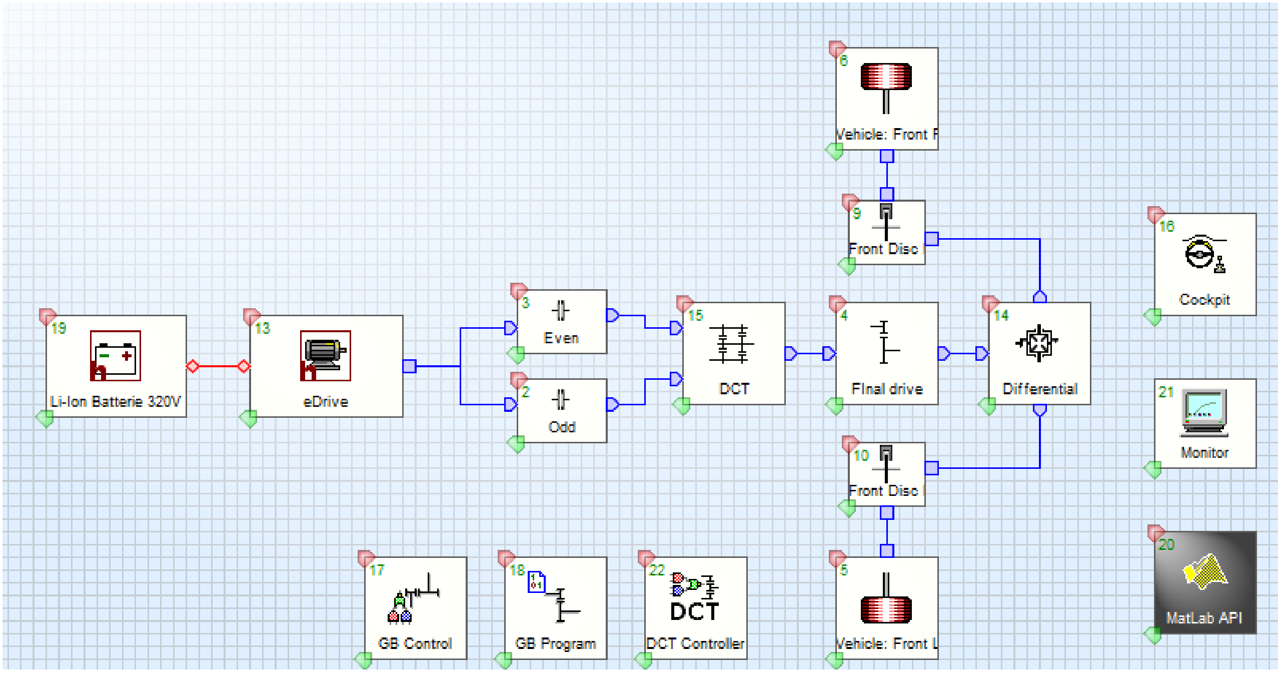

Improvement of Shift Quality in Power-On Upshift using Motor Torque Control

Master's thesis in Automotive Engineering

SIMON HERMANSSON   SHARAN VASANADU

Department of Applied Mechanics
CHALMERS UNIVERSITY OF TECHNOLOGY
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Master's Thesis 2016:13
ISSN 1652-8557
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Cover:
Dual clutch transmission simulation model in AVL Cruise. The model is presented in Section 3.2.

Chalmers Reproservice
Göteborg, Sweden 2016-06-10

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Abstract

The transmission is a very crucial part of a vehicle, especially if the prime mover is a combustion engine, since such a power source has a rather low torque at low engine speeds. There would not be enough torque to accelerate the vehicle at an acceptable rate while still reaching acceptable top speed. There is a need for several gears to achieve this, thus making the transmission mandatory in such vehicles.

Increasing demands on modern vehicles for minimizing the environmental impact are putting high requirements on transmissions. Hybrid solutions integrating electric motors are not only reducing the carbon footprint but also opening up possibilities for improving transmission shift quality.

In this thesis a dual clutch transmission propelled by an electric motor is analysed in the simulation software AVL Cruise 2015 and Matlab Simulink 2010a in order to investigate and improve the shift quality of the transmission using motor torque control.

The transmission is an automatic, powershifting, transmission which means that the torque interruption during the shift procedure is minimal. The four different phases of a shift procedure: filling phase, torque phase, inertia phase and completion phase are studied.

A Simulink program was developed including a proportional integrator (PI) controller for comparing the output torque of the transmission to the desired value and control the electric motor torque in order to achieve a smooth torque curve during the power-on upshift procedure.

It was found in the results from the simulations that the controller is effective in reducing the torque fluctuations during the power-on upshift procedure both for a low and a high load signal applied to the electric motor. In order to reduce the fluctuations even more, a special strategy aiming to improve the fluctuations during inertia phase, was applied. The result was satisfying but can be further improved and is hence a scope for future development.

Key words: Automatic transmission, Powershift, DCT, Torque control, Shift quality, Power-on Upshift, AVL Cruise, Matlab Simulink

Förbättring av växlingskvalitet under uppväxling med positivt ingående moment genom reglering av motormoment
Examensarbete inom fordonsteknik
SIMON HERMANSSON SHARAN VASANADU
Institutionen för tillämpad mekanik
Avdelningen för förbränning
Chalmers tekniska högskola

Sammanfattning

Växellådan är en mycket viktig del av ett fordon, särskilt om kraftkällan är en förbränningsmotor, eftersom en sådan har förhållandevis lågt vridmoment vid låga motorhastigheter. Det skulle inte finnas tillräckligt med vridmoment för att både accelerera fordonet snabbt nog och nå önskvärd toppfart. Det krävs flera olika växlar för att uppnå detta, vilket gör det fullständigt nödvändigt med en växellåda i sådana fordon.

Ökande behov av att minimera miljöpåverkan från moderna fordon ställer höga krav på växellådor. Hybridlösningar med integrerade elmotorer inte bara minskar koldioxidutsläppen utan öppnar också upp nya möjligheter för att förbättra växlingskvaliteten.

I detta examensarbete analyseras en elmotordriven dubbelkopplingsväxellåda med hjälp av simuleringsverktygen AVL Cruise 2015 och Matlab Simulink 2010a med målet att undersöka och förbättra växlingskvaliteten genom att använda reglering av elmotorns vridmoment.

Växellådan är en automatisk så kallad ”powershift-växellåda”, vilket innebär att vridmomentsavbrottet under växlingsprocessen är minimalt. De fyra faserna som en växling består av, fyllningsfasen, momentfasen, tröghetsfasen och avslutningsfasen, analyseras i detta examensarbete.

Ett Simulink-program innehållande en proportionell integrerande regulator (PI-regulator) utvecklades för att kunna jämföra växellådans utgående vridmoment med det önskade värdet och reglera elmotorns vridmoment för att uppnå en mjuk momentkurva under uppväxling med positivt ingående moment.

Resultatet från simuleringarna visade att regulatorn är effektiv på att reducera momentvariationerna under uppväxling med positivt ingående moment både för en låg och en hög lastsignal hos elmotorn. För att ytterligare förbättra momentvariationerna introducerades en speciell strategi med syfte att minska variationerna under tröghetsfasen. Resultatet var tillfredsställande men kan utvecklas ytterligare och är således lämpligt för framtida utvecklingsarbete.

Nyckelord: Automatisk växellåda, Powershift, DCT, Momentreglering, Växlingskvalitet, Positivt ingående moment, AVL Cruise, Matlab Simulink

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Preface

This Master's thesis focuses on improving the shift quality of an automatic, powershifting dual clutch transmission by actively controlling the input torque. AVL Cruise and Matlab Simulink were the tools used to perform the modelling and simulations needed to carry through the project. The work was carried out at AVL Gothenburg from January to June 2016.

We would like to thank our supervisors at AVL, M.Sc Joakim Karlsson and M.Sc Lars Bergkvist, for the help and support during the simulation work and report writing. We want to thank our examiner Professor Ingemar Denbratt at Chalmers University of Technology for the feedback during the report writing and guidance of finding the correct people to ask for technical support. We thank Shabbir Adil at the AVL office for both technical support and social encouragement. Last but not least we want to thank Post doc Jelena Ardic and Post doc Adrian Ilka at Chalmers University of Technology for the guidance provided regarding controller development.

Göteborg, June 2016

SIMON HERMANSSON   SHARAN VASANADU

Nomenclature

Abbreviations

CF	Completion phase
eCVT	Electric continuously variable transmission
FP	Filling phase
IP	Inertia phase
IDC	Indian driving cycle
NEDC	New European driving cycle
Ode	Ordinary differential equation
PID	Proportional integral derivative
SAE	Society of Automotive Engineers
TP	Torque phase
UDC	Urban driving cycle
VDV	Vibration dose value
WOT	Wide open throttle

Roman upper case letters

J	Inertia	kgm^2
T	Torque	Nm

Roman lower case letters

a	Acceleration	m/s^2
t	Time	s
v	Speed	m/s

Greek letters

ω	Rotational speed	rad/s
$\dot{\omega}$	Rotational acceleration	rad/s^2

1 Introduction

One of the most important factors of modern passenger cars is the impact on the environment. In order to have minimal effect on the environment the emissions need to be kept low. The internal combustion engine has been the dominating prime mover for many years and will keep on being a very important factor of both passenger and heavy transport vehicles. No matter how efficient a combustion engine or an electric motor is it will always have certain operation areas where it is as most efficient, that is at certain engine speeds and loads. In order to keep the prime mover running at these conditions as often as possible there is a need of a transmission. Increasing driving comfort demands are driving the development against more complex automatic transmissions with many ratio steps that shift smoothly and without long interruptions of torque flow. A way to use the internal combustion engine even more efficiently is to implement a hybrid concept, where an electric motor is used to propel the vehicle in the velocities where the internal combustion engine is as most inefficient.

1.1 Purpose

The purpose of this Master's thesis is to improve the shift quality of an automatic transmission by controlling the input torque. The software used are AVL Cruise 2015 and Matlab Simulink 2010a, where the former will be used to build the transmission and vehicle model and the latter to control the input torque of the transmission.

The main parameters to study in order to improve the shift quality are the output torque, the rotational acceleration of the transmission output shaft and the vehicle longitudinal acceleration.

1.2 Delimitations

The simulation model includes all necessary components for a full vehicle simulation, such as engine, clutch, transmission, final drive transmission, differential, brakes and wheels but the parameters analysed during the work are only the ones regarding the transmission and its input torque. All other parameters are set to constant values. The analysis is limited to one specific gear shift, which is a power-on upshift.

1.3 Problem definition

There are many aspects in which one can improve the performance of an automatic transmission. In this Master's thesis a specific transmission is chosen and analysed with focus on power-on upshift. The shift sequence, which can be divided into several phases is analysed in physical terms and in the simulation software AVL Cruise. A specific driving cycle is chosen and compared for different parameters of the transmission in order to achieve the best shift quality.

The shift quality can be improved by using the torque from the electrical motor in order to decrease the torque peaks that are created during the shift process. It is very important that the components acting in the different phases of the shift process are exactly timed to achieve optimal shift quality.

2 Literature review and theory

In this chapter the different sources used as background studies for this thesis are presented. There is a lot of applicable information to be found in the SAE International database, but the problem is that those studies are often very specific and hence focusing a bit deeper in areas that not exactly applies to this thesis topic.
The basic knowledge regarding planetary transmissions was encountered in (Kelly, 2012) and (Heisler, 1999) whilst more in-depth information was found in SAE papers such as (Qingkai Wei, 2015) and (Darrell Robinette, 2015). Also internal material from AVL was studied (Karlsson, 2016).

2.1 (Karlsson, 2016)

(Karlsson, 2016) describes the different shift scenarios that occur in an automatic transmission. There are in total nine cases described but only four of them are taken up in this thesis. These four are:

1. Power-on upshift
2. Power-on downshift
3. Motoring downshift
4. Motoring upshift

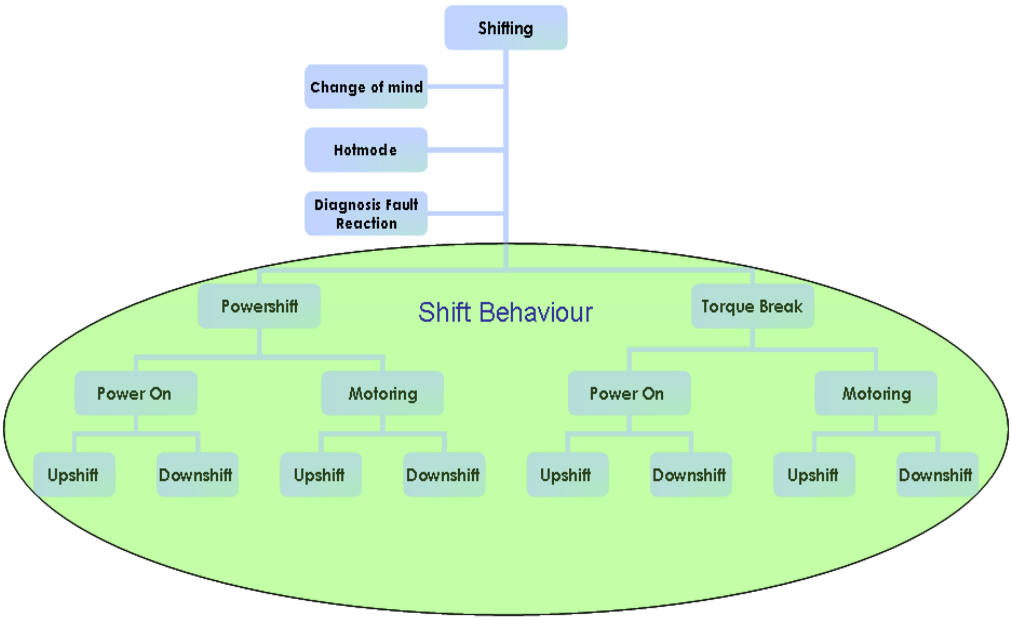


Figure 2-1 - Shifting overview (Karlsson)

The different cases can be seen in Figure 2-1.

In order to control these cases and achieve good shift quality, the following control algorithms need to be considered:

- Oncoming and off-going clutch control
- Gear actuation
- Engine torque control
- Engine speed control

The four cases described are all so called powershifts. This means that there is no torque interruption during the shift process. All cases can be divided into phases dependent on time. For instance the power-on upshift can be described in Figure 2-2. The x-axis shows the time and the y-axis shows torque and speed. The four different phases are described in detail in Section 2.1.1.

2.1.1 Power-on upshift

1. **Filling Phase**
    - Off-going clutch is released until adherence point (which means that it is on the limit to start slipping) and oncoming gear prepared by being engaged in the transmission. The red line shows the off-going clutch and the green line shows the oncoming clutch.
2. **Torque Handover**
    - When this phase starts the clutch torques are kept at reference torque (adherence point).
    - The actual torque handover of the clutches is performed which can be seen as the red line crossing the green.
    - The time for the torque handover is a function of ratio step and the different driving mode settings.

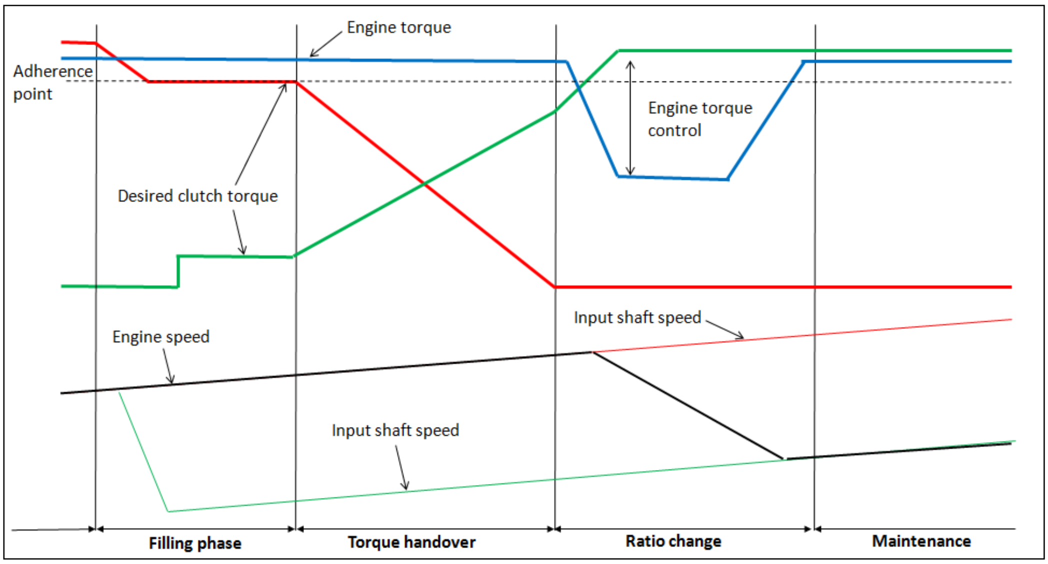


Figure 2-2 - Power-on upshift

3. **Ratio Change**
    - In the start of ratio change phase the oncoming clutch is slipping. Engine speed is still at off-going gear speed. The gear change has only been made physically in the transmission but the change of gear has yet not had an effect to the engine since the oncoming clutch is slipping.
    - The oncoming clutch needs to be closed using a calibrated engagement torque and a closed loop controller for final ratio change.
    - Typically the engine torque can be reduced in an attempt to balance the inertia torque caused by the speed change.

4. **Maintenance**
    - The maintenance phase starts as soon as the engine speed has reached the target speed for next gear.
    - The clutch torque is now set to a defined level, which is a bit higher than the reference torque in order to ensure no slip.

2.1.2 Power-on downshift

In a power-on downshift the inertia phase switches position with the torque phase. The filling phase is joined with the inertia phase as the off-going clutch torque is reduced in order to achieve an increased engine speed. The engine speed increase is desired in order not to get high torque peaks. When the engine has reached target speed the torque handover can start followed by the maintenance phase completing the shift procedure.

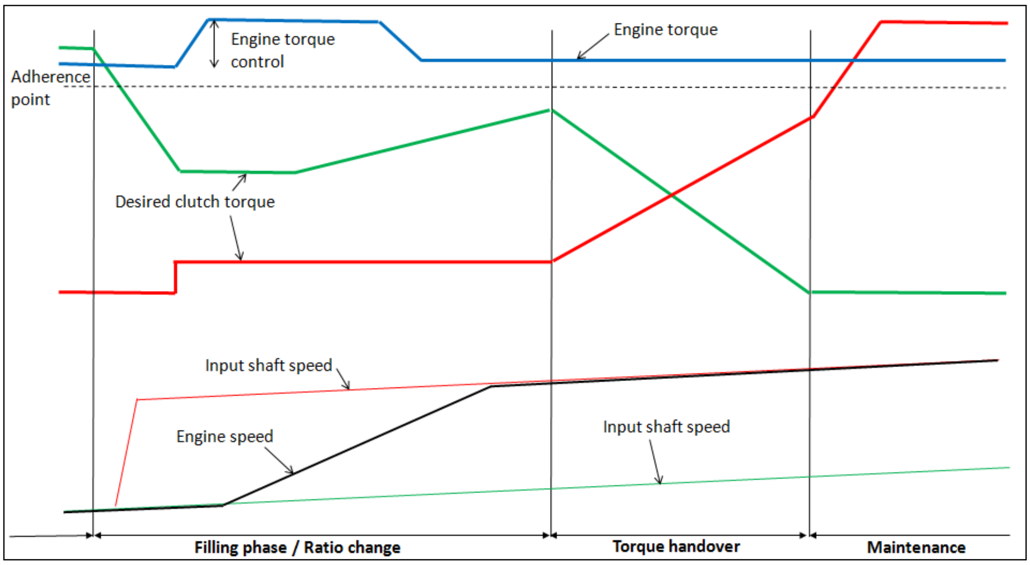


Figure 2-3 - Power-on downshift

2.1.3 Motoring downshift

The motoring downshift, which means that the engine is being propelled by the driving wheels and not vice versa (engine torque is negative), is similar to the power-on upshift. The phases are the same but the negative torque request from the engine during the torque handover has to be replaced by a positive since the engine is changing to a higher speed instead of lower as in power-on upshift. The shift procedure is presented in Figure 2-4.

