Validation of Bus Specific Powertrain Components in STARS

Master's thesis
performed in **Vehicular Systems**

by
Karl Karlsson

Reg nr: LiTH-ISY-EX -- 07/4045 -- SE

December 19, 2007

Validation of Bus Specific Powertrain Components in STARS

Master's thesis

performed in **Vehicular Systems**,
Dept. of Electrical Engineering
at **Linköpings universitet**

by **Karl Karlsson**

Reg nr: LiTH-ISY-EX -- 07/4045 -- SE

Supervisor: **Magnus Neuman**
RBNP - Powertrain Performance,
Bus Chassis Development
Scania CV AB
Erik Hellström
Vehicular Systems, ISY
Linköpings Universitet

Examiner: **Jan Åslund**
Vehicular Systems, ISY
Linköpings Universitet

Södertälje, December 19, 2007

Avdelning, Institution
Division, Department

Vehicular Systems,
Dept. of Electrical Engineering
581 83 Linköping

Datum
Date

December 19, 2007

Språk
Language

☐ Svenska/Swedish

☒ Engelska/English

☐ ____________

Rapporttyp
Report category

☐ Licentiatavhandling

☒ Examensarbete

☐ C-uppsats

☐ D-uppsats

☐ Övrig rapport

☐ ____________

URL för elektronisk version

http://www.vehicular.isy.liu.se
http://www.ep.liu.se/exjobb/isy/07/4045/

ISBN
—

ISRN
LiTH-ISY-EX--07/4045--SE

Serietitel och serienummer
Title of series, numbering

ISSN
—

Titel Validering av Busspecifika Drivlinekomponenter i STARS

Title Validation of Bus Specific Powertrain Components in STARS

Författare
Author

Karl Karlsson

Sammanfattning
Abstract

The possibilities to simulate fuel consumption and optimize a vehicle's powertrain to fit to the customer's needs are great strengths in the competitive bus industry where fuel consumption is one of the main sales arguments. In this master's thesis, bus specific powertrain component models, used to simulate and predict fuel consumption, are validated using measured data collected from buses.

Additionally, a sensitivity analysis is made where it is investigated how errors in the powertrain parameters affect fuel consumption. After model improvements it is concluded that the library components can be used to predict fuel consumption well.

During the work, possible model uncertainties which affect fuel consumption are identified. Hence, this study may serve as foundation for further investigation of these uncertainties.

Nyckelord
Keywords

Bus Model, Fuel Consumption Simulation, Powertrain Analysis, Model Validation

Abstract

The possibilities to simulate fuel consumption and optimize a vehicle's powertrain to fit to the customer's needs are great strengths in the competitive bus industry where fuel consumption is one of the main sales arguments. In this master's thesis, bus specific powertrain component models, used to simulate and predict fuel consumption, are validated using measured data collected from buses.

Additionally, a sensitivity analysis is made where it is investigated how errors in the powertrain parameters affect fuel consumption. After model improvements it is concluded that the library components can be used to predict fuel consumption well.

During the work, possible model uncertainties which affect fuel consumption are identified. Hence, this study may serve as foundation for further investigation of these uncertainties.

Keywords: Bus Model, Fuel Consumption Simulation, Powertrain Analysis, Model Validation

Preface

This work completes my studies for a Master of Science degree in Applied Physics and Electrical Engineering. I have enjoyed working with this project and I have found it very interesting to be able to do my Master's Thesis at a great company like Scania. The work did not get as theoretical as I had expected from a thesis work but instead I have had the possibility to use my more general engineering skills.

Thesis outline

- **Chapter 1** describes the background of this work and the methods used.
- **Chapter 2** gives an explanation of the existing model as it was implemented before the validation started. Additionally, a sensitivity analysis is made by linearizing the model at steady state operating points.
- **Chapter 3** shows the results of the measurements that have been performed.
- **Chapter 4** presents new models based on data from manufacturers and the test results obtained in the previous chapter.
- **Chapter 5** handles SORT specific models which have been developed.
- **Chapter 6** describes the model's possibility to predict fuel consumption. Data collected at an independent test center at Idiada in Spain is compared to simulation results.
- **Chapter 7** gives suggestions to other things of interest that can be studied as an extension of this work.
- **Chapter 8** summarizes the work and the obtained results.

Acknowledgments

First of all I would like to thank my supervisor Magnus Neuman at Scania for his huge support during this work. He has been a great supervisor and co-worker and has always helped me though my questions some times have been simple and obvious for him. He has also shown great patience in my work of getting to know the organization at Scania.

Furthermore, I am very thankful for all the support I have got from the other members of the RBVS group who have made my first impression of Scania to a very nice one! I would also like to express my gratitude to my prior superior Marianne Karlsson who gave me the opportunity to do this work and to entrust me with the task to continue working at Scania's Bus Chassis Development.

My co-supervisor Erik Hellström and examiner Jan Åslund are greatfully acknowledged for their inputs during the work and for proofreading this material.

Further acknowledgments go to my family and my friends for always being there for me, You all know who You are and what You mean to me. Finally but not least I would like to express my love and appreciation to Anna.

Contents

Chapter 1

Introduction

1.1 Background

This Master's thesis is performed in a collaboration with Scania CV AB in Södertälje and the Division of Vehicular Systems at the Department of Electrical Engineering at Linköpings Universitet.

Modeling and simulation have for a long time been of interest in the vehicular industry as an aid in understanding, controlling and optimizing the behavior of vehicular systems. The studied or modeled system can be an engine, some kind of auxiliary device, a drive line component [6] or a whole vehicle model [8, 19].

Scania's final customers, road carriers and bus companies, use their trucks and buses for many hours per day which make fuel costs one of their main outlays. Measures taken to improve fuel economy, will directly be seen as positive measures at these customer's profit and loss accounts. This makes fuel economy an important sales argument in the competitive industry. At Scania, much work has been done to optimize engine and powertrain efficiencies. Studying for example the Vehicular Systems homepage [18], master's theses which directly or indirectly lead to fuel savings have been carried out in collaboration with Scania. Doctoral students have also studied optimizations concerning the subject area, e.g. [9, 14].

When the optimizations on component level are implemented, the next step will be to optimize the whole vehicle setup and use the right type of component for each purpose. There are many ways of combining components to build a vehicle and only one setup can be optimal if the usage of the vehicle is perfectly described. Therefore, it has to be investigated which components to use when there are alternatives. In the end, the final setup is often a result of the analysis of predicted fuel consumption and driveability. Before computational aid was actual, testing and experience were the main ways of getting well working powertrain setups. Since testing is expensive, much can be won if some of the testing can be replaced by simulations. At least, simulations can show if there exists powertrain compositions that directly can be excluded.

As an aid in this work, a work with developing a model library for simulation of long haulage fuel consumption and emissions of trucks was carried out at Scania in 1999-2001 [14]. The work was done as a licentiate work and resulted in a fuel prediction better than 2% when simulation results was compared with real tests. The library has previously been complemented with bus-specific models where a wide range of gearboxes, engines and chassis have been modeled, including automatic gearboxes, which make it possible to simulate almost all setups of buses that can be produced by Scania and its contractors. More bus specific modifications have also been performed, such as a more exact transient behavior and ability to start and stop. One strength of the library is that it is built up by modules which makes it easy to combine different sub-models and build new models to keep the library up to date.

The simulation tool can for example be used to study how the use of different gear shifting programs, rear axle ratios or engines affects the fuel consumption for a specific customer's duty cycle. Further, it can be used to investigate if it would be interesting for Scania to introduce new features or concepts in the drive line, the steering of the engine or its auxiliary components.

A standard for fuel consumption measurements is the SORT [1] standard [17]. UITP, a world wide association of passenger transport operators, has defined a standard for how to measure fuel consumption in urban area bus traffic. When customers inquire for consumption data, they commonly refer to consumption on SORT-cycles. Therefore, a part of this master's thesis handles SORT specific models. Another drive cycle commonly referred to is the Braunschweig cycle [10].

The main tasks of this master's thesis are to validate the model library and improve parts that has not yet been correctly modeled, focusing mainly on city buses with automatic gearboxes. The main problems in the validation process are that tests only can be performed on buses, not on parts like the gearbox or the differential gear etc. This gives uncertainties in the validation because it is not obvious that all faults are found and it is not always clear from which component a fault originates. This will be further explained in the validation part.

1.2 Validation Method

Measurements are carried out to collect data that can be compared with simulation data in the validation. Both data specific for this is used as well as test results performed by an independent test center at Idiada in Spain where

[1] Standardized On Road Test cycles.

fuel consumption was measured for SORT 1 and 2 cycles. The measurements specific for this thesis are steady state speed measurements, acceleration tests and coast down measurements.

At the original work developing the STARS library for trucks described in [14], the simulated fuel consumption could be predicted and differed finally less than 2% from the measured values. This was seen as a very good result and the deviations were said to depend on conditions like road resistance, uncertainies and approximations in the measurements of the road slope and non modeled transient behavior. During this work of validating the new bus specific components, a similar method to the one used in [14] is employed.

The method described in [14] to validate losses in air- and rolling resistance was to insert a strain gauge on the propeller shaft. This is because of different reasons not within the scope of this work. Instead, the torque calculated by the Engine Management System (EMS) is used. This results in other difficulties since both the gearbox's efficiency and the traveling forces are unknown. By measuring the propeller shaft torque, the total sum of the traveling forces can be identified. When the (calculated) engine torque is used, other effects can influence the result. The real engine torque depends on the energy contents of the fuel which can differ from time to time and by using the second method, the gearbox's efficiency is also unknown etc.

Anyway, in addition to this, a sensitivity analysis is made where it is investigated which faults in variables or components that results in the greatest influence of the fuel consumption. This analysis is then used as an aid when analyzing what to keep an eye on in the validation. The analysis can also directly be used to by hand calculate how the change of a parameter affects fuel consumption, e.g. how many percent fuel that can be saved by decreasing rolling resistance with 10%.

One paper that describes another general method of interest is found in [11] where validation metrics are introduced. The metrics are used to describe the agreement between computational results and experiments. The method takes for granted that the input to the system is the same for a simulation as for a real test. This can be be hard to achieve in this particular case where a real driver operates the vehicle in a way that is very hard to model accurately. Further, small deviations between the model and the real bus in e.g. gear change points make the output differ even though the two systems are driven equally. In the article other types of systems have been validated and they are driven at steady state conditions. Computational results are then compared with the measured results and confidence intervals are used to describe the model's agreement with reality.

Chapter 2

Model Description and Sensitivity Analysis

This is a brief description of the model as it was implemented before the validation started.

2.1 Modeling Language

The model library has been implemented in Modelica using Dymola, a commonly used tool for building simulation models. Modelica has the advantage that it is independent of the computational causality, easily lets the user build new models and after the compilation an executable file is created which reads parameters from a file, enabling the use of a user friendly environment. The non causal modeling means that in Modelica, the variables can be implicitly written in the equations. It is not needed to specify which of the variables that are knowns and unknowns. This leads to a system description that has exactly as many equations as unknowns. In some cases for example efficiencies, the unknown variable is interpolated via a lookup table.

2.2 Model Library Components

The main method in the development of the library has been to use data from manufacturers and experts at Scania and implement this information in the equations describing the systems. Most of the equations are written on state space form with linear and nonlinear equations.

Below, the different library components are described. The components have to be combined to a whole bus before a simulation can be made. Each block in the graphical interface represents a real physical part. Since the modeling have been made on component level, it is easy to change a model and compare different setups as long as the interface between the components remains the same.

2.2.1 Ambient

In the ambient model, represented by Figure 2.1, road data can be loaded from a file. It contains information about demanded speed and the slope of the road. The demanded speed is time or distance dependent and the slope of the road is distance dependent. Constant ambient parameters like air temperature and gravity can also be set.

Figure 2.1: *The ambient model where ambient parameters are set eg. temperature, road profile, speed profile etc.*

2.2.2 Bus

The type of bus that should be simulated needs to be set in this model block. Parameters like the number of wheels and axles, front area, mass and the air drag coefficient, C_D. The wheel's inertias affect the acceleration of the bus and the load is split on the different axles. It also has to be specified if the bus is a normal bus or an articulated city bus, which affects the axle load distribution.

This is the place in the model where the sum of forces equals acceleration according to Newtons 2:nd law. In the model, the bus only moves in the longitudinal direction affected by rolling resistance, tractive force and air resistance. See Figure 2.2.

2.2.3 Engine and Engine Management System

From the CAN-bus model (which is an internal communication system used in the vehicular industry) described below, the Engine Management System (EMS) reads the accelerator pedal position. The implementation of how the injected amount of fuel is calculated resembles the production code. The engine model then uses engine speed and fueling to lookup the torque out of the engine using maps from steady state test cell tests. Those maps include the functionality of the turbocharger/turbine unit or the turbocompound unit

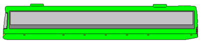

Figure 2.2: *The bus model where Newton's 2:nd law is applied to calculate the acceleration of the bus based on the forces acting on it. The type of bus to simulate has to be specified here. Other parameters are frontal area, air drag coefficient, number of axles, wheels and mass*

at steady state. To achieve good transient behavior, the boost pressure is modeled and the maximum amount of fuel is controlled dependently of the boost pressure.

