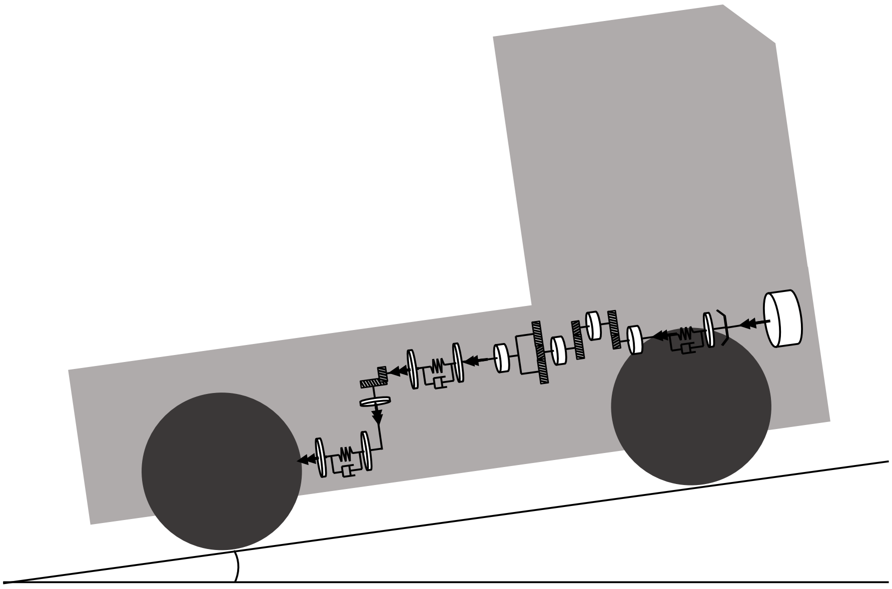

Realtime modeling of driveline dynamics as estimator for control of AMT-gearbox in heavy-duty road vehicle

Bachelor thesis in mechanical engineering

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DEPARTMENT OF
INDUSTRIAL AND MATERIALS SCIENCE

CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2020
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Bachelor's thesis 2020

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Cover: Illustration of truck with driveline.

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Abstract

With Automated Manual Transmissions(AMT), gearshifting is controlled through algorithms controlling both engine and actuators inside the gearbox. The objective is to optimise shift in terms of comfort, energy efficiency and to minimise the time without propulsion. There is relatively low stiffness and high inertia in the driveline of a truck which affects torques and speeds of shafts in the gearbox. Torque is complicated and expensive to measure in production gearboxes. Therefore there is a value in developing algorithms for estimation and prediction. The algorithms modeling the drivetrain needs to be computationally efficient to be run in real time but also include the first order of resonance for the complete driveline. It is shown that a simplified mathematical model of differential equations can replicate the behaviour of the driveline and predict torques and speeds of shafts. Efforts has been made to reduce the number of states in order to reduce the computational load. Comparison of model with data from a test truck shows that accuracy can be achieved for a simplified model with the use of known physical properties for a specific vehicle. The model is also tested with forward Euler iteration method to determine necessary step size to produce a stable prediction.

Keywords: AMT-gearbox, driveline control, real time, dynamic model, estimation, prediction.

Acknowledgements

This work has been carried out during the autumn of 2020 which is not the regular period of writing thesis's at Chalmers. There is currently an unusual high workload for all employees at both Chalmers and AB Volvo due to Corona virus which demands rapid conversion to digitalisation which even strengthen my humility towards acknowledged persons presented here. Thanks to Johan Bankel for organising and supporting this work and providing creative solutions for solving practicalities. I would also like to thank professor Magnus Evertsson at Chalmers IMS for his support and grading of this work. Special thanks to professor Bengt Jacobson for excellence in the area of vehicle dynamics and weekly support, it has been a pleasure to collaborate. Big thanks to Volvo Powertrain group Cluch & Shifting Controls for both technical and practical support and especially to Fredrik Löwgren for enabling and supporting the work. To Oscar Lygnert from Transmission Verification for providing excellent test data and analysing tools.

Andreas Hamilton, Åsa, November 2020

0.1 Nomenclature

0.1.1 Abbreviations

AMT	Automated manual transmission
EECU	Engine electronic control unit
TECU	Transmission electronic control unit
ABS	Anti block system
ICE	Internal combustion engine
EBS	Electronic brake system

0.1.2 Notations

Variables and parameters presented in this thesis are composed in following arrangement:

$$Notation_{Subscript}$$

Notation	Description
ω	Angular speed [rad/s]
θ	Angle [Rad]
F	Force[N]
Fr	Rolling resistance force[N]
M	Torque [Nm]
i	Ratio [$\frac{\omega}{\omega}$]
g	gravity[m/s^2]
$gear$	Current gear [1]
l	Position [m]
CCX	Tire longitudinal slip coefficient[1]
c_x	Tire longitudinal slip stiffness[N/1]
r	Tire radius[m]
e	Tire force horizontal displacement[m]
k	Spring constant[Nm/rad]
b	Damping constant[Nm.s/rad]
a	Exponential loss constant[1]
c	Scalar loss constant[1]
α	backlash angle[rad]
rrc	rolling resistance coefficient[1]
n	Efficiency[1]

Subscript	Description
e	Engine
e,fuel	Engine torque calculated from fuel amount and efficiency.
e,friction	Engine friction torque
e,brake	Engine brake torque
c	Clutch
ck	Clutch spring
cd	Clutch damping
g	Gearbox
gs	Gearbox split section
gm	Gearbox main section
gr	Gearbox range section
gi	Gearbox input shaft
gc	Gearbox counter shaft
go	Gearbox output shaft
p	Propeller shaft
pk	Propeller shaft spring
pd	Propeller shaft damping
f	Final drive
d	Driveshafts
dp	Driveshaft and propeller shaft
w	Wheels
gfi	Gearbox friction input shaft
gfc	Gearbox friction counter shaft
gfm	Gearbox friction main shaft
gfo	Gearbox friction output shaft
zprop	Z-axis, propelled wheels
zunprop	Z-axis, unpropelled wheels

Contents

1
Introduction

1.1 Background

Volvo Powertrain develops drivelines for heavy vehicles focusing on efficiency and drivability. The I-Shift transmission is an automated mechanical gearbox and is the most commonly used in Volvos commercial range of trucks and buses [1]. The embedded computer, “Transmission Electronic Control Unit” (hereafter abbreviated TECU) handles shifting strategies within the gearbox by measuring various signals and controlling actuators and solenoids. From a control technology perspective, it would be optimal to measure speed and torque of all shafts within the gearbox and use ideally fast actuators to maximise shifting precision. Due to technical and product cost limitations, the amount of physically measured signals is limited. Therefore, there is a necessity for calculating and estimating physical properties in real time within the TECU in order to minimise variant specific tuning of control algorithms. In addition to the problem to know the mountainous quantities, it is also necessary to take height for actuators delays. Therefore, a prediction is needed so one sends out requests in time.

1.2 Problem motivating the project

One example of problem that can be addressed with the result of this project is when to disengage the transmission from the engine. The clutch is the connecting link between the engine and the gearbox. To minimize shunt and shuffle behaviour , the clutch should preferably be disengaged when there is no torque passing through[2] [3] . The torque through the clutch is affected by the speed of the engine in relation to the dynamics of the drivetrain and also the dynamics of the complete vehicle. The speed of the engine is measured by the flywheel sensor, and the dynamics of the vehicle affecting the gearbox can be determined by the ABS-sensors of the wheels together with estimated engine torque. The problem addressed in this project is to model the dynamics of the driveline in real time. A prediction of 500ms would be adequate with a good margin in order to foresee dynamics and have time to control actuators.

1.3 Envisioned solution

The envisioned solution is a computer model of the dynamic behavior of the driveline.

1.4 Objective

The objective of the project is to develop a computer model of the dynamic behavior of the driveline. The model will be used to estimate torque and speed of the included components of the driveline and should be possible to execute in real time.

1.5 Deliverables

- Model as set of physically motivated differential-algebraic equations
- A validation of the model versus test data
- A list of improvement areas of the model
- Outline for ASCET implementation

1.5.1 Questions to be answered

- What physical properties are important for creating a consistent model?
- What is the main properties affecting the accuracy of the model?
- What is the limiting factors for accuracy from a real-time perspective?
- How can the model developed be further improved?

1.6 Limitations

- Single clutch version of I-Shift transmission only.
- Only forward gears with high and low-range and crawler gear is included.
- The clutch is an advanced component to model throughout but a simplified version will be included in the project. The level of precision of the clutch model is set by the limitations of resources of the real time system and estimations of its effect on the dynamic behaviour of the driveline.
- The project includes development of computer model from known properties. Delivery of unknown properties is not covered by this project but will be estimated and documented under future work.
- The correlation of the model will be made as accurate as possible considering the resources and data available.
- The complexity of the model is to be limited for the model to execute effectively in a real time system.

1.7 Method

In order to ensure a reliable result, research is first carried out to map the power path of the transmission and the included components are documented in a schematic function description. The location and types of bearings and gears as well as any other frictioning components included in the system are examined. Then data is gathered such as stiffness, inertia, latch and frictions as well as its dependencies such as speed, torque and temperature. Existing data models over the driveline and it's included subsystems are examined to gain knowledge. In addition, suitable output signals that can be correlated with physical measurement are examined. Available signals for input data are evaluated together with its properties such as refresh rate and time delay. With the established data, a computer model is developed. To ensure reliability, the computer model is tested by preforming simulations and comparing results with test data from a physical test truck. Simulations are performed on variants of transmissions that are included within the scope of the project and for which there are available vehicles or measurement data. The outcome from tests/simulations is categorised according to the size of the deviation and is derived by analysis and, if possible, appropriate adjustments are made to the computer model.

2
Driveline description and power paths

2.1 System overview

A heavy truck is built around two longitudinal C-beams that supports the engine/gearbox assembly. The propulsion torque is generated by the ICE and delivered to the transmission through the clutch. The objective of the transmission is to optimise engine speed for the propulsion torque demanded by altering gear ratio. The propulsion torque is thereafter transmitted to the final drive by the propeller shaft. The final drive distributes the propulsion torque through the driveshafts to the wheels. To simplify modelling, it is assumed that the housings of the driveline components are stiff and not rotating with respect to the axis perpendicular to the truck's movement and the driveshafts and wheels are have the same speeds, a.k.a. truck is not turning.

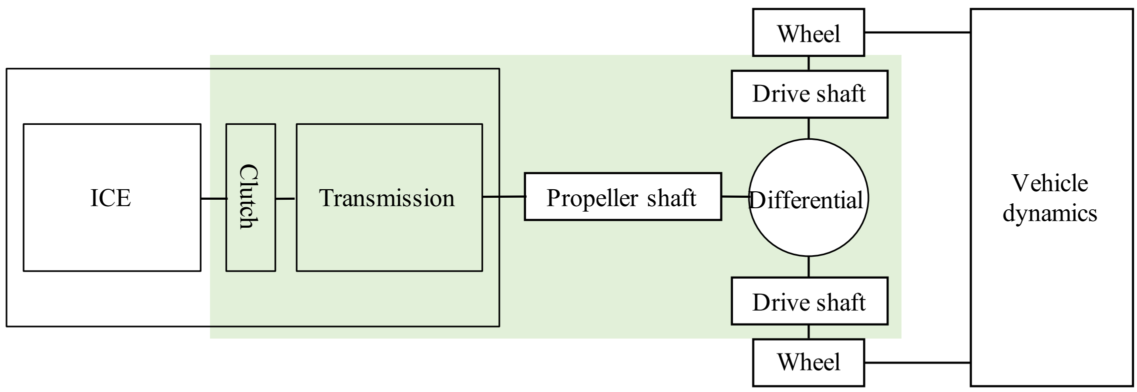

Figure 2.1: System description of objects to simulate

2.2 Function description of gear change

The I-Shift gearbox consist of three areas of gears that can be divided into split section, main section and range section. Torque enters the gearbox through input shaft and reaches split section where it is transferred by counter shaft into main section. Herby torque is transferred to main shaft and thereafter through range section to output shaft. There is also a variant of I-Shift with an extra set of gears called crawler gears. For modelling purpose, it can be seen as an extra split gear that adds a power path to the counter shaft and increased inertia to input shaft and counter shaft. Note that split section and main section shares a gear set and if dog clutch is connected to that set both in split and main section, torque will pass straight from input shaft to main shaft. Countershaft is still connected and turns with its inertia but carries no propulsion torque.

