Master of Science Thesis in Electrical Engineering
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Modeling and control of engaging gears in gearboxes without synchromesh towards specific angles between gear and coupling sleeve

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Abstract

When engaging a new gear in an automated manual transmission (AMT) the gear needs to be synchronized with the main shaft's angular velocity in the gearbox. This is so that the parts can be connected through a cog wheel and torque can be transferred. To synchronize the angular velocities, mechanical synchronization components can be used. These components synchronize the velocities during the engagement and can be used with larger differences in angular velocities. Should no mechanical synchronization components be used it puts higher demands on the components rotating at similar velocities to avoid mechanical wear and ensure that the gear can be engaged. In today's systems without mechanical synchronization components the gear is engaged when the angular velocities are within a certain difference. This leads to random angle connections between the cogs and gaps that are to be engaged on the gear and main shaft. This can lead to extended or incomplete engages should the components connect cog to cog.

This thesis evaluates the possibility to control the angle at which the components connect by using existing sensor signals in the studied system and known parameters. A model of the system is created and simulated to evaluate the probability of predicting the system over the gear engage. Results indicate that it is possible to predict the connection angle close enough to its real value so that a control strategy could be implemented to control the angle to some level.

Acknowledgments

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Contents

Notation

Parameters and variables

Notation	Description
θ	Angle
$\dot{\theta}$	Angular velocity
$\ddot{\theta}$	Angular acceleration
τ	Time constant
J	Moment of inertia
T	Torque
i_{split}	Gear ratio of splitter gears
T_{db}	Braking torque from the disc brake
n_{cogs}	Number of cogs per revolution for Hall-sensors
c	Friction constant for layshaft and input shaft
μ	Friction constant of disc brake
p	Pressure
Q	Flow
F	Force
K_{valve}	Valve pressure constant
r	Radius
t_{delay}	Time delay for the actuators controlling the gear engages

Abbreviations

Abbreviation	Description
LS	Layshaft
IS	Input Shaft
MS	Main Shaft
RPM	Revolutions Per Minute
ZOH	Zero Order Hold

1 Introduction

This technical report will present the master thesis *Modeling and control of engaging gears in gearboxes without synchromesh towards specific angles between gear and coupling sleeve* performed at Scania CV (Södertälje, Sweden). It investigates the possibility to use a control strategy when engaging a gear in the gearbox given the available signals from the system.

1.1 Background

In today's automated manual transmissions (AMT) there are two types of gear engagement, synchromesh and non-synchronous. The velocities to synchronize are the angular velocities of the main shaft and the gear to be engaged. More information about these components are found in chapter 2. In reality the gear is engaged at close to synchronous speeds in both methods but the non-synchronous gearbox has gotten is name due to the fact that it does not rely on mechanical synchronization components. The principle for engaging gear in a non-synchronous gearbox is to have a small difference in the rotating speeds of the cogwheel on the gear to be engaged and the coupling sleeve that is attached to the main shaft. The coupling sleeve is used to engage a gear by being mounted so that it is immovable rotationally on the main shaft but can slide along it. The gears are mounted freely rotating on the main shaft and has cogs on the side which can lock into the coupling sleeve and through this transfer torque.

When the speed difference is less than a specific small value, the control system tries to engage the gear. Due to the difference in the rotating speeds the gear will lock into the cogs on the main shaft if the cogs and gaps do not match instantly. Compared to a synchromesh gearbox which uses mechanical parts to match the speed of the main shaft and the gear, a non-synchronous gearbox does not use any mechanical components to synchronize the rotations which makes it cheaper and lighter. The drawback is that it puts higher demands on the control system since it needs to be more accurate in the gear change to match the rotational velocities.

1.2 Problem formulation

A method of engaging a gear in gearbox without synchromesh is to engage the gear when the angular velocity difference is within a given interval as described in section 1.1. This leads to random connections between the cogs on the coupling sleeve and the cogs on the gear to be engaged. Therefore some of the engagements will be slower because of cog-to-cog collisions before sliding into engaged gear. A too small difference in rotating speeds when engaging a gear might lead to a very slow or incomplete gear change if it collides cog-to-cog. Should an incomplete gear change occur and the gear engage has to start over, the truck might come to a stop if it is currently in a slope upwards and the truck rapidly looses speed when no gear in engaged. A too big difference in rotating speeds could result in the cogs just sliding across each other and never becoming fully engaged. This could also lead to excessive wear on the cogs.

1.3 Method

Should it be possible to control the angle with which the coupling sleeves and gear to be engaged collide, the cases where the engage takes excessive time due to connection cog-to-cog could be avoided. Since there is a time difference between activation of the gear engagement and the actual engagement/collision a prediction of the angle difference have to be performed. This prediction will be based on angles measured by existing Hall-sensors on the shafts in the gearbox. If the prediction can be made sufficient it could be used to avoid prolonged gear engages and/or ensure more efficient gear engages given there is a small speed difference. More information about the system and sensors are found in chapter 2. The time difference will be determined based on data from previous tests on the gearbox.

To investigate the possibility to create such a control strategy, the method is to initially create a model over the studied parts of the gearbox. These are the layshaft's angular velocity and angle, a disc brake connected to the layshaft and the valve controlling the input pressure to the disc brake. This model will be based on known equations of the studied components and constants will be calculated with measurements from a test performed in a rig with the system. The test in the rig has extra sensors attached and the studied signals are presented in chapter 4 and 6. When a satisfactory model of the system is created and compared to measurements, the following step is to implement predictions of the angle and angular velocities of the gearbox's shafts in the control system. Starting with a simple solution based on linear velocity and evaluating its result with the model, different prediction methods will be tested to evaluate their results. The most promising method will then be used in a real system to evaluate the prediction method as well as the control strategy.

Tests and simulations may show that, considering the existing systems in the trucks, it might be difficult to detect the angle of the components sufficiently to use the information when engaging a gear. It could also be the case that the systems is too slow or inaccurate to steer towards the angle of the gear to be engaged. Should this be the case, an analysis of the required update or adding of hardware or software will be performed to give a sense of future possibility to use this concept. The model's sensitivity to modeling errors and disturbances, like sensor noise, is to be evaluated.

If there is a possibility for a control algorithm to predict and steer towards a given angle between the coupling sleeve and the target cogwheel it could also be used in future tests to explore the effect that different "engagements angles" have on wearing on the mechanical components. This is however not a goal for this master's thesis.

1.4 Limitations

To gain more insight regarding the future behaviour of the angular velocity of the main shaft in the gearbox, a model of the driveline from the gearbox to the wheels could be created. This could use current and upcoming information about the slope of the road, rolling resistance, driver behaviour etc. Such a model would require more time to be studied and modelled and is therefore not considered. The angular velocity of the main shaft in the gearbox is therefore considered constant during a gear change in this thesis.

1.5 Thesis Outline

A short description of the thesis outline is presented here.

Chapter 1 Short introduction to the background behind the thesis and its problem formulation.

Chapter 2 The studied system is presented and key components are described further.

Chapter 3 Related and previous work done in the field is presented in this chapter.

Chapter 4 The produced models over the studied system are presented.

Chapter 5 This chapter contains an explanation of the control strategy's objective and the approaches being used.

Chapter 6 The results from simulations and calculations are presented.

Chapter 7 The results and their validity are discussed in this chapter.

Chapter 8 Conclusions and future work.

2 System

In this chapter the studied system is described. The system is to some extent simplified compared to the real system in order to be more general in the approach and also for the purpose of only observing the essential components of the gearbox.

2.1 General layout

In figure 2.1 an overview of the system can be observed. The engine is connected to the input shaft of the gearbox via the clutch. The input shaft transfers torque to the layshaft through the splitter gear. To engage a gear, an actuator is used to move a coupling sleeve on the shaft. The coupling sleeve is immovable rotationally on the input shaft but can slide along it. It can through cogs be connected to either of the two splitter gears to transfer torque to the layshaft. The actuator in turn is maneuvered by two valves connected to a pneumatic system. The layshaft has four different gears that transfer torque to the main shaft. The gears rotate freely on the main shaft when they are not engaged. They are engaged with two coupling sleeves where gear one and two share the same coupling sleeve and gear three and the crawl gear share one. Crawl is the lowest gear and is mostly used only when accelerating from standing still. The coupling sleeves mounted on the main shaft and transfer torque the same way as the splitter gears coupling sleeve.

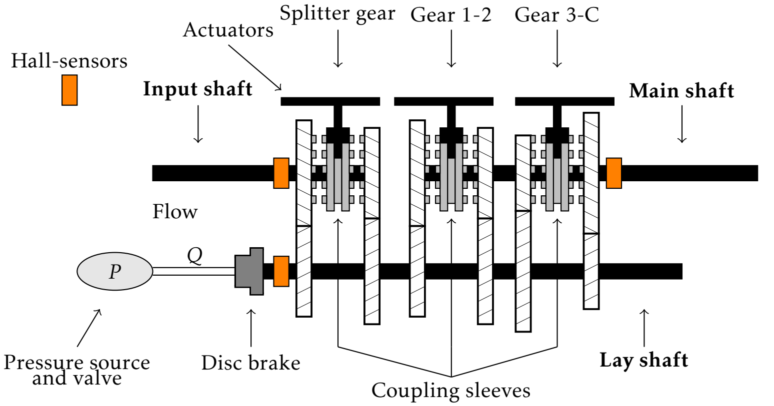


Figure 2.1: *A basic overview of the studied system.*

Connected to the layshaft is a disc brake that will slow down the layshaft when needed. This could be during a gear change to a higher gear when the cogwheel for the gear to be engaged needs to be slowed down to be within a demanded speed interval from the main shaft. The main shaft is connected to a propeller shaft via a gear set called range. This system is not within the scope of this thesis.

On all three shaft's there are angle sensors and from the measurements the shafts rotational speed can be calculated. They are Hall-sensors and they are described in more detail in section 2.5.