2.2.4 Gearbox

The model handles different types of gearboxes and shifting programs. There are two different types of gearboxes modeled, automatic and manual ones. In the model, an automatic gearbox is a gearbox with a hydraulic torque converter and a hydraulic transmission which enables positive tractive force during gear shifts. The manual gearbox on the other hand is a traditional manual gearbox where the tractive force is interrupted during the gearshifts as the driver presses the clutch pedal. Scania has developed a gearbox steering system which can be used to help the driver which make this gearbox work like an automatic gearbox but has the hardware of a manual gearbox. This system is called opticruise and contains gear shifting logic and pneumatic actuators which handles the gear shifting.

The gearbox's efficiency is a function of engine speed, used gear and engine torque. Outputs of the gearbox are propeller shaft speed and torque. If an automatic gearbox is used, it is equipped with a hydraulic torque converter which is used at low gears when the bus starts from stand still. This prevents gearbox damage for vehicles used in traffic with frequent stops. At the stops, an additional clutch (NBS) is used to disengage the powertrain to save fuel. The retarder is a hydraulic brake which can be used to save the brakes in long slopes where the service brakes are at risk of getting overheated. The gearbox model can be seen in Figure 2.3.

The gear shifting logic has been implemented after instructions from manufacturers and depends on engine speed, accelerator pedal position and vehicle acceleration. As in a real bus, the steering unit sends a signal to the engine to reduce the torque at gearshifts. There are a number of different gear shifting programs which are optimized for fuel economy or drive ability. The main

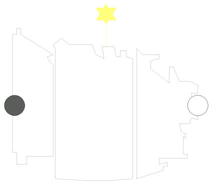

Figure 2.3: *The gearbox model. Contains gearbox steering unit, inertias and losses.*

point is to drive at low engine speeds (high gears) to get a low fuel consumption and to drive at lower gears to achieve good driveability (gradeability, acceleration etc.).

2.2.5 Axle and Wheel

The power out of the gearbox is reduced through losses in the differential gear which has been modeled as a torque and speed dependent efficiency. From the resulting torque, the braking torque and a normal force dependent bearing loss torque is subtracted. On the non driven wheels, only the brake and bearing loss torques act. The resulting tractive force is calculated by dividing the resulting torque with the dynamic rolling radius. The axle and wheel model can be seen in Figure 2.4.

The rolling resistance is proportional to the the normal force and nonlinearly dependent of the vehicle speed.

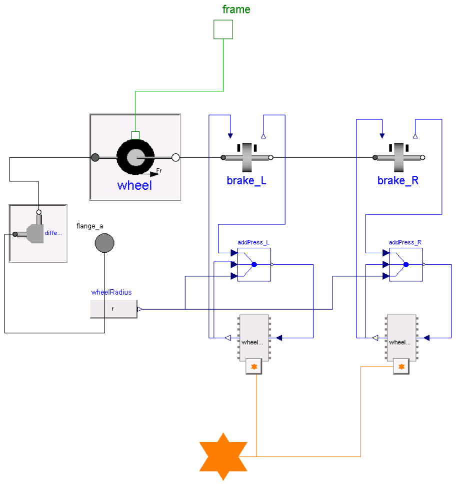

Figure 2.4: *The model of the wheels, brakes and differential gear.*

2.2.6 Auxiliary Devices

Auxiliary devices are: generator, air compressor, engine cooling fan and hydraulic steering pump. Simulations have shown that the auxiliary components consume approximately 5% of the fuel for a long haulage truck [9]. City buses are driven at lower speed and stops more frequently, this figyre can therefore be much greater for a city bus. The torque load on the engine is interpolated via a lookup table or calculated using: $\tau_{aux} = \frac{P_{aux}}{\omega_{aux}\eta_{aux}}$ where aux represents the actual auxiliary device. The auxiliary also load the engine dependently of how they are controlled as in a real bus. An example of this is the air compressor which loads the engine more if the brake pedal is pressed down.

The **engine cooling fan** is driven by a hydraulic pump which loads the engine. The engine temperature is in the model a function of the engine power and the fan speed is controlled dependently of the engine temperature. The torque load is a lookup function of fan speed.

In the model of the **hydraulic steering pump**, the torque that loads the engine depends on the engine speed through a lookup table.

The power out of the **generator** is set to a constant value which results in a constant power loss. The generator efficiency is modeled as constant which results in an engine speed dependent torque load. It is hard to model the real electrical power consumption and the generator efficiency. The interest of modeling the generator more accurately is small since deviations in the model compared to a real generator will only result in small effects on fuel consumption.

The air pressure steering unit (APS) uses dynamic minimum and maximum pressure limits to control the **air compressor**. An example is during braking when the pressure limits are increased which in the ideal case results in a compressor that only loads the engine during braking and therefore consumes no fuel. If the actual system pressure is above the upper limit, the compressor is deactivated, if the pressure is below the lower limit, the compressor is activated. Air consumption occurs at kneeling, door opening, regeneration of the air filter and braking.

2.2.7 Driver

When an automatic gearbox is used in the simulations, the driver model contains two PI-controllers with speed and accelerator dependent constants. Those controllers affect the accelerator and brake pedals. When a manual gearbox is simulated, the production code from Scania’s opticruise has been used for gear shifting logics and the cruise control is used for fuel injection.

2.2.8 Coordinator (CAN-bus)

To let the different models communicate with each other, a coordinator has been implemented where the different steering units can read and write data. Examples of usage are during gear shifting when the gearbox requests a subtractive torque from the engine, or the EMS reading information about accelerator and brake pedal position.

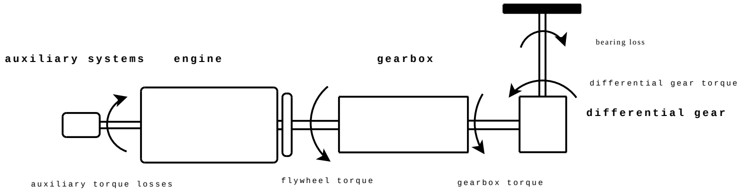

Figure 2.5: *The powertrain model.*

2.3 Steady State Sensitivity Analysis of the Model

To study the influence of how errors in the model parameters affect fuel consumption, an approximation of the forces and torques that act on the bus at steady state speeds is made. This study gives a clue to what parameters are of greatest interest in the validation.

The analysis is made with the existing model described above, with parameters taken from suppliers and handbooks. To really disentangle how an error affects the result, the model and the parameters has to be correct. Since the library has yet not been validated, this could be a big problem but the library is sometimes already used for simulations and gives reasonable results. Therefore the main result of the analysis is probably correct.

Since a city bus is rarely driven at steady state speeds for a long time it can be discussed how well this analysis can be applied on city bus traffic. However, simulations have shown that the simulation tool gives reasonable results also when SORT cycles are driven. Linearization is a common way of analyzing systems and it is a more theoretical approach than just simulating.

Further, the analysis can be used to calculate how much fuel that can be saved if for example the frontal area or total mass is decreased with 5%. For a list describing the variables and abbrevations used, see page 55.

2.3.1 Tractive Force

The tractive force is the force the wheels apply to the ground. At steady state speeds, on an even road, when the engine torque is positive and the engine drives the vehicle forward, the equations describing the tractive force are:

$$F_{tr} = \frac{M_w}{r} \tag{2.1}$$

$$M_w = M_{dg} - \tau_{bl} \tag{2.2}$$

$$M_{dg} = \eta_{dg} \cdot M_{gb} \cdot i_{dg} \tag{2.3}$$

$$M_{gb} = \eta_{gb} \cdot \eta_{TC} \cdot M_{flywheel} \cdot i_{gb} \cdot i_{TC} \tag{2.4}$$

$$M_{flywheel} = M_e - \tau_{steer} - \tau_{gen} - \tau_{fan} - \tau_{comp} \tag{2.5}$$

$$F_{tr} = \frac{\eta_{dg} \cdot \eta_{gb} \cdot \eta_{TC} \cdot i_{dg} \cdot i_{gb} \cdot i_{TC} \left(M_e - \tau_{aux}\right) - \tau_{bl}}{r_w} \tag{2.6}$$

Where τ_{aux} is the sum of the torques from the auxiliary devices. M_{dg}, $M_{flywheel}$, τ_{aux} and M_{gb} are shown in Figure 2.5.

2.3.2 Traveling Forces

The travel resistances are air- and rolling resistance:

$$F_{air} = \rho\, C_D\, A\, \frac{v_{m/s}^2}{2} \tag{2.7}$$

$$F_r = \frac{C_{rr}}{1000}\, N \tag{2.8}$$

$$\begin{aligned} C_{rr} &= C_{rr_{iso}} + C_a(v_{km/h}^2 - 80^2) + \ldots \\ &\ldots\ C_b(v_{km/h} - 80) \end{aligned} \tag{2.9}$$

Were $C_{rr_{iso}}$ is the value the manufacturer Michelin suggests. It is obtained by doing steady state tests of the tire at 80 km/h, described in [1].

Differentiation of Forces

The kinetic equations of the bus at steady state is:

$$0 = m \cdot a = F_{tot} = F_{tr} - F_{air} - F_r \tag{2.10}$$

by a first order taylor approximation of this equation, a sensitivity study can be made by varying variables of interest.

$$\begin{aligned} F_{tot}(v) = F_{tot}\mid_{v=v_{ss}} + \sum_{i=1}^{n}\left(\frac{\partial F_{tot}}{\partial x_i}\mid_{v=v_{ss}} \cdot \Delta x_i\right) + \\ + \sum_{i=1}^{n}\left(\frac{\partial^2 F_{tot}}{\partial x_i^2}\mid_{v=v_{ss}} \cdot (\Delta x_i)^2\right) + \ldots \end{aligned} \tag{2.11}$$

x_i denotes the different variables studied. If the higher order terms are neglected, only the first order derivatives are used and the difference in tractive force, ΔF_{tot}, can be identified as the second term in (2.11):

$$\Delta F_{tot} = \sum_{i=1}^{n} \left(\frac{\partial F_{tot}}{\partial x_i} \mid_{v=v_{ss}} \cdot \Delta x_i \right) \tag{2.12}$$

The differentiation of F_{tot} with respect to the variables studied is done below.

Tractive Force, F_{tr}

$$\frac{\partial F_{tr}}{\partial r} = -\frac{1}{r^2} \cdot M_w \tag{2.13}$$

$$\frac{\partial F_{tr}}{\partial \eta_{dg}} = \frac{1}{r} \cdot i_{dg} \cdot \eta_{gb} \cdot \eta_{TC} \cdot M_{flywheel} \cdot i_{gb} \cdot i_{TC} \tag{2.14}$$

$$\frac{\partial F_{tr}}{\partial \eta_{gb}} = \frac{1}{r} \cdot i_{dg} \cdot \eta_{dg} \cdot \eta_{TC} \cdot M_{flywheel} \cdot i_{gb} \cdot i_{TC} \tag{2.15}$$

$$\frac{\partial F_{tr}}{\partial \eta_{TC}} = \frac{1}{r} \cdot i_{dg} \cdot \eta_{dg} \cdot \eta_{gb} \cdot M_{flywheel} \cdot i_{gb} \cdot i_{TC} \tag{2.16}$$

$$\frac{\partial F_{tr}}{\partial \tau_{aux}} = -\frac{1}{r} \cdot i_{dg} \cdot i_{gb} \cdot i_{TC} \cdot \eta_{dg} \cdot \eta_{gb} \cdot \eta_{TC} \tag{2.17}$$

Air Resistance, F_{Air}

$$\frac{\partial F_{Air}}{\partial C_D} = \rho \cdot A \cdot \frac{v_{m/s}^2}{2} \tag{2.18}$$

Rolling Resistance, F_r

$$\frac{\partial F_r}{\partial Crr_{iso}} = \frac{1}{1000} \cdot m \cdot g \tag{2.19}$$

$$\frac{\partial F_r}{\partial Ca} = \frac{1}{1000} \cdot m \cdot g \cdot (v_{km/h}^2 - 80^2) \tag{2.20}$$

$$\frac{\partial F_r}{\partial Cb} = \frac{1}{1000} \cdot m \cdot g \cdot (v_{km/h} - 80) \tag{2.21}$$

$$\frac{\partial F_r}{\partial m} = \frac{1}{1000} \cdot g \cdot C_{rr} \tag{2.22}$$

2.3.3 Analysis

It is now possible to relate the parameter errors to the difference in produced engine torque i.e. how the engine torque depends on errors in the parameters.