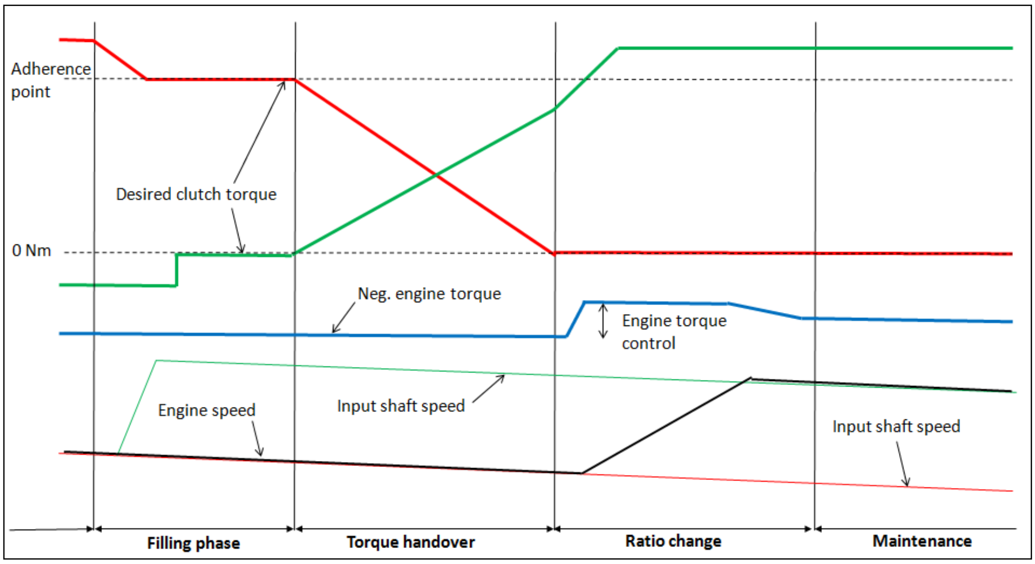


Figure 2-4 - Motoring downshift

2.1.4 Motoring upshift

This scenario is similar to the power-on downshift. The phases are the same but the positive torque request from the engine in power-on downshift has to be replaced by a negative torque request since the engine is changing to a lower speed. The shift procedure is presented in Figure 2-5.

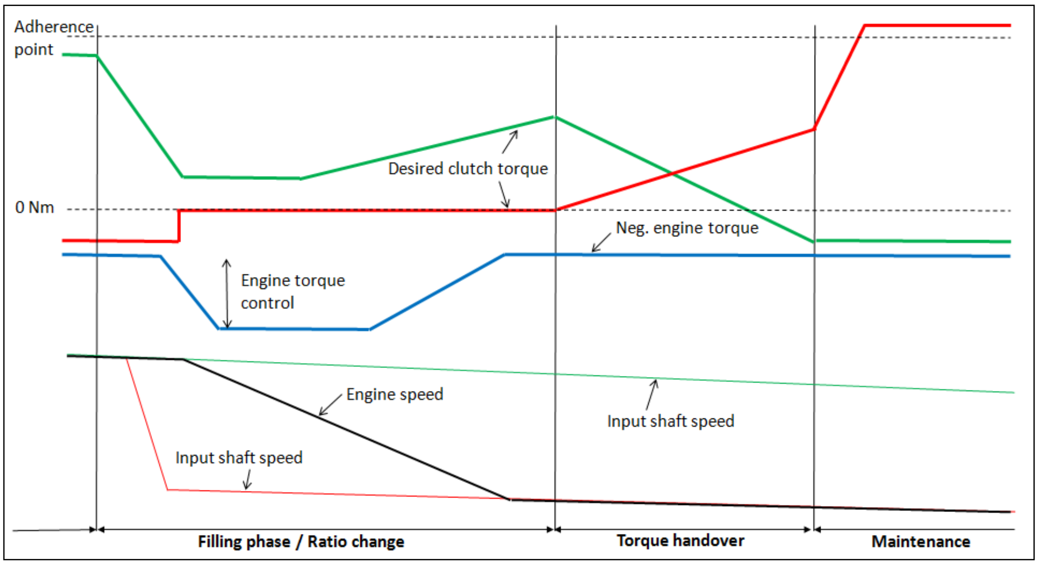


Figure 2-5 - Motoring upshift

2.2 (Heisler, 1999)

Heisler summarizes driveline design in a good way. The part used for this thesis is mainly the chapter about automatic transmissions (chapter 4) where the fundamental mechanics of single stage epicyclic (planetary) gear trains is dealt with. Figure 2-6 shows the principle of running the planetary gear in certain gears. The left part of the figure shows a forward underdrive gear, engaged by locking the ring gear, using the sun gear as input and the planet carrier as output. The right part of the figure shows a reverse underdrive gear. In this case the planet carrier is locked, the sun gear used as input and the ring gear as output.

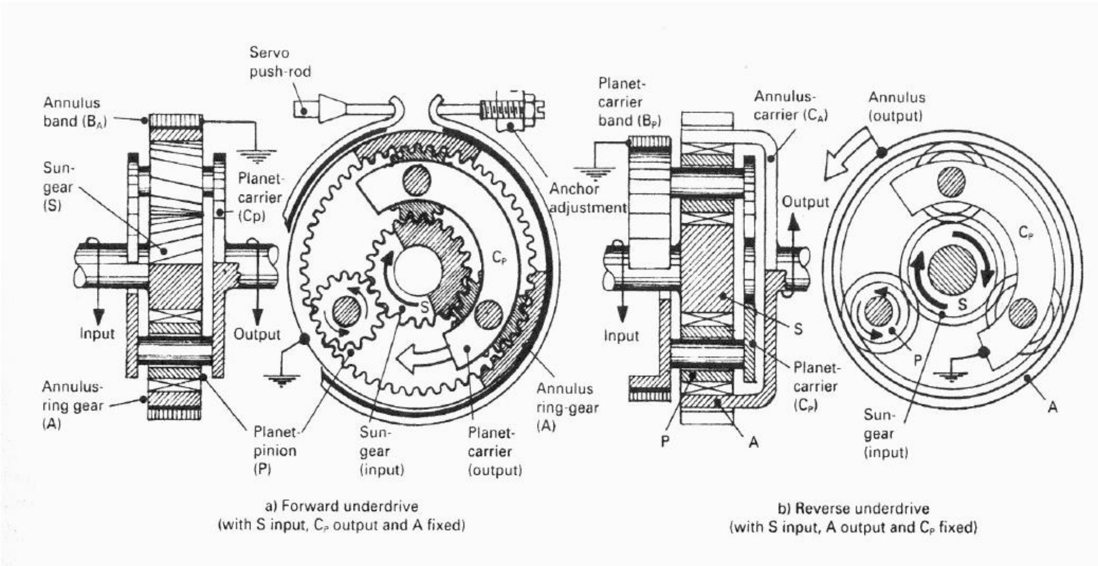


Figure 2-6 - Single stage epicyclic gear train arrangements

2.3 (Kelly, 2012)

This literature explains the basic functionality of a planetary gear. It consists of three elements: the ring gear, the sun gear and a number of planet gears that are held by a carrier (planetary carrier). In order to achieve an output torque one of these three elements needs to be locked whilst the input torque is fed to one of the two remaining elements and the output torque is achieved from the third and last element.

Kinematically this allows for 6 different combinations which all give different speeds (and torques). These are two forward underdrive, two forward overdrive, one reverse underdrive and one reverse overdrive. Normally only two of these are used, since the higher forward overdrive and the reverse overdrive are too high to be of use and the two forward underdrive require complicated switching between the elements. In order to achieve more gear ratios one can use multiple simple planetary gear sets, compound planetary gear sets or combinations of both. There are a lot of different types but one example is the Ravigneaux gear mechanism, which can be seen in Figure 2-7. It has two sun gears (1 and 3) of which both can be inputs, depending on the clutches. The two sun gears are connected to two planetary gears (5 and 6) respectively which are connected to each other through a planetary carrier, that can also act as input. Finally the ring gear (2) acts as the output. This design allows for more usable gears than for instance a single stage planetary gear but is still compact.

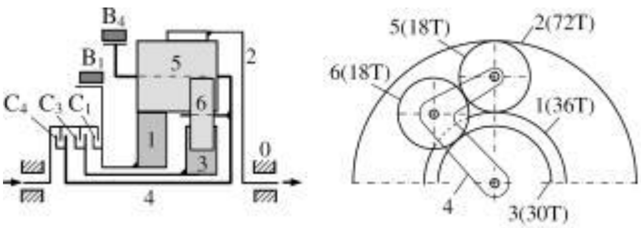


Figure 2-7 - Schematic figure of Ravigneaux gear mechanism (ASME, 2016)

One of the first steps in designing a planetary gear would be to decide the gear ratio, hence the number of teeth of the ring gear and the sun gear. One must also decide the number of pinions. Fewer than three are seldom used. Important parameters influencing this choice is the space available for the planets as well as the balancing of them.

2.4 (Qingkai Wei, 2015)

A specific shift case of an automatic transmission is analysed in this paper. It is the power-on upshift case, which means that the input torque is positive during the shift. Constant input torque is assumed and the output torque is studied during the shift procedure, which is divided in four different parts: original gear, torque phase, inertial torque and new gear. The output torque T_{out} is constant in the first phase but decreasing during the torque phase because of the off-going clutch starting to slip. During the same phase the torque is handed over from the off-going clutch to the oncoming clutch. This torque hand-over is also described by the two curves T_{B2_cap} and T_{B3_cap}. The inertial phase is initiated when the torque is totally handed over to the oncoming clutch. The clutch is then on its way to full engagement, which means that slip is going towards zero. As a result the engaged gear starts to change speed. Since there is inertia involved this will take some time.

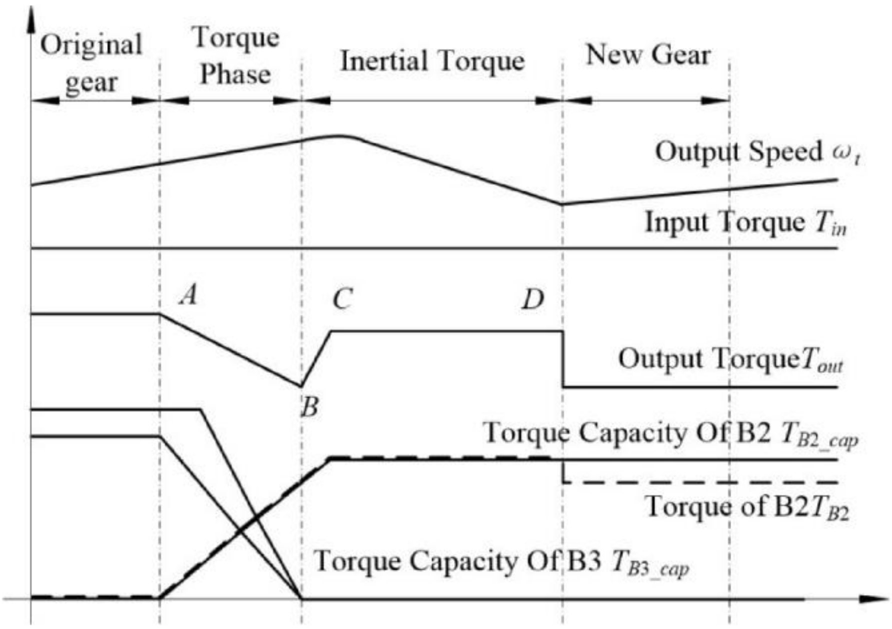


Figure 2-8 - Stages of power-on upshift without control

(Qingkai Wei, 2015) applies a control strategy to decrease the output torque peaks in two steps. The first step is to decrease the input torque during the inertial phase. As can be seen in Figure 2-9 the positive torque peak T_{out} is now almost eliminated.

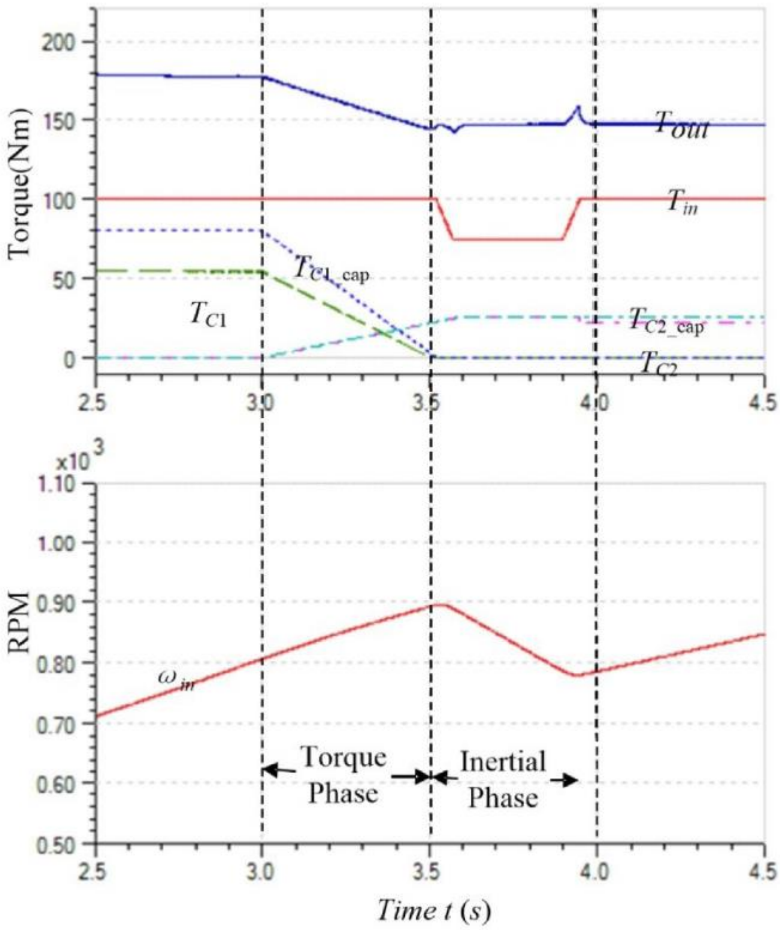


Figure 2-9 - Simulation result of integrated control strategy only in inertial phase

The second step implemented is to increase the input torque linearly in the torque phase (Figure 2-10). This counteracts to the dip in output torque that normally is the result of the torque phase.

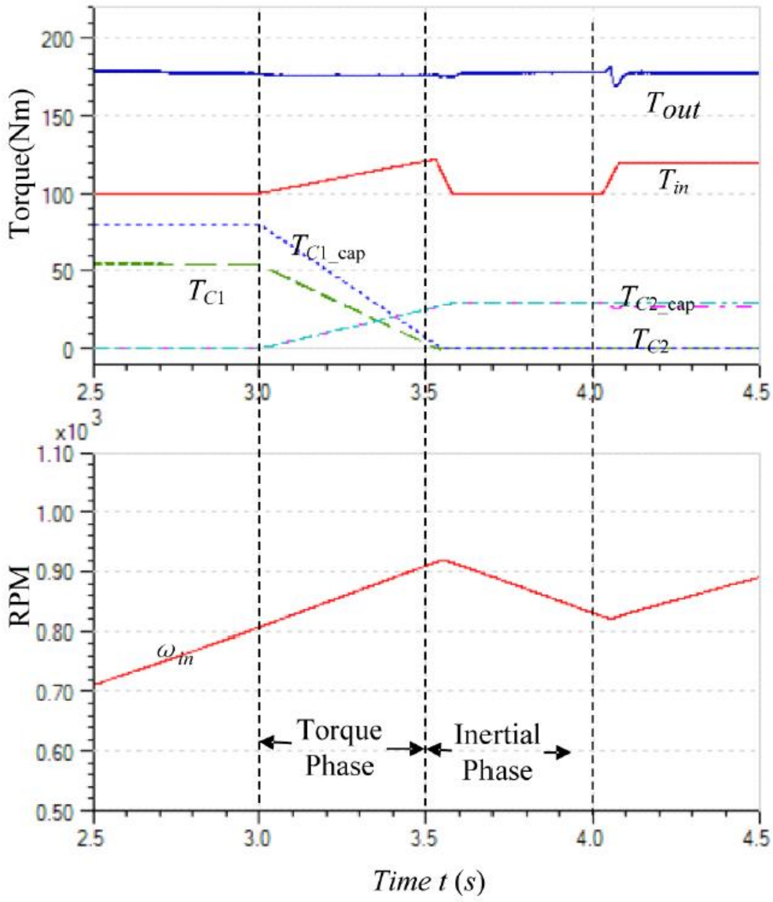


Figure 2-10 - Simulation result, integrated control based on constant output torque

It is important to not only control the input torque, but also the oncoming and off-going clutches during the shift process. The paper concludes that the torque capacity of the oncoming clutch B_2 has to be kept at the critical value (sticking) during the inertia phase in order to transfer the input torque change to the output.

2.5 (Darrell Robinette, 2015)

This paper focuses on shift quality, control and performance in single-transition, clutch-to-clutch upshifts using a simplified automatic transmission model. The reason for doing the research is the increasing number of gears in automatic transmissions creating more complexity in the power flow. Aspects like shift quality and shift time are highly affected which creates a need for investigations like this.

During the torque phase the oncoming clutch increases in torque capacity whilst the off-going clutch loses torque capacity proportionally until the oncoming clutch has reached critical torque capacity. There is no change in transmission input speed. The oncoming clutch is slipping but the off-going is not which means that power is lost in the oncoming clutch.

In the start of the inertia phase the oncoming clutch is slipping and will continue to slip until the end of the phase, when synchronization speed of the input shaft is reached. As a result the power lost from the oncoming clutch reduces from maximum in the start of the phase to minimum in the end. A way to minimize these power losses is to design the transmission in such a way that the power flow will give low lever ratios and small relative speeds across the oncoming clutch.

A method to reduce clutch energy loss during an upshift is to reduce the time of the torque and inertia phases which can be done by reducing the inertias of the rotating parts that go through a speed change. The paper shows that strategies using engine torque control by cutting the fuel supply can significantly reduce the time of WOT (wide open throttle) shifts.

Reducing the shift time does not mean that the shift quality is improved. One has to study for instance vehicle acceleration to determine shift quality. In this paper vehicle acceleration and VDV (vibration dose value) has been analysed in order to do this. Generally large ratio steps, which commonly means low vehicle speeds give high VDV (worse shift quality) whilst small ratio steps and thus high speeds give low VDV. Regarding the shift time there is no obvious and consistent trend but the conclusion is that a longer shift time gives a lower VDV than a shorter shift time.

2.6 (Fischer, 2012)

This book brings up the fundamentals of transmission of vehicles. (Fischer, 2012) describes the need for transmissions in order to achieve the appropriate wheel speed and torque needed to drive the vehicle in the required speeds and torque ranges. The book is mainly focusing on internal combustion engines as prime movers but also brings up electric motors, since hybridization is a technology that is growing and putting more demands on transmissions as well. The transmission is a powerful tool for optimising a driveline in aspects of fuel economy.

(Fischer, 2012) describes a powershift as a shift during which the torque flow through the transmission remains intact. It requires powershift capable transmissions such as step automatic transmissions (AT) or dual clutch transmissions (DCT).

The direction of shift and the direction of torque flow are two important parameters when studying a shift sequence. Shifts into a higher gear (with smaller gear ratio, i) are referred to as upshifts while the contrary are referred to as downshifts. A powered shift is defined as a shift with a positive torque on the transmission input shaft ($M_{An} > 0$), whilst the contrary ($M_{An} \leq 0$) is defined as a coasting shift. The four different shift cases that can be derived from these two criteria are shown in Table 2-1.

Table 2-1 - Shift types (Fischer, 2012)

Table 2.1. Shift types

Gear ratio	$M_{\mathrm{An}} > 0$	$M_{\mathrm{An}} \leq 0$
$i_{\mathrm{neu}} < i_{\mathrm{alt}}$	Power upshift	Coast upshift
$i_{\mathrm{neu}} > i_{\mathrm{alt}}$	Power downshift	Coast downshift

Furthermore (Fischer, 2012) describes these four shift types using a DCT transmission model (Figure 2-11) deriving physical relations from it.

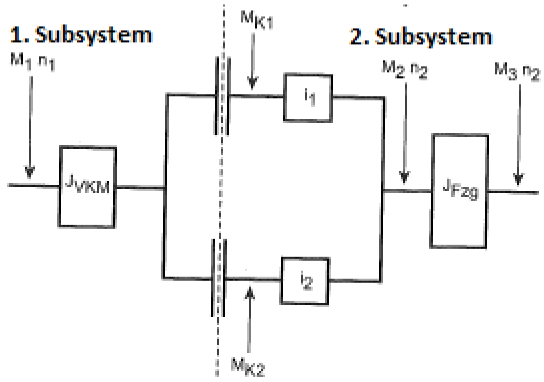


Figure 2-11 - Simulation model for ratio changes with torque fill

Torque equilibrium for the engine-side subsystem:

$$M_1 - J_{VKM}\dot{\omega}_1 - M_{K1} - M_{K2} = 0 \tag{2-1}$$

Torque equilibrium for the vehicle-side subsystem:

$$M_{K1}i_1 + M_{K2}i_2 - M_3 - J_{Fzg}\dot{\omega_{Fzg}} = 0 \tag{2-2}$$

The physical relations are expressed in the four different phases of the shift sequence in the ***power upshift*** sequence without engine torque control:

1. **Preparatory phase**
   Clutch K_1 is initially engaged and clutch K_2 disengaged. $M_{K2} = 0$. Engine torque M_1 is kept constant during the entire shift event. Output torque is determined with:

$$M_2 = i_1 M_{K1} \tag{2-3}$$

   M_{K2}=0 and M_{K1} from 2-1 results in:

$$M_{K1} = M_1 - J_{VKM}\dot{\omega}_1 \tag{2-4}$$

2. **Hand-over phase**
   In this phase the oncoming clutch K_2 starts to engage and torque is transferred via the clutch (M_{K2} increases). The torque on the off-going clutch (M_{K1}) is decreased according to 2-1. While the oncoming clutch K_2 slips, the off-going clutch K_1 sticks. At the end of the phase the engine torque M_1 is totally transferred to the oncoming clutch K_2 which makes the output torque decrease with the new gear ratio i_2 as can be seen in 2-5. Clutch K_1 must be modelled to give a transferrable torque such that:
   - The clutch sticks during the whole hand-over phase
   - The transferrable torque of the clutch is totally dissipated exactly at the end of the hand-over phase. The timing has to be correct.

$$M_2 = i_2 M_{K2} \quad \text{2-5}$$

The output torque level is at its lowest point of the shift procedure towards the end of the hand-over phase.

