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Modeling and Estimation for Dry Clutch Control

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“*A mathematical model does not have to be exact; it just has to be close enough to provide better results than can be obtained by common sense.*”

— Herbert A. Simon

Abstract

Increasing demands on comfort, performance, and fuel efficiency in vehicles lead to more complex transmission solutions. One such solution is the Automated Manual Transmission (AMT). It works just like an ordinary manual transmission but the clutch and gear selection are computer controlled. In this way high efficiency can be accomplished with increased comfort and performance. To be able to control and fully utilize an AMT it is of great importance to have knowledge about how torque is transmitted in the clutch. The transmitted torque in a slipping dry clutch is therefore studied in experiments with a heavy duty truck (HDT). It is shown that material expansion with temperature can explain torque variations up to 700 Nm for the same clutch actuator position. A dynamic clutch temperature model that can describe the torque variations is developed. The dynamic model is validated in experiments, and shown to reduce the error in transmitted torque from 7 % to 3 % of the maximum engine torque compared to a static model.

The clutch model is extended with lock-up/break-a-part dynamics and an extra state describing wear. The former is done using a state machine and the latter using a slow random walk for a parameter corresponding to the clutch disc thickness. An observability analysis shows that the augmented model is fully or partially observable depending on the mode of operation. In particular, by measuring the actuator position the temperature states are observable, both during slipping of the clutch and when it is fully closed. An Extended Kalman Filter (EKF) was developed and evaluated on measurement data. The estimated states converged from poor initial values, enabling prediction of the translation of the torque transmissibility curve. The computational complexity of the EKF is low and it is thus suitable for real-time applications.

The clutch model is also integrated into a driveline model capable of capturing vehicle shuffle (longitudinal speed oscillations). Parameters are estimated to fit an HDT and the complete model shows good agreement with data. It is used to show that the effect of thermal expansion, even for moderate temperatures, is significant in launch control applications.

An alternative use of the driveline model is also investigated here. It is found that the amplitude discretization in production road-slope sensors can excite vehicle shuffle dynamics in the model, which is not present in the real vehicle. To overcome this problem road-slope information is analyzed and it is shown that a third-order butterworth low-pass filter can attenuate the vehicle shuffle, while the shape of the road profile is maintained.

All experiments in the thesis are performed using production HDTs only, i.e. production sensors only. Since all modeling, parameter estimation, observer design and validation are performed with production sensors it is straight forward to implement the results in a production HDT following the presented methodology.

Acknowledgments

I like to thank my beloved girlfriend Elin "White Darling" Julin for always supporting me and cheering me on in my work, you sure made it easier to reach this far. I also have to thank my supervisor Lars "Lasse" Eriksson for reaching this far. I would not even have started without your motivational skills. More thanks go to Karl Redbrandt, my contact at Scania, a busy man that still managed to give plenty of support, not only with measurements but also with ideas and thoughts. More people at Scania that I like to thank are my former contact person, Jörgen Hansson and Kristian Hellgren for valuable clutch information.

Further thanks go to my co-workers at Vehicular Systems for contributing to a pleasant working atmosphere. Special thanks to many roommates, both current, Neda Nickmehr and Jörgen Albrektsson, as well as former, Carl Svärd, Peter Nyberg, and Erik Höckerdal. A special thank also goes to Kristoffer Lundahl for bringing more vehicular interest and vehicular discussions to Vehicular Systems.

A lot of friends and again, my Elin, are thanked for filling up my spare time with activities, which definitely is necessary when progress at work is slow. My family is thanked for creating an environment that nurtured a great science and engineering interest. There are many engineers in my family but, at the time of writing, no licentiates. A special note is made to my dad, Einar Myklebust, who have supported me when developing my practical vehicular skills. And I suppose also to my RX-7 "Black Darling" for constantly breaking down at a rate slightly higher than my "fixing rate". Last, and perhaps least, just to be on the safe side, thanks to anyone who feels forgotten in this acknowledgment!

Contents

Chapter 1

Introduction

In order to propel a vehicle the engine must be connected to the wheels somehow and different driveline solutions are available to create this connection. In a common rear-wheel-drive setup, see Figure 1.1, the tractive wheels (two or more) are connected to a drive shaft each. These drive shafts are, through a final drive (a differential), driven by a propeller shaft that is connected to the transmission. The transmission consists of two parts, the actual transmission part where torque and speed is changed by a gear ratio and a connection between the Internal Combustion Engine (ICE) and transmission that is capable of decoupling the speeds between the engine and transmission. Historically comfort was best accomplished with a classical Automatic Transmission (AT) and high efficiency with a Manual Transmission (MT).

An AT actuates different gear ratios through clutches and brakes that locks different parts of planetary gear sets. The coupling between engine and transmission is handled by a torque converter. These two technologies put together enable seamless gear shifts that can be controlled through simple hydraulics, the AT was put in production as early as 1939, (Nunney, 1998). The drawbacks are lower efficiency, mainly due to pumping of oil in the torque converter, and the increased complexity and cost, compared to an MT.

The MT is coupled to the engine via a clutch that is operated by the driver via the clutch pedal. A simple explanation of the clutch is that it consists of two rotating plates that can be pressed together. When pressed together friction will arise and transmit a torque between the plates, which acts to reduce the speed difference. See Chapter 2 for a more detailed explanation. Gear selection is realized using two shafts, both with a set of cog wheels, that mesh together. One pair of cog wheels, corresponding to a certain gear, can be engaged by a mechanical linkage connected to the gear lever. This type of transmission has a high efficiency and simple construction but requires the clutch to be disengaged during shifting (torque interrupt) and manual input from the driver. For more

Figure 1.1: A rear-wheel-drive driveline from a Scania HDT. The engine, transmission, propeller shaft, differential (final drive) and drive shafts can be seen.

details on possible transmission constructions see Newton et al. (1996) or Nunney (1998).

Increasing demands on comfort, performance, and fuel efficiency in vehicles lead to more complex transmission solutions. The Automated Manual Transmission (AMT) is one way to combine the best from two worlds. It has the same components and basic operation as an MT but the gear selection and clutch operation have been made automatic. This has become possible thanks to technological advances within actuators and computer controllers. The AMT has the benefits of the MT but without the need of driver attention. However it still has the drawback of torque interrupt during gear changes. Another option, capable of removing the torque interrupt, is the Dual Clutch Transmission (DCT). The DCT further improves comfort and performance with the drawbacks of increased complexity and cost. An important part in both an AMT and a DCT is clutch control, which has a profound effect on vehicle performance. A poorly controlled clutch can make starts, stops and gearshifts slow, rough on the hardware or uncomfortable. An example can be seen in Figure 1.2. In this case driveline oscillations, which cause discomfort, are induced by too rapid disengagement/engagement of the clutch. The driveline oscillations can be seen in both the speed graphs and in the acceleration graphs. This type of oscillations can be clearly felt by the driver and passengers.

