Institutionen för systemteknik
Department of Electrical Engineering

Examensarbete

Robust Torque Control for Automated Gear Shifting in Heavy Duty Vehicles

Examensarbete utfört i Fordonssystem
vid Tekniska högskolan i Linköping
av

Henrik Abrahamsson & Peter Carlson

LiTH-ISY-EX--08/4182--SE

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Titel Title

Robust Momentreglering vid Automatiserad Växling i Tunga Fordon
Robust Torque Control for Automated Gear Shifting in Heavy Duty Vehicles

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Sammanfattning
Abstract

In an automated manual transmission it is desired to have zero torque in the transmission when disengaging a gear. This minimizes the oscillations in the driveline which increases the comfort and makes the speed synchronization easier. The automated manual transmission system in a Scania truck, called Opticruise, uses engine torque control to achieve zero torque in the transmission.

In this thesis different control strategies for engine torque control are proposed in order to minimize the oscillations in the driveline and increase the comfort during a gear shift. A model of the driveline is developed in order to evaluate the control strategies. The main focus was to develop controllers that are easy to implement and that are robust enough to be used in different driveline configurations. This means that model dependent control strategies are not considered.

A control strategy with a combination of a feedback from the speed difference between the output shaft speed and the wheel speed, and a feedforward with a linear ramp, showed very good performance in both simulations and tests in trucks. The amplitude of the oscillations in the output shaft speed after neutral engagement are halved compared to the results from the existing method in Scania trucks. The new concept is also more robust against initial conditions and time delay estimations.

Nyckelord
Keywords

driveline, modeling, torque control, automated manual transmission, heavy truck, gear shifting

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Contents

Chapter 1

Introduction

High efficiency, low fuel consumption and good driving comfort are keywords for a modern heavy truck. An automatic transmission has the benefits of good driving comfort and fast gear shifts. Unfortunately, it is expensive and it has a low efficiency and a short lifetime in heavy trucks. The benefits of a manual transmission are better efficiency, lower costs and longer lifetime. The drawbacks are longer gear shifts, clutch wear and that the fuel economy is very dependent on the driver. It is desired to have a transmission with the benefits from both types and with none of the drawbacks. One solution to these problems is the Automated Manual Transmission (AMT). This type of transmission gives you the high efficiency of the manual transmission but also the comfort of an automatic transmission. Scanias AMT is called Opticruise which replaces the manual gear lever with pneumatics and a control system. To perform a gear shift, first the transmission torque must be controlled to zero to be able to disengage the gear. This is done using engine torque control. When the neutral gear is engaged the engine speed is controlled to fit the requested gear. The new gear can be engaged and the torque is restored to the driver's demand, see Figure 1.1.

A critical part of gear shifting with Opticruise is the torque control before disengaging a gear and after engaging the new gear. In order to achieve high efficiency it is important to minimize the time needed for a gear shift without exciting driveline oscillations. The oscillations in the driveline may also lead to problems with disengaging the gear and synchronizing speeds. Therefore it is of high importance to control the torque to zero in such way that the total time for gear shift is minimized without exciting driveline oscillations or making it uncomfortable for the driver. Today's solution works reasonably well but it is of great interest to investigate different solutions that may work even better.

1.1 Objective

The purpose of this thesis is to investigate different control strategies for the torque controller. The problem is to find a strategy that is robust against different types of trucks (with different engines and drivelines) as well as parameter errors. A

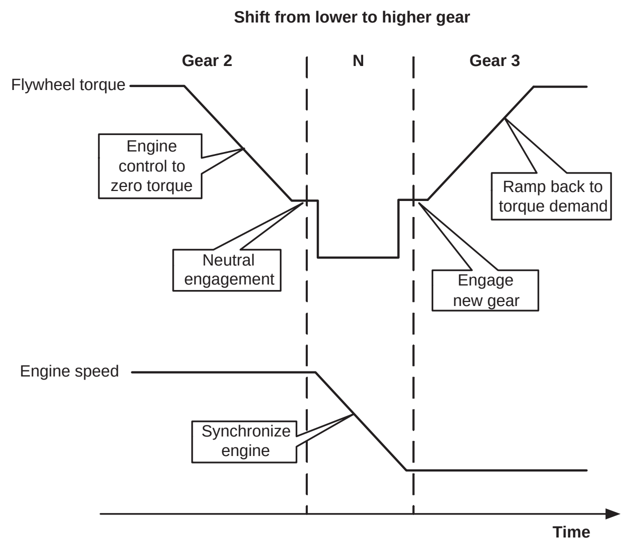

Figure 1.1. Gear shift principle.

plant model of the driveline shall be implemented in Matlab\Simulink in order to simulate the different control strategies. Different types of driveline configurations shall be tested in order to evaluate the robustness. Different types of open-loop controllers that controls the engine torque shall be tested during gear shifting. A closed-loop control using the existing sensors shall be implemented in order to compare this to the existing open-loop strategy. An investigation shall be made if a new torque sensor on the input shaft of the gearbox can improve the results.

1.2 Assumptions and Limitations

Driveline dynamics and gear shifting is a big area, this thesis can only cover a small part of it. In this section the assumptions and limitations of this thesis are listed. If nothing else is stated, these limitations are valid in the whole thesis. The clutch is assumed to be engaged all the time, i.e. the disengaged clutch is not modeled. Wheel slip is not considered. Only the engine control during shift from a gear to neutral gear is considered, i.e. the first phase in Figure 1.1. Only torque control from a higher towards a lower torque is considered because this is the driving situation where the largest shaft torques are found and is most difficult to control. Also the conversion ratio of the gearbox affects how large the shaft torque become, so this thesis will concentrate on lower gears. Only trucks with two driving wheels are considered, i.e. there are only one propeller shaft and two drive shafts. This thesis will focus on simple robust controllers with few tuning parameters to be able to use the controllers in a large variety of drivelines. Therefore controllers that use a model are not considered.

1.3 Notation

The table below summarizes the parameters used in the plant model.

Parameter	Description
Engine	
J_e	Engine moment of inertia
T_{ref}	Reference torque from controller
T_{in}	Flywheel torque
T_{target}	Engine torque that compensates for driveline friction and driving resistance to achieve zero torque in the transmission
τ_{torque}	Torque delay, time delay from T_{ref} to T_{in}
$\dot{\theta}_e$	Enine angular velocity
Transmission	
$J_{t,in}$	Moment of inertia of input shaft
$J_{t,lay}$	Moment of inertia of lay shaft
$J_{t,main}$	Moment of inertia of main shaft
$J_{t,out}$	Moment of inertia of out shaft
J_t	Transmission total moment of inertia
i_{split}	Split conversion ratio
i_{gear}	Gear conversion ratio
i_{range}	Range conversion ratio
i_t	Total conversion ratio of the gearbox
b_t	Total transmission friction
$b_{t,1}$	Transmission friction on input shaft of disengaged transmission
$b_{t,2}$	Transmission friction on the output shaft of disengaged transmission
$\dot{\theta}_t$	Transmission angular velocity
Final drive	
J_f	Moment of inertia of final drive
i_f	Final drive conversion ratio
Drive shaft	
k_{eng}	Stiffness coefficient of engaged driveline
k_{diseng}	Stiffness coefficient of disengaged driveline
c_{eng}	Damping coefficient of engaged driveline
c_{diseng}	Damping coefficient of engaged driveline
T_d	Shaft torque acting on transmission
T_w	Shaft torque acting on wheel (same as T_d)
Wheel	
J_w	Moment of inertia of wheel
m	Vehicle mass
r_w	Wheel radius
T_{dr}	Driving resistance torque
$\dot{\theta}_w$	Wheel angular velocity

Chapter 2

Previous Work

This chapter describes previous work done in other papers in the same field as this thesis. In this thesis "model based control" means that an actual model is used *in* the controller like e.g. MPC (Model Predictive Controller) or feed forward where the inverse of a model is used.

Much work has been done trying to control and damp driveline oscillations with torque control. For this purpose several driveline models have already been developed and explained in different papers. The complexity of the models differ but a common conclusion is that a third order model (two-inertia model with one flexibility) is enough to explain the main behaviour of the driveline and accurate enough for controller design [6], [1].

2.1 Gear-Shift Experiments Without Control

In [6] several gear shift experiments are performed in a heavy duty truck. The gear shifts are performed without torque control on both a stationary driveline and a driveline with a relative speed difference between the transmission speed and the wheel speed. Engagement of neutral gear are then commanded at different times and the behavior of the engine speed, transmission speed and the wheel speed are analyzed and explained. A model of the disengaged driveline is derived to be able to simulate the gear shifts as well.

The experiments show that the stationary driveline and the driveline with a relative speed difference have different characteristics of the oscillations. According to [6] this indicates that a feedback control is needed in order to minimize the oscillations after a gear shift.

2.2 Different Control Strategies

2.2.1 Feedback With PID-Controller

One method to damp out the driveline oscillations is to use a feedback controller. This can be done with an observer in combination with a PID controller structure, with simple tuning rules [6]. The observer is used to estimate the drive shaft torsion since this has a dominating impact on the oscillations in the transmission speed [5]. The general idea in [6] is to observe the drive shaft torsion using a Kalman filter, and control it to zero with the PID controller in order to unload the driveline without exciting oscillations in transmission speed. This gives a much better result than to control the transmission torque which would be a more natural approach. The advantages with drive shaft torsion control are that it is more robust and easier to implement then transmission torque control.

2.2.2 Model Based Controllers

Another method is to use model based linear controllers. This is discussed in [2] where the idea is to develop a control system that takes the engine limitations like e.g. the smoke limiter into consideration. The aim of the controller is to damp out the oscillations that occurs during a tip-in and tip-out (pressing and releasing the accelerator pedal). The resulting control strategy is to use a combination of a feed-forward and a feedback controller. The feed-forward controller consists of a filter and the inverse of the driveline model. The feedback controller consists of a LQ controller with an observer which is used to take care of model errors and disturbances. The highest state penalty in the LQ controller is on the wheel speed. The control system needs to take the engine limitations into consideration so that the feed-forward controller does not calculate a control signal that is out of range for the engine. This is done in [2] by using a reference governor that modifies the reference signal by calculating a command signal to the feed-forward control, such that the feed-forward control signal stays within the engine boundaries. This requires that the reference governor has knowledge about maximum available engine torque and the dynamics of the feed-forward controller. Simulations and field experiments in a heavy truck show that the control strategy damps out the oscillations and is robust against model errors. A problem with the strategy is that the response of the truck becomes slower than with just a feed-forward controller. Also, the reference governor only takes the feed-forward contribution in consideration, so if the feedback controller gives a positive contribution the control signal may violate the engine boundaries despite of the reference governor. Another problem is that the observer for the LQ controller uses a model that has only been parameterized for one certain truck. If the control system is to be used in a more commercial way, the parameterization of the model needs to be adaptable to different vehicles.

2.2.3 Multiple Controllers With Contradictory Aims

A problem with driveline control is that fast engine torque changes during tip-in and tip-out leads to good acceleration but excites driveline oscillations. It is desired to damp out the oscillations and at the same time have good vehicle dynamics (good acceleration response). These two objectives are contradictory and therefore a new approach is investigated in [8]. Two different LQ-controllers, and a Kalman filter to estimate the non-measurable state, are used. The aim of the control system is to maximize the comfort and on the other hand increase vehicle dynamics. One LQ-controller aims for high comfort and the other to increase dynamics. Each of these two controllers calculates a manipulated variable independently from the other. The way of merging both variables depends on the driving situation and must therefore be adaptive online. This is solved by using fuzzy logic where the idea is to support the vehicle dynamics as long as the comfort is not getting too bad (see [8] for more details). The model in the observer must also be adaptive online due to modeling errors, time depending parameters and variable vehicle mass. Therefore a recursive least-square estimation is used to identify the model parameters and adapt the observer. Simulations with a plant model of the driveline show a very good improvement of both comfort and vehicle dynamics at the same time. With the use of the adaptation algorithm the control system is robust against model errors and the difference in construction of the driveline in different trucks. The method is however only tested in a simulation environment with a plant model.

2.2.4 Model Predictive Control With Comfort Evaluation Algorithm

The concept described in Section 2.2.3 uses two controllers that works at the same time in order to increase both driving comfort and vehicle dynamics during tip-in and tip-out. This can cause a conflict during a switching phase and the parametrization of the desired compromise between the controllers is very difficult. Therefore a new concept is introduced in [9] where the theory of model predictive control (MPC) is used in order to achieve an easy parametrization of the conflicting control targets. The two contradictory aims are fused together and an optimal control target is predicted based on the fusion. The concept uses the parameter Δn which is the difference between engine speed and wheel speed. This is not only an indicator of comfort but also for vehicle dynamics. The higher the amplitude of the first oscillation after a tip-in/out the higher the dynamics but the less the comfort. The amplitude is predicted to the first maximum using the MPC and the reference trajectory is calculated to get maximum Δn_0 in order to guarantee acceptable comfort. This means that a reference trajectory with amplitude Δn_0 can be calculated in order to respect both control targets. See [9] for more details.

The concept is evaluated with regard to both dynamics and driving comfort. The performance regarding dynamics is easy to evaluate by just looking at the vehicle acceleration. The evaluation of the comfort needs, however, a more advanced theory. In [9] this is done by using a model of a humans perception of acceleration and jerk in different directions. There is also a chapter suggesting to use neural network in order to approximate human feelings. However, the algorithm that is implemented in order to evaluate the comfort of the control method is based on the human perception model.

The evaluation is done in simulation only, where the results are very good. The comfort is acceptable and the considered time for acceleration is reduced by 85% for a tip-in on gear four compared to the common solution[1] for driveline control. The MPC can also handle the torque delays which gives good result even on higher gears. The drawbacks of the control method is of course the need of a model. In order to use this method in a commercial way the model must be adaptive online to handle model errors and changes in parameters.

[1] Defined in [9].

Chapter 3

Modeling

The driveline can be modeled with different complexity levels. A simple model has several advantages, it is easier to understand the behavior when the equations are simple and there are less parameters to adapt to measurements. The drawback is that it may not fit the measurements correctly. A more complex model could fit the measurements better while there will be harder work to get there. The complexity of the model depends on how much the basic equations are simplified or if nonlinearities are introduced. A basic model of the driveline is described in [3]. An overview of the basic model and equations can be seen in Figure 3.1 and 3.2.

3.1 Basic equations

In this section the basic equations of the driveline will be explained. From these equations different models will later be developed.

The engine

The engine control system receives many different torque-requests and limitations from different parts of the vehicle. They have different priorities, e.g. the smoke-limiter have a high priority and will always limit the torque. During a gearshift the reference torque, T_{ref}, have a high priority and will therefore be assumed to control the engine itself. T_{ref} is sent from the transmission software and is a reference for the flywheel torque. Engine friction, parasitic losses etc. are considered when the amount of fuel is calculated in order to produce the requested torque on the flywheel. Therefore the resulting torque on the flywheel, T_{in}, is already compensated for friction and other losses. This torque will accelerate the engine and the engines moment of inertia is the only thing that needs to be modeled.

The function from T_{ref} to T_{in} is a variable delay depending on the engine speed. The engine control unit have an interrupt to sample the reference torque in every revolution at a specific angle before top dead center (TDC), to be able to calculate the amount of fuel to inject. For an engine with six cylinders the

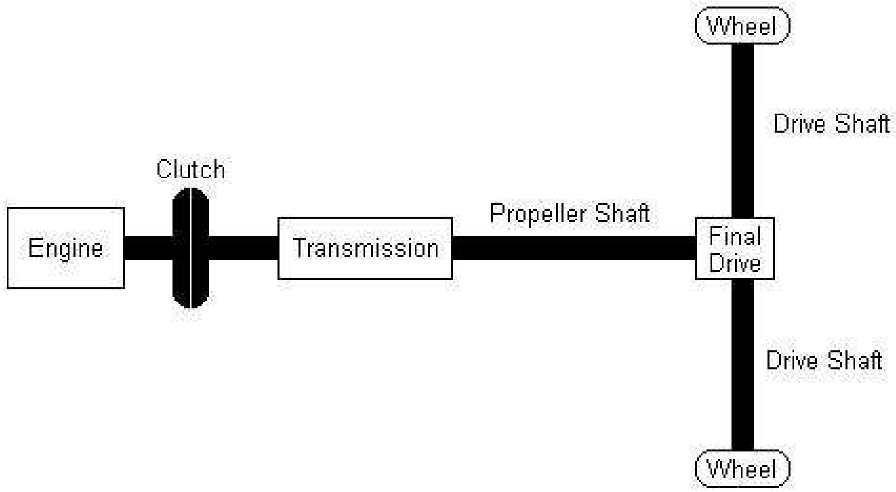

Figure 3.1. The driveline.

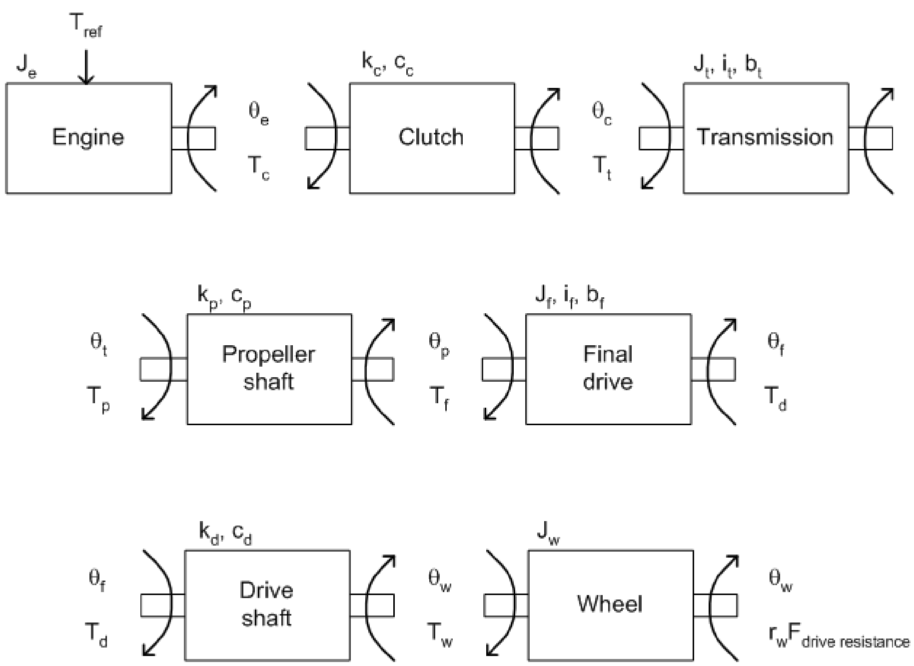

Figure 3.2. The driveline.

reference torque will be sampled three times each revolution. So the worst case of delay caused by sampling will in this case be 120° which translates to time through the engine speed. Besides the sampling delay there are an injection time and an ignition time before the torque is transferred to the flywheel. There is also a transfer delay from the transmission control unit to the engine control unit due to the CAN bus, more about this in Section 5.1. The total torque delay, τ_{torque}, from the reference torque to the flywheel torque will be modeled as a constant delay τ_i added with a variable delay depending on the sampling angle $\alpha_{sampling}$ and the engine speed. This yields:

$$T_{in}(t) = f(T_{ref}, \tau_{torque}) = T_{ref}(t - \tau_{torque}) \tag{3.1}$$

$$\tau_{torque} = \tau_i + \frac{\alpha_{sampling}}{\dot{\theta}_e} \tag{3.2}$$

$$J_e \ddot{\theta}_e = T_{in} - T_c \tag{3.3}$$

Clutch

The clutch transfers the output torque of the engine to the transmission. In ordinary manual transmissions the clutch is disengaged by the driver to achieve zero torque during a gear shift. In an AMT the clutch can be automatically engaged and disengaged if it is used. In this work the clutch will be assumed to be engaged at all time. There is a weakness in the clutch and the clutch can be modeled as a damped spring.

$$T_t = T_c = k_c(\theta_e - \theta_c) + c_c\left(\dot{\theta}_e - \dot{\theta}_c\right) \tag{3.4}$$

Transmission

The transmission basically consists of four shafts where the power is transfered with different cogwheels between the shafts. They are named input shaft, lay shaft, main shaft and output shaft (see Figures 3.3 and 3.4). The power can be transfered with two different cogwheels between the input shaft and lay shaft. These cogwheels define the split gear. The cogwheels that transfer the power between the lay shaft and main shaft define the gear. The last part in the gearbox is a planetary gear called range, that transfer the power between the main shaft and the output shaft. Each part of the transmission has a moment of inertia and a stiffness. The stiffnesses will be moved toward the wheel side, explained later in Section 3.2. There is also a friction term in each part of the transmission. All frictions will be modeled as viscous frictions.

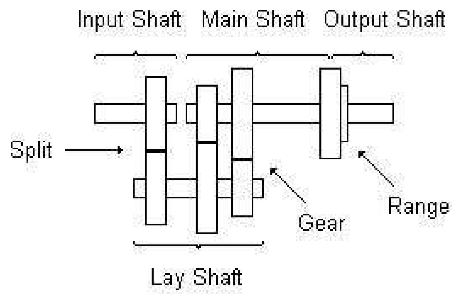

Figure 3.3. The gearbox in principle. A real gearbox has more cogwheels than the picture shows.