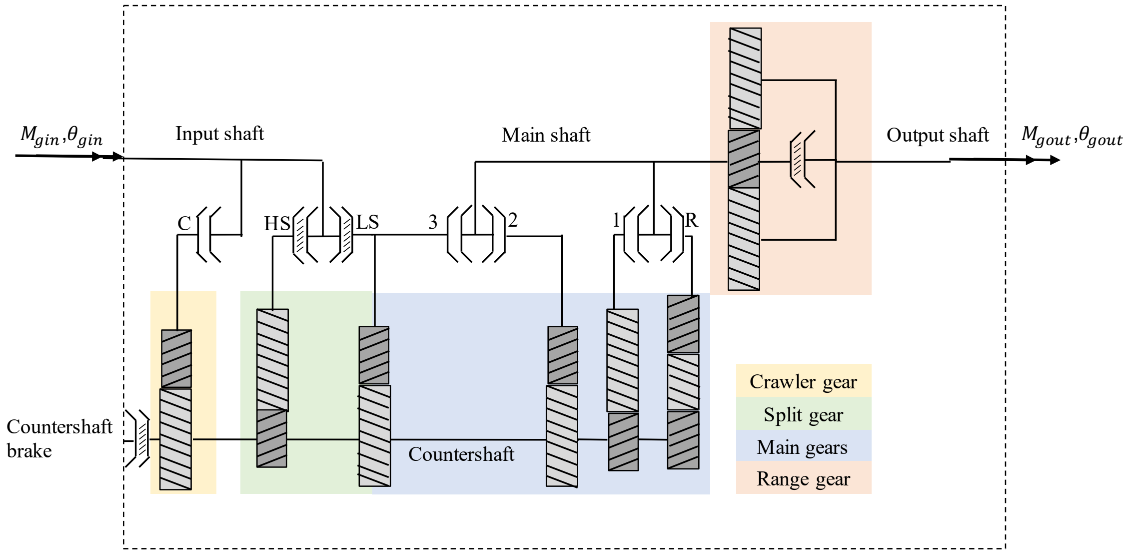

Figure 2.2: Principal design of gears and dog clutches in gearbox.

Typical step by step up shift:

1. Gear change initiated and engine torque is ramped down.

2. Clutch torque reaches zero and clutch disconnects. Estimating when clutch torque is zero is a key problem in this report.

3. Split gear and range are shifted. Counter shaft speed is controlled by counter shaft brake to limit wear on split synchronisation.

4. When countershaft speed reaches synchronisation speed for main shaft, main dog clutch engages.

5. Clutch closes and engine torque ramp up begins

6. Up shift is completed

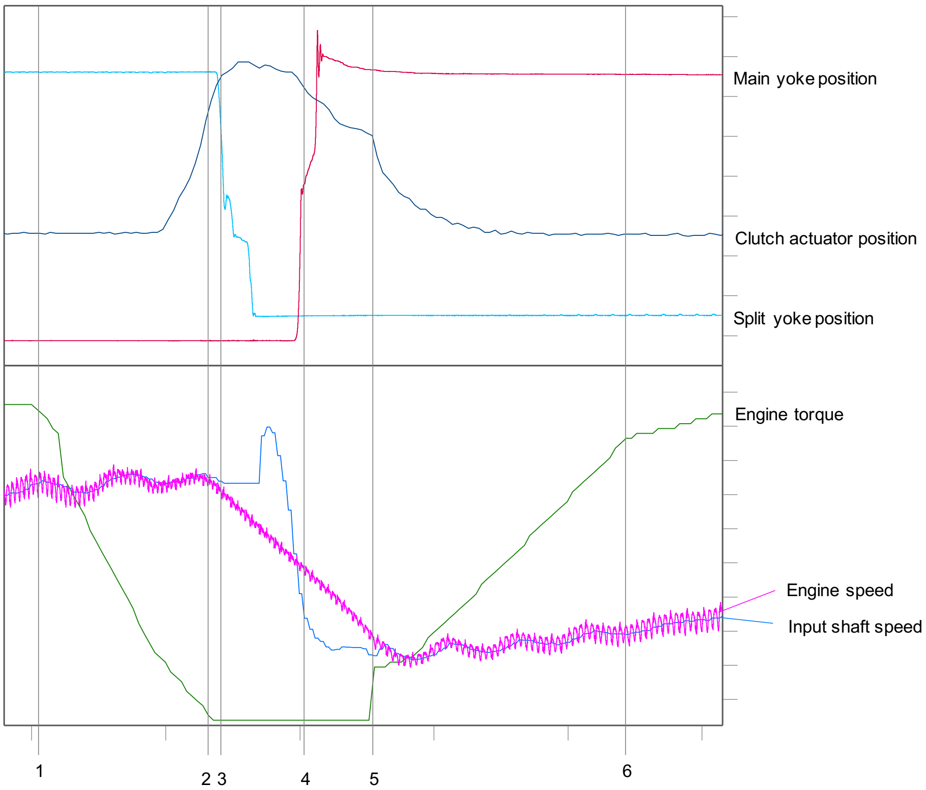

Figure 2.3: Typical up shift with 0.5s between time axis markers. Data from test in real vehicle.

Typical step by step down shift:

1. Engine speed is controlled to release tension in drivetrain and clutch disconnects.
2. Main box is shifted to neutral state
3. Split gear and range are shifted.
4. Clutch is connected and engine speed is controlled to match required speed for main gear synchronisation.
5. When countershaft speed reaches synchronisation speed for main shaft, clutch opens briefly and main dog clutch engages.
6. Down shift is completed

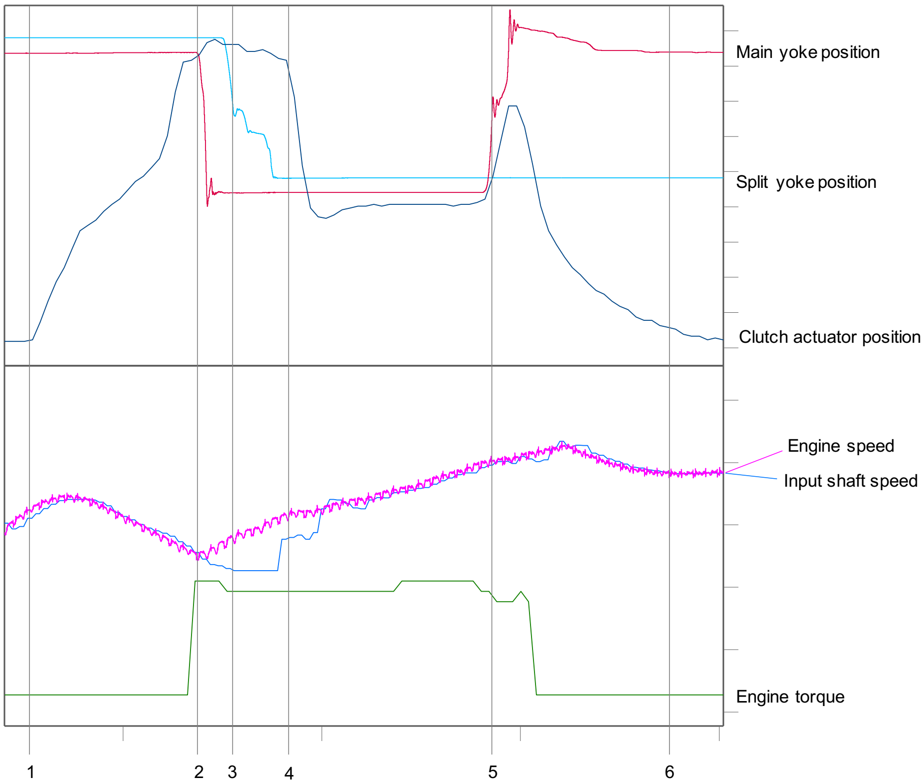

Figure 2.4: Typical down shift with 0.5s between time axis markers. Data from test in real vehicle.

2.3 Component description and their intermediate models

2.3.1 Engine

The simplified engine model uses fuel, inertia and friction to estimate flywheel torque. Propulsion torque from fuel injected and friction torque is available as signals from EECU.

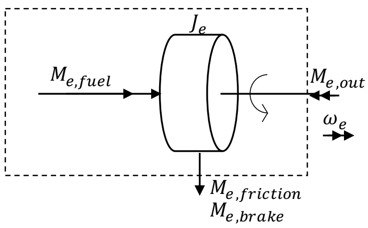

Figure 2.5: Engine model.

$$\dot{\omega}_e \cdot J_e = M_{e,fuel} - M_{e,friction} - M_{e,out} \tag{2.1}$$

2.3.2 Clutch

The clutch consists of a spring and damper in the centre of the friction disc. The damper is of dry friction type but is here modelled as a linear viscous damper.

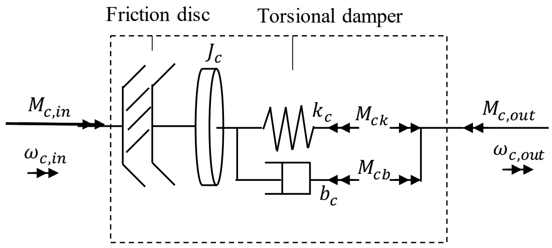

Figure 2.6: Model of clutch.

Equations for closed clutch:

$$
\begin{aligned}
\dot{M}_{ck} &= k_c(\omega_{c,in} - \omega_{c,out}) \\
M_{cb} &= b_c(\omega_{c,in} - \omega_{c,out}) \\
\dot{\omega}_{c,in} \cdot J_c &= M_{c,in} - M_{ck} - M_{cb})
\end{aligned}
\tag{2.2}
$$

2.3.3 Gearbox

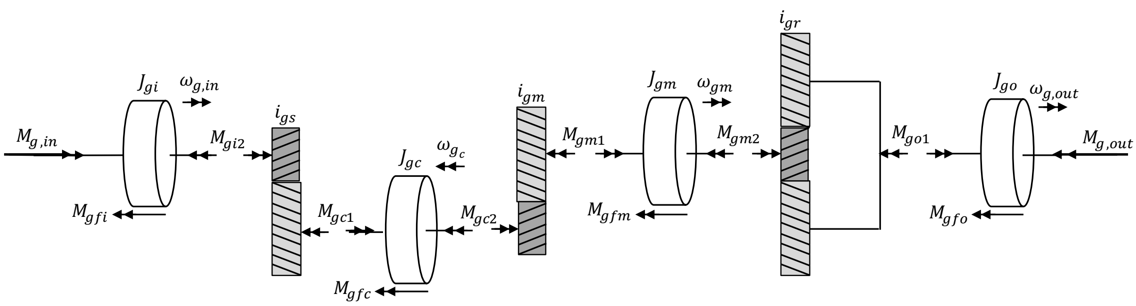

Figure 2.7: Principal design of stiff gearbox with gear engaged.

$$
\begin{aligned}
M_{gi2} &= M_{g,in} - J_{gis} \cdot \dot{\omega}_{g,in} - M_{gfi} \\
M_{gc1} &= M_{gi2} \cdot i_{gs} \\
M_{gc2} &= M_{gc1} - J_{gc} \cdot \dot{\omega}_{gc} - M_{gfc} \\
M_{gm1} &= M_{gc2} \cdot i_{gm} \\
M_{gm2} &= M_{gm1} - J_{gm} \cdot \dot{\omega}_{gm} - M_{gfm} \\
M_{go1} &= M_{gm2} \cdot i_{gr} \\
M_{g,out} &= M_{go1} - J_{go} \cdot \dot{\omega}_{g,out} - M_{gfo} \\
\omega_{gc} &= \frac{\omega_{g,in}}{i_{gs}} \rightarrow \dot{\omega}_{gc} = \frac{\dot{\omega}_{g,in}}{i_{gs}}
\end{aligned}
$$

$$\omega_{gm} = \frac{\omega_{g,in}}{i_{gs} \cdot i_{gm}} \to \dot{\omega}_{gm} = \frac{\dot{\omega}_{gin}}{i_{gs} \cdot i_{gm}}$$

$$\omega_{g,out} = \frac{\omega_{g,in}}{i_{gs} \cdot i_{gm} \cdot i_{gr}} \to \dot{\omega}_{g,out} = \frac{\dot{\omega}_{g,in}}{i_{gs} \cdot i_{gm} \cdot i_{gr}}$$

$$\begin{aligned} M_{g,out} = (((M_{g,in} - J_{gi} \cdot \dot{\omega}_{g,in} - M_{gfi})i_{gs} \\ -J_{gc} \cdot \dot{\omega}_{gc} - M_{gfc})i_{gm} - J_{gm} \cdot \dot{\omega}_{gm} - M_{gmf})i_{gr} - J_{go} \cdot \dot{\omega}_{g,out} - M_{gof} \end{aligned} \tag{2.3}$$

(Eq.2.3) Expressed from $\omega_{g,in}$

$$\begin{aligned} M_{g,out} = M_{g,in} \cdot i_{gs} \cdot i_{gm} \cdot i_{gr} - M_{gfo} - i_{gr} \cdot M_{gfm} - i_{gr} \cdot i_{gm} \cdot M_{gfc} - i_{gs} \cdot i_{gm} \cdot i_{gr} \cdot M_{gfi} \\ -\dot{\omega}_{g,in} \left(\frac{J_{go}}{i_{gs} \cdot i_{gm} \cdot i_{gr}} + \frac{J_{gm} \cdot i_{gr}}{i_{gm} \cdot i_{gs}} + \frac{J_{gc} \cdot i_{gm} \cdot i_{gr}}{i_{gs}} + J_{gi} \cdot i_{gs} \cdot i_{gm} \cdot i_{gr} \right) \end{aligned} \tag{2.4}$$

2.3.3.1 Gearbox losses

In previous studies within the topic of driveline modeling, bearing friction is usually modelled as a damper between axle and transmission housing which makes friction torque a linear function of the axle speed.[8] In reality bearing friction torque is more dependent on the bearing forces than the speed of the axles. The bearings within the transmission is mainly of type conical roller bearings which are well lubricated and gears that are helical which causes both axial and radial load on the shafts and bearings

Losses for gearbox is derived from available test data and is divided into two parts, speed dependent and torque dependent losses[6]. Torque dependent losses will be applied at gearbox input shaft and is only considered when drivetrain is connected from engine to wheels whereas the speed dependent losses are distributed over the internal shafts in the gearbox.

