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Fuel and ride comfort optimization in heavy vehicles

Master's thesis performed in Vehicular Systems

by

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Titel
Title

Bränsle- och komfortoptimering i tunga fordon

Fuel and ride comfort optimization in heavy vehicles

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Sammanfattning
Abstract

 In modern heavy vehicles low fuel consumption as well as good ride comfort and driveability is desired. Assuming that the road altitude ahead of the vehicle is known the optimal control regarding fuel and time consumption can be calculated. However this results in a bang-singular-bang control which decreases the ride comfort by introducing high jerk levels and oscillations in acceleration as well as jerk originating from the dynamics in the driveline.

 In this thesis several methods to supress these behaviours are presented. A qualitative study of the methods impact on ride comfort as well as fuel and time consumption. A driveline model is implemented in Simulink and used for the evaluations. The aim is not to find a optimal strategy but rather to suggest methods and evaluate these as far as can be done in simulation to enable for future test runs.

Abstract

 In modern heavy vehicles low fuel consumption as well as good ride comfort and driveability is desired. Assuming that the road altitude ahead of the vehicle is known the optimal control regarding fuel and time consumption can be calculated. However this results in a bang-singular-bang control which decreases the ride comfort by introducing high jerk levels and oscillations in acceleration as well as jerk originating from the dynamics in the driveline.

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Acknowledgments

Contents

List of Tables

List of Figures

Chapter 1

Introduction

1.1   Background

Due to environmental concerns, customer demands for low cost transportation, and political efforts to reduce dependence upon oil supplying nations fuel efficiency has become a top priority in vehicle development. The average annual mileage of a European class 8 truck is 150,000 km with an average fuel consumption of 32.5 l/100 km [1]. This translates into approximately 30 % of the trucks life cycle cost being the cost of fuel. With this in mind, the gain in reducing the fuel consumption even slightly is obvious.

Dynamic model-based optimization with look-ahead based on altitude data, speed limits, traffic situations etc. is a quite new approach to achieve this but it's already showing a big potential. This method has many advantages. The principle of the approach is to, instead of using a classical PID-based cruise control (CC), calculate an optimal control signal given a vehicle model and information about some of the future disturbances, e.g. road altitude data. This means that the method can be implemented on any vehicle, in development as well as in production.

The solutions to these optimization problems however often result in a control strategy with bang-singular-bang characteristics which, due to high jerk[1] levels, might result in poor ride comfort and driveability. Since driveability and ride experience among with economy and performance are key issues when investing in a vehicle, being able to model and simulate these characteristics are vital in order to be able to incorporate comfort aspects in the fuel and time optimization routines.

The experience of ride comfort and driveability are by definition subjective and therefore they can't be evaluated strictly by simulations. However the magnitude of jerk, driveline oscillations, etc. can be, provided a suitable model is used.

The optimization algorithm is developed to be used in an on-board CC and therefore short calculation times are vital. To achieve this the algorithm is based

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[1]Jerk, denoted by j, is the first derivative of acceleration, unit m/s^3 [2].

on a stiff model of the driveline. However the cause of jerk often lies in the dynamic characteristics of the driveline and thus can't be captured by the stiff model. Subsequentially a more complex dynamic driveline model is necessary in order to be able to evaluate the jerk trough simulations.

1.2 Objective

The objective of this thesis is to suggest and evaluate methods aiming towards limiting the origin of jerk in the pre-existing optimization algorithm presented in [3]. With a major cause of jerk being the dynamic behaviors of the driveline, a dynamic model presented in [4] will be implemented in Simulink and used for evaluation.

In addition to evaluating the different methods impacts on the magnitude of jerk and driveline oscillation, the resulting deviation in cost from the optimal solution are studied.

1.3 Assumptions and Limitations

The origins of jerk and discomfort in a vehicle are many. Being able to simulate all these behaviors requires highly accurate and complex models of the driveline dynamics, vehicle chassis, tires, road surface, drive mission and so forth. This thesis will only focus on discomfort caused by longitudinal jerk originating from the bang-bang regulation and the dynamics of the driveline. Jerk and driveline oscillations caused by gear shifts will not be considered since this is handled by the control algorithms of the automated gear shifting and not directly by the optimization algorithm.

This thesis does not focus on finding the optimal method or parameters for increasing the ride comfort, but rather to qualitatively evaluate the potential in different methods and their impacts on the systems behavior. Findings that later can be used to implement a test rig in an actual vehicle in order to fine tune the parameters and select what methods to use.

1.4 Outline

This thesis is divided into seven chapters. This introductory chapter is followed by a brief summary of previous work carried out in this field. In chapter 3 the dynamic vehicle model used for evaluation is described and this is followed of a brief description of the optimization algorithm in chapter 4. Three different methods for increased ride comfort are proposed in chapter 5. These are then simulated and the impacts on ride comfort as well as cost deviations are presented in chapter 6. The thesis is ended with conclusions and proposals of future work in chapter 7.

Chapter 2

Previous work

2.1 Driveline modeling

Extensive work has been done in order to model and simulate driveline dynamics. Obviously a wide array of complexities exists among these findings, however a common conclusion is that a third order model consisting of two inertias connected by a flexibility is enough to capture the main characteristics of the driveline in terms of oscillations [4].

A model described in [5] based on [4] will be implemented in Simulink and used to evaluate the results from the optimization algorithm. The model and it's utilization will be further described in chapter 3.2.

2.2 Ride comfort and driveability evaluation

By definition, ride comfort and driveability are highly subjective experiences, depending on several different stimuli applied to the human body. Among these are acceleration, jerk, vibrations, seat comfort, noise levels etc. Some studies, aiming towards finding objective metrics of the experience of ride comfort and driveability, has been carried out. Focusing on the longitudinal behavior of the vehicle, the consensus of these are that high levels of acceleration and jerk as well as oscillations in these has a significant negative effect on the ride experience. It is suggested that the magnitude of the jerk affects the experience of acceleration [6]. Oscillations has several complex impacts on the human body from causing slight discomfort to severe nausea [7].

Considering the power-to-weight ratio of a heavy long haulage truck the magnitude of acceleration is highly unlikely to affect the comfort negatively, the exception possibly being emergency braking.