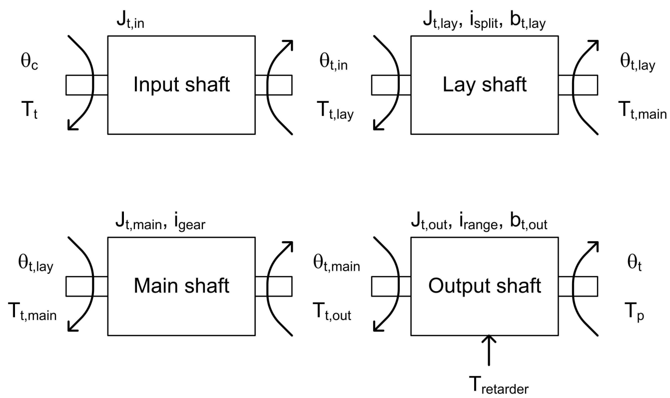

Figure 3.4. The gearbox.

$$\begin{cases} J_{t,in}\ddot{\theta}_{t,in} = T_t - b_{t,in}\dot{\theta}_{t,in} - T_{t,lay} \\ \theta_{t,in} = \theta_c \end{cases} \tag{3.5}$$

$$\begin{cases} J_{t,lay}\ddot{\theta}_{t,lay} = i_{split}T_{t,lay} - b_{t,lay}\dot{\theta}_{t,lay} - T_{t,main} \\ i_{split}\theta_{t,lay} = \theta_{t,in} \end{cases} \tag{3.6}$$

$$\begin{cases} J_{t,main}\ddot{\theta}_{t,main} = i_{gear}T_{t,main} - b_{t,main}\dot{\theta}_{t,main} - T_{t,out} \\ i_{gear}\theta_{t,main} = \theta_{t,lay} \end{cases} \tag{3.7}$$

$$\begin{cases} J_{t,out}\ddot{\theta}_{t,out} = i_{range}T_{t,out} - b_{t,out}\dot{\theta}_{t,out} - T_p \\ i_{range}\theta_t = \theta_{t,main} \end{cases} \tag{3.8}$$

All moments of inertia are seen from the output shaft of the transmission.

Since the stiffness is moved, these equations (3.5)–(3.8) can be simplified to one equation which represents the whole transmission block in Figure 3.2:

$$\left\{ \begin{array}{lcl} J_t\ddot{\theta}_t & = & T_t - b_t\dot{\theta}_t - T_p \\ i_t\theta_t & = & \theta_c \end{array} \right. \tag{3.9}$$

where b_t is the total friction, i_t is the total conversion ratio and J_t is the total moment of inertia in the transmission. When frictions and moments of inertia are moved through conversion ratios, the old values are scaled with the conversion ratio.

$$J_t = J_{t,out} + i_{range}^2 \left(J_{t,main} + i_{gear}^2 \left(J_{t,lay} + i_{split}^2 J_{t,in}\right)\right) \tag{3.10}$$

$$i_t = i_{split} i_{gear} i_{range} \tag{3.11}$$

The friction b_t is calculated correspondingly.

Only a measurement with the total moment of inertia (for each gear) was available at Scania. The other moments of inertia have been calculated from this measurement. Since the conversion ratio changes between the moving parts the total moment of inertia will be different for different gears. The moment of inertia have been measured on the input shaft, but in this thesis the transmission moment of inertia means the one seen from the output shaft.

$$J_t = \frac{J_{t,measured}}{i_t^2} \tag{3.12}$$

Gear Shifting

The cogwheels in the transmission are of different types. The ones on the lay shaft are fix and rotates with the shaft. The cogwheels on the input shaft and the main shaft slides on their shafts and rotates with the lay shaft. A certain cogwheel can be locked to the input shaft and the main shaft respectively. This is done by sliding a collar on the shaft that locks one cogwheel to it. This means that the torque can be transferred different ways through the gearbox which defines the different gears. When no cogwheel on the main shaft is locked to it, no torque will be transferred. This defines the neutral gear.

Propeller Shaft

The propeller shaft connects the gear box with the final drive. It will be modeled as a damped spring.

$$T_f = T_p = k_p \left(\theta_t - \theta_p\right) + c_p \left(\dot{\theta}_t - \dot{\theta}_p\right) \tag{3.13}$$

Final Drive

The final drive is a differential gear with a conversion ratio. In this work the differential is assumed to be locked, i.e. the two drive shafts will not be able to turn relative each other.

$$\left\{ \begin{array}{lcl} J_f\ddot{\theta}_f & = & i_f T_f - b_f\dot{\theta}_f - T_d \\ i_f\theta_f & = & \theta_p \end{array} \right. \tag{3.14}$$

Drive Shaft

In this thesis only trucks with two driving wheels are considered. In these trucks there are two drive shafts, one for each driving wheel. If the final drive is locked, one can say that the drive shafts are connected to the ground through the wheels in parallel. The replacing stiffness simply is the sum of the two stiffnesses, $k_d = k_{d,right} + k_{d,left}$. From here on the drive shafts will be treated as one total drive shaft with the resulting stiffness k_d. The drive shaft is known to have the smallest stiffness in the driveline.

$$T_d = T_f = k_d(\theta_f - \theta_d) + c_d(\dot{\theta}_f - \dot{\theta}_d) \tag{3.15}$$

Wheel

Newton's second law gives the force balance in the longitudinal direction.

$$m\dot{v} = F_w - F_{dr} \tag{3.16}$$

where F_w is the friction force acting on the wheel, and F_{dr} is the driving resistance which can mainly be divided in the following quantities [3]:

- F_{air}, the air resistance is approximated by

$$F_{air} = c_{air}v^2 \tag{3.17}$$

- F_r, the rolling resistance is approximated by

$$F_r = m\left(c_{r1} + c_{r2}v\right) \tag{3.18}$$

- $mg\sin(\alpha)$, is the gravitational force and α is the road slope.

The wheel itself has its own moment of inertia and Newton's second law yields

$$J_w\ddot{\theta}_w = T_w - r_wF_w \tag{3.19}$$

Equations (3.16)-(3.19) together with $v = r_w\dot{\theta}_w$ (no wheel slip) yields

$$\left(J_w + mr_w^2\right)\ddot{\theta}_w = T_w - r_w\left(F_{air} + F_r + mg\sin(\alpha)\right) = T_w - T_{dr} \tag{3.20}$$

The constants c_{air}, c_{r1} and c_{r2} are unknown parameters that need to be identified. The equations can be written as a polynomial, $F_{air} + F_r = A + Bv + Cv^2$ where A, B and C can be identified instead.

3.2 Model Reduction

The equations above is enough to describe the main behavior of a truck. To get a model that is easy to work with, one would like to reduce the model as long as the behavior is satisfactory. When reducing the model the purpose of the model is considered. The model does not necessarily have to function in scenarios that will never happen during a normal gear shift.

3.2.1 Three Inertia Model

The weakest part of the driveline is the drive shaft. The second weakest is the propeller shaft. The clutch and the transmission are known to be stiffer. A three inertia model only contains the drive shaft and the propeller shaft, see Figure 3.5. To motivate this one can look at the dynamics between the engine and the output shaft. Measurements at Scania have resulted in some mechanical data for the transmission stiffness but as seen in Figure 3.6 the difference between engine speed (scaled) and output shaft speed is very small. There is almost a static connection from the engine to the output shaft. But since there are measurements of the transmission stiffness these values are taken in consideration by letting them influence on the propeller shaft stiffness (see Section 4.1.1).

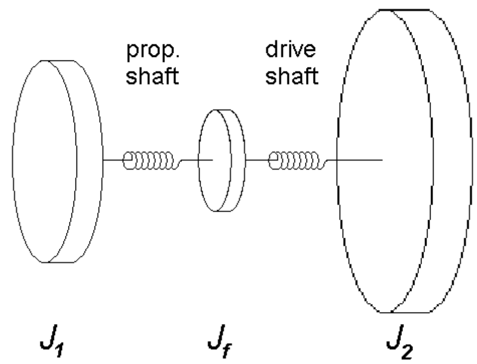

Figure 3.5. The three inertia model with two springs.

The model has the same structure as the one in Figure 3.2 but the engine, clutch and transmission blocks are lumped together. The equations for the three inertia model becomes:

$$T_{in} = f\left(T_{ref}, i_t\dot{\theta}_t\right) \tag{3.21}$$

$$J_1\ddot{\theta}_t = i_tT_{in} - b_t\dot{\theta}_t - T_p \tag{3.22}$$

$$T_p = T_f = k_p\left(\theta_t - i_f\theta_f\right) + c_p\left(\dot{\theta}_t - i_f\dot{\theta}_f\right) \tag{3.23}$$

$$J_f\ddot{\theta}_f = i_fT_f - b_f\dot{\theta}_f - T_d \tag{3.24}$$

$$i_f\theta_f = \theta_t \tag{3.25}$$

$$T_d = T_w = k_d\left(\theta_f - \theta_w\right) + c_d\left(\dot{\theta}_f - \dot{\theta}_w\right) \tag{3.26}$$

$$J_2\ddot{\theta}_w = T_w - T_{dr} \tag{3.27}$$

where

$$J_1 = J_e i_t^2 + J_t \tag{3.28}$$

$$J_2 = \left(J_w + mr_w^2\right) \tag{3.29}$$

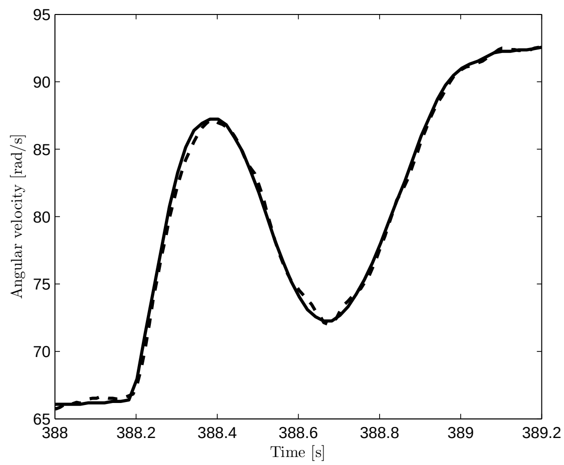

Figure 3.6. Engine speed (solid) and output shaft speed (dashed).

3.2.2 Two Inertia Model

In a two inertia model there is only one rotational spring, i.e. only one weak shaft. The three inertia model consists of three inertias, but the inertia of the final drive, J_f, is less than the other two inertias. The three inertia model will act as if there are two springs in series. Therefore the two springs are lumped together and the moment of inertia of the final drive becomes a part of J_1. How the resulting spring stiffness can be calculated is found in Section 4.1.1. The friction in the final drive, b_f, can not be estimated separately from b_t. The two frictions are lumped together and the result is called just b_t.

The equations for the two inertia model becomes:

$$T_{in} = f(T_{ref}, i_t\dot{\theta}_t) \tag{3.30}$$

$$J_1\ddot{\theta}_t = i_tT_{in} - b_t\dot{\theta}_t - \frac{T_d}{i_f} \tag{3.31}$$

$$T_d = T_w = k_{eng}\left(\frac{\theta_t}{i_f} - \theta_w\right) + c_{eng}\left(\frac{\dot{\theta}_t}{i_f} - \dot{\theta}_w\right) \tag{3.32}$$

$$J_2\ddot{\theta}_w = T_w - T_{dr} \tag{3.33}$$

where

$$J_1 = J_ei_t^2 + J_t + \frac{J_f}{i_f^2} \tag{3.34}$$

$$J_2 = J_w + mr_w^2 \tag{3.35}$$

3.3 Plant Model

The plant model is the model that the control systems will be evaluated against. The model is chosen from the two models described above with regards to accuracy, simulation time and simpleness. The most important thing with the plant model is to capture the main dynamics of the driveline. The drive shaft is the weakest part of the driveline which means that the major dynamics are captured with a model that has drive shaft flexibility (see [5]). Both the three inertia model and the two inertia model should therefore be able to capture the main dynamics but the question is if the three inertia model can capture other important oscillations. In order to compare the two models, parameters for each of the models have been identified with the same measured data. Both models have been implemented in Matlab\Simulink, the three inertia model was also implemented in SimDriveline. SimDriveline is an extension to Simulink with tools for modeling and simulating the mechanics of driveline systems. Instead of modeling the equations directly one uses pre-made models of the driveline components like gears, shafts, torque actuators and sensors. The idea was that this would give a faster simulation time and perhaps a better result than the models implemented in Simulink. Unfortunately the components were complicated to initialize and the simulation result was no better then the original three inertia model. Furthermore the simulation time for the model in SimDriveline was approximately ten times longer than the models in Simulink. Therefore the model in SimDriveline was discarded.

Figures 3.7 and 3.8 show a simulation with the different Simulink models on second and fourth gear. There is also a frequency comparison based on the two models. Here we can see that the three inertia model is very similar to the two inertia model both in time and frequency domain. When compared to measurements the three inertia model does not give a better fit. Furthermore the three inertia model takes longer time to simulate and is more difficult to parameter identify. This is because the additional spring, representing the propeller shaft, give rise to two extra differential equations, which make the system more difficult to solve. In the two inertia model there is only one spring modeled but the stiffness and damping from the propeller shaft and gearbox is included in the calculations (see Section 4.1.1). This makes the two inertia model a better choice since it is equally accurate as the three inertia model but have faster simulation time and is easier to parameter identify. More about simulation and validation of the plant model in Chapter 5. The resulting weakness in the plant model, that consists of the transmission shafts, the propeller shaft and the drive shaft will from here on be called just “the shaft”. The torque transmitted by the shaft will simply be called “shaft torque”.

3.3.1 Engaged and Disengaged Model

To be able to simulate a gear shift well, it is desired to have a model for the disengaged driveline as well. The plant model will switch between the models when neutral gear is engaged. The purpose of the disengaged model is to capture possible oscillations in the driveline after engaging neutral gear. When neutral

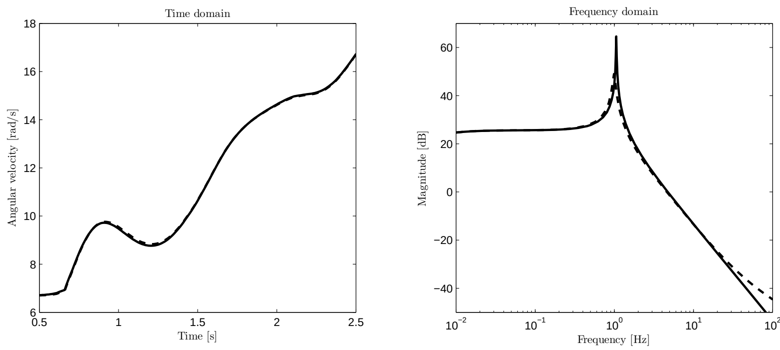

Figure 3.7. Left: Simulated output shaft speed during a tip-in on gear 2 with optimized parameters. Right: Amplitude spectrum of transfer function from T_{in} to shaft torque for gear 2. Two inertia model (solid) and three inertia model (dashed).

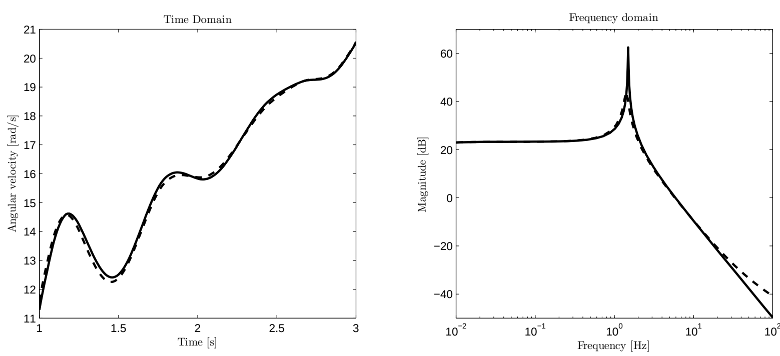

Figure 3.8. Left: Simulated output shaft speed during a tip-in on gear 4 with optimized parameters. Right: Amplitude spectrum of transfer function from T_{in} to shaft torque for gear 4. Two inertia model (solid) and three inertia model (dashed).

gear is engaged the engine is not able to affect the driveline.

Engaged Model

The engaged model is the two inertia model described above. The model are described with Equations (3.30)-(3.35). If the flywheel torque, T_{in} and the driving resistance torque, T_{dr}, are seen as inputs, the system can be written in state space form:

$$A_{eng} = \begin{pmatrix} -\frac{\frac{c_{eng}}{i_f^2}+b_t}{J_1} & \frac{-k_{eng}}{J_1 i_f} & \frac{c_{eng}}{J_1 i_f} \\ \frac{1}{i_f} & 0 & -1 \\ \frac{c_{eng}}{J_2 i_f} & \frac{k_{eng}}{J_2} & \frac{-c_{eng}}{J_2} \end{pmatrix} \tag{3.36}$$

$$B_{eng} = \begin{pmatrix} \frac{i_t}{J_1} & 0 \\ 0 & 0 \\ 0 & \frac{-1}{J_2} \end{pmatrix} \tag{3.37}$$

where

$$J_1 = J_e i_t^2 + J_t + \frac{J_f}{i_f^2} \tag{3.38}$$

$$J_2 = (J_w + m r_w^2) \tag{3.39}$$

with states and input signals

$$\begin{aligned} x_1 &= \dot{\theta}_t \\ x_2 &= \frac{\theta_t}{i_f} - \theta_w \\ x_3 &= \dot{\theta}_w \\ u_1 &= T_{in} \\ u_2 &= T_{dr} \end{aligned} \tag{3.40}$$

Disengaged Model

The disengaged model have two independent parts since the torque chain is broken. The engine only affects the transmission up to the lay shaft. The engine side has two equations, derived from Equations (3.3), (3.5) and (3.6).

$$T_{in} = f(T_{ref}, \dot{\theta}_e) \tag{3.41}$$

$$(J_e + J_{t,in} + \frac{J_{t,lay}}{i_{split}^2})\ddot{\theta}_e = T_{in} - b_{t,1}\dot{\theta}_e \tag{3.42}$$

where $b_{t,1}$ is the transmission friction on the input shaft. These equations have not been modeled in Simulink since the engine control during the disengagement is not part of this thesis.

The rest of the driveline has almost the same equations as the engaged driveline with some modifications:

$$(J_{t,main}i_{range}^2 + J_{t,out})\ddot{\theta}_t = -b_{t,2}\dot{\theta}_t - \frac{T_d}{i_f} \tag{3.43}$$

$$T_d = T_w = k_{diseng}(\frac{\theta_t}{i_f} - \theta_w) + c_{diseng}\left(\frac{\dot{\theta}_t}{i_f} - \dot{\theta}_w\right) \tag{3.44}$$

$$(J_w + mr_w^2)\ddot{\theta}_w = T_w - T_{dr} \tag{3.45}$$

where $b_{t,2}$ is the transmission friction on the output shaft. Note that there are different stiffnesses and damping values in the disengaged model, see Chapter 4.

With the same inputs as for the engaged model the disengaged system can be written in state space form:

$$A_{diseng} = \begin{pmatrix} -\frac{b_{t,1}}{J_1} & 0 & 0 & 0 \\ 0 & -\frac{\frac{c_{diseng}}{i_f^2}+b_{t,2}}{J_2} & \frac{-k_{diseng}}{J_2 i_f} & \frac{-c_{diseng}}{J_2 i_f} \\ 0 & \frac{1}{i_f} & 0 & -1 \\ 0 & \frac{c_{diseng}}{J_3 i_f} & \frac{k_{diseng}}{J_3} & \frac{-c_{diseng}}{J_3} \end{pmatrix} \tag{3.46}$$

$$B_{diseng} = \begin{pmatrix} \frac{1}{J_1} & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & \frac{-1}{J_2} \end{pmatrix} \tag{3.47}$$

where

$$J_1 = J_e + J_{t,in} + \frac{J_{t,lay}}{i_{split}^2} \tag{3.48}$$

$$J_2 = J_{t,main}i_{range}^2 + J_{t,out} + \frac{J_f}{i_f^2} \tag{3.49}$$

$$J_3 = J_w + mr_w^2 \tag{3.50}$$

with states and input signals

$$\begin{aligned} x_1 &= \dot{\theta}_e \\ x_2 &= \dot{\theta}_t \\ x_3 &= \frac{\theta_t}{i_f} - \theta_w \\ x_4 &= \dot{\theta}_w \\ u_1 &= T_{in} \\ u_2 &= T_{dr} \end{aligned} \tag{3.51}$$

It also shows in the state space formulation that the engine is cut off from the rest of the driveline. State x_1 does not influence the other states and vice versa.

Chapter 4

Parameter Estimation

This chapter describes the parameters used in the plant model from Chapter 3. The parameters that are used can be seen in Section 1.3. The majority of the parameters can be measured or calculated from technical data and the other parameters have been estimated with System Identification Toolbox in Matlab (SITB).

4.1 Estimation of Unknown Parameters

Some of the used parameters in the driveline model can not be obtained from technical data at Scania. These unknown parameters are the damping coefficients for the engaged model c_{eng}, and disengaged model c_{diseng}, stiffness for the non modeled components k_u, and the gearbox friction b_t. The resulting stiffness and damping for the engaged model varies with the gears since there are stiffnesses in the clutch and the different shafts in the gearbox. The parameters are estimated using SITB. A state-space model is used where all the known parameters are locked and all the unknown parameters can be varied in order to obtain the best fit to measured data.