2.3.3.1.1 Speed dependent losses Speed dependent losses are presented as $P = c * n^a$; n=axle speed [rpm];c = loss coefficient for specific axle;a = speed dependent loss exponent factor. The equation is hereby rearranged to allow input of ω in [rad/s] and to output torque instead of power.

$$M \approx 46.34 \cdot c \cdot \omega^{(a-1)} \tag{2.5}$$

2.3.3.1.2 Torque dependent losses Torque dependent losses are presented as efficiency distributed by gear set.
Crawler gear together with main section have efficiency $n_{crawler}$.
High and low split are considered to have equal efficiency and is summed together with efficiency for main section.
Split and 1:st gear in main section have efficiency: n_1.
Split and gears 2,3 in main section have efficiency: n_{2-3}.

Direct drive happens if direct split and gear 3 is engaged and have efficiency: n_{direct}. Range unit has increased losses when low range is active with efficiency: n_{lr}.
The efficiencies are thereafter added together to represent complete gearbox according to Algorithm 1.

Algorithm 1: Gear dependent gearbox efficiency

Result: n_{gbx}
if *Crawler* **then**
  $n_{gbx} = n_{crawler}$;
else if *1:st gear* **then**
  $n_{gbx} = n_1$
else
  $n_{gbx} = n_{2-3}$
end
if *Direct split AND 3:rd gear* **then**
  $n_{gbx} = n_{direct}$
end
if *low range* **then**
  $n_{gbx} = n_{gbx} + n_{lr}$
end

2.3.3.1.3 Combined losses The combined losses for the gearbox with gear engaged, with efficiency translated on the input shaft is therefore:

$$M_{gfi} = M_c \cdot n_{gbx} + 46.34 \cdot c_{lis} \cdot \omega_{gin}^{(a-1)} \tag{2.6}$$

$$M_{gfc} = 46.34 \cdot c_{gcs}(\frac{\omega_{gin}}{i_{gs}})^{(a-1)} \tag{2.7}$$

$$M_{gfm} = 46.34 \cdot c_{gms}(\frac{\omega_{gin}}{i_{gs} \cdot i_{gm}})^{(a-1)} \tag{2.8}$$

$$M_{gfo} = 0 \tag{2.9}$$

According to test data, there is no speed dependent loss isolated to the output shaft. The coefficient for the output shaft is probably small enough to be included with the main shaft and therefore here neglected as a separate loss.

2.3.4 Propeller shaft

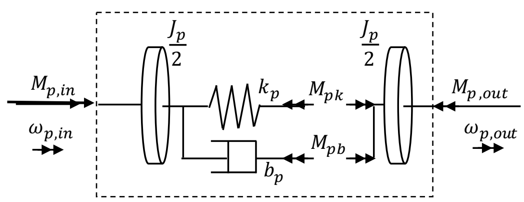

Figure 2.8: Model of propeller shaft.

The inertia of of the propeller shaft is distributed over its full torsional length. For simplification the inertia is hereby split in two parts for the model, attached at input and output.

$$\dot{M}_{pk} = k_p(\omega p, in - \omega_{p,out}) \tag{2.10}$$

$$M_{pb} = b_p(\omega_{p,in} - \omega_{p,out}) \tag{2.11}$$

$$\dot{\omega}_{p,in}\frac{J_p}{2} = M_{p,in} - M_{pk} - M_{pb} \tag{2.12}$$

$$\dot{\omega}_{p,out}\frac{J_p}{2} = M_{pk} + M_{pb} - M_{p,out} \tag{2.13}$$

2.3.5 Final gear and differential

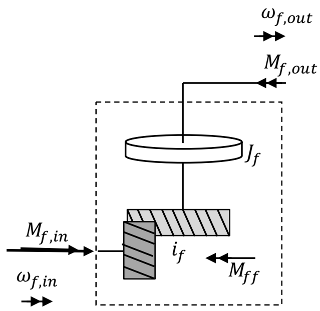

Figure 2.9: Model of Final gear and differential.

Equations for differential:

$$\dot{\omega}_{f,out}J_{f,out} = M_{f,in} \cdot i_f - M_{f,out} \tag{2.14}$$

2.3.6 Drive shafts

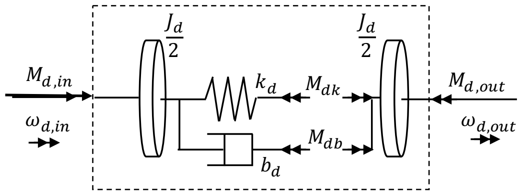

Figure 2.10: Model of drive shafts.

The inertia of of the drive shaft is distributed over its full torsional length. For simplification the inertia is hereby split in two parts for the model, attached at input and output.

$$\dot{M}_{dk} = k_d(\omega d, in - \omega_{d,out}) \tag{2.15}$$

$$M_{db} = b_d(\omega_{d,in} - \omega_{d,out}) \tag{2.16}$$

$$\dot{\omega}_{d,in}\frac{J_d}{2} = M_{d,in} - M_{dk} - M_{db} \tag{2.17}$$

$$\dot{\omega}_{d,out}\frac{J_d}{2} = M_{dk} + M_{db} - M_{d,out} \tag{2.18}$$

2.3.7 Propelled wheels

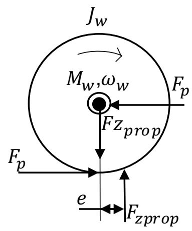

Figure 2.11: Model of wheel.

$$F_p = c_x \cdot slip_x$$

$$c_x = CC_x \cdot F_{zprop}$$

$$slip_x = \frac{r_w \cdot \omega_w - v_x}{|r_w \cdot \omega_w|}$$

$$F_p = c_x \cdot \frac{r_w \cdot \omega_w - v}{|r_w \cdot \omega_w|} \tag{2.19}$$

Normal force parallel displacement due to rolling resistance coefficient:

$$e = RRC \cdot r \tag{2.20}$$

Typical values for truck tire coefficient CC_x ranges from 10 to 15 and $e \approx 0.5\%$ [5]

Acting torque from rolling resistance:

$$M_{rprop} = e \cdot F_{zprop} \tag{2.21}$$

Differential equation for wheel

$$\dot{\omega}_w \cdot J_w = M_w - r_w \cdot F_p - M_{rprop} \tag{2.22}$$

2.3.8 Vehicle motion

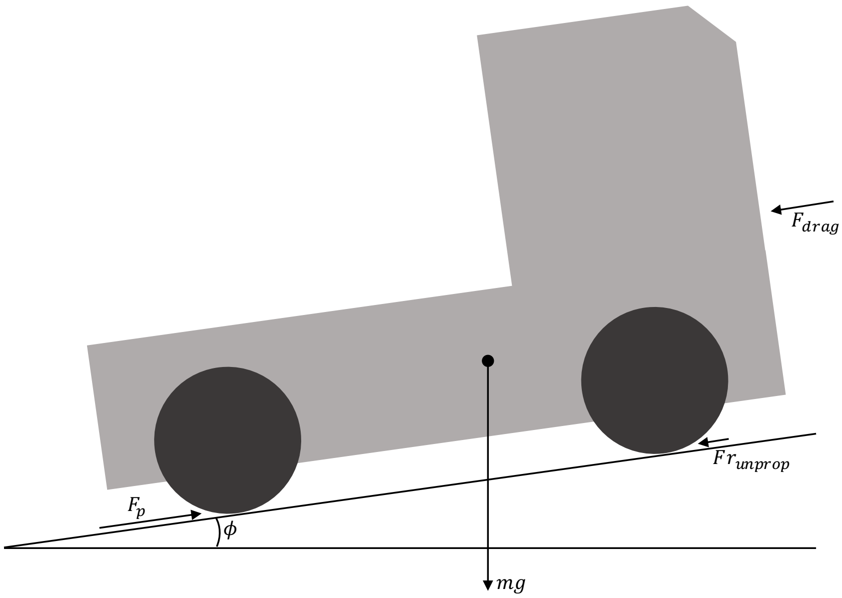

Figure 2.12: Model of complete vehicle.

Rolling resistance unpropelled wheels:

$$Fr_{unprop} = RRC \cdot F_{zunprop} \tag{2.23}$$

Gravitational component in direction of travel:

$$F_g = m \cdot g \cdot sin(\phi) \tag{2.24}$$

Air drag:

$$F_{drag} = \frac{c_d \cdot \rho \cdot A \cdot v^2}{2} \tag{2.25}$$

Collection of resilience forces acting on complete vehicle:

$$F_x = Fr_{unprop} + F_g + F_{drag} \tag{2.26}$$

Differential equation of motion in direction of travel[5]:

$$\dot{v} \cdot m = F_p - F_x \tag{2.27}$$

2.4 Backlash

Backlash occurs in the drivetrain when torsional torque change direction and is located between gears, dog clutches and splines. The size of the backlash will be gear ratio dependent because backlash located after gearbox will have greater influence to the total backlash on low gears than on high gears. Backlash will also vary with temperature and wear over time. The impact of the backlash is an abrupt torque

change when gap is traversed and will make period of oscillations longer when gap traverse is present.[7]
The authors opinion is that backlash may preferably be divided in two parts according to figure 2.13. Algorithms for measuring total backlash is already present within TECU and will not be handled by this thesis. Here follows a proposal of possible improvement by interconnecting gearbox ratio with measured backlash to get a greater data set for filtering.

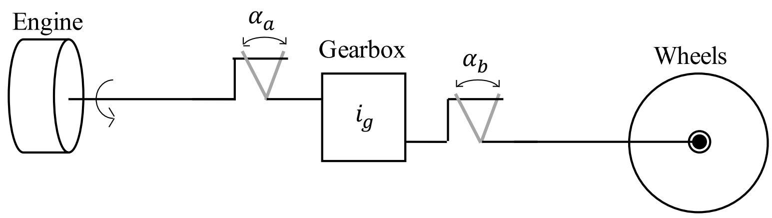

Figure 2.13: Gear ratio dependent backlash.

The theory assumes that the total backlash distributed thorough the drive line can be lumped in two parts, before and after gearbox with following reationship:

$$\alpha = \alpha_a + i_g \alpha_b \tag{2.28}$$

α, i_g is calculated by the TECU but α_a, α_b is initially unknown. Hence there is a need of two equations to determine the unknowns. This can be accomplished by preforming measurement on two different gears. The number of the gear is hereby denoted with a subscript of the gear measured, for example total backlash on gear nr.1 and 3 and it's corresponding gear ratio:

$$\alpha_1 = \alpha_a + i_{g1} \alpha_b \tag{2.29}$$

$$\alpha_3 = \alpha_a + i_{g3} \alpha_b \tag{2.30}$$

Eq(3.7) and (3.8) can be combined and rearranged in order to solve for α_a and α_b

$$\alpha_a = \frac{\alpha_1 - i_1}{1 - \frac{i_1}{i_3}} \tag{2.31}$$

$$\alpha_b = \frac{\alpha_1 - \alpha_3}{i_1 - i_3} \tag{2.32}$$

Eq(3.9) and (3.10) for calculating α_a and α_b apply for every gear by changing subscript denotation 1 and 3 to previous gear and current gear for example. Measurement should be preformed multiple times so that a mean value of α_a and α_b can be achieved. Uncertainties for described method is that the size of the backlash may differ depending on the power path through gearbox, direct drive or through counter shaft. Same uncertainty apply to range gear. This should be investigated by implementing proposed method in a test truck and thereafter analysing variation of results. Backlash theory is not verified in this thesis.

