2018

MODELING AND ANALYSIS FOR DRIVELINE JERK CONTROL

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MODELING AND ANALYSIS FOR DRIVELINE JERK CONTROL

By

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A REPORT

Submitted in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

In Mechanical Engineering

MICHIGAN TECHNOLOGICAL UNIVERSITY

2018

This report has been approved in partial fulfillment of the requirements for the Degree of MASTER OF SCIENCE in Mechanical Engineering.

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Acknowledgements

The report would not have been possible without the guidance of my advisor, Dr. Mahdi Shahbakhti and Dr. Darrell Robinette. Their ideas for this research have been extremely valuable. Dr. Mahdi Shahbakhti has also been extremely supportive throughout the course of my graduate studies and I am very thankful for this constant guidance and patience. I am also grateful to Dr. Darrell Robinette for his constant support. He has used his extensive experience to keenly guide the research and offer timely suggestions. Both, Dr. Mahdi Shahbakhti and Dr. Darrell Robinette, have outlined the scope of this research and have provided resources towards its completion. Their willingness to teach and help with this research has helped in my understanding of the subject.

The research would not have been possible without the support of the team at Ford Motor Company. Special thanks to Dr. Maruthi Ravichandran and Dr. Jeff Doering. Their keen reviews on each step of the process and industry insight have proved to shape this research and make it meaningful. I would like to extend appreciation towards Dr. Mo Rastgaar for agreeing to be a part of the oral examination committee and for your valued comments on this research.

I would like to thank my colleague Prithvi Reddy for co-developing the driveline model and providing report reviews and feedback at every stage. A very special thanks to Reem Merchant for her constant support and encouragement to start the work, persevere with it, and for providing assistance at all times. Also, I would like to thank my friends at Michigan Technological University – Alston Dsouza, Shruti Amre, Siddhant Dange, Siddhant Ruia, Kaushal Darokar for providing constant support and assistance in various aspects of my school life. I would like to wish them best for their future endeavors.

I am very grateful to my family – Mr. Anis Lakhani (father), Late Mrs. Gulshan Lakhani (mother), Mr. Ashif Lakhani (brother), Mrs. Heena Lakhani (sister-in-law), Ms. Aahana Lakhani (niece) – for their unconditional love, blessings and unceasing encouragement, especially throughout the span of the project.

Nomenclature

α_{tb}	Half of the backlash gap between the input and output of transmission [^{o}]
α_{fdb}	Half of the backlash gap between the input and output of final drive [^{o}]
α_{tc}	Half of the backlash gap between the input and output of clutch[^{o}]
θ_c	Clutch output torsional angle [^{o}]
θ_m	Angular displacement of the crankshaft [^{o}]
θ_{ds}	Angular displacement of the drive shaft [^{o}]
θ_{fd}	Angular displacement of the final drive [^{o}]
θ_{fdb}	Final drive backlash position angle [^{o}]
θ_{ps}	Angular displacement of the propeller shaft [^{o}]
θ_r	Gradient of the road [^{o}]
θ_t	Output shaft angular displacement [^{o}]
θ_{tb}	Transmission backlash position angle [^{o}]
θ_w	Angular displacement of the wheel [^{o}]
δ_m	Time delay in torque generation due to combustion [sec]
ρ_w	Density of air [kg/m^3]
τ_m	Time lag due to air breathing dynamics [sec]
A_f	Frontal area of the vehicle [m^2]
a	Rolling resistance coefficient [-]
C_D	Coefficient of drag [-]
b	Rolling resistance coefficient [-]
c_c	Clutch torsional damping [Nm/(rad/s)]
c_{ds}	Drive shaft torsional damping [Nm/(rad/s)]
c_{fd}	Final drive torsional damping [Nm/(rad/s)]
c_{ps}	Propeller shaft input shaft torsional damping [Nm/(rad/s)]
c_{ts}	Transmission shaft input shaft torsional damping [Nm/(rad/s)]
c_w	Wheel torsional damping [Nm/(rad/s)]
F_{DR}	Drag resistance force [N]
F_{GR}	Gradient resistance force [N]
F_{load}	Load force [N]
F_{IR}	Inertial resistance force [N]
F_{RR}	Rolling resistance force [N]
g	Acceleration due to gravity [m/s^2]
i_{fd}	Final drive gear ratio [-]

J_{ds}	Drive shaft rotational inertia [$kg.m^2$]
J_{fd}	Final drive rotational inertia [$kg.m^2$]
J_m	Engine equivalent rotational inertia [$kg.m^2$]
J_{ps}	Propeller shaft rotational inertia [$kg.m^2$]
J_t	Turbine rotational inertia [$kg.m^2$]
J_w	Wheel rotational inertia [$kg.m^2$]
k_c	Clutch torsional stiffness [Nm/rad]
k_{ds}	Drive-shaft torsional stiffness [Nm/rad]
k_{ps}	Propeller shaft input shaft torsional stiffness [Nm/rad]
k_{ts}	Transmission shaft input shaft torsional stiffness [Nm/rad]
k_w	Wheel torsional stiffness [Nm/rad]
K	Capacity factor [$(rad/s)/(Nm)^{0.5}$]
m_w	Mass of the vehicle [kg]
N_m	Engine output speed [rad/s]
N_t	Turbine hub speed [rad/s]
r_{eff}	Effective radius of the tire [m]
SR	Speed ratio [-]
t	Time [sec]
T_m	Torque delivered by the engine [Nm]
T_{req}	Requested torque from the engine [Nm]
T_{fd}	Torque delivered by the final drive [Nm]
T_{load}	Load torque [Nm]
T_t	Torque delivered by the turbine [Nm]
T_{tc}	Torque output after the torque converter clutch dynamics [Nm]
T_{ps}	Torque output at the propeller shaft output [Nm]
T_{ts}	Torque output at the transmission shaft output [Nm]
T_w	Torque output at the wheel [Nm]
TR	Torque ratio [-]
V	Vehicle velocity [m/s]
V_w	Wind velocity [m/s]

Abstract

In modern-day automotive industry, automotive manufacturers pay keen attention to driver's safety and comfort by ensuring good vehicle drivability, feel of acceleration, limiting jerk and noise. The vehicle driveline plays a critical role to meet these criteria. By using high-fidelity simulation tool such as AMESim®, it is now possible to accurately model the vehicle driveline to be tested for different scenarios. With Simulink®, one can develop an efficient torque-based control system to limit the driveline oscillations and the generated noise. So, a joint simulation is used which provides a platform to evaluate the estimators and control system while considering the fast dynamics of the non-linear system.