2.3   Comfort and driveability improvement

Classic CC systems are often based on some sort of PID-regulator which, unlike the optimization routine described in chapter 4, does not generate control signals of bang-singular-bang characteristics. Therefore no need of jerk limitation is necessary in these applications. Driving with the CC disabled a robust and simple but yet effective method of jerk limitation is to simply send the drivers input from the accelerator pedal through a low pass filter (LP filter) before allowing it to affect the engine.

    A different, more complex, approach to improve the ride comfort is to actively reduce the oscillations in the wheel speed caused by the driveline dynamics in tip in/out maneuvers. This can be done by engine controlled damping of driveline resonances. An example of this is presented in [8], where a state-space description of the driveline is used to construct a state-feedback controller.

Chapter 3

Vehicle model

The model capturing the vehicles longitudinal behavior consist of two main components, the engine and the driveline, further described in this chapter. Both components are implemented in a Simulink model and used to evaluate the different methods of jerk suppression.

3.1   Engine

To find the fueling level of the ICE corresponding to the flywheel torque requested by the CC, given the current engine speed, a measured engine map is used. In addition to the flywheel torque, the engine map holds information such as maximum engine speed, maximum fueling level and the engines internal friction torque.

3.2   Dynamic driveline model

To capture the behavior of the driveline, see figure 3.1 and 3.2, a dynamic model is needed. In [4] two models are presented, see figure 3.3; the two inertia model where the drive shafts are considered the only flexible components and the three inertia model where both the drive shafts and the propeller shaft are considered flexible.

In the two inertia model the driveline is modeled as a system of two inertias connected by a rotational spring and damper. The first inertia, J_1, represents the inertia caused by the engine, transmission and final drive lumped together. This is motivated by the stiff clutch and propeller shaft. The second inertia, J_2, represents the inertia caused by the wheels and the vehicle mass. The rotational spring and damper represents the weaknesses and damping of the drive shafts.

In the three inertia model J_1 represents the inertia caused by the engine and transmission, J_2 represents the inertia of the final drive and J_3 represents the inertia of the wheels and vehicle mass. The rotational springs and dampers represents the weaknesses and damping of, respectively, the propeller shaft and the drive shafts.
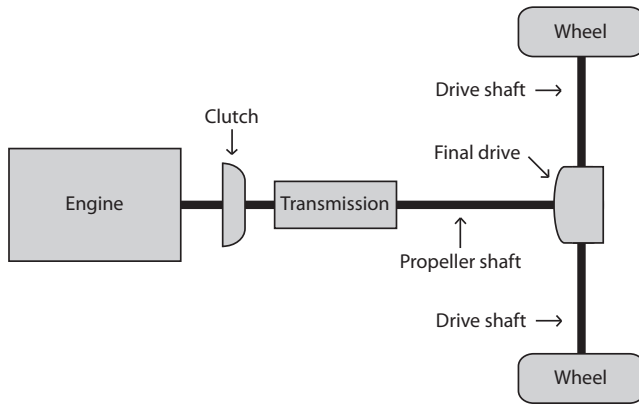
Figure 3.1. The driveline.
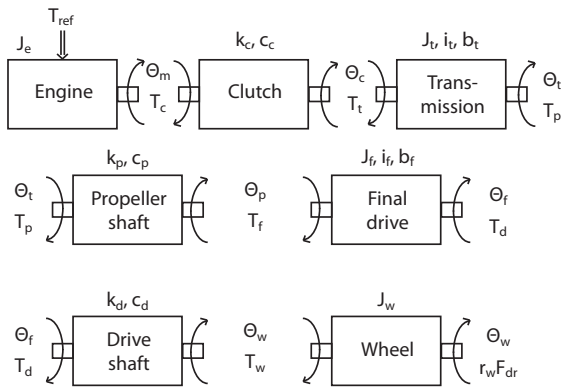
Figure 3.2. The driveline.
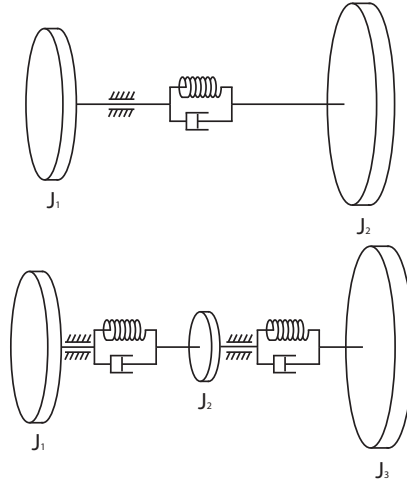
Figure 3.3. The two and three inertia models.

According to [5], the three inertia model offers no additional accuracy compared to the simpler two inertia model and therefore the latter is selected.

3.2.1 Model limitations

As mentioned above the two inertia model only considers the flexibility in the drive shafts. The clutch, propeller shaft and wheels are considered stiff. Gear shifts and reverse driving are not considered and are subsequentially not taken into account in the Simulink implementation.

3.2.2 Model equations

$$T_{in} = f(T_{ref}, i_t \dot{\theta}_t) \tag{3.1}$$

$$J_1 \ddot{\theta}_t = i_t T_{in} - b_t \dot{\theta}_t - \frac{T_d}{i_f} \tag{3.2}$$

$$T_d = T_w = k(\frac{\theta_t}{i_f} - \theta_w) + c(\frac{\dot{\theta}_t}{i_f} - \dot{\theta}_w) \tag{3.3}$$

$$J_2 \ddot{\theta}_w = T_w - T_{dr} \tag{3.4}$$

where

$$J_1 = J_e i_t^2 + J_t + \frac{J_f}{i_f^2}$$

$$J_2 = J_w + m r_w^2$$

and the rest of the notations as in table 3.1.