To be able to control the clutch in a fast and comfortable manner, without causing excessive wear it is of importance to know the torque transmitted in the clutch with high precision. Moreover the clutch torque is of great interest if the clutch is to be integrated in a powertrain control scheme that is of, as common, torque based structure, (Eriksson and Nielsen, 2014). Models have come to play an important role in estimation and control of the transmitted torque, since torque sensors are expensive. Therefore modeling and observation of the clutch torque is in focus here. Since the purpose of the observer is improved clutch control, the model and observer are required to be light-weight enough to run in real-time. Previous experimental investigations of the clutch characteristics, (Velardocchia et al., 1999; Moon et al., 2004; Vasca et al., 2011) and (Deur et al., 2012), have been carried out in test rigs equipped with various sensors. Here the experimental platform is limited to production vehicles without additional sensors. On one hand this choice imposes certain difficulties in setting up useful experiments. On the other hand the resulting model and observer can be directly applied to the intended vehicle, in or after production.

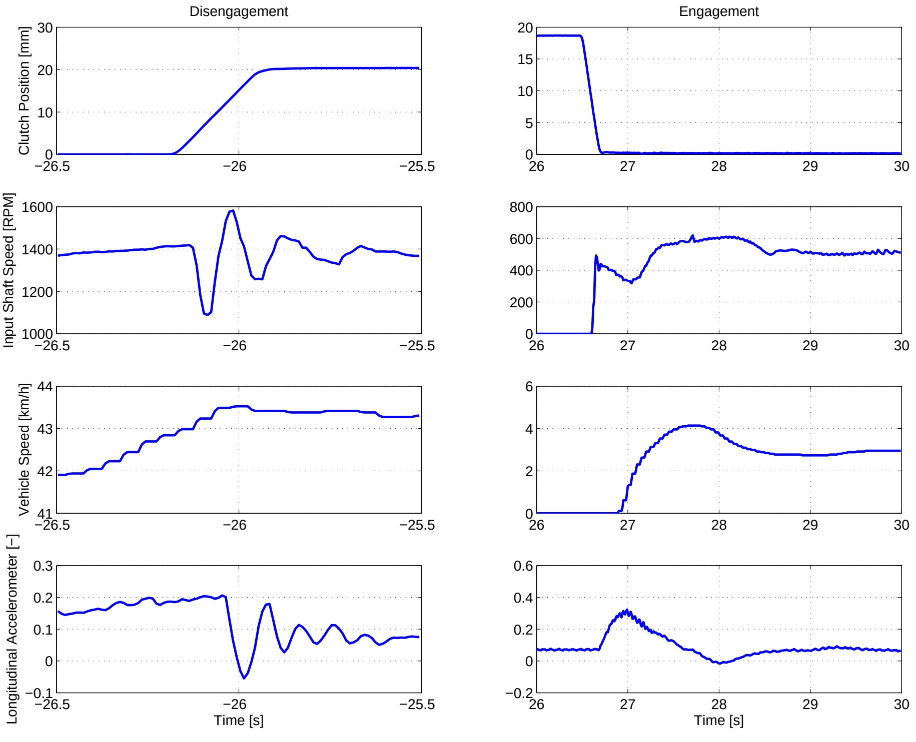

Figure 1.2: The left graphs show measurements of driveline oscillations due to (too) rapid disengagement of the clutch; a scenario possible during gear shifting. The right graphs shows the same measurements but in the case of rapid engagement from standstill, which could be the case if the driver wants a quick launch. In both cases the oscillations could clearly be felt by the driver and passenger.

1.1 Contributions

A set of experiments for determining significant effects in the clutch and their characteristics is proposed. Since the experimental platforms used here are production vehicles the methodology proposed can easily be applied to new platforms without the use of additional test equipment. Moreover the clutch is put in its true thermal environment, between a transmission and a warm combustion engine, which affects the temperature dynamics. Further contributions are found in the attached papers A-D. Their main contributions are summarized below.

Paper A

The main contribution of Paper A is a novel clutch model that includes the temperature dynamics and thermal effects on the transmitted torque during slipping. The model is developed using a method that utilizes production sensors only. The resulting model is simple enough to run in real time.

Paper B

Paper B integrates the clutch model of Paper A into the complete driveline model of Paper D and models clutch lock-up/break-a-part with a simple approach. The complete model is validated against data of HDT launches. The main contribution is the demonstration of the importance of considering the thermal dynamics during vehicle launch.

Paper C

Paper C extends the clutch model of Paper A with a wear parameter corresponding to thinning of the clutch disc. The main contributions are an observability analysis and observer design for the augmented model. The observability of the augmented model is found to be dependent on the mode of the system. An Extended Kalman Filter (EKF) that can observe the temperatures and the wear parameter is designed and tested on data from production vehicles.

Paper D

A driveline model is presented in Paper D and shown to be appropriate for vehicle longitudinal shuffle simulation. The main contribution is an investigation, using the model, of how and why a discretized slope signal needs to be filtered.

1.2 Publications

The following papers are included in the thesis.

Conference papers

- Andreas Myklebust and Lars Eriksson. Torque model with fast and slow temperature dynamics of a slipping dry clutch. Published in *2012 IEEE Vehicle Power and Propulsion Conference*. Seoul, South Korea, 2012. **(Paper A)**

- Andreas Myklebust and Lars Eriksson. The effect of thermal expansion in a dry clutch on launch control. Accepted for publication in *7th IFAC Symposium on Advances in Automotive Control*. Tokyo, Japan, 2013. **(Paper B)**

- Andreas Myklebust and Lars Eriksson. Road slope analysis and filtering for driveline shuffle simulation. Published in *2012 IFAC Workshop on Engine and Powertrain Control, Simulation and Modeling*. Rueil-Malmaison, France, 2012. **(Paper D)**

Submitted

- Andreas Myklebust and Lars Eriksson. Modeling, observability and estimation of thermal effects and aging on transmitted torque in a heavy duty truck with a dry clutch. Submitted to *IEEE/ASME Transactions on Mechatronics*. **(Paper C)**

Chapter 2

Introduction to Clutch and Driveline Modeling

A schematic of the dry clutch and actuator studied in this thesis is found in Figure 2.1. It has an electro-hydraulic actuator and the clutch construction is typical for a dry clutch. Double clutches have slightly altered constructions but the main principle is the same.

The electric motor rotates a worm gear that pushes into the hydraulic fluid, in an MT the clutch pedal would do this. When the opening to the reservoir is passed the hydraulic fluid pushes on a piston that through a lever pulls the throw-out bearing away from the clutch, the depicted clutch is hence called a pull-type clutch.

The throw-out bearing pulls on the fingers of the diaphragm (also called belleville, washer, slotted disc) spring. This spring is radially pivoted and angularly fixed to the clutch cover that is bolted to the flywheel. The bolting of the clutch cover pre-loads the diaphragm spring so that it exerts a force on the pressure (push) plate and thus clamps the clutch disc between the pressure plate and the flywheel. The pressure plate is also angularly fixed to the clutch cover and thereby flywheel. When the throw-out bearing is pulling the fingers of the diaphragm spring load is taken off the pressure plate. As the pressure plate exerts less clamp load, the cushion (flat) spring inside the clutch disc expands leading to a new equilibrium position for all of the clutch linkage. The connection between positions, spring characteristics and bearing force can be seen in Figure 2.2. Through this arrangement a certain actuator positions will correspond to a certain clamp loads, which facilitates control, especially in the manual case.