3. **Synchronization phase**
   The engine speed has to be changed to the new target speed, which is a lower speed for an upshift. In this case it is achieved by increasing the torque on the oncoming clutch K_2 to a higher value than the engine torque. The new engine speed is derived from 2-1, with $M_{K1} = 0$ and expressed as:

$$\dot{\omega}_1 = \frac{M_1 - M_{K2}}{J_{VKM}} \quad \text{2-6}$$

   The reduction of the engine speed causes an additional dynamic torque that is added to the output torque M_2. This torque disappears abruptly when the engine speed deceleration ends, that is when the engine has reached the target speed. This sudden torque change causes vibrations in the driveline.
4. **Completion phase**
   The shift is now concluded. The set clutch torque of the oncoming clutch K_2 is higher than actual required to make sure that there is no slip.

2.7 PID controller

A PID (Proportional integral derivative) controller is used in a feedback loop system wherein the output of a system is measured (Process variable/Output) and is compared with the desired value of the system output (Set point). Then the difference between these parameters is fed into the controller forming a negative feedback loop. The controller then tries to reduce this input error to zero and the way the controller responds can be tuned with the tuning parameters. A PID has three gains; proportional, integral and derivative. The output of the controller is the sum of product of individual gains with the input error. By selecting each of the gains appropriately, the controller is tuned to have the desired rise time, overshoot and steady-state error. Figure 2-12 shows a PID controller schematics.

The *proportional* part of the controller is the one that acts depending on the current error which is then multiplied with the P gain K_p. This gives a corresponding change in the output for a given input change.

The *integral* part of the controller is the part that takes into consideration the duration for which an error is present and acts accordingly. It integrates the error present over time and responds for the accumulated error by multiplying with the I gain K_i. It helps in reducing the steady state error and also the response time of the controller.

The *differential* part of the controller is somewhat a predictive term which senses the error trend by computing the slope of the error. It multiplies the D gain K_d with the error. The D gain controls the overshoot and gives a damping effect to the controller's response and improves the transient response.

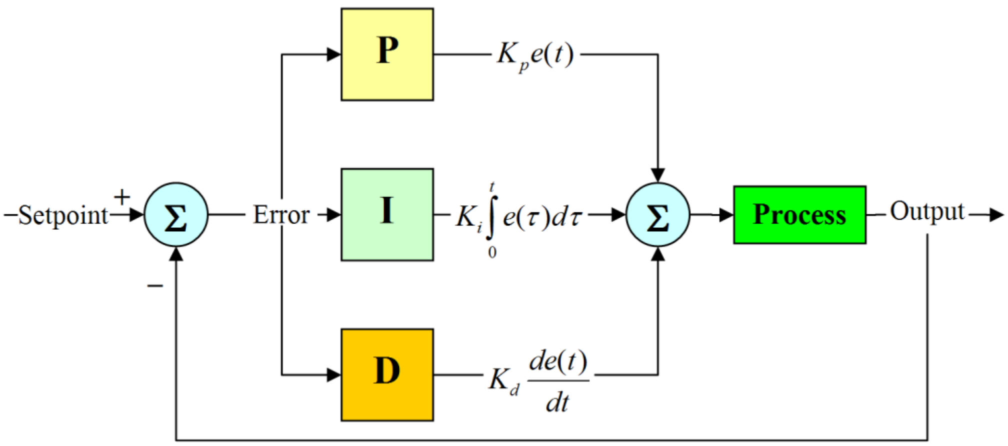


Figure 2-12 - PID Controller (Wikipedia, 2016)

3 Transmissions

When starting the work on this thesis there was a need to find a suitable transmission to perform the simulations on. The main requirement was that it had to be an automatic transmission, hence able to do powershifts. The first suggestion was to use an in-house developed transmission from AVL; the Future Hybrid and the second was a DCT transmission.

3.1 Future Hybrid Transmission

This transmission is special because of its hybrid properties. A small combustion engine is connected as prime mover and on the first ring gear of the Ravigneaux gear set of the transmission there is an electric motor connected. This makes it possible to run the vehicle as a parallel hybrid. The electric motor can be used as single prime mover at such vehicle speeds when the combustion engine would be as most inefficient, for instance when starting off from standstill. The combustion engine can be used as single prime mover as well. This is needed when the battery for the electric motor is drained. There is also a possibility to use both the electric motor and the combustion engine at the same time in order to get maximum power from the powertrain. The transmission layout can be seen in Figure 3-1 (AVL, 2016).

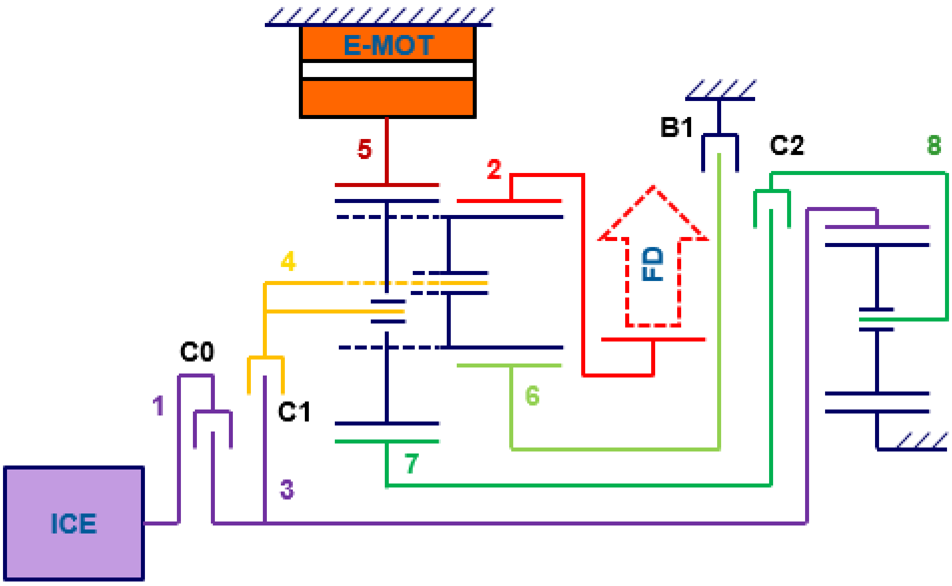


Figure 3-1 - Future Hybrid transmission layout (AVL, 2016)

There are two forward pure electric gears that both are underdrives, hence used in the start off. Moreover there are three different pure combustion engine gears and two gears where both prime movers are used. These are called eCVT (Electric continuously variable transmission) gears, since the electric motor is used to change the ratio for the combustion engine by speeding up or down the ring gear of the Ravigneaux gear set. Hence the combustion engine can be run at its most efficient speeds. Table 3-1 shows the seven available forward gears of the transmission and what clutches and brakes that have to be engaged.

One of the most complex shifts of this transmission is the E1 to G1, since there are two oncoming elements (C0 and B1) and one off-going (C1), compared to the normal for planetary transmissions (one oncoming element and one off-going). The challenge with this type of shift is to control all three elements and still perform a good powershift.

The electric motor gives the transmission a new degree of freedom in such a way that it allows an additional torque input during the shift procedure. It could be used in order to slow down or speed up the ring gear of the Ravigneuax gear set and thereby smoothen out the gear shift in aspects of torque output.

Table 3-1 - Future Hybrid gears

GEAR	C0	C1	C2	B1
E1		X	X	
$E2_a$			X	X
E-CVT 1	X		X	
E-CVT 2	X	X		
G1	X		X	X
G2	X	X	X	
G3	X	X		X

The Future hybrid transmission was modelled in Cruise but there were some difficulties experienced in getting it to run. The modules available in Cruise did not allow a simple way to model the complex transmission layout of the transmission. The model built allowed only some of the gears to run.

3.2 DCT Transmission

The second suggestion of transmission type was the DCT (dual clutch transmission). It is mechanically similar to a conventional manual transmission but the gears are shifted automatically. The gears are organized in two separate shafts; one with all odd gears (1^{st}, 3^{rd} etcetera) and one with all even gears (2^{nd}, 4^{th} etcetera). Between the prime mover and the transmission there are two clutches, one that connects the prime mover with the odd gear shaft and another that connects it to the even gear shaft. When performing a standard shift (for instance 1^{st} to 2^{nd}) the 2^{nd} gear can be synchronized to its shaft when 1^{st} gear is still in use. Then the torque handover can be performed automatically by the two clutches handing over to each other without any torque interruption. A schematic figure of a typical DCT is shown in Figure 3-2.

There are standard models and learning material available in Cruise of DCT models (AST, 2015). The biggest advantage of the DCT model is that it is a simple construction with only one off-going and one oncoming element. This allows an easy way to analyse the effect of different torque input strategies, which is the topic of this thesis.

The transmission model finally used in the simulation work can be seen in Figure 3-3.

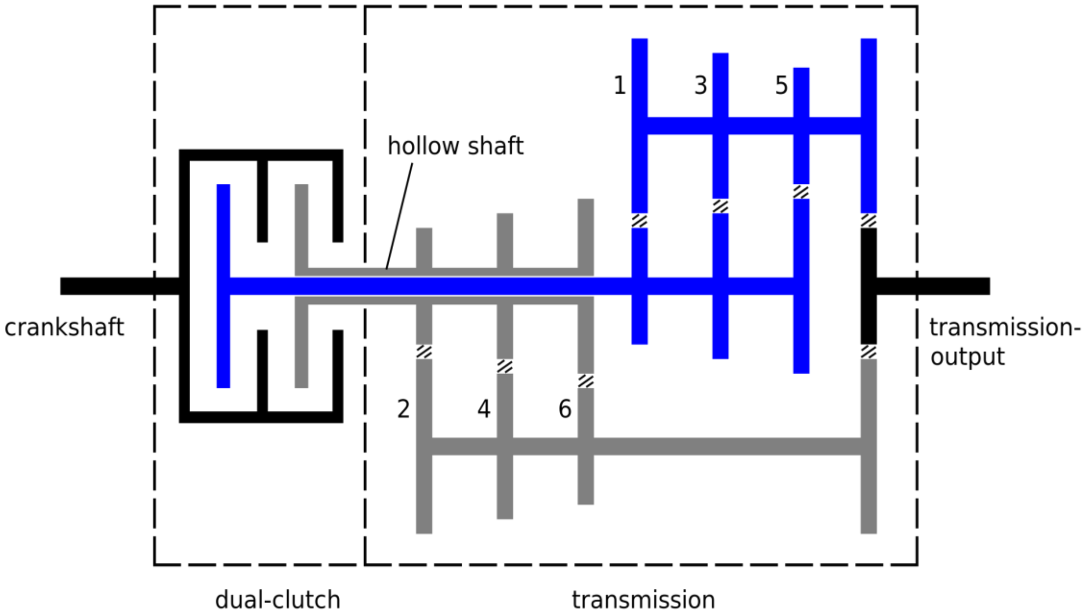


Figure 3-2 - Schematic figure of dual clutch transmission (Wikipedia, 2016)

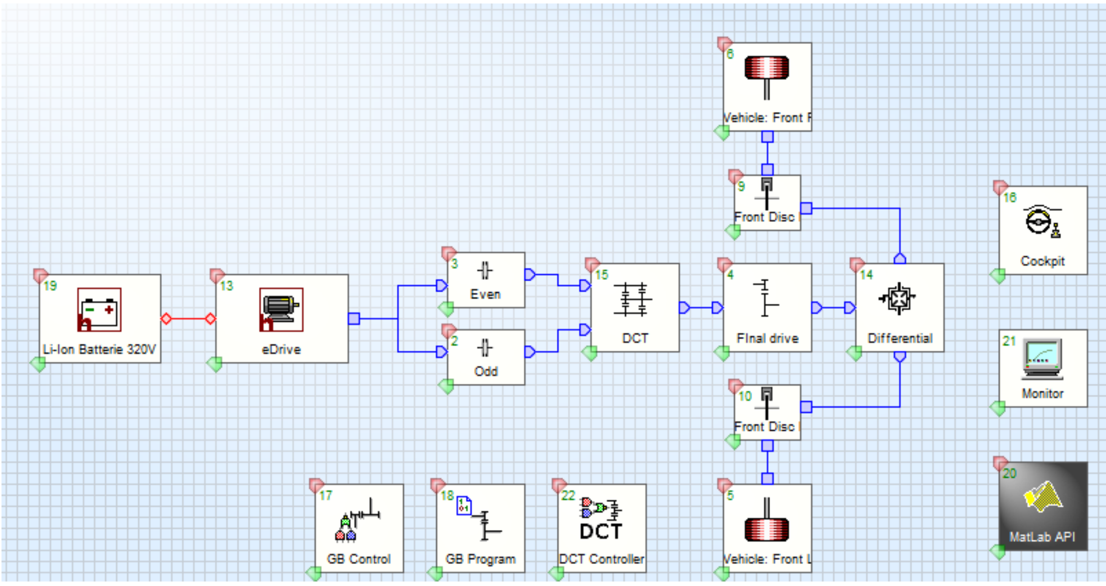


Figure 3-3 – Dual clutch transmission simulation model in Cruise

3.2.1.1 List of parameters

In Table 3-2 the values of the driveline parameters used in the simulations of the DCT transmission can be seen.

Table 3-2 - Driveline parameters

Module	Parameter	Value
Electric motor	Inertia	*0.000 1* kgm^2
	Nominal voltage	*320 V*
	Max torque	*240 Nm (up to 3 000 rpm)*
	Max speed	*10 000 rpm*
Battery	Nominal voltage	*320 V*
	Max voltage	*420 V*
	Min voltage	*220 V*
	Max charge	*10 Ah*
DCT transmission	Number of gears	*2*
	Ratio Gear 1	*4.33*
	Ratio Gear 2	*2.69*
Final drive	Ratio	*4.64*

3.3 Transmission choice

Due to the problems with modelling the Future hybrid transmission and the advantages of the DCT transmission model such as simple construction that directly allows controls strategy changes led to the decision of choosing the DCT as the transmission for the simulation work in this thesis.

4 Methods and materials

In this chapter the method of the work is described. In the start of the simulation work a lot of time had to be spent on getting the interface between the different software AVL Cruise and Matlab Simulink to run. Also a large amount of time has been spent in running simulation models and tuning controller parameters in between the runs.

4.1 Software Installation

AVL Cruise offers different ways to change parameters but in order to actively control certain modules using inputs coming from other modules it is convenient to use another software in an interface with Cruise. This can be done using API (Application programming interface). In Cruise there is an option to use Matlab Simulink in this interface. Every time step the Cruise model calls the Simulink model and gives output values and waits for Simulink to calculate and then receives the result in order to finish the calculations in Cruise. One has to create a Simulink model and then put in the file directory of the Cruise model.

To get Cruise to communicate with Matlab Simulink there are quite some settings that need to be altered. A guide explaining how to do this is to be found in Appendix A: Cruise to Matlab API configuration.

4.2 Cruise Model

The simulation model of the transmission needs to be built in Cruise using the modules available. With the modules it is possible to build a complete vehicle. The modules are for instance engine, transmission, final drive, brakes and wheels. All modules need basic inputs as inertias and dimensions and the more advanced ones, like transmission and engine, need quite some more input parameters such as shift programs, clutch torque curves and engine torque maps. The modules have a set of defined physical equations that govern the functioning of them.

There are some tutorials available in the installation package of Cruise (AST, 2015) that are recommended to go through before one starts with a completely new model.

4.3 Simulation

Cruise is a 0-D and a 1-D program, which compared to a 3-D program, is much faster. There is no need for three-dimensional computation since there are no three dimensional processes (like fluids) analysed. The system evaluated in this thesis can be described by equations that give a single result for every time step.

4.3.1 Profiles and courses

After building the vehicle model the simulation has to be defined regarding course and driver profiles. These components are not part of the block model but defining the environment for the vehicle during the simulation (AST, 2015). Under *Project data => Project*, a *task folder* has to be added. In this folder different types of standard runs *(computational tasks)* can be chosen, such as *Cycle run, Climbing performance* and *Cruising*. These are environments defined by Cruise and are used for testing different kind of situations that a vehicle would be put into in a real driving situation.

In the folder called *Course* it is possible to define environment parameters such as speed limit, altitude, wind velocity and ambient temperature.

Under *task folder* there is a folder called *profile*, defining the course profiles that the vehicle shall follow. For instance there are standardized driving cycles as *NEDC (New*

European driving cycle), UDC (Urban driving cycle) and IDC (Indian driving cycle). There are also a number of test cycles defined by Cruise.

Under *task components* the driver has to be defined as well. The driver should imitate the behaviour of a real driver.

In this thesis the *Cruising task* was chosen using a test way as vehicle course in order to make it possible to do an acceleration from stand still and perform a gear shift from first to second gear. The standard driver defined by Cruise was chosen.

4.3.2 Solver and time step

Cruise offers a number of different solvers. There are variable time step solvers, which are good in that sense that they are not very time consuming since they only use small time steps when the need for details is high, for instance during a shift process. However, since the time step is variable, the solver will not take exactly the same time steps when running the very same simulation a second time. This gave a problem since it was not possible to determine the consistency of the results.

There are also fixed time step solvers. The positive side of such solvers is that they give consistent results, but are also more time consuming than the variable time step solvers.

In this thesis the variable time step solver *Simulation 2* was used in the beginning of the work but since it was found that the results were not consistent it was decided to use a fixed time step solver instead. The solver chosen was *Simulation 4 (Bogacki-Shampine).* When using variable time step solver in Cruise the *ode45* was chosen in Matlab Simulink. When using the fixed time step solver in Cruise the Matlab Simulink solver chosen was *Discrete (no continuous states)* at first, but then changed to *ode3 (Bogacki-Shampine)* since it was the same solver method as the one used in Cruise.

4.3.3 Convergence

The simulation tool approximates curves by linearizing, which is, calculating derivatives in small time steps. The smaller the time steps, the more accurate the results. In order to confirm that the model is giving realistic values it has to converge. By other means, when making the time step smaller, the results should keep being the same.

To control that convergence was reached one model was run with four different time steps but apart from that all parameters were kept constant. The result was analysed in terms of output torque and can be seen in Figure 4-1. The first run was with $\mathrm{t} = 10\mathrm{ms}$ and then decreasing by $5\mathrm{ms}$ and again analysing the output torque. Different results with lower time step means that the simulation has still not converged. The time step has to be decreased until consistent results are reached.

Using a bigger time step than 5ms is not recommended for fixed time step solvers (AST, 2015).

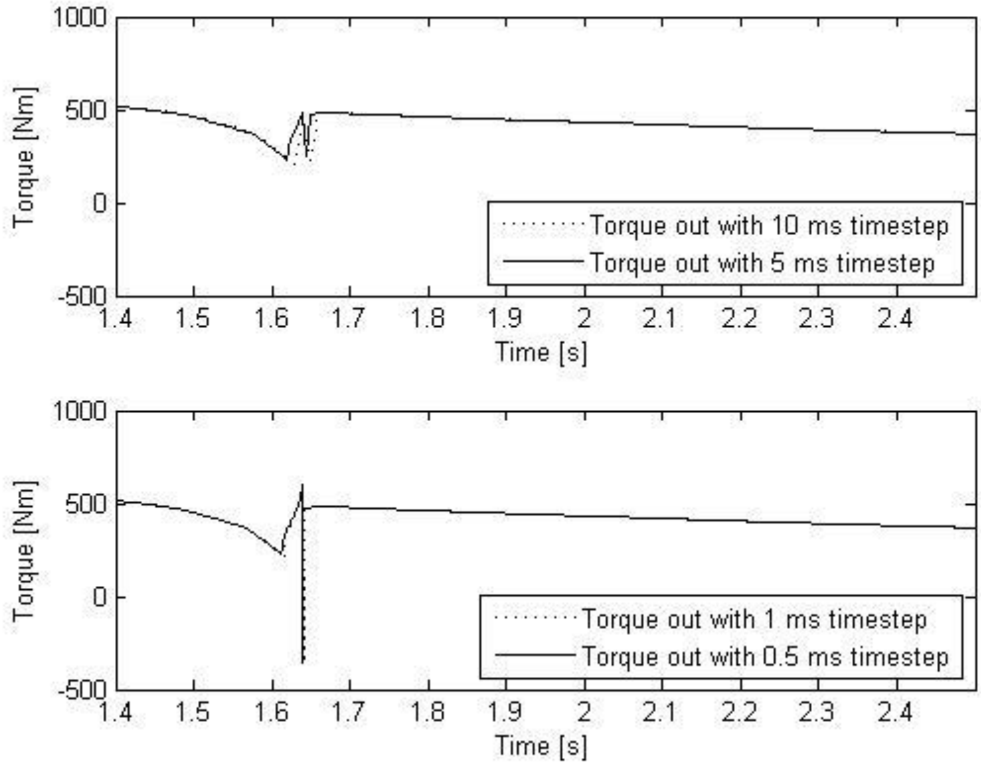


Figure 4-1 - Plot of output torque using different time steps in Cruise

One can see in the upper half of Figure 4-1 that the results for the output torque during the shift sequence for t = 10 ms and t = 5 ms are differing slightly. The difference is even bigger between t = 5 ms and t = 1 ms. The next step, comparing the two curves in the lower half of Figure 4-1 one can see that torque curves are almost identical. This means that the simulation has converged. It was decided to go with t = 1ms for the rest of the simulation work. This was applied both in Cruise and Matlab Simulink.