With the flywheel torque, T_{in}, and the driving torque, T_{dr}, seen as inputs, the system can be presented in the state-space form below:

$$\dot{x} = Ax + Bu \tag{3.5}$$

where

$$A = \begin{pmatrix} -\frac{\frac{c}{i_f^2} + b_t}{J_1} & \frac{-k}{J_1 i_f} & \frac{c}{J_1 i_f} \\ \frac{1}{i_f} & 0 & -1 \\ \frac{c}{J_2 i_f} & \frac{k}{J_2} & \frac{-c}{J_2} \end{pmatrix}$$

$$B = \begin{pmatrix} \frac{i_t}{J_1} & 0 \\ 0 & 0 \\ 0 & \frac{-1}{J_2} \end{pmatrix}$$

and the states and input signals

$$x = \begin{pmatrix} \dot{\theta}_t \\ \frac{\theta_t}{i_f} - \theta_w \\ \dot{\theta}_w \end{pmatrix}$$

$$u = \begin{pmatrix} T_{in} \\ T_{dr} \end{pmatrix}$$

3.2.3   Notations

See table 3.1 for the notations used in the model equations.

Parameter	Description
J_e	Engine moment of inertia
T_{in}	Flywheel torque
J_t	Transmission total moment of inertia
θ_t	Transmission angle
i_t	Gear conversion ratio
b_t	Total transmission friction
J_f	Moment of inertia of final drive
i_f	Final drive conversion ratio
k	Stiffness coefficient of the driveline
c	Damping coefficient of the driveline
T_d	Shaft torque acting on transmission
T_w	Shaft torque acting on wheel

Table 3.1. Notations used in the model equations.

3.2.4   Input signals

The first input signal, u_1, is the flywheel torque acquired from an engine map with the engine speed, calculated from x_1, and the requested fueling level from the optimization algorithm as inputs. The second input signal, u_2, is the torque generated by the resisting force acting upon the vehicle due to air and rolling resistance, road angle and the vehicle inertia. The braking force generated by the vehicles brakes are also included in the term u_2.

3.3   Calculating the jerk

Since jerk is the first time derivate of the acceleration the jerk can be calculated from the state x as

$$j = \frac{1}{m}(\dot{T_d} - \dot{T_{dr}}) \tag{3.6}$$

where

$$T_d = \frac{k}{r_w}x_2 + \frac{c}{r_w}(\frac{x_1}{i_f} - x_3) \Rightarrow$$

$$\dot{T_d} = \frac{k}{r_w}\dot{x}_2 + \frac{c}{r_w}(\frac{\dot{x}_1}{i_f} - \dot{x}_3)$$

and

$$T_{dr} = r_w \cdot \frac{\rho_a}{2} \cdot c_w \cdot A_a \cdot v^2 \Rightarrow$$

$$\dot{T_{dr}} = r_w \cdot \rho_a \cdot c_w \cdot A_a \cdot v \cdot a$$

where

$$v = r_w \cdot x_3$$

$$a = r_w \cdot \dot{x}_3$$

3.4   Determining the systems eigenfrequency

Due to the flexibilities in the drive shafts the risk of resonance is obvious. In order for the CC to be able to avoid causing resonance it's necessary to know the eigenfrequencies of the system. The eigenfrequencies corresponds to the imaginary components of the poles of the linear system in section 3.2.2. With the state-space representation of the model, see equation 3.5, the poles corresponds to the eigenvalues of the A matrix. In 12th gear the imaginary components of the eigenvalues are approximately 33.86, -33.86 and 0. Thus has the system an eigenfrequency of 33.86 rad/s or 5.39 Hz in 12th gear.

Experimentally the eigenfrequency can be determined by presenting the system with a step in the ICE's fueling level. In 12th gear this results in the frequency spectra seen in figure 3.4. The plot reveals a significant peak around 5.4 Hz representing the eigenfrequency of the system at the selected gear.
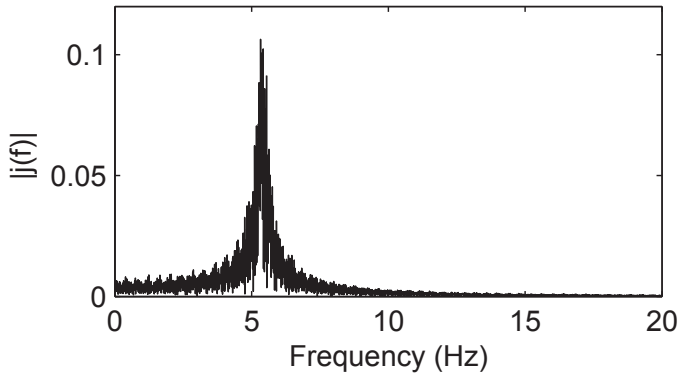
Figure 3.4. Single sided amplitude spectrum of the jerk at 12th gear.

Chapter 4

Optimization algorithm

The optimization algorithm with look-ahead control used for calculating the optimal speed trajectory and gear selections for a given stretch of road as presented in [9].

4.1 Basic principle

The basic principle of the algorithm is to minimize the fuel M and the time T required for a drive mission. The algorithm is designed to control accelerator, brake and gear shift. Spatial coordinates are used to formulate the vehicle model. The model is discretized and dynamic programming (DP) is used for the optimization. Changing conditions during the drive mission due to disturbances, e.g., delays due to traffic or changed parameters such as the vehicle mass means that new optimal solutions must be computed during the drive mission. This is handled by only concidering a truncated horizon in each optimization. By doing so an approximate solution to the global optimization is generated where the accuracy depends of the length of the truncated horizon.

4.2 Look-ahead control

In the predictive control strategy called look-ahead control, knowledge about some of the future disturbances are assumed to be available. In this case the additional information includes the road topography ahead of the vehicle. By predicting the systems behavior, given this topograpyh knowledge, an optimal speed trajectory can be calculated using DP.

4.2.1 Discretization

As mentioned above the models in the algorithm are discretized in order to obtain a discrete process model. The distance of the entire drive mission is divided into M steps.

4.2.2 Receding horizon

By truncating i the entire drive mission of M steps to $N < M$ steps a look-ahead horizon is achieved.

4.2.3 Dynamic Programming Algorithm

The travel cost, J, is calculated by

$$J = M + \beta T \tag{4.1}$$

where β is used to decide the trade-of between trip time and fuel consumption and the objective of the algorithm is to minimize J.

4.3 Vehicle model

With constant gear number, i.e., between gear shifts, the vehicle acceleration is given by

$$\frac{dv}{ds} = f(s, v, g, u) \tag{4.2}$$

where s is position, v is velocity, u are the control signals and g is the gear number. The fuel mass flow is given by

$$\dot{m} = h(v, g, u) \tag{4.3}$$

and the accumulated consumption is obtained by integrating the flow.