2.5 Signals and parameters

Description	**Variable**	**Init. source**	**Freq.[Hz]**	**Delay[ms]**
Engine speed	ω_e	EECU - CAN		
Engine torque	M_{efuel}	EECU - CAN		
Engine friction	$M_{efriction}$	EECU - CAN		
Engine brake torque	M_{ebrake}	EECU - CAN		
Wheel speed	ω_w	EBS - CAN		
Counter shaft speed	ω_{gc}	TECU	>1kHz	Speed dep.
Main shaft speed	ω_{gm}	TECU	>1kHz	Speed dep.
Current gear	$gear$	TECU		Not critical
Split gear ratio	i_{gs}	TECU		Not critical
Main gear ratio	i_{gm}	TECU		Not critical
Range gear ratio	i_{gr}	TECU		Not critical
Clutch position	l_{clutch}	TECU		
Vehicle mass	m	EBS/TECU		Not critical

For complete table, see Appendix 1

Parameter	**Notation**	**Source**
Engine inertia	J_e	Volvo database
Clutch inertia	J_c	Volvo database
Input shaft inertia	J_{gi}	Volvo database
Counter shaft inertia	J_{gc}	Volvo database
Main shaft inertia	J_{gm}	Volvo database
Output shaft inertia	J_{go}	Volvo database
Propeller shaft inertia	J_p	Volvo database
Final drive inertia	J_f	Volvo database
Drive shaft inertia	J_d	Volvo database
Wheel inertia	J_w	Volvo database
Clutch spring constant	k_c	Volvo database
Clutch damping constant	b_c	Volvo database
Propeller shaft spring constant	k_p	Volvo database
Propeller shaft damping constant	b_p	Volvo database
Drive shaft spring constant	k_d	Volvo database
Drive shaft damping constant	b_d	Volvo database

3

Complete Models for simulation

3.1 Description of models

Drivetrain can be divided by clutch, split transmission dog clutch and main transmission dog clutch. Range gear doesn't have a neutral state and crawler gear is modelled as an extra split gear.

Intermediate models:

- e-w: Engine to wheels
- slip: Estimator for slipping clutch
- e: Engine
- c-w: Clutch to wheels
- e-s: Engine to split
- s-w: Split to wheels
- c-s: Clutch to split
- e-m: Engine to main
- m-w: Main dog clutch to wheels
- s-m: Countershaft (split dog clutch to main dog clutch)

Types of connections between intermediate models:
The separated models can be used when gearshift occurs to calculate torque and speed of input/output shaft in gearbox. Separation of models can also be utilised to model backlash of drivetrain.

Clutch	Split	Main	Models	Constraints
x	x	x	e-w	
x	x	x	slip	
o	x	x	e, c-w	$M_c = 0$
x	o	x	e-s, s-w	$M_{gs} = 0$
o	o	x	e, c-s, s-w	$M_c = M_{gs} = 0$
x	x	o	e-m, m-w	$M_{gm} = 0$
o	x	o	e , c-m, m-w	$M_c = M_{gm} = 0$
x	o	o	e-s, s-m, m-w	$M_{gs} = M_{gm} = 0$
o	o	o	e, c-s, s-m, m-w	$M_c = M_{gs} = M_{gm} = 0$

Table 3.1: Possible configurations of model split

Due to limited time, focus of the work are set on complete "engine to wheels" model and split by clutch, "Engine" and "Clutch to wheels models".

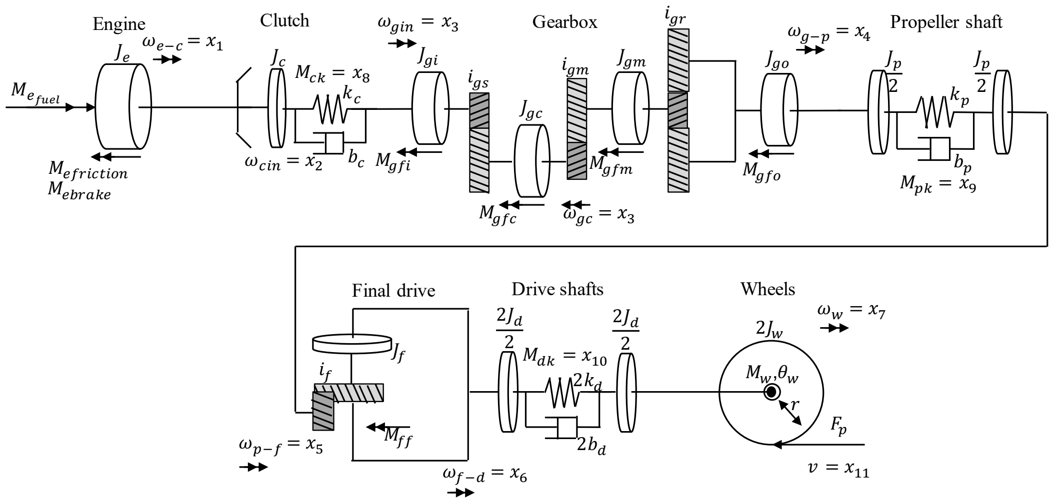

Figure 3.1: Total model. The x-variables shows locations of all state variables that may be utilised by intermediate models

States
Figure 4.1 visualises states needed for modeling all models previously described. However, the specific models utilises just the states within the boundary of the specific model itself and in some cases state reduction is applied within the boundaries.

3.2 Model from engine to wheels (e-w)

when gear is engaged, the included components are connected by following relationships:

$$
\begin{array}{ll}
\omega_e = \omega_{cin} = \omega_{e-c} & M_{eout} = M_{c_in} \\
\omega_{cout} = \omega_{gin} = \omega_{c-g} & M_{cout} = M_{gin} \\
\omega_{gout} = \omega_{pin} = \omega_{g-p} & M_{gout} = M_{pin} \\
\omega_{pout} = \omega_{fin} = \omega_{p-f} & M_{pout} = M_{fin} \\
\omega_{fout} = \omega_{din} = \omega_{f-d} & M_{fout} = M_{din} \\
\omega_{dout} = \omega_{win} = \omega_{d-w} & M_{dout} = M_{win}
\end{array}
$$

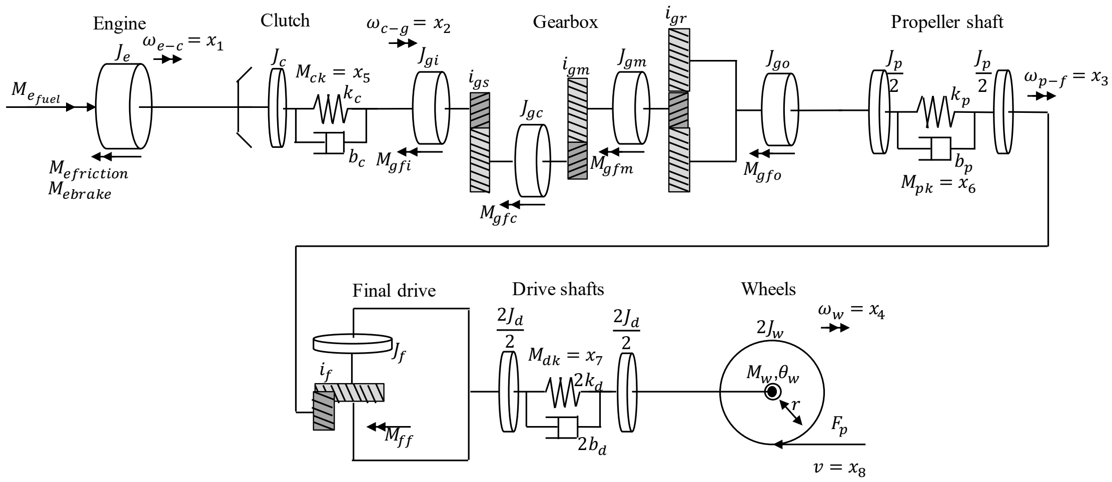

Figure 3.2: Model with gear engaged.

The system of 8 states:

$\omega_{e-c} = x_1 \quad M_{ck} = x_5$
$\omega_{c-g} = x_2 \quad M_{pk} = x_6$
$\omega_{p-f} = x_3 \quad M_{dk} = x_7$
$\omega_w = x_4 \quad v = x_8$

Differential equations for states:

$$\dot{\omega}_{e-c} = \dot{x}_1 = \frac{M_{efuel} - M_{efriction} - M_{ebrake} - x5 - M_{cb}}{J_e + J_c}$$

$$\dot{\omega}_{c-g} = \dot{x}_2 = \frac{x_5 + M_{cb} - M_{gftotx2} - \frac{x_6 + M_{pb}}{i_{gs} \cdot i_{gm} \cdot i_{gr}}}{J_{x2}}$$

$$\dot{\omega}_{p-f} = \dot{x}_3 = \frac{x_6 + M_{pb} - \frac{M_{ff}}{i_f} - \frac{x_7 + M_{db}}{i_f}}{J_x 3}$$

$$\dot{\omega}_w = \dot{x}_4 = \frac{x_7 + M_{db} - r \cdot F_p - e \cdot F_{zprop}}{J_d + 2J_w}$$

$$\dot{M}_{ck} = \dot{x}_5 = k_c(x_1 - x_2)$$

$$\dot{M}_{pk} = \dot{x}_6 = k_p(\frac{x_2}{i_{gs} \cdot i_{gm} \cdot i_{gr}} - x_3))$$

$$\dot{M}_{dk} = \dot{x}_7 = k_d(\frac{x_3}{i_f} - x_4)$$

$$\dot{v} = \dot{x}_8 = \frac{F_p - F_x}{m}$$

Total inertia at x_2:

$$J_{x2} = J_{gi} + \frac{J_{gc}}{i_{gs}^2} + \frac{J_{gm}}{(i_{gs} + i_{gm})^2} + \frac{J_{go} + \frac{J_p}{2}}{(i_{gs} + i_{gm} + i_{gr})^2}$$

Total gearbox friction gathered at x_2:

$$M_{gftotx2} = M_{gfi} + \frac{M_{gfc}}{i_{gs}} + \frac{M_{gfm}}{i_{gs} \cdot i_{gm}} + \frac{M_{gfo}}{i_{gs} \cdot i_{gm} \cdot i_{gr}}$$

Total inertia at x_3:

$$J_{x3} = \frac{J_p}{2} + \frac{J_f + J_d}{i_f^2}$$

$F_x = m \cdot g \cdot sin\phi + Fr_{unprop} + Fr_{prop} + \frac{c_d \cdot \rho \cdot A \cdot v^2}{2}$
$F_p = CCX \cdot F_{zprop} \cdot \frac{r*x4 - x8}{|r*x4|}$
$M_{cb} = b_c(x_1 - x_2)$
$M_{pb} = b_p(\frac{x_2}{i_{gs} \cdot i_{gm} \cdot i_{gr}} - x_3)$
$M_{db} = b_d(\frac{x_3}{i_f} - x_4)$

3.2.1 Validation of model

As reference for model validation, existing measurement data was gathered from previous road testing. The specifications of the truck were compiled and spring constants, inertia, gear ratio e.tc. was derived. The truck had an external measurement system which measured axle speeds and also propeller shaft torque through an torque transducer fitted between output shaft of the gearbox and propeller shaft. All external measurements were recorded at a sample rate of 160kHz and therefore measurements were assumed to have no time delay. The data was examined, cut in length, filtered and re sampled to 100Hz in analysis software FAMOS and thereafter exported in .csv format for simulation and comparison with model.

To validate equations a Python program was written that preforms simulations by iterating and solving the differential equations of the model. The first iteration needs to be set up with initial values for each state. Thereafter, each iteration estimates new initial values for next iteration. The iterations are set to be preformed with the same delta time as the sample rate of the measurement data.

Simulations were preformed for the gear engaged drivetrain with closed clutch and comparison were made with measurements to verify equations, stiffness, inertia and parameter values. To be able to set initial values for simulation, the measurement data was examined and the sequence was set to start when clutch connects after gear shift. When clutch is open the torques through drivetrain is low and hence initial values for torques could be initiated to zero without major deviations to measurement. Initial speeds were taken directly from the measurements. Input for the model is engine torque and engine friction torque. Unknown tire parameters were experimentally calibrated. The parameters that were adjusted and their derived value follows:

Parameter	**Value**	**Description**
r	0.526[m]	Tire radius
CCX	15[1]	Tire longitudinal slip coefficient
RRC	0.0105[1]	Rolling resistance coefficient

Simulations has been preformed on gear number 6 and 12. Gear nr.6 was chosen because it was the lowest gear with a high amount of torque applied. The accuracy of the torque transmitted from EECU is better at high torques and is therefore preferred as a reference for the simulation. Gear nr 12 is the highest available gear, and is also direct drive through transmission which makes it a good choice for validating the model for low gear ratios.