4.4 Controller

The torque control from the electric motor was achieved by building a closed loop negative feedback control system with a PI controller. The controller was built using Matlab Simulink 2010a.

In the simulation model, the Cruise model forms the plant in the control loop but the Simulink model runs as a slave. The inputs to the controller are the electric motor torque, transmission ratio, cockpit load signal, DCT output torque and real-time. All these values are input to the controller from Cruise in discrete time steps. The product of the electric motor torque and transmission ratio was fed as the set point value and this was compared with the corresponding DCT output torque to compute the error. The controller then tries to get this error to zero by giving a corresponding output to the system. The controller output was then added to the cockpit load signal and was sent out to the Cruise model as the modified load signal. It was done so as the only way to control the output torque of the electric motor in Cruise was by altering the load signal.

Initially a Matlab script was used substituting the Simulink controller but the output did not yield satisfactory results and thus the PI controller was adopted.

The Matlab Simulink model can be seen in Figure 4-2.

Referring to (Fischer, 2012), from the theory it was observed that during the inertia phase, when the electric motor speed decreases, there is a torque peak occurring. Thus a new control strategy (Inertia phase strategy) was implemented to counteract this torque rise during the inertia phase as the previous strategy employed did not account for this torque rise. In addition to the input parameters mentioned above, the current gear, electric motor speed and desired gear signals were sent into the model. In this strategy the controller works in the same way as before during the torque phase and in the inertia phase the cockpit load signal is reduced by a value between 48.5% to 50% for default and high load signal respectively and sent as the output.

The orange block in Figure 4-2 computes the error between the desired torque value and the actual torque value. The blue subsystem in the same figure has the algorithm to detect the speed change of the electric motor during the gearshift process and in the green subsystem the cockpit load signal is modified and this is fed as the output into Cruise.

A second method, in an attempt to generalize the inertia phase strategy was worked out. In this strategy, there is an assumption made that the power transferred during the speed synchronisation phase is constant and this is divided by the electric motor speed. Thus a reducing speed leads to an increase in torque and this is the target value of torque to be maintained. The difference between the actual torque output and the target value is calculated (correction load signal) and this is reduced from the cockpit load signal and sent out of the controller.

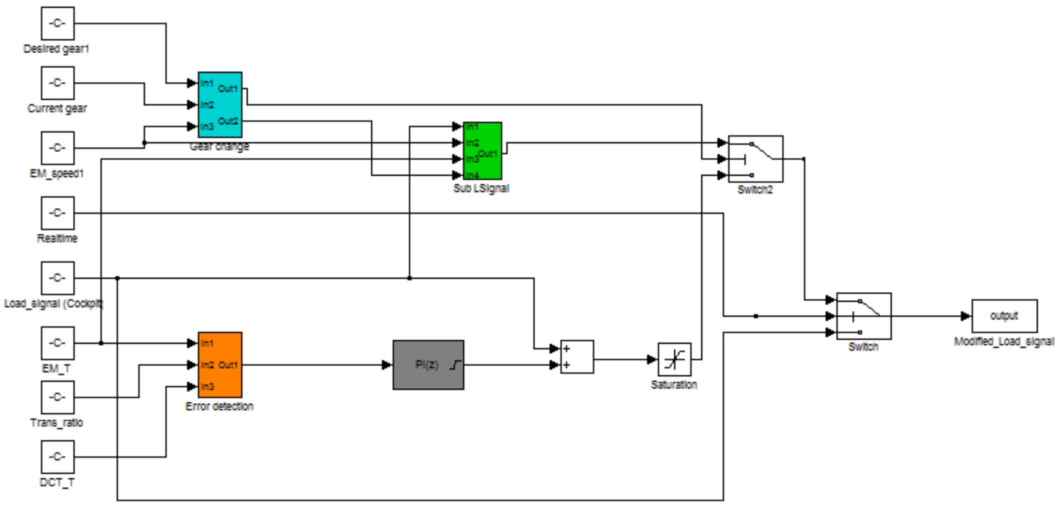


Figure 4-2 - Matlab Simulink model with PI controller

When the motor speed decreases during the speed synchronization, the torque value has to rise accordingly (4-1) assuming constant power. This torque then forms the reference value and is compared with the output torque value and the load signal is modified.

$$P = T * \omega \qquad \text{4-1}$$

4.4.1 Controller Tuning (PI)

The controller was tuned using the *PID tuner* application in Matlab 2014a (Mathworks, 2016). The tuning was done by first identifying the plant and for this a known step input in terms of load signal was given to the Cruise model. The input and the output data for this were exported into the tuner application and thus the plant was identified. Then the response of the system was tuned according to the requirement and the corresponding P and I gain values were obtained.

The model was run for two load cases: default and high load signal. The model was first tuned for the default load signal application and some further manual tuning was required to get the desired torque peak values. The results can be seen in Section 5.1.2. Then the high load signal case was run with the same PI parameters and the obtained results were satisfactory as seen in Section 5.2.2.

There were attempts to tune the controller to the high load signal case to see if the new gain values obtained yielded any better results but the gain values for the default load signal were more effective in controlling the torque peaks. Since the magnitude of the error during each of the load signal cases varies but in the same time interval, a controller tuned for default load signal values might suffice to the present conditions but when the error is high in the other case the gain values might not be optimum for the controller and thus this method was tried.

For the inertia phase strategy the controller was first iterated for its performance for different percentage values of reduction of the cockpit load signal. Initially a common value was being used for both the load signal cases but on subsequent iterations it was found that having separate percentage reduction values for each of the cases deemed to be more effective in controlling the torque peaks. One more possibility is to give an adequate constant value of load signal as the output during the inertia phase instead of altering the cockpit load signal as this becomes independent of the torque request. Due to limitation in the time frame additional investigation on this could not be carried out. This strategy was effective in reducing the build-up of the torques in comparison to the previous strategy.

The initial outcome of the second inertia phase strategy was not satisfactory and so different iterations were performed to further reduce the cockpit load signal until satisfactory results were obtained. This strategy gave acceptable torque peaks for both default and high load signal cases when the correction load signal was increased by 40% of its initial value. The reason for this might be that during the speed synchronisation the dynamic torques that occur were of very high magnitude and to control their amplitudes the output torque of the motor has to be further reduced.

4.5 Validation

The DCT model could be evaluated by comparing it with standard example models but that is not a fair comparison since they have different prime movers. A more adequate comparison is to study the very same model but without controller. Hence models were run without any controller involved and then compared to the very same models but with controllers. The effect of the controllers can be seen directly in the results (Chapter 5).

4.5.1 Parameters for shift quality and acceptance levels

A way to determine shift quality is to measure the vehicle acceleration during the shift procedure. To achieve appropriate results it should be measured in the driver seat since it will express the perception of the vehicle acceleration by the driver.

In this thesis the vehicle model has not been analysed, hence the decision was taken to focus on the acceleration of the output shaft of the transmission, since this is the source creating the acceleration perceived in the driver seat.

Vehicle acceleration:

$$a = \frac{dv}{dt} \tag{4-2}$$

Rotational speed and acceleration of the driven wheels:

$$\omega = \frac{v}{r} \tag{4-3}$$

$$\dot{\omega} = \frac{d\omega}{dt} \tag{4-4}$$

Torque of, for instance, the output shaft of the transmission:

$$T = J \times \dot{\omega} \tag{4-5}$$

Hence, considering the inertia of the rotating parts in the system, the output torque of the transmission can be used as a measurement parameter for determining the shift quality of a given transmission. The results are expressed in terms of output torque of the transmission and then analysed in four parameters:

1. **Amplitude**
2. **Gradient of first positive peak**
3. **(Number of peaks)**
4. **(Frequency of peaks)**

The results are presented in Chapter 5 and more specifically in Table 5-2. The acceptance levels will be different depending on the aggressiveness of the driver. Aggressive driving permits higher acceptance levels.

5 Results

In the following chapter the results from the simulation work is shown.

5.1 Default load signal

This model uses the default load signal from the Cruising task presented in Section 4.3.1. The speed limit used to create the default load signal can be seen in Figure 5-1.

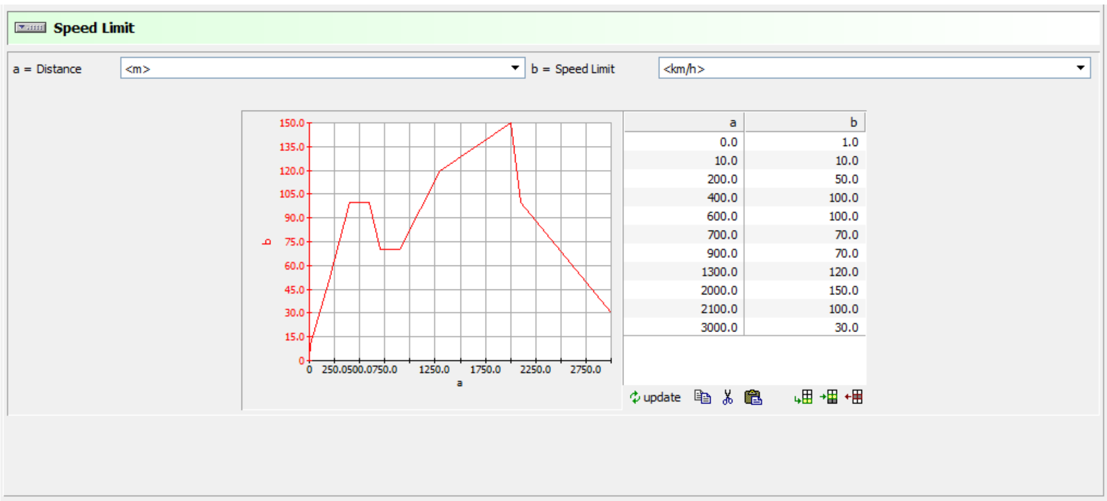


Figure 5-1 - Speed limit for default load signal

5.1.1 Without controller

This model is the standard transmission in Cruise but without using the controller. This means that there is no control of the electric motor in order to reduce the torque peaks. The results are seen in Figure 5-2, Figure 5-3 and Table 5-2.

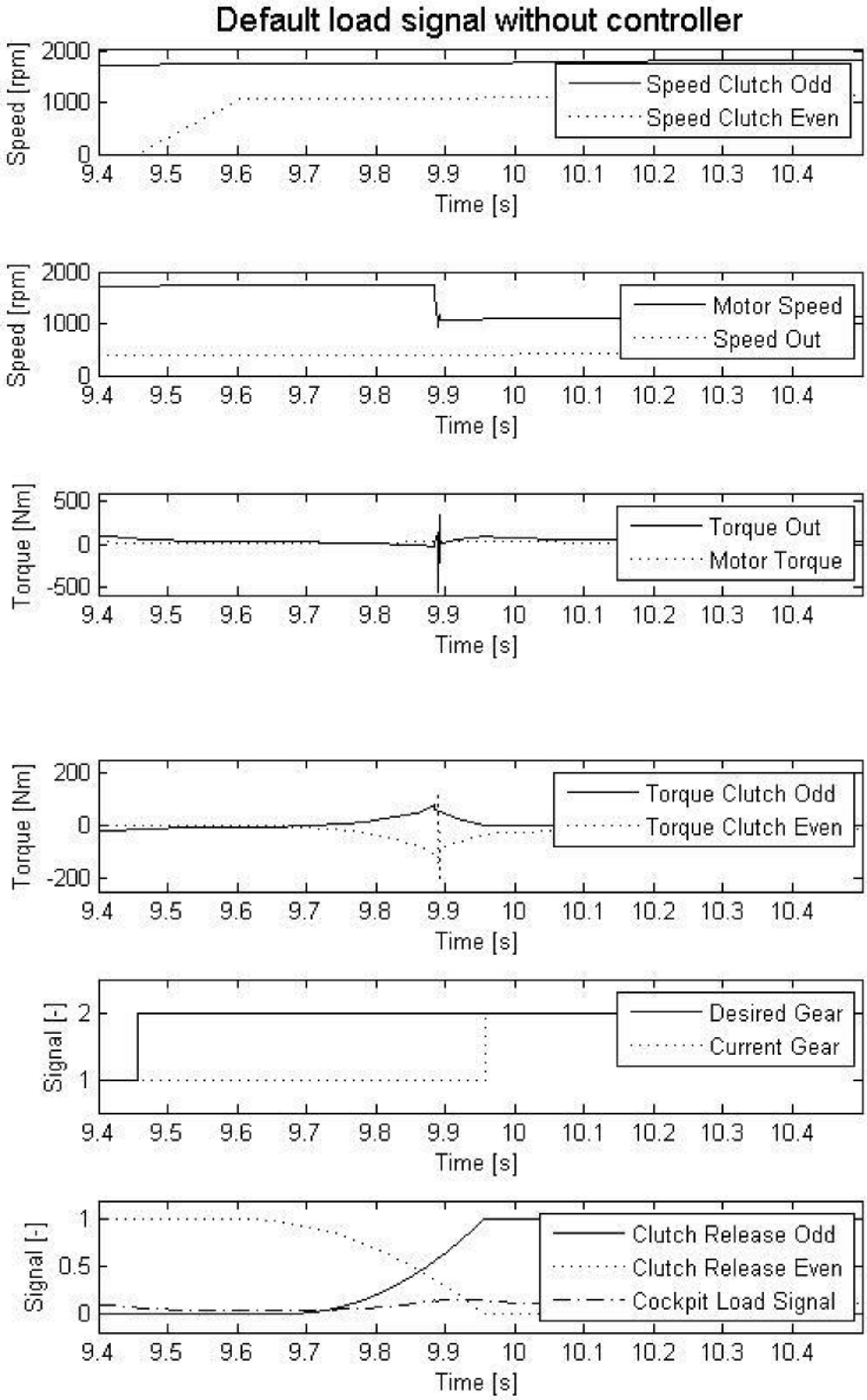


Figure 5-2 - Default load signal without controller

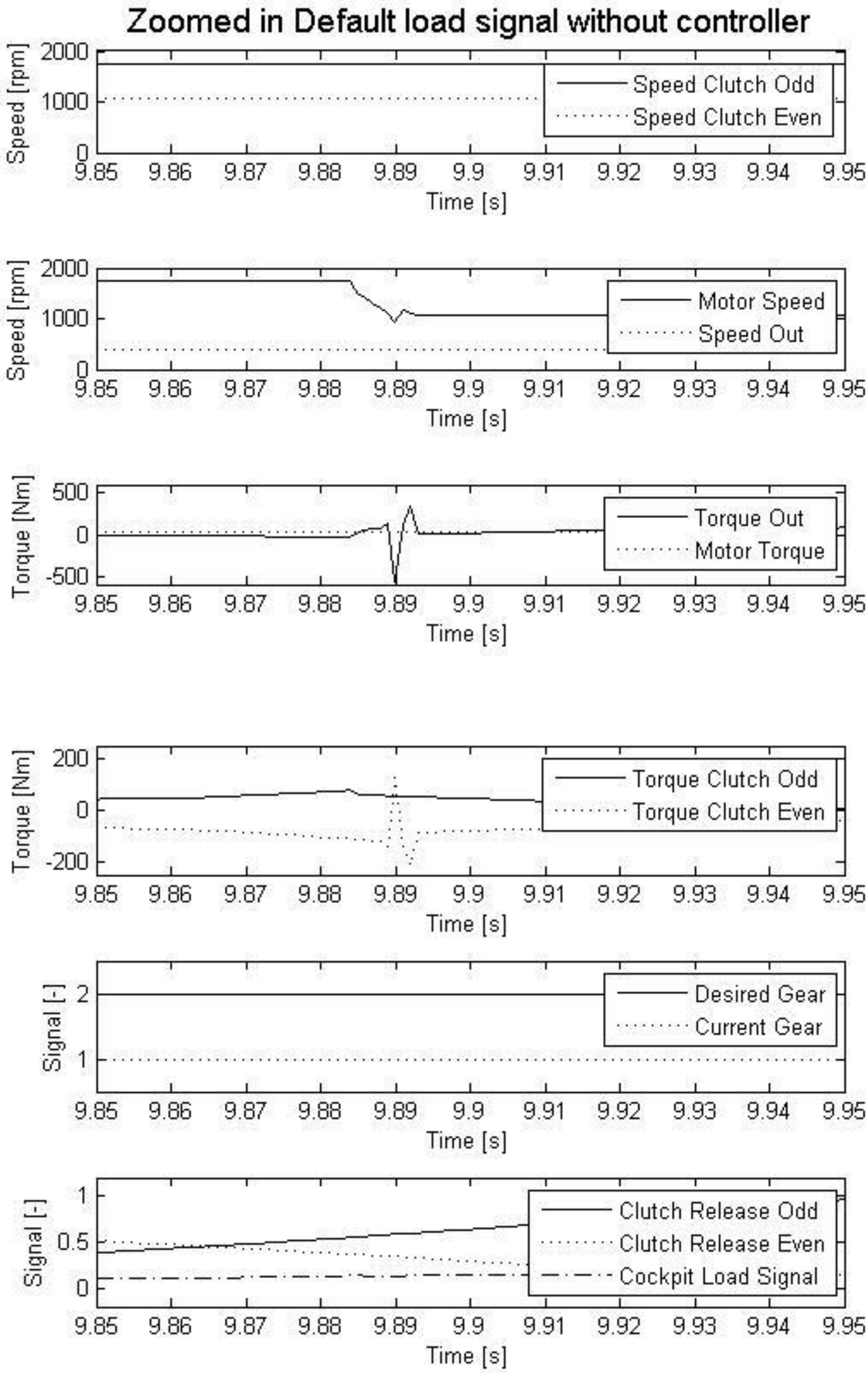


Figure 5-3 - Zoomed in default load signal without controller

5.1.2 With controller

This version contains the PI controller tuned using the Simulink *PID tuner*. The results can be seen in Figure 5-4, Figure 5-5 and Table 5-2.

Table 5-1 shows the tuned controller parameters for default load signal. The method of obtaining these parameters is presented in Section 4.4.1.