4.3.1 Longitudinal model

The longitudinal model of the truck is similar to the one described in section 3. The main difference is that the drive shafts here being considered stiff. The engine torque T_e is given by

$$T_e = f_e(\omega_e, u_f) \tag{4.4}$$

where ω_e is the engine speed and u_f is the fueling control signal. The function f_e is a lookup table originating from measurements. The entire driveline is considered stiff and the resulting conversion ratio of the transmission and final drive $i(g)$ and their efficiency $\eta(g)$ are functions of the engaged gear number, denoted by g. The models of the resisting force consist of three parts, the air drag given by

$$F_a(v) = \frac{1}{2} c_w A_a \rho_a v^2 \tag{4.5}$$

the rolling resistance given by

$$F_r(\alpha) = mg_0 c_r cos(\alpha) \tag{4.6}$$

and the gravitational force given by

$$F_N(\alpha) = mg_0 sin(\alpha) \tag{4.7}$$

With the effective wheel radius denoted r_w

$$v = r_w \omega_w = \frac{r_w}{i(g)} \omega_e \tag{4.8}$$

is assumed to hold. When a gear is engaged this gives

$$\frac{dv}{dt} = v\frac{dv}{ds} = vf(s,v,g,u) = \frac{r_w}{J_l + mr_w^2 + \eta(g)i(g)^2 J_e} \tag{4.9}$$

$$(i(g)\eta(g)T_e(v,u_f) - T_b(u_b) - r_w(F_a(v) + F_r(\alpha) + F_l(\alpha))) \tag{4.10}$$

The states are the velocity v and the currently engaged gear g and the control signals are fueling u_f, braking u_b and gear u_g.

4.3.2 Fuel consumption

The mass flow of fuel, denoted $\dot{m}$ is given by

$$\dot{m} = h(v,g,u) = \frac{n_{cyl}}{2\pi n_r} \omega_e u_f = \frac{n_{cyl}}{2\pi n_r} \frac{i(g)}{r_w} v u_f \tag{4.11}$$

where u_f is the fueling level, ω_e is the engine speed, n_{cyl} is the number of cylinders and n_r is the number of engine revolutions per cycle.

4.4 Ride comfort in current algorithm

In the measured data from the tests carried out in [3], no high jerk levels are evident even though the control signal has a clear bang-bang characteristic. This is most likely a result of the control signal from the optimization being low pass filtered before affecting the engine. In the tests this was unavoidable due to the set-up where the optimal solution was fed as a reference speed for the pre-existing on board cruise control and not allowed to directly control the engine. It is however desirable to let the optimization result be fed directly to the engine in order to ensure a truly optimal control strategy. Simulations on the driveline model shows that this would result in significant jerk levels and therefor a comfort criteria would have to be incorporated in to the optimization algorithm in order to ensure a smooth and comfortable ride.

Chapter 5

Potential methods for increasing the ride comfort

In this chapter three different methods for increasing the ride comfort and their implementation will be discussed. The different methods will be applied to the model and simulated in order to evaluate their influence on the jerk levels as well as the time and fuel consumption. The results of these simulations are presented and discussed in chapter 6.

5.1   Different approaches

5.1.1   Active damping

One efficient way to increase the ride comfort is active damping, mentioned in section 2.3. However, given the complexity of the optimization algorithm, evaluating the controller in every iteration, which is necessary in order to calculate the fuel consumption, would be much too time consuming. Therefore this approach would require a significantly simpler control strategy, not dependent on sensor feedback.

Given that the characteristics of the driveline are known, simply choosing a suitable delay, Δ_t, between the steps in the control signal could be a simple but yet effective way to dampen oscillating tendencies and thereby improve the ride experience. The idea is that instead of presenting the system with one large step in the input signal, causing oscillations in the drive shafts, the change of input will be done in two steps, see figure 5.1. The first step, just like the original step, will induce oscillations in the shafts and the second step is applied to cancel them out. To do this the second step needs to be applied when the force oscillation reaches its minimum, which should be at the shafts torsional equilibrium, i.e., after about half the period time of the systems eigienfrequency, see section 3.4.

However it is reasonable to believe that the characteristics and thereby eigen-frequency of the driveline are likely to change somewhat over it's lifetime or even over a drive mission due to changes in temperature, long time wear in bearings

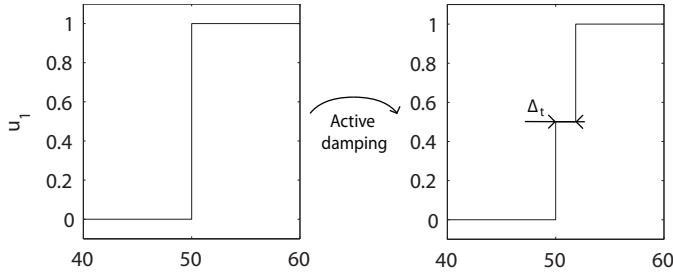etcetera. The possibilities of actually utilizing this method in a real system can therefore be debated.

Figure 5.1. Illustration of the active damping approach.

5.1.2 Limiting the rate of changes in u_1

The current method to ensure a smooth ride is to simply low pass (LP) filter the control signal before it affects the fuel injection, see figure 5.2. As mentioned in the previous chapter, the test runs carried out in [3] has proven this to be an efficient approach to reduce the jerk levels.