2.2 Gear engagement

As mentioned in section 1.1, the system does not have a mechanical synchronization part which would synchronize the gear to be engaged with the main shaft in the gearbox. This is normally done with a conical part between the gears and coupling sleeves which will add friction when the parts are pushed together. This will slow down or speed up the gear to be selected, depending on whether it is an up- or down shift in gears, and lets the gear be engaged when it has the same rotational speed as the coupling sleeve. In this system there is no parts between the gears and coupling sleeves since the system relies on the disc brake and engine to handle the synchronization.

To shift into a new gear, the clutch is first disengaged and the initial gear can be disengaged since there is no or very little torque transferred over the component. The layshaft then needs to be accelerated or decelerated to fulfill the required equation (2.1) where $\Delta\dot{\theta}_{MS}$ is the required difference in rotating speeds and i_{gear} is the ratio between the layshaft's and the gear's rotational speeds. Subscript "LS" stands for layshaft and "MS" for main shaft.

$$\dot{\theta}_{LS} i_{gear} + \Delta\dot{\theta}_{MS} = \dot{\theta}_{MS} \tag{2.1}$$

As mentioned in 1.1, the difference in rotational speed will let the gear connect with the coupling sleeve should the cogs and gaps not match instantly. Should the difference in speed be too high, it might lead to excessive wearing on the components. Should a too small difference be used, the gear might get stuck on the coupling sleeve cog-to-cog without successfully engaging the gear.

If the gear change is to a higher gear, the angular velocity of the layshaft is lowered using the disc brake located on the shaft. To engage a lower gear, the clutch is engaged again and the engine is used to accelerate the layshaft to the target RPM (Revolutions Per Minute). When the gear to be engaged has gained a higher RPM than the main shaft the clutch is disengaged again. This is so that during the disengagement of the clutch and activation of the actuator controlling the coupling sleeve, when the target gear is loosing angular velocity, it doesn't fall below the possible $\Delta\dot{\theta}_{MS}$ range.

2.3 Disc brake

The brake on the layshaft is a disc brake which is controlled/engaged through a pneumatic system. The applied pressure pushes discs on the shaft and brake together. This creates a braking torque on the shaft due to friction since the brake is mounted fixed in the gearbox.

The pneumatic pressure and air mass flow to the brake is supplied from a valve which is controlled by the control system. Connected to the valve is the same pneumatic system as in the rest of the vehicle's pneumatically controlled components. There are different tanks in the vehicle which are connected to a main storage that is driven by a compressor. The pressure from the supply might fluctuate slightly when different components use the system but in this thesis the pressure is regarded as constant as it would require an excessive amount of work to take all the different components connected to the pressure supply into consideration.

2.4 Gear actuators

The coupling sleeves that control which main and splitter gears are engaged are controlled by linear actuators which are driven by two valves each to move the coupling sleeve along the shafts. The valves are connected to the same pneumatic system as the valve controlling the disc brake on the layshaft. To engage a gear, the corresponding valve is activated and when the control system observes that the gear is engaged, the valve is closed and the driving torque on the gear will hold the gear in place. The completed gear engagement can be confirmed with a position sensor on the actuator.

2.5 Angle sensors

The angle sensors that are connected to each rotating shaft consists of Hall sensors and coils. A current is sent through the coil and the generated magnetic field will vary based on the surrounding components. By aiming the sensor towards cogs on a cogwheel on the shaft the magnetic field strength will vary. The voltage from the sensor will thus vary with the frequency of cogs passing the sensor with a sinusoidal appearance. When the signal passes a certain rising value that indicates a ”cog-rise” the current time is stored in an array. The system also stores the time stamps for each ”cog-fall”.

The system reads the value from the Hall sensors work at the frequenzy of 1 MHz which gives timestamps with a accuracy of 1 µs. The array is read every 10 ms. An example of the array for two readings in the control system, where two more cogs has passed the Hall sensor, is shown in figure 2.2.

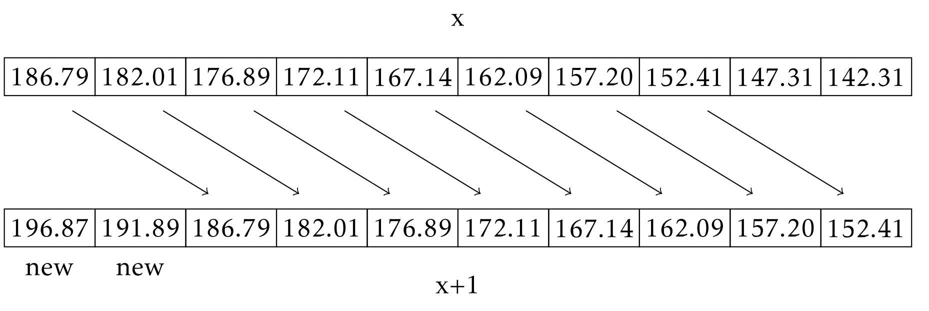


Figure 2.2: *An example set of timestamp arrays for $t_1 = x$ and $t_2 = x + \Delta t$ where two cogs have passed the sensor. Time is in milliseconds [ms] and the update frequency of the array to the control system is $t_2 - t_1 = \Delta t = 10$ ms*

3

Related work

Related work to the thesis is presented in this section.

3.1 Modeling

There has been plenty of work and articles written about modeling of entire drivelines and physics in the engine of trucks and cars. In these studies the gearbox is mostly considered to be a transfer of torque and angular velocity with a ratio from the engine to the propeller shaft. An example of this can be viewed in figure 3.1 below. The goal is often to create a platform for analysis regarding affects to fuel efficiency or drivability and affects on torque at the wheels. Most of these publications use a similar approach and among these are [6] and [13].

The information to gain from these sources is regarding how torque and angular velocities transfers through a transmission. A model of a transmission can be viewed in figure 3.2. Equations (3.1) and (3.2) are commonly used to describe transferred torque and angular velocity. $\dot{\theta}$ is the angular velocity, T is an acting torque and i_z is the ratio of the gears.

$$\dot{\theta}_{IS} = \dot{\theta}_{LS} i_z \tag{3.1}$$

$$T_{IS} i_z = T_{LS} \tag{3.2}$$

Regarding the components in the gearbox described in section 2, these are partly studied and modeled in [8]. However, this is a study performed on a system which uses hydraulics instead of a pneumatic system and focuses more on the hydraulic behaviour of the system. No brake in the gearbox is modeled either.

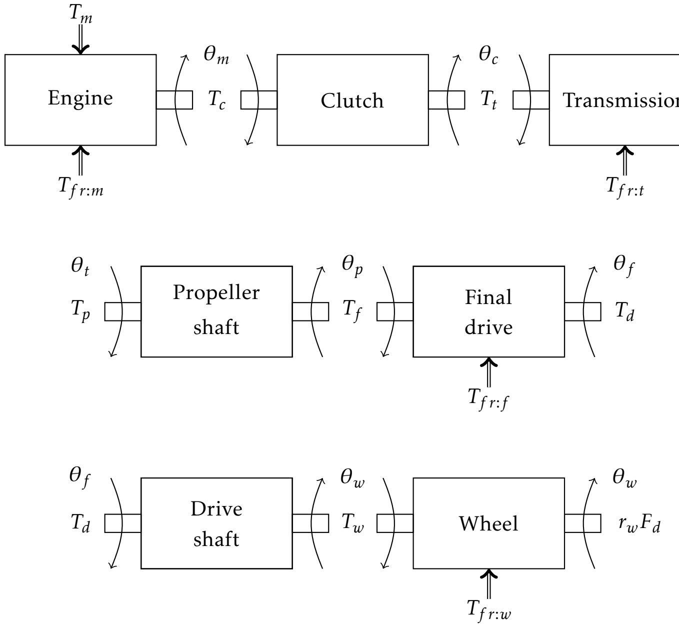


Figure 3.1: *A general driveline model, where the sub-components describes the relations between the incoming and outgoing shaft torques and angles.*

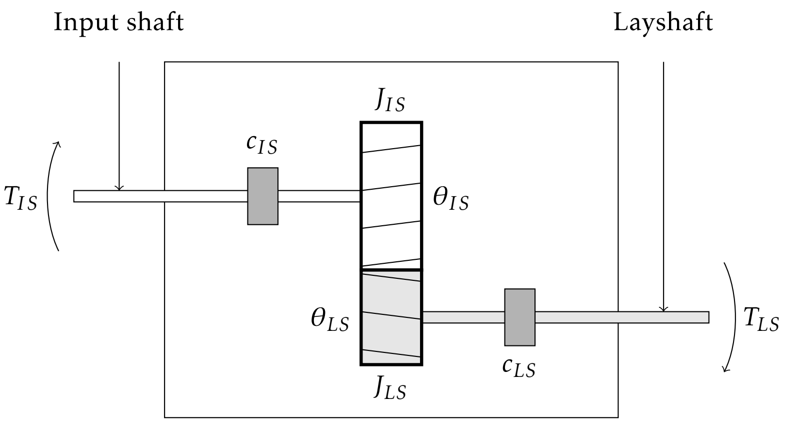


Figure 3.2: *The input shaft describes the torque T_{IS} delivered from the engine. c_{IS} describes the friction acting on the shaft and J_{IS} is the inertia of the shaft and cog wheel. The notations for the layshaft are the same.*

The study is therefore only partly interesting for this thesis. Modeling of disc brakes often follow typical behaviour for such components and information and their physical properties can be found in other literature and in available data.

In [9] a model for the actuators in a AMT gearbox is developed. This includes a model for the clutch as well as a model for an electro-hydraulically operated actuator system that controls the coupling sleeves engaging the selected gears. It is a physically based nonlinear model which focuses on the actuator's dynamics and their integration with the gearbox and driveline. The studied system and the model contains a mechanical synchronization component between the coupling sleeve and the gears. This is modeled with Coulomb friction which will slow down the target gear to match the main shaft's velocity. According to the article, a typical synchronization process takes about 60 *ms* to gain the same speed of the components during a shift to a higher gear and thus slowing down the layshaft. As this thesis studies a system without a mechanical synchronization part and uses a pneumatically driven actuator system, this article can be used as an approach to modeling of a similar system although it will not be exactly the same.