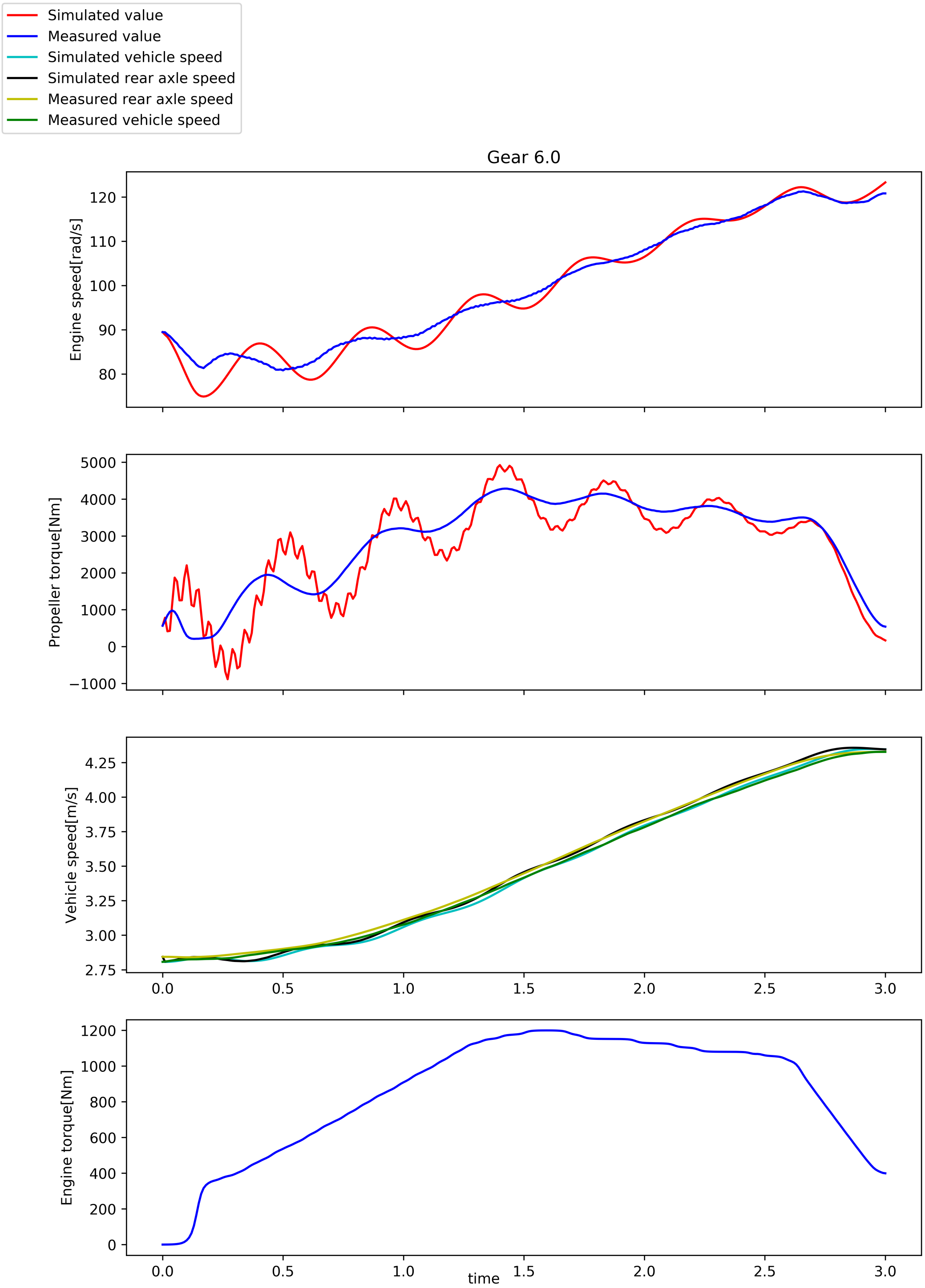

Figure 3.3: Simulation preformed for 6:th gear and comparison to measurement from test truck.

Comparison for gear nr. 6 shows good correlation for mean speeds, torques and period of large scale oscillations. The deviation for the amplitude and phase shift of the large scale oscillations is probably due to initial torque value deviations.

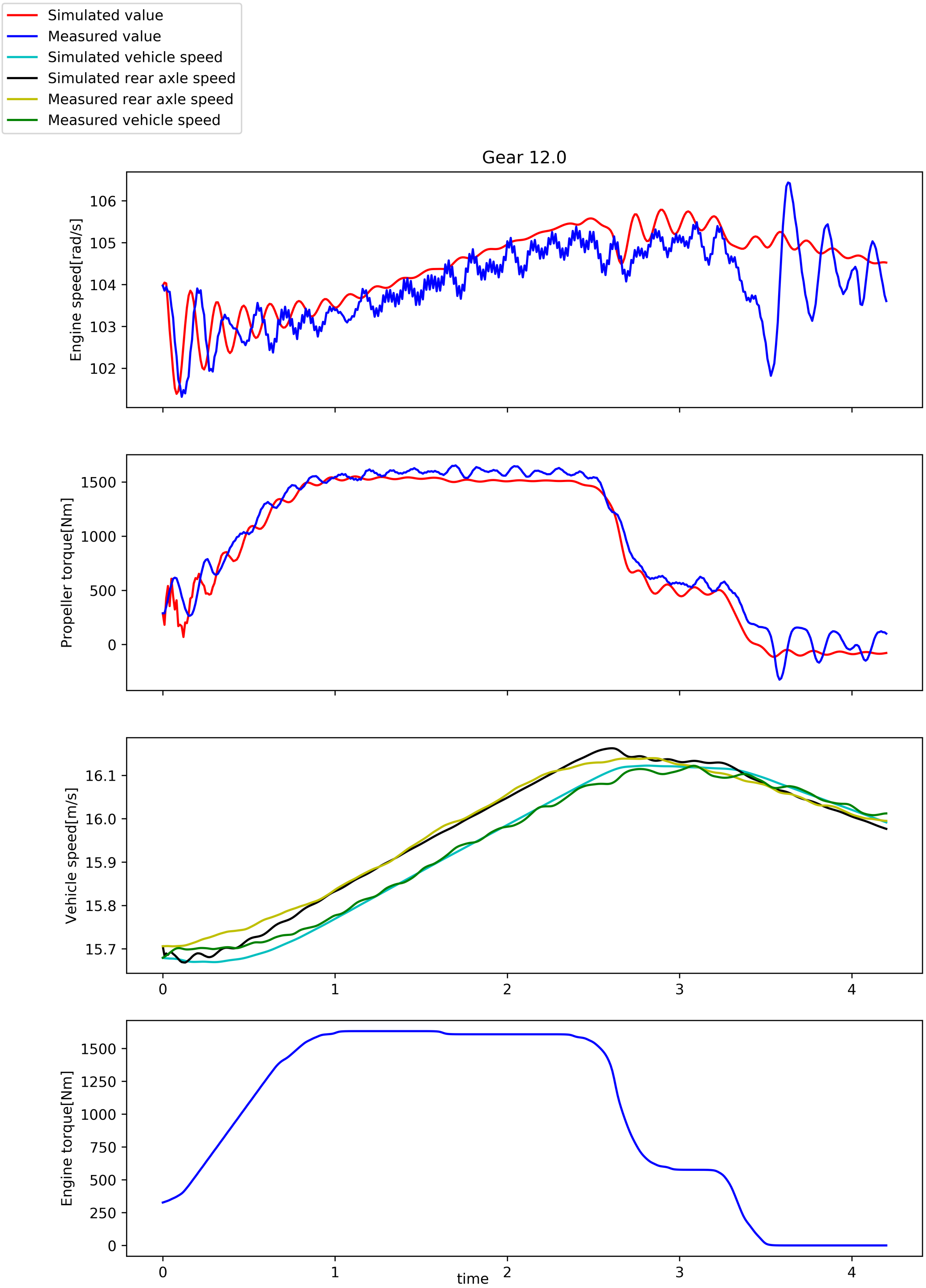

Figure 3.4: Simulation preformed for 12:th gear and comparison to measurement from test truck.

Also gear nr. 12 shows good correlation for mean speeds, torques and period of oscillations.

3.2.2 Reduced model of engine to wheels (e-w)

Experiments were preformed in order to examine if model could be reduced without major loss of accuracy. Reduction of states enhances computational speeds and there is also great benefit in reducing complexity for both understanding and troubleshooting. By adapting spring stiffness in propeller shaft and drive shaft it could be proved that the behaviour of large scale oscillations is not influenced by lumping propeller shaft stiffness and drive shaft stiffness as one as long as final drive ratio is included in the calculation. The model can therefore be reduced to following 6-state system:

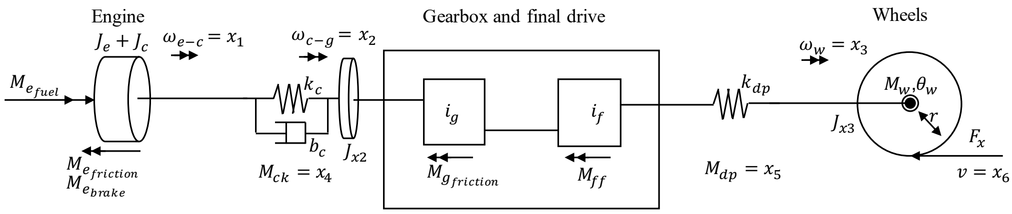

Figure 3.5: Reduced model of drivetrain.

The system of 6 states:

$\omega_{e-c} = x_1 \quad M_{ck} = x_4$
$\omega_{c-g} = x_2 \quad M_{dp} = x_5$
$\omega_w = x_3 \quad v = x_6$

Differential equations for states:

$$\dot{\omega}_{e-c} = \dot{x}_1 = \frac{M_{efuel} - M_{efriction} - M_{ebrake} - x4 - M_{cb}}{J_e + J_c}$$

$$\dot{\omega}_{c-g} = \dot{x}_2 = \frac{x_4 + M_{cb} - M_{gfx2} - \frac{x_5}{i_{tot}}}{J_{x2}}$$

$$\dot{\omega}_w = \dot{x}_3 = \frac{x_5 - r \cdot F_p - e \cdot F_{zprop}}{J_{x3}}$$

$$\dot{M}_{ck} = \dot{x}_4 = k_c(x_1 - x_2)$$

$$\dot{M}_{dp} = \dot{x}_5 = k_{dp}\left(\frac{x_2}{i_{tot}} - x_3\right)$$

$$\dot{v} = \dot{x}_6 = \frac{F_p - F_x}{m}$$

Spring constant driveshaft and propeller shaft:

$$k_{dp} = \frac{1}{\frac{1}{k_d} + \frac{1}{k_p \cdot i_f}}$$

Total gear ratio:

$$i_{tot} = i_{gs} \cdot i_{gm} \cdot i_{gr} \cdot i_f$$

Total inertia at x_2:

$$J_{x2} = J_{gi} + \frac{J_{gc}}{i_{gs}^2} + \frac{J_{gm}}{(i_{gs}+i_{gm})^2} + \frac{J_{go}+J_p}{(i_{gs}+i_{gm}+i_{gr})^2} + \frac{J_f+J_d}{(i_{gs}+i_{gm}+i_{gr}+i_f)^2}$$

Total inertia at x_3:

$$J_{x3} = 2 \cdot J_w + J_d$$

Total gearbox friction gathered at x_2:
$M_{gfx2} = M_{gfi} + \frac{M_{gfc}}{i_{gs}} + \frac{M_{gfm}}{i_{gs} \cdot i_{gm}} + \frac{M_{gfo}}{i_{gs} \cdot i_{gm} \cdot i_{gr}} + \frac{M_{ff}}{i_{gs} \cdot i_{gm} \cdot i_{gr} \cdot i_f}$

$F_x = m \cdot g \cdot sin\phi + Fr_{unprop} + Fr_{prop} + \frac{c_d \cdot \rho \cdot A \cdot v^2}{2}$
$F_p = CCX \cdot F_{zprop} \cdot \frac{r \cdot x3 - x6}{|r \cdot x3|}$
$M_{cb} = b_c(x_1 - x_2)$

3.2.3 Validation of reduced model

The reduced model is hereby verified with the developed Python program in the same way as the complete model. Due to state reduction, information about propeller shaft torque is lost. Propeller shaft torque is instead estimated to be $M_p = \frac{M_{dp}}{i_f}$.