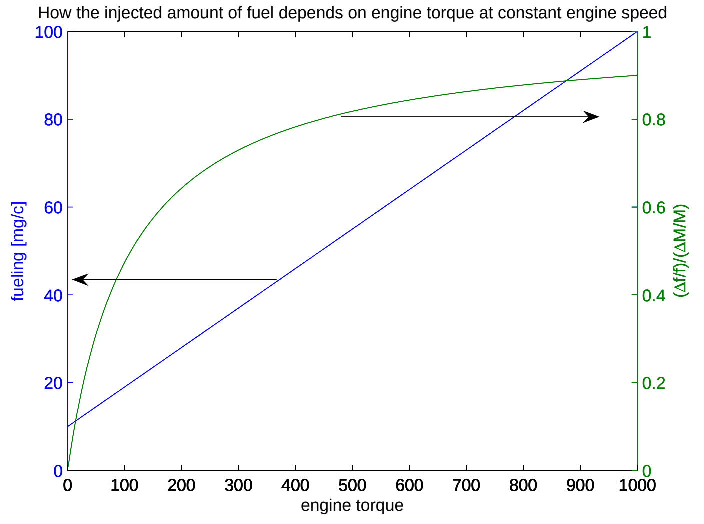

Figure 2.6: *How the injected amount of fuel depends on engine torque. A small amount of fuel is needed to keep the engine running at constant minimum engine speed. At this point the efficiency from the fuel to the power out of the engine is zero. The figure is used to explain the relationship between* $\frac{\Delta f_i}{f_i}$ *and* $\frac{\Delta M}{M}$ *i.e. the ratio of how much the fueling has to be increased when the torque is increased with x%. Later in this report it is shown that the engine torque seldomly exceeds 400 Nm at constant speeds on even roads for a bus equipped with a typical city bus powertrain.*

Further, it is possible to relate the difference in torque with the difference in fueling. The injected amount of fuel is not linearly dependent of engine torque but increases with increasing torque at constant engine speed, see Figure 2.6. A rough estimation seen in the figure is that the injected fuel increases with the order of 0 - 1% when the torque is increased with 1%. This is due to the small amount of fuel that is needed to keep the engine running at a constant speed without producing any torque, the power produced in the engine is needed to overcome the engine's internal friction. If the fueling at zero load is increased, the torque is "infinitely" increased leading to the low value at low loads. For high loads ($M_e >> 0$) the idle fueling can be neglected and the fraction is 1.

The equation describing the deviation of engine torque is:

$$\Delta M_e = \frac{(\Delta F_{tot} + F_{tr} \mid_{v=v_{ss}}) \cdot r_w + \tau_{BL}}{i_{tot} \cdot \eta_{tot}} + \tau_{aux} - M_e \quad (2.23)$$

where τ_{aux} is the sum of the contributions from the auxiliary devices. v_{ss} is the speed at the steady state vehicle speed studied.

The resulting ΔM_e in percent of engine torque at the steady state point is presented below. The variables are changed $\pm\ 5\%$ from their steady state values except from the wind speed that is changed $\pm 5\ m/s$ to show the effect of winds at the test track. The simulations are done in the speed range of 10-60 km/h.

2.3.4 Results and Comments

The results from the calculations are presented in Table 2.3.4. The behavior of the fault, the ”characteristics”, at increasing vehicle speed is presented as well. The absolute result of varying a variable +5% is the same as varying it -5%. This is valid for small deviations from the linearized operating point. The sign of the partial derivative shows if an increase of the parameter leads to an increased engine torque or not. This is obvious (increasing the efficiency results in a decreased engine torque etc.) and is not presented below. The results are strongly dependent on the sign and size of the partial derivatives and the results can be understood analyzing them.

Table 2.3.4: The results of the analysis.

Variable	Max [%]	Min [%]	Characteristics at increased speed
η_{TC}	5	1	increasing
η_{GB}	5	1	increasing
η_{DG}	5	1	increasing
τ_{Steer}	1	< 1	decreasing
τ_{Comp}	1	< 1	decreasing
τ_{Fan}	< 1	< 1	decreasing
τ_{Gen}	4	< 1	decreasing
r	5	1	increasing
τ_{BL}	< 1	< 1	constant
C_D	3	< 1	increasing
$Wind$	35	5	increasing
Crr_{iso}	3	1	max at 30 km/h
$C_b\ C_a$	< 1	< 1	constant
m	2.5	1	max at 30 km/h

Efficiencies

It can be seen that errors in the efficiency of the powertrain components affect the torque equally. The partial derivatives are almost equal for these variables since the different efficiencies are almost equal. The error increases with increasing speed since the flywheel torque increases with vehicle speed to overcome the external forces acting on the bus.

Auxiliary Devices

The torque resulting from errors in auxiliary devices decreases with increasing vehicle speed. The load from the auxiliary components is approximately constant and relatively small which results in a small dependence. The greatest error is in the generator which is the most power consuming component.

Wheel Radius

Increasing vehicle speed and decreasing the variable r gives a reduced need of torque. This depends on the decreased lever the wheel radius represents. Note that the speed also decreases leading to a need of increasing the vehicle speed but it is not taken into account in the calculations. The total ratio of the powertrain can be calculated by dividing wheel speed with engine speed, this error can therefore easily be obtained if it exists.

Bearing Losses

Errors in the bearing losses have a small influence on engine torque, the reason is as for the auxiliary components that the total torque is relatively small (totally approx. 10 Nm).

Air Drag and Load

Errors due to C_D have a relatively small influence of the engine torque at the speeds that are actual for city bus traffic. The fault increases more at high speeds because the partial derivative is proportional to the square of the speed.

The effect of loading the bus with a mass is small and relatively constant. An extra load of 5% results in a need of increased engine torque in the order of 2.5%.

Wind

A remarkable effect is due to the head wind. The engine torque has to be increased with 30 % at high speeds to overcome the additional force due to a head wind of 5 m/s. This has to be considered when the measurements are performed. Simulation results shows that an additional wind speed of 5 m/s results in an increased fuel consumption of 5% for a SORT 1 cycle where the mean speed is below 14 km/h.

Wheel Parameters

The errors in C_a and C_b can be neglected but $C_{rr_{iso}}$ is of greater importance. If the value of $C_{rr_{iso}}$ is correct, the uncertainty in the steady state speed based model might result in a greater fault.

Strength of the Analysis

A small example shows the strength of the analysis: The mean speed of the SORT 1 cycle if the stop times are subtracted is 25 km/h. It is shown above that the wind results in a 30% increase in torque. At a mean operating point the ratio between fueling and torque is 0.2 for a SORT 1 cycle. The total fuel increase would then be: $30\% \cdot 0.2 = 6\%$. This result is to be compared with the 5% increase simulated above.

Chapter 3

Measurements and Calculations of Losses

The sensitivity analysis shows that errors in the powertrain components and travel resistance result in the greatest influence on the fuel consumption. Therefore tests are performed to collect data to validate the existing models or to build new ones. In the calculations below, the model library's equations have been used if nothing else is written.

The first test is made at Björkvik in June 2007. Coast down tests with the gearbox at neutral as well as with the gearbox in normal drive mode where the engine is dragged are made. In addition to this, steady state tests to validate the gearbox model are performed.

In August 2007, new measurements are performed at Scania's test track in Södertälje. The aim this time is to collect more data to assure that the assumed losses detected at the first test did not happen at haphazard.

The results of the tests are presented here. Possible model improvements and explanations to the measurements will be discussed in the next chapter.

3.1 Rolling Resistance

3.1.1 Coast Down Tests

The coast down tests at neutral are performed to get data where the vehicle is unaffected by losses from the gearbox. This data is then supposed to be used to examine the rolling resistance. The kinetic equations of the bus at coast down are:

$$m\,a = F_{tr} - F_r - F_{Air} \tag{3.1}$$

$$F_{Air} = \rho \cdot C_D \cdot A \cdot \frac{v^2}{2} \tag{3.2}$$

where F_{tr} is the tractive force, F_r is the rolling resistance and F_{Air} is the force due to the wind. During the tests, the velocity is measured to be able to calculate the acceleration. F_{Air} is commonly described as above [7]. This gives two unknowns, F_{tr} and F_r. The aim with the coast down tests is to get an expression for F_r. This is only possible if the expression for F_{tr} is known.

In Figure 3.1 the measured speed profiles can be seen. The acceleration has been approximated using equation (A.1). It is not of importance to know how this is done, therefore it is described in appendix A. In Figure 3.2 the calculated acceleration is presented. If the tractive force is zero, this would correspond to the total traveling resistance and F_r can be calculated using (3.1) and (3.2).

At a first glimpse, $F_{tr} = 0$ could be expected when driving with a disengaged gearbox but this can not be said to be valid for a real mechanical system of this type. The torque needed to drag the gearbox at neutral has not yet been modeled, therefore a new model has to be made for this calculation since most systems are influenced by damping and friction forces. The powertrain is affected by forces from the differential gear, the bearings and the gearbox. According to the SKF handbook [3], the bearing losses mainly depend on the load of the axle

$$\tau_{bearing} = N \cdot \mu \cdot \frac{d}{2} \tag{3.3}$$

where N is the load on the bearing, μ is the internal friction, d the diameter and $\tau_{bearing}$ is the torque of resistance. This gives a value of approximately 10 Nm for a 12 tonnes bus.

The torque needed to rotate the gearbox at neutral is not known. A rough estimation is that it is in the order of 20 Nm at 40 km/h (see section 4.1) and decreases with the speed. A simple linear expression is described below.

$$\begin{aligned}
\tau_{GB} &= -\tau_{GB_0} \cdot \frac{v}{v_0} && (3.4)\\
\tau_{bearing} &= -N \cdot \mu \cdot \frac{d}{2} && (3.5)\\
F_{tr} &= (\tau_{GB} \cdot i_{DG} + \tau_{bearing})/r && (3.6)
\end{aligned}$$

With this expression, F_r can now be calculated. With aid of

$$F_r = \frac{C_{rr}}{1000} \cdot N \tag{3.7}$$

the speed dependent term, C_{rr} is calculated and is presented in Figure 3.3. Additionally to the calculation, a 95% confidence interval is calculated since many coast downs were made. Because of the different number of data for a certain speed, the confidence interval deviates more or less even though the values of C_{rr} are close to each other. The teory behind C_{rr} will be further described in chapter 4.

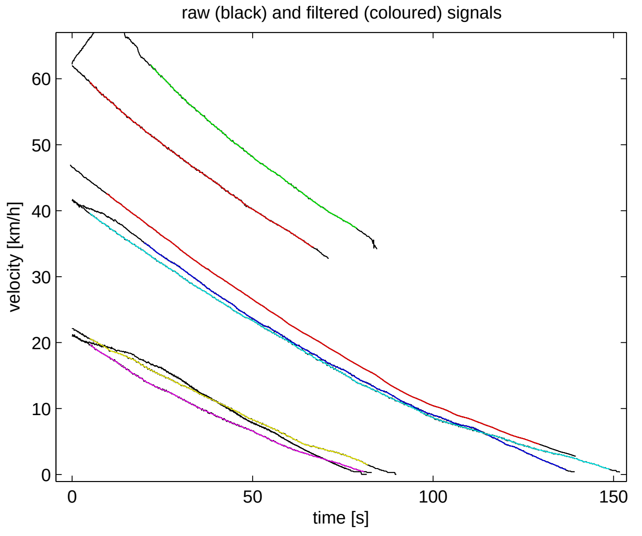

Figure 3.1: *The logged speed profile during the coast downs. The raw signal is plotted in black, the used (filtered) signal is plotted in color. See appendix A for more information regarding the filtering technique.*

3.2 Powertrain Losses

3.2.1 Steady State Tests

By driving the bus at steady state speeds at a constant slope and logging the calculated flywheel torque from the CAN-bus it is possible to do steady state comparisons between measurements and simulations. The tests were carried out in Björkvik on the even landing track and the bus was driven at constant speeds for approximately 20 seconds each in both directions. The test was made during a short time (a few minutes) to minimize the effects of changed rolling resistance that the tire temperature can give rise to.

By comparing the quote engine speed with the vehicle speed $\frac{\omega_e}{v}$ in the simulations and the measurements it has been established that the wheel radius and the gear ratios are correct. This means that the engine speed is the same in the simulations as in the measurements for a given vehicle speed. If the driving torques are the same, the power to drive the vehicle forward will be the same. This would probably lead to equal fuel consumptions in calculations and measurements.