Table 5-1 - Low load signal controller parameters

P [-]	**I [-]**
0.000 204 81	0.000 409 62

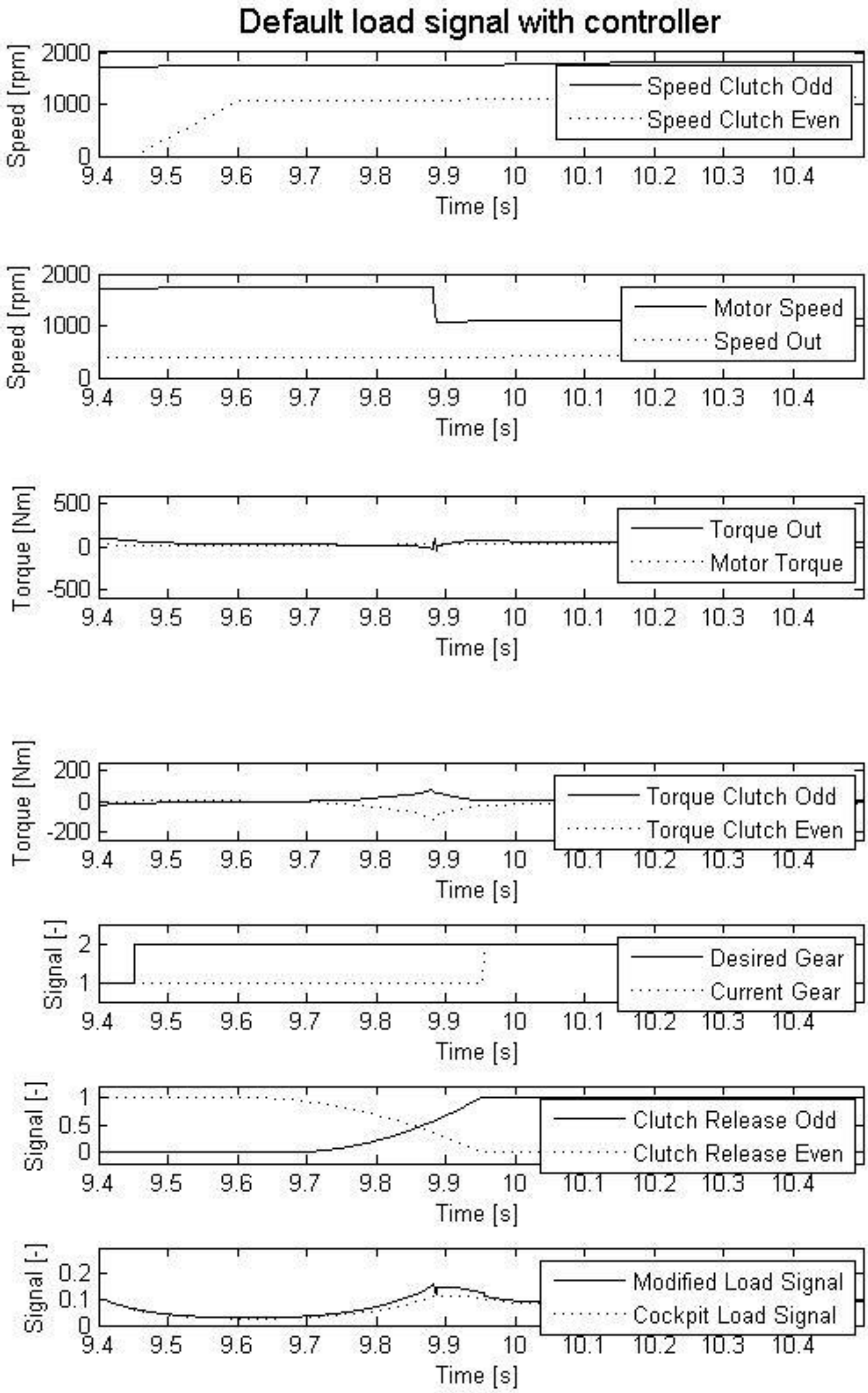


Figure 5-4 - Default load signal with controller

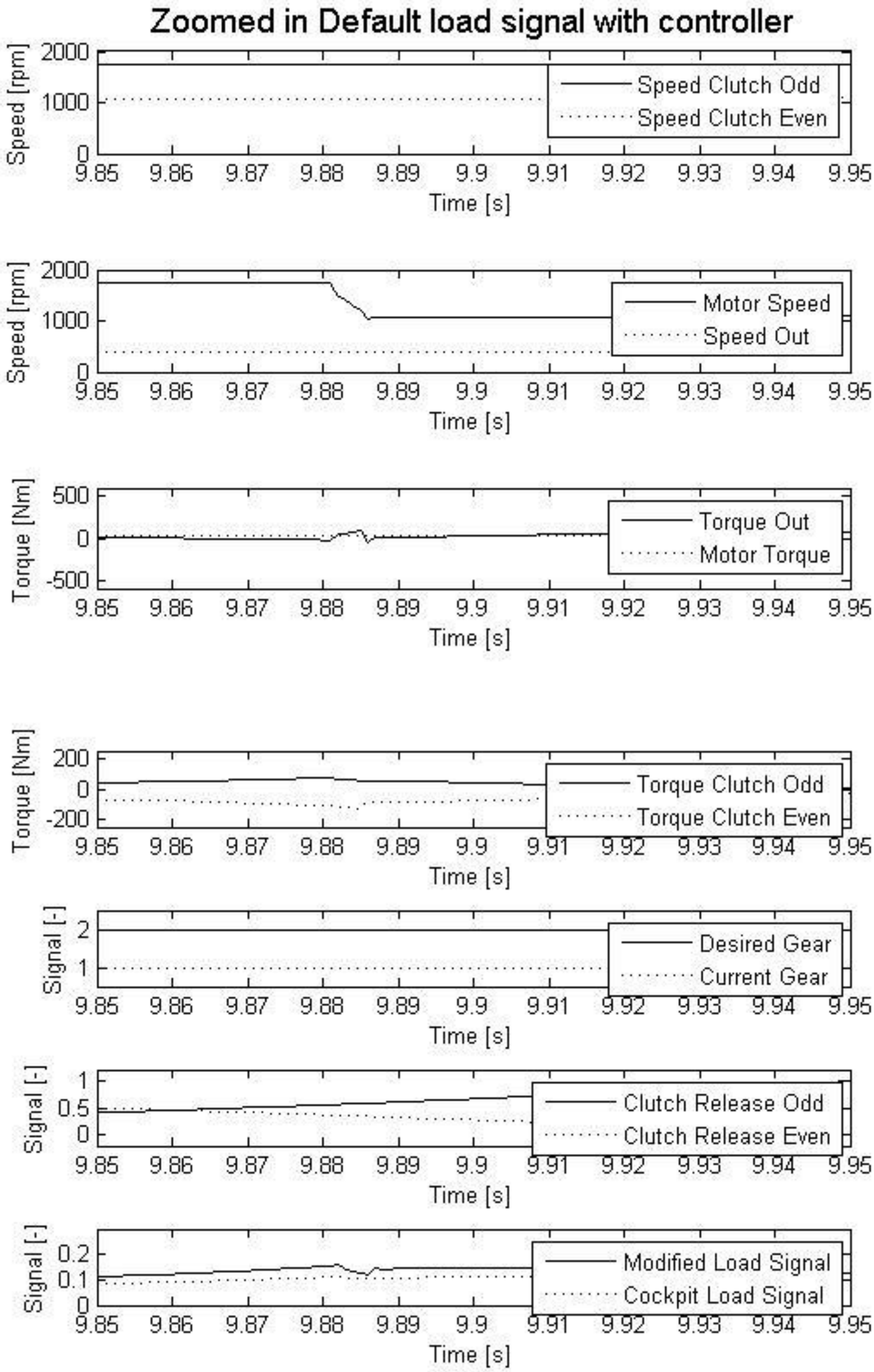


Figure 5-5 - Zoomed in default load signal with controller

5.1.3 Torque comparison

The results in Figure 5-6 show the comparison of the output torque from Figure 5-2 - Default load signal without controller and Figure 5-4 - Default load signal with controller.

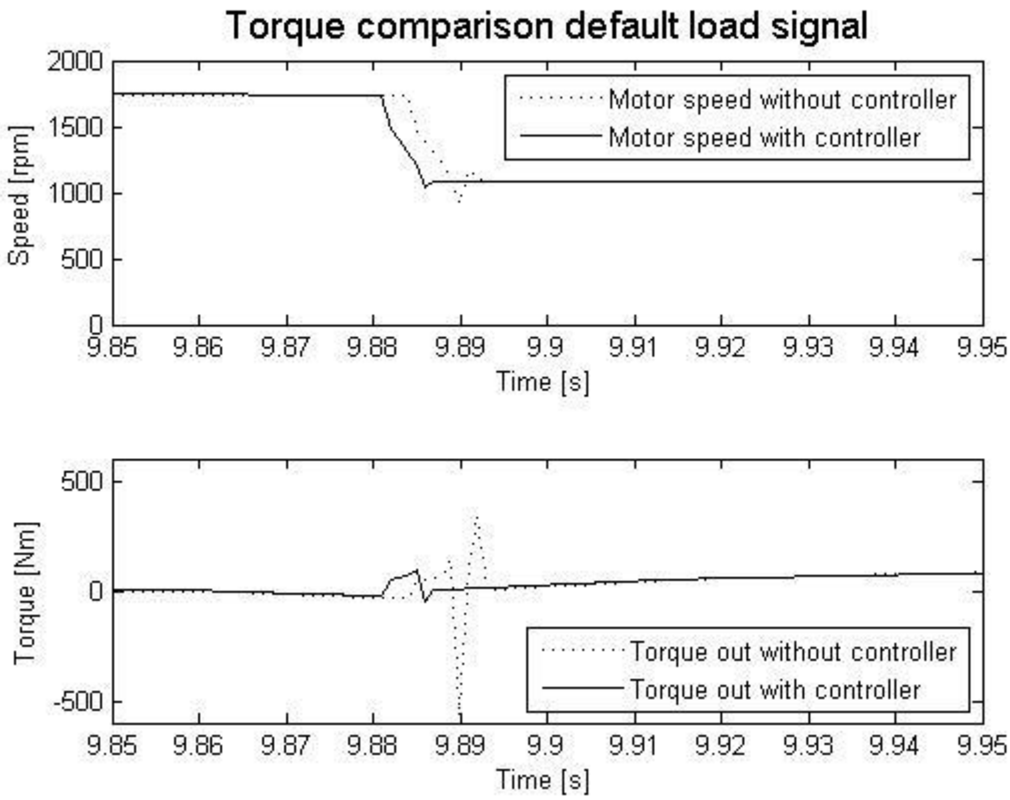
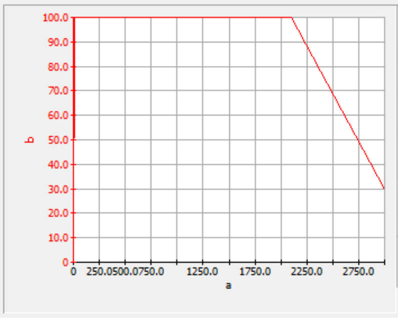


Figure 5-6 - Torque comparison default load signal

5.2 High load signal

In order to analyse how the controller works under different conditions another load signal was used, by changing the speed limit for the driver in Cruise. The new speed profile can be seen in Figure 5-7.



a	b
0.0	1.0
10.0	100.0
200.0	100.0
400.0	100.0
600.0	100.0
700.0	100.0
900.0	100.0
1300.0	100.0
2000.0	100.0
2100.0	100.0
3000.0	30.0

Figure 5-7 - Speed limit for high load signal

5.2.1 Without controller

The results using the high load signal, presented in Figure 5-7, without controller can be seen in Figure 5-8, Figure 5-9 and Table 5-2.

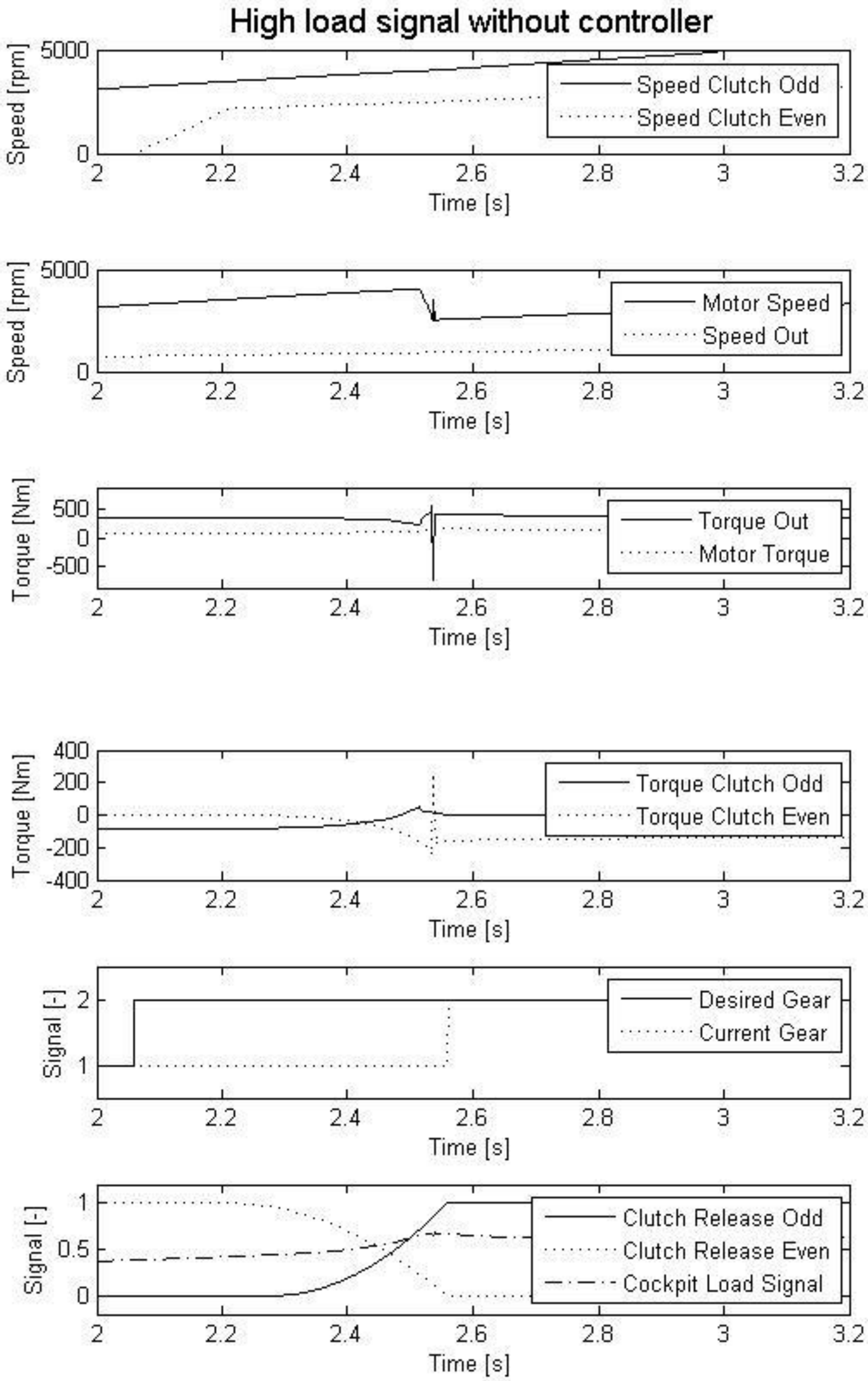


Figure 5-8 - High load signal without controller

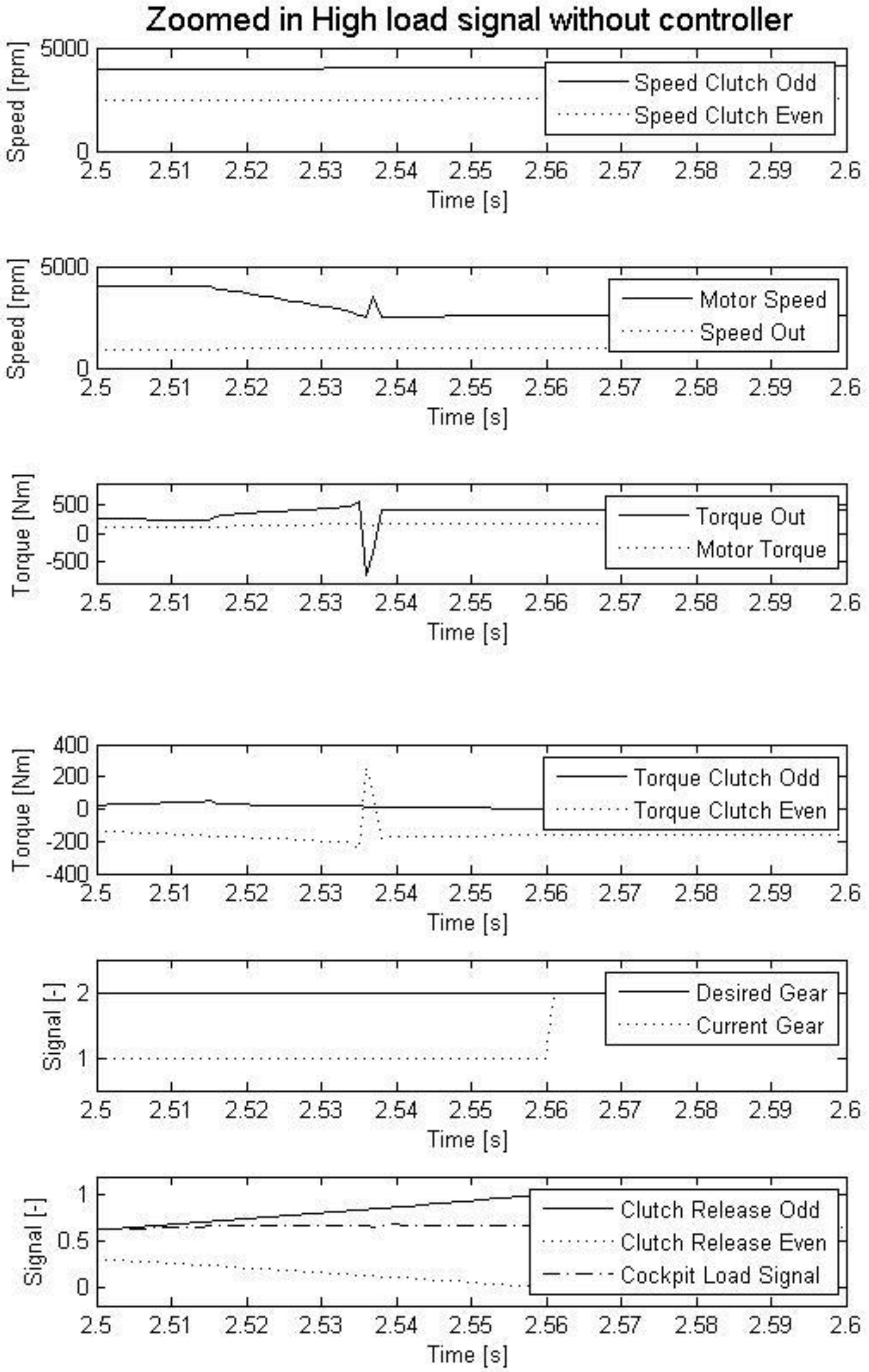


Figure 5-9 - Zoomed in high load signal without controller

5.2.2 With controller

The results for the high load signal model with controller are presented in this section.

Simulations showed that the output torque peaks from the transmission when running high load signal with the same controller parameters as in default load signal were higher than with the low load signal but still acceptable.

The results can be seen in Figure 5-10, Figure 5-11 and Table 5-2.