The clutch disc is angularly fixed to the transmission input shaft and can therefore rotate with a different speed than the engine and flywheel. When the clamp load becomes greater than zero a friction force will arise if there is a speed difference. This force will result in a torque around the crank and input shafts working to reduce the speed difference. With a larger normal force a larger

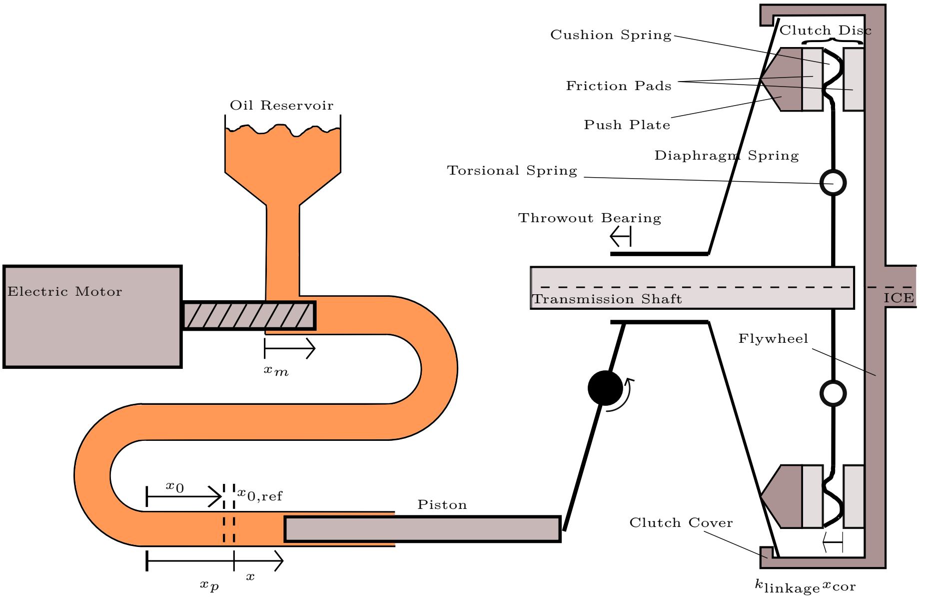

Figure 2.1: A schematic over the actuator and the dry single-plate pull-type clutch studied here. k_{linkage} is the combined ratio of all levers between the piston and the push plate. Sometimes the measurement x is used instead of x_p as a wear compensated piston position that is in the same range as x_m.

torque will be transmitted and if the speed difference gets reduced to zero the clutch will lock up, i.e. static friction will take place. When locked up the clutch acts as a solid unit and transmit the engine torque as long as it does not exceed the stiction torque. The engine torque is oscillative and in order to damp these oscillations there are torsional springs inside the clutch disc. For more detailed descriptions of how a clutch works see Mashadi and Crolla (2012); Newton et al. (1996); Dolcini et al. (2010) or Vasca et al. (2011).

2.1 Survey of Clutch and Driveline Models

When modeling for control purposes not every detail has to be captured. It is sufficient if significant dynamics are captured and preferable if the model complexity can be kept low, (Ljung, 1999). In the case of the clutch the transmitted torque is an important quantity to model. However the importance of clutch torque control becomes clearer if it is put into the wider context of driveline control. For advanced driveline control a driveline model is useful, either for simulation during development or for model based control. The model should capture the dynamics in the driveline, such as shunt and shuffle, that affects the driver (and passenger) comfort.

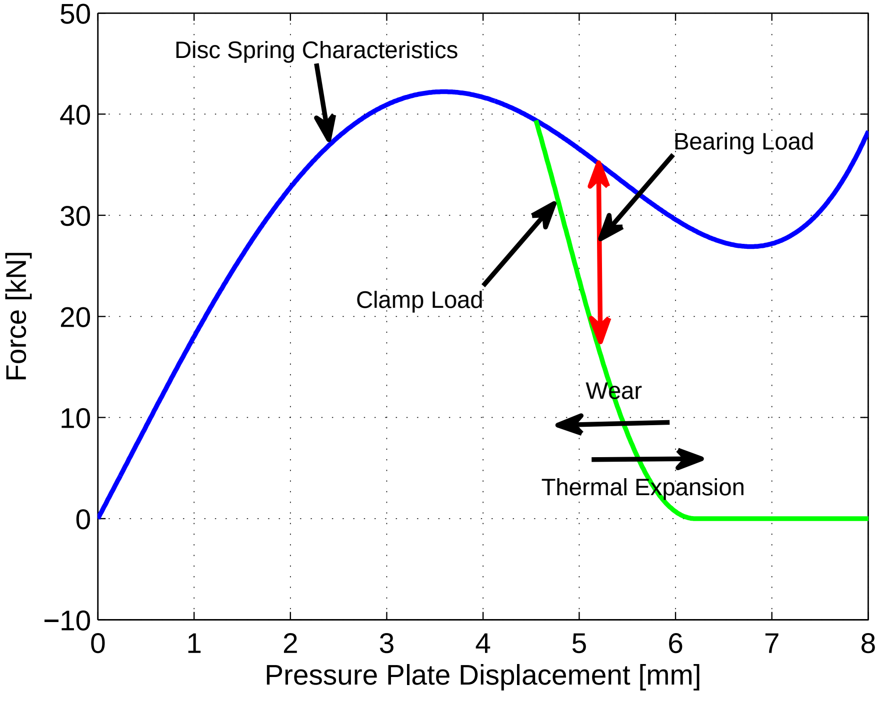

Figure 2.2: Illustration of how the two spring forces in the clutch and the trow-out bearing force interact. The clutch disc spring characteristics are the spring torque at the pivot point divided by the length of the lever between the pivot and the pressure plate. The throw-out bearing force is multiplied with the length of the lever between the bearing and the pivot and then divided with the length of the lever between the pivot and the pressure plate.

2.1.1 Clutch Models

In the literature a wide range of models have been proposed. The most simple models have a clutch torque that is assumed to be a controllable input, see for example Dolcini et al. (2008) or Garofalo et al. (2002). These models rely on the assumption that there is perfect knowledge of how the clutch behaves. More advanced models include submodels for slipping and sticking torques. For example a LuGre model is used in Dolcini et al. (2005) and a Karnopp model in Bataus et al. (2011). The former is a one-state model that captures stick-slip behavior, varying break-away force, Stribeck effect, and viscous friction. The latter includes a dead-zone around zero speed to ease the simulation of stick-slip behavior. The clutch torque during slipping is commonly modeled using a function with the following structure,

$$M_{\text{trans,k}} = \text{sgn}(\Delta\omega)\, n\, \mu\, R_e\, F_N \tag{2.1}$$

where $\Delta\omega$ is the clutch slip (speed), n the number of friction surfaces, μ the friction coefficient, R_e the effective radius and F_N the clamping (normal) force. In these models F_N is often either given as input or a static nonlinear function of clutch position, x, i.e. $F_N = F_N(x)$, see for example Vasca et al. (2011) or Glielmo and Vasca (2000). In particular Dolcini et al. (2010) mentions that a third-order polynomial is suitable to describe the connection between throwout-bearing position and clutch transmitted torque, mainly governed by the cushion spring characteristics. In Deur et al. (2012) μ is fitted to the following regression curve:

$$\begin{aligned}\mu = a_0 + a_1 \left(e^{-a_2\Delta\omega} - 1\right) T_b + (a_3 - a_4 \ln(\Delta\omega))\, T_b^2 \\ + (a_5 - a_6\Delta\omega)\, F_N + a_7 T_b F_N\end{aligned} \tag{2.2}$$

where T_b is the temperature of the clutch body and assumed to be an input signal. $a_0 - a_7$ are curve parameters. F_N is mapped against actuator position and T_b in Deur et al. (2012). More advanced FEM models of the clutch are also found in the literature. To name some Nam et al. (2000) investigates the stresses in the diaphragm spring, Sfarni et al. (2008) has studied how the clutch disc characteristics change with wear and Cappetti et al. (2012b) examines the temperature effect on the cushion spring. Especially temperature distributions are popular to investigate using FEM. Both in clutch parts, Abdullah and Schlattmann (2012a) examines temperature distributions in clutch discs with different grooves and Lee et al. (2007) investigates the temperature distribution in the pressure plate, and in the clutch as a whole, (Abdullah and Schlattmann, 2012b) and (Sun et al., 2013). These FEM models point out interesting effects but are of little use here as the computational complexity is too high.