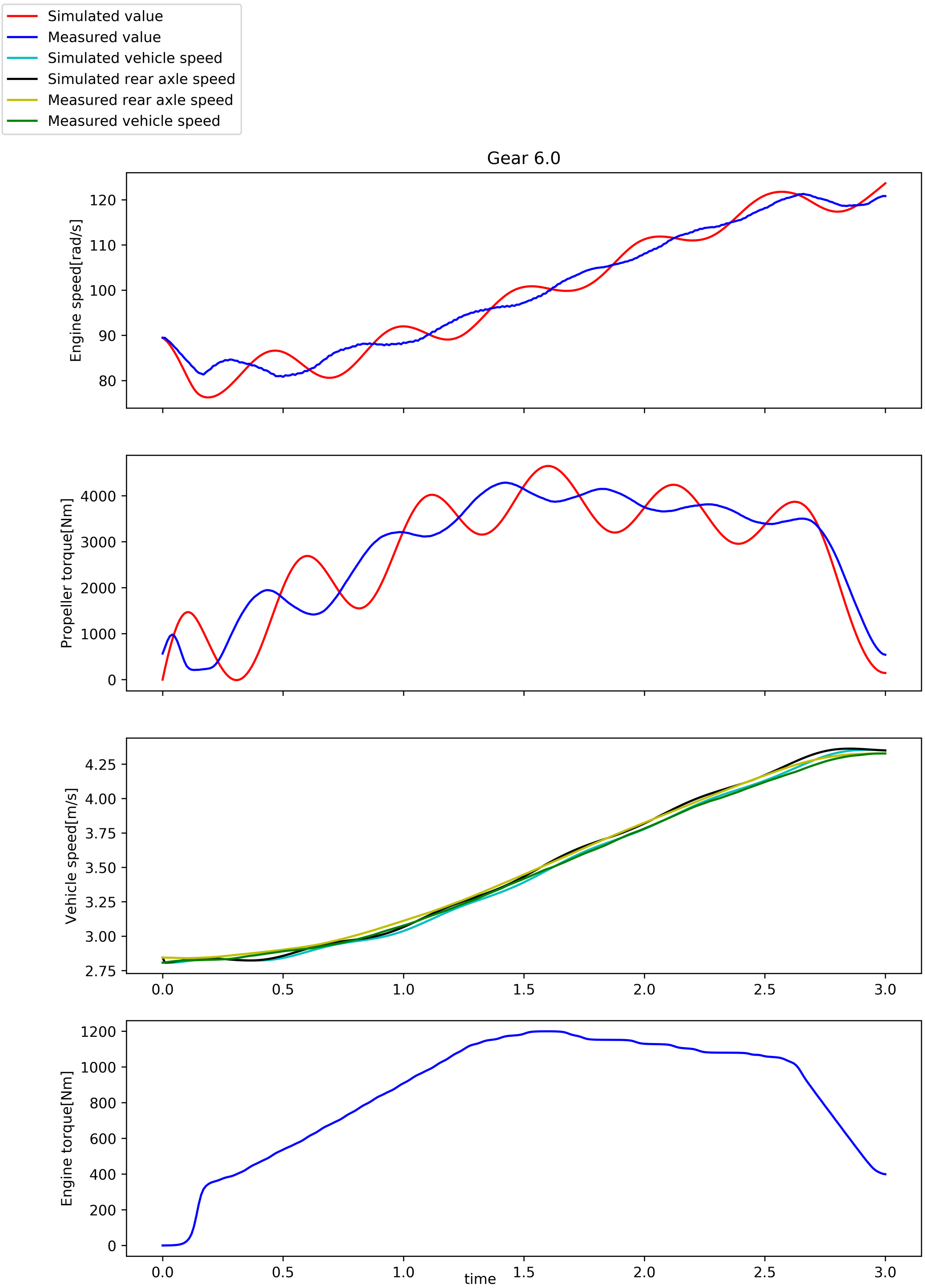

Figure 3.6: Simulation preformed for 6:th gear with reduced model and comparison to measurement from test truck.

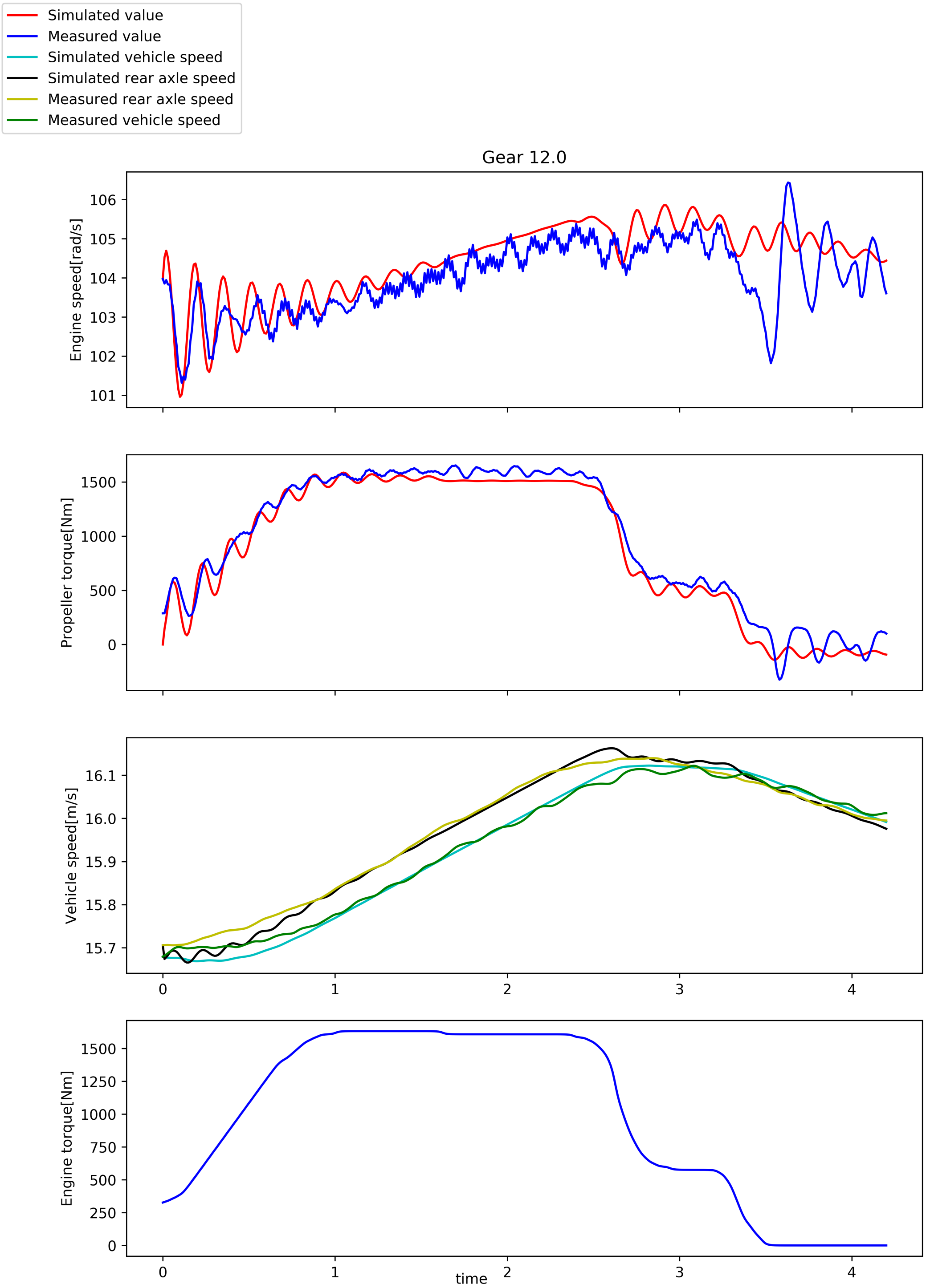

Figure 3.7: Simulation preformed for 12:th gear with reduced model and comparison to measurement from test truck.

The simulations show that the reduced model is adequate for describing large scale oscillations and response for the system. There is still an initial value problem that needs to be solved to get phase and amplitude aligned.

3.3 Model for engine speed while clutch is disconnected (e)

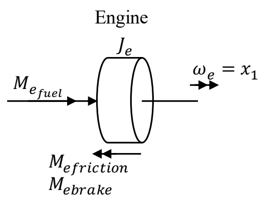

Figure 3.8: Model for engine with disconnected clutch.

$$\dot{\omega}_e = \dot{x}_1 = \frac{M_{efuel} - M_{efriction} - M_{ebrake}}{J_e} \tag{3.1}$$

3.4 Model from clutch to wheels (c-w)

when gear is engaged, the included components are connected by following relationships:

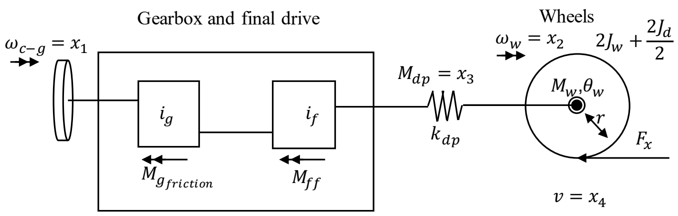

Figure 3.9: Model from clutch to wheels.

States are reduced into following four states:

$\omega_{c-g} = x_1 \quad M_{dp} = x_3$
$\omega_w = x_2 \quad v = x_4$
Differential equations for states:

$\dot{\omega}_{c-g} = \dot{x}_1 = \frac{-M_{gfx2} - \frac{x_3}{i_{tot}}}{J_{x2}}$

$\dot{\omega}_w = \dot{x}_2 = \frac{x_3 - r \cdot F_p - e \cdot F_{zprop}}{J_{x3}}$

$\dot{M}_{dp} = \dot{x}_3 = k_{dp}(\frac{x_1}{i_{tot}} - x_2))$

$\dot{v} = \dot{x}_4 = \frac{F_p - F_x}{m}$
Spring constant driveshaft and propeller shaft:
$k_{dp} = \frac{1}{\frac{1}{k_d} + \frac{1}{k_p \cdot i_f}}$

Total gear ratio:

$i_{tot} = i_{gs} \cdot i_{gm} \cdot i_{gr} \cdot i_f$

Total inertia at x_1:
$J_{x1} = J_c + J_{gi} + \frac{J_{gc}}{i_{gs}^2} + \frac{J_{gm}}{(i_{gs}+i_{gm})^2} + \frac{J_{go}+J_p}{(i_{gs}+i_{gm}+i_{gr})^2} + \frac{J_f+J_d}{(i_{gs}+i_{gm}+i_{gr}+i_f^2})^2$

Total inertia at x_2:
$J_{x2} = 2 \cdot J_w + J_d$

Total gearbox friction gathered at x_1:
$M_{gfx1} = M_{gfi} + \frac{M_{gfc}}{i_{gs}} + \frac{M_{gfm}}{i_{gs} \cdot i_{gm}} + \frac{M_{gfo}}{i_{gs} \cdot i_{gm} \cdot i_{gr}} + \frac{M_{ff}}{i_{gs} \cdot i_{gm} \cdot i_{gr} \cdot i_f}$

$F_x = m \cdot g \cdot sin\phi + Fr_{unprop} + Fr_{prop} + \frac{c_d \cdot \rho \cdot A \cdot v^2}{2}$
$F_p = CCX \cdot F_{zprop} \cdot \frac{r \cdot x2 - x4}{|r \cdot x2|}$

3.5 Slipping clutch torque estimator (slip)

There is many unknown variables affecting the torque transfer through a slipping clutch. The work would be too comprehensive to develop a decent predictive model for torque transfer. Instead, an estimator is here presented that uses transmission input speed to determine the tension build up with the use of the spring coefficient of the driveline.

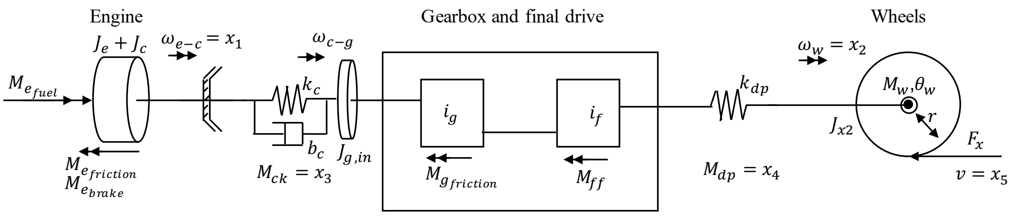

Figure 3.10: Estimator for slipping clutch.(slip)

Input shaft speed is hereby gathered from measured value Differential equations for states and measured input shaft speed ω_{c-g}:

$\dot{\omega}_{e-c} = \dot{x}_1 = \frac{M_{efuel} - M_{efriction} - M_{ebrake} - x3 - M_{cb}}{J_e + J_c}$

$\dot{\omega}_w = \dot{x}_2 = \frac{x_4 - r \cdot F_p - e \cdot F_{zprop}}{J_{x2}}$

$\dot{M}_{ck} = \dot{x}_3 = k_{dp} \frac{(\frac{\omega_{c-g}}{i_{tot}} - x_2)}{i_{tot}}$

$\dot{M}_{dp} = \dot{x}_4 = k_{dp}(\frac{\omega_{c-g}}{i_{tot}} - x_2)$

$\dot{v} = \dot{x}_5 = \frac{F_p - F_x}{m}$

Spring constant driveshaft and propeller shaft:
$k_{dp} = \frac{1}{\frac{1}{k_d} + \frac{1}{k_p \cdot i_f}}$

Total gear ratio:

$i_{tot} = i_{gs} \cdot i_{gm} \cdot i_{gr} \cdot i_f$

Total inertia at x_3:
$J_{x2} = 2 \cdot J_w + J_d$

$F_x = m \cdot g \cdot sin\phi + Fr_{unprop} + Fr_{prop} + \frac{c_d \cdot \rho \cdot A \cdot v^2}{2}$
$F_p = CCX \cdot F_{zprop} \cdot \frac{r \cdot x3 - x6}{|r \cdot x3|}$

3.6 Transition conditions

3.6.1 Clutch

The position of the clutch actuator is measured by the TECU and will be used for deciding which model to use. Slip point of clutch is located at position zero. Positive travel disengages clutch.