In [2] a thorough description is presented of the process of synchronizing the angular velocities when engaging a gear when a synchromesh is used. It also investigates the effect that some of the design parameters of the synchromesh, eg. different cone angles, has on the time to engage a gear and the driving comfort. This is made using a theoretical model of the synchromesh and calculations and simulations are made in MATLAB.

This thesis will provide a more complete model on pneumatically controlled AMTs with a disc brake than any of those found in previous publications.

3.2 Control strategy

There are a number of studies performed on *when*, in a drive cycle, to perform a gear shift. Among these is [10] which studies different methods of estimating optimal gear-shifting to obtain fuel-efficient driving on different drive cycles. This is not relevant to this thesis since the subject of *how* to perform a gear shift, when the gear shift has been initiated, is to be studied.

In [14] different control algorithms for controlling the transferred torque through the gearbox during a change in gears is evaluated. This has partly the same goal as this thesis although it analyses another cause not satisfying gear shifts. It studies control algorithms for the engine to match the gearbox's angular velocity for the new gear to be engaged as well as having good driving comfort and having a fuel-efficient gear change.

No publication is found where the angular difference between the cogs on the coupling sleeve and the target gear is regarded. Since that is this thesis's objective it should have a new approach to the objective of achieving smooth gear changes.

3.3 Sensors

This thesis relies to a major part on the ability of estimating states based on sensor readings. The only sensors used in the studied system are Hall sensors that measure the angle on the shafts in the system. Hall sensors measure the magnetic field close to the sensors, as explained in section 2.5. Since the electric resistance varies in the observed element when the temperature changes, so will also the magnetic field which makes the sensor vary in sensitivity [1]. Because of limitations in the magnetic field, the Hall sensors need to be mounted as close as possible to avoid weak signals and interference from surrounding magnetic fields [1]. Because of imperfect geometry in the production, all Hall sensors have a small offset in the output voltage [11]. This thesis studies a rotating part where the frequency is of more interest than the amplitude. The offset could however affect the angle, but this can be handled by selecting the amplitude at which time stamps are written.

3.4 Event based input

Most of discrete automatic control theory is based on a system where sensor signals comes with a fixed time step. With the angle sensor system described in section 2.5 the measured values, in this case timestamps, are acquired with varying time steps since they will depend on the current angular velocity. Even though the angular velocity is updated with a set time step the angle is updated from the sensors event based. The impact of this characteristic has been studied in a couple of publications. Among these are [5].

In [12] event-based sampling is studied. Different methods for estimation of single resonance frequency in rotational speed signals are presented. A tire pressure monitoring system is produced and analyzed where the input is from a wheel speed sensor where the measurement is event-based.

4 Models

In this chapter the modeling of the studied system is presented. The components chosen to be modeled are presented in chapter 2. The modeling is made in Matlab/Simulink environment.

4.1 Limitations

Apart from the limitations mentioned in section 1.4 there are a set of additional constraints on the model. The model is only valid during a gear change when the clutch is disengaged and no gear is engaged in the gearbox. Consequently no torque is acting on the layshaft from the engine or from the main shaft. The model is chosen to be valid only during this state since the case of engaging a gear only can occur when the clutch and gears are disengaged.

The model is not valid for gear changes where the splitter gear also changes gear. This is due to the dynamics that the engagement of a splitter gear has on the observed states of the layshaft. Since the selected splitter gear has an effect on the moment of inertia, the selected splitter gear is taken into account. This is further described in section 4.2.3

4.2 Overview

The model consists of some of the components from the gearbox. The components that are modeled are the layshaft, the disc brake and the valve controlling the disc brake. Below are the physics and mathematical models that are used in the models presented.

4.2.1 Valves and pressure

The valves controlling the disc brake and actuators that engage gears have the same basic function. The difference in how they operate is that the valve controlling the disc brake affects the system dynamically based on the duration of the input signal. The valves controlling the coupling sleeves have a more constant effect on the system. When a signal is sent from the control system to the valve there is an approximately set time until the coupling sleeve reaches the gear to be engaged. Therefore the valves and actuators that control the coupling sleeve is regarded as a pure time delay.

The pressure p_{valve} and its derivative $\dot{p}_{valve}$ after the valve controlling the disc brake is regarded as a first order system with a time constant and gain. The model of this system is shown in equation (4.1). Since the sensors measuring the pressure in the system measure the absolute pressure the atmosphere pressure p_0 is subtracted in the equation. The gain K_{valve} is known as it is the positive gauge pressure in the tanks in the pneumatic system. The pressure in these might fluctuate some during operations in the valves but in this thesis the pressure is set to be constant from the supply. The time constant τ_{valve} for the pressure is determined using a step response of the component. This is the time from the first reaction in pressure for it to reach 63 % of its final value. $t_{delay,valve}$ is the delay from when an activation signal is sent to the valve until there is a first change in pressure.

$$\dot{p}_{valve}(t) = -\frac{1}{\tau_{valve}}(p_{valve}(t) - p_0) + \frac{K_{valve}}{\tau_{valve}} u_{brake}(t - t_{delay,valve}) \tag{4.1}$$

4.2.2 Disc brake

A general equation often used to model the maximum braking torque from a disc brake T_{db} as a function of the applied pressure is presented in equation (4.2). The friction constant μ, the number of discs N, pressure area A_{db} and outer radius r_o and inner radius r_i of braking pads can all be lumped together to a constant C_{db} when the maximum braking torque $\tau_{db,max}$ for the maximum applied pressure $p_{valve,max}$ is known. Therefore the reduced equation becomes (4.3) and C_{db} is calculated through (4.4). [4]

$$T_{db} = \mu N p_{valve} A_{db}(r_o + r_i) \tag{4.2}$$

$$T_{db} = C_{db} p_{valve} \tag{4.3}$$

$$C_{db} = \frac{T_{db,max}}{p_{valve,max}} \tag{4.4}$$

4.2.3 Layshaft angle and angular velocity

The layshaft angular velocity in the system is modelled by means of Newton's second law for rotating systems as in equation (4.5). J is the moment of inertia for the object, $\ddot{\theta}$ is the angular acceleration and T are the torques acting on the object.

$$J\ddot{\theta} = \Sigma T \tag{4.5}$$

The torques are the torque from the disc brake T_{db} and the viscous friction from the input shaft and layshaft. These are the only torques because of the limitations in the model described in section 4.2. The moment of inertia is the combined moment of inertia for the input shaft and layshaft. The ratio of the splitter gears affects the friction and moment of inertia on the layshaft as shown in equations (4.6) - (4.10) which are based on the objects in figure 4.1. Equations (4.6) and (4.7) refers to the torque on objects. Along with the knowledge about the ratio in equation (4.8) and the equal forces in equation (4.9), these yield the final equation (4.10) which is written is the same way as equation (4.5). $\dot{\theta}$ refers to angular velocity, J is the object's moment of inertia, c is the rotational friction constant, F are the forces that the shafts apply on each other in the contact point between the cogs and T_{db} is the braking torque from the disc brake.

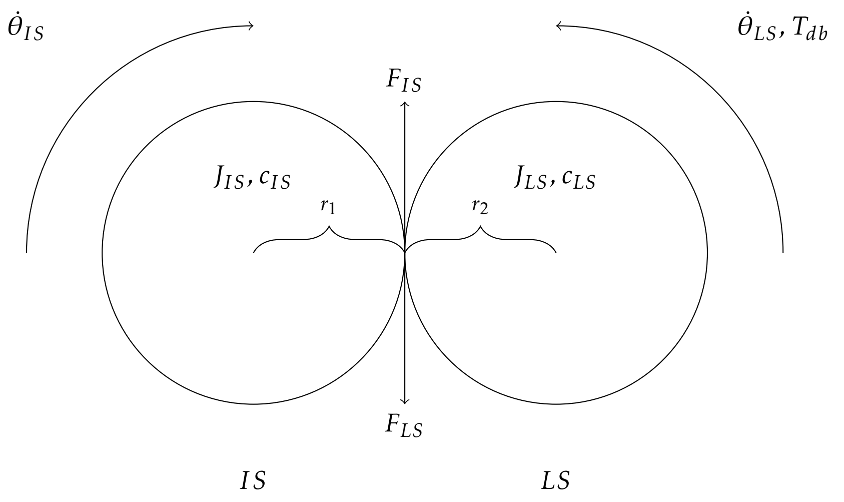


Figure 4.1: *Model for how the friction and moment of inertia affects another axis based on a ratio* $\frac{r_1}{r_2}$.

$$-F_{IS}r_1 - c_{IS}\dot{\theta}_{IS} = J_{IS}\ddot{\theta}_{IS} \tag{4.6}$$

$$T_{db} + F_{LS}r_2 - c_{LS}\dot{\theta}_{LS} = J_{LS}\ddot{\theta}_{LS} \tag{4.7}$$

$$\dot{\theta}_{IS} = \frac{r_2}{r_1}\dot{\theta}_{LS} \tag{4.8}$$

$$F_{IS} = F_{LS} \tag{4.9}$$

$$\ddot{\theta}_{LS}\left(J_{LS} + \frac{r_2^2}{r_1^2}J_{IS}\right) = -\dot{\theta}_{LS}\left(c_{LS} + \frac{r_2^2}{r_1^2}c_{IS}\right) + T_{db} \tag{4.10}$$

The friction from the input shaft's rotation depends on the ratio of the splitter gear but there is no data available for the individual frictions. Due to this the frictions are added together to a single parameter as in equation (4.11).