This report presents the detailed driveline model developed to evaluate the important parameters which affect the driveline of a pickup truck. The model is developed considering the non-linear dynamics of the driveline, torque converter clutch dynamics and the non-linearities in the propeller shafts and the drive-shafts. It is then evaluated at different input conditions for two major test scenarios – tip-in and tip-out. Both scenarios show that the model displays the transmission and final drive backlash dynamics as anticipated in practical scenarios. The wheel speed shown by the results of the model proves that stiffness and damping coefficient of the tires play an important role in predicting the physical behavior of the vehicle. In addition, for the case of a tip-in from negative to positive torque, the effect of flexibilities of the driveshafts is shown as significant by this model. The oscillations caused due to these flexibilities are within 7 – 8 Hz range for evaluation at fifth gear. This frequency of oscillations found in this model is comparable to the results found in the literature.

In future, experimental validation of the current full-order model would provide better understanding of the assumptions considered while developing it. A reduced order model can be derived from the current model which can be further used to develop the estimators and controllers for active reduction of the driveline oscillations. Also, the overall effect of engine mounting system, comprehensive tire model and suspension dynamics on driveline oscillation can be studied.

1 Introduction

Vehicle drivability, dynamic performance (acceleration feel, jerk) and driver's comfort are important metrics to automotive manufacturers along with achieving stringent emission standards and better fuel economy. The driveline (interchangeably used as the drivetrain), being one of the fundamental components of a vehicle's propulsion system, plays an important role to meet these goals. It transfers the torque from the torque generators (engine or electric motor) to the wheels. The main parts of the driveline consist of clutch (torque convertor), transmission, propeller shaft, final drive, drive-shafts and wheels. The schematic of a rear-wheel-drive vehicle is shown in Figure 1.1 where the elements in blue and black are the parts of the driveline. All these parts have different velocities and torques relative to each other. The reason for these different velocities is due to the elastic nature of the parts, which can cause mechanical resonances in the components affecting the functionality and drivability of the vehicle. Drivability refers to the driver's perception of vehicle response to a specific input. It also includes factors such as vehicle handling, vehicle's responsiveness and driver's comfort which are of utmost importance. Hence, it is important to handle such resonances to improve its drivability as well as to reduce mechanical stress and noise.

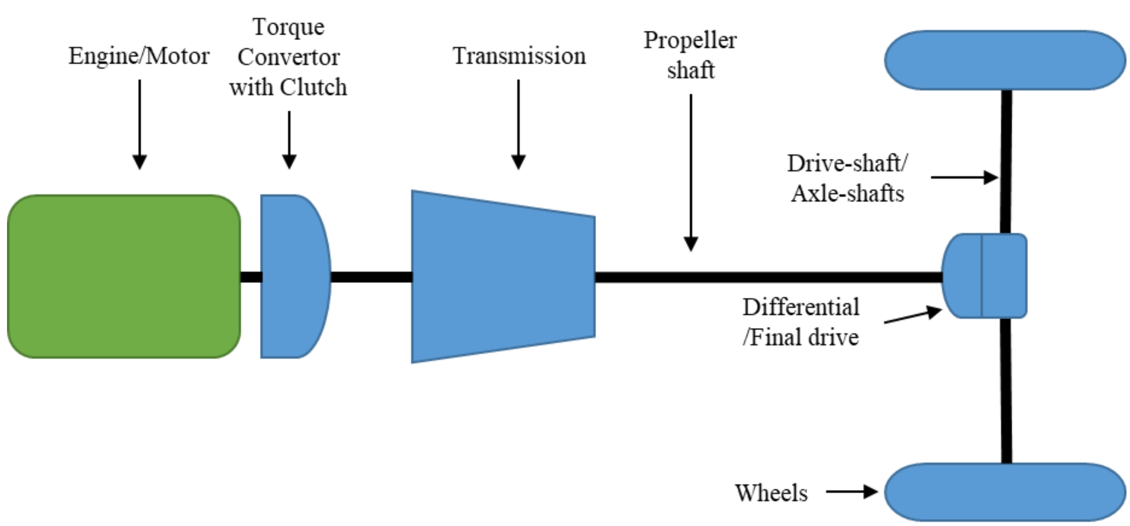


Figure 1.1. Schematic of the driveline of a rear-wheel-drive vehicle

Torque transmitted through these components can be considerable when evaluated in terms of the torsional forces and torsional energies. Especially with high-powered heavy-duty engines, the amount of torque transmitted in the first gear could cause failure of the drive-shafts. If not controlled correctly, these substantially high torsional energies can cause multitude of problems. The three main problems experienced in every vehicle are vehicle shuffle or vehicle surge, clunk-and-shuffle during tip-in conditions, and, oscillations after tip-out conditions (i.e. equivalent to engagement of neutral gear) [1].

1.1 Clunk-and-Shuffle

While driving, it is common for a driver to provide a rapid input to the accelerator pedal in order to obtain either a quick acceleration or to provide more torque to the wheels. These are known as "tip-in" scenarios. The terms "vehicle shuffle" or "vehicle surge" refers to the longitudinal rocking of the vehicle or truck cabin due to the induced driveline