From the steady state tests, the tractive force is calculated using the engine torque and the equations described in section 2.3.1. The result is shown in Figure 3.4. The tractive force is greater than the theoretical force needed to overcome the rolling and air resistance where the rolling resistance is de-

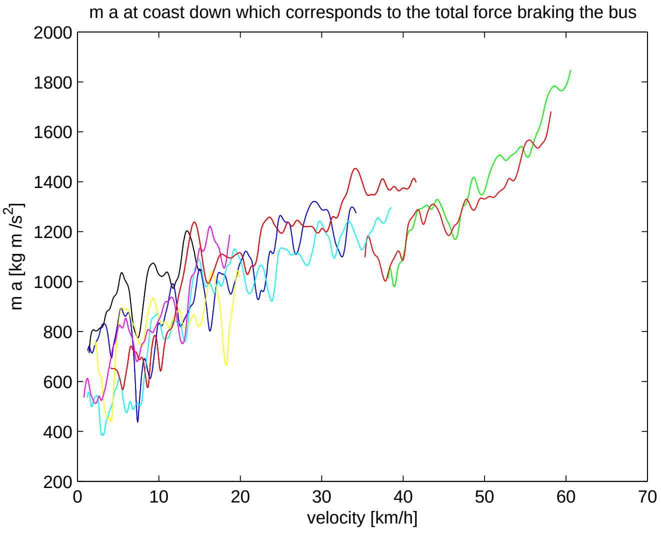

Figure 3.2: $m \cdot a$ *calculated from the data in Figure 3.1at the coast downs. From this data,* F_r *can be calculated using equation* (3.1)*. If the powertrain losses are assumed to be zero, this corresponds to the sum of the Air and rolling resistance. Totally 8 coast downs. The coast downs have been performed with different buses with different masses, therefore the accelerations have been different. When recalculating the forces to a value of* C_{rr}*, the masses of the buses can be taken into account.*

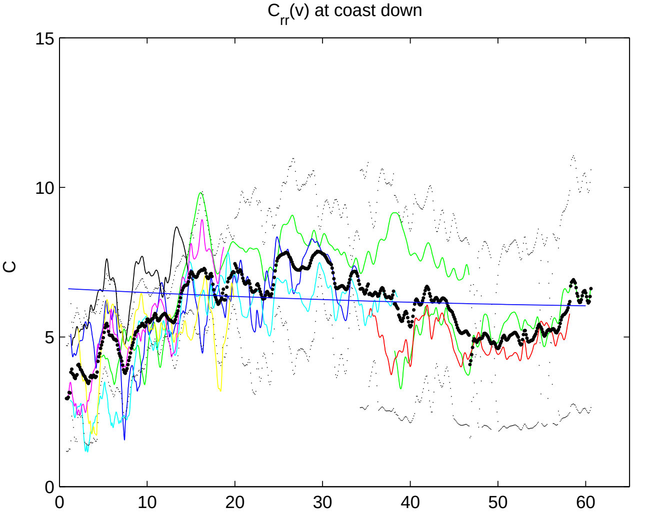

Figure 3.3: *C_{rr} as a function of speed With a 95% confidence level (small black dots). The mean C_{rr} has been calculated (big black dots). The thin blue line is C_{rr} as described by Michelin for the actual tire (only used as a reference). At Björkvik, one coast down test was performed from 60-0 km/h. The calculated C_{rr} from Björkvik did not correspond to the model described by Michelin nor the model described in [7]. Therefore additional coast down tests were performed at Scania's test track. Because of the short length of the track in Södertälje, the tests were performed with different starting speeds and the test was stopped when the bus reached the end of the track. The tests were performed in both directions to avoid disturbances due to gradients or winds. The results of the calculations from the tests in Södertälje shows the same tendency as the from the ones performed in Björkvik.*

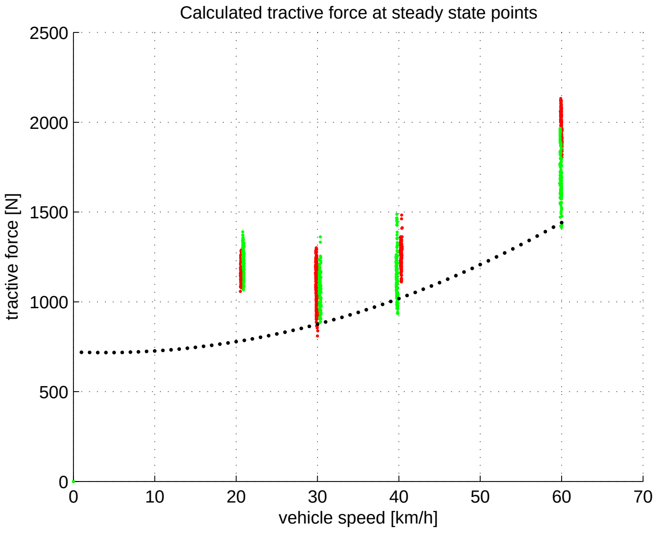

Figure 3.4: *Calculated tractive force for the steady state measurements at Björkvik. The black line is the sum of theoretical rolling and air resistance. The measured tractive force should be equal the theoretical assumption if the power to drive the vehicle is equal in the both cases. This is not the case in this figure.*

scribed by Michelin. In the previous section it was indicated that the rolling resistance could be greater than first assumed.

The difference in C_{rr} was less than 3 kg/tonne for the theoretical and measured value. Using this the difference in tractive force can be calculated.

$$\Delta F_{tr} = \frac{\Delta C_{rr}}{1000} N = 300\ N \tag{3.8}$$

This can perhaps describe the deviations at 30 km/h where C_{rr} seems to deviate at most but not the deviations at the other speeds. It seems as there are other losses in the powertrain that are not yet modeled.

3.3 Conclusions

From the results of the measurements it can be concluded that there exists losses that has not yet been properly modeled. It can be pointed out that they probably originate from the gearbox (powertrain) and the rolling resistance. There has not yet been investigated how to find out from where the losses originates. This will be handled in the next chapter.

Chapter 4

New Models of the Gearbox and the Rolling Resistance

4.1 The New Gearbox Model

The results from the steady state tests shows that the powertrain losses are underestimated in some driving cases as modeled today. This way of measuring makes it hard to calculate the losses by using the collected data. The signals are noisy and the measured torque deviates a lot from the mean value. Instead of building a new model based on the collected data, the approach is to use the data that is known but to calculate the losses in other ways compared to how it previously was done.

Mainly, the losses in the powertrain are supposed to originate from the gearbox. This is a component where the manufacturer has provided data but only data for torques above 600 Nm. At Björkvik the measured torque was in the order of 300 Nm at 60 km/h. This could be a problem since extrapolation is used to find the efficiency of the gearbox at a certain operation point. A simulation is made to estimate the order of how big the error is. It shows that the efficiency of the gearbox always exceeds 90% for a SORT 1 cycle. This can not be said to be true for a mechanical system of this type, in at least some operating point (low load), the efficiency should be very low due to internal friction. Figure 4.1 shows data representative for most gears. It can be seen that the efficiency increases with increasing load and decreases with increasing engine speed. The efficiency seems to be linear at high torques but the distance between the lines increase at low torques. This phenomenon is not taken into account if linear extrapolation is applied.

The equations of the gearbox when the vehicle is driven are, for the old

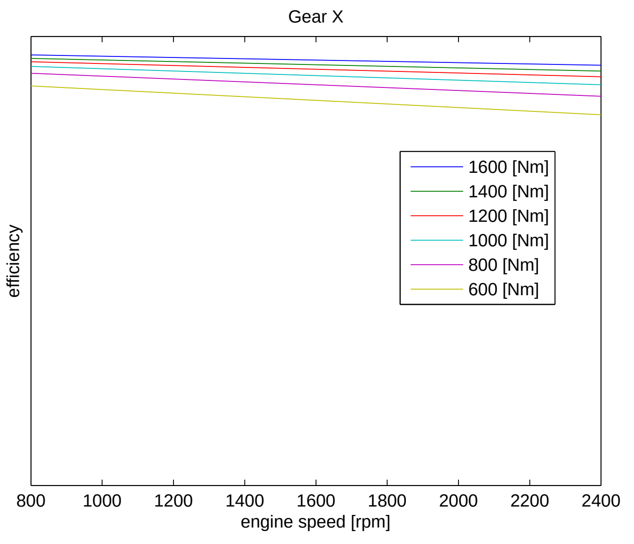

Figure 4.1: *The manufacturers figures of the efficiency of a gear. These figures were used for linear extrapolation of values outside the data range in the old gearbox model.*

gearbox model:

$$\omega_{in} = i_{gb} \cdot \omega_{out} \tag{4.1}$$

$$\tau_{out} = \eta_{gb} \cdot i_{gb} \cdot \tau_{in} \tag{4.2}$$

Where in corresponds to the engine side of the gearbox and out to the propeller shaft side.

The non physical part of this way of modeling the gearbox is that the gearbox will keep running with constant speed if the input power is zero. A real mechanical system of this type is affected by internal friction which would brake the gearbox if it was driven in such an operation point. The internal friction can be seen as a torque braking the gearbox and can be calculated for the operation points given by the manufacturer.

4.1.1 Internal Friction of the Gearbox

To give some background to the way of modeling the internal friction in the gearbox, a general description of friction is given here. For rotating systems, the friction is often modeled as linearly dependent on some representative speed which in this case would be the angular velocity ω of some component of the gearbox.

$$\tau_{fric} = c_1 \cdot \omega + c_0 \tag{4.3}$$

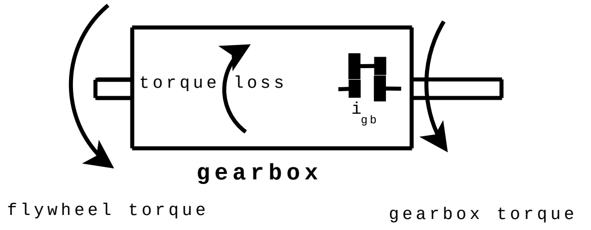

Figure 4.2: *The new gearbox model. The lost torque is a function of engine speed and engine torque and is subtracted from the engine torque before an ideal gear.*

c_1 corresponds to the viscous damping coefficient. Higher order descriptions are also common but it is reasonable to test a low order expression first. If the value of the damping coefficient is taken in a linearized operation point of the system, it could be of interest to include an offset c_0, this offset could be seen as the resting friction of the system.

The gearbox can now be studied. At steady state, the friction can be calculated by using equation (4.4). This results in a torque that brakes the gearbox and corresponds to τ_{fric} in the equation above. It is important to point out that for the operation points in the data range given by the manufacturer, the efficiency is unchanged if this model is used instead. The main advantages of this model is that it is expected that the torque losses are linear with respect to driving torque but not the efficiency.

$$\tau_{loss} = (1 - \eta_{GB}) \cdot \tau_{in} \tag{4.4}$$

The result of these calculations are presented in figure 4.3. The result of the calculations shows that τ_{loss} is increasing with increasing torque and increasing with increasing speed. The exception is gear 4 which is a direct gear with $i_{GB} = 1$ where τ_{loss} seems to be constant for different torques. If this model is used and the extrapolation is made here the model behaves more realistically. The value of η_{GB} decreases to zero for a driving torque equal to τ_{loss} where τ_{loss} is in the order of 20 Nm.

This model also has the benefit that the energy is lost if the driving torque is too low. This was not the case before. Figure 4.2 shows the new gearbox model where the loss has been implemented before an ideal gear.

In figure 4.4, the efficiencies have been recalculated to be able to compare the new gearbox model with the old one presented in figure 4.1. The benefits of the new model is that the efficiency drastically falls when the efficiency decreases below 400 Nm and the efficiency is zero if the driving torque is equal to τ_{loss}.

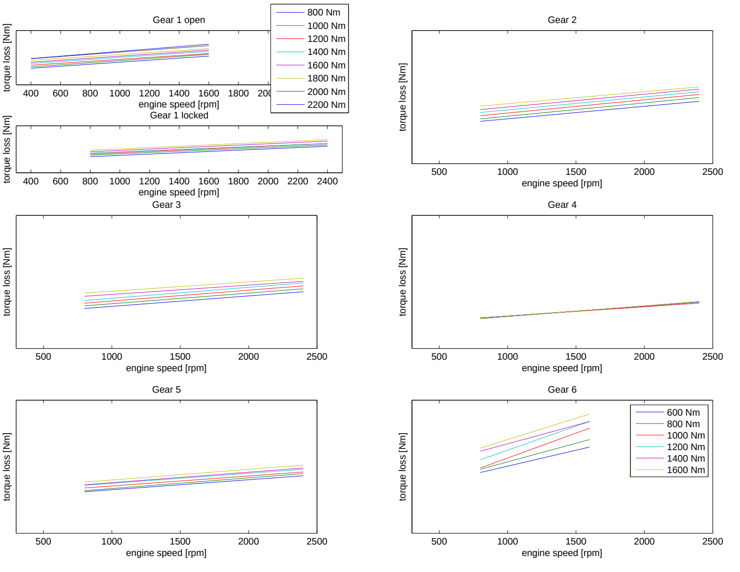

Figure 4.3: *The new torque loss model. The plots are all made with the same (hidden) axle values. Gear 1 with open converter has been mapped for other torques and speeds than the other gears. It seems more plausible that the efficiency is correct if the extrapolation is made from these values of* τ_{loss}.