4.1.1 Stiffness

The total stiffness used in the engaged model is called k_{eng}. This stiffness has been calculated from technical data for the stiffnesses in the clutch, gearbox, propeller shaft and drive shaft. An unknown stiffness parameter k_u has also been estimated and included to compensate for non modeled components and uncertainties in the technical data. All stiffnesses parameters are added in a way that is equivalent to adding different springs in a series. When adding the unknown stiffness parameter, the resulting stiffness becomes somewhat lower than the calculated value from technical data. Table 4.1 shows how much the estimated value differs from the theoretical value on different gears.

In the disengaged model the parameter k_{diseng} is used. This parameter is calculated from mechanical data for the components included in the disengaged model. The only component that varies with the gears in the disengaged model

Gear	**Change in stiffness when adding k_u**
1	-8.9%
2	-10.5%
3	-10.5%
4	-11.3%
5	-8.7%
6	-4.6%
7	-5.4%
8	-7.9%
9	-7.0%
10	-5.4%
11	-3.5%
12	-6.2%

Table 4.1. Change in the total stiffness when adding k_u compared to just using the technical data.

is the conversion ratio of the range gear, therefore the calculated stiffness has the same value for all gears with the same range gear. As for the engaged model, an unknown parameter $k_{u,diseng}$ is included to get the frequencies right. $k_{u,diseng}$ is chosen so the resulting stiffness only varies with the range gear, i.e. $k_{u,diseng}$ has only one value for range high, and one for range low. The resulting parameter k_{diseng} differs much more from the mechanical data than k_{eng}, but to capture the oscillations from measurements it was necessary to tune the parameter. The value of k_{diseng} changes with approximately 80 % when adding $k_{u,diseng}$. The reason for this big difference has not been found, but could perhaps be explained by model errors in the disengaged model, such as backlashes, moment of inertias, etc. This problem should however be investigated more in future work. The total stiffness for two springs in a series is calculated as:

$$k_{tot} = \frac{k_1 k_2}{k_1 + k_2} \tag{4.1}$$

If there is a gear between the springs the total stiffness is calculated as:

$$k_{tot} = \frac{k_1 i^2 k_2}{k_1 i^2 + k_2} \tag{4.2}$$

where i is the conversion ratio. The total stiffness k_{eng} is now calculated as:

$$k_1 = \frac{k_c k_t}{k_c + k_t} \tag{4.3}$$

$$k_2 = \frac{k_1 i_t^2 k_p}{k_1 i_t^2 + k_p} \tag{4.4}$$

$$\tag{4.5}$$

$$k_3 = \frac{k_2 k_u}{k_2 + k_u} \tag{4.6}$$

$$k_{eng} = \frac{k_3 i_f^2 k_d}{k_3 i_f^2 + k_d} \tag{4.7}$$

The total stiffness for the disengaged model k_{diseng} is calculated in the same way as above.

4.1.2 Damping

The total damping coefficient for two damped springs in a series with a gear between, is calculated in the same way as the stiffness i.e.

$$c_{tot} = \frac{c_1 i^2 c_2}{c_1 i^2 + c_2} \tag{4.8}$$

where i is the conversion ratio. The damping coefficients can however not be obtained from technical data. Therefore the total damping, c_{eng} for the engaged model and c_{diseng} for the disengaged model, are estimated directly using SITB.

A problem with the nonphysical values of k_{diseng} and c_{diseng} is that the ratio between them does not coincide with the ratio between the values from the engaged model:

$$\frac{k_{diseng}}{c_{diseng}} \neq \frac{k_{eng}}{c_{eng}} \tag{4.9}$$

As a result of this the shaft torque can change sign when neutral gear is engaged. If the part from the torsion and the part from the speed difference have different signs, the sign of the calculated torque can change when the ratio between the stiffness and the damping changes. This is a problem that makes the shaft torque look discontinuous in simulations, but it was preferred to get the right frequency and right damping of the oscillations in the disengaged model. The engaged model and the disengaged model was validated with measurements from a truck with good results. The problem with the discontinuous shaft torque, due to the non-physical values of the parameters, occurs when switching between the models. Due to lack of time this problem is not further investigated in this thesis but should be included in future work.

4.1.3 Friction

The friction coefficient b_t multiplied with the output shaft speed models the friction in the gearbox and the final drive. This means that the coefficient will vary with different gears. Therefore the parameter is estimated for each gear in order to fit the measured data. In the disengaged model the friction coefficient b_t is divided into $b_{t,1}$ and $b_{t,2}$ where $b_{t,1}$ is the friction coefficient on the input shaft and $b_{t,2}$ is the friction coefficient on the output shaft of the gearbox. A look at the geometry of the gearbox, see Figure 3.3, shows that when neutral is engaged most of the parts rotate with the input shaft. The cogwheels on the main shaft rotates with the lay shaft. The parts that rotates with the output shaft are the main shaft (without cogwheels), the range gear and the final drive. This shows that most of the friction should be placed in $b_{t,1}$ and parameter identification has showed that $b_{t,2}$ is near zero. The friction in other parts of the drive line is included in the driving resistance and the engine friction has already been compensated for in the torque T_{in}.

4.2 Eigenfrequency

The systems eigenfrequency can be calculated from the parameter values. It is mostly affected by the stiffness. To calculate the eigenfrecuency it is easiest to analyze the poles of the system. The eigenvalues for the matrix A_{eng} in Equation (3.36) determine the poles. The corresponding characteristic equation for the eigenvalues is:

$$r^3 + \left(c_{eng}a + \frac{b_t}{J_1}\right) r^2 + \left(k_{eng}a + \frac{b_t c_{eng}}{J_1 J_2}\right) r + \frac{b_t k_{eng}}{J_1 J_2} \tag{4.10}$$

where

$$a = \frac{i_f^2 J_1 + J_2}{i_f^2 J_1 J_2}, \quad J_1 = J_e i_t^2 + J_t + \frac{J_f}{i_f^2}, \quad J_2 = J_w + m r_w^2 \tag{4.11}$$

The solution to Equation (4.10) is very complex. If the friction term is neglected (i.e. $b_t = 0$), the solution of the characteristic equation becomes much simpler. To see how much the friction affects the eigenfrecuency, the transfer function from T_{in} to the shaft torque was calculated using Matlab. The eigenfrequency of the system is the peak response in the amplitude characteristics of the transfer function. The friction was varied from 0 to 200% of its estimated value, and the eigenfrequnecy was only affected less then one percent. Without the friction term the characteristic equation is:

$$r\left(r^2 + c_{eng}ar + k_{eng}a\right) \tag{4.12}$$

The solutions to this equation is:

$$r_{1,2} = -\frac{c_{eng}a}{2} \pm i\sqrt{k_{eng}a - \left(\frac{c_{eng}a}{2}\right)^2} \tag{4.13}$$

$$r_3 = 0 \tag{4.14}$$

The eigenfrequency can now be identified as the imaginary part of the complex conjugated roots.

$$\omega_{eig} = \sqrt{k_{eng}a - \left(\frac{c_{eng}a}{2}\right)^2} \tag{4.15}$$

The relative damping of the system is described by:

$$\varsigma = \frac{c_{eng}a}{2\sqrt{k_{eng}a}} \tag{4.16}$$

Chapter 5

Simulation and Validation

The plant model is implemented in Matlab\Simulink. Data to estimate and validate the model have been measured at Scania. This chapter describes the signals used in the simulations, how different delays affect the simulations and how the plant model is validated.

5.1 Signals

The signals in a truck are transferred on a data bus, with the CAN (Controller Area Network) protocol. There is a limit on how much data that can be extracted from a truck, because of the limits of the bandwidth of the CAN bus. Different signals are sampled with different frequencies depending on the variation of the signal and how important it is to get a real time value of that signal. Most of the signals used in this thesis are sampled with a rate of 100 Hz. The plant model is simulated with continuous time but the input signal T_{ref} and the output signals is sampled with 100 Hz.

5.1.1 Input Signals

The input signals to the model are the reference torque, T_{ref}, the flywheel torque, the signal from the neutral gear sensor and the road incline, α. These are measured signals that are used to drive the model. The road incline is used to calculate the driving resistance, as described in Section 3.1.

The reference signal T_{ref} is zero until a gear shift is ordered by the driver or the control system. The reference torque can not be used until it is nonzero. When the model is simulated, the measured flywheel torque is used until the delayed T_{ref} is nonzero, i.e. a gear shift is ordered. Then the input is switched to the reference signal, which is delayed according to Section 3.1.

$$T_{in}(t) = \begin{cases} T_{ref}(t - \tau_{torque}), & T_{ref}(t - \tau_{torque}) \neq 0 \\ T_{flywheel,measured}(t), & else \end{cases} \tag{5.1}$$

Figure 5.1 shows the three signals in the same picture, the arrow shows where T_{in} is switched from the flywheel torque to T_{ref}.

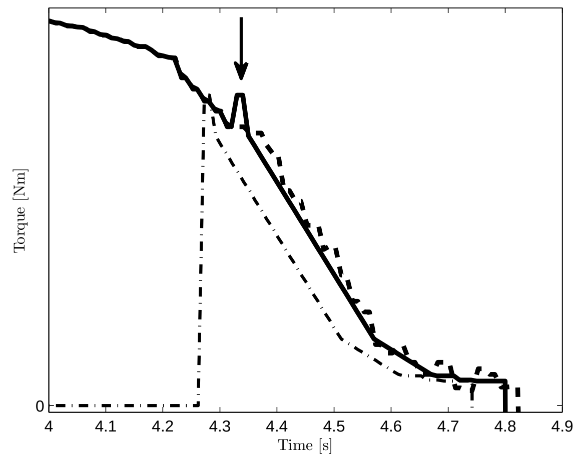

Figure 5.1. Measured flywheel torque (dashed), measured reference torque (dash-dotted) and the simulated flywheel torque (solid). The arrow implies where the input signal is changed from the flywheel torque to the delayed reference torque.

The timing of engaging the neutral gear in the model is a problem. To get the model to disengaged mode at the same time as the real truck is hard. The real truck uses compressed air to change the gear. Therefore the pressure have to be built up in order to move the collars on the shafts before it can disengage a gear. The total delay, called the blow delay, is the time from the blowing starts until the gear actually is disengaged. There is a neutral sensor that indicates when the neutral gear is engaged. But the engagement time according to the neutral sensor and the time that looks to be the real engagement time in measurements of the output shaft speed differs. It can also seem different from shift to shift. However, the neutral sensor will be used since this is closest to the measurements.

5.1.2 Output Signals

The output signals from the model are the flywheel torque T_{in}, shaft torque T_d, output shaft speed $\dot{\theta}_t$ and wheel speed $\dot{\theta}_w$. All signals, except the shaft torque, are also available on the CAN bus in a real truck. Therefore the output signals from the model are sampled with the same rate as the sensors in the truck in order to compare the model output with measured signals from the CAN bus.

5.2 Validation

Recordings from a real truck are used to validate the model. Parameters for the model are taken from technical data and from estimations described in Chapter 4.

The results from simulated and recorded signals for all states of the model can be seen in Figures 5.2 and 5.3. The simulated torsion is hard to validate since the real torsion can not be measured. The values of the torsion is however realistic and in the same range as the values seen in other papers, like in e.g. [5].

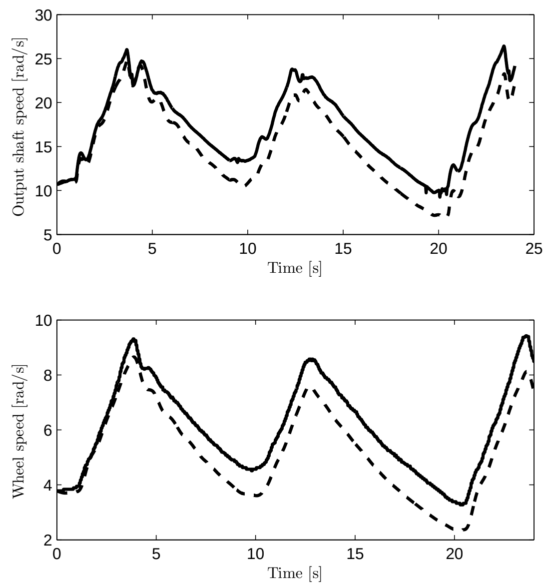

Figure 5.2. Top figure: Simulated (dashed) and measured (solid) transmission speed when driving without gear shift on gear two. Bottom figure: Simulated (dashed) and measured (solid) wheel speed when driving without gear shift on gear two.

The simulated data captures the main behavior but drifts over time when compared to the measured signals. The cause of this is probably poor estimation of driving resistance and friction. This is however not an important issue since the model is used to evaluate gear shifts over a very short period of time. Figure 5.4 and Figure 5.5 show the transmission speed during a shift from gear two and gear four to neutral gear. In Figure 5.6 an FFT (Fast Fourier Transform) of measured and simulated output shaft speed are compared where the simulation uses measured input signals from the same sequence as the measured output shaft speed. The agreement between simulated and measured data is accurate enough in order to evaluate the quality of the gear shift. One problem is that the engagement of neutral gear happens at a different time in the simulation compared to the measurements. This is caused by the inaccuracy of the neutral sensor as mentioned above. The results from the simulations have also showed that the model has better agreement against measurements on lower gears. On the higher gears the drift over time is bigger and the amplitude of the oscillations after a gear shift is inaccurate. However, since this thesis focuses on lower gears where the problem with bad gear shifts mainly occur, the model is accurate enough.

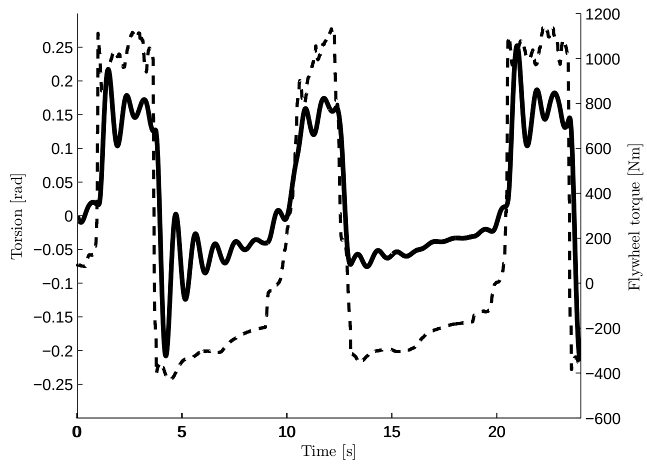

Figure 5.3. Simulated torsion (solid) and flywheel torque (dashed) when driving without gear shift on gear two.

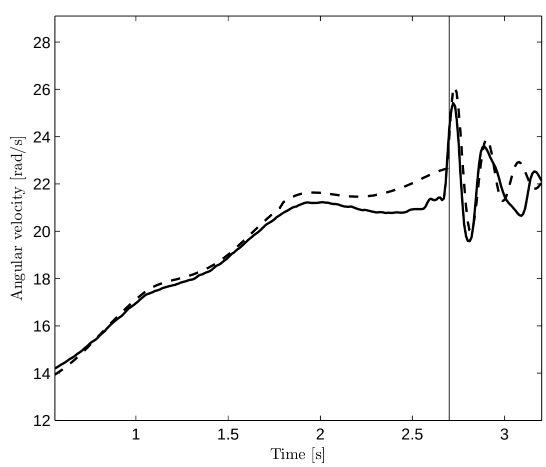

Figure 5.4. Simulated (dashed) and measured (solid) transmission speed during tip-in and neutral engagement on gear two. The vertical line shows when the neutral sensor indicates that neutral is engaged.

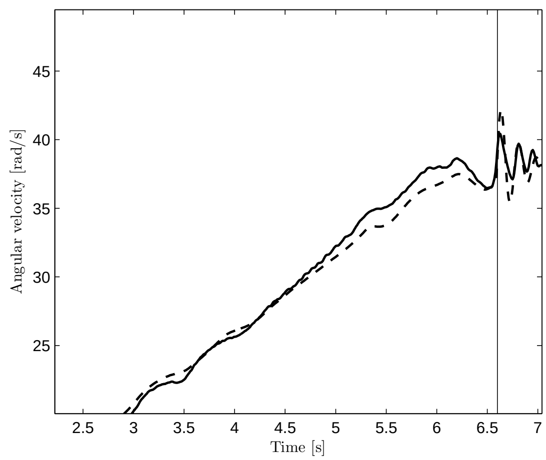

Figure 5.5. Simulated (dashed) and measured (solid) transmission speed during tip-in and neutral engagement on gear four. The vertical line shows when the neutral sensor indicates that neutral is engaged.

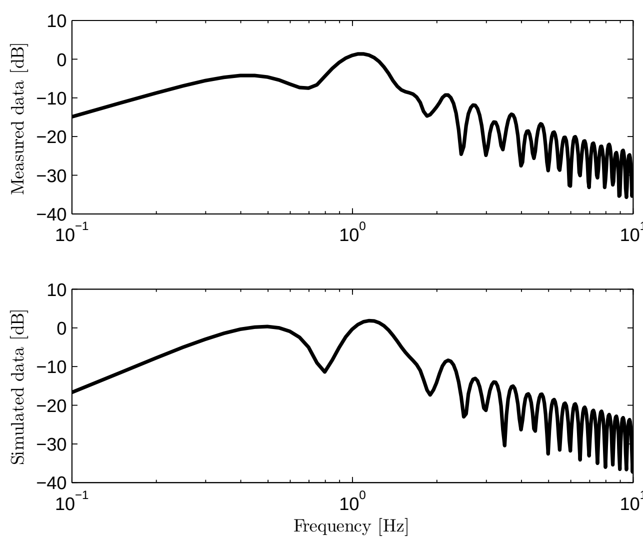

Figure 5.6. Upper: FFT of measured output shaft speed. Lower: FFT of corresponding simulated data.

Chapter 6

Torque Control During Gear Shift

To do a good gear disengagement it is important that the shafts are unloaded. If they are winded up, the speed on the output shaft of the transmission will oscillate after disengagement which makes it harder to engage the next gear. In principle, the problem is to unload a spring without making it oscillate. This chapter describes the idea with torque control, discuss open-loop control versus closed-loop control and describes how the different controllers are evaluated.

6.1 Basic Idea With Engine Torque Control

The torque that should be controlled to zero is the torque transmitted between the lay shaft and the main shaft, because that is where the torque chain is cut off when neutral gear is engaged. In a truck there are dynamics between the transmission torque and the drive shaft torque, mostly in the propeller shaft. These torques are therefore not the same. But if the drive shaft torsion is controlled to zero, the transmission torque will be at least close to zero (See [6]). However, in the plant model where there is only one weakness modeled, these two torques coincides.

When the driving torque is ramped down, the shaft will begin to unwind itself. The driving torque acting on the wheels will be decreased and the vehicle will start retarding because of the driving resistance (if positive i.e. driving on a flat surface or uphill). To get zero torque in the shaft, the retardation of the transmission side must be equal to the retardation of the wheel side. Therefore a negative torque must be acting on the transmission side. The level of the engine torque to achieve this will be called engine target torque, T_{target}. Note that the value of T_{target} does not have to be less then zero, because the friction in the transmission will also give a retarding contribution on the transmission side. If the torque in the shaft is zero when engaging neutral gear, no oscillations will occur. It is not desirable that the shaft torque is negative anywhere during a gear shift since this would apply a braking torque on the wheels. The torque should preferably approach zero from a positive value. T_{target} is calculated online in the truck, and is based on measurements that may be inaccurate. T_{target} is calculated in the transmission control system, and will be seen as a known input signal. In simulation T_{target} can be calculated exactly since the friction and the driving resistance are known. Using Equations (3.31), (3.33), and setting $T_d = 0$, the angular accelerations can be set equal (with a scaling):

$$\ddot{\theta}_t = \ddot{\theta}_w i_f \Rightarrow \frac{T_{target} i_t}{J_1} - \frac{b_t \dot{\theta}_t}{J_1} = -\frac{T_{dr}}{J_2} \Rightarrow T_{target} = \frac{b_t \dot{\theta}_t}{i_t} - \frac{T_{dr} J_1 i_f}{J_2 i_t} \tag{6.1}$$

where J_1 and J_2 are defined by Equations (3.34) and (3.35).

6.2 Controller Evaluation

To decide if a gear shift is good or bad, different aspects are considered. Most important is how the driver experience the gear shift and how the gear shift affects the dynamics of the vehicle. In a real truck things like how it sounds and how it feels can be as important as the behavior of the truck. Since the controllers are developed in models, these things cannot be evaluated very well until they are implemented in a truck. To be able to compare the different controllers in the simulation environment only measurable magnitudes are considered:

- The shaft torque and the speed difference between the output shaft and the wheels just before neutral gear is engaged.
- The amplitude of the oscillations in the output shaft speed relatively the wheel speed after engaging neutral gear.
- The total shift time for neutral engagement.

The shaft torque and the speed difference is measured at the last sample in engaged model, just before the switch to the disengaged. The speed difference is proportional to the derivative of the shaft torque, because the shaft torque mainly depends on the torsion.

$$T_d \approx k_{eng} \left(\frac{\theta_t}{i_f} - \theta_w \right) \Rightarrow \dot{T}_d \approx k_{eng} \left(\frac{\dot{\theta}_t}{i_f} - \dot{\theta}_w \right) \tag{6.2}$$

The amplitude of the oscillation is calculated as the maximum value minus the minimum value after engaging neutral gear. See Figure 6.1. The total shift time is the time from when the gear shift is demanded until neutral gear is engaged.