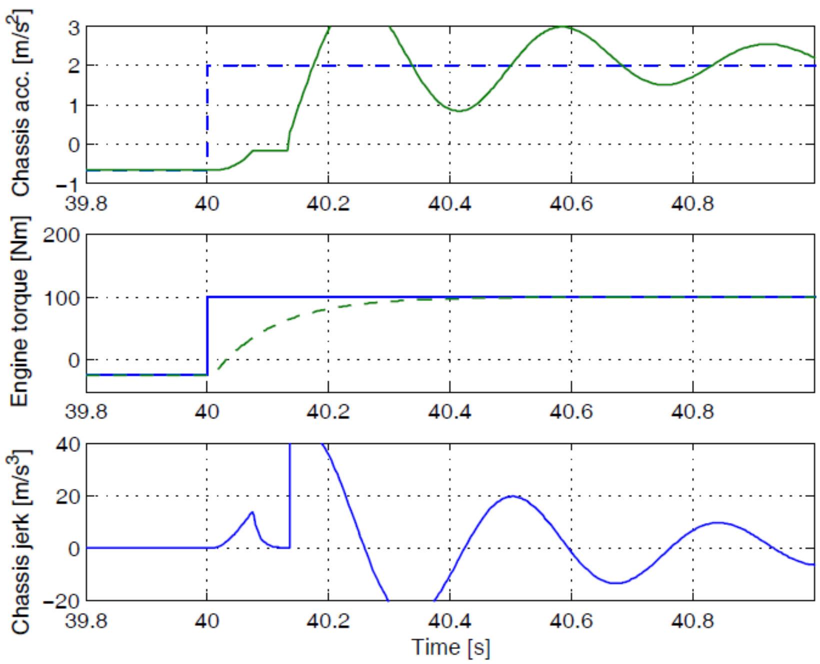


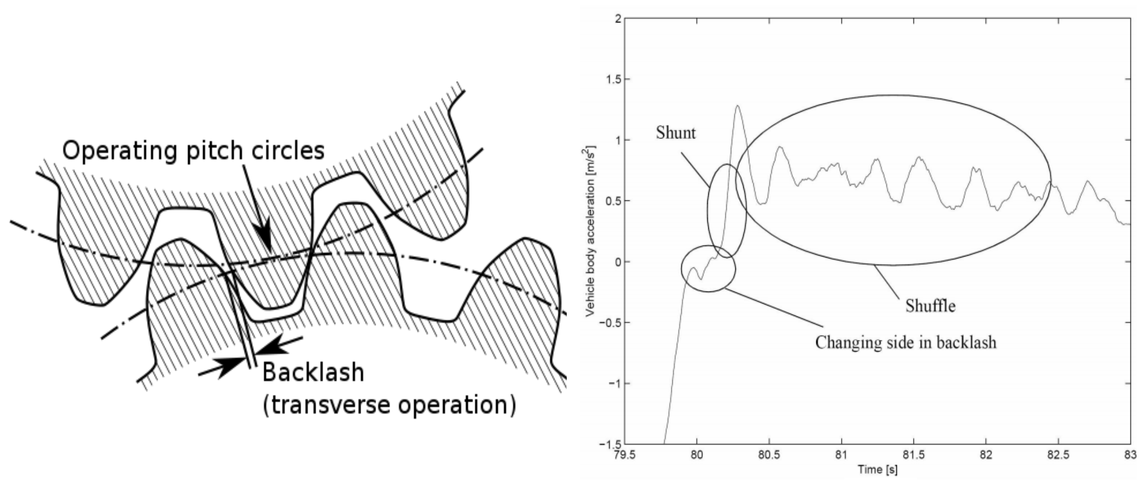
Figure 1.2. A simulated tip-in response of a vehicle is shown. The upper plot shows the driver's desired acceleration (dashed) and the vehicle acceleration (solid). Engine requested torque (solid) and delivered (dashed) torque, is shown in the center plot, while the lower plot shows the vehicle jerk. Adopted from [14]

oscillations when a tip-in occurs from a certain positive torque provided by the engine, shown in Figure 1.2 and Figure 1.4. For heavy-duty trucks or for trucks with towed trailers, this low-frequency (about 5 - 10 Hz) induced driveline oscillations can cause variations in the pitch of the vehicle, affecting the driver's comfort. The "clunk-and-shuffle" or "shunt-and-shuffle" phenomenon can occur when a tip-in occurs from a negative torque state. This triggers backlash traversal across various geared parts of the driveline (e.g. transmission, final drive, transfer case) leading further to provide positive torque to the wheels. The backlash traversal leads to clunk which can be observed when there is a sudden impact between the teeth of the gears in contact. This clunk is the source of loud noise. It also causes the vehicle to jerk (quantified mathematically as time derivative of vehicle acceleration).

The third problem occurs in case of a tip-out scenario wherein the driver releases the accelerator pedal completely and/or the neutral gear is engaged in case of a manual transmission. The torsional energy available in the drive-shaft when driving at a constant speed is liberated as soon as the accelerator pedal is released. The oscillations caused due to the release has negligible effect on longitudinal vehicle motion, but it is quite observable on the transmission shafts. This can cause undesirable noise from the transmission as well as the gear engagement needs to be delayed until the oscillations die out sufficiently. The amplitude of the induced oscillations is proportional to the initial steady state angular speed of the drive-shafts [1]. In an ideal case, if the driveline was completely stiff and the wheel speed be well damped, the driveline oscillations would not exist. But in reality, the torque fluctuations can be felt with an elastic driveline in the vehicle. Thus, it is desired to control the oscillations. The amplitude of these oscillations or the clunk-and-shuffle behavior is used as one of the measures for driver's comfort.

1.2 Effect of Backlash

Backlash can be defined as a play between adjacent moving parts. This mechanical form of a deadband can be better visualized as the gap between the mating teeth of the gears (Figure 1.3). It is a very common problem in the automotive drivelines. This gap is desirable up to a certain extent to prevent jamming between the gears. But an increased amount of the gap (due to extended usage of the component causing erosion) will lead to undesirable noise and impact to the mating surfaces during torque transmission, causing



(a) **(b)**

Figure 1.3. (a) Backlash, in the context of gears is shown (b) Vehicle body acceleration during a tip-in event on a vehicle, reported in [10]

increased wear of the components and sudden acceleration of the vehicle. Backlash also introduces a non-linearity in the control system loop. In [13], the authors have observed backlash to degrade the driveability of the vehicle and affect the performance of the control system. Hence, it is necessary to control or at least compensate for the ill-effects of the backlash with an objective of traversing backlash as soon as possible to retain the fast response of the vehicle.

1.3 Driveline Modeling for Clunk-and-Shuffle – Analysis and Control

Out of various strategies prevalent in the industry, the torque-based control is widely used. The driveline oscillations that are introduced can be reduced by controlling the engine torque request such that the engine inertia works against them. This form of active damping is quite effective for controlling the vehicle shuffle [2]. Many of the available torque-based controllers simply smoothen any sharp changes in the engine torque request to prevent the clunk-and-shuffle in the drivetrain. Although, this method reduces the oscillations to acceptable limits, it results adversely in slower response of the vehicle. To address this problem, an improved torque-based control system is needed which can address the clunk-and-shuffle phenomenon and improve the drivability of the vehicle.

Existing control strategies are difficult and time-consuming to calibrate due to the vast number of parameters involved. Thus, there is a need for an advanced control system to reduce the calibration effort and improve its dynamic performance under several disturbances and environmental variations. The main objective is to characterize the clunk-and-shuffle phenomenon and solve the problems associated with the noise, vibration and harshness (NVH) and the drivability of the vehicles by developing an advanced control strategy. The work for this Master's degree report serves as the first step towards the development and implementation of the control strategy, wherein detailed models of the driveline are developed and evaluated. The aim of this driveline modeling is to find the source of the oscillations and understand the physical parameters affecting them.

1.3.1 Literature Review

A thorough scrutiny of published literature related to automotive driveline development was conducted. The literature search in this report includes sources from journal articles, conference publications, thesis and dissertations. Relevant work done previously for modeling and torque-based control system development for driveline was studied. The main objective of this literature search involves detailed drivetrain modeling, control-oriented model order reduction and development of the control system. This report will address only a part of that literature review related to detailed driveline modeling.

