Design of a Torque Control Strategy for Enhanced Comfort in Heavy Trucks

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Abstract

Heavy trucks vehicles by design often suffer from issues causing driver discomfort. That discomfort is here considered caused by oscillations in the driveline, which are affecting the driver. The driveline oscillations - sometimes referred to as shunt and shuffle - exist because of the sheer amount of power in a heavy truck engine, combined with a driveline where the components are relatively weak. If this engine power is simply output to the driveline without control, the driveline will twist and bend, causing oscillations.

The purpose of this work is to investigate how engine torque can be controlled when large torque changes are requested. The specific situation of interest is when the driver fully depresses and releases the accelerator pedal, referred to as a tip-in and tip-out. The goal is to device a strategy that in this situation prevents any driveline oscillations and thus improves the driving comfort.

What is presented is an investigation of some different strategies for applying torque and what seems achievable, using optimal control. This investigation then leads to a suggested alternative to today's strategy of applying torque. The suggested strategy is implemented in the ECU and tried in different vehicles. This first attempt at an implementation proves itself by reaching the target (maximum) torque in the same time as the conventional strategy, indicating no reduced performance. In several cases, large improvements can even be seen. Even while reaching the target level faster, the strategy manages to reduce the oscillations in the driveline. The goal of this work can therefore be considered achieved.

While the suggested strategy works well in many cases, much work is still required to get it fully functional. New problems have been posed, regarding the subjective notion of how a driver actually wants a tip-in to feel. This might then lead the following research in completely different directions. Seen from a bigger perspective, the main result of this thesis is the fact that only with little effort in simple ways, great gains can be achieved.

Acknowledgements

This thesis work has been performed at the Powertrain Torque Control group at Scania CV AB in Södertälje, Sweden during the fall of 2012. It was initiated because of occuring problems of driver discomfort in some driving scenarios. The task at hand was to investigate how these problems could be prevented.

First, I would like to thank Scania for providing the best possible environment for a fresh engineer. One is here given interesting tasks and is challenged with finding solutions to a wide range of problems, bringing ever so much personal development. It is highly doubtful if a place exists where one can be surrounded by more skilled people and intriguing discussions.

Following the praise of Scania, I would like to thank the group NECE for letting me try to find a solution to one of their specific problems. Out of this group, I would like to extend special gratitude towards my supervisors Martin Evaldsson and Niclas Lerede; Martin for his everlasting positivity and continuous stream of ideas, and Niclas for his infinite amount of knowledge (and patience). Several others at Scania also deserve to have my thanks, but there simply isn't room to name them all.

Furthermore, I would like to thank Prof. Karl-Erik Årzen and Prof. Rolf Johansson at the The Department of Automatic Control at Lund University for accepting me as a thesis student and for examining this thesis.

A special thanks is extended to my family and friends, whom I've seen much too little during the course of the last year. I hope that you still remember me.

Finally, I would like to cite Adam Lagerberg in his attempt to explain to his children what he was doing in his research. As I could never find a better way to describe this line of work myself, I'll borrow his (slightly rewritten) explanation;

"I work to become a doctor, a doctor who cures trucks."

Contents

Chapter 1

Introduction

The aim of this section is to introduce the reader to the problems arising in the design process of a heavy truck driveline and to explain the outline of this thesis.

1.1 Background

In the design of a heavy truck driveline, there are many conflicting goals arising. Some of these goals, for example performance, drivability and fuel economy are somewhat easily measured and can thus be compared and valued against each other. One goal that is not as easily measured, but is of utmost importance to the actual drivers of the vehicle, is the driving comfort of the truck.

There are a lot of factors not stemming from the driveline design and that has nothing to do with torque control that can contribute to increase the comfort in a heavy truck. This is all well, as there are always many pieces that have to fit together to create a finished product. The fact is however that if the power output in the powertrain gives no regard for comfort design, those attempts to achieve good comfort will certainly be ruined. This kind of uncontrolled output gives a powertrain that exhibits what is called vehicle shuffle, which is when the cabin of the truck oscillates back and forth, exposing the driver to uncomfortable jerk. Furthermore, noise from the driveline can sometimes be clearly noticed, which is the result of two sides of a backlash smashing into each other. These rough backlash traversals in the driveline gives rise to the shunt phenomenon, which further increases the shuffle. See figure 1.1 for an example of both shunt and shuffle. To avoid this behavior and thus improve the comfort for the driver, the output engine torque needs to be controlled.

1.2 Purpose and goals

The purpose of this thesis is to investigate methods to control the engine torque fed into the driveline so as to reduce the jerkiness of the vehicle and perform
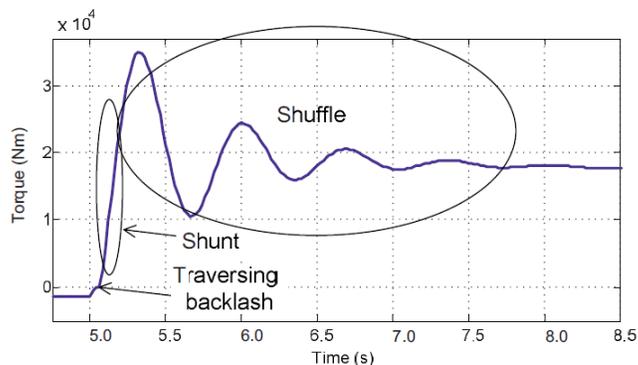
Figure 1.1: An example of the output torque when a step input has been applied, which is the case for a tip-in. The rough backlash traverse in the beginning gives rise to a shunt, where the transmitted torque increases rapidly. The flexibilities in the driveline then causes the following shuffle. [8]

better backlash transitions. The overall goal is thus to find a way of improving the driving comfort while at the same time maintaining a high performance and drivability. Due to the fact that there is great variation in the vehicle properties, it is preferable to find a general torque control strategy. This general strategy might then be feasible for a wide range of vehicles, as opposed to a specific solution designed for a single individual vehicle. The goal is also to device a strategy to be implemented and evaluated in the vehicle ECU if possible.

1.3 Report outline

Chapter 2 describes what heavy truck driveline consists of and some general theory on linear systems, hybrid systems and optimal control.

Chapter 3 describes the development of the driveline model that is to be used as a basis for optimal control and simulation.

In chapter 4, the definitions of the optimal control problems used are stated, and in chapter 5 the resulting solutions to these problems are presented, from which some conclusions can be drawn.

In chapter 6, the suggested strategy is presented and some simulations describing its features and drawbacks are shown.

Chapter 7 presents examples of the performance of the suggested strategy implemented in a truck. The strategy's performance is then compared to the solution used as of today.

In chapter 8, the thesis is concluded with a discussion of the results. Some suggestions for future work are also presented.

Chapter 2

Theory

2.1 Powertrain basics

The powertrain consists of everything needed to propel the vehicle, which essentially means every connected component between and including the engine and the wheels. Fuel is injected into the engine, which is then combusted to produce torque. The torque is transferred through the driveline and reaches the wheels, accelerating the vehicle.

The modular design used at Scania provides a vast number of possible components, which combined means an even greater number of possible drivelines. Each driveline will have specific properties, chosen to suit the operating application of the vehicle. Examples of such operating applications are long haulage, mining, construction and distribution, and each comes with a set of considerations that have to be made.

As an example of a driveline assembly, a wide range of engines are available. Each type of engine provides a certain power and torque output. Following the engine is a clutch, which in turn is connected to a gearbox. The gearboxes have different gearings and might come with or without a retarder. The output shaft of the gearbox is connected to the propeller shaft. These have different lengths, as they are the link between the power production, located in the front of the vehicle and the wheel axles, located in the back. Because of their varying lengths and thicknesses, the propeller shafts will have different stiffness characteristics, as well as other varying properties. The propeller shaft is connected to a final drive, and also these come with different gearings. The final drive distributes the torque to the wheels via the drive shafts. Also the drive shafts have different stiffness characteristics, chosen so that they are able to withstand heavy loads without snapping. The drive shafts are generally considered the weakest - as in the most flexible - part of the driveline, a property that will later prove important. Furthermore, the vehicle can come with or without hub reduction, providing an additional gearing. Finally comes the wheels, putting the power into the ground.

As can be seen in the above description, the possible combinations available with the above mentioned components grow rapidly. This is what leads to the desire to achieve a general solution that might work with each powertrain combination, instead of a specific one, tailored for a single application.

2.2 Linear systems

A linear time-invariant (LTI) system in a state-space representation may be described by

$$\dot{x}(t) = Ax(t) + Bu(t)$$
$$y(t) = Cx(t) + Du(t)$$

The system description relates the input $u(t)$ and the output $y(t)$ through the system matrices A, B, C and D and the system states $x(t)$. The very same system can also be described by its transfer function $G(s)$, relating the input and output through their Laplace transforms, $U(s) = \mathcal{L}\{u(t)\}$ and $Y(s) = \mathcal{L}\{y(t)\}$. The transfer function for the system is then given as $G(s) = \frac{Y(s)}{U(s)}$. How the transfer function and the state-space representation for an LTI system is related is given by the relationship

$$G(s) = C(sI - A^{-1})B + D$$

The input-output relation for a dynamical system can be determined through knowledge of the system. When the differential equations defining the trajectories for the system states are known, the system matrices are quite easily determined. If these differential equations are not linear, the system can often be linearized around a certain operating point in the state space. This linearized system approximation can then be described by the standard LTI formulation.

If knowledge of the system equations does not exist, experiments can be made so that input-output data are gathered by applying for example unit steps or impulses. With this data, frequency response analysis can be made and through the process of system identification the transfer function $G(s)$ can hopefully be estimated.

2.2.1 Second order linear systems

The transfer function for a general second order linear system can be written as

$$G(s) = G_0 \frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2} \tag{2.1}$$

The behavior of the second order system is determined by the parameters ω_n and ξ, describing the system's natural frequency and damping ratio respectively. The natural frequency ω_n describes the rate at which the undamped system response oscillates back and forth. The damping ratio ξ describes the profile

that the system response will have. A damping ratio of 0 will leave the system oscillating at its natural frequency. This is called an undamped system. When the damping ratio is in the range $0 < \xi < 1$ the system is underdamped and will reach the steady state with a slight overshoot. A damping ratio of $\xi = 1$ gives the so called critically damped system, which will reach the steady state level precisely without overshoot. A damping ratio of $\xi > 1$ is called overdamped and such a system will not show any overshoot, but it will take longer to reach the steady state level than in the critically damped case. The steady state gain $G(0)$ is given by G_0.

2.3 Hybrid systems

Throughout this thesis, a hybrid system will have nothing to do with adding electric motors to the powertrain. What is meant is instead dynamical systems that change their properties over different regions of the space spanned by states and inputs. In this thesis, a hybrid system will be considered equivalent to a piesewise affine (PWA) system, and the two might be interchanged.

A PWA system consists of a combination of different affine dynamics. Each of these dynamics is defined over a specific region of the state space and will in this region follow certain trajectories. The PWA system description can generally be stated as

$$
\begin{array}{rcl}
\dot{x}(t) &=& A_i x(t) + B_i u(t) + f_i \\
y(t) &=& C_i x(t) + D_i u(t) + g_i
\end{array}
$$

if $[x(t), u(t)] \in \Pi_i$, where Π_i is the currently active polyhedral set (region) that the state and input belongs to. As one can see, the PWA description closely resembles the common description of a linear system, with the difference of the changing system matrices.

2.4 Optimal control

The problem of optimization occurs frequently in many different areas. One of the main reasons is probably that many real life problems are effectively posed as optimization problems. When the problem is solved at a single instance in time, standard optimization theory is used. When the problem is posed over a sequence of time however, dynamical optimization theory - also known as optimal control - is used. The solution to the optimal control problem is a sequence of inputs in time that renders the minimum value of some defined cost functional, described below.

Applications of optimal control can be found frequently in various fields of engineering such as robotics, process control and perhaps especially aeronautical systems, the field that started the development of and brought optimal control

to prominence. Optimal control is also effectively used in other professional fields such as economics, logistics and biomathematics. [3]

2.4.1 Problem structure

The general optimal control problem can be formulated as (see [4, 3])

$$
\text{minimize } \Phi(x(t_f)) + \int_{t_i}^{t_f} L(t, x(t), u(t)) \, \mathrm{d}t
$$

$$
\text{subject to } \dot{x} = f(t, x(t), u(t))
$$
$$
x(t_i) \in S_i
$$
$$
x(t_f) \in S_f
$$
$$
u(t) \in U(t), \, t_i \le t \le t_f
$$
$$
\psi(x(t_f)) = 0
$$

where Φ is a cost for deviating from the wanted final state, L is the dynamical cost that is integrated over the selected timespan and $\psi(x(t_f))$ determines constraints that has to be fulfilled at the terminal time. $U(t)$ is a time-varying set, determining the input signals $u(t)$ available at time t. S_i and S_f define sets which contain the initial state and the final state respectively. $f(t, x, u)$ describes the propagation of the system states.

2.4.2 GPOPS

The tool used in finding the optimal control solution is the MATLAB toolbox GPOPS (see [9, 10, 11, 12, 13, 14, 15, 16, 17, 18]), developed at the University of Florida. It uses an hp-Adaptive Pseudospectral method in turning the problem into a non-linear programming problem and then uses (a limited version of) the SNOPT software which solve these kind of problems efficiently. From the perspective of this thesis, GPOPS provides a convenient interface for defining and solving optimal control problems.

The toolbox has no built-in functionality for hybrid systems. Different system dynamics and constraints can however be defined by, instead of using the hybrid system description, formulating the optimal control problem as a multiphase problem. Each of the phases then uses the currently active dynamics and constraints, as in a regular problem description. The phases are linked by linkage constraints and the optimal control can be found for the whole sequence of phases. This procedure works well in this application when some basic knowledge of the problem at hand is attained, so that a general picture of the sequence of states and controls is known.

Chapter 3

Modeling

Mathematical modeling of a powertrain, based upon the generalized Newton's second law, has been done in numerous previous works. A thorough and descriptive modeling process can be found in for example [1]. The referenced work provide a basis for many other papers of driveline modeling and control to extend upon. For each of the components of the driveline, assumptions of what will affect the control and modeling performance are made. A few models of different complexities are designed and compared in simulation. An important conclusion (with regard to this thesis) in [1] is that the basic driveline model, consisting of only a flexible drive shaft, provides a good enough model for control design. For verification of the control design, the derived controllers are simulated against more detailed driveline models, incorporating for example clutch nonlinearities and propeller shaft flexibilites. It is thus shown that although the controllers are based upon simple models, the design is assumed to be robust enough to handle a more complex system. The important conclusion from [1] applicable to this thesis is that a simple driveline model can be used, still providing a sufficient amount of detail.