The current optimization algorithm actually include a penalty for the absolute value of the vehicle acceleration, $\left| \frac{du_1}{dt} \right|$. However the algorithm generates a control signal using a position grid with a resolution of 50 meters, giving a update frequency of 0.5 Hz at 90 km/h. It's impact on jerk levels can therefore be neglected and the term is only used to give a more stable velocity profile. In order for the derivate limitation to have any effect on the jerk levels, a significantly shorter interval between the set points is required, i.e., a finer position grid would be needed.
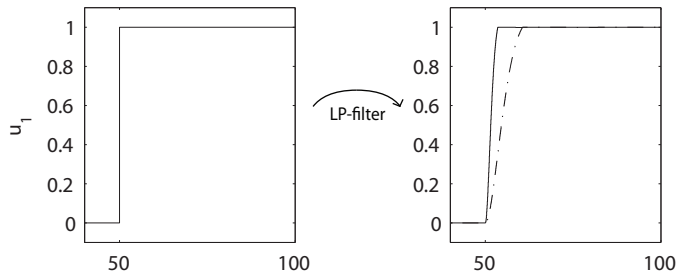

Figure 5.2. The effect of LP-filtering with two different cutoff frequencies on u_1.
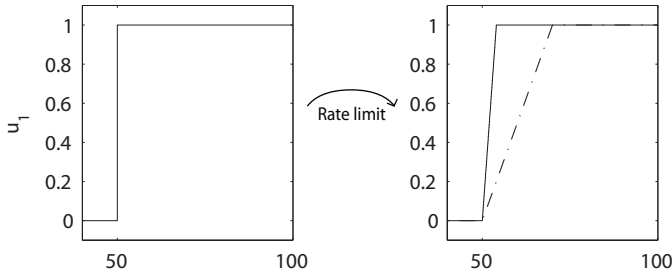
Figure 5.3. The effect of limiting $\left| \frac{du_1}{dt} \right|$ to, respectively, 1 and 5 units/s on u_1.

5.1.3   Interpolation

Interpolating the discrete steps in the control signal is another approach. A linear interpolation with a maximum slope could be an alternative to the method discussed in section 5.1.2 above, i.e., limiting the slope of the signal, while cubic interpolation could be another.

Cubic interpolation still results in small discrete steps in the signal but with a fine interpolation interval the characteristics of u_1 after interpolation is similar to the characteristics after LP-filtering, compare figure 5.4 and figure 5.2. Figure 5.4 shows the characteristics of the control signal after a fine and course cubic interpolation, respectively.
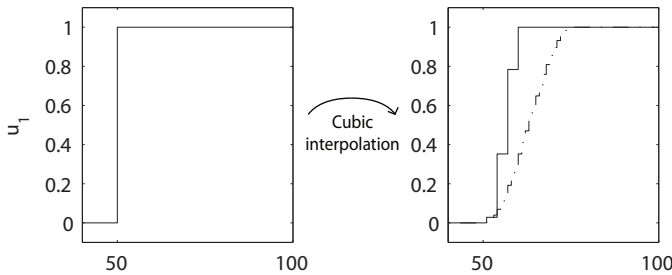

Figure 5.4. The effect of coarse and fine cubic interpolation on u_1.

5.2   Differences in implementation

The implementation of the methods above can be done in different ways. One way is to simply apply the strategies directly to the optimal solution that's been calculated by the algorithm, i.e., post-process the optimal solution. This means that the optimization algorithm can't affect the amount of smoothing but the change in time and fuel consumption would be considered by including this to the optimizations vehicle model, i.e., the algorithm still produces the optimal control signal

given the changed conditions. As discussed above, in section 5.1.2, another way is to implement the penalties or constraints directly in the optimization. Allowing for the optimization to affect the level of smoothing, comparing this against the traveling cost.

5.3   Evaluating the methods

The first property of the methods to be evaluated is whether it has the desired effect on the jerk and oscillations or not. This evaluation does not require the post-processing methods to be incorporated in to the optimization, only applied on the control signal before simulating the vehicle. Thereafter one can proceed to evaluate the deviation from the optimal time and fuel consumption.

Chapter 6

Simulation results

In this chapter the simulation results are presented. Every method is simulated with a set of different parameters in order to give a good indication of it's potential. In addition to the impacts on jerk levels and oscillations, the impacts on the travel costs are evaluated. This is done by comparing the simulated total cost to the cost of the optimal solution, see equation 4.1.

The road profiles used in the simulations are measured altitude data from several road segments of highway E4 between the cities of Södertälje and Norrköping in Sweden, see Appendix A. In the evaluation of ride comfort improvement only the simulation results necessary to capture the characteristics are presented. In the evaluation of cost deviation however, the presented results are an average of simulations done over all the road profiles.

The systems potential energy is not the same in the final state as in the initial state, neither is the momentum. The latter is due to the fact that the control signal is calculated by the optimization algorithm based on the stiff driveline model. The behavior of this model is not identical with the behavior of the dynamic evaluation model. This deviation can be corrected by adding a feedback controller based on the speed error to the control signal generated. In this qualitative study however, preserving the control signals bang-singular-bang characteristics are higher appraised and therefore this method is not used. The comparison of costs (i.e. the time and fuel consumption) is done based on the cost of a reference simulation of the dynamic driveline model with no manipulations done to the control signal and thus the comparison is still valid. As for the former, simulations has also been carried out over symmetrical road segments to circumvent this although these simulations did not result in any significant changes of the cost deviation.

6.1 Simulation with no comfort improvement

For reference the model is simulated with the unmodified control signal generated by the algorithm. As mentioned above, simulations and comparisons has been done over several stretches of road. However the simulation results over the northbound Järna segment, see figure 6.1 and 6.2, is enough to capture all the essential behaviors. Hence only simulations of this road segment will be presented in this report.