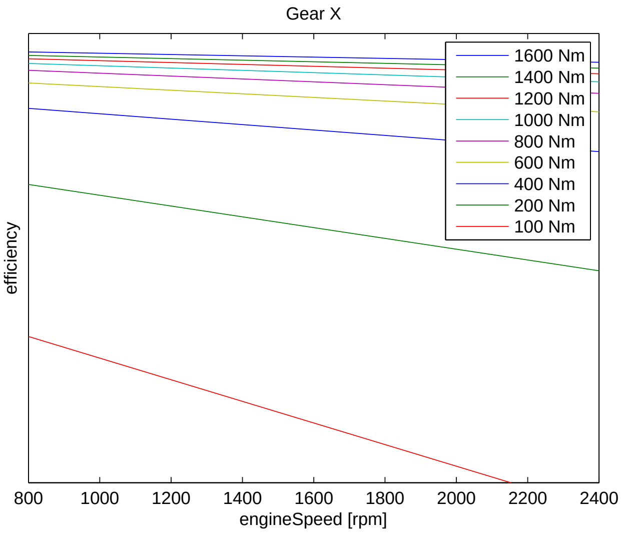

Figure 4.4: *New efficiency, complemented with low torque efficiency after model improvements. The efficiency drastically falls when the torque decreases below 400 Nm. Note that the efficiencies for high torques are the same as before. The same scale as in figure 4.1 has been used.*

4.2 Rolling Resistance

Rolling resistance is normally described as a function of speed and normal force. The speed dependence is at steady state speeds strongly coupled to the tire temperature. Normally it is described as below:

$$F_r = C_{rr}(v, T) \cdot N \tag{4.5}$$

There are different approaches to describe the physics behind C_r. Normally steady state behavior [1] used by Michelin and transient behavior [7]. The steady state approach states that the tire temperature and inflation pressure increase with speed. The increased temperature leads to an increased inflation pressure which results in a lower value of C_r and the rolling resistance at increasing speed. The tire's mechanical construction can lead to deviations from this theoretical assumption. These phenomena normally appears at high speeds.

The transient approach states that the value of C_{rr} increases with increasing speed when the temperature is held constant [7]. The value of C_{rr} is in these approaches normally proportional to the speed square.

Sandberg [14] combined these theories and presented a model valid for both transients and steady state conditions:

$$\begin{aligned}
C_r(T, v) &= C_{r0}(T) + C_{r1} \cdot (v^2 - v_{sc}^2) && (4.6)\\
v_{sc} &= g_{sc}^{-1}(T) && (4.7)\\
\frac{dT}{dt} &= -\frac{1}{\tau} \cdot (T - T_{sc}) && (4.8)\\
T_{sc} &= g_{sc}(v) && (4.9)
\end{aligned}$$

Here, C_{r0} is the temperature dependent term and v_{sc} the steady state speed corresponding to the current tire temperature. g_{sc} is the function relating v and T. The time constant τ was experimentally found to be in the order of $1000\ s$ for a running truck tire [5]. At stops, the physics seemed to deviate from the physics for a running tire. The cooling of the tire changed and the time constant differed from the one obtained for a running tire.

When reading literature describing the physics of a tire, nothing has been found that explicitly deals with rolling resistance of a tire at low speeds. A possible explanation to this could be that rolling resistance historically has been more interesting in the truck industry where the main interests are long haulage tests and simulations at relatively high speeds. In urban bus industry there are other effects (mainly load) that affect the fuel economy more than rolling resistance of a vehicle and the interest has therefore been low in studying the rolling resistance at low speeds.

At the coast down tests, one could expect that the appearance of C_{rr} would be as described by Wong since the tire temperature could be expected to be relatively constant during the short time the tests were performed. As seen in figure 3.3 the value of C_{rr} differs a lot from the one described by Michelin. This is not remarkable because of the different approaches and the model described by Michelin is only used as a reference. What is remarkable is that the appearance does not satisfy the theory presented by Wong neither.

Below it is tried to find the explanation to this.

4.2.1 Explanations of the Value of Rolling Resistance

There are many uncertainties that can affect the final result. The main sources of disturbances when the measurements are carried out this way are:

- **Powertrain losses** - if modeled incorrectly the resulting C_{rr} will be wrong since the powertrain losses are taken into account. When using a strain gauge on the propeller shaft, only the bearing losses has to be modeled.
- **Geography** - the shape of the curve could be explained by an unknown road gradient.
- **Wrong method** - is the method used really a reliable method for this purpose?
- **Wrong model of air resistance** - a wrongly modeled air resistance will result in a wrongly shaped curve since the model of air resistance is used to calculate C_{rr}.

If the value of C_{rr} is ignored for a while, it can be discussed how the factors above influence the final result.

Powertrain losses have been modeled in different ways. They have been set to constant values and linearly dependent on gearbox speed. The result is that the absolute value of C_{rr} changes a little but the characteristic shape of the curve remains.

Geography The presence of a force depending on road gradient has a big influence on the calculated C_{rr}. If the equation used before to describe the coast down behavior is complemented with the road gradient dependent force F_{grad} we get:

$$m\,a = F_{tr} - F_r - F_{Air} - F_{grad} \tag{4.10}$$

$$F_r = F_{tr} - F_{Air} - m\,a - F_{grad} \tag{4.11}$$

$$\frac{C_{rr}}{1000} \cdot m \cdot g = F_{tr} - F_{Air} - m\,a - m\,g\,\sin\alpha \tag{4.12}$$

$$C_{rr} = 1000 \cdot \frac{F_{tr} - F_{Air} - m\,a}{m\,g} - 1000 \cdot \alpha \tag{4.13}$$

small angles have been assumed in the last equation. The slopes were at least small enough not to be detected by a human eye. A small slope with a

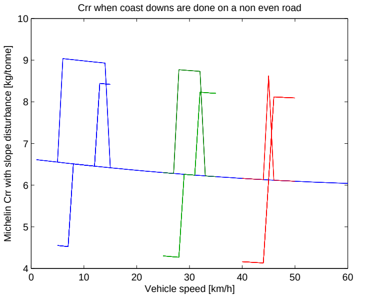

Figure 4.5: *If C_{rr} would be as described by Michelin (the blue line) then this is the effect if the coast downs are performed in a neat slope or hill. Because of the difference in speed, the time to pass the slope gets shorter. Therefore it seems as if the hill gets shorter in this figure. The slopes correspond to a gradient of less than* $0.2^\circ = 0.3\% = 0.0035rad$

gradient of $\alpha = 1^\circ = 0.017\ rad$ will correspond to an increase in C_{rr} of 17 according to the calculations which could perhaps explain the deviations. If the coast downs ended in neat downward slopes, the calculated C_{rr} would be smaller than in reality.

The results of how a slope or a hill would appear in Figure 3.3 can be seen in figure 4.5.

A problem is that the test track in Södertälje was too short. The coast downs had to be performed with different starting speeds to be able to get data from 60 to 0 km/h. This resulted in that the ending speed was 40 km/h in two of the tests and 0 in the 5 last ones. The slope would therefore be seen also at 40 km/h. The coast downs were also performed in both directions which would result in the opposite value when driving in the opposite direction.

Wrong way of measuring F_r. There are many works and articles done that uses coast down tests to model travel resistances. The accuracy of these results can be discussed but it is a commonly used way of measuring the traveling resistances. In [12] a more accurate test procedure for measuring traveling resistance by coast down analysis is explained. The parameters are calculated based on the time it takes to coast from one reflective tape to another and the method describes how the time it takes to coast a certain distance can be measured more accurately. The method used in this thesis is to use speed sensors with high accuracy which gives an accurate description of the deceleration. In [12] the test track has to be divided into a fix number of sections. It is not sure that the results obtained here would have been obtained if the method described in [12] was used.

Wrongly modeled air resistance. The model of air resistance is commonly found in automotive literature describing the subject area. The above described uncertainties can not totally describe the difference in C_{rr}. The

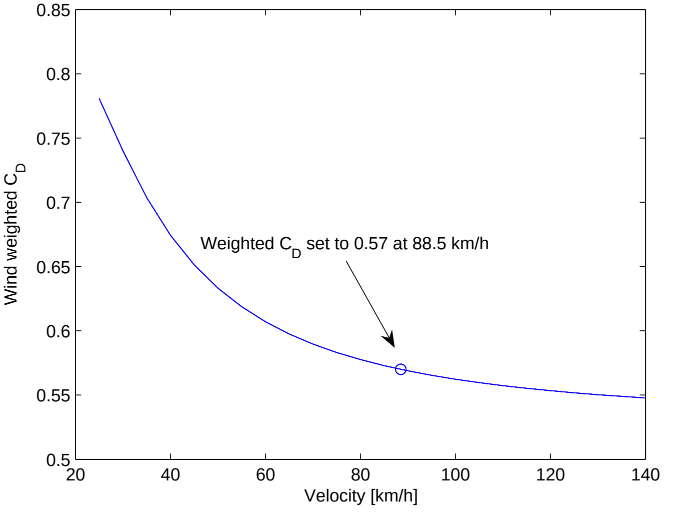

Figure 4.6: *The result of the analysis of how C_D depends on the speed when side winds are taken into account. C_D was previously modeled as a constant.*

last thing that can overturn the new model of rolling resistance is the value of C_D. It has been assumed to be constant but its dependence on the Reynolds number is not neglecteable [4]. The question is whether the relation can be said to be constant for velocities close to the ones commonly driven in city bus traffic or if it is just a linearization at a certain velocity.

The aerodynamicists at the group RTTF at Scania were consulted to clear up the difficulties. Their results show some interesting facts which can be seen in Figure 4.6 [1]. The results from the analysis are:

- C_D depends on side winds.
- The dependency is stronger if the edge radius of the bus are decreased.
- C_D can partly describe the deviation seen in C_{rr}.

C_D, and the force due to air drag is increased with more than 30% at low speeds. How much does this result affect the value of C_{rr} that was calculated in chapter 3? Differentiating (4.13) with respect to C_D yields:

$$\frac{\partial C_{rr}}{\partial C_D} = -\frac{1000}{m\,g} \cdot \rho \cdot A \cdot \frac{v^2}{2} \tag{4.14}$$

and

$$\Delta C_{rr} = \frac{\partial C_{rr}}{\partial C_D} \cdot \Delta C_D \tag{4.15}$$

[1] C_D was set to 0.57 at 88.5 km/h in the analysis. Calculations are done according to [2]

Evaluating this equation in the operation points of interest yields:

- $\Delta C_{rr}(v_{km/h} = 0) = 0$
- $\Delta C_{rr}(v_{km/h} = 42) = -0.6$
- $\Delta C_{rr}(v_{km/h} = 88.5) = 0$

This is not enough to describe the results obtained in section 3.1.1 but it gives an explanation to from where the deviations may originate. The results obtained by RTTF also show that the dependence on the edge radius of the bus is big. The value of C_D's dependence on speed has been taken as an weighted average of wind speed and side wind. During the coast downs in Södertälje, a side wind of approximately 5-8 m/s was present. This could be more than the ”weighted average” and it is not investigated here how much more this could affect the results. If C_D's dependence on side winds are greater than the results obtained by RTTF shows, this might explain some of the results in Figure 3.3.

4.2.2 Proposed Model of Air- and Rolling Resistance

When simulations are to be performed where the vehicle operates at steady state speeds the model described by Michelin is to be preferred. In simulations where mainly transients are studied e.g. city bus traffic, the model described by equation (4.6) to (4.9) will probably give the most reliable simulation results.

Sandberg claimed the time constant of a truck tire to be in the order of 1000 s. It was also assumed that the energy developed in the tire depended mainly on vehicle speed. It is probable that it also depends on load if the load is varied much from the values studied in the thesis. The steady state temperature would then be lower if the axle load is lowered. This would increase the coefficient of rolling resistance. Though, the tire's mechanical construction is the same which indicates that the time constant remains the same.

One simplification due to the big time constant would be to model the tire temperature as constant if the mean speed is relatively constant (or the simulated cycle is short). The tire temperature (and the coefficient of rolling resistance) would then depend on the (predicted) mean speed as in equation (4.16). For example, if a SORT 1 cycle should be simulated, C_{rr} should depend on the mean speed which is 13 km/h.

$$C_{rr} = C_r(v_{mean}) + C_a \cdot (v^2 - v_{mean}^2) \tag{4.16}$$

With the velocities expressed in km/h the value of C_a is $0.23 \cdot 10^{-3}$ according to Wong [7]. If the mean speed is 30 km/h, C_{rr} varies with 0.828 between 0 and 60 km/h which is approximately 15% with $C_{rr_{iso}} = 5.5$.

C_D's dependence on vehicle speed also has to be taken into account. The calculations performed by RTTF showed that C_D is near constant if no side winds are present but that the dependence on side winds is big. If the simulation results should agree with realistic circumstances, C_D should be a function of vehicle speed.