What the driver perceives as discomforting is the acceleration and jerk of the vehicle, i.e. the value of the shaft torque and its derivative acting on the wheels. As stated by Equation (6.2) the shaft torque derivative is proportional to the speed difference. The wheel speed is constant or at least linear during the control time, so variations in the output shaft speed is a measure of how discomforting the gear shift is perceived by the driver.

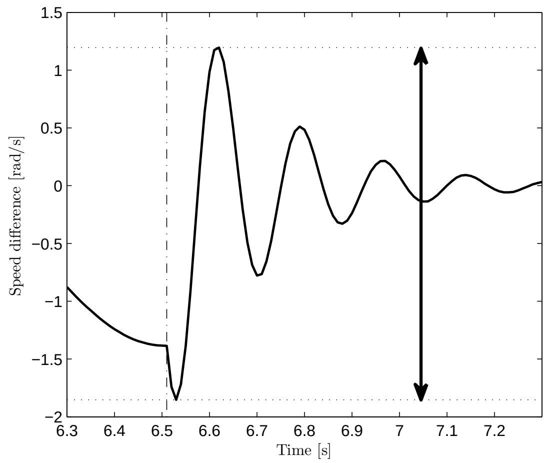

Figure 6.1. How the amplitude is calculated to evaluate shift quality. $\dot{\theta}_t i_f - \dot{\theta}_w$ (solid) and time when neutral is engaged (dash-dotted).

6.3 Open-Loop Control vs Closed-Loop Control

The main reason for using open-loop control is the simpleness of the controller. A major problem with open-loop control is that the initial state of the driveline is unknown when starting a gear shift. If the driveline is stationary when starting a gear shift it is possible to find a open-loop control that works well enough. However, if the driveline oscillates at the start of the gear shift the result will generally be bad since the open-loop control is incapable of adapting. The timing is critical when controlling the torque to zero, and a good open-loop control is dependent on the ability to foresee the oscillations of the driveline which is impossible if the initial state is not stationary. The torque delay and the blow delay, see Section 3.1 and 5.1.1, also have to be well known to be able to time the neutral engagement.

The advantage with closed-loop control is that it can handle initial conditions and also if there were model errors during the controller development. When using closed-loop control the control delay has to be considered. The control delay is the time from a measurement until the time when the control signal based on that measurement affects the vehicle. In a truck the control delay mostly consists of CAN-delays and the torque delay. If the system is too fast relative the control delay the closed system will be unstable. This is a problem since the control delay is almost the same for each gear. For lower gears the control delay is $5 - 10\,\%$ of the systems period. For higher gears, when the stiffness increases, the system becomes faster and the control delay becomes a bigger part of the period. For a fast system a simple P-controller will make the system oscillate, since its control signal is based on too old measurements. In open-loop control, this does not affect the shape of the control signal, it only causes a delay. Another problem with the feedback controller is the measurement signals. Measurement signals from a truck are noisy and may have a bias. This requires some sort of signal filtering in order to implement the feedback controller. The most interesting signals are also not measurable. The aim of the controllers is to control the shaft torque to zero but the torsion in the driveline is not measurable in todays trucks (how such a sensor could contribute is investigated in Chapter 10).

6.4 Parameter sensitivity

To evaluate the robustness of the implemented controllers, the parameter sensitivity will be investigated for each controller. The physical magnitudes, e.g. shaft stiffness or different delays, that affects the design parameters of the controllers are varied. The plant model is kept constant during all simulations (the parameters are only changed in controller code). Also the sensitivity against initial conditions will be investigated. To test different initial conditions the model is simulated during a tip-in maneuver and gear shifts are commanded after 1 s, 1.25 s, 1.5 s and 1.75 s, see Figure 6.2. The results of the parameter sensitivity investigations will be presented in the section of each controller respectively.

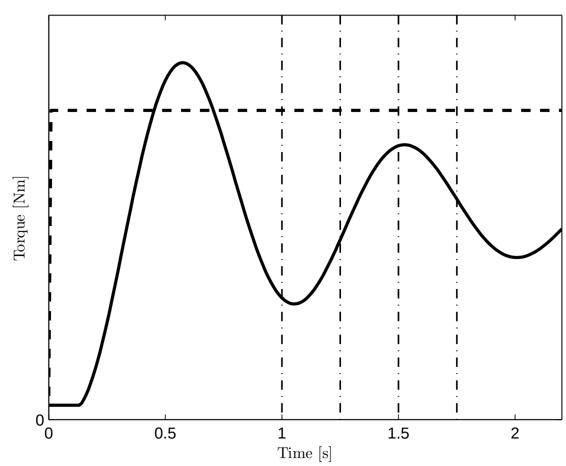

Figure 6.2. A step in the reference torque (dashed and scaled) results in oscillations in the shaft torque (solid). Gear shifts are commanded at different times after the step to investigate how initial conditions affect the controllers.

Chapter 7

Open-Loop Control

This chapter describes different open-loop controls that are easy to implement in both a simulation environment and in a heavy duty truck. The idea behind each controller is explained together with simulation results and parameter sensitivity analysis. The control strategies with the best simulation results was also implemented and tested in a truck. The results from the tests are presented in Chapter 9

7.1 Single Linear Ramps

A simple way of controlling the shaft torque to zero is to use a linear ramp and simply decrease (or increase if the truck is retarding) the engine torque until it reaches the value T_{target}. One important issue when controlling the torque with a ramp is to decide the slope of the ramp. The shaft torque will oscillate around a state of equilibrium which is depended on the value of the reference torque. If the reference torque is continuously ramped down the shaft torque will begin to oscillate around the slope of the ramp, see Figure 7.1.

This means that if the ramp is performed on a time equal to half of the systems period, the shaft torque will be zero at exactly the same time as the ramp has reached T_{target}. This is however only true in the two inertia model where the system only has one frequency which is the resonance frequency. In a more complex model or in the reality the system will have several overtones. The overtones will cause a small dissonance in the system if they are not integer multiples of the resonance frequency and that means that the statement above will be false. However, a frequency analysis of measured data on gear two shows that the first overtone is nearly an integer multiple of the resonance frequency. The resonance frequency on gear two was 1.05 Hz and the first overtone was 2.14 Hz. Therefore the method of controlling the torque with regard to the systems period works reasonably well in a truck. Figure 7.2 shows a simulation with torque control where the ramp is performed in half a period.

The problem with performing the ramp in half a period is the value of the shaft torque derivative at the end of the ramp. In Figure 7.2 it is clear that the

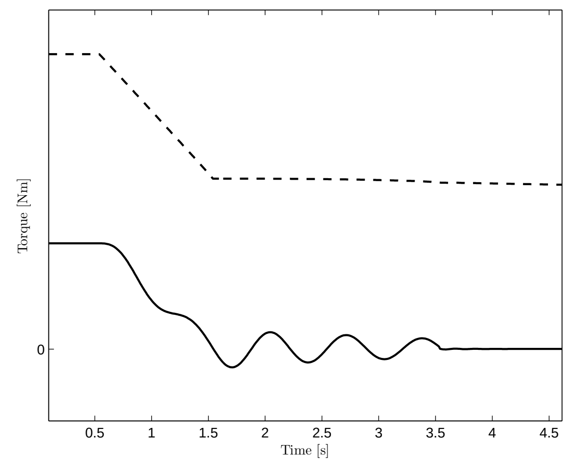

Figure 7.1. Reference torque T_{ref} (dashed) and shaft torque T_d (solid) during torque control.

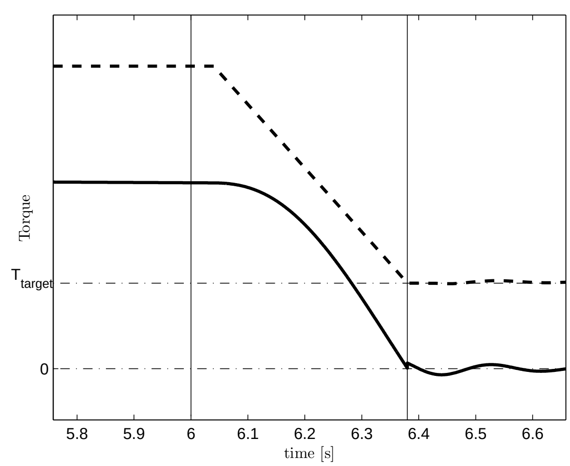

Figure 7.2. Flywheel torque T_{in} (dashed) and shaft torque T_d (solid) during torque control in half a period.

derivative has a large negative value at the end of the ramp. If the timing for engaging neutral gear is incorrect the driveline oscillations will be very bad since the value of the shaft torque is decreasing rapidly. This means that the method is very sensitive for error in the blow delay. A more robust method would be to control the torque, as well as the derivative of the torque, to zero. This makes the timing for neutral engagement less crucial since the shaft torque will not continue to decrease rapidly. If the ramp is performed in a time equal to the systems entire period the shaft torque derivative will be closer to zero than with the ramp performed in half a period. This is due to the fact that the shaft torque oscillates around the slope of the ramp. In Figure 7.3 the ramp is performed in one period. It is clear that the shaft torque derivative is closer to zero at the end of the ramp compared to Figure 7.2.

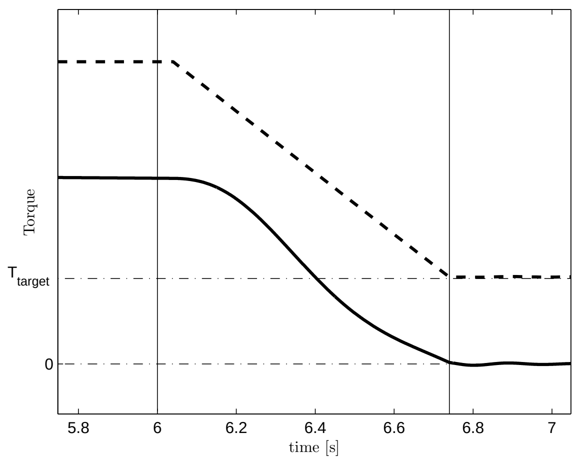

Figure 7.3. Flywheel torque T_{in} (dashed) and shaft torque T_d (solid) during torque control in one period.

Performing the ramp on a time equal to the systems period gives small oscillations in the output shaft speed. One period is however a long time for engaging the neutral gear. The period changes with the gear and on lower gears the period is in the range of one second. This is a long time considering that in order to complete the gear shift there will be additional time for synchronizing the speeds, engaging the new gear and restore the torque to the drivers demand (see Figure 1.1). Therefore it is of great interest to investigate if other methods can control both the shaft torque and the derivative to zero in a faster time than one period.

Something should also be mentioned about ramps with constant slopes. In the ramps described above the systems period was used to calculate the slope of the ramp. Another method is to have a constant slope and let the control time vary. This type of method is hard to evaluate since it will depend heavily on the torque difference between the start torque and the target torque. The smaller the difference, the faster the control time will be. Since the shaft torque oscillates

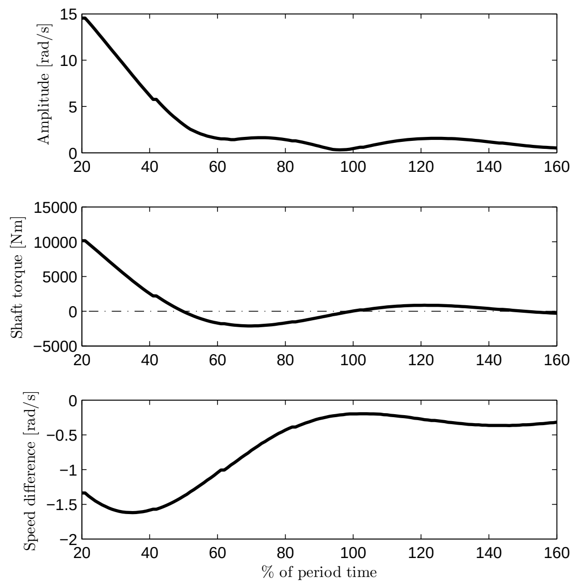

Figure 7.4. Amplitude of output shaft oscillations (top), shaft torque before neutral engagement (middle) and speed difference before neutral engagement (bottom) during a simulation sweep where the period is changed.

around the slope of the ramp, the torque derivative will sometimes be large and sometimes small depending on the control time. Furthermore, if the control time differs from a multiple of half a period, the shaft torque will not be zero at the end of the ramp since it oscillates. Therefore the ramp should have a gentle slope in order to damp out the oscillations for this method to work.

7.1.1 Parameter Sensitivity

The only design parameter for the single linear ramps is the period time. The accuracy in the period time affects the quality of the gear shift. Figure 7.4 shows the amplitude of the oscillations of the output shaft, the shaft torque and the speed difference just before a gear shift, as functions of the period time. Amplitude, shaft torque and speed difference are the ones discussed in Section 6.4. The torque passes zero at each multiple of a half period as mentioned above. The oscillations however seems to be caused by a combination of the missed target torque and the derivative of the torque at neutral engagement.

Table 7.1 shows how the different parameters affects the period. The period time is calculated using Equation (4.15). The parameters that affects the period the most are the stiffness, k_{eng} and the moment of inertia, J_{tot}. The parameter J_{tot} is the same for all trucks with the same gearbox and can be calculated with

Parameter	Change in parameter value	Change in period
k_{eng}	+10/-10 %	-4.7/+5.5 %
	+20/-20 %	-8.2/+12.0 %
	+50/-50 %	-18.5/+42.4 %
c_{eng}	+10/-10 %	+0.1/-0.1 %
	+20/-20 %	+0.3/-0.25 %
	+50/-50 %	+0.9/-0.5 %
m	+10/-10 %	+1.3/-1.5 %
	+20/-20 %	+2.4/-3.4 %
	+50/-50 %	+5.1/-11.7 %
J_{tot}	+10/-10 %	+3.4/-3.7 %
	+20/-20 %	+6.4/-7.7 %
	+50/-50 %	+14.3/-23.2 %

Table 7.1. Parameter sensitivity of the systems period with respect to different parameters. The period time is calculated using Equation (4.15).

good accuracy. The parameter k_{eng} can also be calculated from technical data but with less accuracy. The difference between the calculated value and the estimated value for k_{eng} was shown in Table 4.1 where the maximum difference was 11.3 %. An error of that magnitude in k_{eng} generates an error of 6.0 % in the period on gear four. Such an error would mean a change of 40 ms in the period on gear four, which is such a small change that it is hard to detect in the oscillations of the output shaft speed during simulation. Figure 7.4 also shows that such a small change in the period does not affect the simulation result in a significant way.

As all open-loop controllers, the ramp controller is sensitive for non stationary initial conditions. Figure 7.5 shows gearshifts at different times after a step in the reference torque using a ramp performed in one period. The oscillations are worse when the gear shift is commanded at 1 s and 1.5 s compared to the one commanded at 1.25 s and 1.75 s. This is because the shaft torque, see Figure 6.2, is furthest away from its state of equilibrium at those times. Figure 7.6 shows gearshifts with the half period ramp. In this case the output shaft speed oscillations are even larger since there is shorter time to damp out the oscillations during the ramp.

Another important thing to investigate is how sensitive the controller is for error in the blow delay. If the blow delay is underestimated the neutral will be engaged too late and if it is overestimated the neutral will be engaged too early. In order to test the linear ramp controllers sensitivity against error in the blow delay, several simulations were performed where the blow delay in the controllers was changed. As expected, the ramp performed in half a period was much more sensitive than the ramp performed in one period since the shaft torque derivative is closer to zero for the entire period ramp. Figure 7.7 shows the shaft torque at neutral engagement with different errors in the blow delay. The closer the shaft torque derivative is to zero at the end of the control time, the more robust the controller is against underestimation of the blow delay.

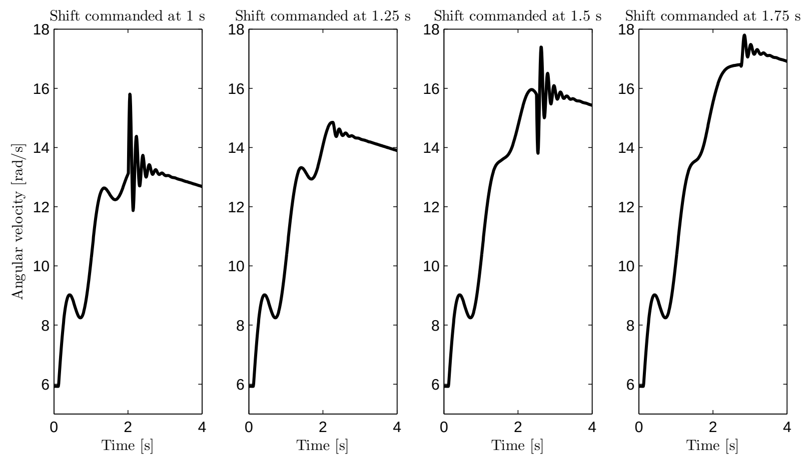

Figure 7.5. The transmission speed for different gear shifts with the ramp performed in an entire period. A gear shift was commanded at 1.0 s, 1.25 s, 1.5 s and 1.75 s after a step in the reference torque.

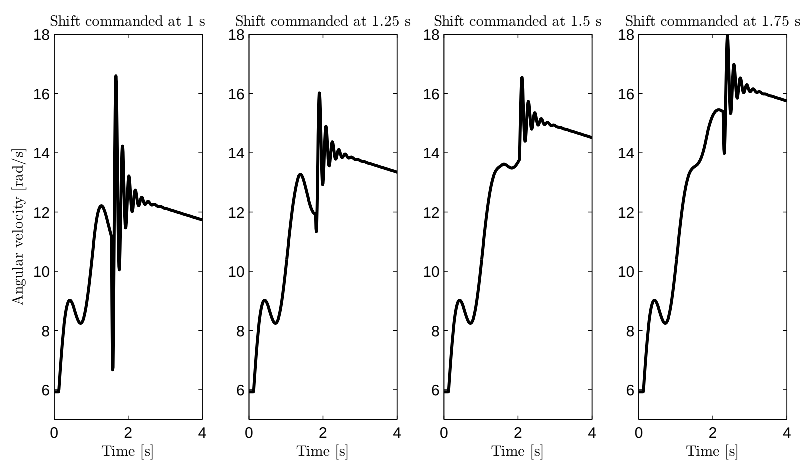

Figure 7.6. The transmission speed for different gear shifts with the ramp performed in half a period. A gear shift was commanded at 1.0 s, 1.25 s, 1.5 s and 1.75 s after a step in the reference torque.

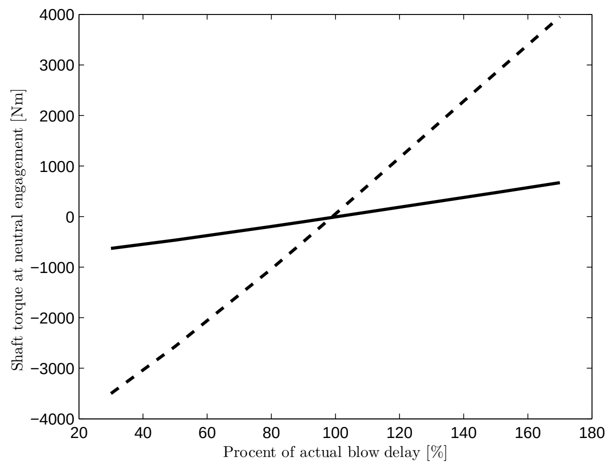

Figure 7.7. The shaft torque at neutral engagement for entire period ramp (solid) and half period ramp (dashed) with error in the blow delay.

7.2 Linear ramps with breakpoints

The ramp controller can also make a control signal that consist of two ramps with different slopes. The idea with using two ramps is to control the shaft torque aggressively in the beginning and then decrease the torque more slowly at the end to achieve a shaft torque derivative that is closer to zero at the end of the control time. The problem with using two linear ramps is where to place the breakpoint. To test where to place the breakpoint, two simulation sweeps on the second gear were made where the breakpoint was altered along both the time- and the torque-axis. The total control time was a half period in the first simulation sweep and one period in the second. The amplitude of the oscillations in the output shaft speed and the absolute value of the shaft torque and its derivative at neutral engagement was measured each iteration. The optimal placement of the breakpoint in the first simulation sweep differed when comparing the amplitude and the absolute value of the shaft torque. The best absolute value of the shaft torque at neutral engagement was achieved when the breakpoint was placed along the single ramp performed in half a period. This means that the single ramp is the best alternative to achieve a shaft torque close to zero when the control time is half a period. The torque derivative can however be closer to zero using multiple ramps with different slopes. The best amplitude of the oscillations in the output shaft speed was measured when the breakpoint between the two ramps was placed at 2.9 % of the total control time (10 ms) from the start of the control and on a level of 71 % from the target torque (see Figure 7.8). The difference from the method with one single ramp in half a period is that the derivative of the shaft torque is closer to zero when neutral is engaged. The shaft torque has however passed zero due to the steep slope before the breakpoint. Figure 7.9 shows the amplitude plotted against the breakpoint level and breakpoint time when the total control time is a half period.