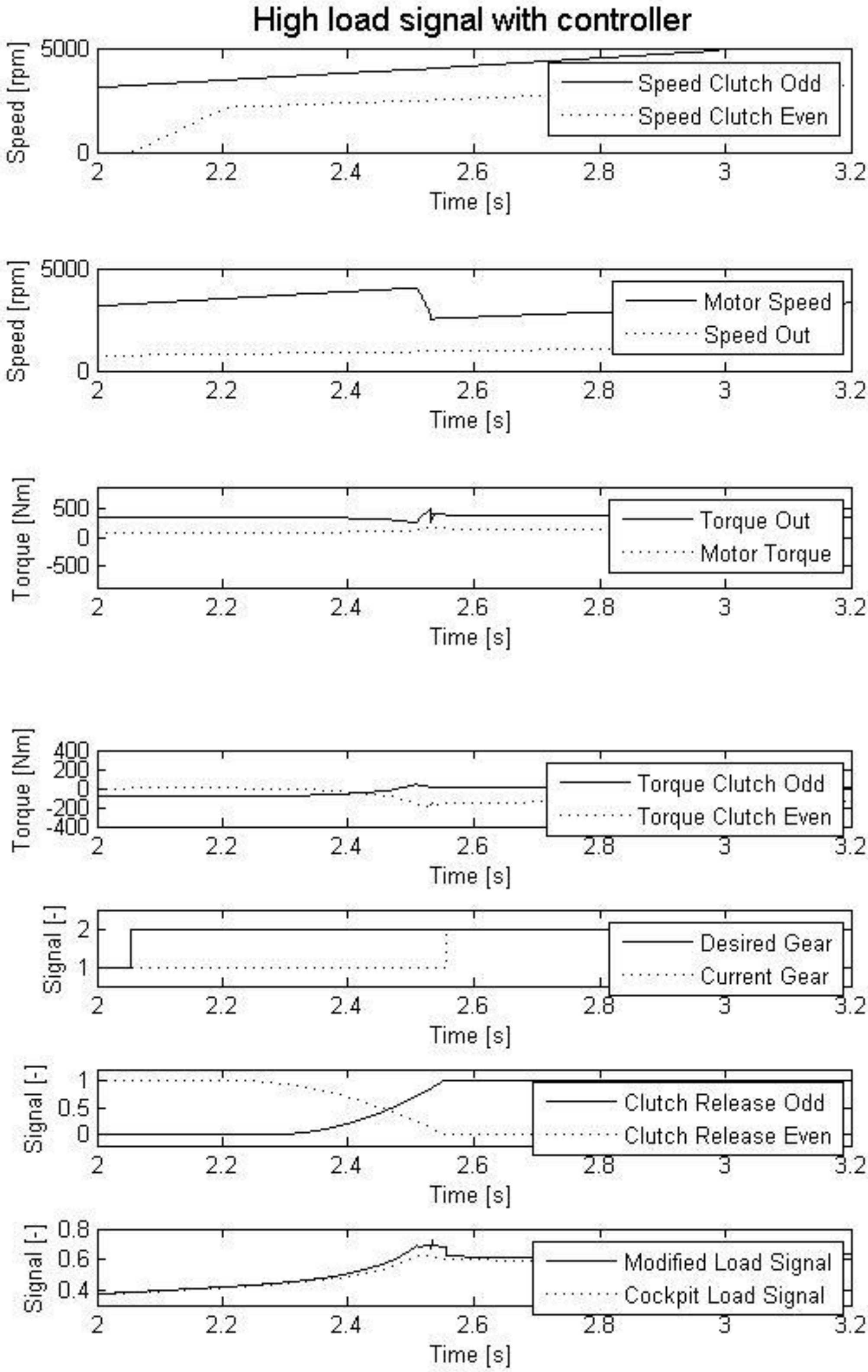


Figure 5-10 - High load signal with controller

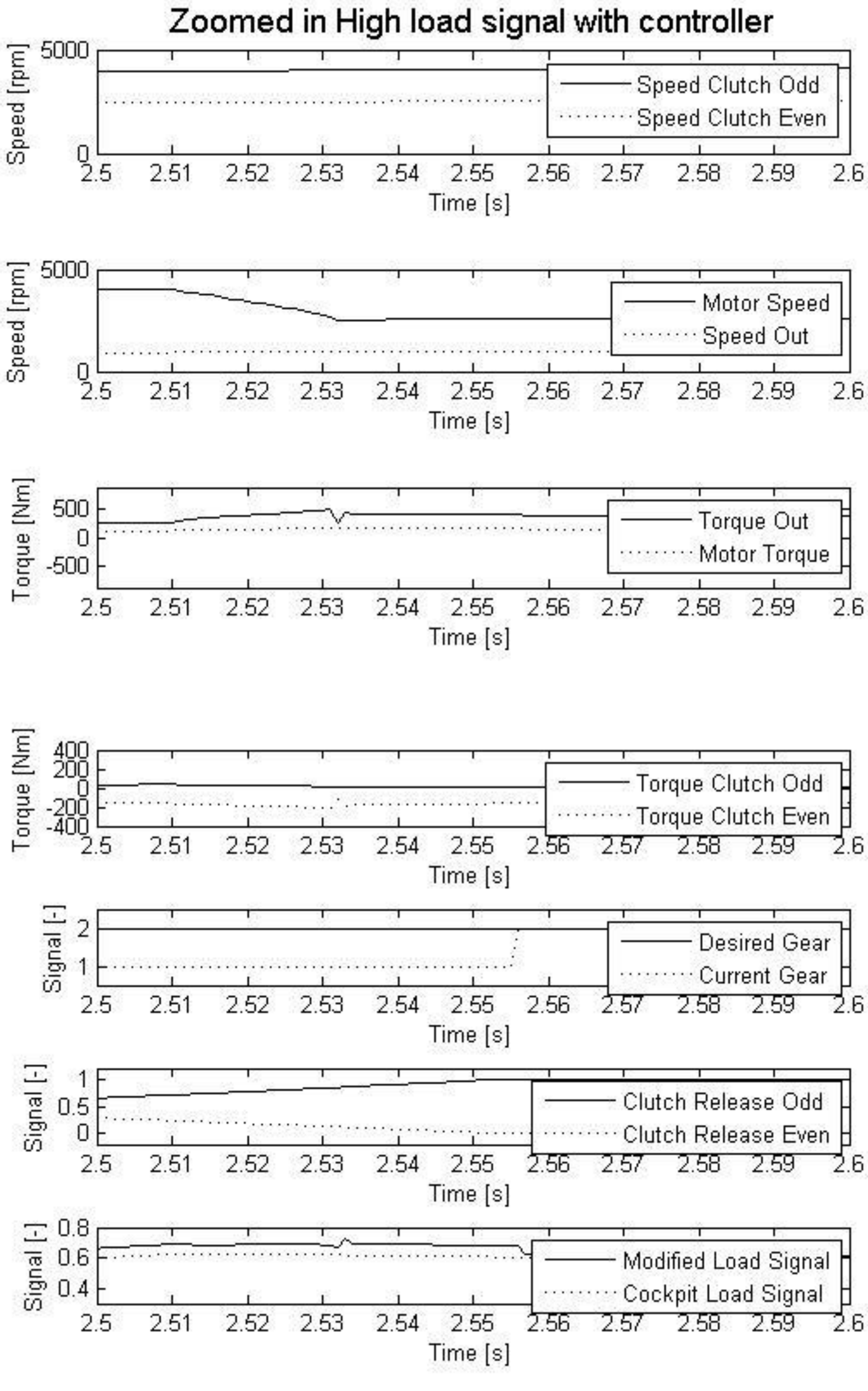


Figure 5-11 - Zoomed in high load signal with controller

5.2.3 Torque comparison

The results in Figure 5-12 show the comparison of the output torque from Figure 5-8 - High load signal without controller and Figure 5-10 - High load signal with controller.

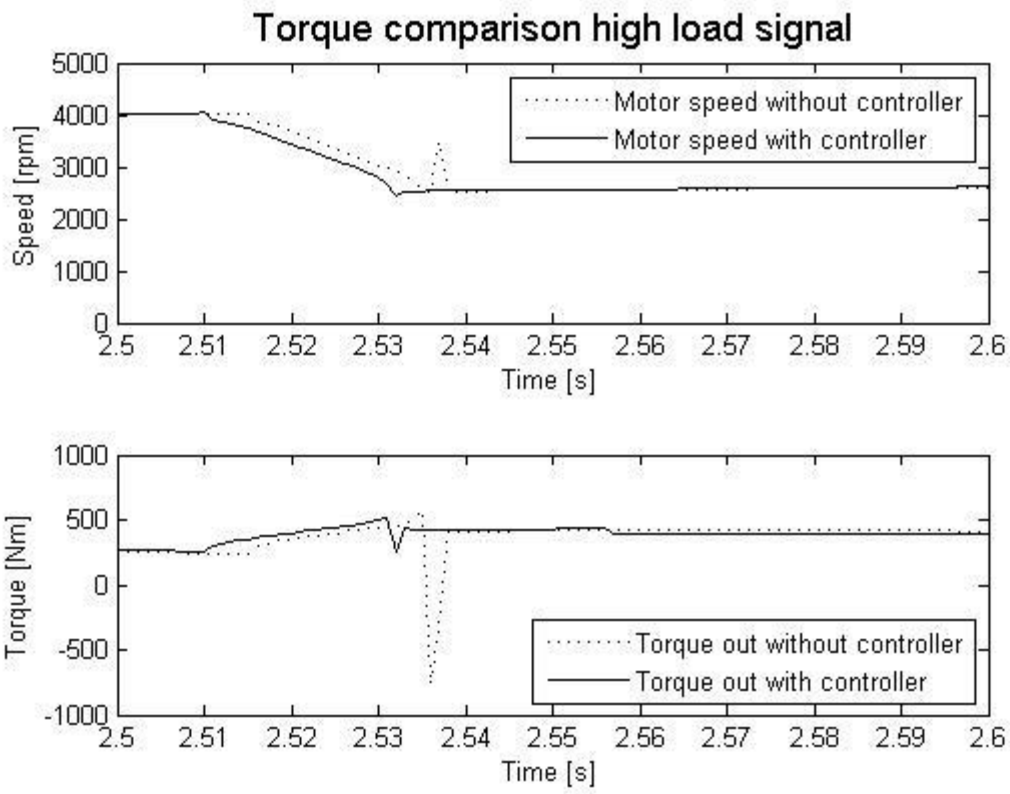


Figure 5-12 - Torque comparison high load signal

5.3 First inertia phase strategy

In order to find out if the torque peaks during the inertia phase could be decreased even more, a strategy of lowering the modified load signal more in the inertia phase than in the rest of the shift sequence, was implemented.

5.3.1 Default load signal

Figure 5-13 presents the comparison of default load signal with controller (Figure 5-4) and the inertia phase strategy of the very same load signal.

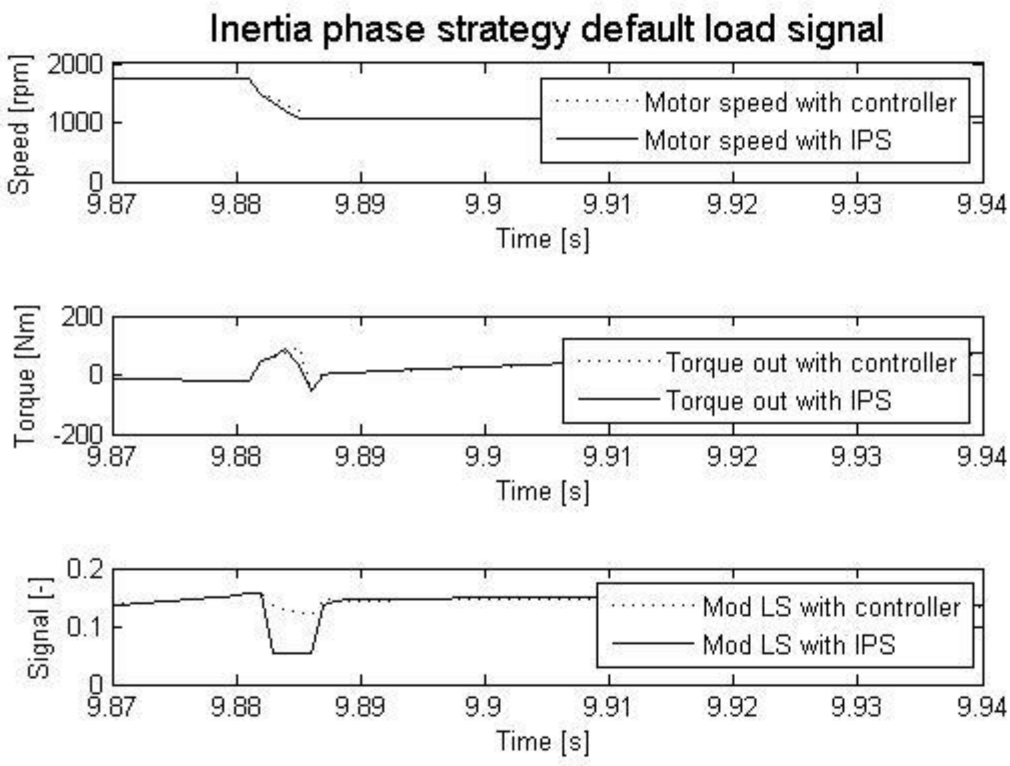


Figure 5-13 - Inertia phase strategy default load signal

5.3.2 High load signal

Figure 5-14 presents the comparison of high load signal with controller (Figure 5-10) and the inertia phase strategy of the very same load signal.

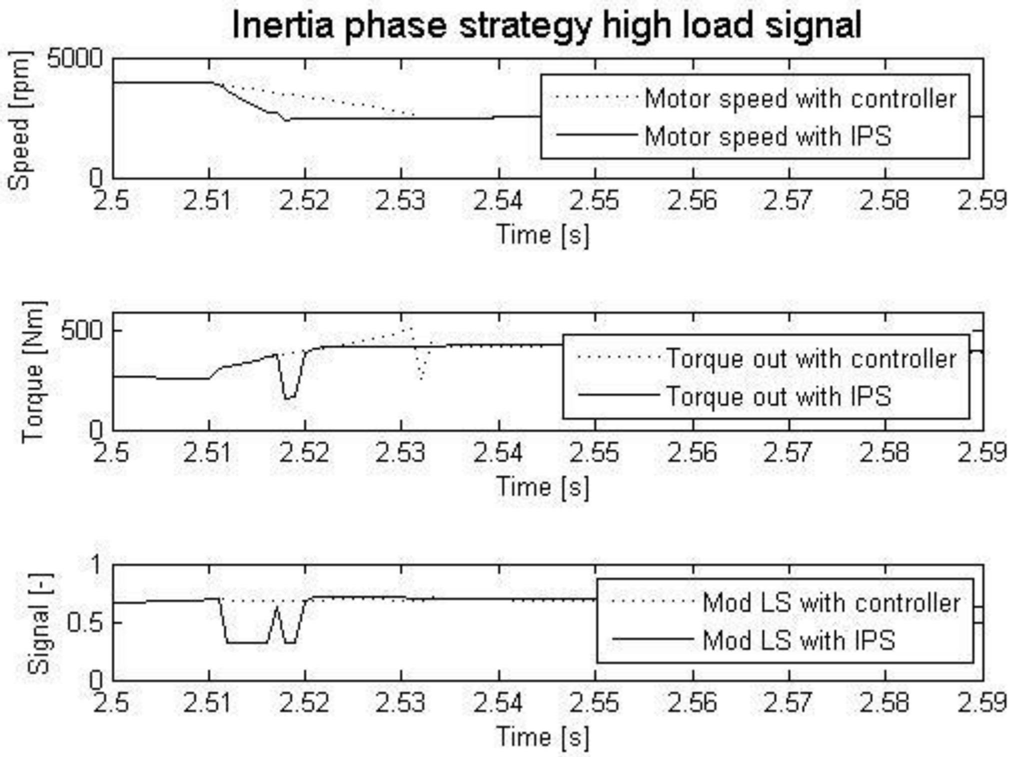


Figure 5-14 - Inertia phase strategy high load signal

5.4 Second inertia phase strategy

The results of the second inertia phase strategy are presented in this section.

5.4.1 Default load signal

The results from the second inertia phase strategy for default load signal can be seen in Figure 5-15.

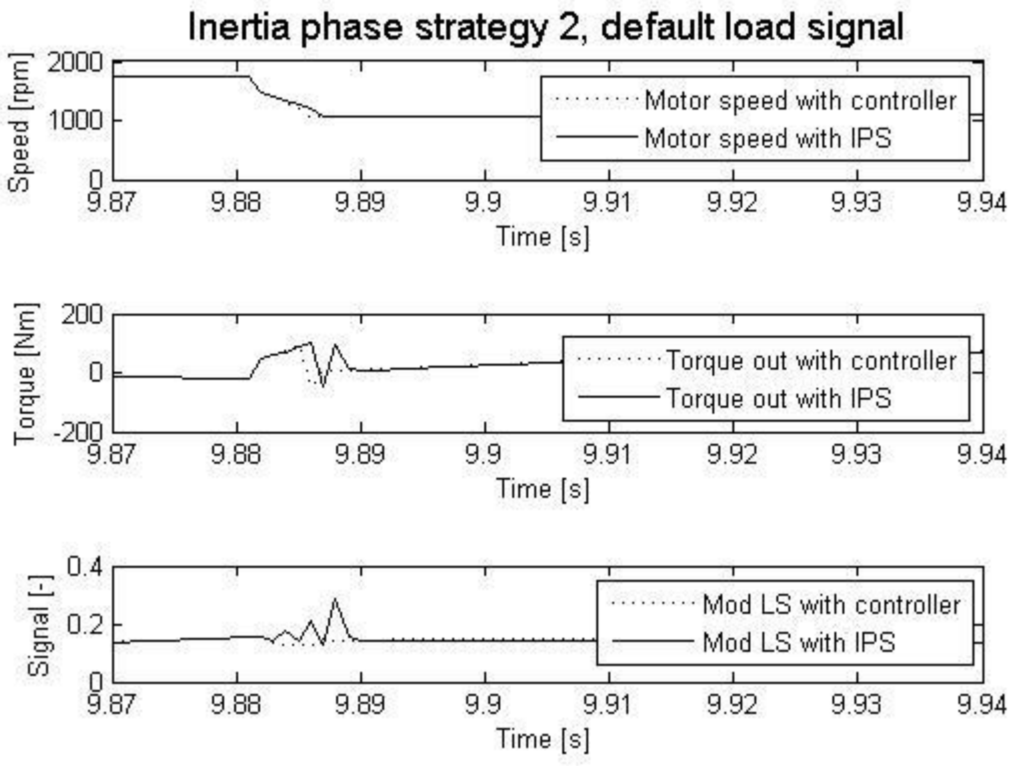


Figure 5-15 - Inertia phase strategy 2, default load signal

5.4.2 High load signal

The results using the second inertia phase strategy with high load signal can be studied in Figure 5-16.

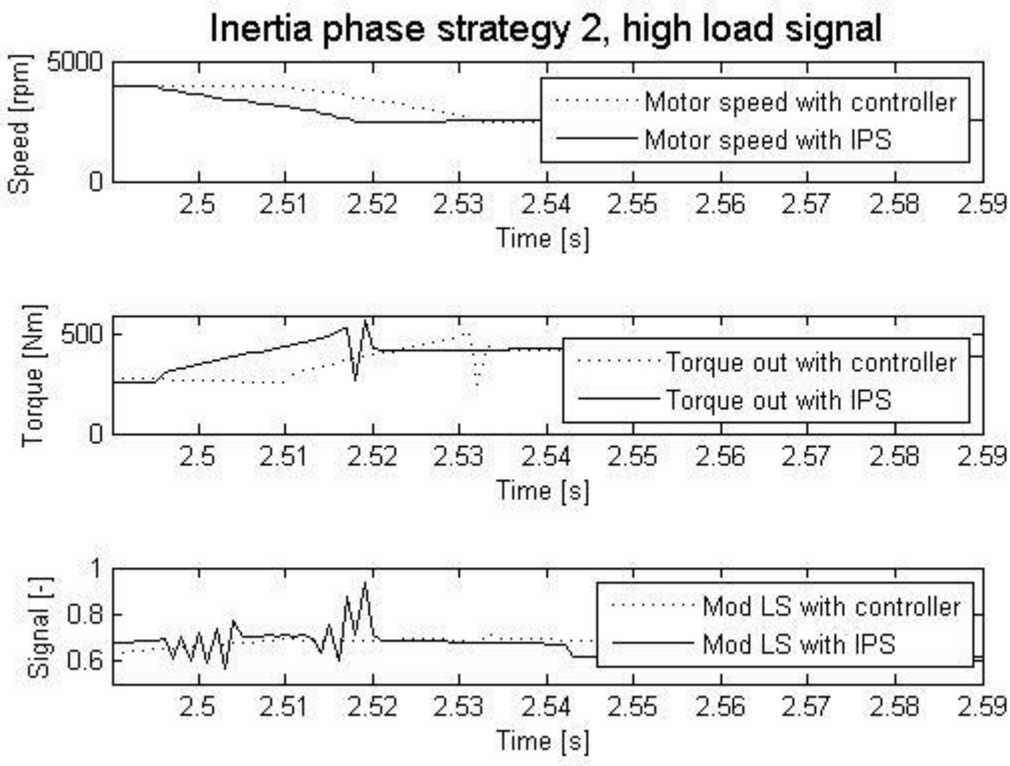


Figure 5-16 - Inertia phase strategy 2, high load signal

5.5 Clutch control strategy

Of the different methods of controlling the transmission that were presented in Section 2.1 only one was thoroughly experimented in this thesis, namely the torque control strategy. However, at an early stage of the thesis, some simple simulations were run testing clutch control strategy. The simulations were run with the standard DCT model (with combustion engine) and the results showed no improvement in changing from the standard clutch timing applied in Cruise.

5.6 Results table

Table 5-2 shows a summary of the results of shift quality parameters achieved for the different controllers applied in this thesis. Table 5-3 shows the results in percentage.

Table 5-4 explains the abbreviations used in Table 5-2.