1.3.1.1 Driveline Modeling

In published literature, there are detailed studies related to driveline modeling. Relevant work includes Hayat et al. [8] (Figure 1.4). In their study, they proposed an alternative modeling approach for better assessment of the driveability requirements of the customer during the early phase of the vehicle design cycle. Three different models for capturing the powertrain and driveline dynamics were proposed. The applicability of each of these models would depend on which stage the vehicle design cycle project would be in. During the design and development phase, a detailed linear model, but not of a reduced order, is required for the design of the sub-systems. To formulate the control strategy, they suggest a second model of a reduced order which should be capable to capture the approximate vehicle behavior but with lower degree of freedom in order to reduce the computation time. For the validation phase of the project, according to Hayat et al. [8], a third, more detailed, non-linear model is required to accurately assess the physical behavior of the vehicle. It will also help refine the design of the sub-systems as well as the control strategies.

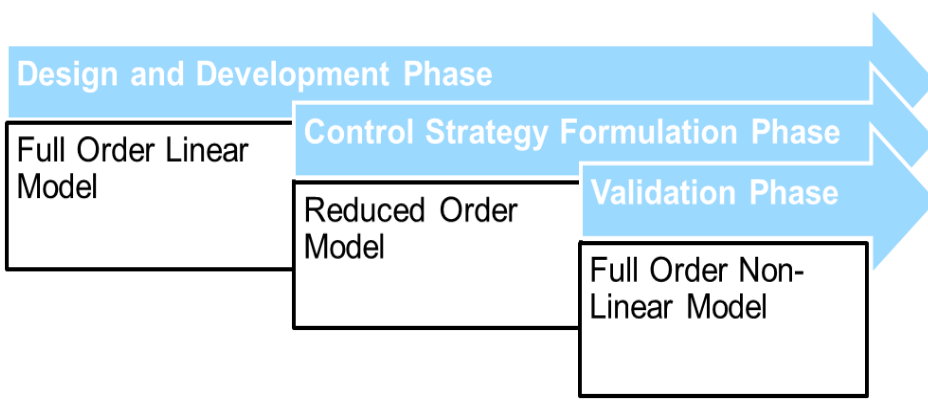


Figure 1.4. Structure of Hayat et al. [8] modeling approach

A thesis study conducted by Pettersson [2] serves as a good resource for understanding the principles required for powertrain and driveline modeling with regards to its control applications. Many other following works including Eriksson and Farahani [3], Tunhag [4], Nezhadali [5] and Myklebust and Eriksson [6] were based on the research conducted by Pettersson. As his models were based on a rear-wheel-drive vehicle with a manual transmission, the components that were modeled included engine, clutch, manual gearbox, propeller shafts, drive-shafts and wheels. These higher degree of freedom models were made by considering these as an arrangement of multiple rotating inertias connected to the flexible shafts with inherent damping associated to them. Pettersson developed three models with a new complexity added to each new model over the previous one asshown in Figure 1.5. First model created was a linear model with viscous friction added only to the transmission and the final drive while the other components such as clutch, propeller shaft and drive-shafts were modeled stiff. The clutch flexibility was added to the first linear model to develop the second one. The third model was built in with a static non-linearity in the clutch model. He also conducted the experiments to verify the model on lower gears as it provides high torque transfer condition leading to high amplitude of driveline oscillations. Based on these experiments, he concluded that the key source of flexibility in the driveline is associated to the components in between the transmission output shaft and the wheels. Thus, he considered a stiff driveline up to the transmission output shaft and incorporated flexibility only in the propeller shaft and the drive-shafts. Non-linearities such as backlash were not considered in his research.

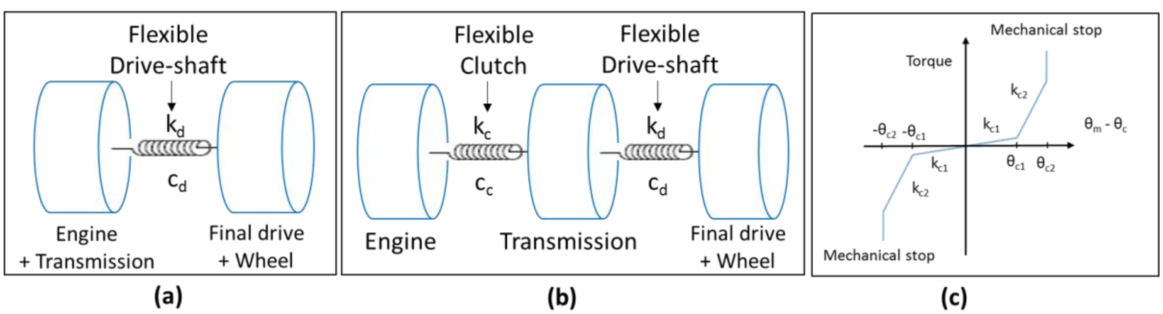


Figure 1.5. Schematic of the three models developed by Pettersson [2] where
(a) Model 1: Only drive-shaft has damped torsional flexibility (stiffness, k_d and damping coefficient, c_d), (b) Model 2: Clutch modeled as linear flexibility (stiffness, k_c and damping coefficient, c_c) along with damped torsional flexibility for the drive-shaft (c) Model 3: Non-linear characteristics added to the clutch with and flexible drive-shafts.

Karlsson [10] studied a Volvo passenger car and suggested two models. The first model was a linear third order model that omitted the clutch dynamics and backlash. The second model was a more complex non-linear model in which the clutch dynamics and backlash were included. He assessed the performance of these models by applying different types of torque inputs.

Bellomo, et al. [7] aimed at studying vehicle NVH by formulating a detailed object-oriented model of the powertrain and the driveline. The authors identified two major types of noises from the vehicle, one is the noise made by the freely rotating components in idle, in-gear drive and in coast mode. Another source is the impact noise from the gears due to backlash traversal. The model developed incorporates non-linearities in the system such as clutch hysteresis as well as backlash in the gears, providing insight into the modeling with a perspective of reduction in driveline oscillations. The authors also investigated the effect of different model parameter such as stiffness, damping and inertia of clutch, wheels and drive-shafts as well as clutch hysteresis and vehicle mass.