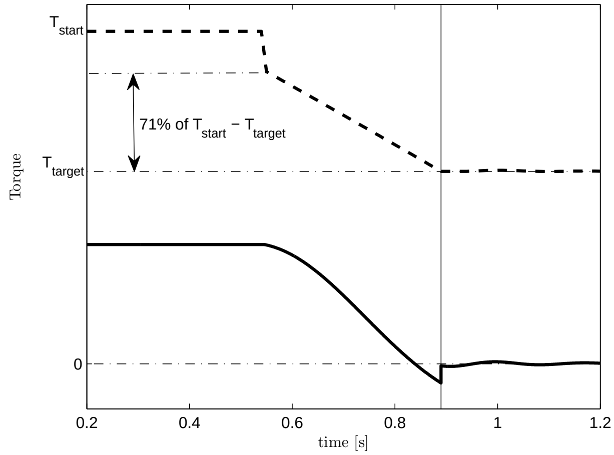

Figure 7.8. Flywheel torque T_{in} (dashed) and shaft torque T_d (solid) during torque control with two ramps in a total control time of half a period on the fourth gear. The discontinuity in the shaft torque at gear disengagement was discussed in Section 4.1.2

In the second simulation sweep, where the total control time was one period, both the amplitude and the absolute value of the shaft torque was smallest when the breakpoint was placed along the single ramp performed in one period. This is due to the fact that both the shaft torque and the derivative is closer to zero when the total control time is one period compared to half a period and therefore nothing is gained from adding a breakpoint in this case.

Also in this case the ramps could be defined by different slopes instead of time in order to control the torque change. By using constant slopes it is easy to calculate the position of the breakpoint. However, as mentioned above, constant slopes makes the controller unpredictable. Since the shaft torque oscillates around the slope, the value of the shaft torque and its derivative at neutral engagement will depend on the torque difference between flywheel torque and target torque at the start of the torque control. Even if the driveline is stationary when the gear shift is commanded, there is no certainty that the shaft torque will be close to zero at the end of the ramp when using constant slopes.

The conclusion of this control strategy is that the results are not better than with a single ramp. If the total control time is less than one period, a breakpoint can make the shaft torque derivative closer to zero. However, simulation sweeps show that the single ramp in half a period achieves the smallest absolute value of the shaft torque at neutral engagement. Furthermore the optimal placement of the breakpoint varies with different gears, total control time, damping and stiffness. There is also no obvious way how to calculate the optimal position even if the stiffness and damping are known. This makes the control method very hard to

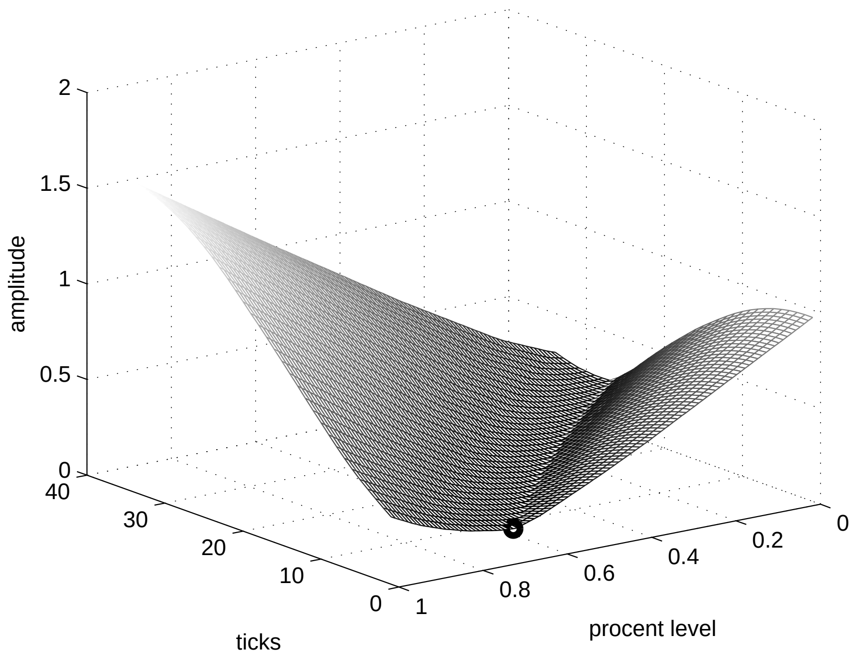

Figure 7.9. The amplitude of the oscillations in the output shaft speed after neutral engagement from gear four. The x- and y-axis shows where the breakpoint was placed. The smallest amplitude was measured when the breakpoint between the two ramps was placed one tick (10 ms) from the start of the control and on a level of 71 % from the target torque.

tune and also very sensitive for different drivelines.

7.2.1 Paremeter Sensitivity

The design parameters for the linear ramp with breakpoint are the position of the breakpoint and the total control time. As mentioned above the optimal position of the breakpoint varies with the control time. However, when the total control time was half a period, the optimal solution was the single ramp performed in half a period. With a control time in one period, the optimal solution was a single ramp performed in one period. Therefore the sensitivity analysis is the same as in Section 7.1.1. Since simulation of the model is used to calculate the position of the breakpoint, simulation sweeps needs to be done for every other control time that is used. This is time consuming and also suggests that the method is too sensitive to use in a heavy duty truck, therefore it will not be done in this thesis.

7.3 Sinus Controller

A sinus controller was implemented in order to test if something could be gained by a smoother start and finish of the ramp. If the control signal is a cosinus-function with a period that is twice as large as a specified control time, the control signal (reference torque) will look like the one in Figure 7.10. When a linear ramp was used the reference torque had very rough edges at the beginning of the control and at the end where neutral was engaged. The sinus controller gives a more smooth reference torque but the part in the middle has a steeper slope than the ramp from the ramp controller, as shown in Figure 7.11. The smoothness at the end of the reference torque makes the shaft torque derivative closer to zero, but the steeper slope in the middle makes the shaft torque decrease faster than with the single ramp. In order to test how long the control time should be, two simulation sweeps on two different gears were made where the control time was changed each iteration. The amplitude of the oscillations in output shaft speed and the value of the shaft torque at neutral engagement was evaluated each iteration. The smallest amplitude and absolute value of the shaft torque was achieved when the control time was equal to half of the systems period in both simulation sweeps. Figure 7.11 shows the shaft torque when torque control is performed with the sinus controller and the ramp controller in half a period. The results from the two different controllers are almost identical but one can see that the the sinus shaped signal makes the shaft torque decrease a bit slower in the beginning only to miss the zero torque at the end. This is due to the fact that the slope of the middle part in the sinus shaped signal is too steep. It is also clear that the control time is too short to improve the shaft torque derivative. With a longer control time the torque derivative is closer to zero but the shaft torque misses the target which induce large oscillations in the output shaft speed.

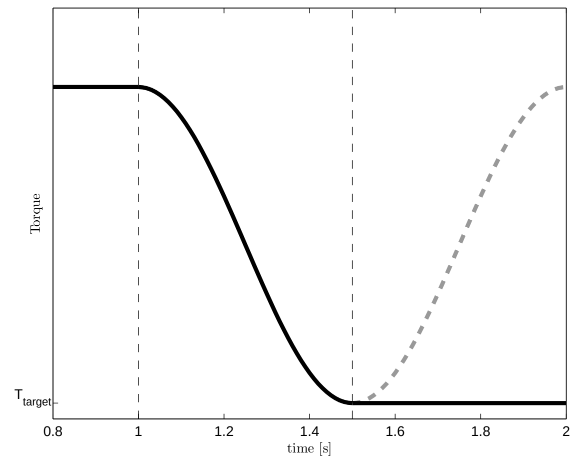

Figure 7.10. The reference torque T_{ref} is computed with the cosinus-function during the control time and is then set to T_{target}.

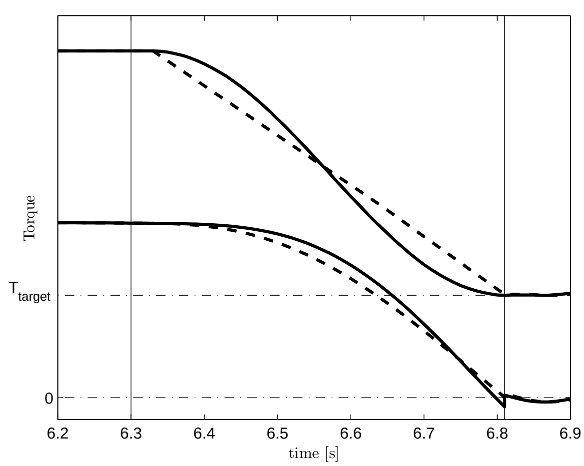

Figure 7.11. Comparison between the ramp controller (dashed) and the sinus controller (solid). Flywheel torque T_{in} and shaft torque T_d during torque control.

7.3.1 Paremeter Sensitivity

The sinus controller has only one design parameter and that is the control time. The frequency of the cosinus signal is calculated from the specified control time. However, the best result was achieved when the control time was equal to half of the systems period. Therefore the sinus controller is depended on the systems period in the same way as the ramp controller. How variations in different parameters effects the period has already been investigated in Section 7.1.1, and will therefore not be discussed here. Like all open-loop controllers the sinus controller is sensitive for non stationary initial conditions and error in the blow delay. The sensitivity for bad initial condition was tested in the same way as for the single ramps. Figure 7.12 shows how the initial condition of the shaft torque effects the oscillations in the output shaft speed. Since the control signal from the sinus controller is very slow at the beginning, the sinus controller is more unpredictable regarding initial condition than the ramp controller. The shaft torque will continue its movement until the steep slope in the middle of the control signal forces it to decrease, which means that large oscillations will occur if the initial condition is not stationary. This is illustrated in Figure 7.13 where a gear shift is commanded when the initial condition of the shaft torque is not stationary. When the control time for the sinus controller is half a period, the shaft torque characteristics is the same as when the single ramp in half a period is used. Therefore the sinus controller is equally sensitive for an error in the blow delay as the half period ramp. The absolute value of the torque derivative is large at the end of the control time which means that the shaft torque will miss zero if the neutral is engaged at the wrong time. Figure 7.14 shows how the value of the shaft torque at neutral engagement is effected of an error in the blow delay.

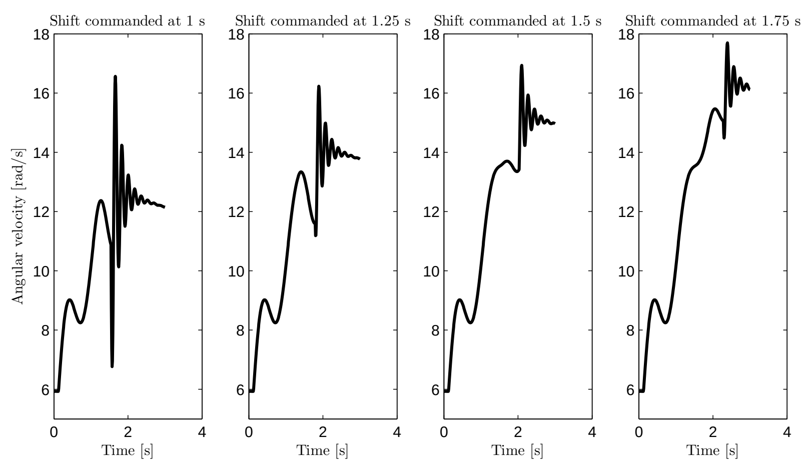

Figure 7.12. The transmission speed for different gear shifts. A gear shift was commanded at 1.0 s, 1.25 s, 1.5 s and 1.75 s after a step in the reference torque.

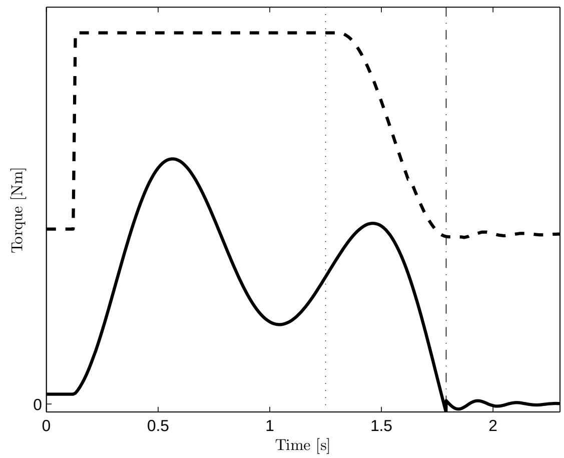

Figure 7.13. The shaft torque (solid) and input torque (dashed) when a gear shift is commanded after a step in the input torque. The vertical lines shows when the gear shift was commanded (dotted) and when the neutral was engaged (dash dotted).

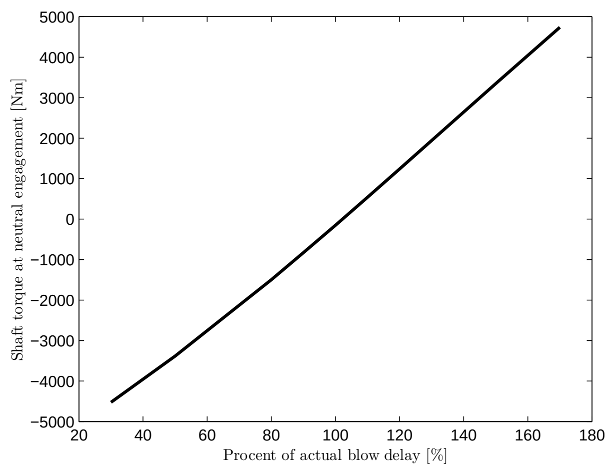

Figure 7.14. The shaft torque at neutral engagement with error in the blow delay.

7.4 Half Step Controller

When changing the load of a spring, or in this case a weak shaft, it will start to oscillate around its new state of equilibrium. It is easier to illustrate in terms of a longitudinal spring, see Figure 7.15. The spring is attached to a fix point and a force, F_1, is acting on the other end making the springs' state of equilibrium $x_{0,1}$. The idea of the half-step controller is to find a load, F_2, that results in a state of equilibrium, $x_{0,2}$ between the starting one and the one for the unloaded spring. When the load is changed to F_2 the spring will start to oscillate, which makes the force produced by the spring to oscillate as well. When the force oscillation reaches its minimum, which should be around zero, ideally the load can be removed at x_{target} without any oscillations. This means that the tension in the spring is used to swing by the middle state of equilibrium and then slow down before the turning point. See Figure 7.16. For an undamped spring the level for F_2 is exactly half the starting load, but because the spring is damped that level has to be lowered.

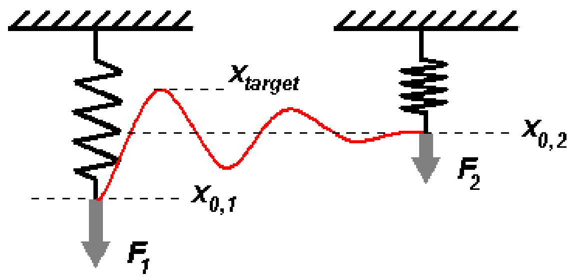

Figure 7.15. Change of load on a damped spring.

The same method works for rotational springs, i.e. the shafts in the driveline, by just replacing the forces with torques and the speed of the end with the angular velocities. But in a truck it is not that easy, there are some problems that have to be considered. The level where the middle step should be placed can be calculated exactly when driving stationary, with constant engine torque. The engine torque which corresponds to the step level is the torque that makes the shaft torque to be exactly zero at the turning point of the first oscillation. By solving equations (7.1) and (7.2) the engine torque and the time where the shaft torque is zero are found.

$$T_d(T_{in}, t) = 0 \tag{7.1}$$

$$\dot{T}_d(T_{in}, t) = 0 \tag{7.2}$$

The equations are derived and solved in Appendix A. Despite simplifications and assumptions the calculations become complex. They depend on model parameters and that the driveline torque is known at the beginning of the gear shift. The time calculated is close to half of the systems period. It should have been exactly a half period if the initial conditions were completely stationary. With the calculated values, the simulated gear shift performs well, see Figure 7.17. This is

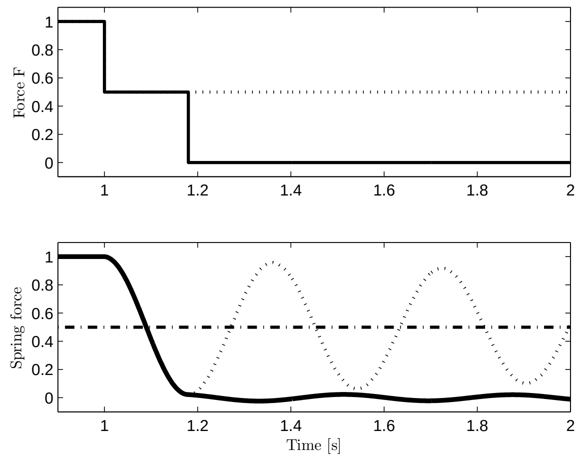

Figure 7.16. Change of load on a damped spring. Top: controlling force. Bottom: Spring force.

however not a realistic method to use in order to determine the step level since it requires knowledge of all the driveline parameters, the torsion and the initial conditions. A better approach is to use the fact that for an undamped system, the step level will be exactly 50%. The larger the relative damping of the system (see Equation (4.16)), the lower the step level will be. A step level of 50% is therefore a good starting point from which the step level can be lowered to compensate for the systems relative damping.

Instead of using half the systems period, a more robust method of using the speed difference to determine when to take the last step to T_{target} was implemented. It is important to take the step at the shaft torques minimum to avoid oscillations. Since the shaft torque can not be measured, the speed difference that is measured is used instead to decide when to take the step. If the speed of the truck is stationary at the start of the gear shift the speed difference will be zero. When the driving torque is decreased the output shaft speed will slow down making the speed difference negative. When the shaft torque has passed its state of equilibrium for the middle step the output shaft speed will accelerate again. The speed difference will increase again towards zero, and a minimum can be found, see Figure 7.18. When the speed difference has made its dip and leveled up again to zero, the torque will be near its minimum and the neutral can be engaged. Because of the torque delay in the engine, the step has to be made before the speed difference has reached its target. Therefore the time it takes for the speed difference to reach its minimum is assumed to be the same as the time it takes from the minimum back up to the level it started from. See the two times indicated in Figure 7.18.

The benefits with the half step controller is that it is fast, it is performed in a half period, and that the derivative of the shaft torque is closer to zero at the

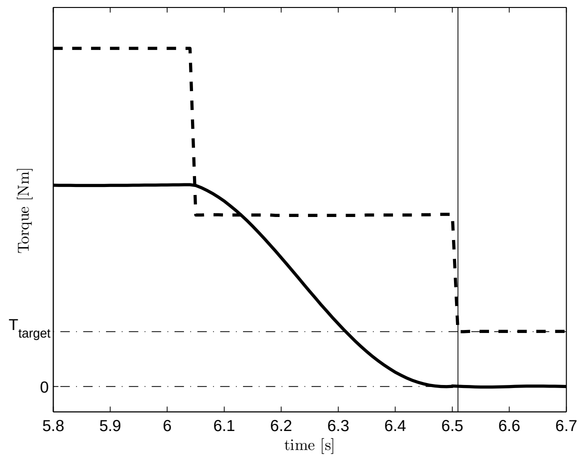

Figure 7.17. Flywheel torque T_{in} (dashed and scaled) and shaft torque T_d (solid) during gear shift with step controller.

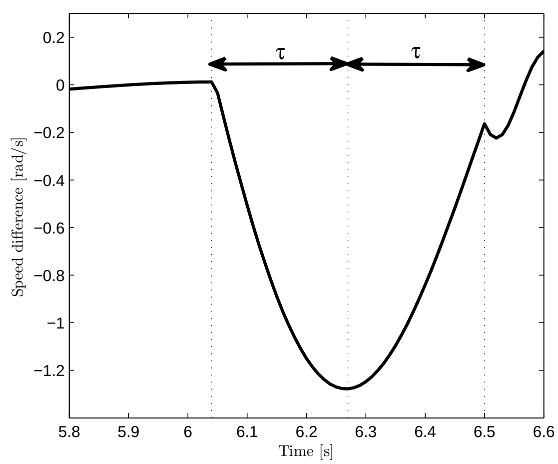

Figure 7.18. The speed difference, $\dot{\theta}_t - \dot{\theta}_w i_f$, during a gear shift using the step controller.

Change in step level	**Shaft torque before neutral [Nm]**	**Amplitude [rad/s]**
-20%	-2245	2.2
-15%	-1688	1.5
-10%	-1117	0.9
-5%	-555	0.5
0%	+7	0.7
+5%	+577	1.3
+10%	+1139	1.9
+15%	+1705	2.5
+20%	+2276	3.2

Table 7.2. How the step level affects the shaft torque and amplitude when gear shifting from gear two.

engagement of the neutral gear then for the linear ramps.

7.4.1 Parameter Sensitivity

The controller is very sensitive for changes in the step level, which is its only design parameter. A small change in the step level can cause a missed target torque by several hundred Nm. Table 7.2 shows how the change in step level affects the shaft torque just before the neutral engagement and the amplitude of the output shaft oscillations after engaging neutral. The step level is affected mostly by the stiffness k_{eng} if the step level is calculated as in Appendix A. In the calculations of the step level, a change of k_{eng} changes the step level equally much. This means that if the physical values of k_{eng} is used, the target torque will be missed by more then 1000 Nm. Mass, damping and moment of inertia do not affect the step level as much as k_{eng}, a change in those parameters only affect the step level a few percent.

If the torque delay is overestimated, i.e. the controller thinks that it is longer then its real value, the controller will take the step to T_{target} too early. Since the derivative of the shaft torque is small near neutral engagement, taking the step too early does not effect the performance much. If the torque delay is increased 50 %, it only causes the shaft torque to miss zero with a small amount of torque (6.5 Nm measured at the engine). As seen in Figure 7.17, the step to T_{target} and the neutral engagement is performed at the same time. So as long as the blow delay is correct an underestimation of the torque delay will not affect the gear shift because the neutral will be engaged anyway independently of T_{ref}. An overestimated blow delay has about the same effect as overestimated torque delay. But since the blow delay is twice as long as the torque delay, a percentage change of this will correspond to twice as large absolute time change. If both delays are overestimated the shaft torque will start to swing back up, and zero will be missed.