In Velardocchia et al. (2000) and Wikdahl and Ågren (1999) simpler temperature models are established. However these two models do not include the effect of the temperature on $M_{\text{trans,k}}$. The models in Velardocchia et al. (2000) and Wikdahl and Ågren (1999) could be combined with the model presented in Deur et al. (2012), where the clutch temperature is an input, in order to give the clutch torque. Although that approach requires a map of the normal force as function of actuator position and temperature, whereas the approach in Paper A, (Myklebust and Eriksson, 2012b), is completely model based. Furthermore the clamping load and temperatures, which is not available in production vehicles, need to be measured during the parameter estimation procedure. In Paper A, (Myklebust and Eriksson, 2012b), only sensors available in production vehicles are used.

2.1.2 Driveline Models

When modeling for control purposes it is common in the literature to include one or more flexibilities. In Pettersson (1997) active shuffle damping and clutch-less gear-shift control is studied. There the model includes flexibilities in the clutch, propeller shaft and drive shafts. The clutch torsion springs are modeled both in a linear and a piece-wise linear fashion. Also Fredriksson and Egardt (2003) looks at clutch-less gear-shift control, but this time in a transmission without synchronizers. There only the drive shaft flexibility is modeled. Garofalo et al. (2002) and Dolcini et al. (2008) study optimal control of clutch engagement. For that the former models two flexibilities in the driveline, one before the transmission and one after. Whereas the latter models only the flexibility after the gearbox. A clutch-by-wire system is built in Moon et al. (2004), there a non-linear clutch and drive-shaft flexibilities are used. Clutch judder is examined in Crowther et al. (2004) using a driveline model with two flexibilities. A paper about AMT modeling in general, (Lucente et al., 2007), states that three flexibilities are useful, one in the engine, one before the transmission and one after.

Chapter 3

Experimental Observations and Model Structure

As seen in Chapter 2 there are different ways to model the clutch. In order to chose a suitable model structure an experimental platform and data is required. Here Scania Heavy Duty Trucks (HDTs) with standard production sensors are used. A variety of trucks have been used but the two most used are Ara and Ernfrid. Both are equipped with a 14-speed AMT and a 16.4 liter V8 capable of producing 3500 Nm. Ara weighs 105 tonnes and Ernfrid 21 tonnes. The clutch was described in Chapter 2. A list of the measurement signals used is found in Table 3.1 and their respective location in the driveline can be seen in Figure 3.1. Hence no exact torque measurement, is available. In order to get a measurement of the clutch torque the reported engine torque is used with added compensation for inertia effects of the engine and flywheel.

The main benefit of this setup, with only production sensors, is that the resulting model and observer is directly applicable on a production truck. Furthermore the procedure for experiments, model building, model validation, and

Table 3.1: A list over the different measurements used in the HDTs

Notation	Explanation
ω_e	Engine speed
ω_t	Transmission input-shaft speed
ω_p	Transmission output-shaft speed
ω_w	Tractive-wheel speed
v	Vehicle speed, actually speed of the free rolling (front) wheels
T_{coolant}	Engine coolant temperature
T_{amb}	Ambient temperature
x_m	Clutch actuator motor position
x_p	Clutch actuator piston position
M_e	Engine torque, reported from ECU

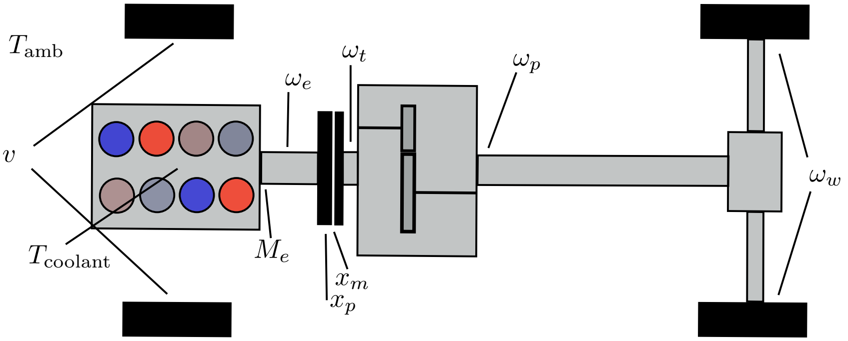

Figure 3.1: Sketch of a typical HDT driveline and where the sensors are placed. See Table 3.1 for an explanation of the notation.

observer evaluation can easily be performed on other vehicles equipped with a dry clutch. On the other hand, the drawback, compared to a test stand, e.g. Velardocchia et al. (1999); Moon et al. (2004) or Deur et al. (2012), is fewer and less precise sensors. All the cited test stands consist of a stand-alone dry clutch that is cooled towards the stagnated room-temperature air. In a truck there might be an airflow, the clutch is installed below the truck floor, bolted to the transmission, and especially the flywheel is bolted to a 1 tonne, ~ 90 °C warm engine. Naturally all this leads to different temperature dynamics.

3.1 Motivating Experiment

As a first investigation of the clutch characteristics the actuator position has been ramped back and forth several times while the clutch torque has been measured. Every second ramp pair the actuator has moved a shorter path. This is performed in order to study possible hysteresis effects. The results from such an experiment is presented in Figure 3.2. There the ramps are seen to follow a curve that can be fitted to a third order polynomial, (Dolcini et al., 2010; Myklebust and Eriksson, 2012b). However the polynomial is different for each ramp, there is clearly some dynamics present.

3.2 Actuator Dynamics

The clutch actuator has a built in position controller, the benefit of this setup is that effects causing hysteresis in the actuator force, e.g. friction or the diaphragm spring, will not effect the actuator output/clutch torque relation, (Deur et al., 2012). Furthermore a position measurement of the piston effectively makes the piston position the output of the actuator. Since the actuator output is known the clutch torque can be modeled without knowing the actuator dynamics.