$$c_{LS} + \frac{r_2^2}{r_1^2}c_{IS} = c_{tot} \tag{4.11}$$

The ratio between the radius in equation (4.10) is the same as the ratio i_{split} on the splitter gear between the input shaft and the layshaft as shown in equation (4.12).

$$\frac{r_2^2}{r_1^2} = \frac{1}{i_{split}^2} \tag{4.12}$$

This is used in equation 4.10 which yields equation (4.13) as the relation between the angular acceleration $\ddot{\theta}_{LS}$, the angular velocity $\dot{\theta}_{LS}$ and the braking torque from the disc brake T_{db}. The braking torque result in a negative torque and thus slows down the angular velocity. The sign is (+) since the equation could work with a torque that speeds up the velocity.

$$\ddot{\theta}_{LS} = -\frac{c_{tot}}{J_{LS} + \frac{J_{IS}}{i_{split}^2}}\dot{\theta}_{LS} + \frac{T_{db}}{J_{LS} + \frac{J_{IS}}{i_{split}^2}} \tag{4.13}$$

The relation between the angular velocity $\dot{\theta}_{LS}$ and the angle θ_{LS} is self-explanatory and used as shown in equation (4.14) in the state-space model.

$$\dot{\theta}_{LS} = \frac{d}{dt}\theta_{LS} \tag{4.14}$$

4.3 State-space model

The equations from previous sections can now be used to form the state-space model of the system. The states for the system is shown in equation (4.15a) and (4.15b) shows the input signal to the system. The state-space model is presented in equations (4.15c) - (4.15e).

$$x = \begin{bmatrix} \theta_{LS} \\ \dot{\theta}_{LS} \\ p_{valve} - p_0 \end{bmatrix} \tag{4.15a}$$

$$u = u_{brake}(t - t_{delay,valve}) \tag{4.15b}$$

$$\dot{x} = Ax + Bu \tag{4.15c}$$

$$A = \begin{pmatrix} 0 & 1 & 0 \\ 0 & -\frac{c_{tot}}{J_{LS}+\frac{J_{IS}}{i_{split}^2}} & \frac{C_{db}}{J_{LS}+\frac{J_{IS}}{i_{split}^2}} \\ 0 & 0 & -\frac{1}{\tau_{valve}} \end{pmatrix} \tag{4.15d}$$

$$B = \begin{pmatrix} 0 \\ 0 \\ \frac{K_{valve}}{\tau_{valve}} \end{pmatrix} \tag{4.15e}$$

4.4 Main shaft

As mentioned in the limitations in section 1.4, the angular velocity of the main shaft in the gearbox is considered to be constant during a gear shift. Therefore, equation (4.16) describes the modeled angular velocity of the main shaft. K_{MS} is a constant value that is initiated at the start of every gear shift.

$$\dot{\theta}_{MS} = K_{MS} \tag{4.16}$$

4.5 Angle sensors

During the work with modeling and simulating the system there is no data available from the angle sensors described in section 2.5. To simulate the sensor signals with arrays of timestamps the system is initially simulated with data from a test where the existing control system estimates the angular velocity of the layshaft. The estimated angle during each gear shift is then used to create a sinusoidal wave with the same output as the Hall sensor would have. This is done by multiplying the angle with the number of cogs that passes the sensor each rotation. The sine value of this angle will have the same frequency as the signal from a Hall sensor on the shaft as shown in equation (4.17).

$$u = \sin\left(n_{cogs} \cdot \theta\right) \tag{4.17}$$

By updating the value with the same frequency as the real Hall sensors operate with and reading the time whenever the value value becomes for example $u > x$, timestamps can be written in an array like the one existing in the real system. The array is then read with the same frequency as the control system work with. The initial value θ_{init} will change when the timestamps are written. As the main purpose of the model is to create timestamps and the initial angle of the cogwheel at start-up is unknown, the initial value is set to $\theta_{init} = 0$.

5

Control Strategy

This chapter explains the theory behind and the implementation of the control strategy for the gear engage in the studied gearbox. It contains sensor signal filtering and processing and prediction of the system over the time horizon from the delay t_{delay} from the actuators controlling the coupling sleeves. The different approaches to prediction is presented in this chapter and their result is presented in chapter 6.

5.1 Objective

The objective of the control strategy is to use the available sensor signals to control the gear change such that specific angle differences between the gear to be engaged and the coupling sleeve is obtained. This is to avoid prolonged gear engagements as described in section 1.3. The engagements is to be performed when the layshaft is within the RPM-difference described in chapter 2. This is when the control strategy is intended to be used. Previously the gear has been engaged as soon as the RPM-difference is achieved.

To accomplish this, the current states of the observed system are first estimated as described in section 4.3. These states are then used to try to estimate the system's behaviour during the time it would take the actuator to engage a new gear as described in section 2.4. The theory behind the state observer is presented in section 5.2 and theory regarding the prediction methods used is presented in section 5.4. The prediction methods are used to predict the observed states over the gear actuator's delay t_{delay}.

5.2 State-space observer

To be able to control the system as intended, some sort of observer of the systems states needs to be implemented and the design of the observer can use the state-space model as a starting point. As mentioned in chapter 2, the angle sensors return arrays with timestamps related to different angles on the shafts. These signals contains some sort of noise from the measurement. To obtain a better estimation of the true states of the system, a filter can be used to decrease the effect of noise on the signals. With knowledge about the system and the state-space model in section 4.3 there are different methods for designing an observer. The chosen method in this thesis is to use a Kalman filter since it is one of the most well-documented approaches for estimating states in a linear system.

5.2.1 Kalman filter

A Kalman filter is an estimator that combines knowledge about the system and input from sensors to calculate an estimate of the systems states. It computes the optimal observer gain that minimize the effect from measurement and process noise given that the noise is uncorrelated and additive zero-mean white noise [7].

The process that a Kalman filter uses can be split into two parts; *predict* and *update*. An example system is presented below in equation (5.1). x_k, u_k, w_k, y_k, and v_k are the states, input signals, process noise, output signals and measurement noise respectively for the time step k. w_k and v_k are as mentioned above assumed to be uncorrelated and additive zero-mean white noise with covariances Q_k and R_k. If Q_k and R_k are unknown they can be viewed as filter parameters which can be set to give the filter the wanted characteristics.

$$\begin{cases} \dot{x}_k &= A_k x_k + B_k u_k + w_k \\ y_k &= C_k x_k + v_k \end{cases} \tag{5.1}$$

First the *predicted* states $\hat{x}$ and the estimated state error covariance matrix P for the next time step k are calculated based on the last time step $k-1$. These have to be initialized for the first time step i.e x_0 and P_0. P_0 and x_0 can be regarded as a design parameter. For time step k the equations are as shown below in equations (5.2)-(5.3).

Prediction

$$\hat{x}_{k|k-1} = A_k \hat{x}_{k-1|k-1} + B_k u_k \tag{5.2}$$

$$P_{k|k-1} = A_k P_{k-1|k-1} A_k^T + Q_k \tag{5.3}$$

The next step is to *update* the states by taking a new measurement y_k and from this calculate a new state $\hat{x}_{k|k}$ based on the current time step. This is made by the

Kalman gain K which can be calculated by equation (5.5). The estimated states are then updated with this Kalman gain in equation (5.6) by the use of $\tilde{y}_k$ from equation (5.4). Finally the error covariance matrix P is updated in equation (5.7).

Update

$$\tilde{y}_k = y_k - C_k \hat{x}_{k|k-1} \tag{5.4}$$

$$K_k = P_k C_k^T \left(C_k P_{k|k-1} C_k^T + R_k \right)^{-1} \tag{5.5}$$

$$\hat{x}_{k|k} = \hat{x}_{k|k-1} + K_k \tilde{y}_k \tag{5.6}$$

$$P_{k|k} = (I - K_k C_k) P_{k|k-1} \tag{5.7}$$

These equations are used along with a discretized version of the models from chapter 4 to be able to estimate the states of the system.

5.3 Event-based input and implementation of Kalman filter

To handle the sensor signal's event-based input described in section 2.5, a version of a Kalman filter is created. As the estimation should work with the same frequency as the rest of the control system the filter is made to work in a 10 ms loop. The filter reads the latest and previous timestamp arrays to determine the amount of new timestamps. It also needs the previous array to calculate the difference from the latest time stamp in the previous array and the first new one in the new array. The system's other signals to the state space model described in section 4.3 are also read as well as the previous error covariance matrix from equation 5.7.

The discrete-time state-space models matrices A_d and B_d are created as a function of the time step Δt beforehand in MATLAB with the command 'c2d'. The Q and R matrices are set to give the filter wanted characteristics where it follows the true value as close as is achievable. The Q matrix is also a function of the time step and therefore varies in each iteration as shown in equation (5.8).

$$Q_k = (w_k \Delta t)(w_k^T \Delta t) \tag{5.8}$$

The filter steps through the new time stamps and for each time stamp the new time step is used to calculate the new $A_d(\Delta t)$ and $B_d(\Delta t)$ matrices for the filter. The filter also contains an internal counter k_{count} to keep track of how many timestamps has been registered in total. This is to know the angle that each time stamp relates to. This generates the measured value y by equation (5.9). θ_{Hall} is the angle between each cog passes the Hall sensor.

$$y_{k_{count}} = k_{count} \cdot \theta_{Hall} \tag{5.9}$$

The following step is to predict and update the filter as described in subsection 5.2.1. The current states can be estimated through taking the difference in the current time and last timestamp to update the Q, A_d and B_d matrices and repeat equation 5.2.