Figure 6.1 shows the entire simulation, whilst figure 6.2 shows an enlargement of the tip-in and tip-out maneuvers at, respectively, 150 and 1250 meters. High peak to peak jerk levels as well as extensive oscillations are evident.
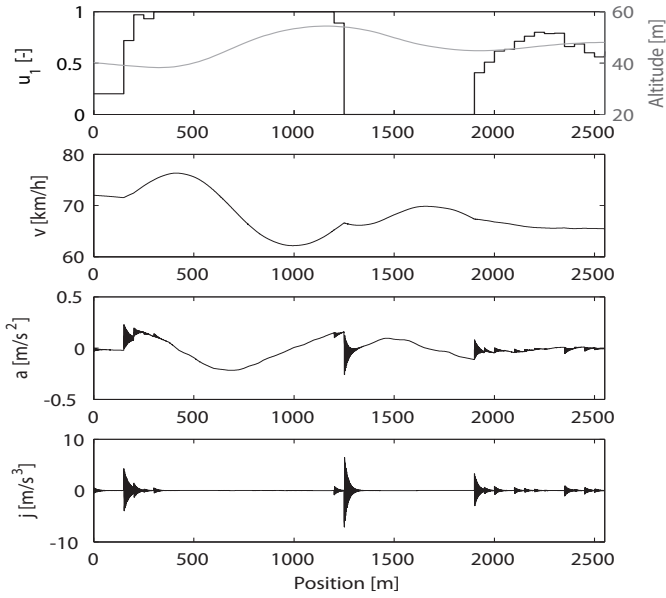
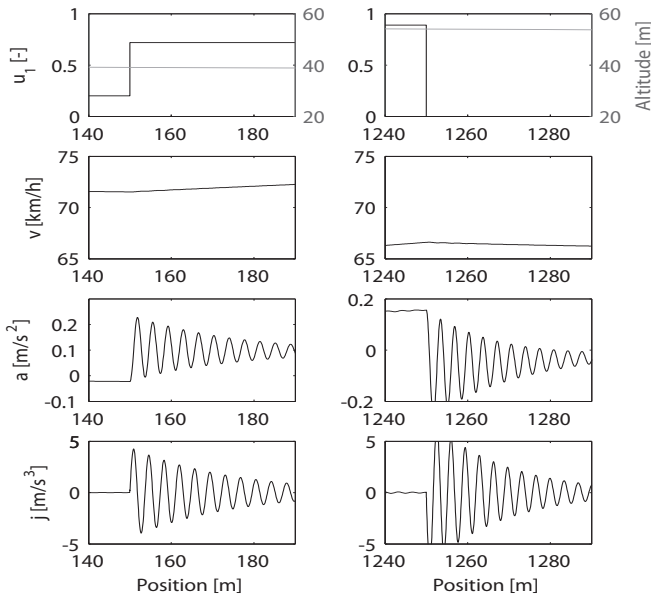**Figure 6.1.** Simulation result with no comfort improvement.

Figure 6.2. Simulation result with no comfort improvement.

6.2 Simulation with active damping

The simulation result of the active damping approach, described in section 5.1.1, is presented below in figure 6.3, figure 6.4 and table 6.1.

6.2.1 Impact on the ride comfort

The comfort effects of implementing the active damping described in section 5.1.1 and applying it to the optimal control signal are shown in figure 6.3 and figure 6.4. As predicted, simulations shows drastic decrease in the peak magnitude of the jerk as well as significant suppression of oscillations in jerk as well as acceleration suggesting a considerable comfort improvement.

In the tip-out maneuver at 1250 meters, see figure 6.4, the method's Achilles' heel can be seen. The time interval, Δ_t, is slightly too long resulting in significantly decreased suppression of oscillations in both jerk and acceleration. However compared to the reference simulation, figure 6.2, the improvement is still obvious.

6.2.2 Impact on the cost function

As can be seen in table 6.1, in addition to the increased ride comfort, th average trip cost actually *decreases* by approximately 0.02%. This can be explained by three factors; The optimization algorithm generates an approximation of the optimal solution, this means that a slightly better solution may exist. The optimization is done over a stiff driveline but in the dynamic driveline a fraction of the generated work is consumed in the damped oscillations. The driveline model used for optimization is not identical with the driveline model used for evaluation, this means that the optimal solution for the optimization algorithm's model might deviate slightly from the evaluation model's optimal control signal.

	Δ_{time}	Δ_{fuel}	$\boldsymbol{\Delta_{cost}}$
Active damping	0.0271%	-0.0526%	$\boldsymbol{-0.0229\%}$

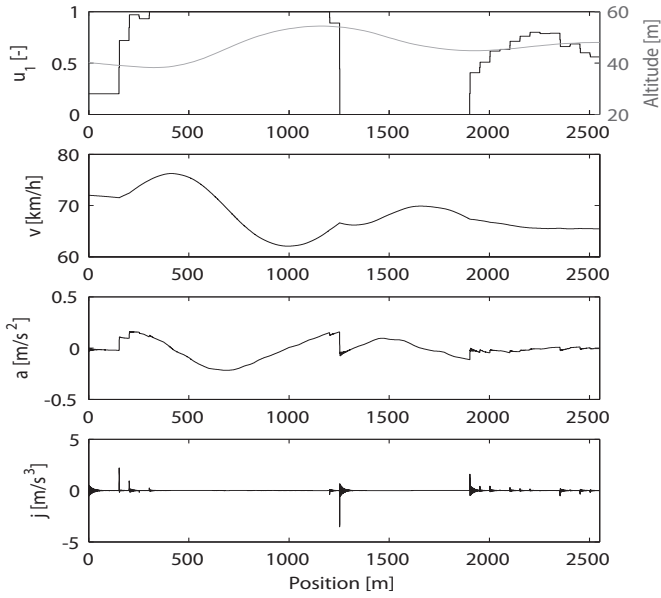**Table 6.1.** Deviation from the optimal solution using active damping.

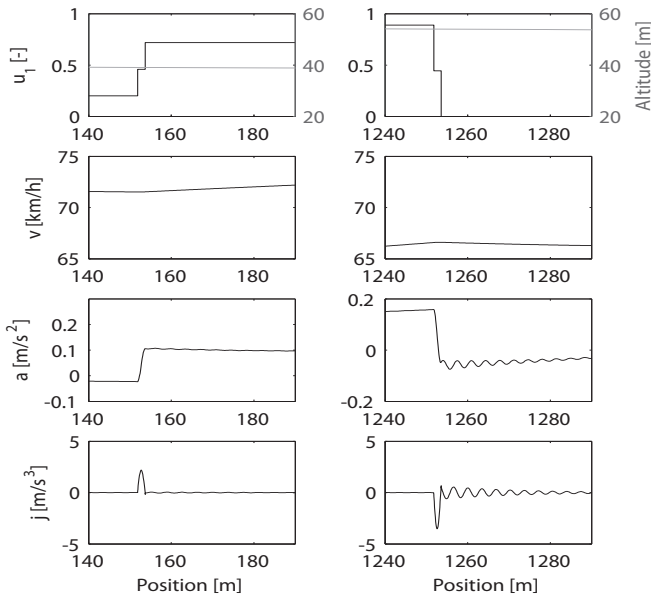**Figure 6.3.** Simulation result using active damping.

Figure 6.4. Simulation result using active damping.

6.3   Simulation with rate limitation

The rate limit method is simulated with three different rate limits, $\left|\frac{du_1}{dt}\right| \leq 0.5, 1.0$ and 3.0, represented by, respectively, black, dark gray and light gray in figure 6.5 and figure 6.6.

