving capability, the controller behaviour was not coincidal with the expected behaviour. Since the optimization algorithm considers the entire route, the speed is affected by the topography miles ahead. For example, fuel can be saved by lowering the cruising speed on the start of the route if it is benificial to raise the speed in another part of the route. Due to the computational complexity of the optimization, an actual LACC has a limited prediction horizon and can not save fuel this way.

When penalizing deviations in speed instead, the LACC acts as a conventional CC when the inclination of the road is small. The speed only deviates from the referance speed in the actual slopes where fuel can be saved. This is more consistant with both the LACC implemented in STARS and with Scania Active Prediction.

8
Summary and conclusions

A simulation tool for fuel consumption estimation is constructed and validated. The vehicle model is simplified in order to achieve a low simulation time, yet a good accuracy is obtained. Using a time step of 0.1 second, an hour of driving is simulated in approximately one second. The validation shows that the accuracy of the estimations are within three percent for a haulage driving task.

In order to obtain the low simulation time, some vehicle control systems, such as the automatic gear shifting strategy, are simplified in order to reduce their computational complexity. The effects of using the simplified gear shifting strategy has been studied and the influence on fuel consumption is less than 0.1 percent for haulage driving. The model of the mode changing algorithm has not been validated in this thesis. However, the relatively good estimate of the fuel consumption suggests that it works well.

In the comparison with STARS, some discrepencies is observed. During static driving conditions, the engine torque is estimated as slightly higher than in STARS which gives a higher estimate of the fuel consumption. However, due to the difference in how the fueling is calculated, a lower estimate of fuel consumption is obtained during transient engine operation. These effect are rather small and during haulage tasks, the fuel estimate does not deviate more than 1.5 percent from STARS' estimate.

The largest difference from how the fuel consumption is currently estimated in VO is that the vehicle's longitudinal motion is simulated during the entire driving task. This enables implementation and simulation of fuel saving technology. A general look-ahead cruise controller is therefore implemented. It uses dynamic programming to find a fuel efficient speed profile and the behaviour of the controller has been shown consistant with the one implemented in STARS.

Even though gear shifting is not treated as a part of the optimization problem, fuel is saved even if the reference speed is varied. Furthermore, by returning the lowest allowed speed as the optimal speed when no feasable solution is found, the state space grid can be kept small, even if the vehicle does not have the performance to always drive within the specified speed range.

8.1 Future work

The most important work to be done is deciding how the simulation model will be implemented in VO. The current input parameters are not enough to perform a dynamic simulation. Either a speed profile will have to be an input along with the topography, or the speed limit of the route. The latter would require a more advanced driver model that lowers the vehicle speed before the speed limit is reduced. Either way, some information about the speed has to be an input to the simulation.

A more extensive model of the exhaust aftertreatment system would be beneficial. If more signals from the SCR-model were available, a more realistic mode changing algorithm could be used. An algorithm based on the engine torque and the temperature in the SCR-catalyst was developed, but no suitable method to model the heat transfer in the muffler was found. Without introducing several additional states in the model, no accurate way of depicting the catalyst temperature was achieved.

In section 5.5 the simulation model was proved stable and accurate during haulage driving tasks, even for longer time steps. If the time step were made variable, the simulation speed could be increased considerably by increasing the time step during static driving conditions. However, a different approach than the one discussed in section 3.4 has to be made, since it caused the simulations to become unstable in combination with a PID controller.

Appendix

A
Drive cycles

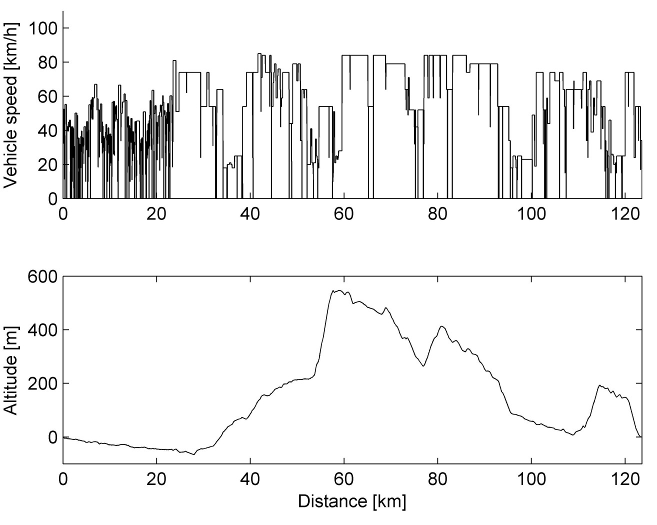

Figure A.1: *The ACEA inter-urban bus cycle.*

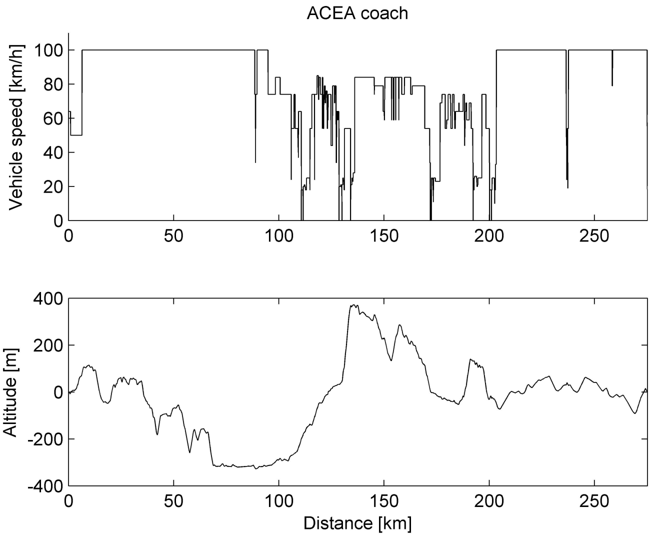

Figure A.2: *The ACEA coach cycle.*

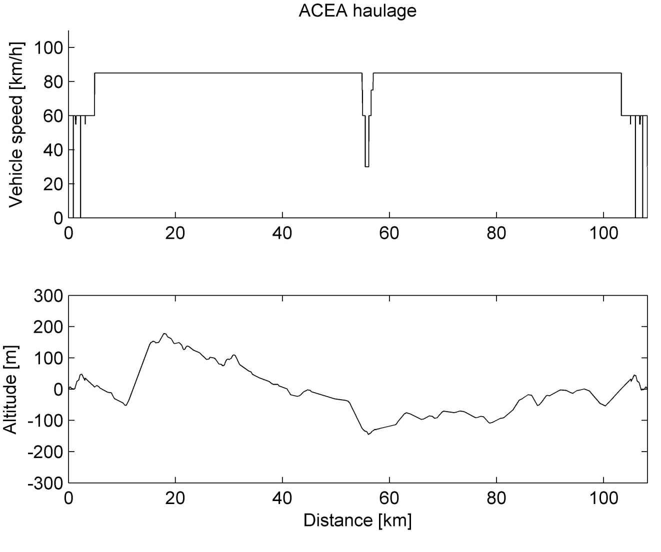

Figure A.3: *The ACEA haulage cycle.*

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