A way of comparing the estimated value, from the Kalman filter, to the calculated one, from the control system, is to calculate the RMS value from their difference from the true value. RMS is calculated through equation 5.10 and is a common way of calculating the standard deviation. The error x_n is the error in each time step and is calculated through equation 5.11 where x_{true} is the true state and x_{KALMAN} is the estimated state.

$$x_{RMS} = \sqrt{\frac{1}{N}\sum_{n=1}^{N} |x_n|^2} \tag{5.10}$$

$$x_n = x_{true} - x_{KALMAN} \tag{5.11}$$

5.4 Prediction

To achieve the goal of controlling a gear shift so that the contact between the coupling sleeve and the gear happens at a certain angle between the parts it is necessary to predict how the system acts. From the time where a decision is taken in the control system to engage a new gear there is a delay until the coupling sleeve reaches the cogwheel on the gear. This is due to pressure build-up in the cylinder that is connected to the valve controlling the coupling sleeve, time for moving the sleeve etc. This prediction is used to decide when to engage a gear.

The time t_{delay} it takes from the system sends out the signal to the actuator controlling the coupling sleeve to when the actuator reaches the gear needs to be estimated. This is to know the horizon of the needed prediction. This is done with available data from a test in a rig with the system where multiple engagements on each gear are performed. The measured delays are presented in chapter 6 and the mean time will be used in the predictions.

To predict the systems behaviour during this time delay t_{delay} there exists many ways of discretizing the model. The chosen methods are *Linear angular velocity*, *Euler prediction* and *Zero Order Hold prediction*. The *Linear angular velocity* and *Euler prediction* are simpler methods initially used to understand demands on the accuracy of the predictions. The *Zero Order Hold prediction* is based on the discrete state-space model's matrices A_d and B_d described in section 5.3.

5.4.1 Linear angular velocity

To initially gain an understanding of the system and how much the states vary over the time delay t_{delay} the simple discretization method of linear angular velocity is evaluated. When the conditions for engaging a gear is met regarding RPM-difference, disc brake not engaged etc. at time t_{engage} the last derivative of the angle velocity is kept constant for the prediction horizon t_{delay}. This is presented in equation (5.12).

$$\ddot{\theta}_{LS}(t) = \ddot{\theta}_{LS}(t_{engage}) = k\ ,\ t_{engage} \leq t \leq t_{engage} + t_{delay} \tag{5.12}$$

5.4.2 Euler prediction

A basic way of discretizing a model is the Euler method (or Euler Forward Method). This is a rather simple method which is shown in equations (5.13) and (5.14). Implemented in a state-space model the equation is used as in equation (5.15). Given a start value y_0 the equation steps forward and calculates the derivatives of the function at every step with step length h. The derivatives are multiplied with h and added to the previous value. The smaller the step length h the closer the final value will be to the real state but also the more calculations need to be performed. The error in the prediction therefore also becomes larger as the predicted horizon grows since each step will add more error.

$$\frac{dy}{dt} = f(t, y) \tag{5.13}$$

$$y(t+h) = y(t) + hf(t, y(t)) \tag{5.14}$$

$$x(t+h) = x(t) + h(Ax(t) + Bu(t)) \tag{5.15}$$

5.4.3 Zero Order Hold prediction

Another way of discretizing a continuous model is Zero Order Hold (ZOH). This method assumes that the input signal is constant during the step length of the discretization. For an example system shown in equation (5.16) the same ZOH-discretized system is calculated through equation (5.17). k is a known counter and h is the step length during which the system's states and input signals are constant. The definitions of Φ and Γ are presented in equation (5.18). [7]

$$\begin{cases} \dot{x} &= Ax + Bu \\ y &= Cx + Du \end{cases} \tag{5.16}$$

$$\begin{cases} x(kh+h) &= \Phi(h)x(kh) + \Gamma(h)u(kh) \\ y(kh+h) &= Cx(kh) + Du(kh) \end{cases} \tag{5.17}$$

$$\begin{cases} \Phi(h) & = e^{Ah} \\ \Gamma(h) & = \int_0^h e^{As} ds \; B \end{cases} \tag{5.18}$$

There are different ways of calculating Φ and Γ. In this thesis, the MATLAB command 'c2d' is used to calculate these matrices. The benefits of using ZOH prediction is that it is an exact calculation of the states and outputs of the model in every point kh in time given that the requirement of constant input signal during the time step is met.

6 Results

In this section the results from the thesis work is presented. Due to excessive amounts of test results from the validation of models and predictions, only a handful of the results are presented in these sections to summarize the result.

6.1 Models

The models' output signals are compared to measured signals from a real system. The models are run in a Simulink model with a variable time step and an automatic solver chosen by Simulink.

6.1.1 Valve and pressure

Presented below are graphs where the pressure in the cylinder between the valve and the disc brake is viewed over time. The sequence in figure 6.1 is during a gear shift to a higher gear when no change in splitter gear is performed.

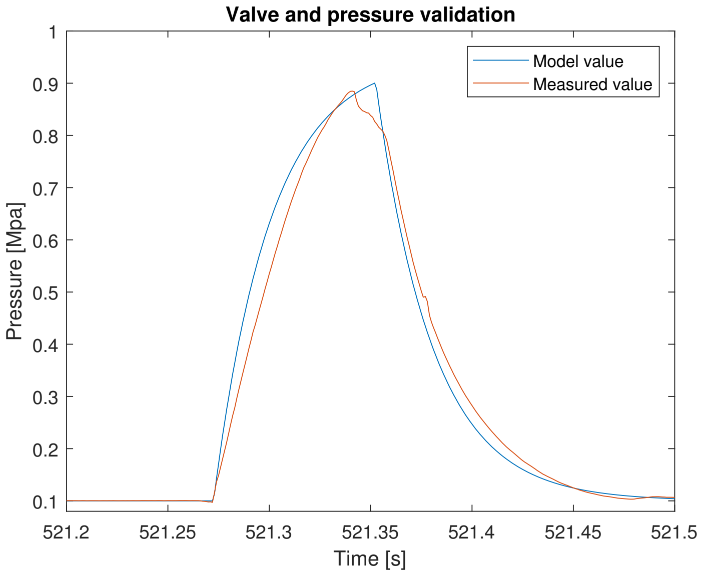


Figure 6.1: *Validation of the valve and pressure to disc brake. It is the absolute pressure which is shown. The model follows the measured value with slight variations.*

The modeled value varies some from the measured value during the build-up and decrease in pressure. At maximum pressure there is some fluctuations in the measured value which can be related to changes in flow in the system which the model is not designed for.

6.1.2 Braking torque

In the available data from the previous test, there was no measurement on the torque produced from the disc brake. The validation can be to verify that the produced torque is the same as the specified maximum torque $T_{db,max}$ as the part was modeled after when the maximal pressure to the brake is applied. In figure 6.2 a braking sequence is presented. The torque is negative since it is a braking torque on the layshaft. During normal operations the brake is disengaged and the layshaft has reached the targeted angular velocity before maximum torque is reached. This test is performed on the model with a step signal at $t = 0$ to the valve controlling the pressure to the brake. The initial delay is the delay from the model of the valve as described in section 4.2.2.

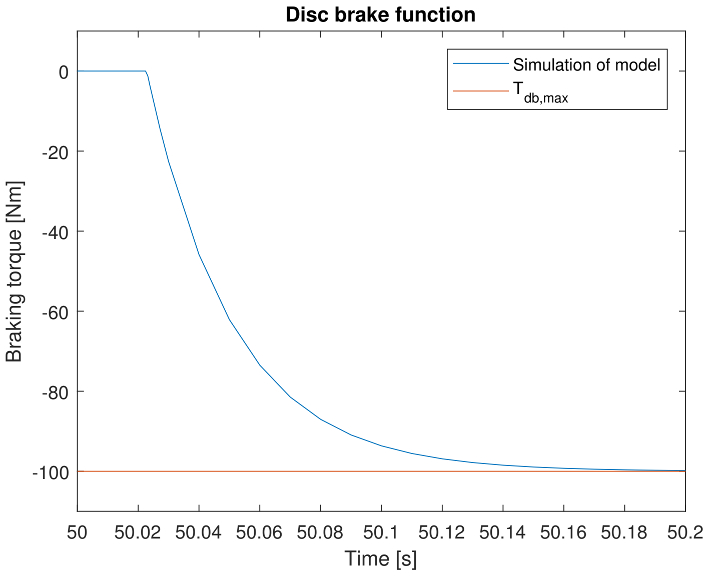


Figure 6.2: *The function of the braking torque from the disc brake. Since there is no data available from a real system only the value from the model is viewed.*

The measured disc brake has a specified max torque of 100 Nm when the maximum pressure from the system's pressure source is applied. This confirms that the model produces the intended torque.

6.1.3 Layshaft angular velocity

The model in Simulink is, as mentioned earlier, only valid when the clutch and main gears are disengaged. When simulating against the measured signal the model therefore is initialized each time the conditions are met. In figure 6.3 the model for the angular velocity from equation 4.13 is compared to the measured signal during a gear shift to a higher gear where the disc brake is engaged to slow down the velocity of the layshaft. The model is initialized when the real system is disengaged, the disc brake then engages and the velocity is slowed down. Shortly after the brake is disengaged in the real system the new gear is engaged and the layshaft then assumes the same velocity, with the gear's ratio, as the main shaft. The model over the velocity continues to slowly lose velocity due to the modeled friction in the system.