6.3.1   Impact on the ride comfort

Studying figure 6.6, the peak to peak jerk level is significantly smaller than for the reference simulation. Considerable oscillations, however, still are induced.

6.3.2   Impact on the cost function

As can be seen in table 6.2, the cost deviation decreases with increasing rate limit. Note that a higher rate limit results in a smaller manipulation of the optimal solution, hence the cost deviation decreases with increasing limit.

$\left\|\frac{du_1}{dt}\right\|$	Δ_{time}	Δ_{fuel}	$\boldsymbol{\Delta_{cost}}$
≤ 3.0	-0.0308%	0.0578%	$\mathbf{0.0247\%}$
≤ 1.0	-0.0901%	0.1738%	$\mathbf{0.0753\%}$
≤ 0.5	-0.1756%	0.3453%	$\mathbf{0.1509\%}$

Table 6.2. Deviation from the optimal solution using rate limitation.
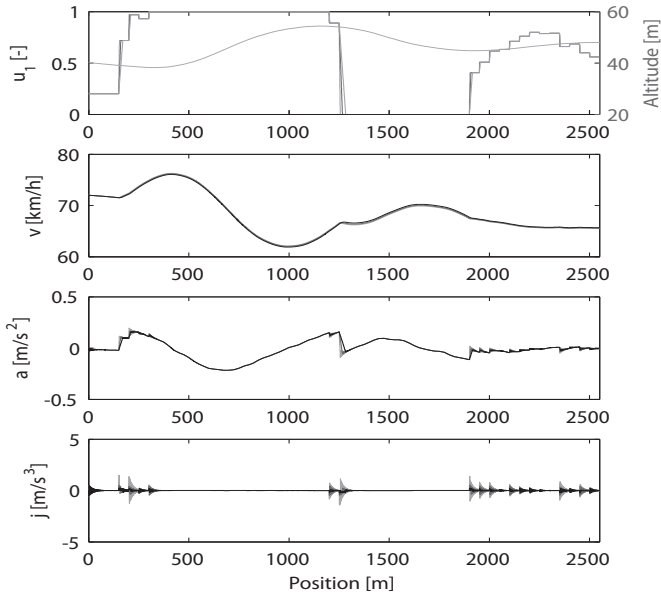
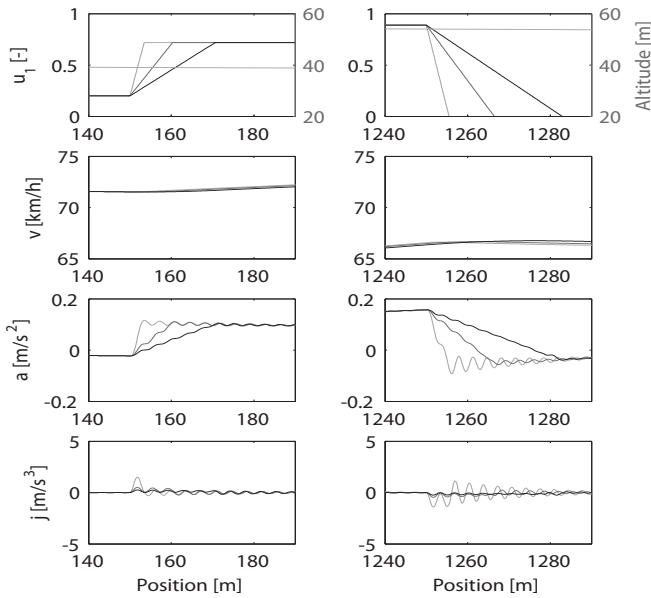**Figure 6.5.** Simulation result using rate limitation.

Figure 6.6. Simulation result using rate limitation.

6.4   Simulation with interpolation

As mentioned in section 5.1.3, cubic interpolation is the method most similar to the LP-filtering used in the actual test runs. Three simulations has been carried out with decreasing steepness in the cubic interpolation.

6.4.1   Impact on the ride comfort

Smoothing the discrete steps in the control signal by cubic interpolation results in the behavior seen in figure 6.7 and figure 6.8. The jerk magnitude is drastically lowered as well as the oscillations of the acceleration and jerk. The acceleration in figure 6.8 shows only minor ripple after each step.

6.4.2   Impact on the cost function

Simulation	Δ_{time}	Δ_{fuel}	$\boldsymbol{\Delta_{cost}}$
1	-0.0611%	0.1174%	$\mathbf{0.0508\%}$
2	-0.1006%	0.1959%	$\mathbf{0.0852\%}$
3	-0.1395%	0.2742%	$\mathbf{0.1198\%}$

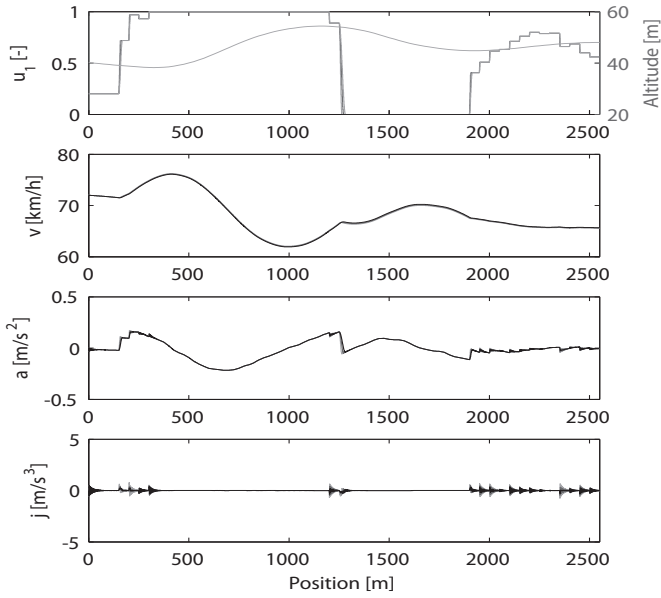**Table 6.3.** Deviation from the optimal solution using cubic interpolation.