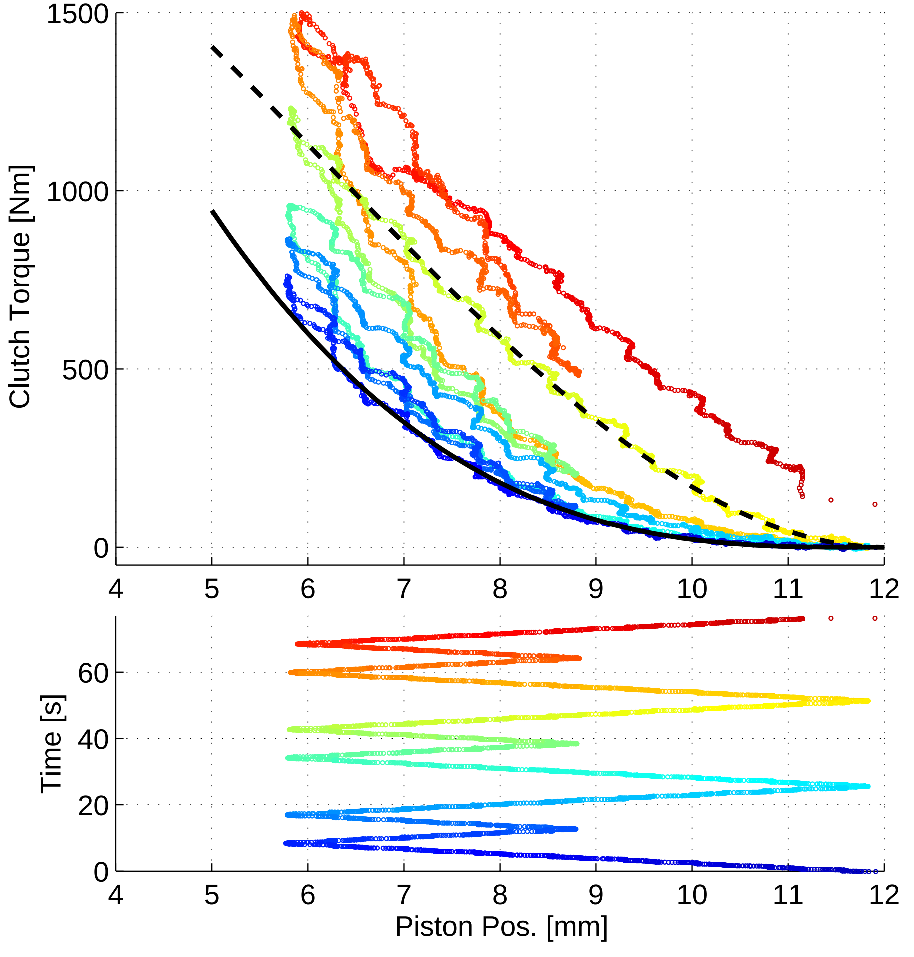

Figure 3.2: The clutch position, x, has been ramped back and forth while the clutch torque has been measured, top plot. The transmitted torque clearly depends on something more than the clutch position, in particular there is a drift with time. The time scale can be viewed by following the position into the lower plot. The time scale is also color coded, blue=0 s and red=75 s. The torque difference between the first and last ramp is up to 900 Nm. The two black lines are two possible parameterizations of a third order polynomial.

However for control applications the limitations of the actuator need to be considered. Conveniently the motor in the actuator is both accurate and fast, see Figure 3.3. The motor position can therefore be modeled as the requested position rate limited to 82 mm/s.

Some dynamics exists in the hydraulics of the actuator but the piston still

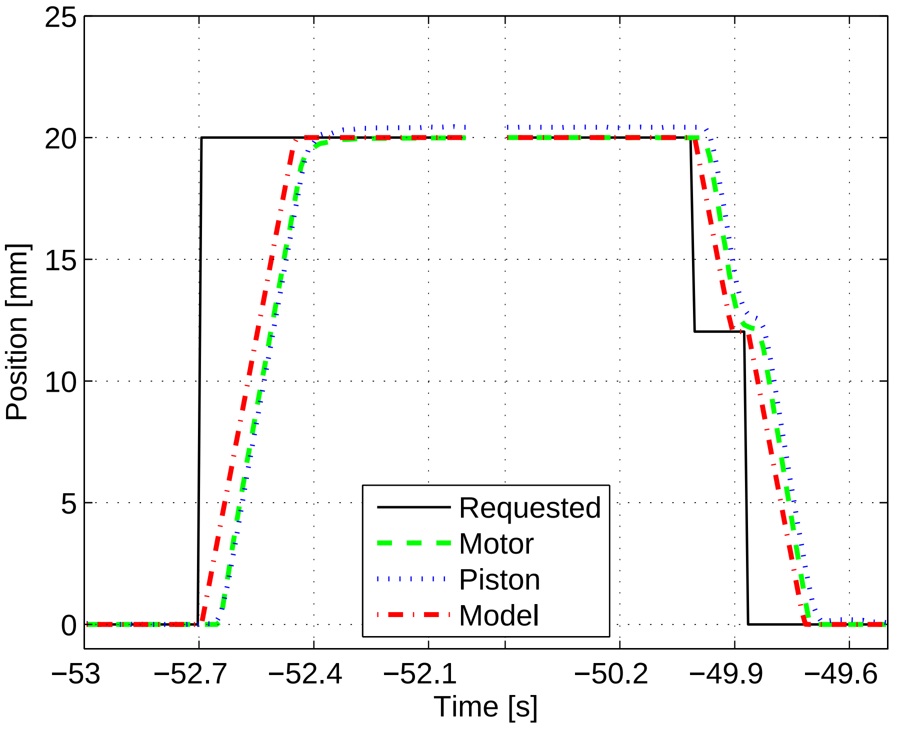

Figure 3.3: A step change in actuator (motor) position has been demanded. The corresponding response in both motor and piston position can be seen. The actuator is fast and can be modeled as a rate limiter of 82 mm/s.

moves at approximately the same rate as the motor, so control of the piston position have similar performance. During intense use of the clutch the hydraulic oil can be heated and thereby expand. This will cause a change in the relation between the motor and piston position. However since the piston position is measured and controlled this is of small concern. It should be noted that the clutch is located inside the bell housing (the housing surrounding the clutch) whereas the actuator is outside the bell housing. Therefore the temperature dynamics of the actuator has little to do with the temperature dynamics of the clutch discussed in Section 3.7.

3.3 Clutch Variables

Several factors could be the reason for the dynamics seen in Figure 3.2. First lets recall (2.1) from the previous chapter.

$$M_{\mathrm{trans,k}} = \mathrm{sgn}(\Delta\omega)\, n\, \mu\, R_e\, F_N \tag{2.1}$$

Looking at (2.1) one factor that directly affects the clutch torque is the clamp load. The clamp load is dependent on the actuator position but also on other factors, mainly the cushion spring characteristics. These characteristics are reported to have a decreasing slope with temperature in Cappetti et al. (2012a). However in Cappetti et al. (2012b) thermal expansion in the axial direction of the spring is shown to be a more significant effect. Sfarni et al. (2008) reports about steeper cushion spring characteristics with wear. From a graph in Hong et al. (2012) the normal force can be seen to be speed dependent. In Dolcini et al. (2010) that speed dependency is said to be due to centrifugal forces acting on the diaphragm spring. The works Mashadi and Crolla (2012); Moon et al. (2004) and Szimandl and Németh (2012) report of hysteresis in the diaphragm spring, that could lead to hysteresis in the normal force. A decrease in diaphragm spring force with temperature is reported in Sun et al. (2013). The decrease is especially pronounced for temperatures above 200 °C. In Mattiazzo et al. (2002) a temperature and wear dependency of the normal force/bearing position characteristics is shown. The wear is reported to affect the characteristics both through thinning of the disc, as in Figure 2.2, and through fatigue of the diaphragm spring. In Mattiazzo et al. (2002) the normal force decreases with temperature. In Deur et al. (2012); Cappetti et al. (2012b) and Hoic et al. (2013) the change in normal force/bearing position characteristics is explained by thermal expansion of clutch parts and the normal force is increasing with temperature. Although with an exception for low forces in Hoic et al. (2013). There the normal force is decreasing with temperature due to the expansion of a return spring, which is not present in the more common setup studied here.