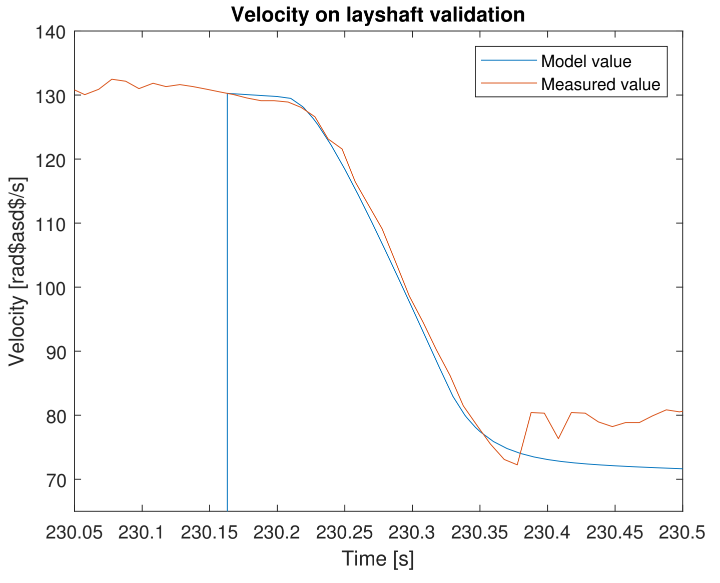


Figure 6.3: *Validation of the angular velocity compared to the measured value. The model follows the real systems based on the same signal as good as expected. When the real system engages the new gear and assumes the main shaft's velocity the model continues to loose velocity.*

6.2 Gear actuator time delay

To find the delay from the signal from the control system to engage a gear to the connection between the cogs on the gear to be engaged and the coupling sleeve, data from a previous test is used. By creating a histogram over the delay during all gear engages in the test, an estimated value is set to $t_{delay} = 45\ ms$. The histogram is presented in figure 6.4.

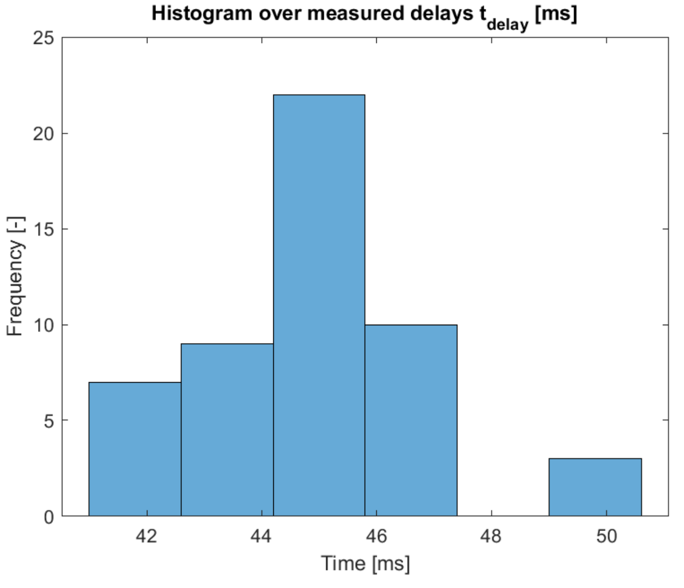


Figure 6.4: *Histogram over the delay t_{delay} for all gear engages during a test in a rig.*

There are some dynamics in the system that makes the delay vary with about ± 5 ms. During the evaluations on the prediction methods t_{delay} is assumed to be 45 ms. In the sensitivity analysis in section 6.4 the effect from variation in t_{delay} is evaluated.

6.3 Predictions

The different methods to predict the states, during the delay for the actuators controlling the coupling sleeves to reach the gears, are evaluated with simulations on the created model. The purpose of the predictions is to estimate the angle of the layshaft after the delay t_{delay} to be able to able to control the engagement angle with the coupling sleeve. When the model is valid, that is when the gearbox is disconnected from the clutch and gears, the predictions are made on angle and angular velocity, delayed and then compared to the model after it reaches the predicted time. The angle is also translated into the relation between the cog gaps on the gear to be engaged and the coupling sleeve's cog gaps. The test is initialized when the model is valid, in the same way as in the results for the models. The main shaft's angular velocity is initialized and assumed to be constant over the gear shift.

6.3.1 Linear angle velocity

The linear angle velocity method is described in section 5.4.1. The results are similar at every gear shift to a higher gear and an example is presented in figure 6.5. Since the predicted signal is delayed, the predicted signal's value for the model's initialized angular velocity is initated 45 *ms* later than the simulated signal.

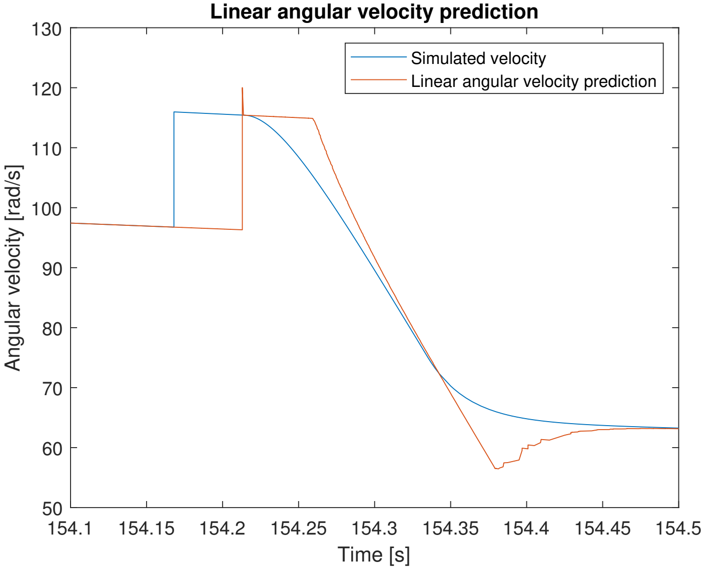


Figure 6.5: *Linear angular velocity prediction compared to the simulated value. Since a linear prediction is sensitive to changes in acceleration the prediction is delayed during engagement/disengagement of the disc brake.*

The method is sensitive to changes in acceleration and since the gear is to be engaged when the disc brake is disengaged and the acceleration changes due to the change in torque acting on the layshaft the prediction is slow to assume a similar value as the simulated velocity. This makes it ineffective at predicting the angular velocity when a gear is to be engaged.

6.3.2 Euler Prediction

The Euler prediction method is described in section 5.4.2. As for the linear prediction the prediction is similar at each gear shift and an example is presented in figure 6.6.

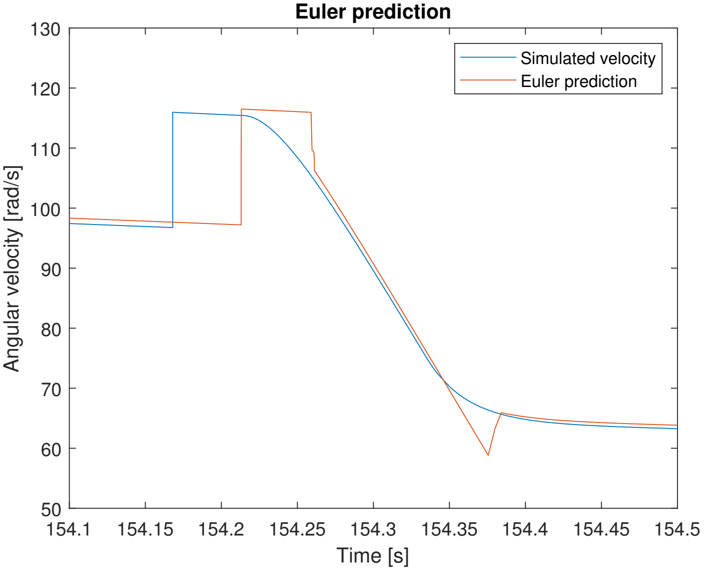


Figure 6.6: *Euler method prediction compared to the simulated value.*

Since the Euler method takes model states and input signal into account, the prediction has a delay of 45 *ms* when the input signal to the disc brake is changed. This is the reason for the delay in the prediction at the start and end of the brake's active time. This is acceptable since a gear is not engaged before the brake is disengaged and the prediction assumes a value closer to the simulated velocity. Should the splitter also change gear the prediction varies slightly after the brake is disengaged due to the change in moment of inertia as described in section 4.2.3.

6.3.3 ZOH prediction

The ZOH prediction method is described in section 5.4.3. In figure 6.7 the same gear shift as for the linear angular velocity and Euler method is presented for the ZOH prediction. As with the Euler method there is a delay during the engagement/disengagement of the disc brake.

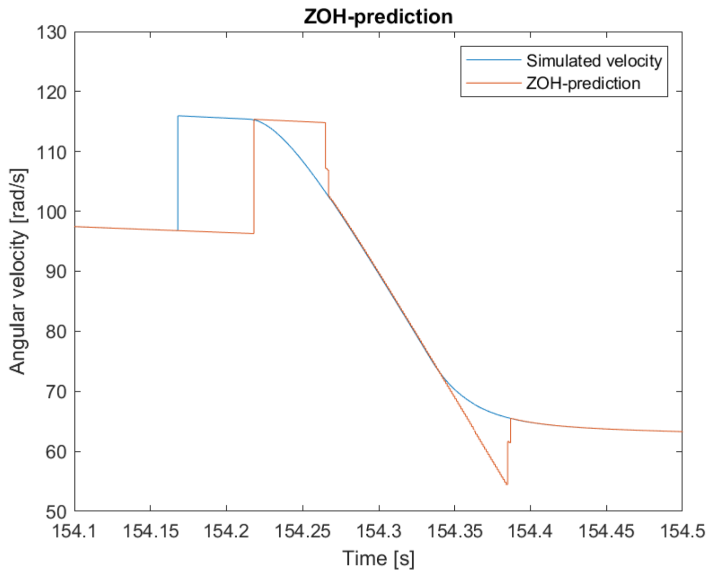


Figure 6.7: *ZOH-prediction compared to the simulated value.*

Because the ZOH prediction is based on the model that is simulated for the comparison, the prediction assumes exactly the same value as the simulation in each sampling point, except when the input signal change value, as expected.

6.4 Sensitivity analysis

To evaluate the effect that different error sources has on prediction over the delay t_{delay} a sensitivity analysis is performed. The delay is important since the angle and angular velocity need to be estimated during it to calculate when to activate the valve controlling the gear engagement and there is no way to control the engagement after the valve is activated. Different errors are introduced in the model the moment when the signal to the disc brake is set to zero after decelerating the layshaft to engage a higher gear. The effect of the errors over the delay t_{delay} is then evaluated.