Another problem with the step controller, since it uses measurement signals, is the control delay. If the driveline is faster then the torque delay, i.e. the time indicated as τ in Figure 7.18 is smaller then the torque delay, the step controller will miss the minimum of the shaft torque before it can change the engine torque. The stiffer driveline, the shorter the period of the system becomes. Since the driveline stiffness increases for higher gears, the step controller works better for lower gears.

Figure 7.19 shows the simulated output shaft speed for the different initial conditions described in Section 6.4. When the shaft torque is below its mean level the ending torque will be above zero when neutral is engaged at the shift commanded at 1 s. This is because the step controller uses the initial torsion to "swing" the moment of inertia past the state of equilibrium.

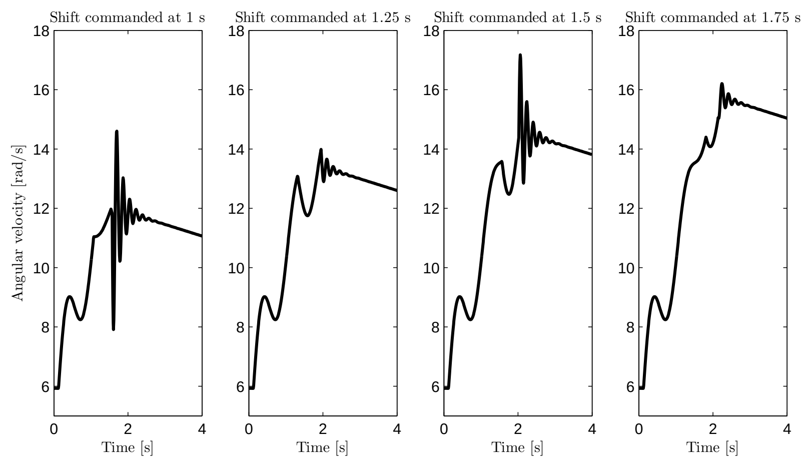

Figure 7.19. The transmission speed for different gear shifts. A gear shift was commanded at 1.0 s, 1.25 s, 1.5 s and 1.75 s after a step in the reference torque.

The step controller is very sensitive for its step level and initial conditions, but it is fast and can be used to get a low torsion with a small derivative. This can be used when speed is preferred over comfort.

Chapter 8

Closed-Loop Control

Closed-loop control is more robust against initial conditions, since measurements are fed back to the controller. But other problems occur instead, such as measurement noise and problems caused by time delays. This chapter describes the signals that can be used for feedback, the different control strategies implemented and how changes in different parameters affect the controllers.

8.1 Measurement Signals

To be able to get a good feedback, it is necessary that the feedback signals are reliable and that they have a good signal-to-noise ratio. There are several sensors in the truck, most of them with a sampling rate of 100 Hz. Other signals are created using observers. To get a good feedback controller, information about the driveline torsion is desired. As mentioned in Section 6.3 neither the shaft torque nor the torsion are measurable. This means that some other way has to be found that can give information about the shaft torque. Equation (6.2) shows that the speed difference is a good measurement of the shaft torque derivative. This will oscillate with the same period as the shaft torque, but with a different phase. Figure 8.1 shows the shaft torque and the speed difference during the torque control with the step controller. Since the drive shaft is the weakest part of the driveline it is natural to measure the speed difference between its ends. That leaves just the wheel speed at one end, but at the other end one can choose the engine speed, input shaft speed or the output shaft speed. These three speeds are almost equal due to the stiffness in the gear box, but the output shaft speed was chosen because the mechanical connection is not broken between the wheels and the output shaft when neutral gear is engaged.

The wheel speed is calculated as the vehicle speed divided with the wheel radius assuming no slip occurs, i.e. $v = r_w\dot{\theta}_w$. The vehicle speed is calculated in the brake control system based on the angular speed at the rolling wheels. Since the calculations are based on more then one sensor the signals have to be calculated and filtered which causes delays. However, the wheel speed is almost constant or at least linear during a gear shift which makes it possible to use the signal anyway.

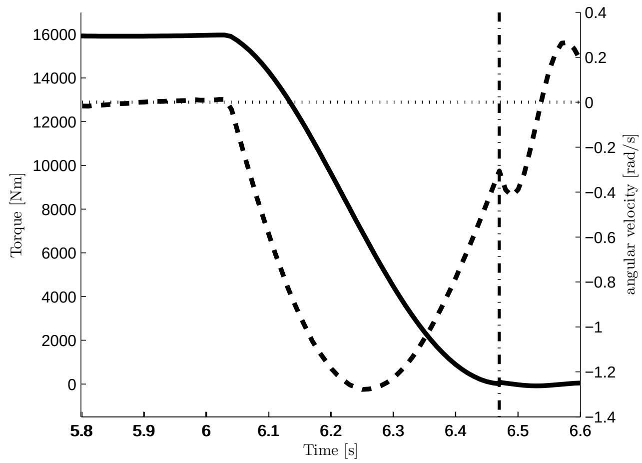

Figure 8.1. The shaft torque (solid) and the speed difference (dashed), $\dot{\theta}_t - \dot{\theta}_w i_f$, during simulated torque control with the step controller. The speed difference is a good approximation of the shaft torque.

If it is linear it just causes an offset error. Another problem that causes an offset error is different uncertain constants, e.g. the wheel radius, that are used in the calculations. The wheel radius is adapted in the brake control system since the wheel radius actually varies. The variation in the wheel radius is caused by e.g. different air pressure in the tires. A third problem is that the vehicle speed is only transmitted over CAN at 20 Hz, which makes the speed difference contain a 20 Hz noise component. To solve these problems the speed difference is filtered using a second order Butterworth band pass filter. The filter must preserve the trucks resonance frequencies to be useful, because it is used for feedback. The cut-off frequencies are chosen as 0.05 Hz and 15 Hz to adjust the offset error (damp low frequencies) and to damp noise, for example the 20 Hz component. The order of the filter is chosen as low as possible to prevent any larger control delays. Figure 8.2 shows the filters' amplitude characteristics and the filtered and unfiltered speed difference. The filtered signal varies around zero instead of around an offset error and the 20 Hz noise is damped to about half the amplitude.

8.2 D-Controller

The idea with the D-controller is that it should control the derivative of the shaft torque to zero. In Chapter 6 it was explained that the speed difference between the output shaft speed and the wheel speed is proportional to the shaft torque derivative. When the speed difference is zero, the shaft torque derivative is approximately zero. Therefore the speed difference is used as a feedback signal, so instead of using the derivative of the shaft torque, the controller uses the approx-

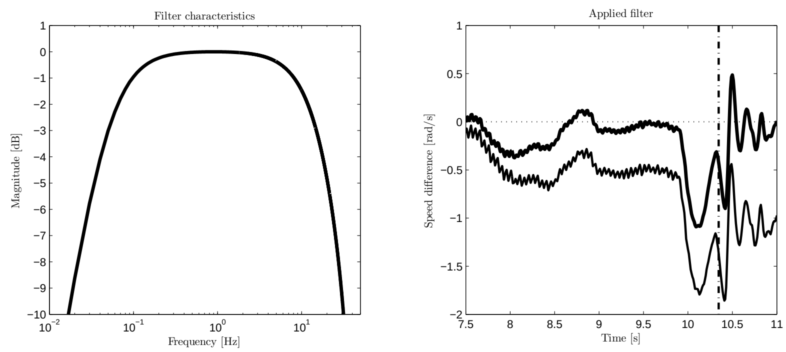

Figure 8.2. Left: The Butterworth bandpass filter characteristics. Right: The speed difference used as feedback signal, unfiltered (thin) and band pass filtered signal (bold).

imated derivative. This way there is no need to calculate the derivative from the measurements. Figure 8.3 shows the controller structure. The parameter K_d is divided by the conversion ratio of the gear box since the speed difference is proportional to the derivative of the shaft torque, but it is the engine torque that is controlled. K_d is tuned in simulations. When the D-controller is satisfied, i.e. the shaft torque derivative has been controlled to zero, the output will be zero. Therefore the target torque is added to compensate for the friction and the driving resistance. The reference signal can also be adjusted to combine an open-loop controller with the D-controller. More about this later on in the chapter.

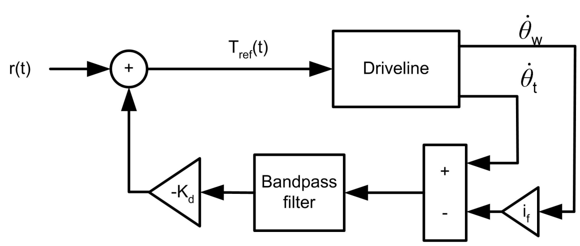

Figure 8.3. The driveline with a D-controller. The reference signal $r(t)$ is used to add the target torque or combine the D-controller with an open-loop strategy.

Since the speed difference is still noisy, a direct feedback of this would make the control signal noisy too. To avoid this, a deadzone is introduced to truncate the signal to zero for small values. The deadzone is chosen to handle most of the noise on measured signals. Figure 8.4 shows the behaviour of the controller. The problem with using just the D-controller is that the speed difference is small at the beginning of the gear shift, so the output of the controller will simply be T_{target} at the beginning. This large step in the flywheel torque will make the output shaft speed decrease very fast which feels uncomfortable for the driver. The D-controller should be used together with a reference to prevent the discomforting fast torque

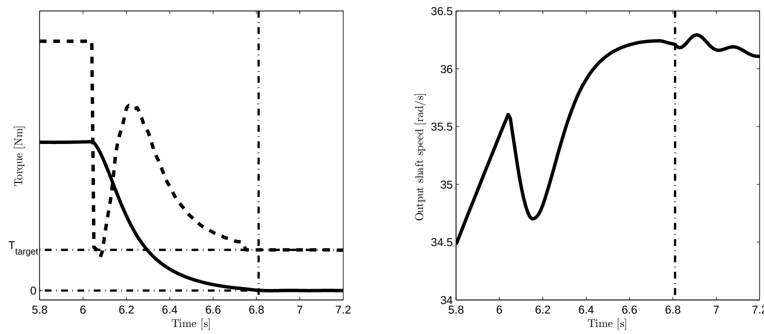

Figure 8.4. Left figure: Torque control with the D-controller, flywheel torque (dashed and scaled) and shaft torque (solid). Right figure: Output shaft speed during torque control with the D-controller.

decrease. Instead of just adding T_{target} some kind of decreasing shape can be used.

A problem with feedback, either only the D-controller or if some combination with a reference is used, is to decide when to start building up the pressure to engage neutral gear. A criteria to decide when the controller is finished is needed. In this thesis both a time out and a torque criteria is used. If the control signal is less than 50 Nm from the target for at least 80 ms or if the control time exceeds the time out, the neutral is signaled to be engaged i.e. the pressure build up is starting.

8.2.1 Parameter Sensitivity

The worst thing that can happen when using feedback is that the closed loop system becomes unstable. This can happen if K_d is too big or when the torque delay is too long, i.e. the control delay is so long that the sign of the derivative has changed before the control signal is actuated. To analyze the stability one can look at the poles of the closed-loop system. When K_d is increased, the poles are moving towards the positive side of the real axis and the system becomes unstable when the real parts of the poles become positive. It is also interesting what happens when K_d is varied within the stable area. K_d is varied from its tuned value to see how sensitive the controller is against variations in the parameter. Table 8.1 shows the result of the investigation. The magnitude of the parameter K_d affects how much the shaft torque is slowed down, so a larger value will slow down the shaft torque too much and the target torque will be missed. Since the criterion when to engage neutral is based on the magnitude of the control signal, the gear shift is faster for smaller values of K_d.

To test how much the torque delay could be varied, the torque delay was changed to being just a constant delay, i.e. setting $\alpha_{sampling}$ to zero and vary τ_i in Equation (3.2). The longer the torque delay becomes, the more oscillations will occur in the closed system until it becomes unstable. Figure 8.5 shows how well the controller works for three different torque delays. The torque delay has

Change in K_d	**Shaft torque before neutral [Nm]**	**Amplitude [rad/s]**	**Shift time**
-30%	-601	0.9677	0.68
-20%	-351	0.4815	0.69
-10%	-133	0.1154	0.72
-5%	-69	0.0935	0.74
0%	-9	0.1580	0.75
+5%	+28	0.2008	0.78
+10%	+76	0.2607	0.82
+20%	+140	0.3497	0.86
+30%	+203	0.4259	0.92

Table 8.1. Changes in the control parameter K_d for the D-controller.

been exaggerated in these simulations to test the limits. The torque delay is most

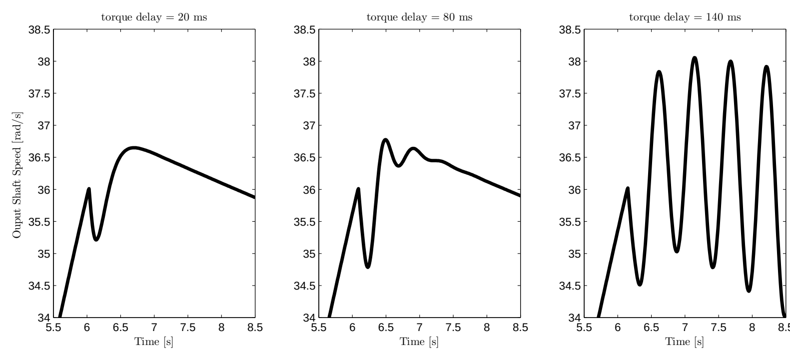

Figure 8.5. The transmission speed for different gear shifts with the D-controller. The torque delay was varied and the neutral was not engaged.

often between 30 ms and 60 ms, but for higher gears when the system is faster the delay is a problem. Since a part of the delay is CAN-delays the problem would be reduced if the torque control was moved to the engine control system, which would erase at least one tick in the CAN bus. Since the air pressure build up starts when the D-controller is satisfied, the controller is not sensitive against the blow delay. The derivative of the shaft torque is controlled toward zero, so the shaft torque will not vary much at the end of the control time.

In order to test the sensitivity against initial conditions, the gear shift is commanded at different times after a torque step, see Figure 6.2. When looking at the results in Figure 8.6 it should be compared with the corresponding pictures for the open-loop controllers, Figures 7.5, 7.12 and 7.19. The D-controller shows big improvements on handling initial conditions. The amplitude of the oscillations is at least halved compared to the ramps performed in one period, and the D-controller

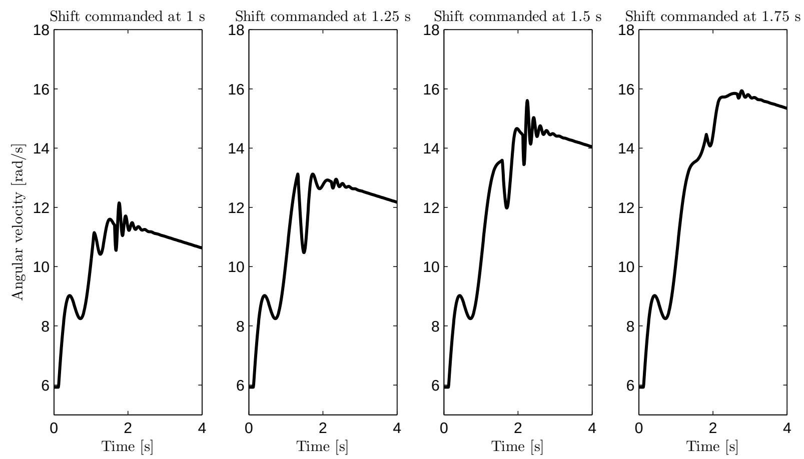

Figure 8.6. The transmission speed for different gear shifts with the D-controller. A gear shift was commanded at 1.0 s, 1.25 s, 1.5 s and 1.75 s after a step in the reference torque.

is also faster then those ramps on the lowest gears.

8.3 The D-controller with a Feed Forward

The major problems with the linear ramps described in Chapter 7 is the sensitivity against initial conditions and how to determine the systems period. These problems can be avoided by using the D-controller in addition to the linear ramp. The linear ramp will drive the shaft torque to zero and the D-controller will damp oscillations in the shaft torque due to non stationary initial conditions. In this case the D-controller does not add target torque to the control signal since the target torque is included in the ramp. The D-controller will also drive the shaft torque derivative to zero at the end of the ramp even if the control time differs from the systems period. This means that the slope of the ramp can be constant instead of a constant control time. If the slope of the ramp is too steep the fast torque change will feel uncomfortable since it effects the jerk of the vehicle (time derivative of the acceleration). Therefore a constant slope is preferred in this case since this gives the possibility to control the jerk. Another advantage with constant slope is that it is faster when the initial engine torque is close to the target torque. To test how different ramp slopes affects the combined controller, a simulation sweep varying the slope of the ramp was done. The D-controller was active through the whole ramp. As long as the D-regulator was finished by itself, i.e. the time out did not stop the controller, the quality of the gear shift in terms of output shaft speed oscillations and shaft torque before neutral was independent of ramp slope. The only difference between the different ramp slopes is how long the torque control takes. The total control time decreases the steeper the ramp slope is. If only shift time is considered, the best controller is the D-controller added with just T_{target}, but the reason to use a combination was to increase the comfort, so there have to be a trade-off between fast gear shifts and comfort. Figure 8.7 shows three simulations with different ramp slopes. It is clear that a steep ramp slope gives a faster gear

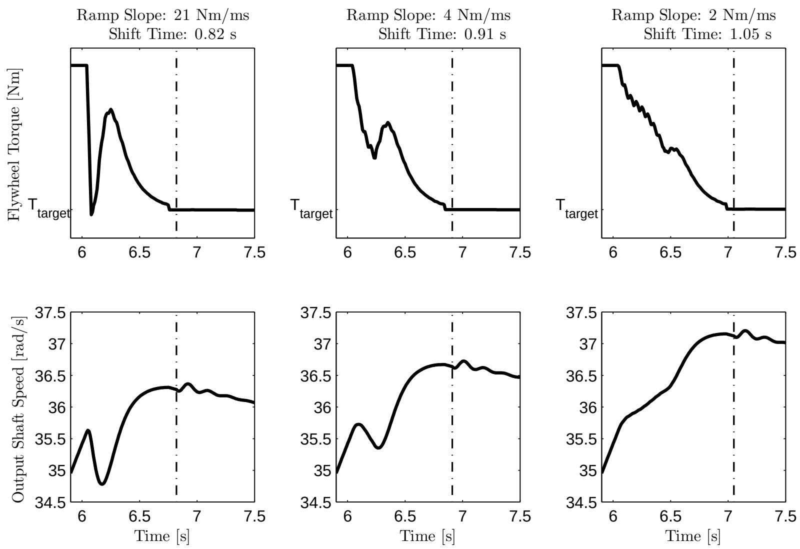

Figure 8.7. Three neutral engagements with a ramp plus D-controller with different slopes of the ramps. Top figures show the flywheel torque and the bottom figures show the output shaft speed. The vertical lines represents neutral engagement (dash dotted). The oscillations in the output shaft speed is about the same, but the sink is deeper the steeper the slope is.

shift, but at the same time a deeper, discomforting sink in the output shaft speed (before neutral is engaged). The reason why the sink in the output shaft speed is discomforting is because it is proportional to the shaft torque derivative which effects the wheels. Tests at Scania have also showed that an accelerometer on the driver seat responses to the oscillations in the output shaft speed which means that this is a good measure of comfort. With different trade-offs the controller can be used for different driving situations, e.g. when it is more time critical a steep slope is preferable.

Another issue that needs to be considered when combining the linear ramp with the D-controller is that the ramp and the D-controller have conflicting targets. The ramp aims at controlling the torque to zero and the D-controller is controlling the torque to a stationary point. Therefore the D-controller should only be turned on at the end of the ramp where it is important to improve the value of the shaft torque derivative and to damp the effect of non-stationary initial conditions. To test the timing when to turn on the D-controller during the torque control, a medium ramp slope is chosen and then the point where the D-controller is turned on is varied. It varies from being active the whole ramp to be turned on at the end of the ramp. Steep ramps will decrease the shaft torque faster, and if the D-controller is turned on too late it will not be able to control the torque to zero without making it pass zero first. As mentioned in Section 6.1 this should be avoided since it will apply a braking torque on the wheels. If the D-controller is turned on when 25% of the ramp remains, or earlier, it can handle most types of steep ramps. The shift time is shorter the later the D-controller is turned on. Figure 8.8 shows a gear shift with a medium ramp slope with the D-controller turned on when 25% of the ramp remains.

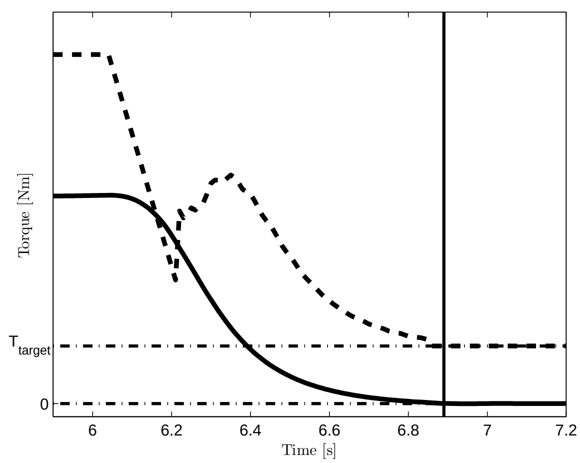

Figure 8.8. Torque control with the D-controller combined with a ramp with constant slope. The D-controller is activated at 25% of the ramp.