However the normal force is not the only component of (2.1) that varies. It is generally recognized that μ can depend on temperature, slip speed, and wear while R_e can depend on temperature and wear as well, (Velardocchia et al., 1999; Sun et al., 2013). Sun et al. (2013) has data showing an increase of μ with temperature until 200 °C, after that μ decreases. In Vasca et al. (2011) a slip speed dependency of μR_e is shown, this was especially pronounced for slip speeds below $\sim$100 RPM. R_e has been assumed constant in Deur et al. (2012) and there μ decreases with temperature and increases with slip for a low to medium normal force but for a high normal force no clear trend is seen. The result is explained using (2.2).

It can be difficult to separate which parameter in (2.1) that is the reason for a change in $M_{\mathrm{trans,k}}$. Therefore the clutch torque is often studied as a lumped model. In Velardocchia et al. (1999) $M_{\mathrm{trans,k}}$ is seen to decrease with temperature and $\Delta\omega$. However there are large variations with wear. In Ercole et al. (2000) $M_{\mathrm{trans,k}}$ initially decreases with temperature for low temperatures and then increases for medium and high temperatures. Similarly there are variations with wear and in addition temperature-torque hysteresis are reported.

A summary of the effects and dependencies above is to express (2.1) as,

$$M_{\text{trans,k}} = \text{sgn}(\Delta\omega)\, n\, \mu(\Delta\omega, T, t)\, R_e(T, t)\, F_N(x, T, \omega_e, t) \tag{3.1}$$

where T is the temperature(s) and t represents wear (with time). Since production trucks are used as experimental platforms, the number of sensors is limited. This reduces the possibilities to separate the effects in (3.1). Therefore a lumped model has been studied here.

$$M_{\text{trans,k}} = M_{\text{trans,k}}(\Delta\omega, \omega_e, T, t, x) \tag{3.2}$$

However the more general model (3.1) gives some insight that can be used for setting up experiments and selecting the internal structure of the model (3.2). In order to investigate how the different dependencies of (3.2) affect the torque a number of experiments have been performed. The experiments have been designed to isolate different effects from each other and to step through the dependencies of (3.2).

3.4 Wear

Investigations of wear requires a lot of time consuming measurements. Luckily wear is usually a slow process that is not noticeable is single experiments. Therefore wear is omitted from the model structure,

$$M_{\text{trans,k}} = M_{\text{trans,k}}(\Delta\omega, \omega_e, T, x) \tag{3.3}$$

However in a production application wear have to be dealt with through e.g. adaption. The only wear effect that has been observed throughout this thesis is thinning of the clutch disc. A method for adapting the disc thickness is proposed in Paper C, (Myklebust and Eriksson, 2013b).

3.5 Slip

In order to investigate the slip speed, $\Delta\omega$, dependency experiments have been carried out as follows. A truck has been driven to an uphill and has been kept stationary by slipping the clutch at a constant position. This way the vehicle will accelerate forwards if more torque is transmitted and backwards if less torque is transmitted. Furthermore it is easy to, at least initially, keep the input shaft speed stationary this way, and by doing so the engine speed can be used to control the slip speed. The change in slip speed will of course have a corresponding change in dissipated power and hence the temperature rate. In order to isolate the slip dependency from temperature dynamics a fast change in slip speed is desirable. Therefore slip-speed steps are made and an abrupt

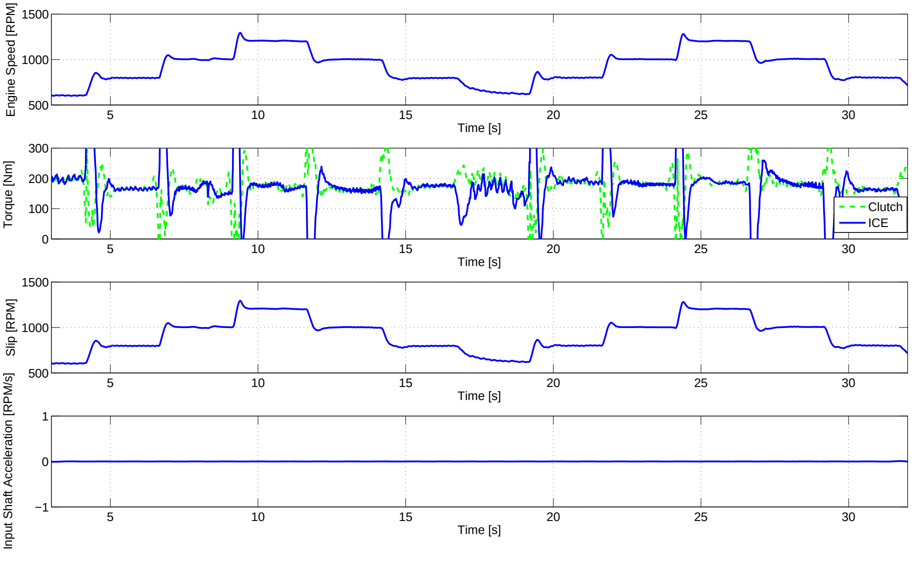

Figure 3.4: Slip steps are made through engine-speed control when the truck is held stationary in a slight uphill using a constant clutch position. No reaction is seen in neither clutch torque or input shaft acceleration.

change in torque/input shaft acceleration should be seen if there is a slip-speed dependency, see Figure 3.4. Note that the engine torque has a transient during the edges of a step, due to the acceleration of the engine. A smaller transient is present in the clutch torque signal due to imperfections in the compensation for the acceleration. No significant change is seen in the input shaft acceleration and no correlation between steady-state clutch torque and slip speed is seen neither, Figure 3.5. Based on this experiment (3.3) can be reduced to,

$$M_{\text{trans,k}} = M_{\text{trans,k}}(\omega_e, T, x) \tag{3.4}$$

This is somewhat in contrast to the data in Velardocchia et al. (1999) where a slip dependency is shown. The slip has only been investigated at relatively large speeds and does therefore not contrast the data in Vasca et al. (2011) where the slip dependency only is present below 100 RPM.

3.6 Rotational Speed

Centrifugal effects in the diaphragm spring can be studied with the same experiment as in Section 3.5 since the engine speed (and thereby diaphragm spring speed) is varied whereas the other parameters of (3.4) are kept constant. Hence no effects are seen in the torque in Figure 3.4 no speed dependency is

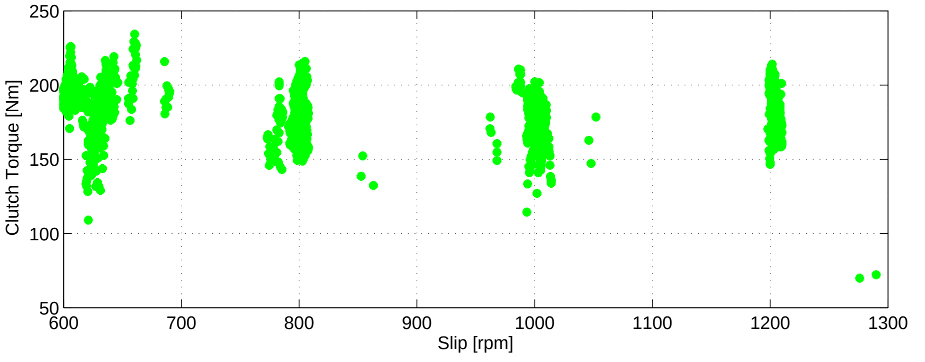

Figure 3.5: The data points from Figure 3.4 when the engine speed is constant. No correlation is seen between clutch torque and slip speed.

present either. The model can be further simplified into:

$$M_{\text{trans,k}} = M_{\text{trans,k}}(T, x) \tag{3.5}$$

There is a risk that an effect of ω_e is present but canceled by an effect of $\Delta\omega$. In order to fully investigate this the experiment of Section 3.5 would need to be repeated with different input-shaft speeds, which would give a different combination of slip speed and engine speed. This kind of experiments have however not been performed due to practical difficulties with controlling the input-shaft speed to a constant non-zero value.