3.1 Driveline modeling

In this thesis, two different driveline models are investigated. The basic model has two rotational masses connected by a damped torsional spring. The first rotational mass represent the masses of the components on the engine side of the drive shaft; the engine, the clutch, the gearbox and the propeller shaft, where the engine is the dominant contribution. The other rotational mass represent the masses at the load side of the drive shaft; the mass of the vehicle itself, the wheels and the drive shaft. The dominating mass on the load side is naturally the vehicle mass. Both rotational masses has a damping factor representing rotational friction. The gearbox provides a conversion with ratio i, increasing the torque and decreasing the rotational speed.

The other driveline model is very much similar to the first one, the only

difference (however important) being an added backlash. Backlash is introduced in the driveline from a range of various components. Among these, the main causes of backlash are gear play in the gearbox and differential. However, as a means to reduce the vibrations in the flywheel and clutch, weak springs are added. Looking at this in less detail, from a perspective where greater torque is output, the behavior of these weak springs are much like that of an additional backlash. Even though the backlashes in the driveline are several and they enter in different ways, they will for this application behave approximately as one. The need for an as simple model as possible therefore leads to the assumption of a single backlash of appropriate size. It is assumed to be located between the two sides of the driveline, in connection with the flexible shaft.
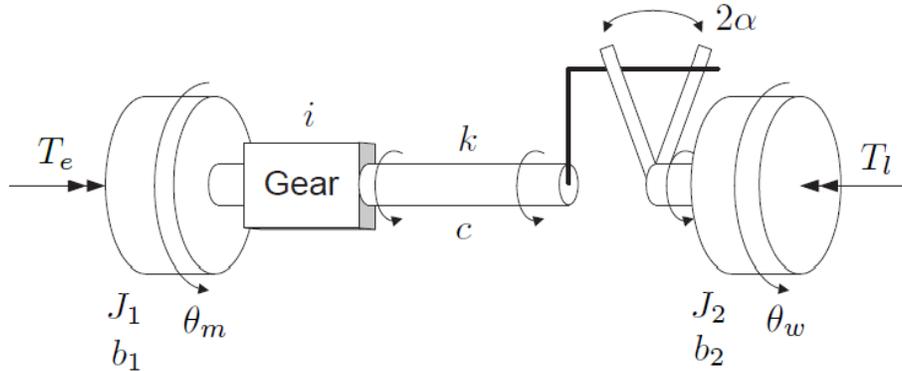
Figure 3.1: A simple sketch of the driveline model, as seen in [1].

The two differential equations describing this simple driveline model are

$$J_1 \dot{\omega}_m = T_e - b_1 \omega_m - T_w/i \tag{3.1}$$

$$J_2 \dot{\omega}_w = T_w - b_2 \omega_w - T_l \tag{3.2}$$

where J_1, J_2 represent the rotational masses, T_e, T_w, T_l the engine, wheel and load torque, b_1, b_2 the rotational friction components, i the total conversion ratio of the driveline including both the gearbox and the differential gearing, and ω_m, ω_w the rotational speed at the engine and load side respectively. θ_m and θ_w correspondingly describe the angle of the rotational masses, and $\theta_d = \theta_m/i - \theta_w$ thus describe their difference, scaled to the wheel side of the driveline. This means that the twist of the driveshaft will be described by θ_d.

The engine torque T_e is the torque that the engine produces, which is the only input to the driveline that is available for control. This signal is thus controlled to give T_w the wanted profile. The load torque T_l represent all forces acting on the opposite direction of travel. These forces are the air resistance, the rolling resistance and the negative torque resulting from road slope.

3.1.1 Model simplifications

Some of the main simplifications made when developing this model are that the delay from requested to produced torque in the engine is neglected. Furthermore, the dynamics of the engine are not modeled, so the requested torque is instantly produced. The load torque is assumed constant, but is in fact proportional to the vehicle speed squared, due to the air drag. At low speeds this assumption is however fairly correct, as can be seen in figure 3.2. When compared to the actual load torque with a quadratic speed dependence, the relative error is less than 2 % for the specified range of vehicle speeds. Considering the gear ratio engaged at these speeds, which might be in the range of 5-20, the absolute load torque error scaled to an engine torque error is far lower than the resolution of the engine's output torque. This error, considered the most important one regarding the load torque, thus shouldn't make any big impact on the overall model. Finally, any wheel slip is also neglected.
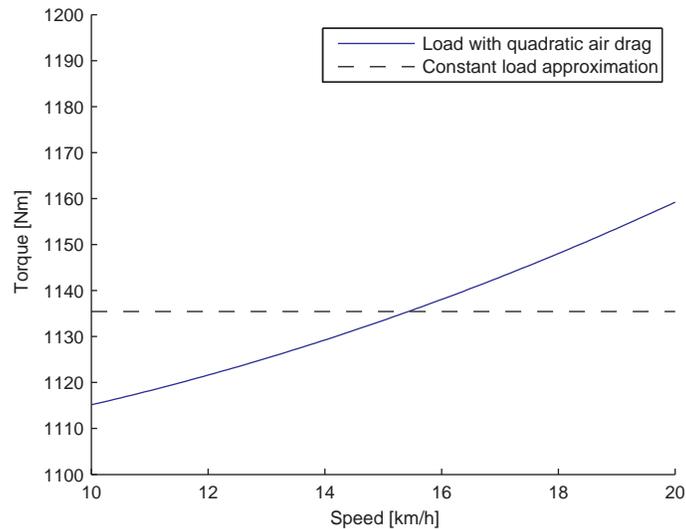
Figure 3.2: A comparison between the constant load approximation and the load with quadratic term for the air drag.

3.2 Backlash modeling

In [2], some different models for backlash are presented. Two of these models are described in short below, which are the simple and commonly used dead-zone model and the physical model developed in [2]. In all backlash models, the inertia of the flexible shaft itself is assumed to be significantly less than both the load and motor inertia, an assumption that holds for this powertrain configuration.

3.2.1 The deadzone model

The deadzone model is a simple and commonly used way to describe backlash. The transferred torque is described as a function of the shaft displacement, θ_d, and its derivative (assuming shaft damping is considered). The shaft displacement is defined as the angle difference between the motor and load side, scaled to the load side so that

$$\theta_d = \frac{\theta_m}{i} - \theta_w$$

The equation for the transferred torque is

$$T_w = \begin{cases} k(\theta_d - \alpha) + c\dot{\theta}_d, & \theta_d > \alpha \\ 0 & |\theta_d| < \alpha \\ k(\theta_d + \alpha) + c\dot{\theta}_d, & \theta_d < -\alpha \end{cases} \tag{3.3}$$

where k and c represent the shaft stiffness and damping respectively and α represents the backlash size.

The simplicity of the deadzone model is appealing, but it also comes with its disadvantages. When the shaft by the backlash has damping, the deadzone model deteriorates. "Unphysical" pull forces of the wrong sign are realised before the backlash is entered, which is shown in [2].

3.2.2 The physical model

As is shown in [2] the "Physical Model" is the backlash model providing the most realistic representation. The downside of the model is the additional state required to describe the backlash angle. As stated in section 3, the complexity of the driveline model is best kept to a minimum. The additional state is therefore a serious downside in some cases, making this added detail somewhat less valuable. In other cases where computational speed is not a problem however, this would be the model of choice.

Introduce θ_b for the backlash angle and define $h_{bl} \equiv \dot{\theta}_d + \frac{k}{c}(\theta_d - \theta_b)$. The backlash angle for the physical model is then updated according to

$$\dot{\theta}_b = \begin{cases} \max(0, h_{bl}), & \theta_b = -\alpha \\ h_{bl}, & |\theta_b| \leq \alpha \\ \min(0, h_{bl}), & \theta_b = \alpha \end{cases} \tag{3.4}$$

representing a limited integrator with time derivative h_{bl}. The transferred torque for the physical model is then given by

$$T_w = k(\theta_d - \theta_b) + c(\dot{\theta}_d - \dot{\theta}_b) \tag{3.5}$$

A comparison of the delivered torque in the deadzone and the physical model is performed in simulation in [2], and another comparison can be seen in figure 3.3, where an approximate engine torque step has been applied. The wheel torque for both models is shown. As stated for the deadzone model, the torque

takes on a positive value before hitting zero during the backlash passage, something that is clearly not possible. The time of contact and the torque delivered at impact is for the deadzone model however almost exactly as good as for the physical model, as shown in [2]. In this tip-in/tip-out application, a well-modeled contact torque after leaving the backlash is considered more important than modeling the correct torque when entering the backlash. The deadzone model is therefore considered viable to use, despite its shortcomings.
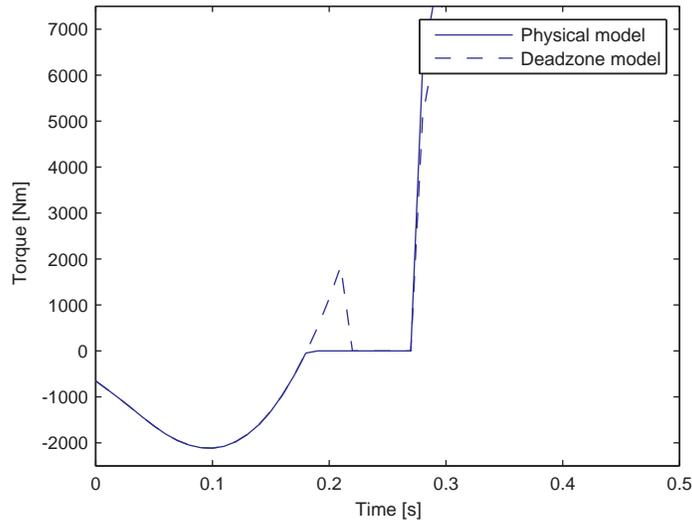
Figure 3.3: A wheel torque comparison for the two different models of backlash.

3.3   State space representation

In this section, state space representations for the previously derived models will be described.

3.3.1   Driveline model

Using the state vector $x = \left[\theta_m/i - \theta_w, \dot{\theta}_m, \dot{\theta}_w \right]^T$, $u = T_e$, $z = T_w$ and $l = T_l$ gives the following state space realization for the linear driveline model without backlash

$$\begin{aligned}
\dot{x} &= Ax + Bu + Hl \\
&= \begin{pmatrix} 0 & \frac{1}{i} & -1 \\ -\frac{k}{J_1 i} & -\frac{1}{J_1}(b_1 + \frac{c}{i^2}) & \frac{c}{J_1 i} \\ \frac{k}{J_2} & \frac{c}{J_2 i} & -\frac{1}{J_2}(b_2 + c) \end{pmatrix} x + \begin{pmatrix} 0 \\ \frac{1}{J_2} \\ 0 \end{pmatrix} u + \begin{pmatrix} 0 \\ 0 \\ -\frac{1}{J_2} \end{pmatrix} l \\
y &= Cx = \begin{pmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \end{pmatrix} x \\
z &= Mx = \begin{pmatrix} k & \frac{c}{i} & -c \end{pmatrix} x
\end{aligned}$$

The transfer function for this LTI system from the controlled input u and the load input l to the performance output z is given by

$$G_{DM}(s) = \frac{i(k + sc)((J_2 s + b_2)u + (J_1 s + b_1)l)}{i^2 s(J_1 s + b_1)(J_2 s + b_2) + (k + sc)((J_1 i^2 + J_2)s + (b_1 i^2 + b_2))}$$

This expression is simplified by disregarding the slowest poles and zeros resulting from the friction components by setting $b_1 = b_2 = 0$. We thus concentrate on the main dynamics of the system and get the simplified transfer function

$$\tilde{G}_{DM}(s) = \frac{c}{J_1 i} \frac{s + \frac{k}{c}}{s^2 + J_{rec}cs + J_{rec}k}$$

from engine torque to wheel torque, where

$$J_{rec} = \frac{1}{J_1 i^2} + \frac{1}{J_2}$$

is the reciprocal sum of the two inertias. Relating this expression to equation 2.1, describing the general second order system, it is easy to see that the steady-state gain for the driveline model is

$$\tilde{G}_{DM}(0) = \frac{J_2 i}{J_1 i^2 + J_2}$$

and that the natural frequency and the damping ratio is given by

$$\omega_n = \sqrt{J_{rec}k} = \sqrt{\left(\frac{1}{J_1 i^2} + \frac{1}{J_2}\right)k} \tag{3.6}$$

and

$$\xi = \frac{c}{2}\sqrt{\frac{J_{rec}}{k}} = \frac{c}{2}\sqrt{\frac{\left(\frac{1}{J_1 i^2} + \frac{1}{J_2}\right)}{k}} \tag{3.7}$$

This model will further on be called the Driveline Model (DM).

3.3.2 Deadzone model

When the backlash is considered, the model is no longer linear. The dynamics of the system change when the transition between the respective modes of the model is made, as is seen in the equations describing the two backlash models 3.3 and 3.4. For the deadzone backlash model three distinct modes exist over the state space, and the transition between them depend on the state of the drive shaft torsion. The modes are

- connection on the negative side (co-) for $x_1 \leq \alpha$

- no connection, i.e. backlash mode (bl) for $|x_1| < \alpha$

- connection on the positive side (co+) for $x_1 \geq \alpha$

For each mode a separate state space description is needed. Using the deadzone model for the backlash, the same system states as for the DM model can be used, rendering the following hybrid system representation

$$\dot{x} = \begin{cases} A_{co+}x + Bu + f_{co+}, & co+ \\ A_{bl}x + Bu + f_{bl}, & bl \\ A_{co-}x + Bu + f_{co-}, & co- \end{cases}$$

$$y = Cx$$

$$z = \begin{cases} Mx - k\alpha, & co+ \\ 0, & bl \\ Mx + k\alpha, & co- \end{cases}$$

where

$$A_{co+} = A_{co-} = \begin{pmatrix} 0 & \frac{1}{i} & -1 \\ -\frac{k}{J_1 i} & -\frac{1}{J_1}(b_1 + \frac{c}{i^2}) & \frac{c}{J_1 i} \\ \frac{k}{J_2} & \frac{c}{J_2 i} & -\frac{1}{J_2}(b_2 + c) \end{pmatrix}$$

$$A_{bl} = \begin{pmatrix} 0 & \frac{1}{i} & -1 \\ 0 & -\frac{b1}{J_1} & 0 \\ 0 & 0 & -\frac{b2}{J_2} \end{pmatrix}$$

$$B = \begin{pmatrix} 0 \\ \frac{1}{J_1} \\ 0 \end{pmatrix}$$

$$f_{co+} = \begin{pmatrix} 0 \\ \frac{k\alpha}{J_1 i} \\ -\frac{k\alpha + l}{J_2} \end{pmatrix}, \ f_{co-} = \begin{pmatrix} 0 \\ -\frac{k\alpha}{J_1 i} \\ \frac{k\alpha - l}{J_2} \end{pmatrix}, \ f_{bl} = \begin{pmatrix} 0 \\ 0 \\ -\frac{l}{J_2} \end{pmatrix}$$

$$C = \begin{pmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \end{pmatrix}$$

$$M = \begin{pmatrix} k & \frac{c}{i} & -c \end{pmatrix}$$

This model will be referred to as the deadzone model (DZM).