Evaluation of the influence of the drivetrain elements on longitudinal dynamics of the vehicle using mathematical models was carried out by Sorniotti [9]. This approach has been taken into consideration for the current work. In this research, five different non-linear models of the driveline were developed which studied the effect of each of the components viz., dynamics of the clutch torsional damper (with non-linear stiffness and hysteresis of varying amplitude), tire flexibility, driveshaft stiffness and damping, details of which are

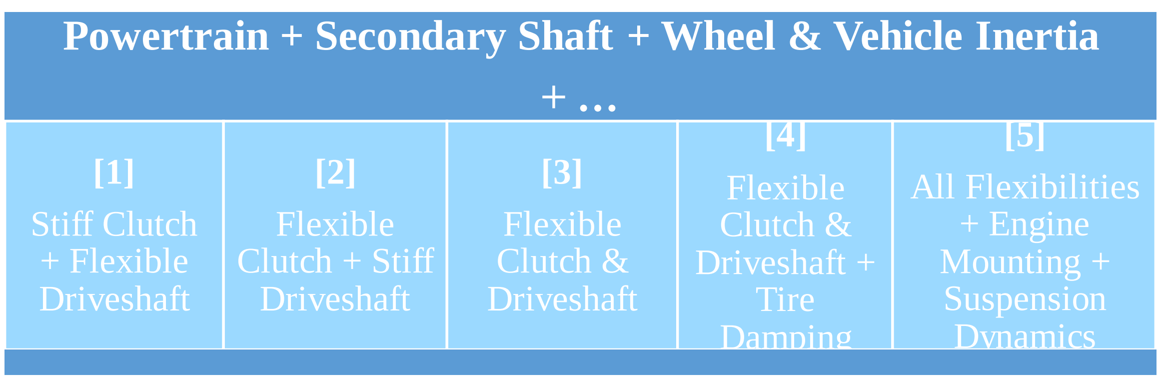


Figure 1.6. Summary of Sorniotti's [9] five driveline models

shown in Figure 1.6. Also, the suspension dynamics along with powertrain mounting system has been modeled to capture the longitudinal and vertical dynamics of the sprung and unsprung mass. Furthermore, Sorniotti [9] also discusses reduced-order linearized models and its analysis in the frequency domain.

1.4 Objectives and Organization of this Technical Report

This report aims to develop a driveline model for Ford F150 pickup truck. This model is used as an analysis tool to understand clunk-and-shuffle, and effect of backlash status and position on performance of F150 driveline. The report is organized as follows:

Chapter 2 covers the platform used for the physical and mathematical modeling of the vehicular driveline. The baseline vehicle used for modeling and the analytical equations behind the physical models of the components are also described in Chapter 2.

Chapter 3 discusses and analyzes the simulation results for the effect of each driveline components and observe the full system behavior in terms of driveline oscillations, vehicle acceleration, etc.

Chapter 4 provides a summary of this work and lists the conclusions along with future work.

2 Modeling

Driveline modeling is needed to evaluate and capture the situations where the velocities and torques along the vehicle components are different at the inputs than the outputs. The type of the models (linear or non-linear) and the level of details they contain depend on the purpose and the applications in which it will be used. As the objective of this study is to simulate the driveline dynamics, the physical model made using the AMESim® software is explained in the subsequent sections. As the fundamentals of the component modeling remains the same, the underlying equations used in the modeling of each of the components will also be discussed in the subsequent sections.

2.1 Software Platform

This section briefly explains the software products used for the work done in this report.

2.1.1 LMS AMESim®

LMS Imagine.Lab Advanced Modeling Environment for performing Simulations, known as AMESim® (Siemens Product Lifecycle Management Software Inc., Version 15.2, 2017) is a multi-domain one-dimensional system simulation software used for modeling, analysis and performance prediction of engineering systems. Based on the latest version v15.2, AMESim® offers up to 40 libraries to model the non-linear time-dependent analytical equations representing the appropriate system behavior. As most of the physical domains are modeled using the libraries containing pre-defined set of blocks with input and output ports, the causality is imposed when the outputs of one component block is connected to the inputs of another component block.

The modeling and simulation of the system in AMESim® is done in four steps:

- Sketch mode: The component blocks with built-in analytical equations are pulled from the library and are joined together to form a system.
- Submodel mode: Physical submodels for each component block is chosen.

- Parameter mode: The values of the parameters for each submodel of the system are set and the model compilation takes place.
- Simulation mode: The simulation is performed, and the results can be visualized.

2.1.2 Simulink®

Simulink® (The MathWorks, Inc., Version 8.9, Release2017a, 2017) is similar to AMESim® software as it is also a graphical programming software used for modeling, simulation and analysis of dynamical multi-domain systems. But Simulink® is a highly suitable and widely used for model-based design and control systems development applications. Its capability for generating S-functions (system functions) for real-time implementation of complex systems had made it easier to use as a tool for embedded system work. It is also easier to use Simulink® software for data-processing.

2.1.3 Combined simulation

AMESim® and Simulink® can be used together to perform the simulations to maximize the potential advantage offered by both platforms. There are two ways to achieve this. First is importing the AMESim® model as S-functions into Simulink® (known as the AMESim®-Simulink® interface). Second is by importing the Simulink® model into AMESim® (known as the Simulink®-AMESim® interface). These two methods, where the models are

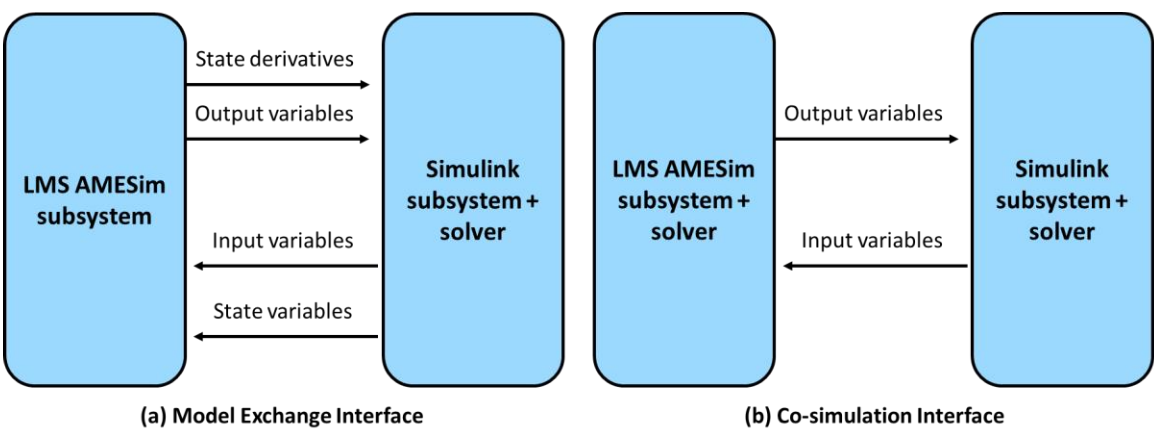


Figure 2.1.Details of the information exchange between the two types of the interfaces of AMESim® and Simulink®. Adopted from [11].

uploaded without their respective solvers is termed as the Model Exchange interface, while the method of combining the two software with solvers is called as the Co-simulation interface [11] [12]. Co-simulation basically means that the solvers from two different software packages are imported together with the corresponding software's model to another software. The information exchange that occurs between the two software for the two types of interface is shown in Figure 2.1.