Algorithm 2: Constraints on clutch torque by clutch position when gearbox is in gear

if *Clutch position > 0* **then**
  $M_c = 0$;
  Separate Engine & Clutch to wheels models (e),(c-w)
else if *0 >= Clutch position > no slip position* **then**
  Slipping clutch torque estimator (slip)
else
  Engine to wheels (e-w)

3.7 Validation of combined models with transition conditions

As seen while simulating "Engine to Wheels"(e-w) model, the main problem and cause of deviation of the simulation are unknown initial values. Speeds are measured but torque isn't and needs to be estimated. Therefore there is a great benefit of preforming state reduction as shown in chapter 3.2.2 and hence reducing the initial value problem. When clutch is open, initial condition for clutch torque becomes trivial, zero. When clutch starts to slip, the torque is then calculated from the stiffness of the driveline. When clutch goes to fully closed the estimated values are used as initial values for simulating the connected model (e-w). Intermediate models were connected with described transition conditions in chapter 3.6 and simulated and compared with measured test data.

Note that there were no available measured clutch torque in data for figure 3.11. Denoted clutch torque is measured propeller shaft torque divided by gear ratio in the gearbox and therefore deviates because of influencing friction and inertia inside the gearbox. It is still the best available estimate for comparison with simulation.

Green area in figure 3.11 utilises separate models "Engine" and "Clutch to wheels" while clutch is open and clutch torque therefore is zero. When clutch then travels towards engagement, model shifts to "Slipping clutch torque estimator" with initial conditions from previous models. When clutch no longer slips, simulation switches to "Engine to Wheels" model, also with forwarded initial conditions.

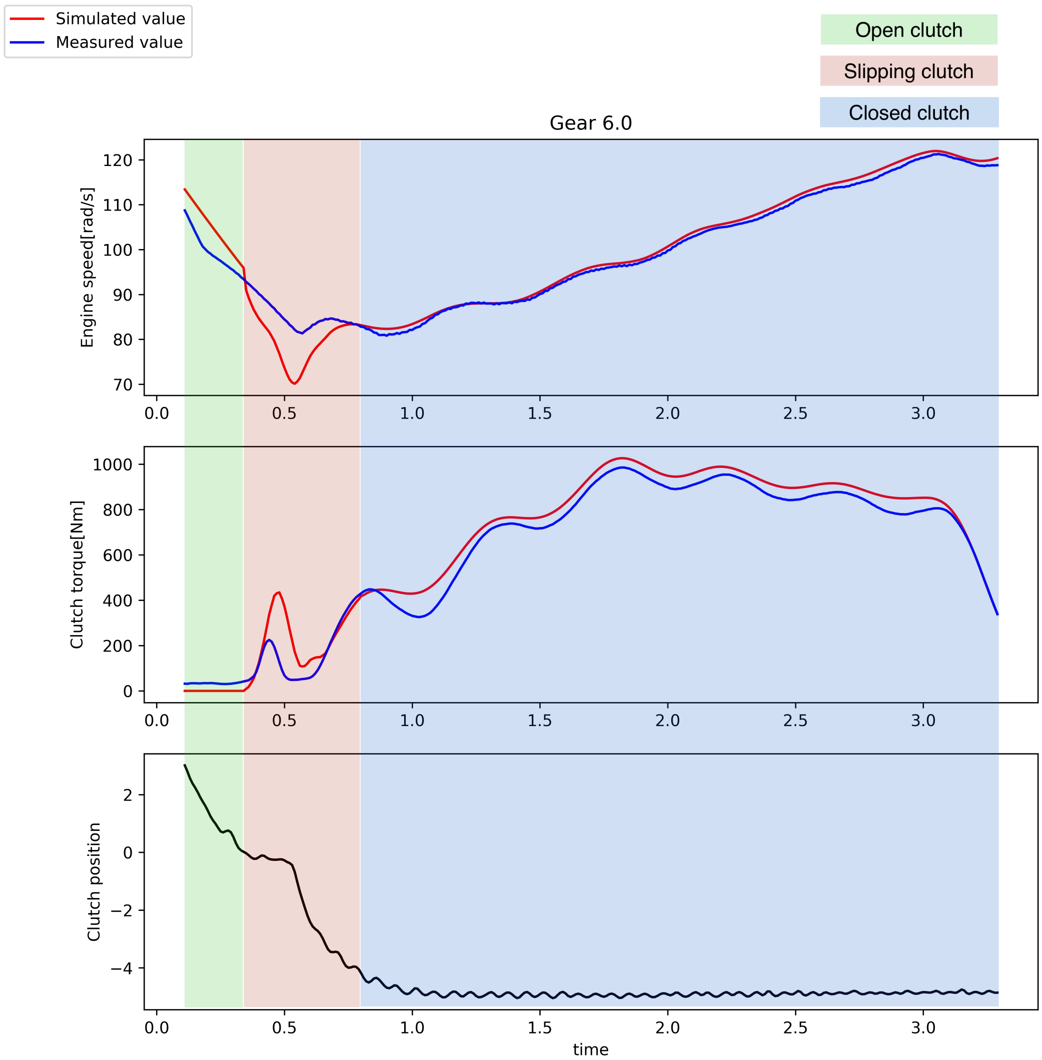

Figure 3.11: Simulation with models with separated, slipping and closed clutch

Simulations starting with open clutch and estimate of clutch torque through slipping phase shows a significantly better correlation with measurements than previous simulations preformed due to less deviation of initial values.

4

Implementation in a real time system

Simulations preformed previously to verify models has been utilising implicit solution methods for solving the differential equations numerically.

Implicit solution methods
The solution utilises information both from current time step and from next time step by additional computation. Usually by fix point iteration. Implicit solution methods produces a stable but computationally demanding solution for each time step.

Explicit solution methods
The solution utilises information known at current time step to make an explicit prediction for next time step. Explicit solution methods are computationally less demanding for each time step but may result in an unstable solution.
Stability of solutions is heavily influenced by step size. Decreased step size improves stability.[10]

4.1 Explicit forward Euler method

The forward Euler method is simple to implement in programming and computationally efficient for each time step. It is therefore hereby tested to analyse if the method may be used to produce a stable prediction of 500ms.

The test is preformed with the same simulation as in figure 3.11 and additionally, a forward Euler solution is started at time t = 2.0.

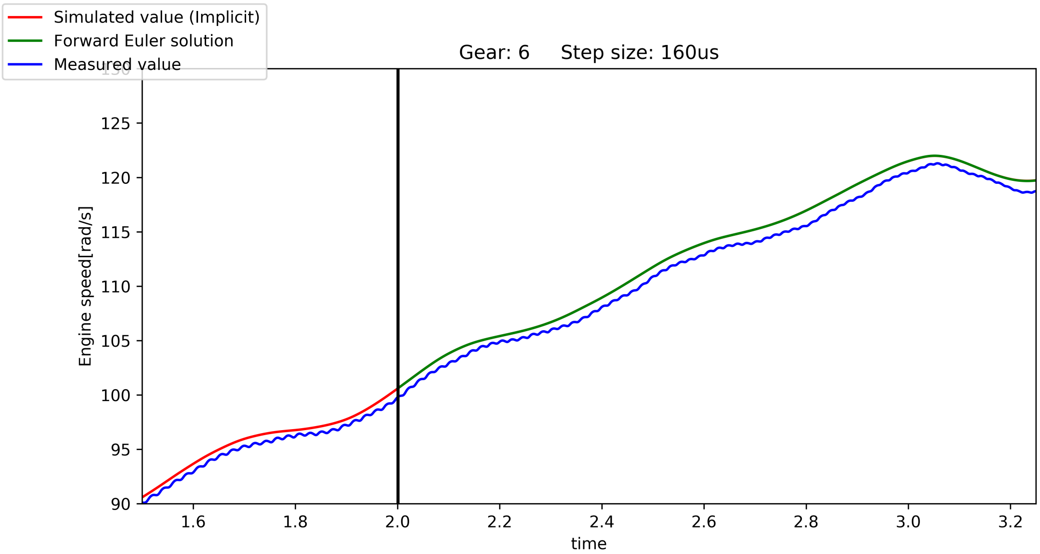

Figure 4.1: Simulation with forward Euler solution starting at t=2.0 and step size 160μs

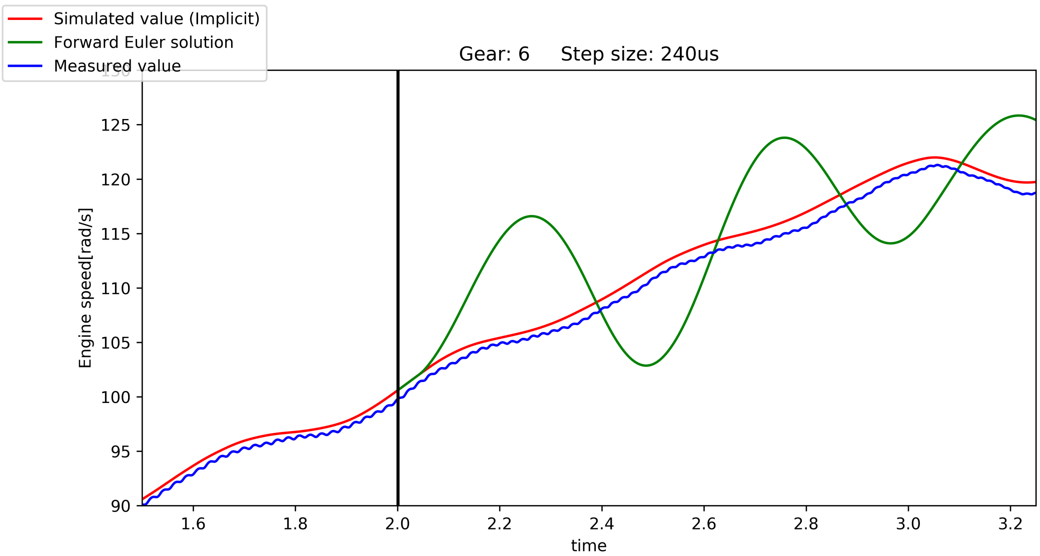

Figure 4.2: Simulation with forward Euler solution starting at t=2.0 and step size 240μs

Figure 4.1 and 4.2 shows that the explicit forward Euler solution needs short time steps in order to produce a stable result. Conclusion is that to be able to utilise forward Euler solution, the step size needs to be 160μs or less or the complexity of the model needs to be further reduced.
Hereafter one state is reduced by removing the wheel slip dynamics from the previous reduced "e-w" model according to following description:

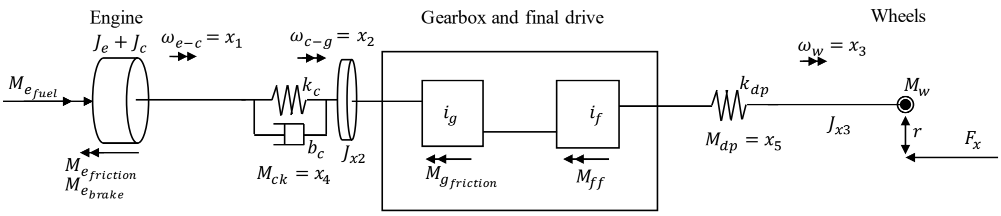

Figure 4.3: Reduced model of drivetrain without wheel slip.