3.3.3 Physical model

The physical backlash model introduces an additional state, describing the backlash angle. The state vector is thus extended to $x = \left[\theta_m/i - \theta_w, \dot{\theta}_m, \dot{\theta}_w, \theta_b\right]^T$, θ_b representing the backlash angle. For the physical model two modes exist; the system can be in either connection or backlash mode. The two modes are defined over different parts of the state space, summarised below.

- connection on the negative side, with negative torque (co);

$$\theta_b = -\alpha \wedge h_{bl} < 0$$

- connection on the negative side, moving into the backlash (bl);

$$\theta_b = -\alpha \wedge h_{bl} \geq 0$$

- no connection (bl);
$$|\theta_b| < \alpha$$

- connection on the positive side, moving into the backlash (bl);

$$\theta_b = \alpha \wedge h_{bl} \leq 0$$

- connection on the positive side, with positive torque (co);

$$\theta_b = \alpha \wedge h_{bl} > 0$$

The state space representation then becomes

$$\dot{x} = \begin{cases} A_{co}x + Bu + f, & co \\ A_{bl}x + Bu + f, & bl \end{cases}$$

$$y = Cx$$

$$z = \begin{cases} M_{ph}x, & co \\ 0 & bl \end{cases}$$

where

$$A_{co} = \begin{pmatrix} 0 & \frac{1}{i} & -1 & 0 \\ -\frac{k}{J_1 i} & -\frac{1}{J_1}\left(b_1 + \frac{c}{i^2}\right) & \frac{c}{J_1 i} & \frac{k}{J_1 i} \\ \frac{k}{J_2} & \frac{c}{J_2 i} & -\frac{1}{J_2}\left(b_2 + c\right) & \frac{k}{J_2} \\ 0 & 0 & 0 & 0 \end{pmatrix}$$

$$A_{bl} = \begin{pmatrix} 0 & \frac{1}{i} & -1 & 0 \\ 0 & -\frac{b_1}{J_1} & 0 & 0 \\ 0 & 0 & -\frac{b_2}{J_2} & 0 \\ \frac{k}{c} & \frac{1}{i} & -1 & -\frac{k}{c} \end{pmatrix}$$

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$$B = \begin{pmatrix} 0 \\ \frac{1}{J_1} \\ 0 \\ 0 \end{pmatrix}$$

$$f = \begin{pmatrix} 0 \\ 0 \\ -\frac{l}{J_2} \\ 0 \end{pmatrix}$$

$$C = \begin{pmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{pmatrix}$$

$$M_{ph} = \begin{pmatrix} k & \frac{c}{i} & -c & -k \end{pmatrix}$$

This model will be referred to as the physical model (PHM).

3.4   Physical engine constraints

The physical constraints that come into play in this application are mainly limits on the torque that the engine can produce. The absolute maximum and minimum values of the available engine torque limits are assumed constant, although they in reality depend on for example engine speed. In addition, there are also dynamic constraints on the engine torque, which are caused by the smoke limiter. This limiter is used to disallow combustion in certain modes of operation that would otherwise produce emissions outside of the legal limits. The limiting level of the smoke limiter depends on the boost pressure of the supercharger, which has to be built before the full engine torque is available.

The effect in practice during a tip-in is that the maximum torque level takes some time to build. Therefore, the upper limit of the engine torque during a tip-in is modeled as a timevarying function, while the lower limit is assumed constant. During a tip-out, the boost pressure is already high, so the torque limits are set only by the hard engine constraints. The set of available input signals $U(t)$ is therefore defined by

$$U(t) = \{u | u_{min} \le u \le u_{smoke}(t)\}$$

where

$$u_{smoke}(t) = \begin{cases} u_{max}(0.6 + 0.1t^2) & 0 \le t \le 2 \\ u_{max} & t > 2 \end{cases}$$

so that the smoke limiter level starts at 60 % of the maximum available torque and then has a quadratic buildup over 2 s of the final 40 %. A plot showing the torque limits can be seen in figure 3.4.

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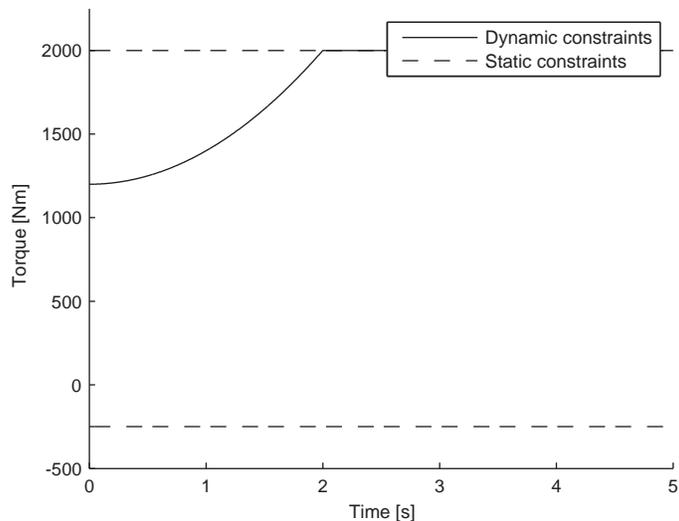
Figure 3.4: The dynamic and static input signal constraints.

3.5   Scaling

To achieve better numerical stability in the calculations, the model is scaled such that each signal and state in the model achieves values approximately in the range [-1, 1]. Before any scaling is made, the mean value of the signal or a stationary value of the model is usually removed. A stationary value for the DM is calculated for a specified wheel speed ω_w and calculated load l, giving

$$A\bar{x} + B\bar{u} + Hl = 0$$

This stationary point is then removed from the model equation, so that the model varies about its stationary value, giving

$$\dot{x} = A(x - \bar{x}) + B(u - \bar{u})$$

Now, an effective scaling can be made. Using the coordinate transformations

$$
\begin{aligned}
D_x\tilde{x} &= (x - \bar{x}) \\
D_u\tilde{u} &= (u - \bar{u}) \\
D_y\tilde{y} &= (y - \bar{y}) \\
D_z\tilde{z} &= (z - \bar{z})
\end{aligned}
$$

where the $D_{\cdot}$ matrices are diagonal matrices containing the (approximate) maximum value of each corresponding signal, the new coordinates will have about the same magnitude and will vary in the specified range. This results in better numerical robustness and makes for better comparisons as to which signals

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differ the most from its wanted value. The resulting, scaled system is given by

$$
\begin{aligned}
\tilde{x} &= D_x^{-1} A D_x \tilde{x} + D_x^{-1} B D_u \tilde{u} \\
\tilde{y} &= D_y^{-1} C D_x \tilde{x} \\
\tilde{z} &= D_z^{-1} M D_x \tilde{x}
\end{aligned}
$$

Also the DZM and PHM are scaled accordingly.

3.6   System identification

The modeled system is known only up to a certain degree. While there are numerical values available for most of the system parameters, some are simply not available for measure or are hard to come by. The most obvious ones belonging to this category are the friction parameters and the shaft damping. Yet, while there are comparatively exact parameters available for the shaft flexibilities, vehicle weight, road slope etc. also these parameters contain uncertainties. These uncertainties can result from many different factors. One of the most important is the simplified model used.

For example, as the model description contains only a single flexible shaft, this modeled flexibility has to incorporate any and all "real" flexibilities appearing in a driveline, including for instance the propeller shaft which is also known to be rather flexible. Also components that would seem stiff, such as the shafts in a gearbox, are in fact flexible. Thus, however exact the value of the drive shaft stiffness k is, using this known value for the model flexibility might make the model output completely different from the available measurement data. This example can be extended to most other parameters in the model, perhaps with a few exceptions such as the transmission ratio i. To remedy the problem of parameter discrepancies, a grey-box system identification is made. The known numerical values are used as initial guesses for the parameters in the identification algorithm.

The identification routine is performed using MATLAB and the System Identification Toolbox [19]. Setting up a grey-box model is done using the command `idgrey` and identification of the parameters using `pem`. The `pem` routine uses optimization to minimize the cost function

$$
V_N(G, H) = \sum_{t=1}^{N} e^2(t)
$$

where $e(t)$ is the difference between the measured output and the predicted output of the model [20]. The parameters minimizing this cost function are then the ones chosen to best describe the model.

3.7   Model validation

The data for which the model is estimated is a sequence of tip-ins and tip-outs. The vehicle travels in a span around 11 and 19 km/h and gear 4 is engaged. At

first, the linear driveline model was estimated using the measured data. After this, the backlash was added to the model. The size of the backlash is arbitrarily chosen, but a first estimate was chosen using the estimates found in [7] together with discussions with experienced engineers working at Scania. The value was then somewhat tuned to a value that looks reasonable in simulation. Just as for the other parameters in the model, modeling the exact size of the backlash is not considered important as this parameter widely varies between vehicles.

A simulation of the DZM can be seen in figure 3.5. The resulting model of the driveline is not perfect, but it can be seen that the main characteristics of a heavy truck driveline are well described. The oscillating behavior is clear and a notable backlash can be seen at 4 and 6.5 s. To have a model that describes the driveline in full detail is not considered important, as it is the overall behavior from changes in the model parameters and a general implementable strategy for any driveline that is sought. A model of low order and complexity is therefore prioritized.
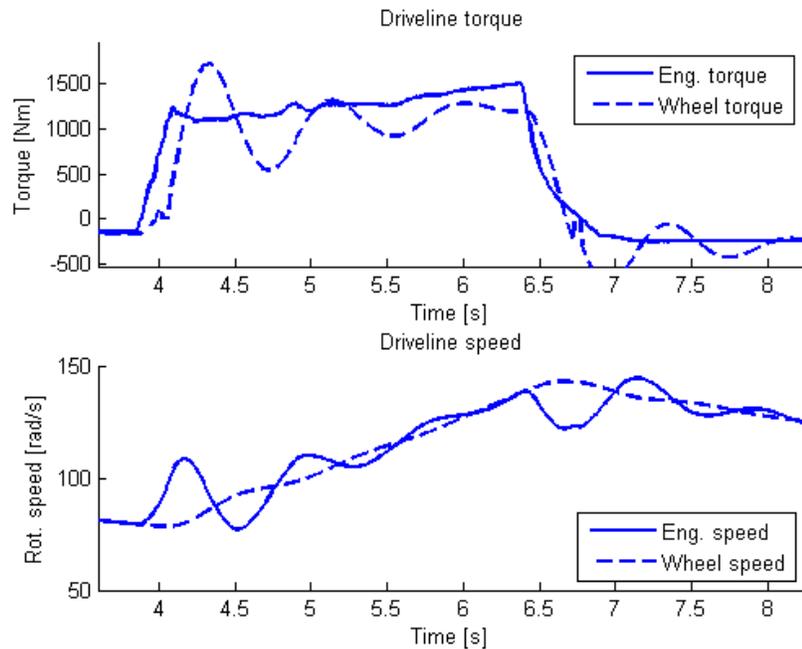
Figure 3.5: Simulation of the model with a deadzone backlash.

The result of the linear model fit can be seen in figure 3.6. The model fit measure for the linear driveline (without backlash) gives a value of 75.38 % for the engine speed and 69.69 % for the wheel speed. The reason for evaluating only the linear model and not the model with backlash is simply that the System Identification Toolbox doesn't have any functionality for hybrid systems. If the

model fit values are on a satisfactory level can be debated, but as previously mentioned the model well enough describes the main characteristics. It should be further noted that the difference between the output from the model and the measurements does not occur in a single tip-in/tip-out sequence, but over several such cycles. As the model is to be used to investigate single events of tip-in and tip-outs, the need for greater detail becomes even lower.
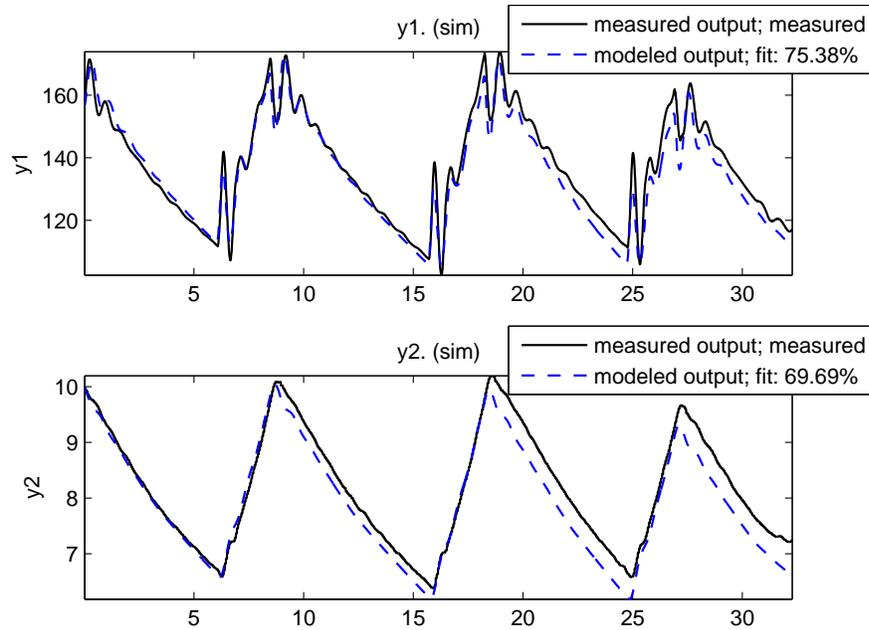
Figure 3.6: The result of the 'compare' command in MATLAB System Identification Toolbox, displaying the engine and wheel speeds. The model error grows larger over a sequence of tip-ins and tip-outs.