Chapter 5

SORT Specific Models

”Founded in 1885, UITP is the world wide association of urban and regional passenger transport operators, their authorities and suppliers” [1]. This association has developed a standard with reproducible test cycles for on road tests in order to measure fuel consumption [17]. The standard proposes three different ”idealized” drive cycles which can be combined to fit to a bus operators route. The three cycles are named SORT 1, SORT 2 and SORT 3 where SORT 1 is the most urban like cycle with a mean speed of 13 km/h and SORT 3 is the most suburban like cycle with a mean speed of 26 km/h, see Figure 5.2.

The SORT standard also describes how the cycles are to be driven. It contains values for the minimum acceleration at full-throttle and the constant retardation for different SORT cycles.

5.1 The New SORT Driver Model

To be able to simulate SORT cycles properly, a new driver model is introduced. The driver is able to start and stop and to wait a specified time at each stop. Demanded speed is a function of driven distance to make the comparisons between simulations and measurements easier. This is a more true model of how a speed profile is setup in reality when driving on a road or when SORT-cycles are driven. In these cases, the speed profile are distance dependent. Since the fuel consumption per kilometer is of main interest, the driven distance plays a significant role. How to create a SORT cycle is described in the next section. In the old model, the driver had a time dependent speed profile to follow. This approach leads to different travel distances for different bus setups which can affect the result.

Since the SORT cycle should be made at full throttle in the accelerations, the demanded speed is implemented as a step from the starts. The signal from

[1] Part of the foreword of the SORT standard [17]

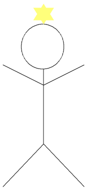

Figure 5.1: *The driver model*

the controller sometimes exceeds the possible maximum accelerator pedal position. A simple anti wind up PI-controller is implemented by modifying the existing controller. The difference from the original driver is that the integrator holds its value during the time the output signal reaches its maximum value. The method is a commonly used way of solving the problem with reset wind up in controllers and can be found in [16].

5.2 Construction of The Perfect SORT Cycle

The new SORT driver needs a distance dependent speed profile which has to be recalculated from the time dependent variables described in the SORT standard. How this is done is described below.

$$a = \frac{d\,v}{d\,t} = \frac{d\,v}{d\,s}\frac{d\,s}{d\,t} = \frac{d\,v}{d\,s}\,v = \frac{d}{d\,s}(v^2(s)/2) \tag{5.1}$$

Integration of (5.1) with respect to distance gives:

$$\int_{x_0}^{x} a\;ds = \int_{x_0}^{x} \frac{d}{d\,s}\frac{v^2(s)}{2}\;ds \tag{5.2}$$

which, at constant acceleration yields

$$a\;(x - x_0) = \frac{v^2(x)}{2} - \frac{v^2(x_0)}{2} \tag{5.3}$$

$$v(x) = \sqrt{2a(x - x_0) + v^2(x_0)} \tag{5.4}$$

the stop durations are saved in a table

$$stopTime = \begin{bmatrix} 1\;0 \\ 2\;20 \\ 3\;20 \\ 4\;20 \end{bmatrix} \tag{5.5}$$

where stop number 1 is the time to wait at the start.

This results in a driver who stops within 5 cm at each stop and always waits the right time according to the SORT standard, corresponding to how a driver behaves in a measurement.

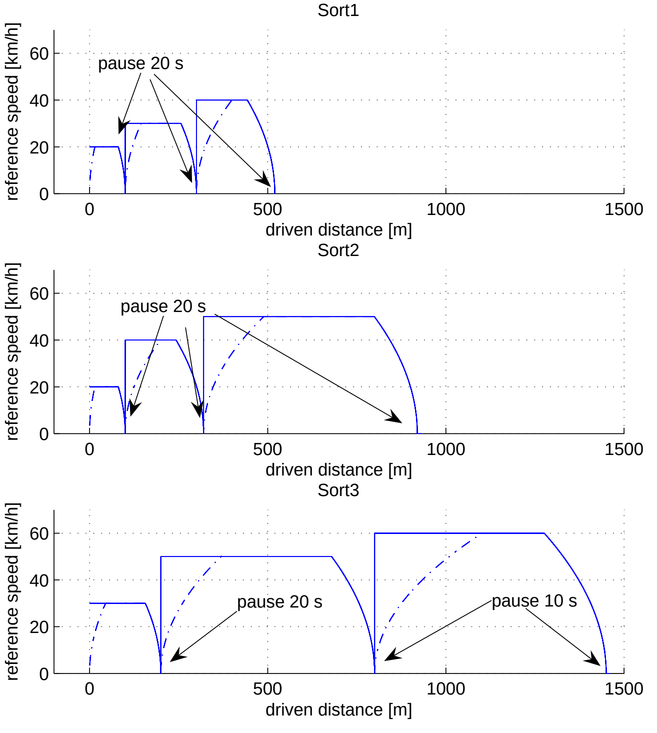

Figure 5.2: *The distance dependent speed profile for the SORT cycles. The line is the step reference speed and the dash dotted line is the speed profile calculated as function of minimum (constant) acceleration.*

5.3 The Driver's Influence on Fuel Consumption

An additional study of the driver's influence of fuel consumption is made to give an understanding of the validation on SORT-cycles. It is said that the driver can save up to 20 % fuel by eco-driving-training [13]. The main ideas of eco-driving are to drive the vehicle at full throttle in the accelerations, skip gears, drive on low engine speeds, use the engine brake instead of braking and plan the stops more in advance etc.

5.3.1 Driver's Behavior on a SORT-cycle

To investigate how much the driver can influence the fuel consumption, simulations are performed with three different driver profiles. The first driver is the normal SORT-driver who drives the vehicle at full throttle until the demanded speeds are reached. The second driver is an aggressive driver, pressing the accelerator pedal a little too long which gives the result that this driver gets into the constant speed phase with a speed that is a little too high. Instead of braking, the driver presses the pedal a little less and the bus decelerates during the whole constant speed phase. There is also a third, wimpy, driver, who eases the accelerator pedal a little too early in the acceleration, resulting in a need for a gentle acceleration during the constant speed phase. To be able to compare these drivers, the mean speed is the same in the simulations. The results can be seen in Figure 5.3. Note that the only behavior that differs between these three drivers is the behavior during the constant speed phase. The study is made to give background to the results given in the next chapter where the driver behavior during the constant speed phase is one of the factors influencing the result.

The driver profiles drive 520 m in 125 s (they all stop within 3 cm and 0.2 s) which gives a mean speed of 15 km/h. The lowest fuel consumption is achieved by the driver accelerating a little too much. The third driver presented above has the highest fuel consumption. Compared to the "normal" driver, the fuel consumption differs with $\pm 3\%$ for these two drivers.

Driver type	**Normalized Fuel Consumption**
Normal	1
Agressive	0.97
Wimpy	1.03

Explanation

The results obtained above agrees with the ideas of eco-driving. This calculation is based on equal mean speeds of the different drivers. The efficiency of the powertrain components and engine is higher at high torque loads which benefits a driver who drives with a high loaded engine. If the mean speed is lowered, the fuel consumption normally decreases but this calculation shows

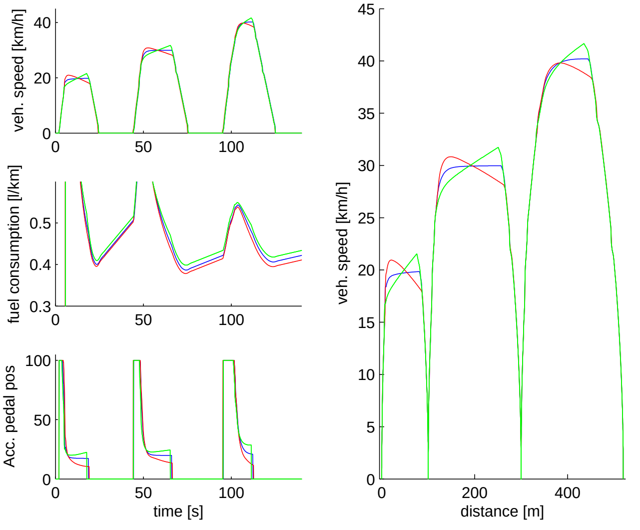

Figure 5.3: *The results of the simulations with three different drivers. The lowest fuel consumption is achieved by accelerating a little too long and ease the accelerator pedal to decelerate during the constant speed phase.*

that even small deviations from the perfect driven sort cycle results in big differences in fuel consumption. One could expect the engine efficiency to increase for the third driver but the increase is too small to compensate for the energy needed for the acceleration.

These results should be considered in the comparison of two buses which have driven the SORT cycles. If one of them accelerates faster than the second one, the fast one will probably have a higher fuel consumption if the two buses are equally loaded, and it will definitely reach a higher mean speed. If the customer only looks on these figures, he will probably buy the bus with the lowest fuel consumption. It is not sure that this bus will have the lowest fuel consumption on this customers route. This has not been mentioned in the SORT-standard and can be misleading for customers unaware of this.

5.3.2 Acceleration Limiter

During some of the tests to collect data, it was discussed why the fuel consumption decreases when a sort cycle is driven at low acceleration compared to a sort cycle driven at maximum acceleration. As said before, according to the eco-driving principle the accelerations should be performed at high acceleration. Why are pupils teached to drive that way if it is not fuel saving? This is a perfect task to study with aid of the model library.

An acceleration limiter is implemented in the EMS. The equations describing the dynamics of the vehicle are:

$$\dot{x}_1 = x_2 \quad (5.6)$$

$$\dot{x}_2 = -\Psi(x_2) - \Phi(x_1) + \frac{u}{m} \quad (5.7)$$

Where x_1 is the driven distance and x_2 is the vehicle speed. Ψ and Φ are non-linear functions of vehicle speed and distance. The control signal u is the tractive force. If Φ is neglected (it is identified as the road gradient dependent term), $\frac{u}{m}$ can be chosen so that:

$$\frac{u}{m} = \Psi(x_2) + r \quad (5.8)$$

where r is the reference value a_{lim} and Ψ is identified as the total driving resistance. In [15], this method is described as computed force or computed torque where u is chosen so that the system is perfectly linearized and the state variable will follow the reference value exactly if Ψ is calculated correctly. In this particular case where the driving resistance is known, u can be calculated by calculating the required engine torque and estimate the powertrain efficiency.

$$\Psi(x_2) = \frac{F_r + F_{Air}}{m} \quad (5.9)$$

The tractive force is as before:

$$F_{tr} = \frac{M_e \cdot i_{tot} \cdot \eta_{tot}}{r_w} \quad (5.10)$$

and the torque the engine has to produce ($M_{e_{req}}$) to achieve the demanded acceleration can now be calculated from (5.9) and (5.10):

$$M_{e_{req}} = \frac{(\Psi + r) \cdot m \cdot r_w}{\eta_{tot} \cdot i_{tot}} \quad (5.11)$$

$M_{e_{req}}$ can be achieved by requesting an injected amount of fuel by interpolation in a lookup table with engine speed and requested torque as inputs.

$$f_i = lookup(M_{e_{req}}, \omega_e) \quad (5.12)$$

The result of the controller can be seen in Figure 5.4. The disturbances are due to gearshifting where the controller is deactivated. On the first gear, a small disturbance from the hydraulic torque converter can be seen. On the 2:nd and 3:rd gear, the acceleration limiter works properly. On gear 4, the engine is no longer able to produce the requested torque to overcome the reference acceleration. The efficiencies in the powertrain are assumed

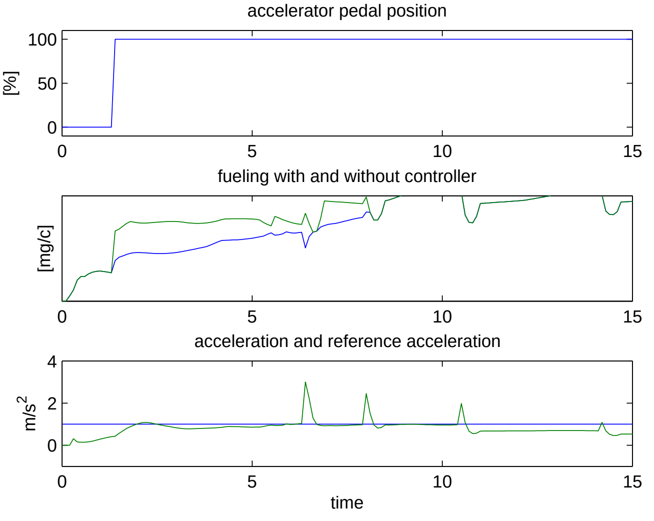

Figure 5.4: *The result of controlling the maximum injected amount of fuel to limit the acceleration by using a exactly linearising controller.*

to be constant. Deviations from this value will be handled by an integrator integrating the fault. The output of the integrator is added to the calculated f_i. i_{tot} is available from the CAN bus and F_r is assumed to be constant in the simulations, deviations are handled by the integrator. The result shows a decreased fuel consumption with 1.5 % for a SORT 1 simulation and 1.35% for the Braunschweig cycle.