States:
$\omega_{e-c} = x_1 \quad M_{ck} = x_4$
$\omega_{c-g} = x_2 \quad M_{dp} = x_5$
$\omega_w = x_3$
Differential equations for states:

$$\dot{\omega}_{e-c} = \dot{x}_1 = \frac{M_{efuel} - M_{efriction} - M_{ebrake} - x4 - M_{cb}}{J_e + J_c}$$

$$\dot{\omega}_{c-g} = \dot{x}_2 = \frac{x_4 + M_{cb} - M_{gfx2} - \frac{x_5}{i_{tot}}}{J_{x2}}$$

$$\dot{\omega}_w = \dot{x}_3 = \frac{x_5 - F_x \cdot r}{J_{x3} + m \cdot r^2}$$

$$\dot{M}_{ck} = \dot{x}_4 = k_c(x_1 - x_2)$$

$$\dot{M}_{dp} = \dot{x}_5 = k_{dp}(\frac{x_2}{i_{tot}} - x_3)$$

Spring constant driveshaft and propeller shaft:
$$k_{dp} = \frac{1}{\frac{1}{k_d} + \frac{1}{k_p \cdot i_f}}$$

Total gear ratio:
$$i_{tot} = i_{gs} \cdot i_{gm} \cdot i_{gr} \cdot i_f$$

Total inertia at x_2:
$$J_{x2} = J_{gi} + \frac{J_{gc}}{i_{gs}^2} + \frac{J_{gm}}{(i_{gs} + i_{gm})^2} + \frac{J_{go} + J_p}{(i_{gs} + i_{gm} + i_{gr})^2} + \frac{J_f + J_d}{(i_{gs} + i_{gm} + i_{gr} + i_f)^2}$$

Total inertia at x_3:
$$J_{x3} = 2 \cdot J_w + J_d$$

Total gearbox friction gathered at x_2:
$$M_{gfx2} = M_{gfi} + \frac{M_{gfc}}{i_{gs}} + \frac{M_{gfm}}{i_{gs} \cdot i_{gm}} + \frac{M_{gfo}}{i_{gs} \cdot i_{gm} \cdot i_{gr}} + \frac{M_{ff}}{i_{gs} \cdot i_{gm} \cdot i_{gr} \cdot i_f}$$

$$F_x = m \cdot g \cdot sin\phi + Fr_{unprop} + Fr_{prop} + \frac{c_d \cdot \rho \cdot A \cdot (x_3 \cdot r)^2}{2}$$
$$M_{cb} = b_c(x_1 - x_2)$$

The reduced model without wheel slip is hereby solved with forward Euler solution starting at time t=2.0.

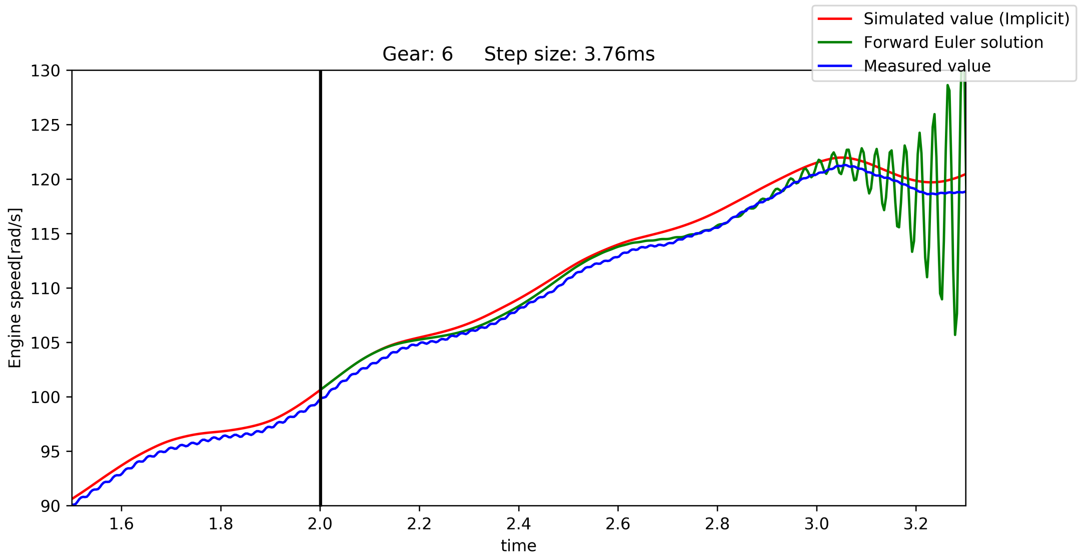

Figure 4.4: Simulation with forward Euler solution starting at t=2.0 and step size 3.68ms

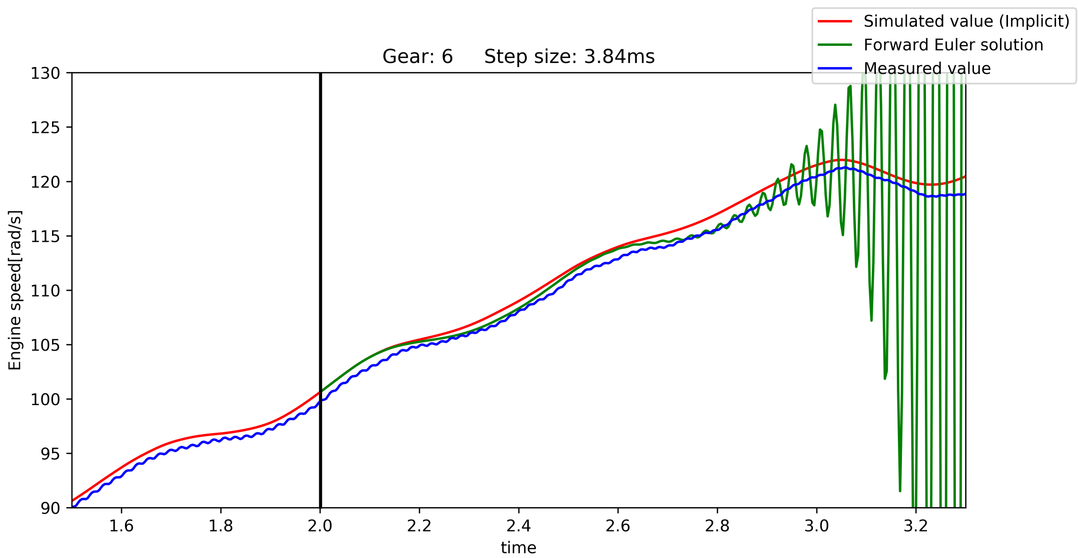

Figure 4.5: Simulation with forward Euler solution starting at t=2.0 and step size 3.84ms

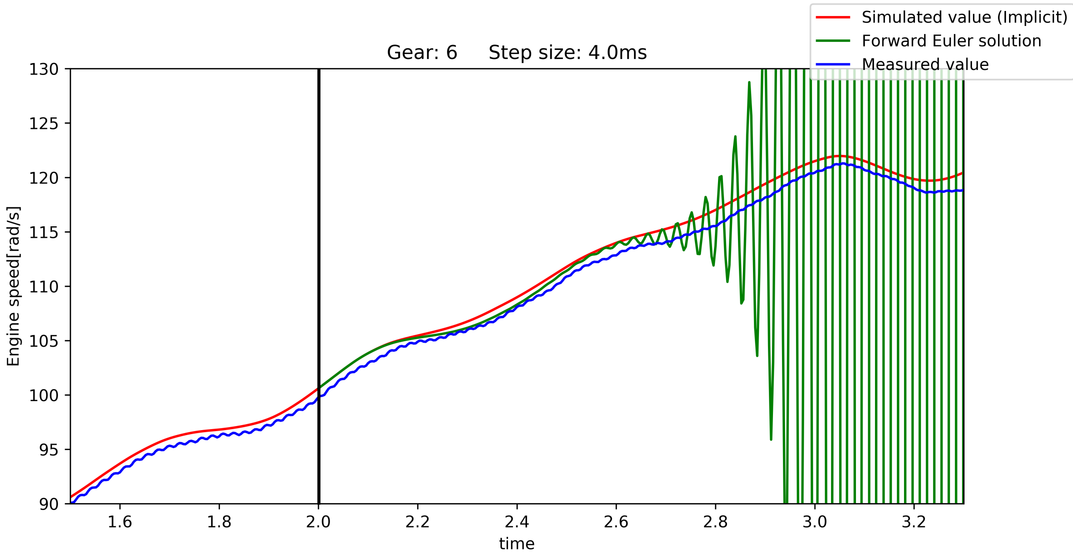

Figure 4.6: Simulation with forward Euler solution starting at t=2.0 and step size 4.0ms

Figure 4.3, 4.4 and 4.5 show that the time steps for forward Euler should be less than 4ms in order to predict 500ms. The reduction of wheel slip from the model results in an possible increment of step size with a factor of 22 but the curve also start so deviate because the wheel slip is ignored.

5

Results

This work presents models that can be used to estimate and predict behaviour of the driveline based on physical data available at Volvo. Transition conditions are used to switch between different models when clutch is actuated. The models show good correlation with measurements from a test truck when initial conditions can be determined. If explicit forward Euler solution are to be used within the TECU for prediction, the step size for iterations needs to be small. For the 6-state "engine to wheels" model including clutch flexibility, lumped propeller and driveshafts and wheel slip model, the step size must be 160μs or less. By reducing model to five states, hence removing wheel slip model, the step size can be increased 22 times to 3.6ms with loss of model precision.

6
Discussion

Driveline modeling is a very comprehensive topic. it is easy to underestimate the importance of reductions and simplifications. Function development by utilisation of mapping and variant cases tends to be a straight forward approach to start off but when variants and circumstantial effects grows, also the the size of the functions grow and gets complicated. Modelling of behaviour from the physical and mechanical properties is an approach that can reduce the number of values needed calibration and by that simplifying the variant specific tuning of the control algorithms.

This work presents models that can be used to estimate and predict behaviour of the driveline based on physical data available at Volvo. All parameters for the control algorithms is not yet available within the TECU. To be able to utilise the described algorithms, variant specific physical parameters needs to be implemented. An alternative method is to develop functions estimating the stiffness and inertia from actual driving by measuring torsion and first order oscillations of the driveline.

Results show that a a simplified model of the drivetrain with spring coefficient from propeller shaft and drive shaft lumped together replicates the behaviour from a test truck in an accurate manner when initial conditions is known and simulation occurs during a restricted interval of time. Therefore restrictions needs to be put in place when implementing model in TECU. The size of the tolerance of a prediction will grow with the length of the simulation because of integration error and is also accelerated by deviation of initial values. Therefore it is important to establish credible initial values to restart simulation as often as possible. For example it may be possible to determine clutch torque directly from engine torque under steady, high load conditions. There should therefore be a function in place analysing if initial values for next simulation should be from last iteration of model or from new estimations. The chosen method and step size for solving differential equations is also important aspects for achieving accuracy.
To execute the models in a real time system, there must be an ODE (ordinary differential equation) solver in place. The easiest solver to implement is the explicit forward Euler method. The downside is that the step size needs to be very small in order to produce a stable result. Chapter 4 describes the stability issues and step size needed to produce a stable prediction. Chapter 4 also describes how a further state reduction increases the length of the time step 22 times with kept stability. This reduction decreases the computational load by more than 22 times because of the increment in time between iterations and less computations at each iteration.

The downside with removing wheel slip model that the wheel speed is no longer torque dependent and will have a torque dependent deviation. Implicit methods demand more computations for each time step but can increase the length of the time steps which may be beneficial for the total computational load.

Methods of implementation in TECU were examined as follows. ASCET does not naively support matrix operations which complicates the process. Implementation could be preformed graphically by loops, deviates and iterations to produce a forward Euler method but will probably grow in complexity, especially with the short time steps needed for forward Euler. A preferred strategy would be to implement the equation system directly through ESDL-code and C-code snippets for matrix operations and test different methods for solving the ODE:s and comparing computational load.

Limiting factors for accuracy of driveline modeling:

- **Initial conditions:**
  The accuracy of initial condition greatly affect the simulation. It is important to have functions in place to determine initial values and their accuracy.

- **Method and step size chosen for solving differential equations:**
  As described in chapter 4, the solver needs to produce a stable result.

- **Unforeseen events:**
  Events that is hard to predicted like drivers input or irregularities of the road will affect the accuracy. Therefor, functions need to be in place to analyse credibility when the prediction is to be utilised.

- **Deviation of physical properties:**
  Properties like tire friction, air pressure and radius will affect the accuracy but with a prediction horizon of 500ms, the impact will be small.

7 Future work

- **Model in ESDL language with state space**
  Build the models in ASCET. While ASCET does not naively support matrix's, it can be done through ESDL-language combined with C-code snippets.

- **Test iteration methods**
  Test different iteration methods, both explicit and implicit and compare computational load.

- **Analyse deviation of model without wheel slip**
  Preform simulations for different gears with the reduced model without wheel slip and analyse results.

- **Filter input values**
  Input values should preferably to be filtered before applying to model because even minor deviations impose oscillations to the system. A problem with filtering is that it imposes a delay of the signal. Therefore work needs to be preformed in order to find the best compromise.

- **Implement physical properties in TECU**
  Implement vehicle specific properties in TECU for inertia's, spring coefficients and dampings.

- **Implement all intermediate models**
  Implement all intermediate models described in Chapter 3.1 to have a complete estimation/prediction of all included shafts in the gearbox during gear shifting.

- **Implement and test split backlash method**
  Implement method and preform tests to analyse deviation of measurements of method for split backlash as presented in chapter 2.4

- **Analyse torsion in frame an engine mounts**
  Analyse the size of torsional rotation/stiffness between engine/gearbox assembly and rear axle, or ground plane on a truck. What is the size of the resulting dynamics impact on the behaviour of the driveline?

- **Implement retarder**
  Implement transmission out retarder into model.

- **Multiple propelled axles**
  Implement multiple propelled axles into model to cover more variants.

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