Similar driveline models that are of higher detail are derived in [1]. There, additional flexibilities in the propeller shaft and the clutch nonlinearities are added to the model. Should the need exist, the same model extensions could be done also for this application.

Chapter 4

Problem formulation

To investigate the optimal strategy of applying engine torque so that the vehicle behaves in the specified way, optimal control is used. The optimization is made over the space of all available control signals, i.e. the available engine torque. The engine torque is output so that a defined cost functional is minimized. This cost functional can be defined rather freely, but different costs give different behaviour regarding computational complexity of solving the optimization problem. Common choices of cost functionals in this application incorporate terms such as the time to reach a target set (for instance a reference torque or a backlash angle position), the wheel torque and/or its derivative and different combinations of the model states.

One of the most important aspects in optimal control is defining the problem, so as to get the wanted solution. What is optimal thus depends on what is asked for, i.e. how the problem has been set up. The choice of cost functional and constraints of the problem is what determines the solution, so care has to be taken in defining the problem.

The goal with a tip-in is to go from a negative wheel torque when the vehicle is rolling freely, to maximum torque in an as short time as possible. This is done by first reaching the backlash after some time of engine braking. When the backlash is reached, it is traversed as quickly as possible. When traversing the backlash, care must be taken so that the preferred behavior regarding the comfort issues is achieved. When the two sides are connected, so that the positive backlash side is reached, torque is applied so that the wanted wheel torque profile is achieved.

4.1 Reaching the backlash

Going from the phase of engine braking to the backlash, comfort issues are neglected due to the fact that no major (negative) drive shaft torsion has been built up. The assumption is therefore that the torque contribution from the release of the negative torsion is small enough not to affect the performance

and feel of a tip-in. Assuming that the backlash should simply be reached as fast as possible, the maximum available engine torque is applied so that the engine side of the driveline is accelerated. With this approach, the transfer into the backlash is trivial. For completeness, as the solution to the whole tip-in sequence is saught, and for the ability to set different starting points, this phase is however also posed as an optimal control problem. The formulation becomes

$$\min_{u} t_f^1$$
$$s.t. \ \dot{x} = A_{co-}x + Bu + f_{bl-}$$
$$x(0) = x^0$$
$$u(t) \in U(t), \ \forall t \geq 0$$
$$x_1(t_f^1) = \text{-}\alpha$$

where t_f^1 denotes the end of the first, pre-backlash phase.

4.2   Backlash traversal

The backlash phase starts after a period of engine braking so that the vehicle is and has been decelerating for some time, i.e. negative wheel torque has been output. The main goal of the phase is to reach the positive side of the backlash as fast as possible. However, when the backlash has been traversed and the engine and load side meet, their speed difference will produce a step in the wheel torque. This torque step change is what causes the shunt phenomenon and will, at least to some extent, also affect the cabin and thus the driver. Some of the shunt effect can be tolerated, but the goal is to keep the size of this torque step below some specified level. Therefore, a constraint that limits the relative speed between the load and engine side at the time of contact is imposed. A similiar approach can be found in [5].

The optimal control problem to be solved for the backlash phase is

$$\min_{u} t_f^2$$
$$s.t. \ \dot{x} = A_{bl} + Bu + f_{bl}$$
$$x(0) = x^0$$
$$u(t) \in U(t), \ \forall t \geq 0$$
$$x_1(t_f^2) = \alpha$$
$$\left| x_2(t_f^2)/i - x_3(t_f^2) \right| \leq q_{\text{bl speed difference}}$$

where t_f^2 denotes the end of the second phase, and the other notations are as before. $q_{\text{bl speed difference}}$ is introduced and sets a limit for how large the speed difference between the two sides of the driveline is allowed to be.

4.3   Torque control

While the problem definitions in the first two phases are somewhat trivial, the strategy of applying engine torque to achieve the best wheel torque profile is not as clear. While simply applying engine torque more carefully certainly reduces any torque oscillations, it also lessens the response of the vehicle and reduces the vehicle performance. To find a suitable solution, a balance between the two extremes of full comfort and maximum performance has to be found.

The torque reference is most often given in engine torque. For example, the maximum available engine torque is to be output when the driver fully depresses the accelerator pedal. To be able to calculate feasible solutions to these torque control problems, the reference engine torque is recalculated to a reference wheel torque. As some torque is used to accelerate the engine and some disappear due to friction, all torque that the engine produces can not be transferred to the wheels. These terms therefore have to be subtracted from the reference engine torque, so that the wheel torque reference become

$$r = r(T_e^{ref}, i, \omega_m, \dot{\omega}_m) = i(T_e^{ref} - J_1\dot{\omega}_m - b_1\omega_m)$$

If the wheel torque reference is simply given as

$$r = iT_e^{ref}$$

the problem becomes infeasible and no solutions can be found.

To find a solution that produces the wanted torque profile, some different optimal control problems are designed. Based on the solutions to these problems, a strategy will be developed and used for implementation.

4.3.1   Torque rate-of-change damping strategy

As a first example, one way to reduce the torque oscillations is simply to reduce the rate-of-change of the torque so that no oscillations are induced. Adding a cost on the wheel torque derivative, $M\dot{x}$, achieves this purpose. High derivative values occur when there are oscillations, and these high values will thus be avoided. The result should therefore be a steadily rising wheel torque. The derivative cost will make the wheel torque change slowly enough so that no oscillations occur, while the final time cost will make sure that the reference torque is reached reasonably fast. The desired torque profile is then designed by tuning the two weight parameters q_{time} and q_{der}. A high value of q_{time} gives a faster response while a higher value of q_{der} is meant to give a more comfortable tip-in. Making sure that the reference torque is reached at the final time is done by adding the final time torque constraint $Mx(t_f) = r$.

The optimal control problem is stated as

$$\min_{u} q_{\text{time}} t_f + q_{\text{der}} \int\limits_{0}^{t_f} (M\dot{x})^2 \, dt$$

$$s.t. \ \dot{x} = f(x, u)$$
$$x(0) = x^0$$
$$u(t) \in U(t), \ \forall t$$
$$Mx(t_f) = r$$

4.3.2 Torque rate-of-change damping strategy with reference cost

As in the previous example, a cost for the torque derivative is added so that torque oscillations are avoided. To achieve the requested reference torque, deviations from this reference value is also punished. This will bring the wheel torque closer to the reference value already from the beginning, which should provide additional performance compared to the previous example. The wheel torque should with this strategy be able to reach higher values, still without oscillations, and thus a better torque profile will be achieved.

To alter the solution behavior, tunable weights for the two conflicting goals are introduced. If achieving the reference torque sooner is considered more important, the relative size of the corresponding weight q_{torque} is increased. If a more cautious and comfortable solution is wanted, the relative size of q_{der} is increased. Note that it is only the relative sizes of the weights that is important, so the same solution could very well be found using only a single weight. Two different ones will however be used for clarity.

As a cost for deviating from the reference is now active during the whole sequence, there is no need for a final time constraint and final time cost. The reference torque will be reached according to its relative importance compared to the torque derivative. The final time cost and constraint previously used is therefore removed.

The torque control phase is finished as soon as the reference wheel torque has been reached. To hinder any following oscillations from arising, the speed difference between the two sides is kept below a certain acceptable level. With zero speed difference the shaft torsion will remain constant, thus keeping the wheel torque constant. Therefore any following oscillations will be reduced with a small enough speed difference. This acceptable level, chosen by tuning $q_{\text{final time speed difference}}$, becomes an additional design parameter.

The optimal control problem thus becomes

$$\min_u \int_0^{t_f} q_{\text{torque}}(Mx - r)^2 + q_{\text{der}}(M\dot{x})^2 \, dt$$

$$\begin{aligned}
&s.t. \ \dot{x} = f(x, u) \\
&\quad x(0) = x^0 \\
&\quad u(t) \in U(t), \ \forall t \\
&\quad |x_2(t_f)/i - x_3(t_f)| \leq q_{\text{final time speed difference}}
\end{aligned} \tag{4.1}$$

4.3.3 Equal torque strategy

As long as there is an explicit cost on the torque derivative, one might reason that the damped response is achieved in a forced way. Perhaps there are ways to achieve non-oscillating torque profiles without actually forcing the solution to be damped.

A completely stiff driveline without any flexibilites would produce as wheel torque the input engine torque directly, less the torque resulting from acceleration of the engine and its friction. This driveline could therefore achieve exactly the requested wheel torque profile, as the wheel and engine torque would be equal. This reasoning leads to the idea behind the strategy of applying engine torque so that it matches the dynamical wheel torque during the torque control sequence. The assumption is that the more equal the two torques are, the more comfortable the tip-in will be. To get a high-performance tip-in, a cost for deviating from the reference torque is also added. This solution will therefore be the result of a balance between equalizing the two torques and applying maximum torque for performance. The optimal control problem becomes

$$\min_u \int_0^{t_f} q_{\text{reference torque}}(Mx - r)^2 + q_{\text{difference}}(Mx - iu)^2 \, dt$$

$$\begin{aligned}
&s.t. \ \dot{x} = f(x, u) \\
&\quad x(0) = x^0 \\
&\quad u(t) \in U(t), \ \forall t \\
&\quad |x_2(t_f)/i - x_3(t_f)| \leq q_{\text{final time speed difference}}
\end{aligned} \tag{4.2}$$

Also here, weighting parameters are added to tune the solution and a speed difference constraint is introduced at the final time to limit any following oscillations.

Chapter 5

Optimal control solutions

5.1 Tip-in

In this section, solutions to the three optimal control problems described in chapter 4 is presented. The starting state used in these solutions is chosen from the measured data used for estimating the model, and is taken right before one of the tip-ins is applied. There is a slight negative twist of the driveshaft after some time of free rolling, where the wheel torque has thus been negative and no engine torque has been output. This also means that the driveline is on the negative side of the backlash, so that the backlash initially has to be traversed. The vehicle is rolling at approximately 11 km/h and gear 4 is selected, as in the data used when estimating the model. The engine and wheel speed is approximately equal. This situation will be the same for every following case.

5.1.1 Backlash traversal

Before the torque phase has been entered, the very same problem definition is used for all three cases of torque control. The solution for the backlash traversal can be seen in the beginning of figures 5.1, 5.2 and 5.3. Before the backlash is entered, full torque will simply be output so as to enter the backlash as fast as possible. During the backlash phase, the engine speed is initially accelerated by applying maximum engine torque until its local peak value is achieved. At this point maximum negative torque is applied, effectively braking the engine side so as to "catch up" with the load side at the end of the backlash passage. This reduces the speed difference between the two sides to the specified level, giving a soft impact. When the backlash is safely traversed, the torque phase is entered and engine torque is applied so that the wheel torque reference is reached using the specified criteria.

5.1.2 Derivative damping strategy

As one can see in figure 5.1, the result of the derivative damping strategy is looking a bit too cautious. While there are indeed no oscillations, which is the main goal, there is room for improvement. Looking at the plot of demanded engine torque, one realizes that the torque reduction is probably greater than it needs to be, leading to a significant performance loss. Starting with a higher torque demand and then utilizing the dynamics of the driveline could provide a better torque profile. With a ramped engine torque, the dynamics of the driveline are not utilized at all. Instead, the torque rate of change is simply low enough that no oscillations are induced.

This only shows that you get the results you ask for. Only by defining the problem in the correct way, one will find the intended solution. As previously stated, a big effort when using optimal control lies in designing the problem. As the solution of a simple ramp is what is often used as of today when changing torque levels, this should be improved upon and the optimal control problem must be further modified to achieve that goal.
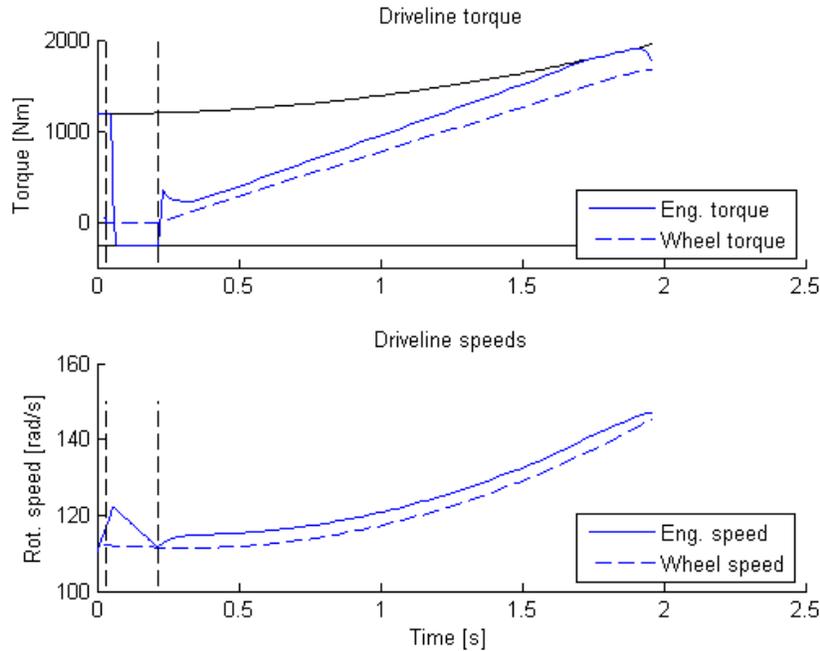
Figure 5.1: The solution to the derivative damping optimal control problem. The backlash is entered at the time of the first vertical line and positive contact is achieved at the second.