6.4.1 Model parameters

In figure 6.8 a graph over the effect that different model parameter errors has on the angle after the delay t_{delay} is presented. To get a better view over how it affects the ability to aim for certain angles between the cogs and gaps on the gear to be engaged and the coupling sleeve, the resulting error in angle is presented as a percentage of a cog gap. The moment of inertia on the input shaft is analyzed with both splitter gear engaged. For errors introduced on the layshaft the high splitter gear is used to make the error bigger to analyze the worst case since the input shaft's moment of inertia will affect the model less.

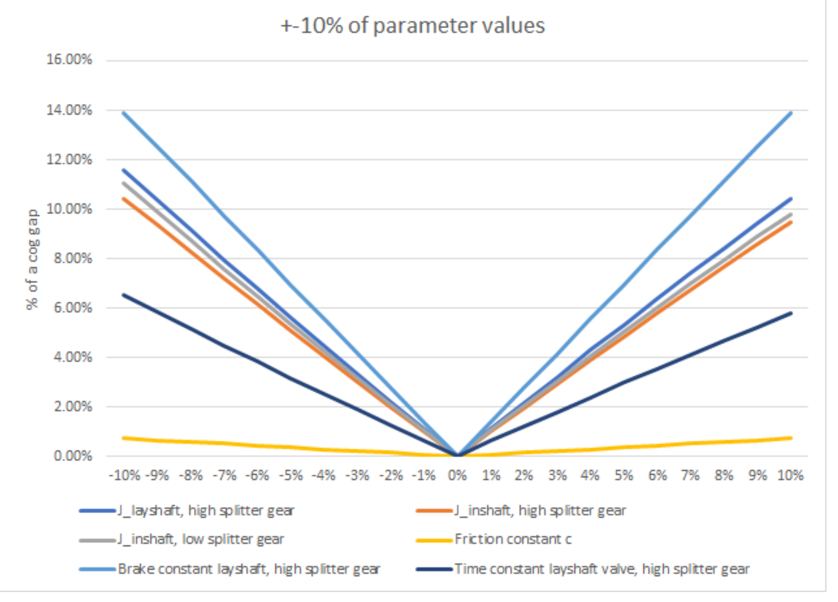


Figure 6.8: *The effect that errors in model parameters has on the angle of the layshaft after* t_{delay}.

The brake's friction constant μ on the disc brake has the biggest effect on the angle. The different moments of inertia has a slightly smaller effect. Notably, the friction constant c, which represents both frictions on the layshaft and input shaft added together, has the smallest effect on the angle.

6.4.2 Angle and angular velocity

The error on the angle and angular velocity on the layshaft scales differently in relation to the angle of the main shaft, based on which gear is to be engaged. In figure 6.9, the errors created by errors in initial θ_{LS} are presented based on which gear is to be engaged. Should the error grow larger than 100 % there is a very limited probability to control the engagement angle which is why the graph is capped at 100 %

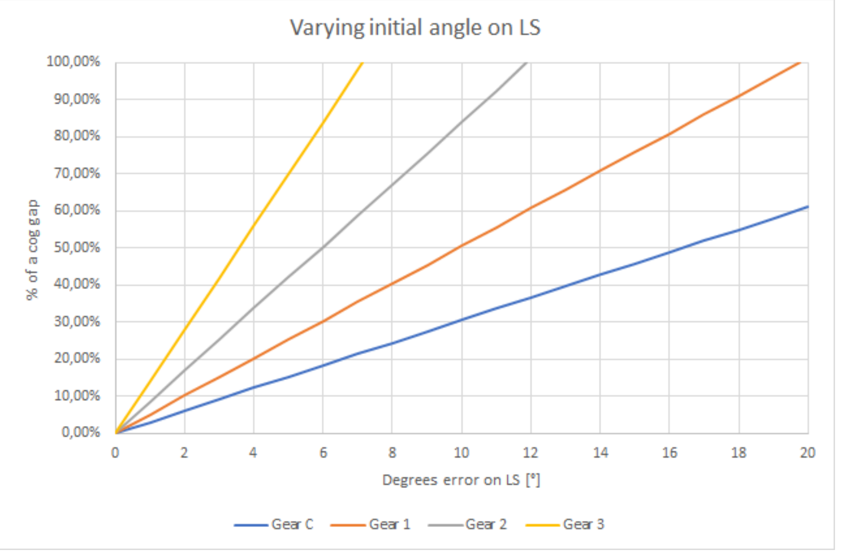


Figure 6.9: The effect that errors in initial θ_{LS} has on the angle of the layshaft after t_{delay} based on which gear is to be engaged. Negative angles would result in the same as the positive values, which is why only positive angles are presented.

In figure 6.10, the errors created by errors in initial $\dot{\theta}_{LS}$ are presented based on which gear is to engaged.

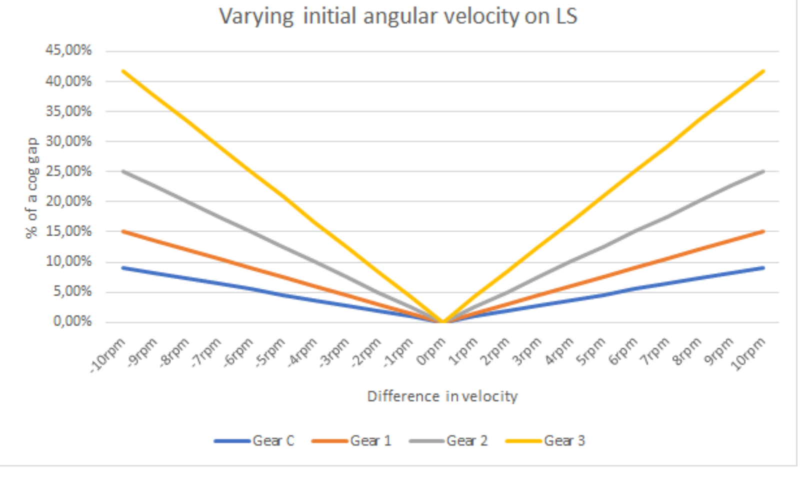


Figure 6.10: The effect that errors in initial $\dot{\theta}_{LS}$ has on the angle of the layshaft after t_{delay} based on which gear is to be engaged.

6.4.3 Actuator delay

In this section, variations in the actuator delay t_{delay} is evaluated. In figure 6.11 is a graph over varying errors in t_{delay} with different used RPM-differences between the layshaft and main shaft. The delay has a varying effect on the final angle based on which desired angular velocity difference that is used. It also depends on the gear's ratio. In the figure only the angle difference on the layshaft is presented.

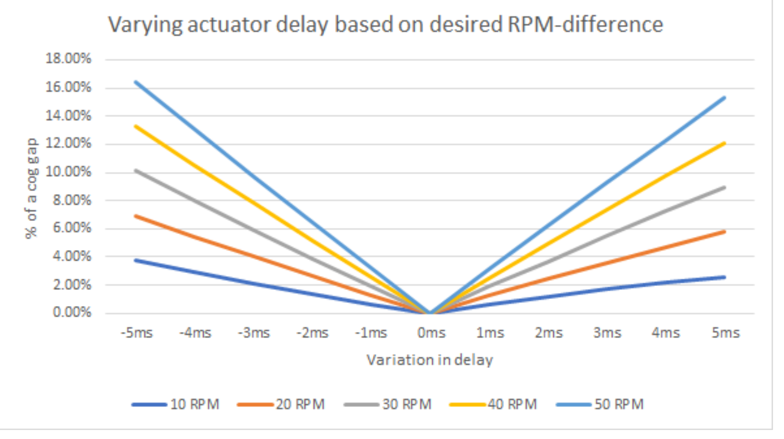


Figure 6.11: *The effect that errors in the delay t_{delay} of the gear engagement has on the angle of the layshaft after t_{delay}.*

As presented in section 6.2, the time from the signal to engage a gear to the time when the coupling sleeve reaches the cogwheel on the gear be engaged varies with about ± 5 *ms*. This produces a difference in angle on the layshaft around $4 - 16$ % of a cog gap based on which difference in angular velocity is used. The reason for the larger errors for the negative variations in delay is due to changing retardation on the layshaft during the engagement. Since the analysis is based on the instant the signal to the disc brake is set to zero, the deceleration will continuously decrease which leads to bigger differences in velocity for negative errors in t_{delay}.

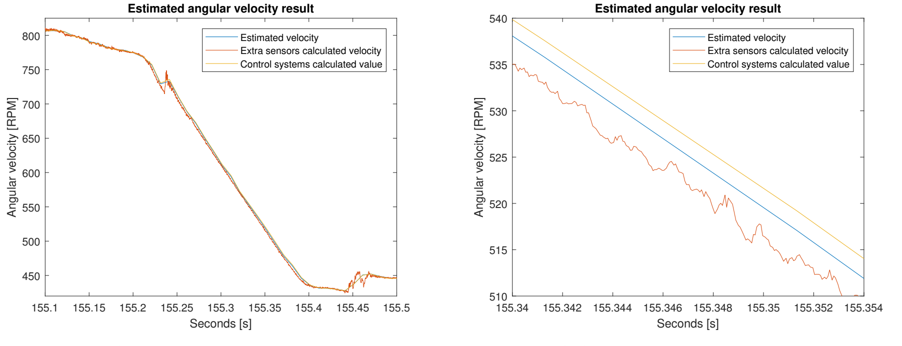
6.5 State estimation

In this section the results from the estimation of the layshafts angle and angular velocity are presented. The Kalman filter is implemented in MATLAB/Simulink as described section 5.3. Data is available from a test in a rig where the timestamp arrays are saved. The rig is also equipped with an extra sensor on the layshaft with much higher resolution and update rate than the Hall sensor which measures the angle and angular velocity. The update rate and resolution on the extra sensor makes these sensor values close to the actual angle and angular velocity.