It is also possible to have different weights on the two control signals from the ramp and the D-controller and let the weight of the ramp linearly decrease and the weight of the D-controller linearly increase. This means that the ramp will be the dominating control signal in the beginning and the D-controller the dominating control signal at the end of the control time. Simulation with this kind of controller showed no improvements, so it was discarded.

8.3.1 Parameter Sensitivity

The design parameters in the combined controller is K_d, the ramp slope and the timing when to turn on the D-controller. How the ramp slope and the timing when to turn on the D-controller affects the results has been discussed above. K_d was varied in the same way as for the D-controller. The controller was configured as the one in Figure 8.8. Table 8.2 show the results, which is about the same as for the D-controller. The difference is that the combined controller takes longer time than the D-controller because the comfort is considered.

Sensitivity against torque delay and blow delay is the same as for the D-controller. The controller will become unstable if the torque delay is large com-

Change in K_d	**Shaft torque before neutral [Nm]**	**Amplitude [rad/s]**	**Shift time**
-30%	-559	0.8773	0.80
-20%	-315	0.4354	0.81
-10%	-107	0.1035	0.85
-5%	-40	0.0698	0.86
0%	0	0.1031	0.99
+5%	+43	0.1575	0.91
+10%	+91	0.2171	0.95
+20%	+142	0.2813	1.00
+30%	+208	0.3610	1.06

Table 8.2. Changes in the control parameter K_d for the D-controller combined with linear ramp.

pared to the systems period. Figure 8.9 shows the combined controller during a gear shift with different initial conditions. The combined controller shows addi-

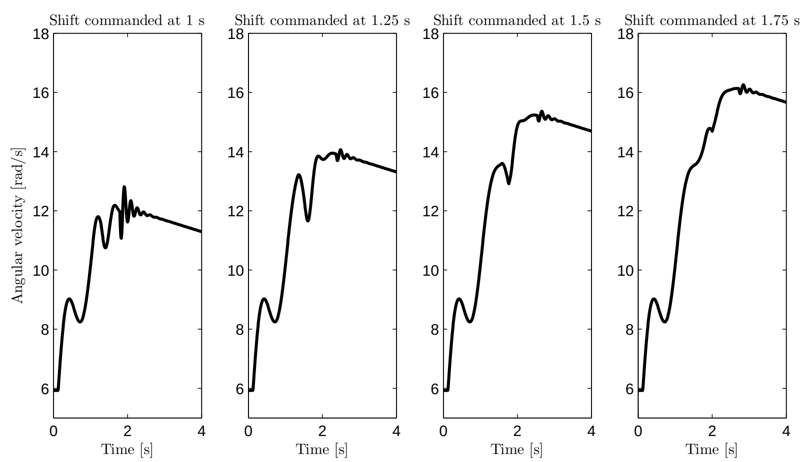

Figure 8.9. The transmission speed for different gear shifts with the combined controller. A gear shift was commanded at 1.0 s, 1.25 s, 1.5 s and 1.75 s after a step in the reference torque.

tional improvements compared to the D-controller. The sinks in the output shaft speed before neutral engagement is smaller and the oscillations are smaller after neutral engagement. This is because the oscillations are damped during the ramp as well, but the combined controller uses more time before neutral is engaged.

Chapter 9

Tests in a Truck

Some of the controllers described in Chapters 7 and 8 were also implemented and tested in two different vehicles. The vehicles have different engines, shafts and gearboxes since the first vehicle is a trailer-truck with a trailer attached and the second vehicle is a truck. The results and conclusions from the tests are summarized in this chapter. The vehicles are described more in Appendix B.

9.1 Entire Period Ramp

The ramps described in Section 7.1 was tested in both the trailer-truck Elvira and the truck Lollipop. In simulation, the shaft torque and the amplitude described in Chapter 6 was used to evaluate the controllers. Since there is no torque sensor to measure the transmission torque in a real truck, the amplitude was the only measurable indication of controller performance. In the truck Elvira it was noted from the measurements that the target torque was overestimated and therefore an offset on the target torque was used in all tests with Elvira. The control strategy with the entire period ramp felt good on lower gears but the time from gear shift command to neutral engagement becomes very long (approximately 1 s on gear two). This strategy is therefore suited in a driving situation where comfort is more important than time. On higher gears the system is much faster and one period on gear ten in Elvira is about 210 ms which instead is too short. When ramping down the torque on such a short time the driver cabin will dip due to the rapid torque change in the flywheel torque. Even if the oscillations in the output shaft speed are small it feels very uncomfortable when the driver cabin is dipping. This problem can be solved by performing the ramp in two entire periods on the higher gears, which also would make the transmission torque derivative close to zero at the end of the ramp. A recording of the speed difference between the output shaft speed and the wheel speed, when making a gear shift from gear two with the entire period ramp, is shown to the left in Figure 9.1. The total control time from gear shift command to neutral engagement was 1.09 s for the entire period ramp controller and 0.61 s for the existing controller in Elvira. The oscillations after neutral engagement are compared between the existing control strategy and the entire period ramp strategy. In simulation the existing control strategy makes the transmission torque miss the target but the overestimation of the target torque in the truck compensates for this. Nevertheless it looks like the transmission torque has passed zero since the output shaft speed first increases at neutral engagement. When the entire period ramp is used, the offset of -130 Nm is added to the target torque. In this case the offset seems to be too large since the output shaft speed first increases at neutral engagement also with this strategy. The real target torque is however believed to be close to the estimated target torque with the offset added since the output shaft speed behaves differently every other gear shift. As seen in Figure 9.1 the oscillations has a smaller amplitude with the entire period ramp strategy. This makes the comfort better but at a cost of a longer time for a gear shift.

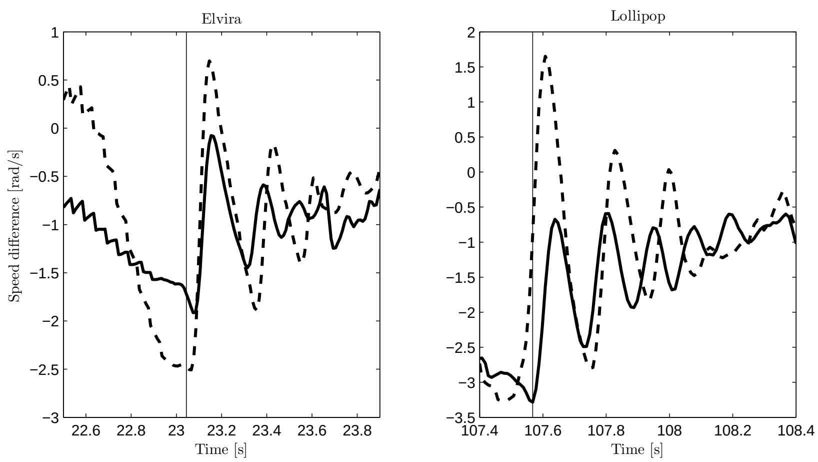

Figure 9.1. Speed difference after neutral engagement for entire period ramp controller (solid) and existing controller (dashed) in the trailer-truck Elvira (left) and the truck Lollipop (right). The vertical lines represent neutral engagement.

The ramp controller was also implemented in the truck Lollipop. The idea was to test the robustness of the controller when used in a truck with a completely different driveline configuration than the simulation model. The periods that were used in Elvira were calculated from the simulation model where the parameters have been optimized against measurements from Elvira. The truck Lollipop has however not been modeled or parameter optimized and therefore only technical data was used to calculate the period for this truck. The eigenfrequency for each gear was derived from equations (4.2), (4.11) and (4.15), where the stiffness k_{eng} was calculated from technical data of the stiffnesses in the clutch, gearbox, propeller shaft, intermediate propeller shaft[1] and the drive shafts. The damping c_{eng} was set to a fix part of k_{eng} to achieve the same relative damping for each gear.

[1]Shaft placed between the gearbox and the propeller shaft to shorten the length of the propeller shaft and to prevent the angle of the universal joint from becoming too great.

The damping has also little effect on the period, so the value of c_{eng} is not that critical. The vehicle mass that is used in Equation (4.11) was constant and known during the tests and therefore the calculation of the eigenfrequencies was done offline.

The speed difference when making a gear shift from gear two in Lollipop is shown to the right in Figure 9.1. The total control time from gear shift command to neutral engagement was 0.95 s for the entire period ramp controller and 0.56 s for the existing controller in Lollipop. The amplitude of the oscillations is smaller but at a cost of a longer time for a gear shift. The good results also show that the technical data are accurate enough to be used in a calculation of the eigenfrequency. The vehicle mass has however an impact on the eigenfrequencies and the mass changes very much in a heavy duty truck when e.g. changing the load or attaching a trailer. The calculations must therefore be done online, where a mass estimation is available, in order to use this method in a commercial product.

9.2 Half Period Ramp

The half period ramp was also tested in both Elvira and Lollipop. On the lower gears the results from the half period strategy are very similar to the existing method. A recording of the speed difference between the output shaft speed and the wheel speed when making a gear shift from gear two in the trailer-truck Elvira is shown to the left in Figure 9.2. The total control time from gear shift command to neutral engagement was 0.62 s for the half period ramp controller and 0.59 s for the existing controller in Elvira. The amplitude of the oscillations and the time for a gear shift when using the half period controller are in the same range as for the existing method on this gear. A recording from Lollipop when making a gear shift from gear one with both the existing controller and the half period controller is shown to the right in Figure 9.2. The total control time from gear shift command to neutral engagement was 0.65 s for the half period ramp controller and 0.81 s for the existing controller in Lollipop. The time for a gear shift with the existing controller is depended on the difference between start torque and target torque whereas the half period controller has a constant time for each gear. When the torque difference is big, the half period controller is faster but still achieves an equal or smaller amplitude in the oscillations of the output shaft speed compared to the existing controller. On higher gears the half period controller is too fast and the rapid torque change feels uncomfortable. The method can however be used in driving situations where a fast gear shift is more important than comfort on lower gears while another method such as the entire period can be used on higher gears.

9.3 Step Controller

In Chapter 7.4.1 parameter sensitivity analysis showed that the step controller is very sensitive against error in the step level. Nevertheless, the controller was implemented in Elvira and Figure 9.3 shows the speed difference when using the step controller on gear four in the trailer-truck Elvira. In this case the amplitude

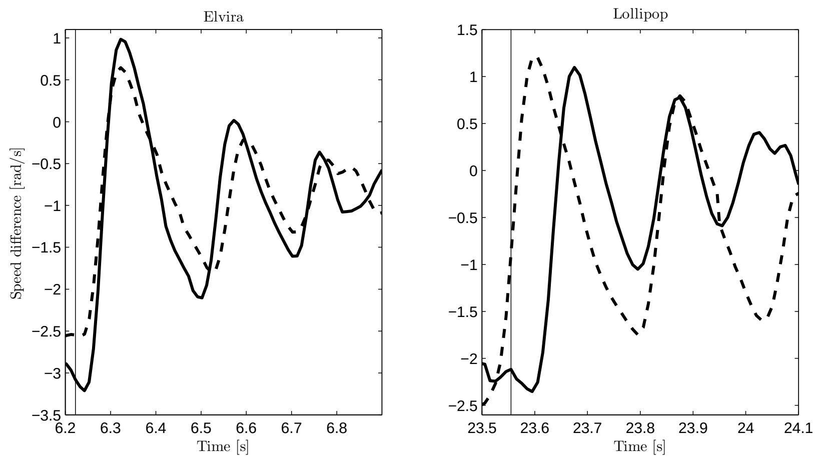

Figure 9.2. Speed difference after neutral engagement for half period ramp controller (solid) and existing controller (dashed) in the trailer-truck Elvira (left) and the truck Lollipop (right). The vertical lines represent neutral engagement.

of the oscillations was bigger with the step controller compared to the existing method. The results are however varying when using the step controller. In some cases the amplitude of the oscillations are nearly zero and in some cases the amplitude is twice as big as for the existing method. The fast torque change when making the step also makes the driver cabin dip but the time for a gear shift is very short. The conclusion is however that the step controller is too sensitive to be used for torque control in a heavy truck.

9.4 D-controller

The D-controller showed very good performance in simulations and was much more robust against initial conditions than the open-loop controllers. The feedback signal in simulation is however perfect and as discussed in Chapter 8 the feedback signal in a real truck is noisy and has an offset. It is therefore very interesting to test the controller in both Elvira and Lollipop to see if the bandpass filter and the deadzone described in Chapter 8 can eliminate these problems. The first strategy that was tested was to control with only the D-controller, i.e. the control signal will have a shape like the one in Figure 8.4. The total control time from gear shift command to neutral engagement was 0.93 s for the D-controller in all cases. For the existing controller the total control time was 0.58 s in both cases in Elvira and 0.56 s in both cases in Lollipop. The tests were performed on a plain surface and several gear shifts were made with both the D-controller and the existing controller in Elvira and Lollipop. Figure 9.4 shows the speed difference at neutral engagement when using the D-controller compared to the existing controller when making a gear shift from two different gears in Elvira and Lollipop. Figure 9.5

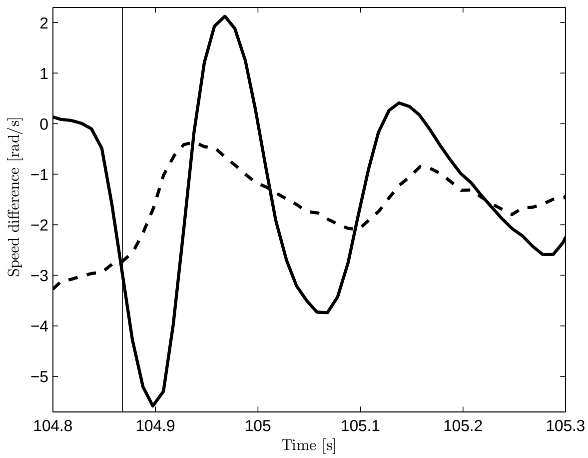

Figure 9.3. Speed difference after neutral engagement for step controller (solid) and existing controller (dashed) in the trailer-truck Elvira. The total control time from gear shift command to neutral engagement was 0.57 s for the step controller and 0.54 s for the existing controller. The vertical line represents neutral engagement.

shows a typical torque change when controlling with the D-controller.

The oscillations are better damped when using the D-controller and the controller is much more robust against initial conditions compared to the existing controller. The drawback with using the D-controller is the longer control time. As discussed in chapter 8 it is hard to know when to start building up the pressure to be able to disengage the gear. During these tests both criteria that were mentioned in Section 8.2 were used. This made the D-controller slower than the existing controller but the oscillations were better damped. A better criteria can probably shorten the control time and still damp the oscillations satisfactory but such an investigation is left for future work. Figure 9.5 shows the control signal and the flywheel torque when using the D-controller in the truck Lollipop. The shape of the control signal looks like the one in simulation but the abrupt torque change when making the step to target torque feels uncomfortable for the driver. Therefore a combination of the ramp controller and the D-controller was implemented and tested in the trailer-truck Elvira. Figure 9.6 shows the control signal and the flywheel torque when using a combination of the ramp controller and the D-controller. The ramp changes the torque in a more comfortable way and the D-controller makes the control more robust against initial condition and error in the blow delay. Figure 9.7 shows the speed difference at neutral engagement when using a ramp plus the D-controller for torque control on two different gears. The total control time from gear shift command to neutral engagement was 0.90 s for the D-controller and 0.59 s for the existing controller on gear two. On gear eight the total control time was 0.92 s for the D-controller and 0.58 s for the existing controller. In these tests the ramp was defined by a constant time but it could

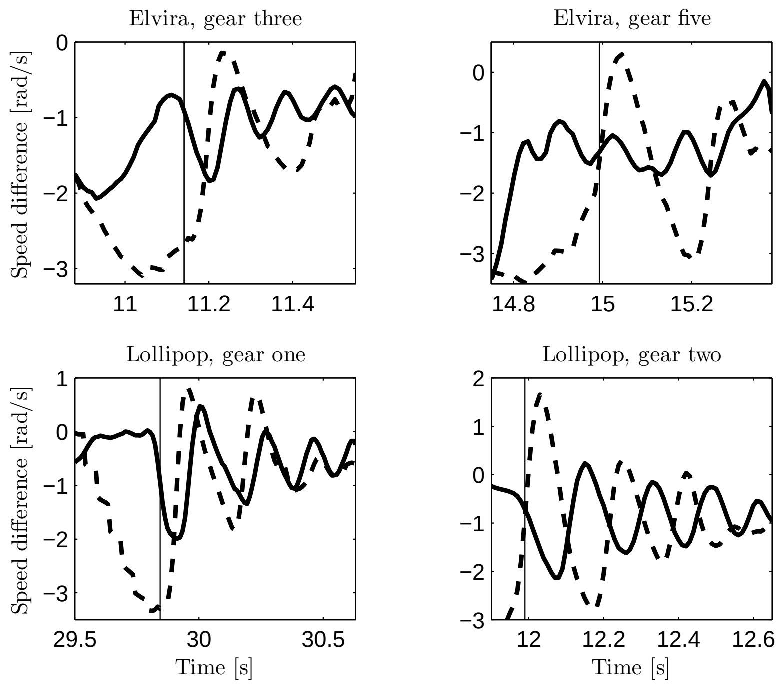

Figure 9.4. Speed difference after neutral engagement for the D-controller (solid) and existing controller (dashed). The vertical lines represent neutral engagement.

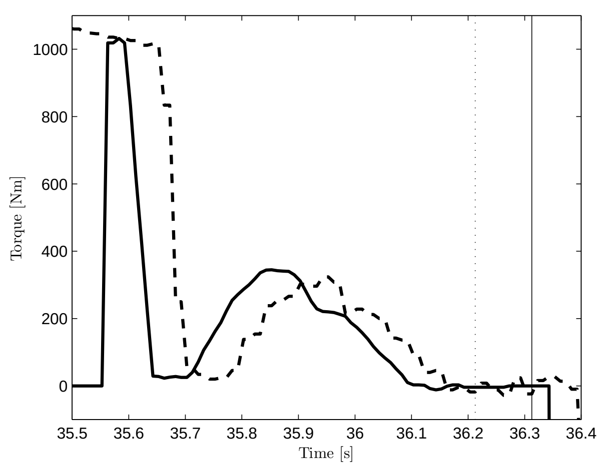

Figure 9.5. Flywheel torque (dashed and scaled) and reference torque (solid) when using the D-controller.

be defined by a constant slope instead. The D-controller will control the torque derivative to zero so there is no need to perform the ramp in a time equal to the systems period. The oscillations are better damped when using the ramp plus the D-controller compared to the existing controller and the torque change feels much more comfortable than with just the D-controller. This method also uses the criteria for building up the pressure that was mentioned in Section 8.2, which makes the control time longer than for the existing method. The comfort when using the ramp plus D-controller is improved compared to the existing controller and if the control time can be shortened with a better criteria, the proposed controller becomes very competitive since it can handle bad initial conditions and error in the blow delay much better than the existing controller.

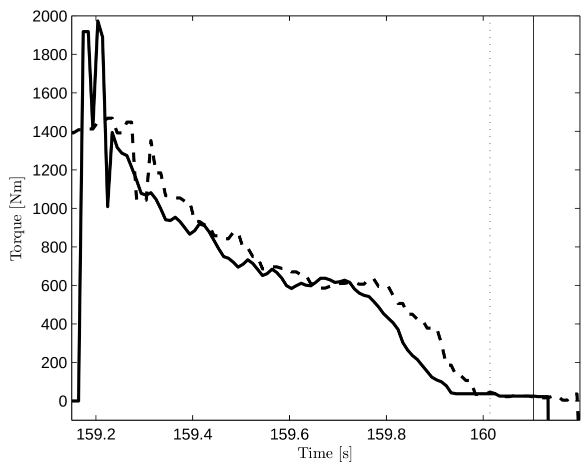

Figure 9.6. Flywheel torque (dashed and scaled) and reference torque (solid) when using a ramp plus the D-controller.

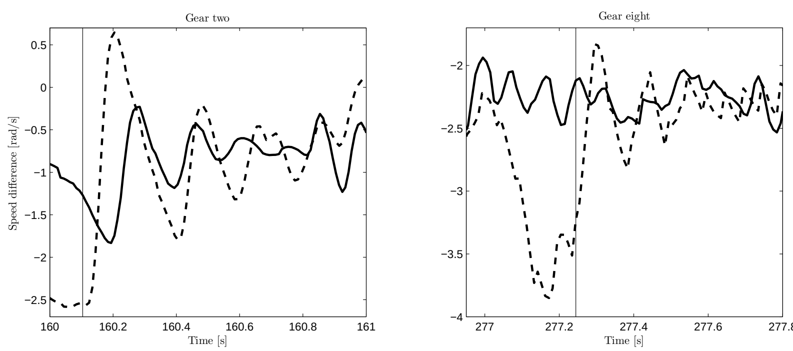

Figure 9.7. Speed difference after neutral engagement when making a gear shift with a ramp plus the D-controller (solid) and the existing controller (dashed) in the trailer-truck Elvira. The vertical lines represents neutral engagement.

Chapter 10

Contribution From a Torque Sensor

This chapter investigates if something can be gained by using a torque sensor on the propeller shaft. The idea was to use a physical sensor, but due to delivery delays it was not available in this thesis. Therefore the investigation was made in a simulation environment only. The focus of this thesis was to implement controllers based on present measurable signals and due to lack of time this investigation was not completed and no parameter sensitivity was tested.