Figure 2.1 shows the schematic of the interfaces. The model exchange interface shown Simulink® provides the inputs, equations and state variables to AMESim®, and then AMESim® calculates the state derivatives and the output variables. The process of this transfer of information from Simulink® to AMESim® is controlled by the Simulink® solver alone.

As the ultimate objective of the project is to develop a model-based control strategy to control the vehicular jerk as well the noise and vibrations from the automotive driveline, a Model exchange interface of AMESim® and Simulink® was used to model the vehicle and its driveline as shown in Figure 2.1(a). The communication structure of the model exchange interface is shown in Figure 2.2. For the work done in this report, the physical modeling of all the driveline components was done using AMESim®, while Simulink® was used to

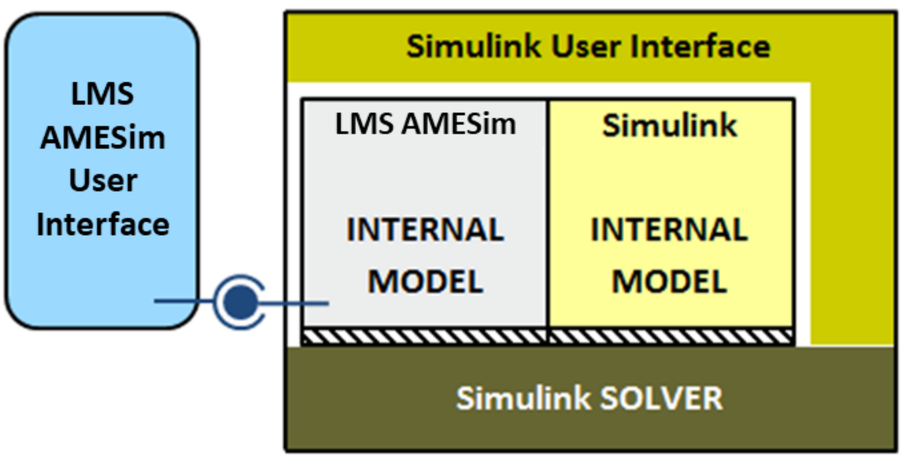


Figure 2.2. Communication structure for the combined AMESim®-Simulink® platform using the model exchange interface. Adopted from [11], [12].

define the driver torque requests along with simplified first-order engine dynamics. To convert the AMESim® model to an S-function post modeling, Visual C++ (Version 2013) is used since it is the only compiler compatible to create S-function of the physical model with AMESim® ver15. Simulink® solver is used to solve the equations for both AMESim® and Simulink® models. Hence, to capture the fast-dynamics of the AMESim® model (stiff system), an ode15s (variable step) solver was used. Simulink® will also be used to develop the control system in future. After constructing the model in AMESim®, the values of the model parameters are modified if needed and then S-function is generated. The new S-function is provided to the Simulink® model. Using Simulink's solver, the simulation is run, and the results obtained are analyzed using the respective software. The procedure to simulate the model developed for the present work using the Model exchange interface is shown in Figure 2.3.

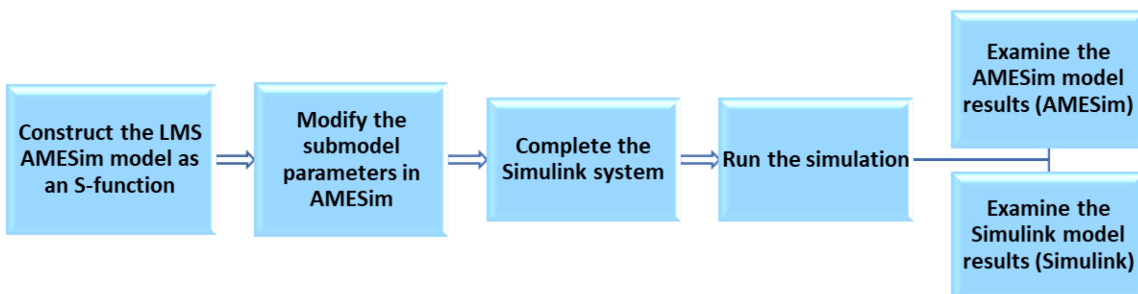


Figure 2.3. Procedure to prepare and execute the simulation on the combined AMESim®-Simulink® platform. Adapted from [11], [12].

2.2 Assumptions and Limitations in the model

The domain of driveline dynamics is widespread. This report covers a small portion of it. Assumption made for the purpose of this study is that the vehicle moves in the longitudinal direction only. Hence, only the longitudinal dynamics of the vehicle is considered for this study. Also, the vehicle is subjected to tip-in and tip-out maneuvers only. The model is made for Ford F150 which is a rear-wheel-drive, engine-driven vehicle with a 10-speed automatic transmission.

In a real world, the powertrain mounts are flexible, allowing it to rotate a couple of degrees with variations in the torque. However, for this study, the powertrain mounts are assumed to be stiff. Also, the transmission is modeled as rigid and friction in the kinematic pairs (excluding the clutch) is neglected in the model. The gear-shifting strategy is not incorporated in to the model for simplification and all the simulations are done at a fixed gear. Differential is locked all the time. Wheel slip in not considered in the model since it dampens the jerk level felt by the driver and hence the model predictions will not be accurate.

This report does not deal with suspension modeling or the effects of the suspension dynamics on driveline oscillations. Parameter identification and experimental validation of the model is proposed as future work for this study.

2.3 Modeling Equations

The Newton's second law equation is the underlying principle used to model the components. The model has diverse physical effects from combinations of rotating inertias with damped shaft flexibilities. These combinations can create sub-models that are interfaces between torques and rotational velocities. The torque-based control structure model is well-suited for this. The modelling objective is to filter the main causes of the oscillations in the engine, transmission and wheel speeds from these diverse physical effects.