3.7 Temperature

A variation of the experiment of Section 3.5 can be used to isolate the thermal effect. The truck is placed in an uphill and the clutch is controlled to a point where it can keep the truck stationary. Movement of the truck is not desirable as it can make the clutch lock up (and the slip to vary). The power dissipated will heat the clutch and if that affects the transmitted torque the truck will start to move unless the clutch position is controlled to keep a constant torque. Therefore the clutch torque is put under feedback and the clutch position is measured. In Figure 3.6 it is seen to increase as more energy is dissipated into the clutch. There are clearly some temperature dynamics present.

When performing experiments for identifying the thermal dynamics the cool down process is also of interest. However a position that transmits a slipping torque can not be measured if the clutch is to cool off. One possibility then is to investigate how the contact point (kiss point), where pressure plate and clutch disc first meet, changes with cooling. This feels intuitive as the kiss point is an important parameter in clutch control. However, due to inexactness in the torque measurement and the cushion characteristics of the clutch transmissibility curve

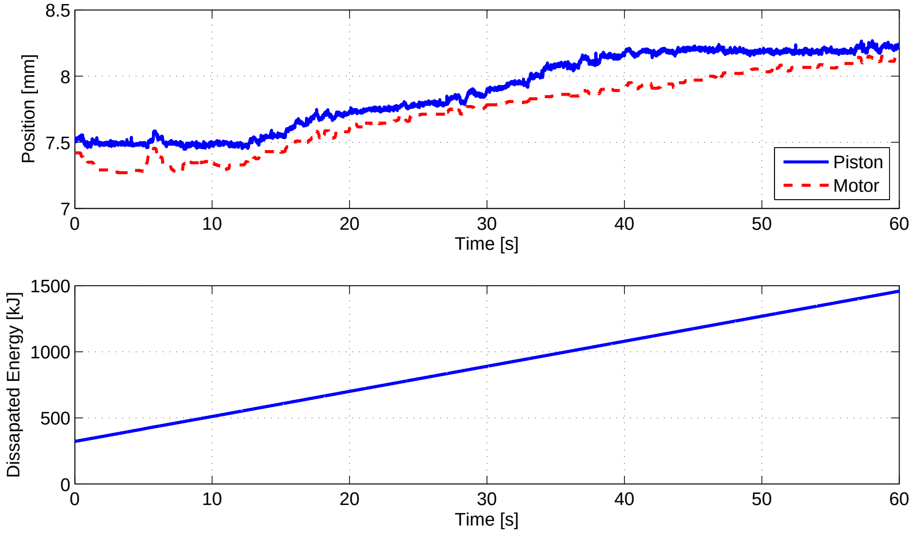

Figure 3.6: Slip and torque through the clutch has been held constant. To keep them constant the clutch position has to be varied. The motor position and piston position vary differently. The position increases with dissipated energy.

around zero torque, the exact position of the kiss point is hard to detect. One way to ease detection is to engage the clutch when neutral gear is engaged. This way the input shaft speed will respond to the transmitted torque quickly. This type of experiments have shown the kiss point to jump around, see Figure 3.7. This is thought to be because the clutch disc can slide axially along the transmission input shaft and stop in an arbitrary position in the gap between pressure plate and flywheel. The friction along the shaft can provide a sufficient clamp load to transfer enough torque to spin the input shaft when in neutral gear, but not enough to transmit any significant torque for the truck dynamics. Therefore the exact value of the kiss point is not of great concern, the point where clutch torque gets significant is of greater concern.

An alternative measure is the position where the clutch is fully closed, here called the zero position. This point is easy to determine, when the actuator motor is fully backed out (motor position zero, hence the name zero position) the clutch will, due to the diaphragm spring, be fully closed. If the clutch is closed when neutral gear is engaged no significant energy is dissipated into the clutch. The zero position does therefore give a more stable measurement, see Figure 3.7. The zero position increases when heat is dissipated into the clutch and it decreases exponentially when no heat is dissipated. A hypothesis is that thermal expansion of the clutch is the reason for the change in zero position. Therefore a rough estimate has been made of the thermal expansion.

The clutch mass is estimated as the flywheel and pressure plate, m_c =

$m_{fw} + m_{pp} = 47 + 34 = 81$ kg. They are made out of cast iron and according to Nording and Österman (1999) the expansion coefficient is $e_{ci} = 11 \cdot 10^{-6}$ 1/K and the specific heat capacity $c_p = 500$ J/(kg K). If 4 MJ of energy is dissipated into the clutch through slipping, as in Figure 3.7, the clutch will heat up with approximately 98 K and expand by 0.11 %. The clutch is about 100 mm deep, which means it will expand by 0.11 mm. At the actuator this is seen as 0.96 mm (ratio of 8.8 measured in drawings). When comparing this number with the change in the zero position seen in Figure 3.7, the numbers are in the same order of magnitude. During the warm up the zero position increases with ~ 1.2 mm and during cool down it decreases ~ 0.8 mm asymptotically. This gives further support for the hypothesis that this effect is due to the thermal expansion of the clutch.

In order to observe the zero position not only when the clutch is cooling but also when the clutch is heated the following procedure has been used. The clutch has been slipped for a short time while the truck has been in gear and kept stationary by the parking brake. After the slipping phase the clutch has been closed in neutral gear so that the zero position can be measured. The slipping phase and measurement phase is alternated like this for an extended period of time. The result of such an experiment can be seen in Figure 3.8. A simple temperature model can be fitted to this data, see Paper A, (Myklebust and Eriksson, 2012b), or Paper C, (Myklebust and Eriksson, 2013b), for details.

However if the dynamics seen in Figure 3.8 is compared to those seen in Figure 3.6 two things can be noticed. First the change in position is much greater in Figure 3.6. Second the time constant in Figure 3.6 is much smaller. This smaller time constant could correspond to a smaller thermal mass, e.g. the clutch disc. The thermal behavior of the cushion spring inside the clutch disc is investigated in Cappetti et al. (2012b) and could explain the faster dynamics seen here. A second expansion term with faster dynamics is useful in order to explain the clutch thermal dynamics, again see Paper A, (Myklebust and Eriksson, 2012b), or Paper C, (Myklebust and Eriksson, 2013b), for details.