5.1.3 Derivative damping with reference following

With the reference cost added, the solution, which can be seen in figure 5.2, looks a bit different. The requested performance boost is achieved, as a large torque impulse is output right at the beginning of the torque control phase. With a higher numerical resolution of the solution, this impulse would have been even more clear. Following this impulse, a higher torque level is achieved throughout the sequence by staying closer to the reference at all times. One can see that the driveline dynamics are now better utilized, as an initial bump in the engine speed is created. This should be compared to the solution in figure 5.1 where a constant speed offset was the result. This speed bump results in a fast build-up of the drive shaft torsion and thus the wheel torque. The speed difference is then slowly reduced, and with that also the torque derivative. The wheel speed fully catches up to the engine speed at the end of the sequence, giving no further torque change.
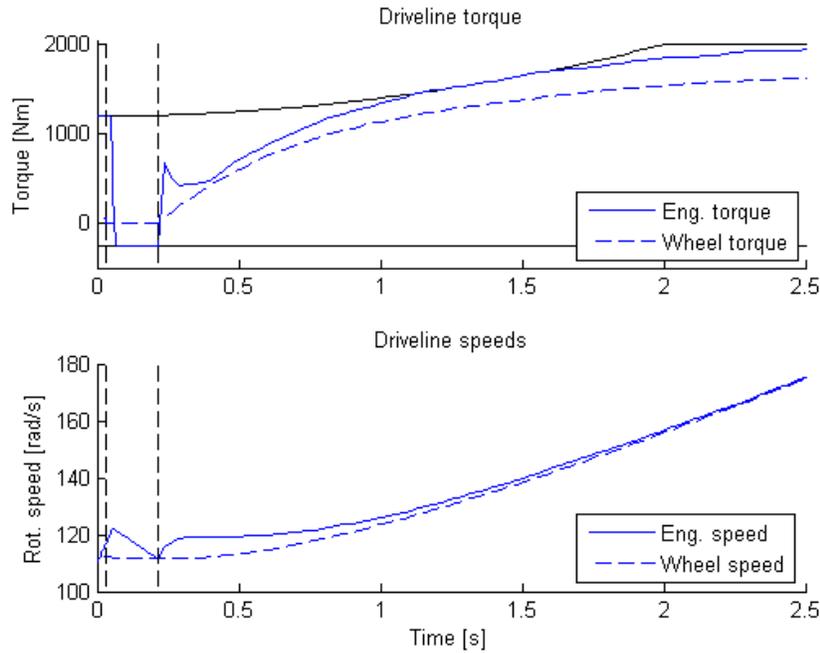
Figure 5.2: The solution to the derivative damping with reference cost optimal control problem.

5.1.4 Equal torque strategy

In figure 5.3, the solution to the strategy with equal torque can be seen. What can be noted is that this solution shows an even more aggressive buildup of

the wheel torque than the one shown in figure 5.2. Initially, a torque step is produced. This should be compared to the previous strategies, that showed an initial impulse. After this, the engine torque is ramped up to the dynamic limit, which it then follows until maximum torque is achieved. As intended, the resulting wheel torque follows the engine torque. There is still no oscillation in the wheel torque, but a plateau is entered when the maximum torque limit is reached. This can be seen as a slight hint of unwanted behavior, as a smooth torque buildup is preferred. A less aggressive solution, with a higher $q_{\text{difference}}$ or lower $q_{\text{reference torque}}$, would however reduce this tendency.
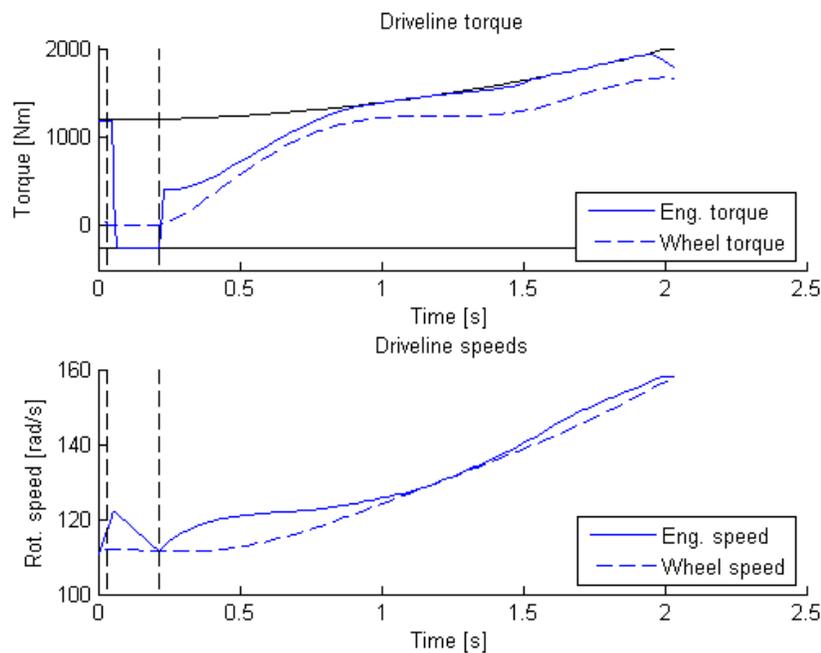
Figure 5.3: The solution to the equal torque optimal control problem.

A comparison of the solutions using different weights in the equal torque optimal control problem can be seen in figure 5.4. The relative size between the two design parameters $q_{\text{reference torque}}$ and $q_{\text{difference}}$ has been changed so that the ratio $q_{\text{difference}}/q_{\text{reference torque}}$ is 5 times greater in the less aggressive solution.

A comparison where different speeds at the contact point at the end of the backlash are allowed can be seen in figure 5.5. In the aggressive solution, the backlash is traversed faster and ends with a large speed difference. Compared to the less aggressive solution, a higher torque has been output, which results in a higher vehicle speed at the end of the sequence. The reason that there are no oscillations even though the backlash traversal is very rough is that torque is output with perfect timing and amplitude right at the contact point. With a
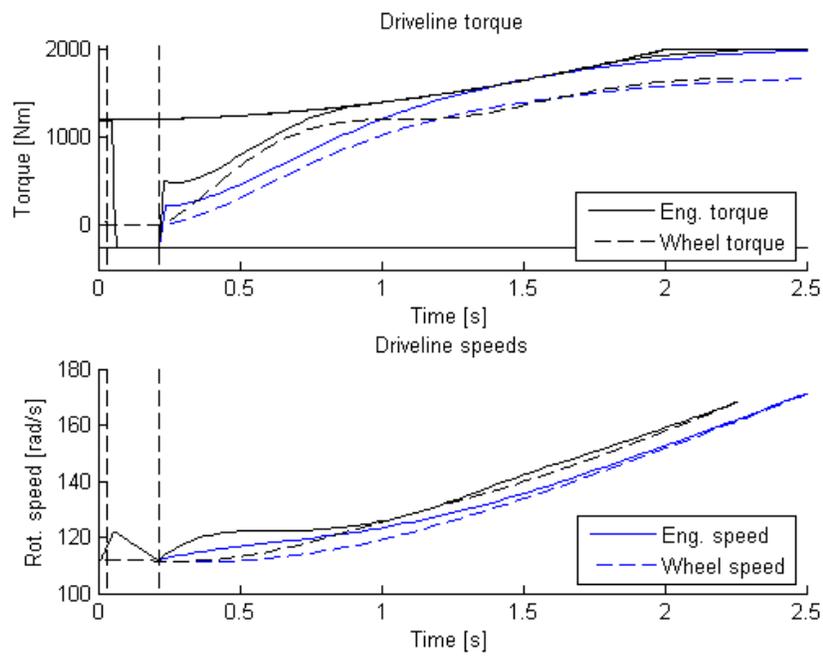
Figure 5.4: Two different equal torque optimal control problems compared, using different weighting parameters. The aggressive solution (higher initial torque) uses a ratio $q_{\text{difference}}/q_{\text{reference torque}}$ that is 5 times greater than the less aggressive solution.

less than torque control, the shunt phenomenon would be observed.
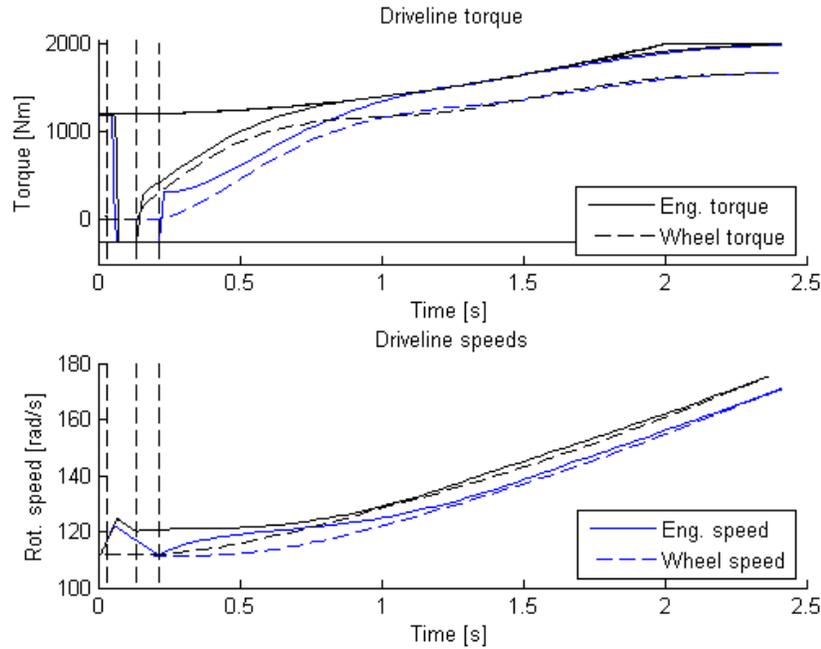
Figure 5.5: Two different solutions to the equal torque optimal control problem is compared, using different values of the $q_{bl\ speed\ difference}$ parameter.

5.2 Optimal solution summary

Different drivelines with different parameters for inertia, stiffness, damping, etc, will give different solutions to these optimal control problems. Changing the design parameters also allows one to tune the solutions, which will give different results. The solutions shown in this chapter does however give some suggestions as to what is achievable and what a good torque profile might look like. Goals to aim for in a strategy used for implementation could for example be to control the speeds relative to each other, to attempt to achieve the timing of the initial torque impulse, to follow the reference torque in a special manner and when in the backlash to succeed with the time-optimal bang-bang control. One should however remember that these optimal control solutions should only be seen as a guidance to what is achievable. There is a large step that needs to be taken between having these optimal control results and implementing successful strategies.

A problem that arises when trying to directly translate the optimal results is that they are achieved for models with fully known parameters. In a real

application, this is not the case, and these optimal open-loop results can usually not be expected nor achieved in practice. This is especially clear in the backlash traversal. While the optimal solution of bang-bang control is simple and intuitive, implementing this strategy in a real application would (currently) not be possible. The strategy relies on knowledge of the backlash size, which is not available. There is no way to know this size exactly, as the backlash in the driveline is composed of several different components, as described in section 3.1.

One way to improve the situation and make the strategy implementable would be to implement a backlash observer, such as in [6, 7]. In [7] however, one of the conclusions regarding such an estimator was that a too high bandwidth was required, which further resulted in too much processing power needed for the observer to be run online in the ECU. Where this problem came from and whether it was due to their specific implementation is not mentioned. Other conclusions stated in their work was that it was doubtful if the backlash estimation actually provided any major improvements. With these comments, the implementation of such an observer will not be considered for this work, as the vehicle environment in their work was very similar to the setup here.

An alternative solution to the backlash problem is to make a conservative assumption of the backlash size and then control the relative speed to zero at about the impact point. This would still give a rough impact, but could mean an improvement to the open-loop strategies used today. The lower quality regarding sampling speed of the wheel speed signal also needs to be kept in mind, but as the wheel speed is approximately constant during the short time of a backlash passage, this is assumed to be less of a problem.

Chapter 6

Torque control strategy

As is previously stated it is important (or at least highly beneficial) to have the implemented strategy be as simple as possible, so that the processing power and memory usage of the ECU is saved for other important tasks. The optimal control results have provided suggestions as to what kind of torque control strategy that is suitable. When considering the implementation of this strategy, some conclusions can be drawn as to which one is the most preferable. A strategy based on one of the three optimal control problems can be readily described for implementation. The suggested and implemented strategy is explained in this section.

6.1   The implemented strategy

It is assumed that the output engine torque is approximately the same as the one requested from the engine, so that any delays and accuracy issues are disregarded, as is stated in section 3.1.1. The engine torque can therefore be considered known. Under the assumption that the engine inertia is greater than the other inertias on the engine side combined, an approximation to the currently transmitted wheel torque is already calculated in the ECU. The signal is calculated basically using the acceleration of the engine and the known engine inertia, together with the previously output engine torque compensated for any delays. Rearranging equation 3.1 and disregarding the friction term gives

$$T_w^{calc} = i(T_e - J_{engine}\dot{\omega}_e)$$

which will give the approximate wheel torque when the mentioned assumption about the inertias hold. As is stated in section 3.1, this is the case. Also the transmitted wheel torque can thus be considered (approximately) known by calculation. With this, both signals needed for the equal-torque strategy are available. Due to the idea's intuitivity and simplicity - keeping the torques equal - a strategy based on these two signals is the method of choice.

The reasoning behind the strategy is based on applying torque in a way so that it cannot differ much from the torque currently transmitted in the driveline. In this way, no large, sudden torque changes can be made, and then no shunt and shuffle will be induced. Therefore, a smooth torque build-up to the requested level should be achieved. This strategy will in essence result in a modified rate limiter, that instead of using the old input signal uses the previous output to limit the following input.

To achieve the small difference between the two torques, the next sample input torque is calculated simply by applying an offset to the transmitted wheel torque, according to

$$T_e(k) = T_w^{calc}(k-1)/i + f(k)$$

or in state space variables

$$u(k) = z(k-1)/i + f(k)$$

where f is the chosen offset. The result is that the difference between the two torques can be (reasonably) freely chosen by designing the offset. The offset is in its simplest case a single design parameter, but is modified to tune the torque profile. The initial approach here is to increase the torque offset as the maximum torque limit is approached, following the results from the optimal control solutions. Other examples of tuning the offset, one could want different offsets at different torque levels; a specific offset when in the backlash; to alter the size of the offset dynamically to react to certain situations and so on.

By using the wheel torque approximation, a lot of information can be gained. As all applied torque will be used to accelerate the engine when there is no connection between the two sides of the driveline, the wheel torque must then be zero. When the calculated wheel torque signal is sufficiently close to zero, one can therefore assume that the driveline is in the backlash. With this knowledge, one can also implicitly know when the backlash have been traversed, seeing an increase in the wheel torque signal. When this happens, it is known that the torque applied after this point will actually be transferred through the driveline and will not be used only to accelerate the engine side so that it roughly collides into the load side, resulting in shunt behavior.