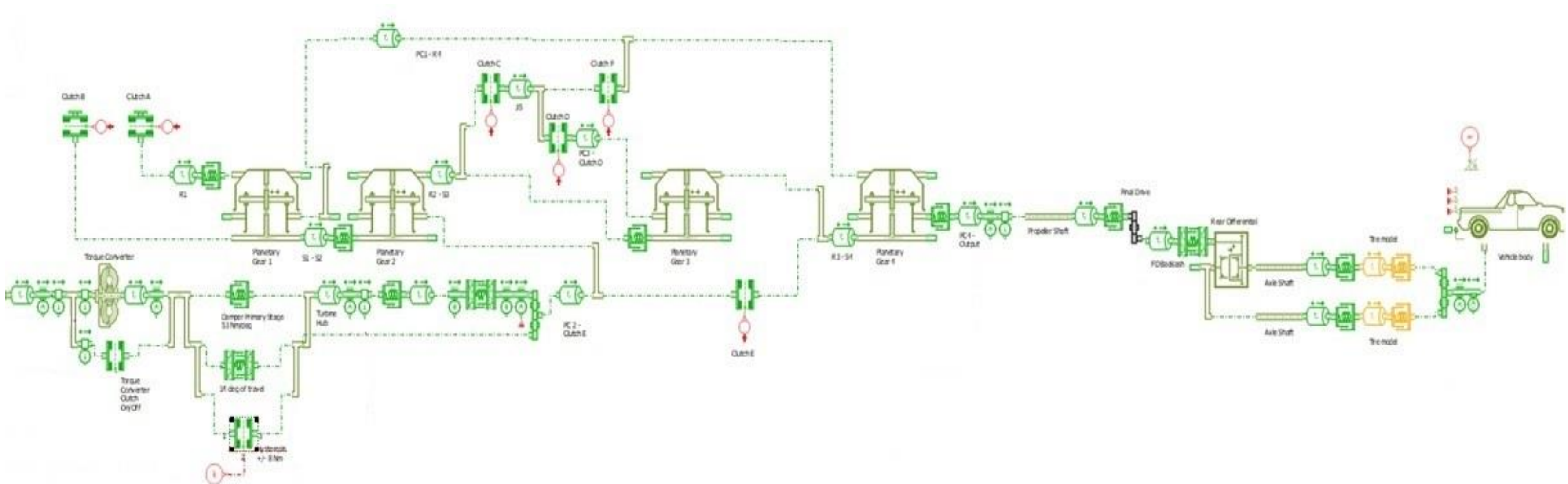


Figure 2.4. Schematic model of Ford F150 truck made using AMESim® software

The inputs to the model include the driver requested torque and other loads, such as the driving resistance acting on the wheels.

The non-linear driveline model of the vehicle can be simplified as a spring-mass system (Figure 2.4) when the vehicle is run at a fixed gear with clutch engaged. A detailed discussion of model features and their fundamental equations will be explained in the following sub-sections.

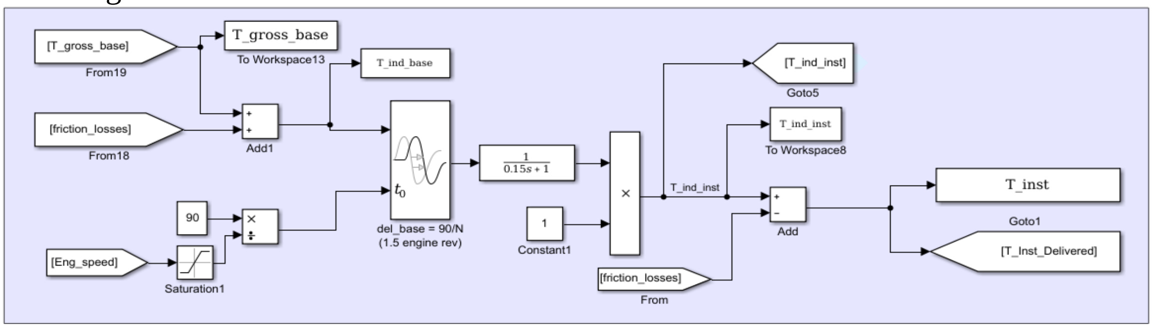


Figure 2.5. Schematic model of the first-order engine dynamics in Simulink®

The requested torque is provided using the signal builders in the Simulink to a simplified first-order engine dynamics block. The modified torque output from the Simulink® model, termed as ‘Torque delivered’ throughout this report, is provided to the LMS AMESim® model. This torque goes through the following components modeled in AMESim® - engine modeled as an inertia, torque converter (with a clutch), 10-speed transmission, propeller shaft, final drive, two drive-shafts, two tires and a longitudinal vehicle dynamics block.

LMS AMESim® captures the physical behavior of the components in the model using Newton’s second law of motion along with some underlying assumptions. These assumptions help us find out what physical effects of the components affect the driveline

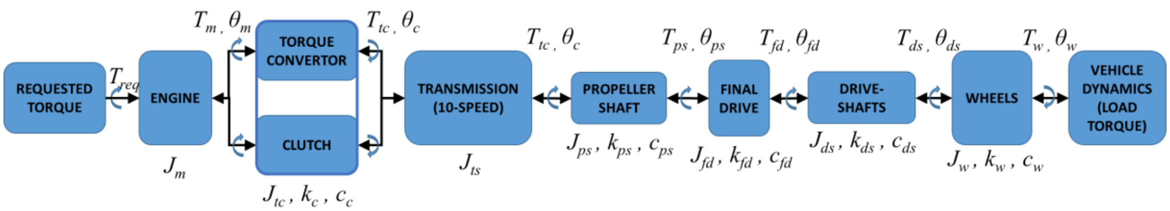


Figure 2.6. The block-level diagram of the complete AMESim® - Simulink® model

oscillations. Basic force/torque-based equations are used to model each component. Then each component is linked to each other using the relations between the inputs and outputs of the equations to get a complete driveline model.

2.3.1 Engine

The torque generated by the engine is characterized as the reduction of the indicated torque by the amount of torque lost in pumping, due to friction, and in accessory drive loads. The detailed model of the engine and its corresponding losses are neglected in this evaluation as its effect on the driveline model is small or slow. So, it was simplified to a first-order engine-dynamics model which takes into consideration only the delay in torque generation caused due to combustion and lag due to air manifold filling dynamics. The delay in torque generation due to combustion (δ_m) is engine speed dependent while the lag caused due to the manifold filling dynamics can be considered as a first-order lag with a time constant (τ_m) [15]. In Simulink®, the driver requested torque is provided to the engine dynamics model using the signal builders. The first order engine dynamics block is shown in Figure 2.5. The engine dynamics can be represented as follows:

$$\dot{T}_m(t) = \frac{1}{\tau_m}(-T_m(t) + T_{req}(t - \delta_m(t))) \tag{1}$$

where, T_m is the torque delivered by the engine, T_{req} is the requested torque from the engine.

The torque delivered from the engine block goes to the AMESim® model in which the first block consists of a lumped inertia (J_m) for reciprocating and rotating engine components. The inertia block in AMESim® is a simple rotary load with external torque applied at the input and output ports. The block internally calculates the rotary velocity. Its corresponding rotary acceleration is computed using the formula:

$$\ddot{\theta}_m = \frac{T_m - T_{req}}{J_m} \tag{2}$$

where, θ_m is the angular displacement of the crankshaft.