The timestamps arrays are used to estimate the current states and the data from the extra sensor is used to validate the result of the estimation. The estimated angular velocity can be compared to the angular velocity from the extra sensor and the one calculated in the existing control system. The estimated angle can only be compared to the angle from the extra sensor.

6.5.1 Angular velocity on layshaft

In figure 6.12 a gear shift to a higher gear is performed and the layshaft's angular velocity is decelerated. Subfigures 6.12a and 6.12b both show the same deceleration. Figure 6.12b is a zoomed-in version of figure 6.12a to be able to see the difference between the estimated value and the control system's value. The angular velocity $\dot{\theta}_{LS}$ on the layshaft is plotted over time for the extra sensor, the estimated value and the control system's calculated value. The estimated value is initialized when the model is valid, that is when the clutch and gears are disengaged. The velocity is converted to RPM to easier compare the results to the sensitivity analysis.



(a) Full brake sequence *(b) Zoomed in sequence*

Figure 6.12: *Test result from estimation of angular velocity $\dot{\theta}_{LS}$ on the layshaft, converted to RPM. The estimated value of the velocity is slightly closer to the value from the extra sensor than the control systems calculated value.*

As mentioned in section 5.3, the RMS value of the error in estimated value is calculated to evaluate the Kalman filter. The error in estimated value is calculated with the value from the extra sensor as the ”true” value. The values used are during the gear shift when the model is valid. The result is presented in table 6.1 for two gear shifts. The table also contains RMS values if the control system is used instead of the estimated angular velocity. The estimated value has a slightly lower RMS value for both gear shifts compared to the angular velocity from the control system. This points to that the estimated value is slightly better at estimating the angular velocity than the current control system.

Table 6.1: *RMS-values for the control system and estimated value for velocities.*

	Signal	RMS value [RPM]
Gear shift 1	Control system	5.74
	Estimated value	4.34
Gear shift 2	Control system	5.95
	Estimated value	4.30

6.5.2 Angle on layshaft

The same test is performed for the angle on the layshaft as the angular velocity that is described in subsection 6.5.1. The same values for the Q and R matrices are used. In figure 6.13 the same gear shift as in figure 6.12 is shown but for the angle on the layshaft. Since there is no calculation of the angle in the existing control system, only the extra sensor's value and the estimated value are shown. The RMS values for the two gear shifts are presented in table 6.2 converted to degrees to more easily compare the results to the sensitivity analysis.

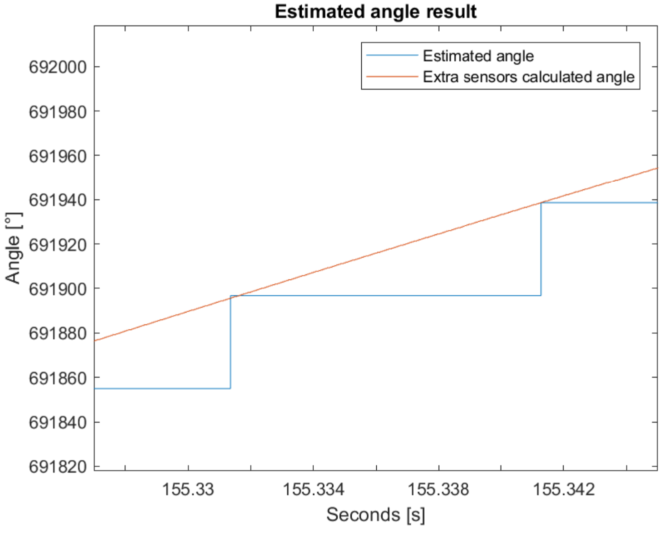


Figure 6.13: *Test result from estimation of angle θ_{LS} on the layshaft. The estimated value of the angle follows the measurement from the extra sensor closely.*

Table 6.2: *RMS values for the estimated value converted to degrees.*

	Signal	**RMS value [°]**
Gear shift 1	Estimated value	0.5147
Gear shift 2	Estimated value	0.5464

7 Discussion

In this chapter the result from chapter 6 is discussed. The result is evaluated and the thesis goals are assessed if they are met. Should they not have been met, the reason is discussed and what could have been done differently.

7.1 Models

The models created for the system have all met the expected results. There are slight variations from the real system which is considered to be because of dynamics in the system that not could be modeled during the thesis work, as can be seen in figures 6.1 and 6.3. As an example, variations in pressure from the valve controlling the pressure and flow to the disc brake could be based on usage of other components connected to the same pressure source. Such components have not been taken into regard and could lead to variations between the model and the real system.

The disc brake's friction constant μ is set to a constant value which could lead to some variations from the real system. Friction constants are sometimes modeled with regard to the temperature since it is temperature dependent [3]. A constant value of the friction was thought to be enough during the modeling.

As can be seen in the sensitivity analysis in section 6.4.1, the friction constant c does not effect the angle after t_{delay} to a big extent for errors in c that are less than 10 %. Other variations to the angular velocity could come from imperfect components in the real system where friction could arise from mechanical wearing. These are considered to be neglectable and the created model is working sufficiently to the expectations.

7.2 Predictions

The created predictions functioned as one might expect. The easiest approach of linear angular velocity was initially tested since it is the simplest of the predictions to create. However, the result presented in 6.3.1 shows that the method is not suitable for estimating the velocity after the disengagement of the disc brake and therefore is a poor choice of method for this thesis' objective since this is when the gear is to be engaged.

The Euler method uses the model and therefore is closer to the simulated value. With a small enough time step in the calculation the result would be closer but demand unnecessary calculation. The ZOH prediction also uses the model to calculate the predicted states of the system and needs fewer calculations. Since it is an exact representation of the model at each time step it is a better method of predicting the system's states. The only drawback would be that the discrete matrices A_d and B_d would need to be calculated for each prediction horizon t_{pred} that the states' value is desired to be predicted for which can be difficult to do live in a real system. Depending on the frequency of which the states and predictions can be calculated in the control system and the frequency of which the system can send out a signal to engage the new gear, there might be need for different prediction horizons t_{pred}. Should A_d and B_d be calculated beforehand they need to be stored in the system which could take up excessive storage in the control system.

7.3 Sensitivity analysis

The results from the sensitivity analysis on the model parameters justifies the assumption to regard the friction constant on the input shaft and layshaft as one since errors smaller than 10 % result in angle errors smaller than 1 % after t_{delay}. As expected an error on the moment of inertia of the input shaft has a larger impact on the angle with the lower splitter gear engaged since it will scale up the error based on equation 4.13.

Figures 6.9 and 6.10 shows the faults created by errors in estimation of θ_{LS} and $\dot{\theta}_{LS}$. They are also compared for which gear that is to be engaged which shows the difference in how the faults scale to different gears. It suggest that the task of engaging gears at certain angles between the gear and coupling sleeve should be easier to implement on gear changes to the crawl gear and first gear.

7.4 State estimation

The estimated values that are compared to the test with an extra sensor on the layshaft in section 6.5, in figures 6.12 and 6.13, indicate that the created Kalman filter from section 5.2.1 estimates values closer to the "true" values from the extra sensor compared to the control system. This is after some tweaking in the Q and R matrices to get closer values. The angular velocity difference compared to the true value in figure 6.12 along with the sensitivity test suggests that the resulting error in angle on the layshaft after t_{delay} would be around $4 - 18$ % of a cog gap based on which gear that is to be engaged. The estimated angle is very close to the true value in figure 6.13. Compared to the sensitivity test it indicates that the the resulting error in angle on the layshaft after t_{delay} would be around $1.5 - 7$ % of a cog gap based on which gear that is to be engaged.

Due to limited time at the end of the thesis work, this subject is not fully explored and more work could be spent on determining the validity of the estimations. Ideally more data would have been compared and additional sensor signals could have been used. Should more time have been spent on implementation of the Kalman filter a better estimation of the states could probably have been achieved.

8 Conclusion

8.1 Conclusion

The thesis has performed an investigation regarding the possibility to control the engagement of a gear in an AMT gearbox so that certain angles between cogs on the gear to be engaged and coupling sleeve are achieved. In this work a model over the studied system has been created based on knowledge about the system and mathematical models. The model contains sub-models for the angle and angular velocity on the layshaft in the gearbox, the torque from the disc brake located on the layshaft and the valve and pressure that engages the disc brake. The model produces simulated values of the system's states that are satisfactorily close to the real system's values.

Three different methods for predicting the system's states during the time delay t_{delay} are evaluated. These are *Linear angular velocity*, *Euler prediction* and *Zero Order Hold prediction*. The ZOH prediction produces a satisfactory prediction over the delay. A Kalman filter is designed based on the created model over the system and implemented to estimate the system's states closer to the real value than today's estimations. The studied states are the angle and angular velocity on the layshaft, the braking torque from the disc brake and the pressure supplied to the disc brake by the valve. Results indicate that it is possible to estimate the states close enough to their true values so that a control strategy could be implemented with regard to the result from the sensitivity analysis.

8.2 Future work

To get a better estimate on the angular velocity on the layshaft and main shaft, the Kalman filter could be updated to include sensor signals from the Hall sensors on the input shaft, main shaft and propeller shaft and use sensor fusion. Since the ratio between the gears are known, the sensor on the input shaft could be used to improve the layshaft's angle and angular velocity. Even though the main shaft is assumed to have constant velocity, the value could be better estimated the same way as the layshaft with known ratios on the range gear that connects the main shaft and propeller shaft.

To implement the control strategy on a real system, the relative angles between the gears and coupling sleeves needs to be initialized during each first engagement of each gear. When the system has started up, the Hall sensor signals will just contain information regarding when cogs pass the sensors, not the current angle. When a gear has been engaged for the first time after a start up, the angle needs to be stored for the main shaft and layshaft for next engagement to be able to control the connection between cogs and gaps between the gears and coupling sleeves. The strategy could then be implemented in a real system to analyze the strategy further.

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