6.2   Feedback analysis

The implemented strategy results in a feedback system as the resulting wheel torque is used to calculate the following engine torque to be output. The feedback, however, can not be considered as a negative feedback as is used in the standard case, because the calculated input signal (engine torque) is continuously increased upon an already increasing output signal (wheel torque). The reason that this approach works is that the strategy is not used as an actual means to control the system, but only to limit the output below an even higher torque request. The output torque cannot exceed the requested level, as other

modules in the ECU will take over control as soon as the engine torque passes the torque requested by the accelerator pedal. Furthermore, the hard constraints mentioned in section 3.4 is also limiting the output torque.

6.2.1 Backlash effects

When applying torque following this strategy, it is clear that the backlash will not be traversed in an optimal way. The fact is though that the simple strategy still provides some of the benefits that an active strategy would achieve. If a lot of torque is applied before the two sides of the driveline is in contact, the shunt phenomenon appears. With this strategy, the demanded torque is limited to a low value until the wheel torque starts to build up. This can only happen after contact is achieved, as the wheel torque must be zero when there is no contact. The result is that the demanded torque will remain at a low level until the backlash is definitely traversed. Only after this, the demanded torque is allowed to increase. The shunt problem is not eliminated, but the effects of it will be limited and can be controlled by changing the size of the offset during the backlash traversal. If a fast but less comfortable backlash passage is wanted, a high offset can be chosen. A lower offset gives a more comfortable but also slower traversal.

6.2.2 Torque control effects for tip-in

The effect of the implemented strategy is that no more torque than the driveline can handle is applied. No energy is temporarily built up and stored in the driveline flexibility, which means that no such energy can be released to create oscillations. Even though it intuitively might seem that limiting the torque like this would decrease the performance considerably, taking a long time for the torque to reach the maximum value, this is not the case. As torque is demanded "in phase" with the driveline, maximum torque can still be reached relatively fast. The time to achieve the reference torque is tuned simply by choosing the gain of the calibrated offset.

What turns out to be a problem that causes oscillations is when the engine torque meets the torque limits at a rough angle, so that a torque level higher than is available is demanded. This will happen when a too large offset is used right before the impact point. To solve this problem, the offset could be modified to decrease when the distance to the torque limit becomes sufficiently small. Then the offset will decrease to a level that will not make the demanded torque larger than the available torque. By tuning this torque decrease correctly, the torque build-up should smoothly enter the maximum level and remain there without oscillations.

6.2.3 Delay margin

When a feedback system contains time delays, some problems might occur. If the delay is sufficiently small, no special action usually needs to be taken. If the

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delay is somewhat greater, the performance of the control might degrade. If the delay is large enough, the result might be an unstable feedback loop. Therefore, a measure for when this instability occurs is sought.

When a higher gear is engaged, the gear ratio in the driveline is lowered. This causes the driveline to become stiffer, thus increasing the natural frequency of the driveline, according to equation 3.6. With a higher frequency, less margin for delay exist. The eigenfrequency of the driveline is calculated for each gear according to equation 3.6 and is shown in figure 6.1. One can see that it remains low for the low gears and then grows rapidly with higher gears.
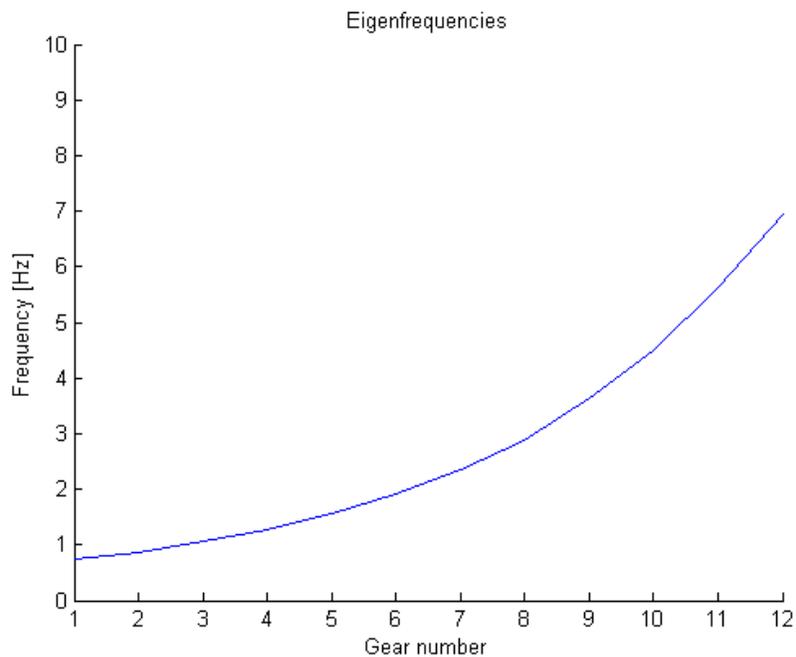
Figure 6.1: The eigenfrequency for each gear.

As a complement to figure 6.1, see figure 6.2. In it, also the delay margin for the driveline is shown. This was (as well as other stability margins) calculated using the MATLAB command 'allmargin', see [21]. Along with the delay margin the delay from requested to produced torque, called the injection delay, and the transmission delay over the CAN bus is plotted. One can see that the delay margin decreases when higher gears are used. This should however not significantly affect the behavior as long as the CAN bus delay doesn't exist. As we are then well below the margin, the conclusion so far is that while the output engine torque is calculated close to the engine, so that any delays are minimized, there is a margin great enough to allow for feedback. When large delays exist (because they are necessary) however, problems might occur with

stiff drivelines at higher gears. The applicability of feedback needs to be further investigated in those cases.
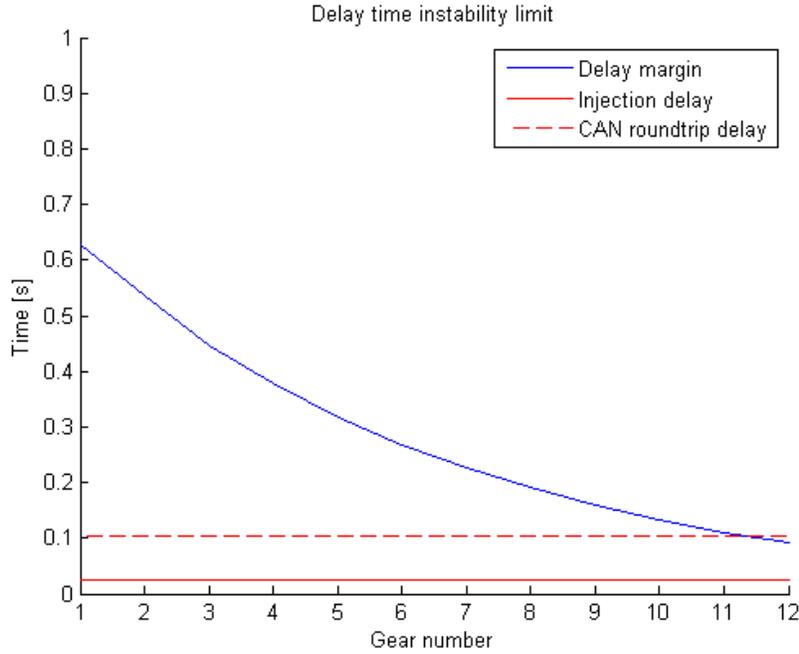
Figure 6.2: The delay margin decreases with higher gears. The margin should be large enough when only the injection delay exist. Adding a CAN delay however, problems might occur at higher gears.

6.3   Simulation

The implemented strategy is here simulated. In these simulations some delay effects neglected during the modeling, as mentioned in section 3.1.1, are introduced. In all of the simulations the delay from requested to output torque exists, the so called 'injection delay'. This delay is modeled as

$$t_{injection} = \frac{\theta_{crankshaft}}{\omega_m} + 20\,\mathrm{ms} \qquad (6.1)$$

where $\theta_{crankshaft}$ is simply a uniformly distributed random number representing the crankshaft angle and ω_m is the engine speed. $t_{injection}$ will range between 20 ms and some 30 ms. In some simulations, also an additional significant delay is added. This is used to model transmission over the CAN bus.

In figure 6.3 a simulation of the strategy can be seen. The vehicle is initially traveling at approximately 11 km/h and gear 4 is engaged, freely rolling. Note

the initial delay in the engine torque right at the start of the sequence, marking the injection delay from requested to produced torque. As can be seen, a low torque is requested as long as the driveline is not connected, up until 0.28 s. When there is torque transferred through the driveline, the output engine torque is allowed to increase. The maximum torque limit is reached with a small angle, therefore softly, causing only a hint of an oscillation.

Comparing this simulation of the suggested strategy to the solution to the optimal control problem seen in figure 5.3, one can see that they bear a resemblance. The difference lies in the backlash traversal, which here cannot use the bang-bang approach because of the stated backlash uncertainties.
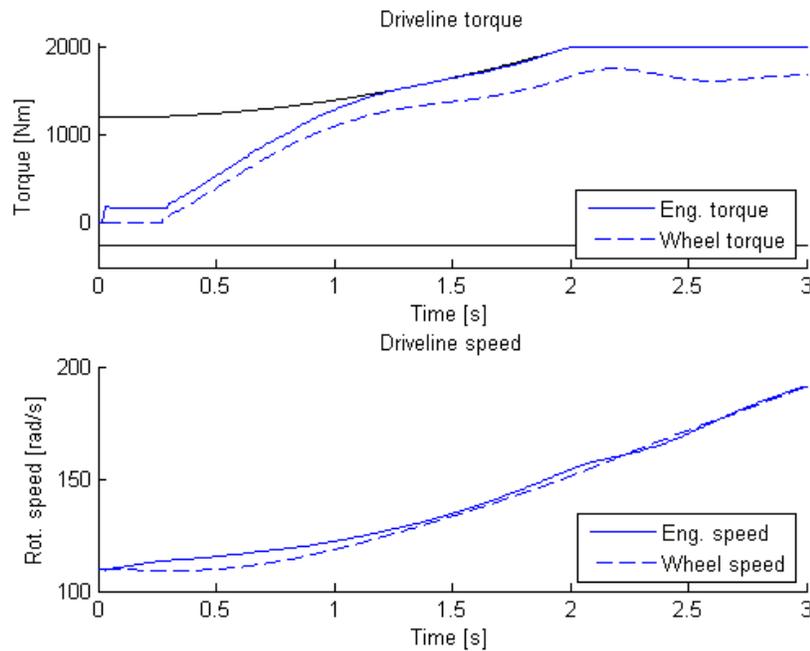
Figure 6.3: A simulation using the implemented strategy.

In figure 6.4, the same simulation is performed. The difference here is that a higher overall gain for the offset is used. As can be seen, the maximum torque limit is reached at a greater angle. This induces an oscillation as the output engine torque can not keep accelerating the engine because of the limitation. The engine speed will slow down as the drive shafts untwist, which will reduce the transmitted torque. When the wheel torque has decreased, there is once again torque available for an offset and the output engine torque and thus the transmitted torque will increase up to the maximum level once again.

From these simulations, it can be seen that this simple limitation set on the control signal can achieve the wanted behavior as long as the offset is correctly
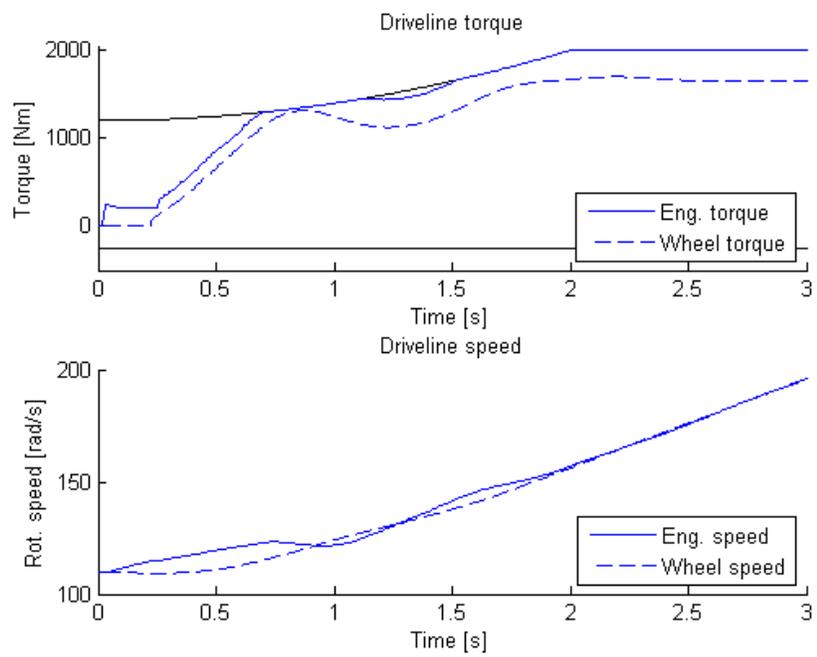
Figure 6.4: A simulation using the implemented strategy with a higher gain.

tuned. With already available signals and an implementation requiring no complex calculations, the goal of no oscillations may be achieved. The strategy will therefore be further tested by implementation in a vehicle.

6.3.1 Simulation of LQ control

As a comparison to the suggested strategy, an LQ-controller for the torque control phase is developed, i.e. when the driveline is connected on the positive side of the backlash. A cost on the derivative is applied and the system is extended to include integral action so that the reference torque will be achieved. It should be remembered that this controller is (linearly) optimal, and a similar controller (LQ/LQG) would have probably been used if only the torque control phase was considered and no backlash existed. In [1], an LQG-controller is used both for speed control and for gear change control.