This contradicts the recommendations from eco-driving. The explanation is that the mean speed is decreased with 3.6 and 2.6 % respectively when the controller is activated. A lowered mean speed results in lower travelling forces and in most cases a lowered fuel consumption. This is positive, but a bus driver will probably increase the constant speed a little to reach the next bus stop in time. This is not taken into account in the simulations. Therefore, an acceleration limiter will not always reach its purpose with saving fuel. The limiter will also have the best effect in a low loaded bus. A fully loaded bus with a weak engine will never reach the limited acceleration. Perhaps could a limiter be interesting as a comfort feature. A limiter could increase the comfort and reduce the effects of a driver pushing the accelerator pedal too much. On the other hand, it could give the drivers a bad opinion of Scania busses as slow and powerless.

Chapter 6

Model Validation

In this chapter the results from simulations are compared with measured data to validate the new models. Steady state tests were done at Björkvik, those are used to validate the total driving resistance including the powertrain, air and rolling resistance. Fuel consumption measurements were performed at Idiada. The simulated results are compared with the figures logged at Björkvik and Idiada. Since the driver at Idiada drove the vehicle a little different at each cycle, the mean value of the results from Idiada is presented including the statistical level of confidence.

6.1 Comparison of Steady State Torques

At Björkvik, the bus was driven at constant speeds and the driving torque was logged. Simulations show that the total ratio of the powertrain is correct at all gears. This results in equal engine speeds at equal velocities and gears. Figure 6.1 shows that the torque needed to drive the bus forward is similar in the simulation and the measurements. Deviations could be due to C_D and small slopes as discussed in chapter 4.

Since the engine speeds are equal, the power to drive the bus forward is equal. The gearbox model was built without using data from the measurements, the model of traveling friction was developed without the use of a gearbox model. The results shows that the new gearbox model describes the low load losses well and the losses are modeled better than in the original model.

If the manufacturer supplies Scania with more efficiency data, the uncertainties due to the gearbox can be improved.

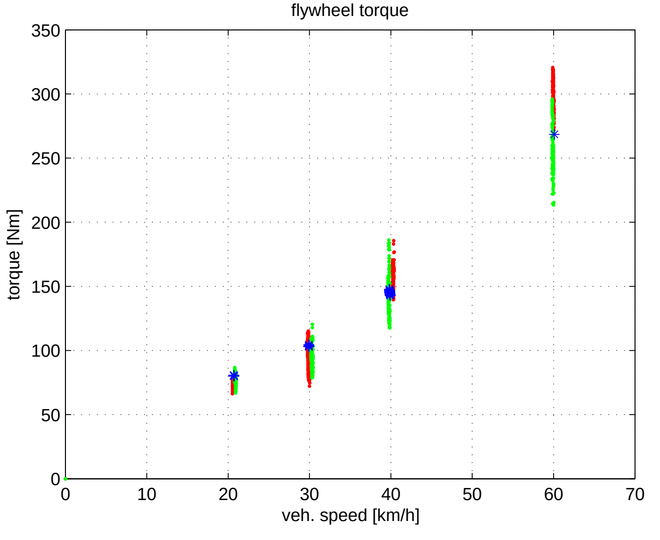

Figure 6.1: *Validation of steady state torques after model improvements. Simulated results (blue) and measured results from Björkvik. The engine speeds are the same which means that the power is equal at steady state speeds using the new gearbox model and the new tire model. There are some small deviations which might depend on model uncertainties. Possible other explanations are winds or gradients in the road. The ratio between fueling and torque is 0.2:1 which results in small errors even though deviations exists. The deviations in the measurements are very big and it would be desirable to collect data during a longer time to get more data which would give a better statistical basis.*

6.2 Acceleration and Gear Shifting

Even if the steady state torques agree well in measurements and simulations, the model's possibility to predict fuel consumption will be poor if the transient behavior of the model is badly modeled. Figure 6.2 shows that the vehicle speed increases more during gearshifts in the model than in the measurements. Calculations show that the sum of the kinetic energy of the powertrain and the bus is the same before and after a gearshift when the energy from the fuel during the 0.5 s long gearshift is subtracted. This effect can be felt in a bus but it can not be seen in the measurements. The gearbox is built up by hydraulic components. When gear shifting occurs, valves are opened which enables power to be transfered between two gears. It seems as the efficiency of the gearbox during the time the valves with different ratios are open is lower than when just one valve is open [1]. In Figure 6.3 the efficiency of the gearbox is set to 40% of its normal value during gearshifts which seems to capture the main effects. In the first figure, the set speeds are reached 5-10 m too early resulting in a decreased fuel consumption. The difference in fuel consumption is approximately 0.01 l/km. After model improvements this figure seems to be correct. From Scania's point of wiew, it is not interesting to model the gearbox efficiency during gear shifts more accurate than this.

It can also be seen that the gear shifting points are correct modeled. Deviations depends on acceleration, accelerator pedal position and speed. As seen in the figures, the amount of fuel that is injected during the gear shifts is correct too.

6.3 Accuracy of Fuel Prediction on SORT Cycles

SORT cycles have been driven at Idiada in Spain with a typical city bus powertrain. The simulations are performed using the same engine, speed profile and other figures as at Idiada. In section 5.3 it was discussed how the driver's behavior influences fuel consumption. Keep that in mind when studying the results from Idiada where the driver is having problems with keeping the accelerator pedal at a constant level. The driver in the simulations has been modeled to be as ideal as possible according to the SORT-standard. This means, it will keep the accelerator pedal fully pressed down until the demanded speed is reached. To fit the simulated speed profile to the measured ones, the mean speed of the speed profile is increased a little by adjusting the constant speed.

Since there are many parameters that can affect the results it has to be discussed how well the measurements can predict the final value. The driver can influence the fuel consumption a lot. There are three main results that can be seen in the figures.

[1] The manufacturer has been asked for information regarding this. As this is written no more facts have come to Scania's awareness.

- The driver who did the measurements at Idiada had problems with keeping the accelerator pedal position steady. The deviations also seems to be greater than one could expect from a driver driving on a flat road. This does probably increase the fuel consumption. (The consumption would decrease if the tests were performed as in the simulations)

- The driver often accelerates or decelerates during the constant speed phase. In mean, this behavior probably results in very small deviations from the real ”perfect” value [2].

- The driver releases the accelerator pedal a little earlier than the simulated driver. This means that the real driver uses the last 10-15 meters to coast down. It would result in an increased fuel consumption if the driver kept the accelerator pedal pressed down the last meters.

- Other things that could affect the result are wind and road effects. These parameters were perfect during the measurements according to the test protocol.

In mean, those effects are small and will only influence the consumption very little.

6.4 Results

The final results of fuel consumption from the simulations and Idiada are presented below, normalized with the simulated result from the SORT cycles which are named Γ_1 and Γ_2:

Drive cycle	**Simulated result**	**CAN integrated result**	**IDIADA measure**
SORT 1	Γ_1	$(0.988 \pm 0.020)\,\Gamma_1$	$0.984\,\Gamma_1$
SORT 2	Γ_2	$(1.022 \pm 0.025)\,\Gamma_2$	$102.5\,\Gamma_2$

The CAN integrated result is presented with a 95% confidence level since the calculations are made on all of the available Idiada measurements. The last result is the official result from Idiada which was measured using special equipment for such purposes.

If the ratio between the simulations and the CAN integrated result is calculated, using the value with a 95% level of confidence that gives the greatest deviation, the deviaitons between the measured and the simulated results are: 3% at the SORT 1 cycle and 5% at the SORT 2 cycle. Compared to the measured figures the deviations are 2 and 3 % respectively.

In general, the results show that the library can be used to do accurate predictions of fuel consumption, which lie within the variations of the real

[2]The results in section 5.3 indicate this result

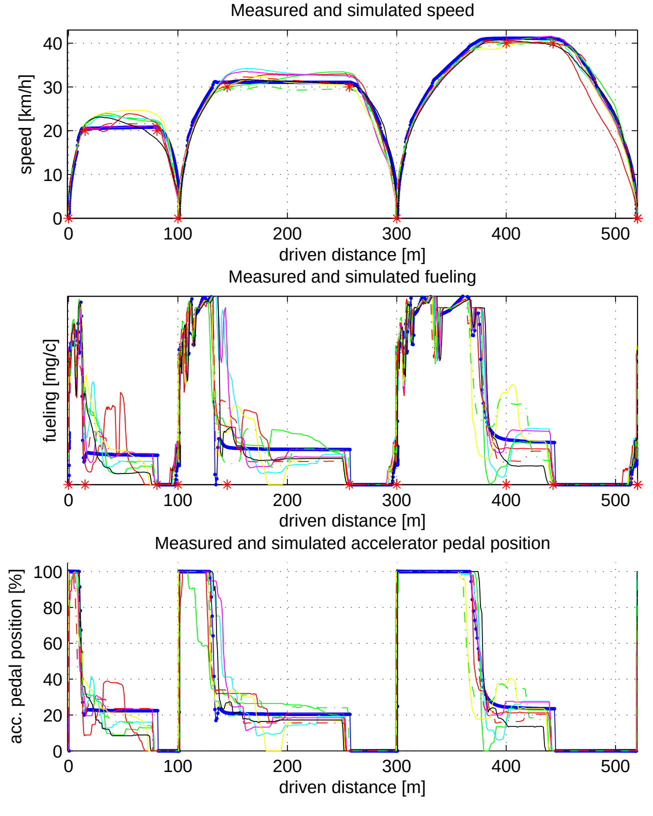

Figure 6.2: *Validation of the fuel consumption for a SORT 1 cycle after model improvements. The simulated result is the blue dotted line, all others are from Idiada. Red stars are cones which describe where to start each trapeze, where the speeds have to be reached and where to start braking.*

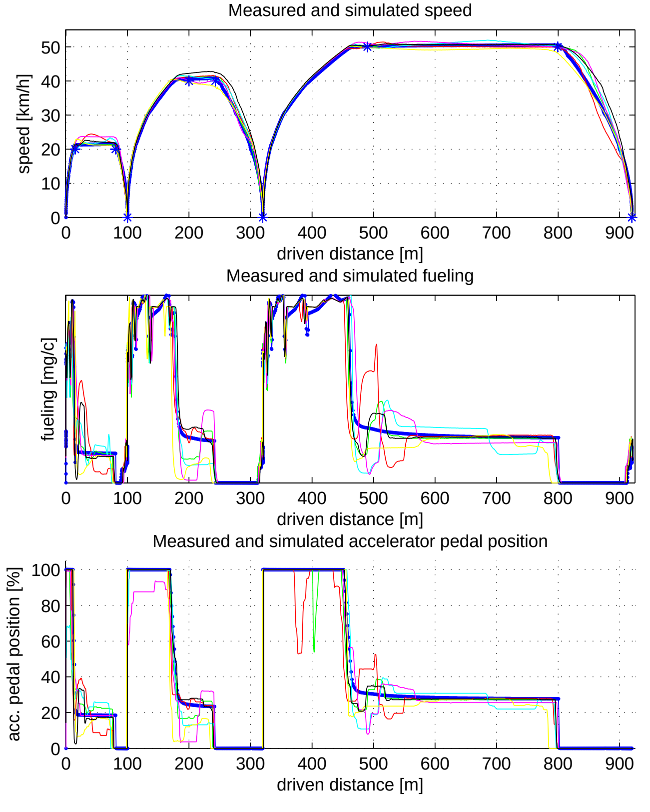

Figure 6.3: *Validation of the fuel consumption for a SORT 2 cycle after model improvements. The simulated result is the blue dotted line, all others are from Idiada. Blue stars are cones.*

driver [3] and as long as the simulated figures are within the ones a real driver causes, it must be seen as a very good result. Trucks and coaches can be equipped with cruise controllers which make the deviation between different drivers relatively small. In city bus traffic, the driver has to press the accelerator pedal which results in even greater deviations between different drivers.

Finally it can be established that the CAN signal describes the real consumption well for SORT cycles. The 2% deviations seen in [14] could not be reproduced in this work. It is probably easier to achieve good values at constant speeds compared to these transient cycles. Transient modeling also results in more simplifications and the deviations due to the speed profile are greater than in the steady state speed case.

[3] The maximum and minium integrated CAN values from Idiada are: SORT 1: $0.94\ \Gamma_1$, $1.02\ \Gamma_1$ SORT 2: $0.96\ \Gamma_2$, $1.03\ \Gamma_2$

Chapter 7

Future Work

The main achievement in this validation is that the model library is shown to be useful for accurate predictions of fuel consumption. The most useful and reliable figures are probably relative results. It is always risky to spread information regarding absolute fuel consumption. Figures and facts can rapidly change to hearsays where the background information is lost. Relative results will also probably still be correct even if the absolute consumption differs between a computational and a measured result since some of the errors are canceled.