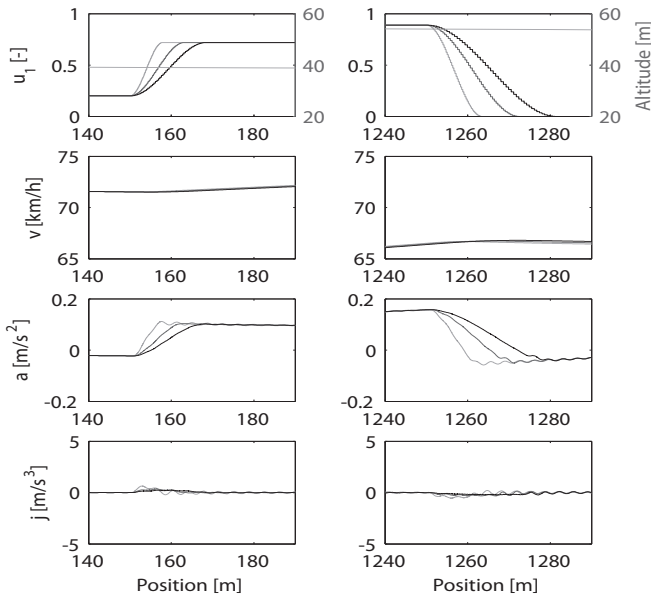**Figure 6.7.** Simulation result using cubic interpolation.

Figure 6.8. Simulation result using cubic interpolation.

Chapter 7

Conclusions and future work

7.1 Conclusions

In the simulations, all the, in this thesis, suggested methods has shown considerable improvements in the ride comfort and driveability at small or no cost increment. As could be expected the average cost increases with increasing manipulation of the optimal control signal.

The cubic interpolation and the rate limiting methods results in similar cost deviations, however the cubic interpolation results in significantly lower oscillations and jerk levels.

In the simulations the method using active damping appear superior to the other methods in preventing oscillations but the simulations also reveal the method's major weakness. Implementing this method into a real system, no doubt, offers some challenges. One way around the issue of robustness could be to only use the value of the systems eigenfrequency in the optimizations cost calculation and rely on sensor feedback when applying the actual control signal. This would result in a good approximation of the cost combined with low calculation times in the optimization as well as high accuracy in the suppression of oscillations.

7.2 Future work

As mentioned in section 1.2, the objective of this thesis is to suggest and evaluate methods for future implementation. Through simulations the potentials of the different methods has been evaluated. However, due to the subjective nature of comfort experience, finding the optimal method and parameters would require actual test runs, based on this qualitative study, to be carried out.

Bibliography

[1] M Schittler. State-of-the-art and emerging truck engine technologies for optimized performance, emissions, and life-cycle costing. In *9th diesel engine emissions reduction conference. U.S.Department of Energy, RI, USA*, August 2003.

[2] Sybil P. Parker, editor. *McGraw-Hill Dictionary of Scientific and Technical Terms*. McGraw-Hill Professional, New York, 6 edition, 2002.

[3] Erik Hellström, Maria Ivarsson, Jan Åslund, and Lars Nielsen. Look-ahead control for heavy trucks to minimize trip time and fuel consumption. *Control Engineering Practice*, 17(2):245 – 254, 2009.

[4] Magnus Pettersson. *Driveline Modeling and Control*. PhD thesis, Linköping University, SE-581 83 Linköping, May 1997.

[5] Henrik Abrahamsson and Peter Carlson. Robust torque control for automated gear shifting in heavy duty vehicles. Master's thesis, Linköping University, SE-581 83 Linköping, 2008.

[6] P.R. Grant and B. Haycock. Effect of jerk and acceleration on the perception of motion strength. *Journal of Aircraft*, 45(4):1190–1197, 2008.

[7] Xi Wei and Giorgio Rizzoni. Objective metrics of fuel economi, performance and driveability - a review. *SAE Technical Paper Series*, 2004.

[8] Lars Eriksson and Lars Nielsen. *Vehicular Systems*. Linköping University, SE-581 83 Linköping, 2006.

[9] Erik Hellström, Jan Åslund, and Lars Nielsen. Design of a well-behaved algorithm for on-board look-ahead control. IFAC World Congress, Seoul, Korea, 2008.
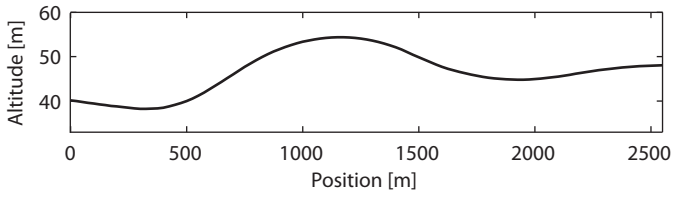
Appendix A

Road profiles used for simulations

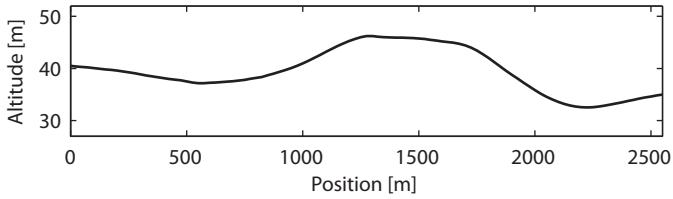**Figure A.1.** Road profile of the northbound Järna segment.

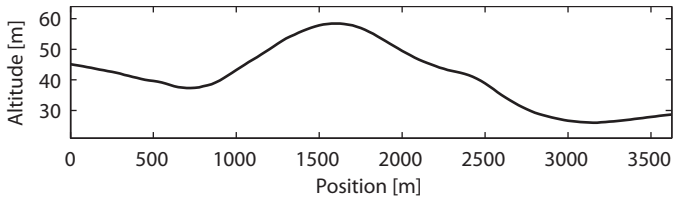

Figure A.2. Road profile of the southbound Järna segment.

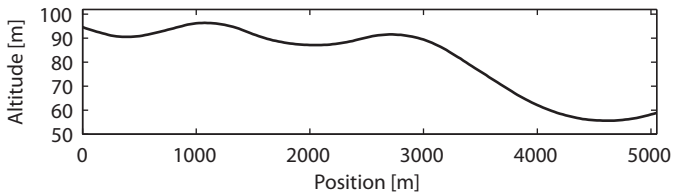**Figure A.3.** Road profile of the northbound Hållet segment.

Figure A.4. Road profile of the southbound Stavsjö segment.