What can be seen in figure 6.5 is that the nonlinear switching between the backlash and contact causes problems, as is expected. When starting in the backlash, the linear properties that the LQ-controller was designed for does not hold. Torque will be output but the system will not behave as expected. This causes an oscillation in the control signal.
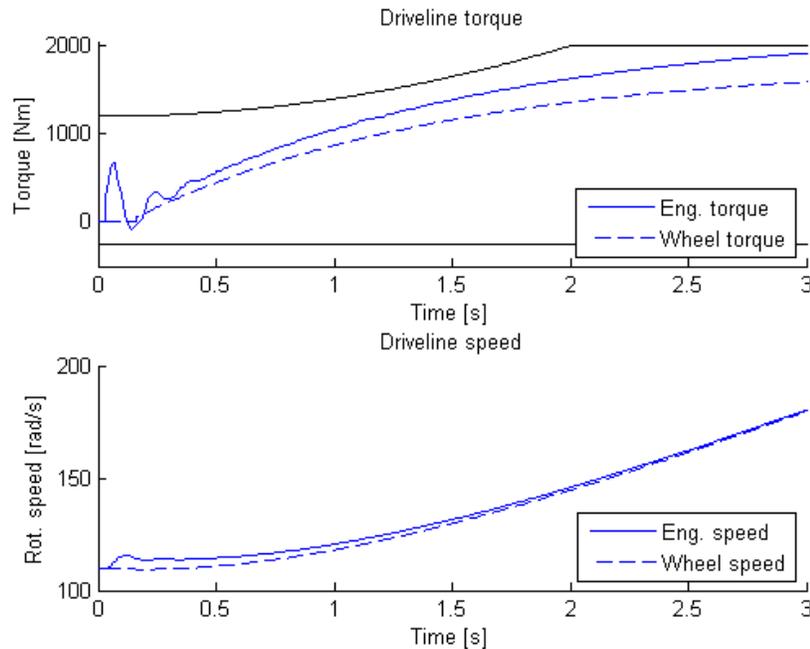
Figure 6.5: An LQ controller is used for reaching the maximum torque. A small oscillation in the engine torque can be seen, caused by the backlash.

When the LQ controller uses a higher gain (as in a lower cost on the control

signal), the oscillation is worse. This can be seen in figure 6.6. These two examples display the problems due to nonlinearities caused by the backlash when linear control is used.
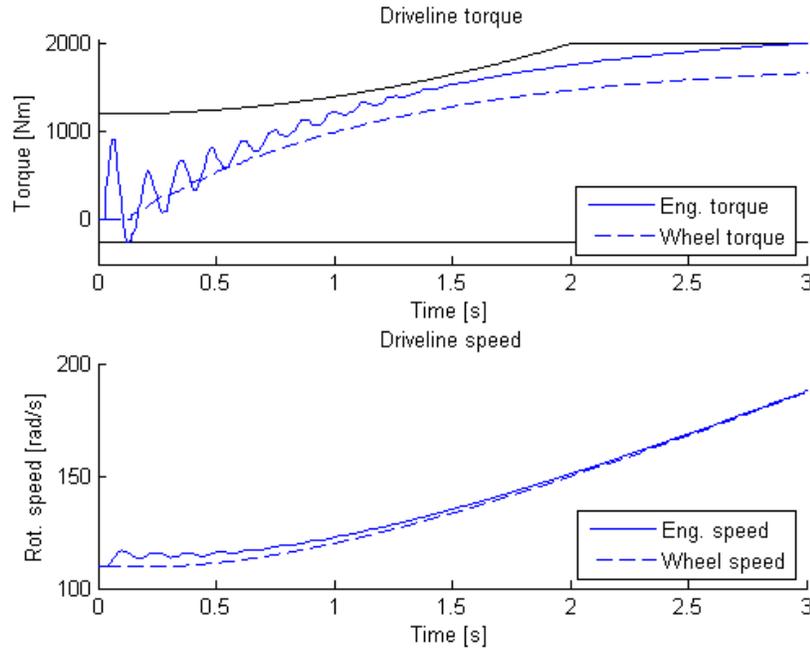
Figure 6.6: An LQ controller is used for reaching the maximum torque. The higher gain gives a greater oscillation in the engine torque.

6.3.2 Delays added to the loop

When adding an additional delay to the feedback loop, the strategy's performance deteriorates, as expected from the analysis in section 6.2.3. This delay is added to represent transmission over the CAN bus. These transmissions occur when signals are sent between different control units in the vehicle, connected via CAN. Should another unit control the engine torque using the suggested strategy, the following problems will in some cases occur.

In figures 6.7, 6.8 and 6.9, different delays are added to the simulation. The gear used in these simulations is number 10. The injection delay still exists, and ranges between approximately 20-30 ms, as described by equation 6.1. One can see (figure 6.7) that the strategy works as intended when there is no delay also at a higher gear. The torque is ramped up to the maximum limit and stays there for the rest of the sequence, without oscillations.

When a delay of 80 ms is added to the loop, the engine torque begins to oscillate (see figure 6.8). This is simply due to the fact that too old values of
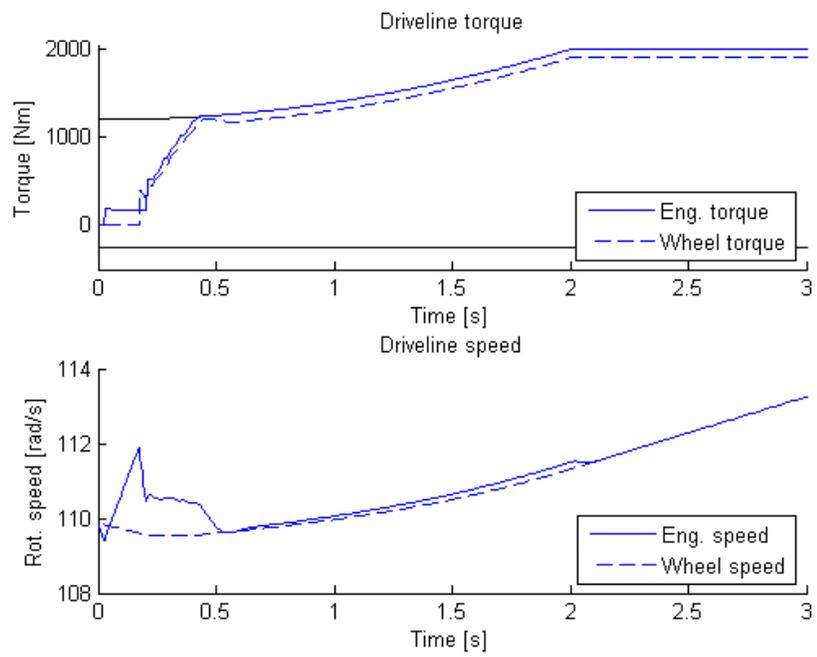
Figure 6.7: The strategy is used when gear 10 is engaged. Only the injection delay exist.

the wheel torque is used when calculating the next engine torque to output. 80 ms is an approximate value reasonably close to how long a real round-trip (two transmissions) over CAN would take. This behavior would not be acceptable.
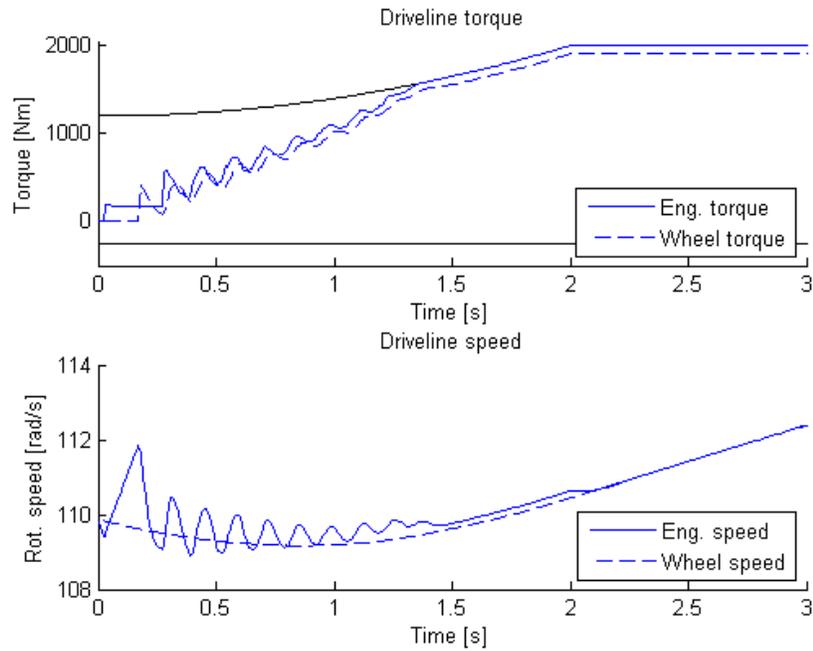
Figure 6.8: The strategy is used when gear 10 is engaged. In addition to the injection delay, a delay of 80 ms is added to the loop to represent transmission over the CAN bus. A consistent oscillation can now be observed.

To further point out the problems that a delay might cause, the added delay is increased even further, to 150 ms. A 150 ms delay is too large to be reasonable - far greater than the transmission time over the CAN bus - but clearly displays the mentioned instability. The result is shown in figure 6.9.
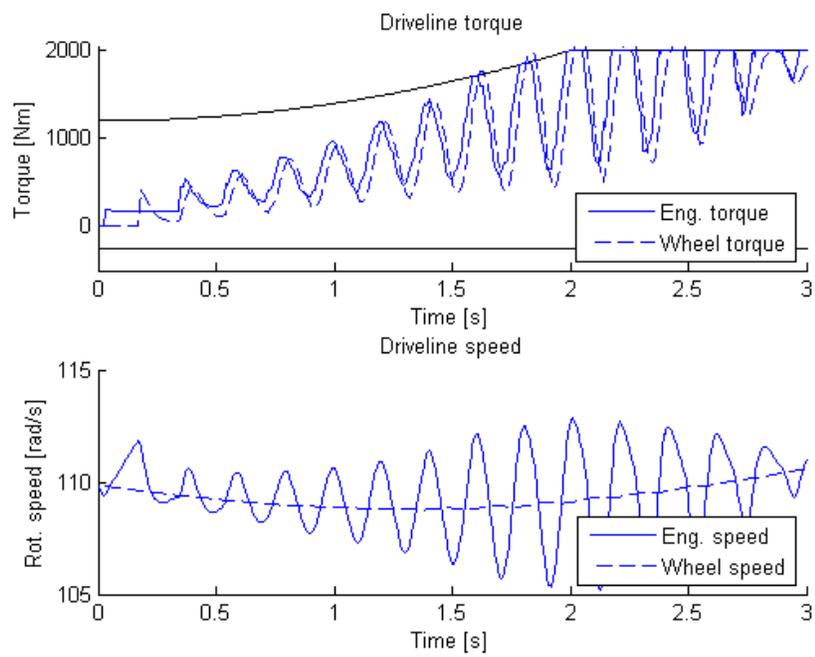
Figure 6.9: The strategy is used when gear 10 is engaged. A delay of 150 ms is added to the loop, in addition to the injection delay. An oscillation with increasing amplitude can be observed, displaying an unstable loop.

Chapter 7

Truck measurements

Measurements of the suggested strategy (figures 7.1, 7.2 and 7.3) together with examples of the tip-in functionality used today is presented. The strategy is active and limits the applied torque from the initial torque demand up until the maximum torque limit is reached. As the signal that is calculated in the ECU and is used as an approximation to the wheel torque is not filtered for our application, the calculated torque demand will neither be. The reason for not filtering is simply to avoid any delay effects. This unfiltered torque demand does seem to work well in this application, but whether a noisy torque demand has any adverse effects on surrounding systems needs to be further investigated. The amount of delay that this strategy can handle in practice also needs to be further investigated, but some conclusions can be drawn from the analysis in section 6.2.3.

In figure 7.1, the implemented strategy is active between times 190.1 and 190.5 s, using an offset of approximately 100 Nm ramped up to 200 Nm when the maximum limit is reached. During the initial backlash traversal, a minimum torque level of 100 Nm is applied to avoid a negative torque demand. This holds until the backlash is passed and the two sides of the driveline are connected. At this instance, when the wheel torque starts to increase and is fed back to the engine, the demanded engine torque starts to increase and will do so until the maximum limit is reached. No oscillation can be observed.

In figure 7.2, the same situation as in figure 7.1 can be observed. Also here the torque offset is ramped up from 100 Nm to approximately 200 Nm when the maximum limit is reached. Neither in this situation any oscillation can be observed.

In figure 7.3 gear 8 is used instead of the previous 6. Also here, the torque offset is ramped from 100 to 200 Nm over the course of the tip-in. When reaching the maximum torque limit, a small torque oscillation occurs. This is due to the fact that the requested torque level can not be reached, because of the current limitations. The strategy should here be modified so that a softer approach to the maximum limit is achieved.