The temperature dynamics can consist of several temperature but in common

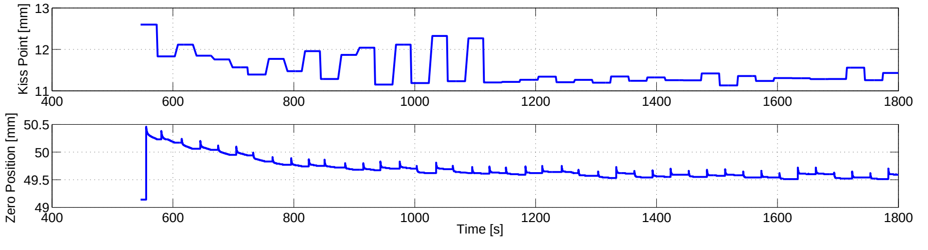

Figure 3.7: About 4 MJ of energy has been dissipated in the clutch. Afterwards the truck has been standing still with neutral gear engaged while the kiss point and zero position has been monitored.

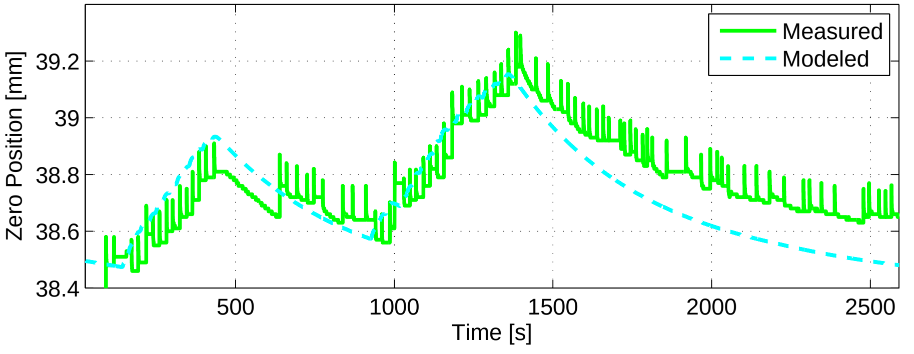

Figure 3.8: The clutch has been heated through slipping and then left to cool down. The zero position has been recorded as a measure of the expansion. A simple temperature model can explain the behavior.

for all temperatures will be that energy is gained through the power dissipated in the clutch,

$$P = M_{\text{trans,k}}\,\Delta\omega \tag{3.6}$$

and energy is dissipated to the ambient air and the ICE, but not so much the transmission, (Wikdahl and Ågren, 1999). Since the truck is moving and the clutch is spinning there will be forced convection. Therefore also the vehicle and clutch speed are possible variables.

$$\dot{T} = f\left(T, P, T_{\text{ICE}}, T_{\text{amb}}, \omega_e, v\right) \tag{3.7}$$

where T is a temperature vector, T_{ICE} is the coolant temperature of the ICE, T_{amb} is the ambient temperature and the engine speed is used as clutch speed.

3.7.1 Vehicle Speed Dependency

The speed of the truck directly affects the speed by which air is flowing past the bell housing. This forced convection will naturally increase the cooling of the clutch. In most experiments performed here the truck has been held stationary in order to keep the clutch from locking up. Although this should be of smaller concern when using the observer of Paper C, (Myklebust and Eriksson, 2013b), since the bell housing temperature only affects the clutch torque indirectly through the ambient conditions of the flywheel and pressure plate. Equation (3.7) can be reduced to,

$$\dot{T} = f\left(T, P, T_{\text{ICE}}, T_{\text{amb}}, \omega_e\right) \tag{3.8}$$

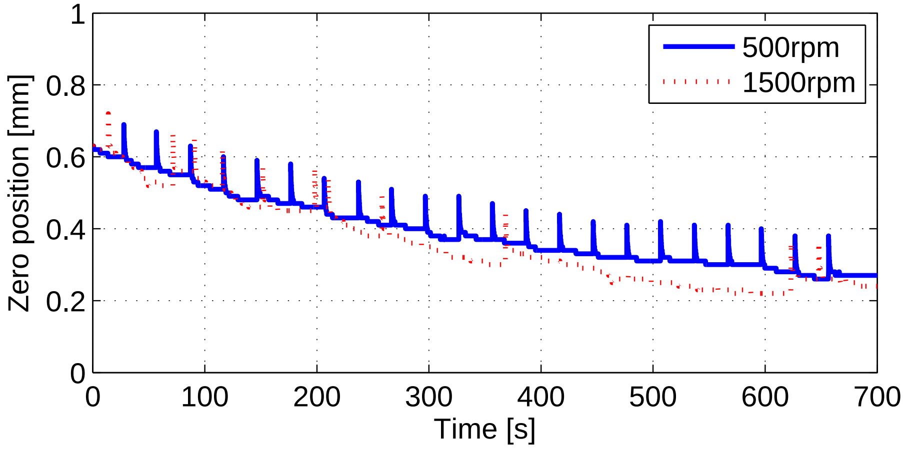

Figure 3.9: Zero position measurements when the clutch is cooling. In one case the engine speed has been 500 RPM and the coolant temperature has been 77 °C. In the other case the engine speed has been 1500 RPM and the coolant temperature has been 82 °C. The two trajectories are similar. The zero position has been adjusted with an offset to compensate for different wear levels in the two experiments.

3.7.2 Engine Speed Dependency

If the clutch is spinning at a higher speed there should be more forced convection and the clutch body should cool faster towards clutch housing temperatures. Therefore experiments have been carried out where the zero position has been measured during cooling of the clutch with two different engine speeds, and hence two different clutch body speeds. Looking at the data in Figure 3.9 there is no significant difference in the curve shapes indicating that the clutch speed has no significant effect on the clutch temperature. Therefore (3.7) can be further reduced to,

$$\dot{T} = f\left(T, P, T_{\mathrm{ICE}}, T_{\mathrm{amb}}\right) \tag{3.9}$$

Using the observer from Paper C, (Myklebust and Eriksson, 2013b), it can be verified that both measurements start at the same temperature level. For the 500 RPM case and the 1500 RPM case the clutch body temperature is 109 °C and 107 °C, respectively and the clutch housing temperature is 83 °C and 85 °C, respectively. The engine coolant temperature was measured to 77 °C and 82 °C respectively.

3.8 Model Summary and Usage in Papers

In this and the previous chapter a literature survey of dry-clutch models and experiments is summarized into the model structure (3.2). Thereafter a set of experiments are designed to investigate which variables in (3.2) that are significant. In all experiments the platform has been a production truck driven on roads. Therefore this set of experiments can easily be adapted to new platforms. After performing the experiments the model structure is:

$$\begin{aligned} \dot{T} &= f\left(T, P, T_{\mathrm{ICE}}, T_{\mathrm{amb}}\right) \\ M_{\mathrm{trans,k}} &= M_{\mathrm{trans,k}}\left(T, x\right) \end{aligned} \tag{3.10}$$

This model structure is developed and parameterized in Paper A, (Myklebust and Eriksson, 2012b). The model that results from Paper A, (Myklebust and Eriksson, 2012b), is the foundation for Paper B, (Myklebust and Eriksson, 2013a), and Paper C, (Myklebust and Eriksson, 2013b). In Paper B, (Myklebust and Eriksson, 2013a), simulations of the clutch model and the driveline model from Paper D, (Myklebust and Eriksson, 2012a), are used to demonstrate the importance of considering the thermal dynamics in clutch control applications. One way to do that is to build an observer using the model. This is done in Paper C, (Myklebust and Eriksson, 2013b), where a clutch torque and temperature observer is designed.

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