10.1 Basic PD Structure

In the model the propeller shaft and the drive shafts are lumped together which means that they transfer the same torque. If they transfer the same torque a measurement on the propeller shaft is proportional to the drive shaft torque. In reality, the inertia from the final drive introduce some dynamics between the propeller shaft and the drive shafts. The dynamic effect is however very small as discussed in Section 3.2.2, so the assumption that the shafts transfer the same torque is reasonable. The shaft torque is used as feedback to the controller in the simulation environment.

With a torque sensor it is possible to have a PD-structure on the controller with the shaft torque as a feedback signal. In Chapter 8 the D-controller was introduced where the feedback signal was the speed difference between the output shaft speed and the wheel speed, i.e. the torsion derivative. When multiplied with the stiffness parameter, this is a good approximation of the shaft torque derivative (Equation (6.2)). Therefore the shaft torque derivative is approximately zero when the speed difference is controlled to zero. When using another reference signal than zero for the shaft torque derivative, the stiffness parameter k_{eng} need to be known if the speed difference is used as a feedback signal. It is therefore preferable to have a regular PD-structure where a derivative of the shaft torque is calculated in the controller, see Figure 10.1. This means that the signal from the torque sensor can not be too noisy and that a derivative algorithm with a filter is used in the controller. Since the concept of a PD-controller is only investigated in a simulation environment it is not possible to analyze the real sensor signal. In simulation it is possible to have a regular PD-structure or use the speed difference and no major difference was detected between the two methods when a zero reference for the torque derivative was used. Due to lack of time no other reference for the shaft torque derivative has been investigated but as mentioned above this requires that k_{eng} is known which makes a regular PD-structure preferable.

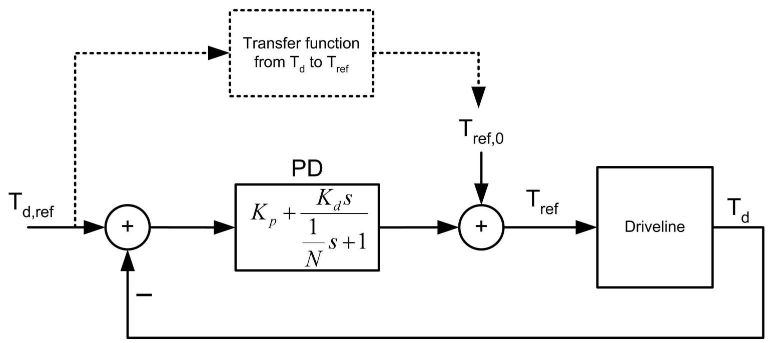

Figure 10.1. A regular PD-structure where the shaft torque derivative is calculated and low pass filtered in the controller. The reference for the engine torque $T_{ref,0}$ can be calculated using feedforward, but a simple solution is to use a ramp as $T_{ref,0}$.

In this application the engine is the actuator. The objective of the controller is to control the shaft torque to zero, i.e. control the engine torque to T_{target} in a suitable way. This means that both a reference signal for the shaft torque and the engine torque is needed, so when the error is zero there is still a reasonable control signal for the engine in order to produce the desired shaft torque. The reference for the engine torque can be calculated using feedforward from the reference signal for the shaft torque, see Figure 10.1. This requires a transfer function from T_d to T_{ref}. Unfortunately this is complicated since there are time delays between these two torques which would result in a non causal transfer function. Since the reference signal for the shaft torque is known, the problem is solvable but due to lack of time this has not been implemented in this thesis. A simple way of solving this problem is to have a ramp as reference to the engine torque independent of the shaft torque reference, so that the shaft torque will continue to decrease at a slow rate even when the error is zero. Figure 10.2 shows the structure of the PD-controller with a reference on both the shaft torque and the engine torque. With this structure the reference for the shaft torque derivative is zero and the reference for the engine torque is a single ramp as described above.

It is now possible to test different reference signals on the shaft torque. The preferred shape of the shaft torque is like the one in Figure 10.3. This would make the controller robust against error in the blow delay and still achieve zero torque in a reasonable control time. A sinus shaped signal should therefore be a suitable reference signal for the shaft torque. A regular PD-controller would

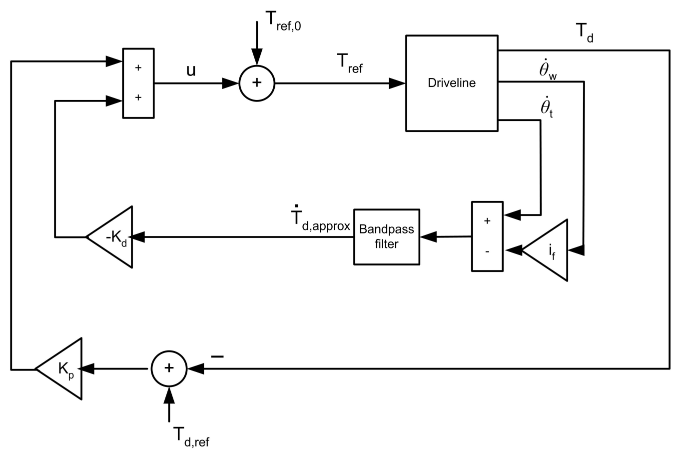

Figure 10.2. The structure of the PD-controller. In order to achieve a non-zero reference torque for the engine, the signal $T_{ref,0}$ is required. A simple solution is to choose $T_{ref,0}$ as a ramp with a reasonable slope and control the shaft torque with the other reference signal.

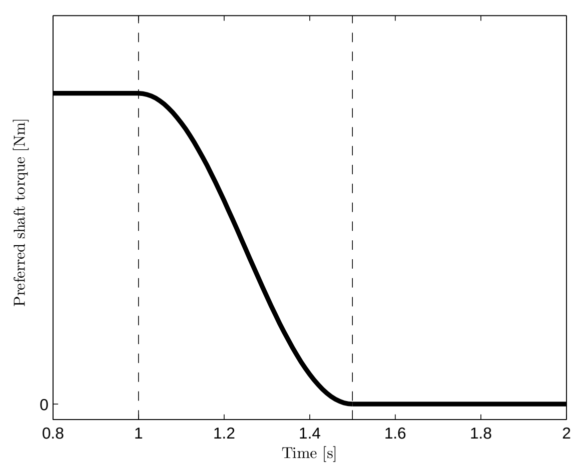

Figure 10.3. The preferred shape of the shaft torque.

calculate the reference to the shaft torque derivative automatically, but when the speed difference was used the derivative reference was set to zero.

10.2 Simulations

A problem with the D-controller is that it can level the shaft torque before it has reached zero and then be very slow at the end of the control time. With a PD-controller, the P-part will counteract the D-controller while there is a fault in the torque. Figure 10.4 shows the PD-controller and the D-controller when the references for both the P-part and D-part are zero. The neutral was not engaged in this simulation. There is an improvement with the use of the PD-controller.

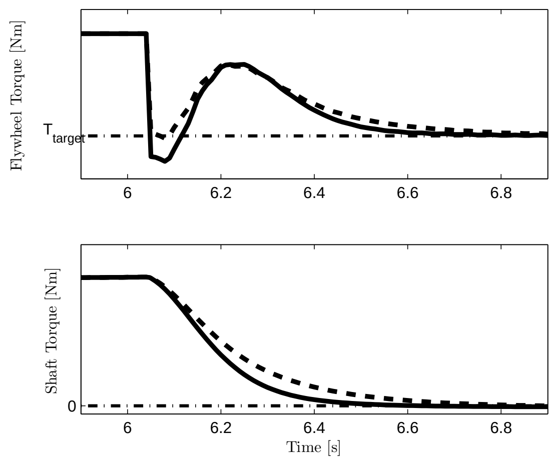

Figure 10.4. The difference between PD-controller (solid) and the D-controller (dashed).

If the same argument for engaging neutral is used as described in Section 8.2 the PD-controller engages neutral 20 ms before the D-controller. The drawback is that the shaft torque get a larger derivative which feels more uncomfortable for the driver.

Figure 10.5 shows the two controllers combined with a linear ramp. A ramp slope was chosen, and both the PD- and the D-controller was activated during the whole ramp. As torque reference in the PD-controller a sinus-shaped signal (Figure 10.3) was chosen. Half of the period in the sinus shaped signal corresponds to the time it takes for the ramp to reach the target torque. Here too, the PD-controller is faster than the D-controller, but the difference is very small. Tests have been made where the P-part of the PD-controller was increased but then the shaft torque passed zero before it stabilized. The conclusion based on these simulations is that the P-part does not have much influence on the gear shift performance. A well tuned D-controller works as well as the PD-controller. The knowledge about the shaft torque can perhaps be used in other ways, e.g. control without estimating the target torque or to decide the transition condition for the D-controller. The problem with the existing transition condition is that it can be fulfilled without the shaft torque near zero and therefore the torque sensor can be used to make a more robust criteria.

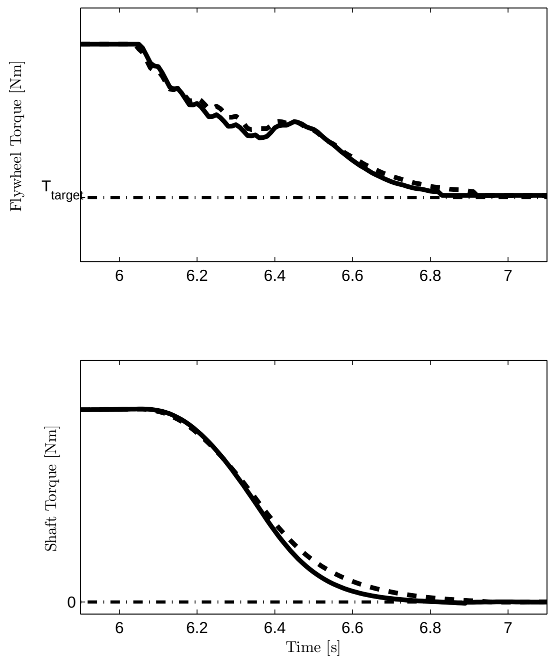

Figure 10.5. The difference between PD-controller (solid) and the D-controller (dashed).

Chapter 11

Conclusions and Future Work

11.1 Conclusions

In this thesis several open-loop strategies were presented and evaluated. The aim of the controllers is to control the shaft torque to zero so the neutral can be engaged without inducing driveline oscillations. For a more robust controller it is also important that the shaft torque derivative is close to zero since the timing for engaging neutral then becomes less crucial. When using a ramp performed in a time equal to the systems period, both the shaft torque and the shaft torque derivative will be close to zero at the end of the ramp. However, on lower gears the entire period ramp is very slow which suggests that this strategy should only be used on lower gears in a driving situation where comfort is more important than a fast gear shift. The ramp can also be performed in half a period which makes the shaft torque zero at the end of the ramp but the shaft torque derivative will be non-zero. This makes the method more sensitive against error in the blow delay but the total time for a gear shift becomes small. One problem with the ramp controller is that the systems resonance frequency must be known in order to calculate the period. This makes the controller sensitive against different driveline configurations. Tests with different trucks show however that technical data of inertias, stiffnesses and wheel radius together with an estimation of the vehicle mass is sufficient to approximate the systems period. The entire period ramp significantly reduces the amplitude of the oscillations compared to the existing method but the control time is longer on lower gears. The step controller is another concept that controls both the shaft torque and its derivative to zero. This step method controls the torque to zero very fast but the abrupt torque changes in the flywheel torque makes the driver cabin dip which feels uncomfortable for the driver. The method is therefore suited in a driving situation where a fast gear shift is more important than comfort. The major problem with the step controller is how to calculate the step level to achieve the correct state of equilibrium. Parameter sensitivity show that the controller is very sensitive to changes in the step level and to estimate the step level is a non trivial task. Other open-loop control strategies was developed such as the sinus controller and the linear ramp with breakpoints. These control strategies were implemented and tested in both Simulink and a real truck but was discarded due to bad performance and high parameter sensitivity.

A closed-loop control based on a feedback signal that is proportional to the shaft torque derivative was also developed and investigated. The simulation results with this D-controller was very promising and the shape of the shaft torque looked good. Tests in a truck showed that the real feedback signal was noisy and had an offset. This problem was reduced with a bandpass filter and a deadzone. Even though the oscillations in the output shaft speed was heavily damped, the D-controller felt uncomfortable in a truck due to the rapid torque change produced by the control signal. Therefore the D-controller was combined with a feed forward control in the shape of a ramp. This combination made the shaft torque decrease more slowly which increased the comfort and still damped the oscillations satisfactory. A problem with the closed-loop control is to decide when to start building up the air pressure in order to disengage the gear at the correct time. The control time for the closed-loop control was considerably longer than for the existing method, but it is believed that a better criteria to start the air pressure build up could reduce the control time for the closed-loop controller. In simulations the total control time was also shortened by controlling the torque with just the ramp in the beginning and then turning on the D-controller when 25 % of the ramp remains. Due to limited time this was however never tested in a truck. The major advantage with the proposed closed-loop control is that it only uses standard truck sensors and few tuning parameters. The strategy does not use any model or observer and is therefore very easy to implement in different vehicles. The only tuning parameters for the controller are the slope of the ramp, the procent level for turning on the D-controller and the feedback gain.

11.2 Future Work

The reason for the big difference in estimated data and mechanical data for the stiffnesses in the disengaged model should be investigated in future work. There is something that affects the stiffnesses that has not been modeled, like e.g. backlash or shaft inertia. The knowledge about the stiffnesses is fundamental for creating an accurate driveline model.

To make the ramp controller more robust against different driveline configurations and wear in the driveline, an adaption algorithm to estimate the systems period online is needed. A better criteria for starting the air pressure build up should also be investigated to improve the control time for the closed-loop control. In order to use the closed-loop control in a commercial product, more tests must be performed with different driveline configurations. Also the best slope of the ramp to achieve a good trade off between comfort and total time for a gear shift must be determined in different driving situations.

More work has to be done regarding the contributions from a torque sensor to draw any final conclusions. Simulations show that the torque sensor can be used to shorten the control time in a feedback controller and it is also possible to have more advanced reference signals on the shaft torque. Further investigations however needs to be done in order to determine if the contributions from the torque sensor outweighs the drawbacks with adding an extra sensor to the vehicle.

Bibliography

[1] M. Bruce. *Powertrain Modelling and Experimental Validation in a Heavy Duty Vehicle*, volume 1. AVEC '04, 7th International Symposium on Advanced Vehicle Control, April 2004.

[2] M. Bruce. *Model Based Powertrain Control, Damping of Oscillations in a Heavy Duty Vehicle*. Deparment of Signals and Systems, Chalmers university of techonolgy, 2005. Licentiate, Technical report No. R021.

[3] L. Eriksson and L. Nielsen. *Vehicular Systems*. Vehicular System, Department of Electrical Engineering, Linköping University, 2007.

[4] T. Glad and L. Ljung. *Reglerteori*. Studentlitteratur, 2 edition, 2003. ISBN:978-91-44-03003-6.

[5] M Petterson. *Driveline Modeling and Control*. Department of Electrical Engineering, Linköping University, 1997. Dissertation No. 484.

[6] M. Petterson and L. Nielsen. *Gear Shifting by Engine Control*, volume 8. IEEE Transactions on Control Systems Technology, 3 edition, 2000.

[7] L. Råde and B. Westergren. *Mathematics Handbook*. Studentlitteratur, 1 edition, 2004. ISBN:91-44-03109-2.

[8] L. Webersinke, L. Augenstein, and U. Kiencke. *Adaptive Linear Quadratic Control for High Dynamical and Comfortable Behavior of a Heavy Truck*. SAE Paper 2008-01-0534, Reprinted From: Transmission and Driveline (SP-2147), 2008.

[9] L. Webersinke, D. Feth, M. Hertweck, A. Vietinghoff, and U. Kiencke. *Optimization of Heavy Truck's Driveline Performance via Model Predictive Control rated by a Comfort Evaluation Algorithm*. 17th World Congress, The International Federation of Automatic Control, 2008.

Appendix A

Calculation of Step Level

Given the plant model described in state space formulation with the shaft torque as output and the states defined by equation (3.40),

$$\dot{x} = Ax + Bu \tag{A.1}$$

$$y = Cx \tag{A.2}$$

$$C = \begin{bmatrix} \frac{c_{eng}}{i_f} & k_{eng} & -c_{eng} \end{bmatrix} \tag{A.3}$$

it can be transformed to transfer function with the formula, described in [4].

$$G(s) = C\left(sI - A\right)^{-1} B \tag{A.4}$$

But this formula requires that the initial conditions are zero. If the initial conditions are considered there will be a matrix with initial conditions as well. Equation (A.1) Laplace transformed with its initial conditions.

$$\dot{x} = Ax + Bu \overset{\mathcal{L}}{\Longrightarrow} sI\,X(s) - \begin{pmatrix} x_{1,0} & 0 & 0 \\ 0 & x_{2,0} & 0 \\ 0 & 0 & x_{3,0} \end{pmatrix} - A\,X(s) = B\,U(s) \tag{A.5}$$

If the initial conditions are seen as constant input signals, formula (A.4) can be used.

$$T_d(s) = Y(s) = C(sI - A)^{-1} \underbrace{\begin{bmatrix} B & I \end{bmatrix}}_{\tilde{B}} \underbrace{\begin{bmatrix} U(s) \\ x_{1,0} \\ x_{2,0} \\ x_{3,0} \end{bmatrix}}_{\tilde{U}} \tag{A.6}$$

To be able to calculate the step level the following assumptions are made:

- The speed difference is zero just before the gear shift, $x_{1,0} = x_{3,0} i_f$.
- The torsion is constant and known just before the gear shift, $x_{2,0} = \alpha_{d0}$.

- The driving resistance and the friction is constant during the gear shift.

This means that T_{in} can be calculated for the case with no friction or driving resistance, and the target torque can be added later to compensate for them. By inserting T_{in} as a step at time zero and the initial conditions the expression for T_d becomes:

$$T_d(s) = \frac{\frac{i_t}{i_f J_1}\left(k_{eng} + s c_{eng}\right) T_{in} + k_{eng} s^2 \alpha_{d0}}{s\left(s^2 + c_{eng} a s + k_{eng} a\right)} \tag{A.7}$$

The expressions for T_d in the time domain is then obtained by calculating the inverse Laplace transform using standard Laplace transform pairs and the Euler formulas [7].

$$\begin{aligned} T_d(t) &= k_{eng} e^{-C_1 t}\left(-\frac{C_1}{\omega_0}\sin(\omega_0 t) + \cos(\omega_0 t)\right)\alpha_{d0} + \\ &+ \frac{i_t}{i_f J_1 a}\left(1 + e^{-C_1 t}\left(-\frac{C_1}{\omega_0}\sin(\omega_0 t) + \cos(\omega_0 t)\right)\right) T_{in} + \\ &+ \frac{i_t c_{eng}}{i_f J_1 w} e^{-C_1 t}\sin(\omega_0 t)\, T_{in} \end{aligned} \tag{A.8}$$

where

$$J_1 = J_e i_t^2 + J_t + \frac{J_f}{i_f^2} \tag{A.9}$$

$$J_2 = J_w + m r_w^2 \tag{A.10}$$

$$a = \frac{i_f^2 J_1 + J_2}{i_f^2 J_1 J_2} \tag{A.11}$$

$$C_1 = \frac{c_{eng} a}{2} \tag{A.12}$$

$$\omega_0 = \sqrt{k_{eng} a - C_1^2} \tag{A.13}$$

From this expression $\dot{T}_d$ is derived. To calculate the step level and the step time the following system of equations is solved.

$$\begin{cases} T_d(T_{in}, t_{step}) &= 0 \\ \dot{T}_d(T_{in}, t_{step}) &= 0 \end{cases} \tag{A.14}$$

$\dot{T}_d$ and the expressions for T_{in} and t_{step} are obtained using symbolic toolbox in matlab. The resulting expression of T_{in} is:

$$T_{in} = \frac{k_{eng} a i_f J_1 e^{-C_1 t_{step}}\left(\cos(\omega_0 t_{step}) - \frac{C_1}{\omega_0}\sin(\omega_0 t_{step})\right)}{i_t\left(e^{-C_1 t_{step}}\left(\cos(\omega_0 t_{step}) - \frac{C_1}{w}\sin(\omega_0 t_{step})\right)\right) - i_t}\alpha_{d0} \tag{A.15}$$

where J_1, J_2, a, C_1 and ω_0 is defined above. This gives an absolute value of T_{in}. Finally the step level can be calculated as

$$stepLevel = \frac{T_{in}}{T_{start} - T_{target}} \tag{A.16}$$

Appendix B

Truck Information

B.1 Elvira

Elvira is a trailer-truck equipped with a 12,7 liter 6-cylinder diesel engine. It has a maximum torque of 2300 Nm and maximum power of 440 hp. The gearbox has 12 normal gears, two crawl gears and is equipped with the Opticruise system. During the tests a loaded trailer was attached so the total weight was about 38,5 tons.

Figure B.1. The trailer-truck Elvira.

B.2 Lollipop

Lollipop is a truck equipped with a 9,3 liter 5-cylinder diesel engine. It has a maximum torque of 1400 Nm and maximum power of 280 hp. The gearbox has 8 normal gears and is equipped with the Opticruise system. During the tests the truck was loaded so the total weight was about 15,5 tons.

Figure B.2. The truck Lollipop.