Table 5-2 - Shift quality parameter results for the different controllers

Strategy	T_{max} [Nm]	T_{min} [Nm]	T_{amp} [Nm]	Grad [Nm/ms]
DL no controller	341.62	-570.40	912.02	35.98
DL with controller	99.38	-50.63	150.01	28.91
HL no controller	572.17	-762.24	1334.41	16.84
HL with controller	522.75	250.65	272.10	12.64
DL Inertia phase strategy	88.43	-55.93	144.36	36.96
HL Inertia phase strategy	380.30	158.80	221.50	17.58
DL Inertia phase strategy2	103.41	-49.84	153.25	25.17
HL Inertia phase strategy2	573.00	256.49	316.51	12.55

Table 5-3 - Percentage reduction of torque amplitude and gradient

Strategy	T_{amp}	Grad
DL with PI controller compared to without controller	84%	20%
HL with PI controller compared to without controller	80%	25%
DL Inertia phase strategy compared to PI controller	4%	-28%
HL Inertia phase strategy compared to PI controller	19%	-39%
DL Inertia phase strategy2 compared to PI controller	-2%	13%
HL Inertia phase strategy2 compared to PI controller	-16%	1%

Table 5-4 - Abbreviation list for Table 5-2

Abbreviation	Explanation
DL no controller	Default load signal without controller
DL with controller	Default load signal with controller
HL no controller	High load signal without controller
HL with controller	High load signal with controller
DL Inertia phase strategy	Default load signal with controller using inertia phase strategy
HL Inertia phase strategy	High load signal with controller using inertia phase strategy

Figure 5-17 shows how the shift quality parameters are defined. The torque amplitude is calculated according to 5-1.

$$T_{amp} = T_{max} - T_{min} \tag{5-1}$$

The gradient of the first torque peak is derived according to 5-2.

$$Grad = \frac{d_y}{d_x} \tag{5-2}$$

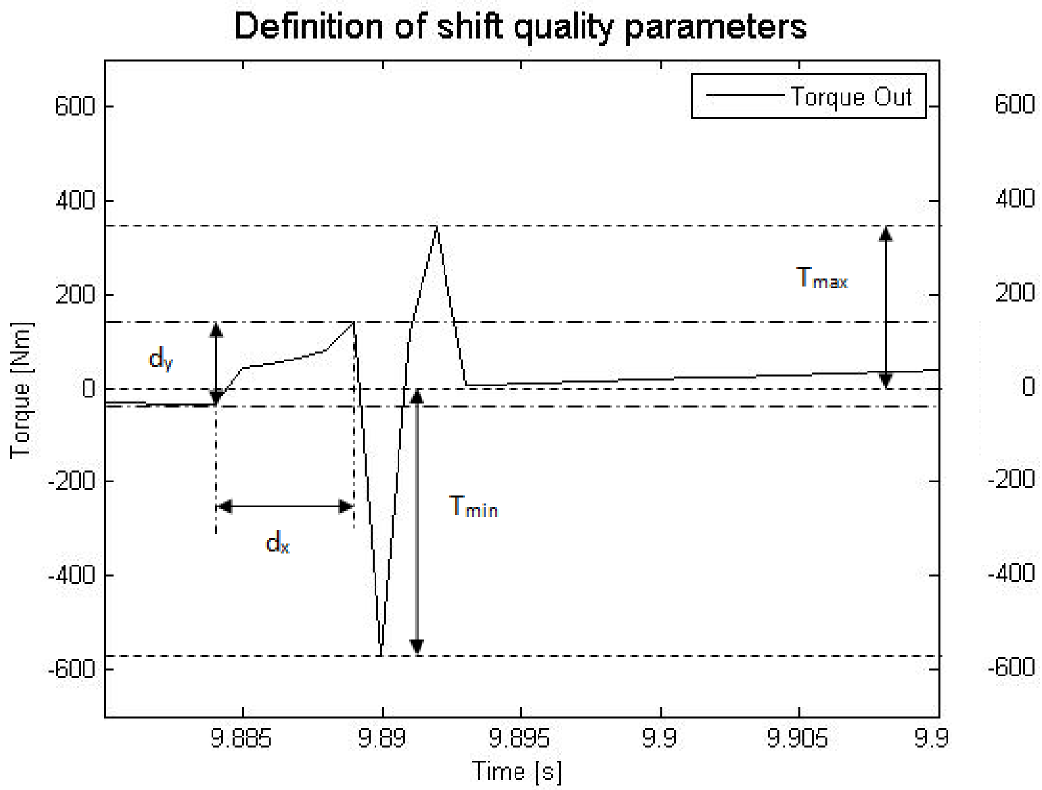


Figure 5-17 - Definition of shift quality parameters

6 Discussion

In this chapter the results presented in Chapter 5 are analysed and discussed. The theory from Chapter 2 Literature review is used for comparison in order to see if the results of this thesis aligns with earlier studies done within the field.

6.1 Default load signal

The default load signal results are discussed in this section focusing on the output torque peaks.

6.1.1 Without controller

In Figure 6-1 the default load signal simulation is shown with the phases that the shift sequence consists of. The abbreviation FP means filling phase, TP is torque phase, IP inertia phase and CE means completion phase. FP starts when the oncoming clutch (even) reaches critical torque, which can be seen as the speed of the clutch starts changing. When filling phase is completed the torque phase starts, which is defined by the off-going clutch ramping down from reference value (clutch release odd starts to increase from zero). This means that clutch even is starting to release. Slightly after 9.85 seconds the actual handover happens as the clutch release curves crosses each other. Inertia phase starts when the electric motor starts to change speed, as the oncoming clutch is gaining torque. Inertia phase is defined by the components' inertias changing speeds.

Compared to the theory presented in the Literature review the inertia phase is rather short in these results. This is explained by the low inertia of the electric motor compared to the inertias of the combustion engines used in Literature review. The electric motor has an inertia of $1 \times 10^{-4} kgm^2$ compared to the combustion engine used in the standard DCT model (AST, 2015), which has an inertia of $0.134\ kgm^2$.

The big torque peaks are located in the inertia phase. The positive peak is expected according to theory because of the engine speed reduction (Fischer, 2012). The negative peak directly afterwards is most probably because of the electric motor undershooting in speed at the end of the inertia phase.

In the torque phase, before the inertia phase, the theory expects a reduction of engine torque (Qingkai Wei, 2015). This can be seen by studying the output torque after 9.8 seconds in Figure 6-1.

The reason why the output torque is not lower after the shift than before as expected is most probably because of the default load signal having a lower value before the shift than afterwards.

The high input clutch torque (Torque clutch even) in Figure 5-3 is due to the clutch almost reaching its critical torque, which means that it is almost transferring all torque from the motor to the output shaft, at the same time as the big output torque peak occurs. The torque peak is sensed in the clutch but with a reduction of the present gear (2.68 for the oncoming gear). Deviations are present as the clutch is still slipping.

Results from the default load signal without controller (Figure 6-1) are used to validate the results for the models with controller.

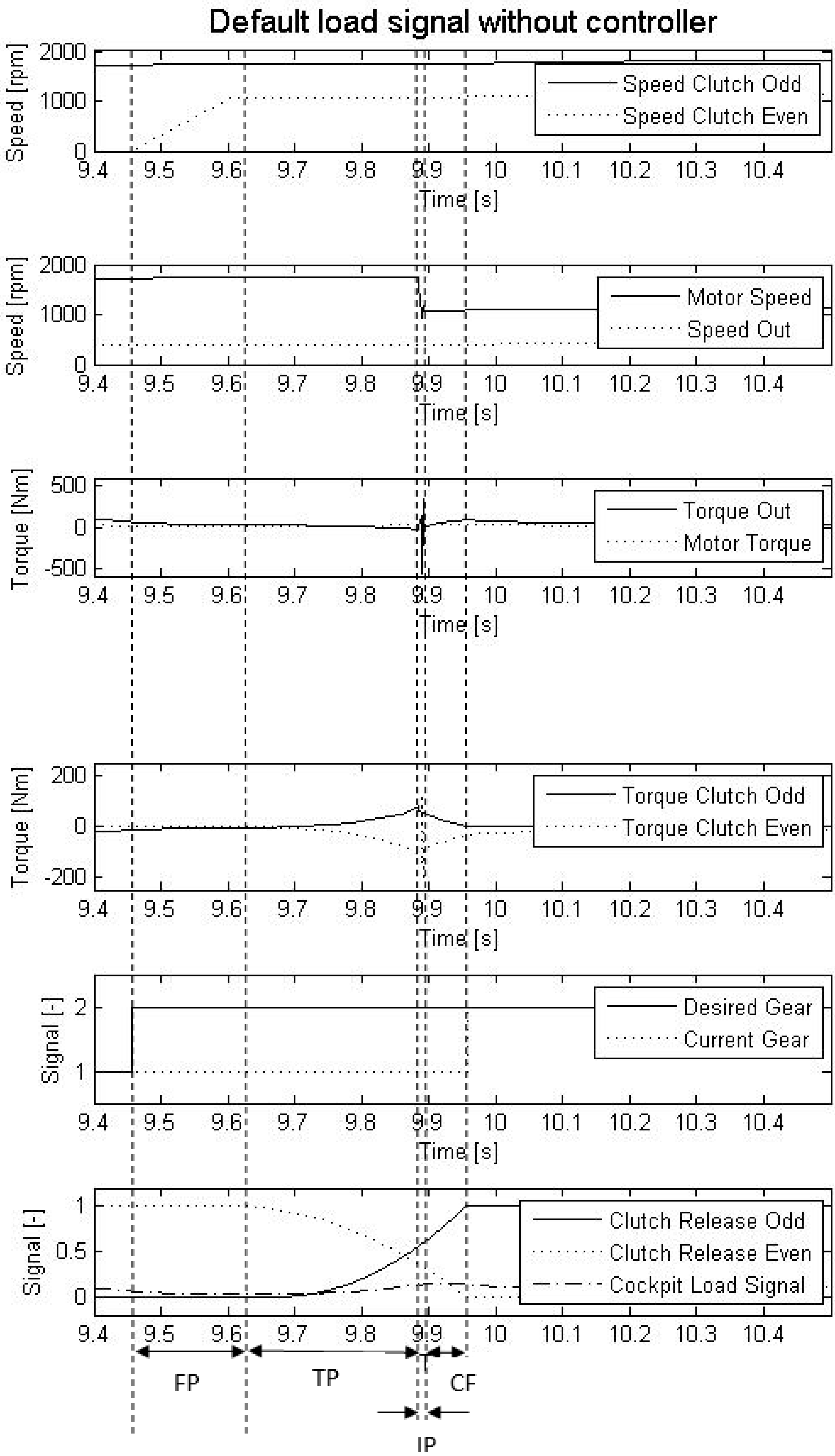


Figure 6-1 - Default load signal with plotted phases

6.1.2 With controller

One can see that the big difference between using the controller and not is the reduction of the torque peaks in the inertia phase, and mainly the big negative peak (Figure 5-2 and Figure 5-4). It can be observed that the controller is altering the load signal in accordance with the error and as a result the torque is regulated from the electric motor and this results in eliminating the electric motor speed peak as well. Figure 5-3 and Figure 5-5 show more detailed plots of the torque peaks.

Table 5-2 shows that the torque amplitude comes down from 912 Nm to 150 Nm using the controller. This is a very big reduction caused by the controller. The big negative peak in the model without controller can also be explained by uncertainties in the simulation results; it is a single peak value and hence not as certain as if there would have been two or more simulation steps with equal values.

6.1.3 Torque comparison

In order to more clearly see the influence of the controller a plot showing the output torque curves of the model with controller and the one without in the same figure (Figure 5-6). One can see that the output torque is reduced significantly during the inertia phase. The big negative peak is eliminated and the negative peak of the motor speed is reduced. The reason for the lower torque peaks is most probably the controller registering the error between desired torque and actual torque and compensating for it. When the peak is positive it gives a lower load signal and vice versa when the peak is negative.

The gradient of the first positive peak is reduced from 35.98 Nm/ms to 28.91 Nm/ms, which means that the rate of torque change is not as large with the controller as without.

The number of torque peaks and the frequency of them are the same as without controller.

There is a mismatch of around 3 milliseconds between the two curves. This is most probably due to simulation technical issues from Cruise.

6.2 High load signal

The high load signal model, compared to the default, differs such that the output torque is higher and that the vehicle is doing a faster acceleration. The load signal is more flat than the default load signal but it is higher after the shift than before, which can be seen by studying the cockpit load signal in Figure 5-1 and Figure 5-7.

6.2.1 Without controller

This version gives the highest output torque amplitude of the models run with the electric motor in this thesis, which can be seen in Figure 5-8. It is mainly the negative peak that increases compared to the default load signal. The behaviour of the output torque is according to theory during the torque phase (decreasing at 2.4 seconds, see Figure 5-8). During the inertia phase the output torque coincides with theory except the positive peak of motor speed at the end of the inertia phase (Figure 5-9). This speed peak is most probably due to the oncoming clutch (clutch even) reaching its critical torque too abruptly.

6.2.2 With controller

With the presence of the controller both the positive and the negative peaks during the inertia phase are reduced drastically. In the torque phase the torque drop is reduced somewhat, because of the modified load signal giving a slightly higher value than the cockpit load signal in order to reduce the torque drop (Figure 5-11). The reduction of the negative torque peak in the inertia phase is explained by the controller giving a higher modified load signal than the cockpit load signal in order to counteract the peak.

6.2.3 Torque comparison

Studying Figure 5-12 one can see that, in comparison with the default load signal, the high load signal case is showing low torque peaks, and especially the negative peak. The positive peak of the motor speed is reduced in the case with controller.

The gradient of the first positive peak is reduced from 16.84 Nm/ms to 12.64 Nm/ms, which means that the rate of torque change is not as large with the controller as without.

The number of torque peaks and the frequency of them are the same as without controller.

The mismatch of around 3 milliseconds is present in the high load signal case as well.

6.3 First inertia phase strategy

In this strategy one can see that the controller cuts down the load signal during the synchronisation phase and thereby reducing the positive torque peak (Figure 5-14) for the high load signal case but the lower limit of the torque peak is reduced further. The speed transition is not very smooth compared to the case with PI controller but the absolute value of the torque peaks is reduced. The same strategy for the default load signal case is effective to a lesser extent in reducing the absolute value of the torque peaks but the negative peak is decreased further (Figure 5-13) and the gradient is also higher (Table 5-2). But the speed transition is smoother in this case.

6.4 Second inertia phase strategy

From the results in Section 5.4 one can see that the second inertia phase strategy is effective in reducing the gradient of the peaks for both load signal cases in comparison with the previous strategies but the amplitude of the torque peaks is higher. From Table 5-2 it can be seen that the strategy works more effectively for the default load signal case in controlling the gradient. It is observed that the speed transition in this strategy is not very smooth compared to the previous two strategies. The oscillations in the modified load signal are most probably a result of the speed variation of the electric motor because of the way the controller is designed in the second inertia phase strategy (using the motor speed as input). The speed transitions in both the inertia phase strategies are not smooth as in the first strategy (PI Controller) since the motor torque variation is of greater magnitude during the speed synchronisation phase and thus resulting in stepped reduction of motor speed.

6.5 Clutch control strategy

The clutch control is one of the most important parts of a DCT. The timing of the off-going and oncoming clutch as well as the behaviour of the clutch torque during the actuation time is a factor that can improve the shift quality. However, the topic of this thesis is to use torque control to improve the shift quality. Therefore only some simple simulations of clutch control strategy were run with results showing that changing from the standard clutch timing used in Cruise does not improve the shift quality by the means of torque amplitude during the shift procedure.

6.6 Analysis of the controller

- The simulations are run at a very high sampling frequency and the controller is tuned to respond to this high frequency and this may not be applicable if it is to be implemented into a real system.
- For both the load signal cases during the shift it is observed that there is a build-up of error. This is due to the fact that during the shift, since neither of the clutches are completely engaged and the torque transferred is always lesser than the set point value, and the controller not being aware of this, it is increasing the electric motor torque in order to compensate for this error and thus resulting in torque build up and thereby increasing the positive torque peak once the oncoming clutch is engaged. This build-up of the error is reasonably controlled when a PID is used but the negative peak is higher. Thus a PI controller is being used instead.

6.7 Experimental uncertainties

This section reflects over possible error sources causing results that might not be valid to full extent.

6.7.1 Simulation settings

One source of error that is influencing this type of simulation work is the choice of simulation settings. Even though convergence was achieved there can still be issues with that in different simulation models since the convergence was not controlled for every model. A result could be that the single peak values might not be fully realistic. However, a smaller time step than 1ms was tried, but still giving the same result.

6.7.2 Transmission mechanics

A large ratio step generally results in lower shift quality than a small ratio step (Darrell Robinette, 2015). Since the shift analysed in this thesis is 1^{st} to 2^{nd} gear the ratio step is rather high (For exact values, see Table 3-2), which means that the conditions for this shift in particular is not the best for achieving the best shift quality.

6.8 Recommendations for future work

- Development of a controller with functionality applicable to different types of shift scenarios.
- Gain scheduled controllers can be used wherein few controllers can be tuned for different load signal cases and then a lookup table can be formulated to get specific gain values for the corresponding load signal values.
- Further development of the inertia phase control strategy in order to extend the influence of the controller during torque phase to reduce torque peaks.
- If the controller is made in Matlab Simulink, work on getting the interface between it and Cruise should be taken up in an early stage of the project. A lot of time had to be spent on getting the interface between the two software to run.
- The system dynamics can be modelled in more detail by the addition of shafts with their stiffness between components in the powertrain and the delay in the motor response time to the controller signals can also be included in the further development process in order to obtain a more realistic model.

7 Conclusions

The power-on upshift for a standard DCT is simulated in AVL Cruise and a controller is designed to control the output torque of the electric motor during a power-on upshift procedure in order to improve the shift quality. After close examination of the obtained results it is concluded that the controller is effective in keeping the torque fluctuations within acceptable limits for both default and high load signal cases with the controller being more effective for the default load signal case.

Inertia phase strategy is effective in controlling the positive torque peaks during the inertia phase but the lower end of the torque peak is reduced as an effect. The effect of the control strategy is more evident for the high load signal case in comparison with the default load signal case. The second inertia phase strategy is better in reducing the gradient of the first positive torque peak compared to the first inertia phase strategy. On the other hand it gives a bit higher torque amplitude.

The results show that torque amplitude and the gradient of the first positive torque peak can be reduced significantly using input torque control hence improving the shift quality.

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Appendix

Appendix A: Cruise to Matlab API configuration

AVL Cruise can be connected to Matlab Simulink using API (Application programming interface). In order to do so there are some settings that has to be done to the computer that the programs are installed on. The list and the figures that are presented in this section explain step-by-step how to proceed with these settings.

1. Cruise 2015 can be run as 32- or 64-bit program. Depending on what bitness your Matlab is you will have to set the bitness in Cruise. This is done according to (Figure A).
   For 32-bit, choose the following file:
   C:\Program Files (x86)\AVL\CRUISE\v2015\CRUISE\v2015\bin\bin.ia32-unknown-winnt_i12v10\ cruise_m
   For 64-bit, choose the following file:
   C:\Program Files (x86)\AVL\CRUISE\v2015\CRUISE\v2015\bin\ bin.x86_64-unknown-winnt_i12v10\ cruise_m
2. The path in Cruise needs to be set in order for Cruise to find Matlab.
   a. Open Cruise
   b. Click on Vehicle Data=>options=>configuration (Figure B)
   c. Click on Environment=>Path (value) (Figure C)
   d. Add the following three paths coming from your Matlab installation folder (Figure C), (Figure D), (Figure E).**Observe! It has to be corresponding to where you have your Matlab installation files!**
3. Matlab regserver has to be set:
   a. Open the command prompt: *"Windows+R"* and type *"cmd"*.
   b. Change directory to where you have your Matlab installation files, e.g C:\
      i. To reduce the path to only C:\, type *"cd .."*. See (Figure G)
      ii. Type *"cd Program Files (x86)"* (You can do it faster by just typing *"cd P"* and then use tab to scroll to correct name.
      iii. Keep on changing directory until you reach the win32 folder. The path should then look something like this:
      C:\Program Files (x86)\MATLAB\R2010a\bin\win32
      iv. Type *" matlab /regserver"*
4. The path in your Windows environment variables need to be set:
   a. Go to My Computer
   b. Click System properties (Figure F)
   c. Click Advanced System Settings (Figure H)
   d. Click Environment Variables (Figure H)
   e. Open the path under System Variables that is called "Path" (Figure I)
   f. Add the bold part of the path shown in bullet A. **Observe! It has to be corresponding to where you have your Matlab installation files! There has to be NO empty spaces anywhere but within the folder names (e.g *"Program Files"*).**
   g. Restart the computer.