Still, there exists some work that can be done:

The value of C_D and how it depends on vehicle speed for a typical city bus and a coach is of interest. If it would be possible to test a bus in a wind tunnel it would give a great input to the model. Perhaps is it enough and would give reliable results to simulate this dependency.

The value of C_{rr} should be further investigated. The simulations are performed with a wide range of loads and speeds. It has not been investigated how C_{rr} or the temperature depend on load if the load is varied a lot. If simplifications has to be made, the most reliable results would probably be reached by using steady state values for long haulage simulations and city bus traffic simulations with a value proportional to v^2. (4.6) could be used with a fix value of C_{r0} which depends on the mean speed of the cycle since the temperature would be constant.

This study has only dealt with fuel consumption issues. If the model in the future will be used for driveability analyses, other things have to be investigated eg. stiffnesses in the powertrain components which could induce driveline vibrations etc.

To achieve reliable results and a good understanding of the simulation results it is of greatest importance to validate new models. The developers of the model library should continuously compare simulation results and available data to get a good understanding of the library's possibility to predict fuel consumption and to find possible model errors. Finally it is a very powerful simulation tool but as with all models where simplifications have been made it can be misleading if the tool is used by persons who have no or little understanding of the model's area of validity. This understanding can only be obtained by using the tool as described above.

Chapter 8

Conclusions and Reflections

8.1 Conclusions

A model library consisting of bus specific powertrain components has been developed at Scania. The library is based on a work primary done for long haulage simulations of trucks and has been modified and developed to catch the transient behavior of vehicles. In this work it has been tried to figure out the bus specific library's ability to predict fuel consumption for the standardized SORT cycles.

In the first part of the thesis a theoretical analysis of the model at constant speeds on an even road is made. The analysis points out which parameters that result in the greatest influence on fuel consumption. It shows later in the report that this is a powerful tool to by hand easily calculate how the change of a parameter influences fuel consumption.

After the analysis, practical measurements is performed which points on some uncertainties in the powertrain components. Because of the way the measurements is carried out, it is not possible to directly determine from where the faults originates. By studying the gearbox model it is supposed that this model underestimates the losses at low loads. Low loads is common in city buses with high differential gear ratios and this model uncertainty is therefore crucial. It is assumed that the losses are linear in a new torque loss model. With the new model the efficiency of the gearbox drastically falls when the driving torque approaches the assumed internal friction torque named τ_{loss}. In the thesis it is discussed why this model probably describes the real losses of the gearbox better.

Coast down tests are also made. Those are used to compare the modeled traveling forces with the simulated ones. New models of air and rolling resistance are proposed.

In the comparisons with fuel consumption tests, the models ability to predict fuel consumption is investigated. Because of the real drivers big deviations from the speed profile, the mean values with confidence levels are used to compare the models ability to predict fuel consumption. By visual inspection of the graphs it is seen that the model catches transient behavior well. Finally it is established that the model library can be used to do precise fuel consumption analyses of SORT 1 and 2 cycles that results in less than 5% deviations from the measured values. Those results are better than the variations caused by the real driver when the same drive cycle is driven many times. The main uncertainties are powertrain losses, traveling frictions and difficulties in modeling the real driver when the comparisons between simulations and measurements are made.

8.2 The Author's own Reflections

The model is said to predict fuel consumption on SORT 1 and 2 cycles well. There are no results that contradicts the assumptions that the model will predict fuel consumption well on SORT 3 or other transient cycles as well.

Much of the deviations between simulations and measurements originate in difficulties describing a certain duty cycle or a certain driver's behavior. Even if a real and a simulated driver in mean have the same measured values eg. mean velocity or mean accelerator pedal position, an increased fuel consumption can certainly depend on traffic type etc. Effects like these can affect the simulation result a lot if the real driver sometimes operates the vehicle at operating points with poor efficiency but in mean in an operating point with good efficiency.

To only optimize a vehicle for good fuel economy will result in poor driveability and a good understanding of the driver's acceptance and expectations is therefore important. The simulated driver is not optimized for good fuel economy measures but only to follow the speed profile well. Big profits can therefore be made by teaching the real driver the principles of eco-driving.

The new model of C_D takes side winds into account. It has been shown that these effects are big and can not be neglected. When predicting fuel consumption it can be discussed whether the calculations should be performed at "ideal" conditions or not. The assumption that C_D is constant is a good assumption as long as there are no side winds but the result will not be reliable for real conditions in general.

There are also some arguments for using simulations instead of real measurements:

- The simulations will always give the same results since the ambient conditions are always the same. It can also take long time before the new ambient conditions can be achieved in a real test.

- The simulations are fast and cheap and no modifications have to be made on test vehicles.

- In measurements, a single person's behavior can affect the final result a lot. If two different tests are performed with different drivers the result will probably differ more than the deviations caused by one driver.

References

[1] ISO standard 18164:2005. www.iso.org.

[2] SAE standard 1252.

[3] SKF handbook. www.skf.com/portal/skf/home/products.

[4] Bruce R. Munson Donald F. Young and Theodore H. Okiishi. *A brief introduction to fluid dynamics.* 3 edition, 2004.

[5] M. Gamberg. Mätning av temperaturer i däck för verifiering av ny rullmotståndsmodell. Master's thesis MMK 2003:39 MPK 493, Institutionen för maskinkonstruktion, KTH, Stockholm, Sweden, 2003.

[6] Alexander Banerjee Jochen Köhler. Usage of modelica for transmission simulation in zf. www.Modelica.org/events/Conference2005/.

[7] Wong J.Y. *Theory of Ground Vehicles*. 3 edition, 2001.

[8] Mike Dempsey Michael Tiller, Paul Bowles. Development of a vehicle model architecture in modelica. pages 75–86, 2003. www.Modelica.org/Conference2003/papers.shtml.

[9] Pettersson N. Modeling and control of auxiliary loads in heavy vehicles, December 2004.

[10] Diesel net. www.dieselnet.com/standards/cycles/braunschweig.html.

[11] William L. Oberkampf and Matthew F. Barone. Measures of agreement between computation and experiment: Validation metrics. *Journal of Computational Physics*, March 2006.

[12] V A Petrushov. Improvement in vehicle aerodynamic drag and rolling resistance determination from coast-down tests. Technical report, The Central Scientific Research Automotive Engines Institute (NAMI), Moscow, Russia, 1998.

[13] STR. Ecodriving. www.str.se/Miljo/Ecodriving/Korsatt.

[14] Sandberg T. Heavy truck modeling for fuel consumption simulations and measurements, December 2001.

[15] L. Ljung T. Glad. *Reglerteori Flervariabla och olinjära metoder*. Studentlitteratur, 2 edition, 2003.

[16] L. Ljung T. McKelvey A.Stenman J. Löfborg T. Glad, S. Gunnarson. *Digital Styrning Kurskompendium*. Linus & Linnea, February 2003.

[17] UITP. SORT - Standardised On Road Test Cycles. www.uitp.org/publications/index2.cfm?id=1.

[18] ISY Vehicular systems. www.fs.isy.liu.se/Publications/mastersthesis.html.

[19] Han Zhixiong Ma Xiaoping Bi Shu and Xixia Liu. Transient modeling for heavy tracked vehicle performance and fuel consumption under accelerating conditions. *SAE Journal of Computational Physics*, March 2006.

Notation

Symbols used in the report

Variables, parameters and names

v	speed
$v_{km/h}$	speed in km/h
$v_{m/s}$	speed in m/s
r_w	wheel radius
ρ	density
C_D	air drag coefficient
C_{rr}	coefficient of rolling resistance
A	frontal area
μ	coefficient of friction
$_w$	wheel
$_{dg}$	differential gear
$_{gb}$	gearbox
$_{TC}$	transfer case
$_e$	engine
$_{tr}$	tractive
$_r$	rolling
$_{tot}$	total

Forces, torques and efficiencies

M_x	torque in powertrain component x
τ_x	torque load of auxiliary device x
F_x	force in component x
N	normal force
η_x	efficiency of component x
i_x	ratio of component x
ω	angular velocity

Appendix A

Signal Processing

The speed signal logged from the CAN-bus is noisy and has to be filtered.

This is made using a 3:rd order Butterworth filter implemented in Matlab with a normalized cut off frequency of 3/50 . The filter properties can be seen in Figure A.2. The used Matlab function is filtfilt which returns a zero phase filtered velocity signal with correct initial conditions.

The filtered velocity signal is then used to calculate the acceleration using (A.1). If 5 sec. is considered a long time, it should be set in relation to the time it takes to coast down from 50 km/h to 0 which is almost 2.5 minutes. It should also be mentioned that other ways of calculating the acceleration have been tested but it has been concluded that this method works well.

$$a(t) = \frac{v(t + 2.5) - v(t - 2.5)}{5} \tag{A.1}$$

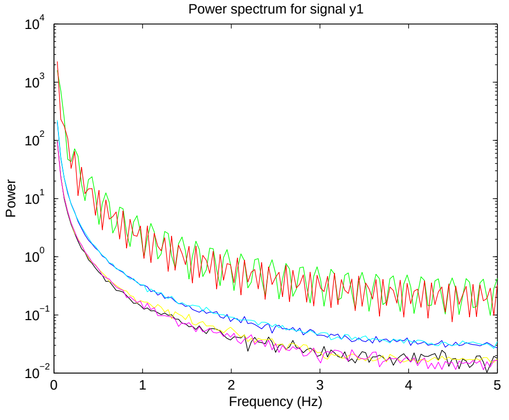

Figure A.1: *Spectral analysis of the velocity signals. Generally it can be said that the signals logged at high velocities contain more energy.*

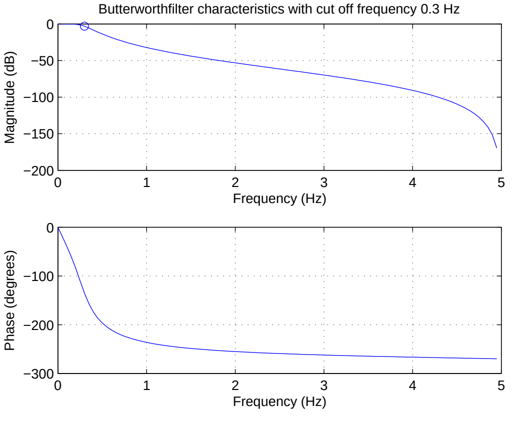

Figure A.2: *The characteristics of the filter used to filter the velocity signal. The cut off frequency has been marked with a ring (-3dB)*

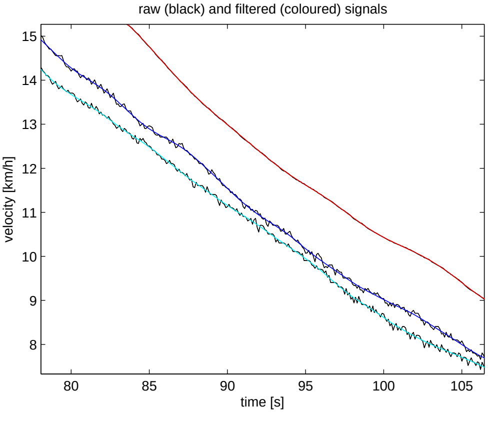

Figure A.3: *The non filtered (black) and the filtered signals (coloured). This is a zoomed variant of the plot seen in Figure 3.1.*

Appendix B

Model Changes

This appendix deals with the model changes made during this work.

B.1 Gearbox

Instead of extrapolating the efficiency from the data given by the manufacturer which results in gearbox efficiencies above 90% at all driving cases, an internal friction called τ_{loss} is calculated. When studying τ_{loss} it seems as the losses are linear with respect to torque. Therefore, the extrapolation gives more reliable results if the extrapolation is made with respect to this quantity. With this model, the efficiency of the gearbox is zero at $M_{flywheel} = \tau_{loss}$ which is approximately 25 Nm at low engine revs.

An additional gearchange fix is made. The efficiency of the gearbox is set to 40% of the normal efficiency during gearshifts. The automatic gearbox is a hydraulic gearbox and the fix seems to capture the main effects of the gearchanges.

In an e-mail sent to the manufacturer of the gearbox, the uncertainties were pointed out. New figures will not be sent to Scania within the timescale of this thesis.

The advantages of the new gearbox model are:

- Power is lost even if the power into the gearbox is zero.
- The gearbox has lower efficiency at low load.
- The loss model is technically intuitive.

B.2 Differential Gear

The inputs to the loss model had been mixed up in the previous model. The torque and the speed have now been changed.

B.3 Traveling Resistances

The new traveling resistance gives correct driving torques. Works have been initialized to figure out how the air drag coefficient depends on vehicle speed for different buses.

B.4 Driver

The new driver's behavior is distance dependent instead of time dependent as before. This leads to equally driven distances even if different buses are driven. This makes the l/km figure more comparable. The new driver also handles steps without controller wind up.