Comparing these results to the strategy of today, one can see that an im-
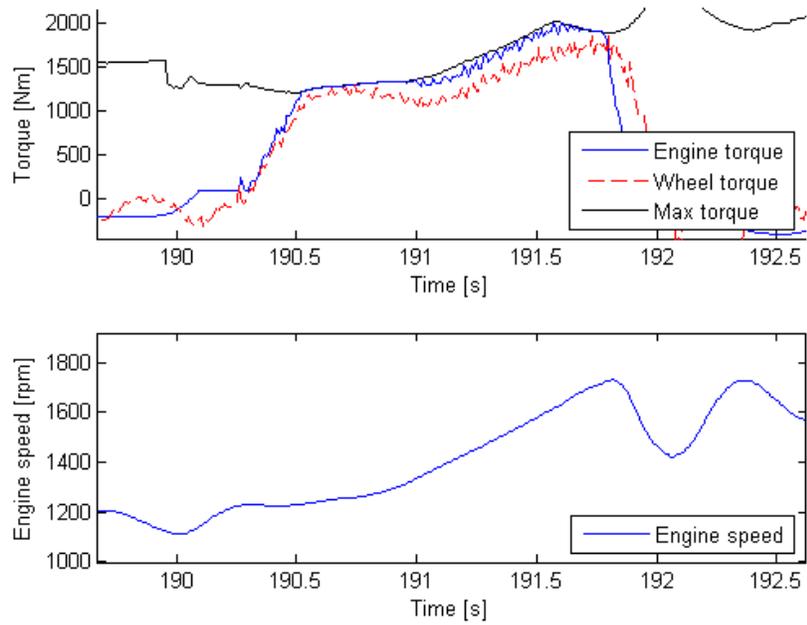
Figure 7.1: The suggested strategy is active and limiting the demanded torque between 190.1 and 190.5 s. The vehicle is traveling at 16 km/h using gear 6 when the accelerator pedal is pressed.
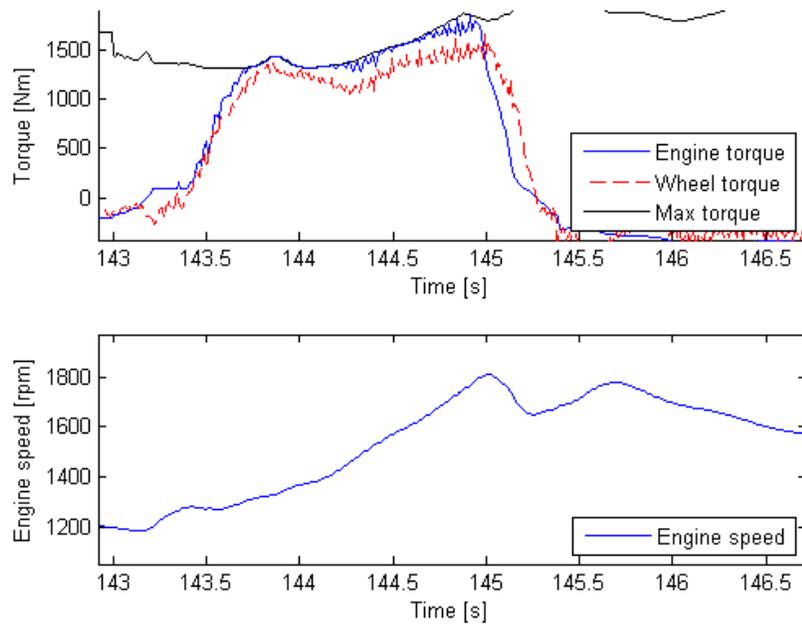
Figure 7.2: The suggested strategy is active and limiting the demanded torque between 143.2 and 143.7 s. The vehicle is traveling at 16 km/h using gear 6.
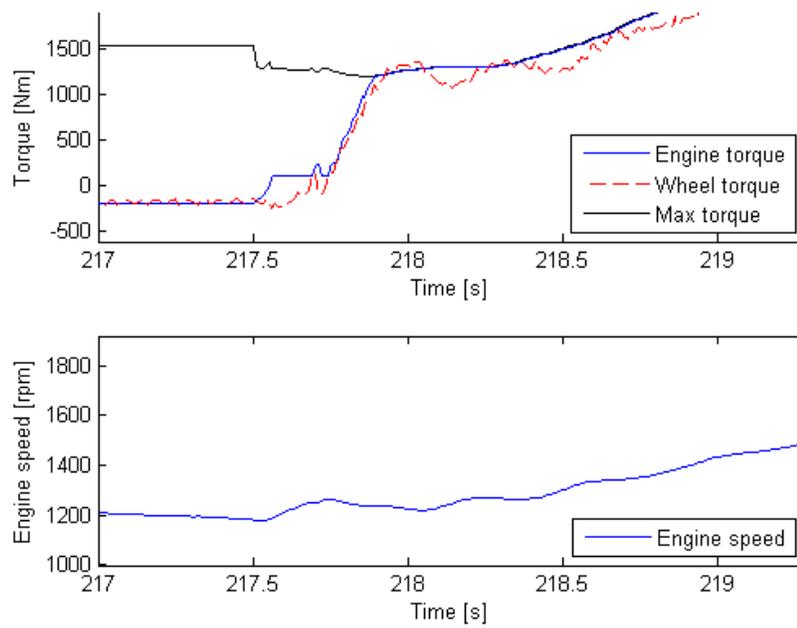
Figure 7.3: The suggested strategy is active and limiting the demanded torque between 217.6 and 217.9 s. The vehicle is traveling at 25 km/h and gear 8 is engaged. Some oscillation in the wheel torque can be seen, caused by a too large angle between the engine torque and the maximum torque.

provement regarding the oscillations in the wheel torque have been achieved. Figures 7.1 and 7.2 where the suggested strategy is used display the same driving situation as in figure 7.4 where the strategy of today is used and they can thus be compared. By looking at the wheel torque, one can see that in figure 7.4 a high engine torque is output while the two sides is not yet connected. This, combined with "too much" torque being output to the driveline thereafter, causes the following oscillation.
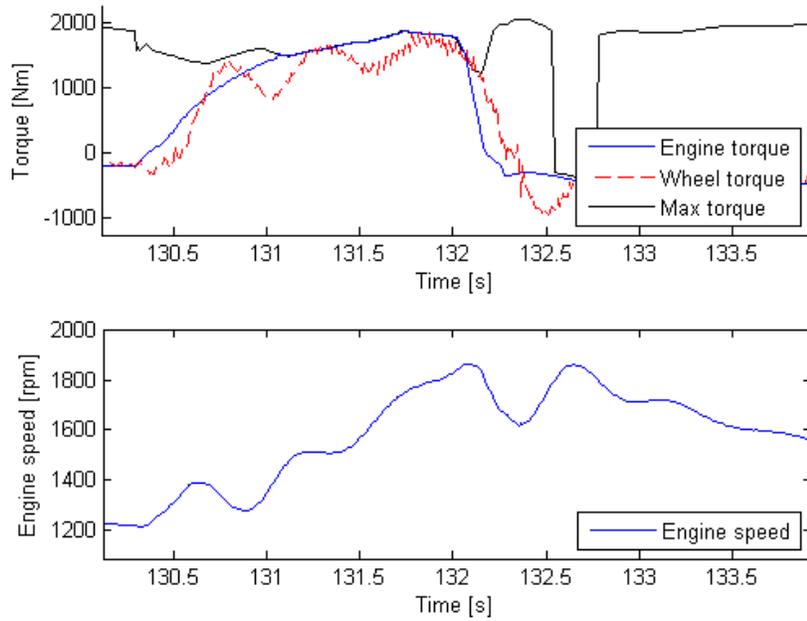
Figure 7.4: An example of the strategy used today. The vehicle is traveling at 16 km/h and gear 6 is engaged. Even though the torque demand is shaped to avoid oscillations, too much torque is demanded in the interval 130.3 to 130.6 s.

Another example of today's strategy is seen in figure 7.5. Once again, the high torque level applied during the backlash gives a rough impact between the two sides of the driveline, causing a shunt. The high torque level thereafter does not help the situation, and so the wheel torque oscillates.

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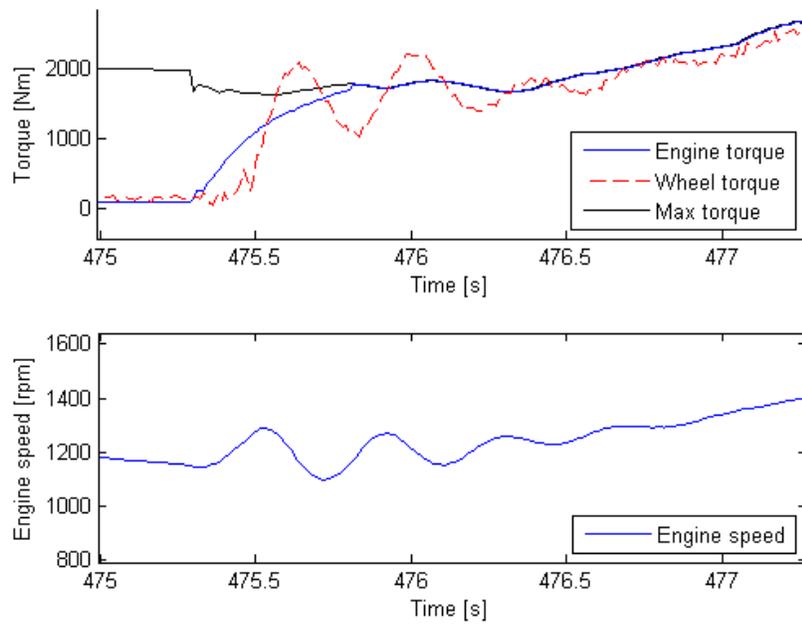
Figure 7.5: Another example of the strategy used today. The vehicle is traveling at 23 km/h and gear 8 is engaged. Too much engine torque is output in the interval 475.3 to 475.5 s. The rough backlash traversal and the high torque level causes an oscillation.

Chapter 8

Discussion and conclusions

With the suggested strategy, some of the main goals of this work have been achieved. As shown in section 7, the strategy manages to prevent oscillations where the strategy of today does not. The strategy has thus proven to be able to provide the requested increase in comfort, seen as reduced oscillations in the wheel torque. In addition to this, it also manages to do this in a way that doesn't reduce the performance. Less torque is initially output, but this is compensated for by then reaching the maximum torque faster than the strategy used today.

As previously stated, this torque control strategy doesn't imply a lower performance, measured as the time it takes to reach maximum torque. As long as the gain of the calibrated offset is chosen large enough, this time can be tuned to meet today's implementation, still without inducing oscillations. However, experienced drivers have mentioned a feeling of a lack of vehicle response using the strategy. As the initial shunt that usually comes when applying torque by pressing the accelerator pedal is now removed, the driver will no longer get this feedback. The time it takes for the driver to feel that the vehicle responds has therefore increased. However, what it is exactly that introduces this feeling is as of now unclear. Possible causes could be the missing initial shuffle that influences the cabin, the sound of the engine being delayed or simply the softer torque build-up throughout. What is the actual cause and how to add to the response feeling needs to be further investigated.

The delays shown in simulation (section 6.3.2) will need to be considered if the strategy is to be further developed. As is discussed in section 6.2.3 the delays should for this driveline not result in any major complications as long as the calculation of engine torque is done close to the engine so that any transmission delays are avoided. This is the case today when the torque requested from the accelerator pedal is determined, and therefore there should be no problems in controlling tip-ins and tip-outs.

The delay problem becomes significant first when another unit wants to request torque using the suggested strategy, so that the torque request has to be transmitted over a slow channel. The sensitivity to delays however occur mostly on higher gears where the driveline acts stiffer. When the driveline is

stiffer, the problem of discomfort (due to shunt and shuffle) is not as significant, due to the higher frequency of oscillation which will not affect the driver as much. Therefore a strategy for damping oscillations, such as the one suggested here, is not as important for those higher gears. Maybe the suggested strategy can be complemented with some open-loop strategy not utilizing feedback when delays exist and discomfort is less of a problem. Then comfort could be achieved where needed, and the problems of delays could be avoided when they exist. As is previously stated, also the functionality used today manages to prevent oscillations in most cases.

Finally, it should be clearly stated that the two examples of the implementation of today (figures 7.4 and 7.5) are somewhat extreme cases where this implementation cannot manage to prevent oscillations. Most of the tip-ins today look far better than this. However, you also want to avoid these exceptional cases where failures occur, so that you can achieve good comfort every time the driver presses the accelerator pedal. With the suggested strategy, tuned to have the correct offset parameters, no failed tip-ins have so far been observed.

8.1   Future work

Some suggestions for future work that have arisen are here listed;

- As the backlash traversal is not very sophisticated, there is room for improvement. A well designed backlash estimator that estimates both the current backlash size and position sufficiently well, such as can be found in [6], would help achieve this. Then the offset could be controlled so that the impact between the two sides become even softer. Active strategies following the time-optimal bang-bang approach would also be an alternative to be implemented.

- Much research using linear, nonlinear, switching, optimal, hybrid and maybe other fields of control analysis for controlling drivelines with and without backlashes has been and is still being performed. One common approach for control of a system having a backlash is for example to analyse the problem using describing functions. A lead-compensation can then be designed to avoid any limit cycles. The result is a phase-advance controller that better handles the backlash. This and several of the other suggested control strategies should be compared to and evaluated against the simple strategy suggested here. While managing to improve the situation as of today, more refined and robust strategies that have been carefully analyzed can certainly exist. This has to be determined.

- An investigation of what vehicle behavior that is desired from the users point of view is needed. It is often easy to specify hard measures; time to max, amplitude of oscillation, rate of change and so on, much like what is done in this work. Another approach would however help when developing functionality that has a direct impact on the end users subjective impressions. When the goal is to achieve greater comfort, a "softer" measure that

handles comfort directly should also be developed if it doesn't already exist. Then these softer measures could somehow connect to the hard values directly, for the sake of convenient development as that is what is conventionally used. Without these measures, it is always somewhat unclear what actually gives an improvement. A clear example comes with the result presented in this thesis; even though the amount of oscillations are removed or reduced, there can instead be a feeling of lack of response. The users evaluation thus becomes ambiguous. While some think the gained comfort is an improvement, others think the lack of response means a degradation. What needs to be determined is in what direction the hard measures should be tweaked so as to improve on the soft measures.

- How to achieve a feeling of response to solve the problem mentioned in the previous discussion needs to be further investigated.

Appendix A

Abbreviations

ABS	Anti-lock Braking System
CAN	Controller Area Network
ECU	Engine Control Unit
LTI	Linear Time-Invariant
PWA	Piecewise Affine (System)

Table A.1: Abbreviations used throughout the thesis.

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Title and subtitle

Design of a Torque Control Strategy for Enhanced Comfort in Heavy Trucks (Design av dynamisk momentreglering för komfortabel framdrivning av fordon)

Abstract

Heavy trucks vehicles by design often suffer from issues causing driver discomfort. That discomfort is here considered caused by oscillations in the driveline, which are affecting the driver. The driveline oscillations – sometimes referred to as shunt and shuffle - exist because of the sheer amount of power in a heavy truck engine, combined with a driveline where the components are relatively weak. If this engine power is simply output to the driveline without control, the driveline will twist and bend, causing oscillations.

The purpose of this work is to investigate how engine torque can be controlled when large torque changes are requested. The specific situation of interest is when the driver fully depresses and releases the accelerator pedal, referred to as a tip-in and tip-out. The goal is to device a strategy that in this situation prevents any driveline oscillations and thus improves the driving comfort.

What is presented is an investigation of some different strategies for applying torque and what seems achievable, using optimal control. This investigation then leads to a suggested alternative to today's strategy of applying torque. The suggested strategy is implemented in the ECU and tried in different vehicles. This first attempt at an implementation proves itself by reaching the target (maximum) torque in the same time as the conventional strategy, indicating no reduced performance. In several cases, large improvements can even be seen. Even while reaching the target level faster, the strategy manages to reduce the oscillations in the driveline. The goal of this work can therefore be considered achieved.

While the suggested strategy works well in many cases, much work is still required to get it fully functional. New problems have been posed, regarding the subjective notion of how a driver actually wants a tip-in to feel. This might then lead the following research in completely different directions. Seen from a bigger perspective, the main result of this thesis is the fact that only with little effort in simple ways, great gains can be achieved.

Keywords

Classification system and/ or index terms (if any)

Supplementary bibliographical information

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