A. C:\ProgramData\Oracle\Java\javapath;C:\Program Files\Common Files\Microsoft Shared\Windows Live;C:\Program Files (x86)\Common Files\Microsoft Shared\Windows Live;C:\Program Files (x86)\NVIDIA Corporation\PhysX\Common;%SystemRoot%\system32;%SystemRoot%;%SystemRoot%\System32\Wbem;%SYSTEMROOT%\System32\WindowsPowerShell\v1.0\;C:\Program Files (x86)\EgisTec MyWinLocker\x86;C:\Program Files (x86)\EgisTec MyWinLocker\x64;**C:\Program Files (x86)\MATLAB\R2010a\runtime\win32;C:\Program Files (x86)\MATLAB\R2010a\bin\win32**;C:\Program Files\MATLAB\R2014a\runtime\win64;C:\Program Files\MATLAB\R2014a\bin;C:\Program Files (x86)\QuickTime Alternative\QTSystem;C:\Program Files (x86)\Windows Live\Shared;C:\Program Files\Broadcom\Broadcom 802.11 Network Adapter;C:\Program Files\Intel\WiFi\bin\;C:\Program Files\Common Files\Intel\WirelessCommon\;C:\Program Files\Microsoft Windows Performance Toolkit\;C:\Program Files (x86)\Skype\Phone\;C:\Program Files (x86)\MATLAB\R2010a\bin

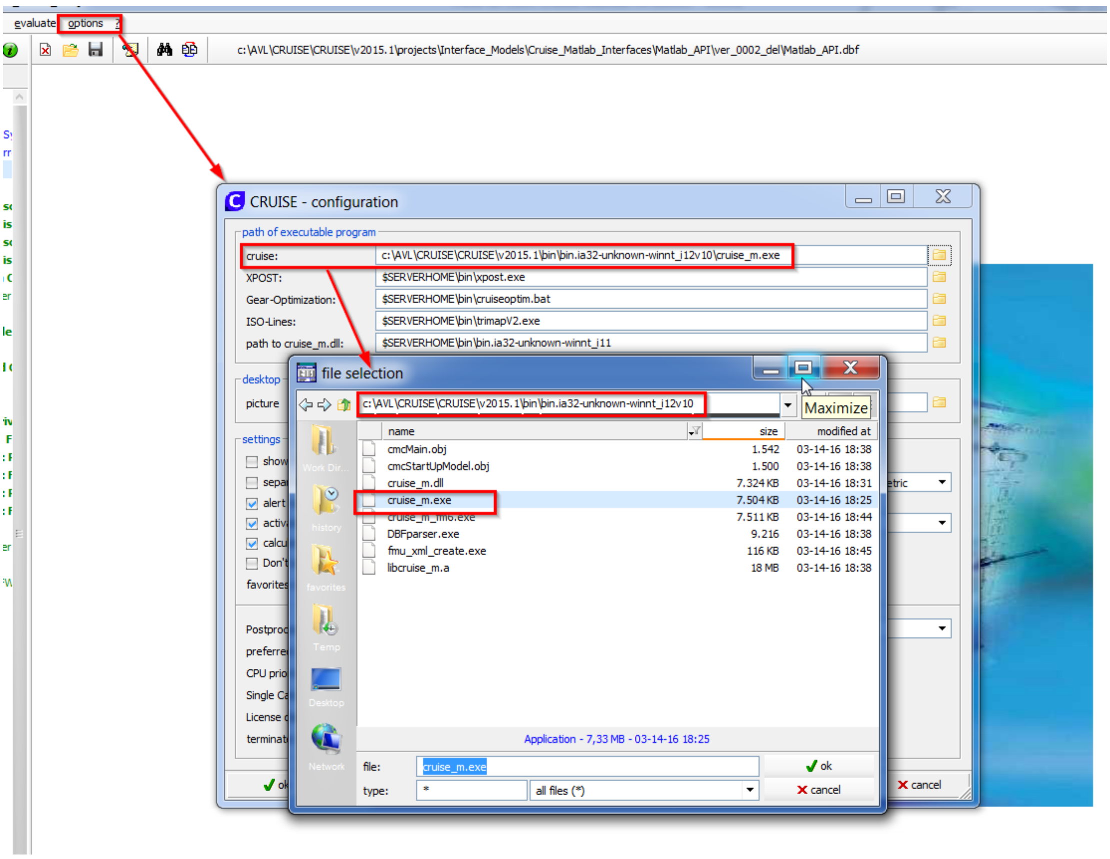

Figure A - Cruise bitness

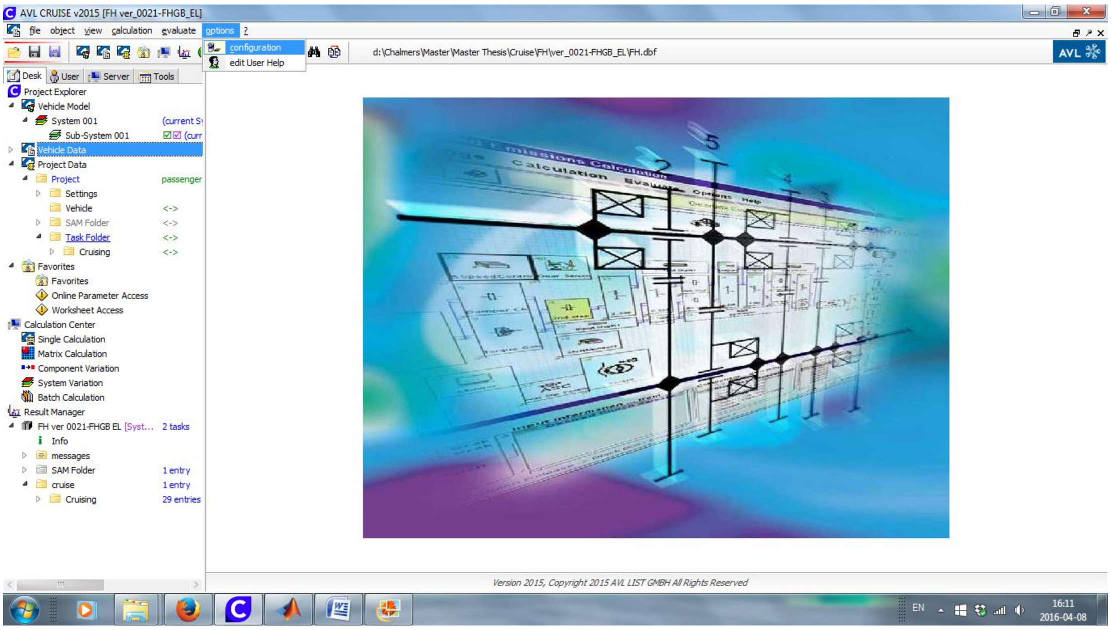

Figure B – Cruise path configuration

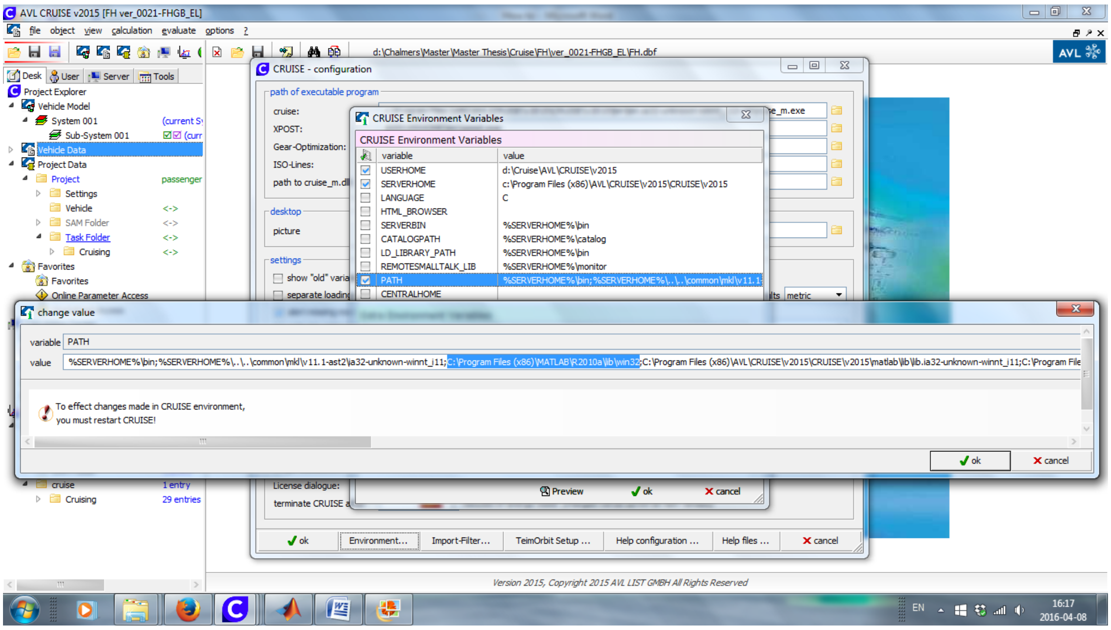

Figure C – Cruise to Matlab path 1

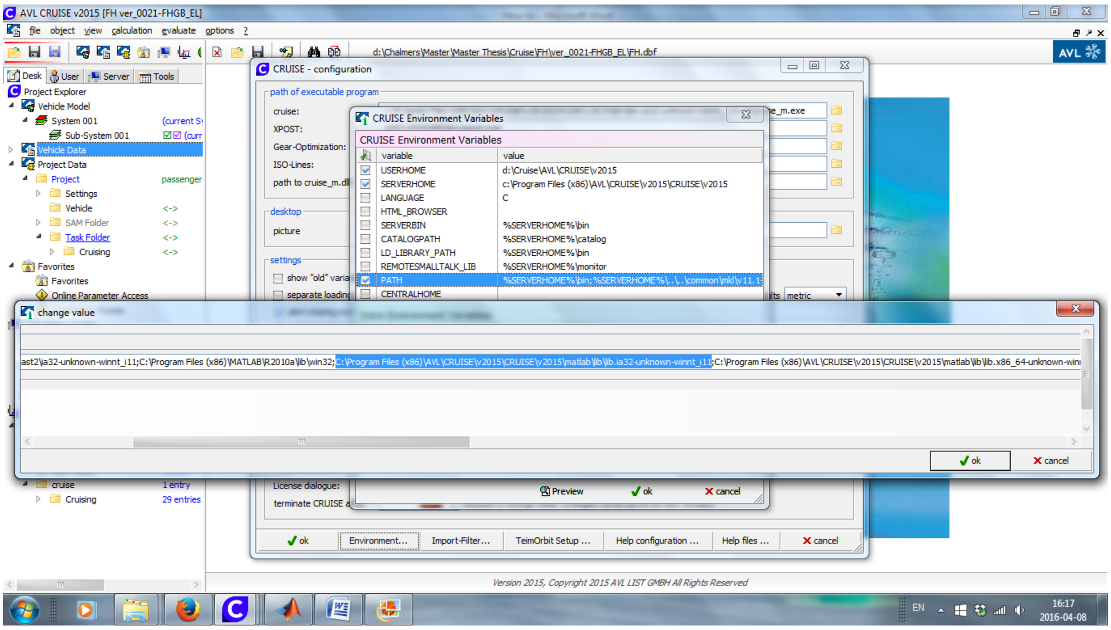

Figure D – Cruise to Matlab path 2

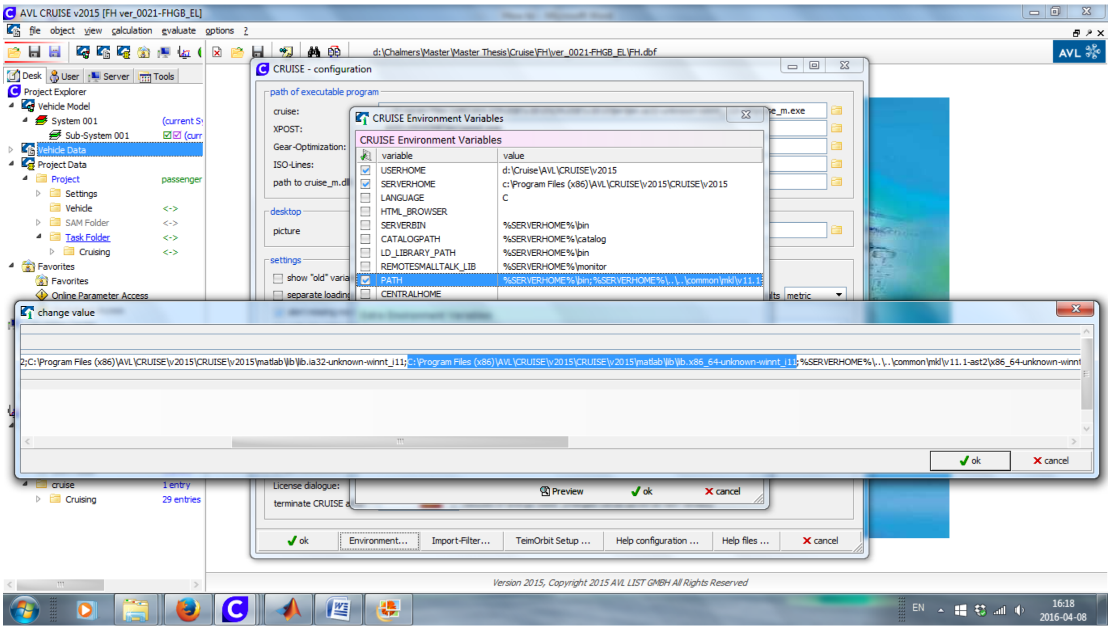

Figure E – Cruise to Matlab path 3

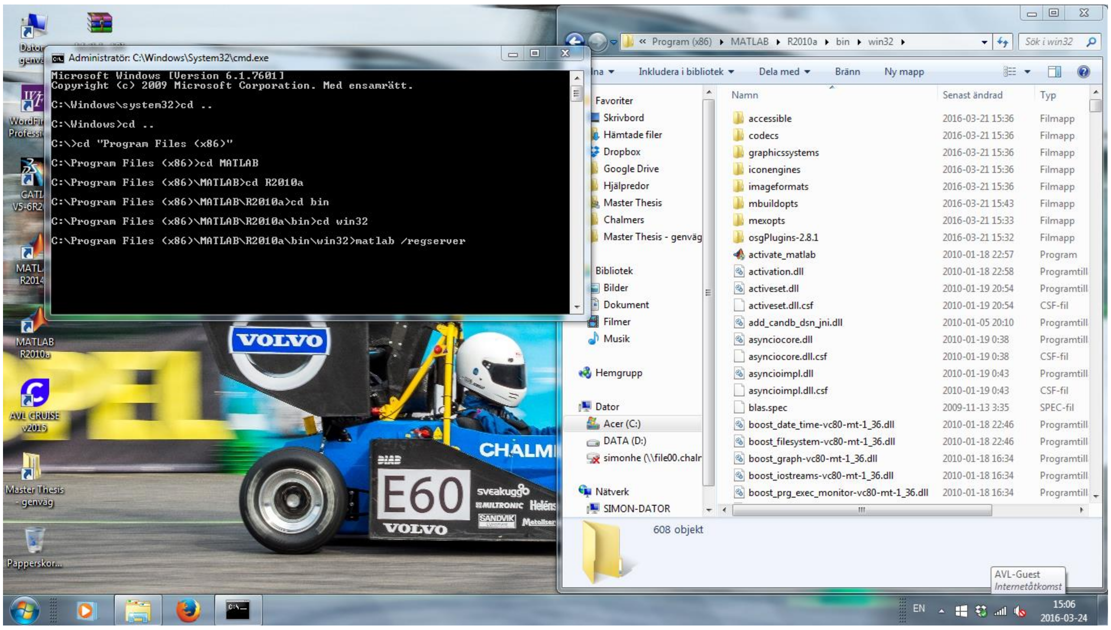

Figure G – Matlab regserver configuration

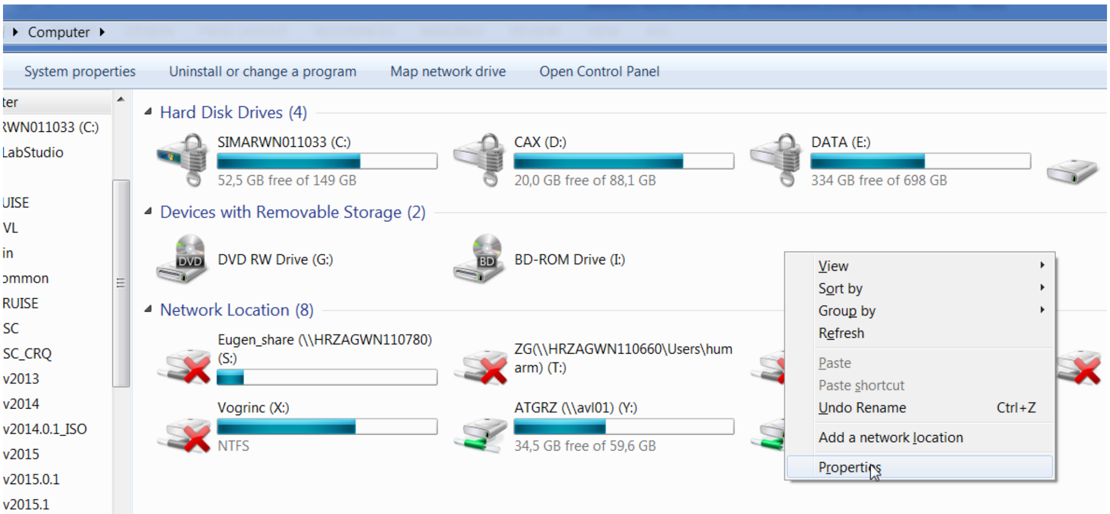

Figure F – Windows path configuration 1

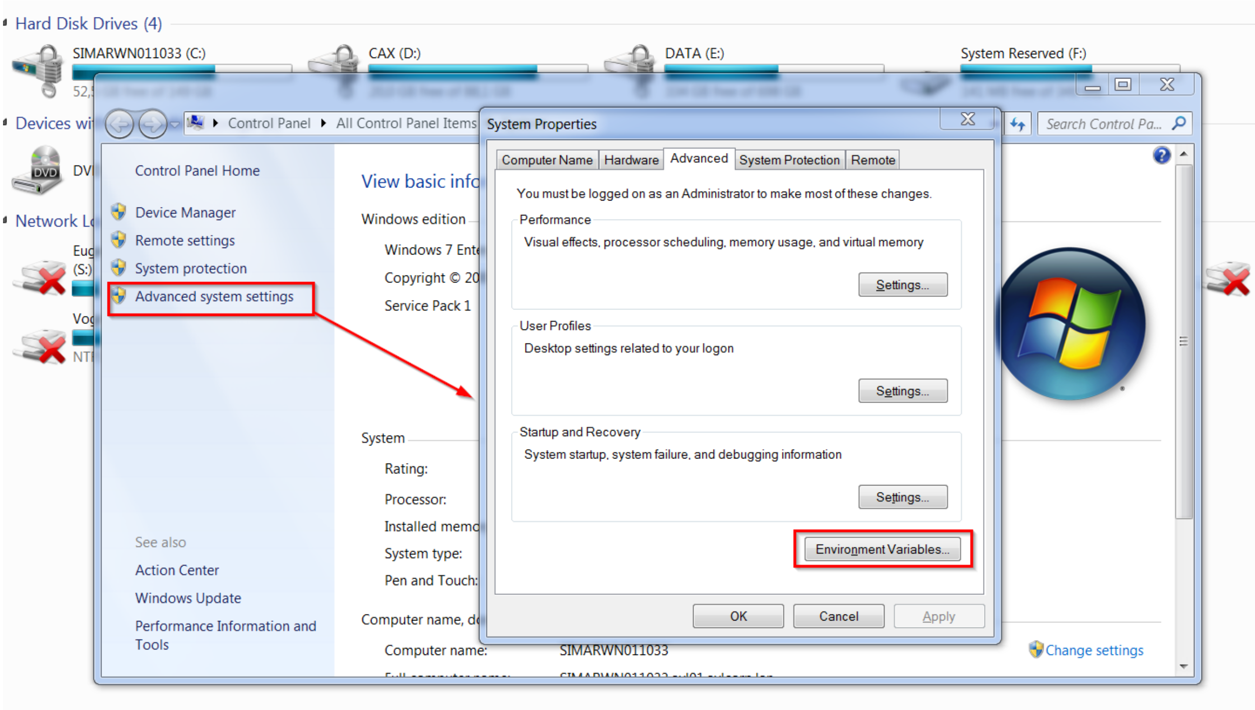

Figure H - Windows path configuration 2

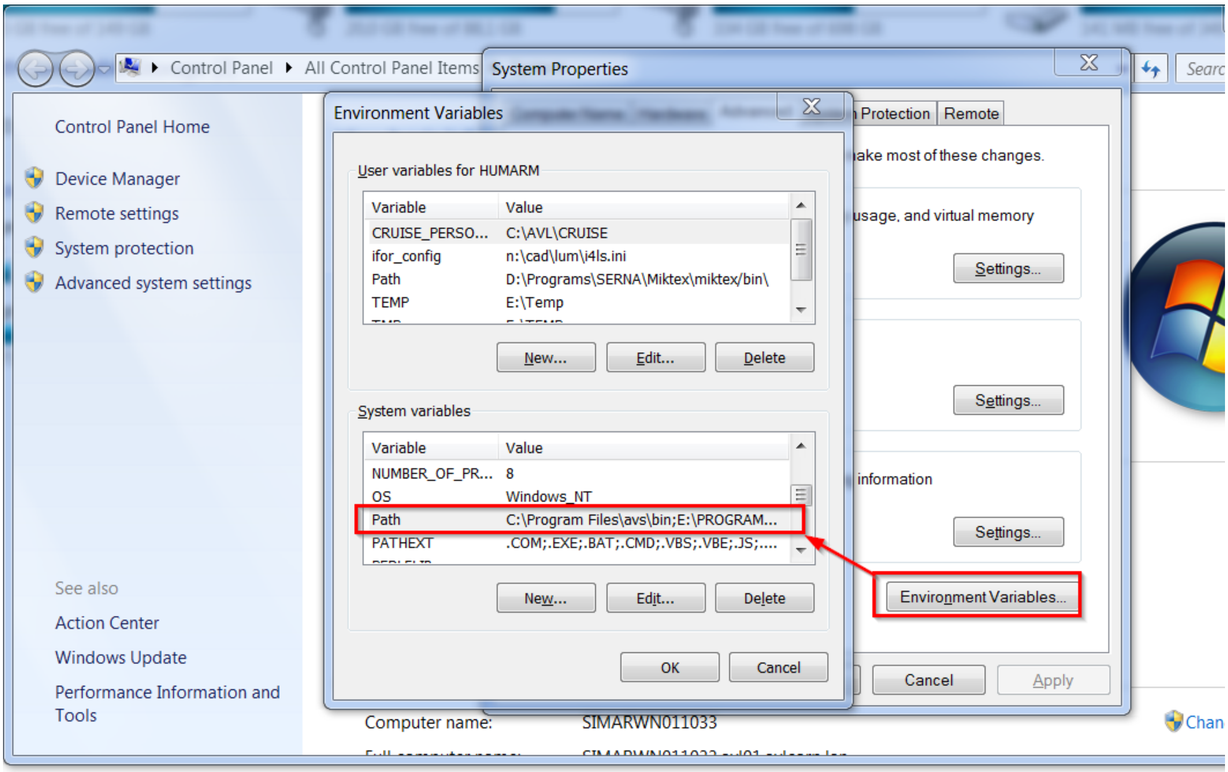

Figure I - Windows path configuration 3