2.3.2 Torque Converter

Vehicles with automatic transmission have torque converter as the connecting link between the impeller is rigidly coupled to the crankshaft of the engine. Hence, it rotates at the same speed as that of the engine. The turbine is another part of the torque converter. It is locked with the gearbox through the transmission input shaft. Turbine and impeller are immersed in oil. This oil acts as a hydraulic coupling between the two components. The torque transfer occurs as the oil is spun by the impeller, which eventually rotates the turbine, and vice versa. This hydraulic coupling allows the torque to be transferred between the engine and the transmission mechanism.

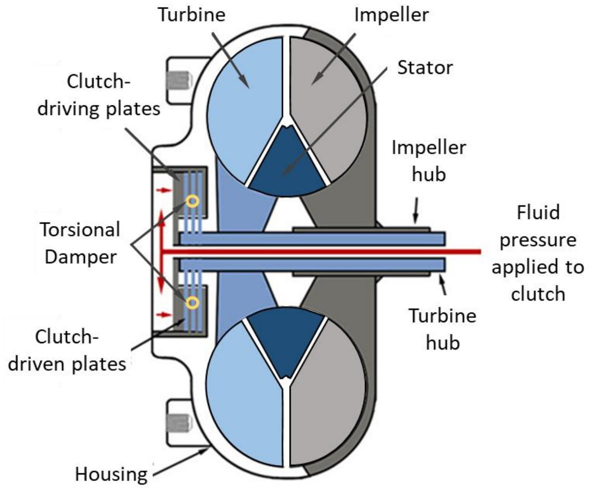


Figure 2.7. Schematic of a lock-up torque converter.

The torque converter model for our application considers the inertias of the impeller (pump), turbine, stator and fluid lumped together as one inertia. Based on the need of the application, a simplified model can be made for the torque converter which has its static representation in the form of a plot of torque ratio and input capacity factor, both versus speed ratio. The underlying relations for the torque converter can be written as [18]

$$SR = \frac{N_t}{N_m}, \quad TR(SR) = \frac{T_t}{T_m}, \quad K(SR) = \frac{N_m}{\sqrt{T_m}} \tag{3}$$

where, N_m = impeller hub speed (engine output speed), N_t = turbine hub speed, T_m = impeller (input) torque (engine output torque), T_t = turbine (output) torque, SR = speed ratio, TR = torque ratio, as a function of the speed ratio, K = capacity factor, as a function of the speed ratio.

The torque converter clutch is modeled in a parallel connection to the torque converter to capture its appropriate physical behavior (Figure 2.8). It consists of the pressure plates, friction plates, and clutch disc with torsional damper and a release mechanism. Next block in line is the turbine inertia (J_t). Moreover, to capture different modes of clutch operation, the capacity factor for the clutch (K_c) is also taken into consideration while modeling. The three modes of clutch operation are open, slipping and closed or locked condition. Following fundamental equations will describe its behavior in each mode –

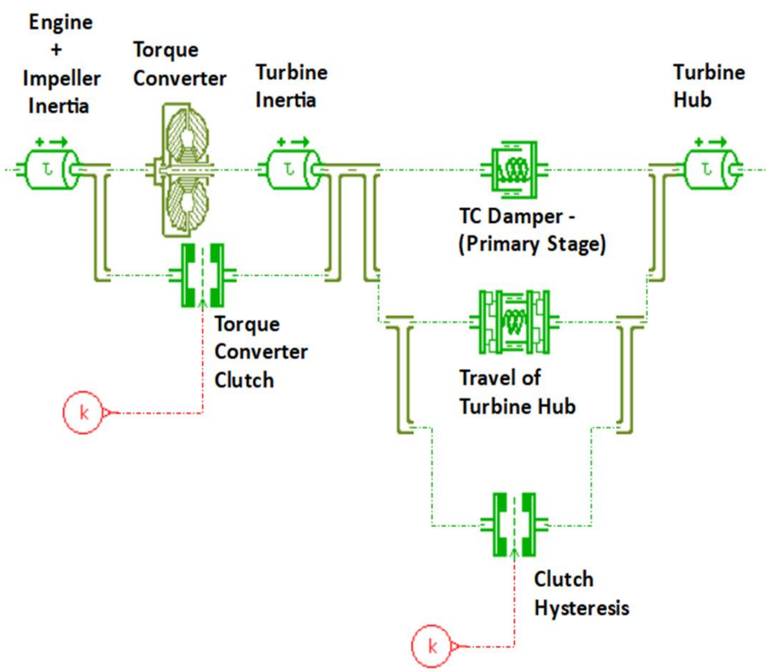


Figure 2.8. The AMESim® model for engine inertia, torque-converter, clutch and corresponding clutch-dynamics

Open condition –

$$T_t = T_m\left(TR\left(SR\right)\right) \tag{4}$$

$$T_t = \left(\frac{N_m}{K\left(SR\right)}\right)^2 \left(TR\left(SR\right)\right) \tag{5}$$

Slipping condition –

$$T_t = T_m + \left(TR\left(SR\right) - 1\right)\left(\frac{N_m}{K\left(SR\right)}\right)^2 \tag{6}$$

Locked condition –

$$T_t = T_m \tag{7}$$

2.3.3 Torque Converter Clutch Dynamics

The dynamics of the torque converter clutch is characterized using three components in parallel as given below:

1. The flexibility of the clutch can be modeled using the spring-damper block as:

$$T_{tc} = k_c(\theta_m - \theta_c) + c_c(\dot{\theta}_m - \dot{\theta}_c) \tag{8}$$

   where, T_{tc} = torque transferred by torque converter clutch, k_c = clutch torsional stiffness, c_c = clutch torsional internal damping, θ_c = the clutch output torsional angle.
2. The rotary clearance block (backlash element), connected in parallel to the spring-damper block, captures the torsional compression effects of the torque converter clutch springs by providing a gap or clearance corresponding to a relative angular displacement between the impeller shaft (input) and the turbine hub (output). It can be represented using the dead-zone backlash model using the equation given by Karlsson [10]:

$$T_{tc} = \begin{cases} k_c(\theta_c - \alpha_{tc}), & \theta_c > \alpha_{tc} \\ 0, & |\theta_c| < \alpha_{tc} \\ k_c(\theta_c + \alpha_{tc}), & \theta_c < -\alpha_{tc} \end{cases} \quad (9)$$

where, $2\alpha_{tc}$ = total clutch backlash.

3. To accurately characterize the friction damping of the system, the clutch hysteresis block is added as the third component. Hysteresis in clutch is attributed to both internal and external friction damping phenomenon. This hysteresis arises due to, not only the elastic forces within the mechanical parts, but also the non-elastic effects resisting generation and propagation of deformations and movements. It does not allow the parts to come back to their initial states. It can be perceived as an energy dissipation which occurs when the cyclic loading forces are applied to the clutch. The torque output of all three components are added to obtain the torque delivered to the transmission.

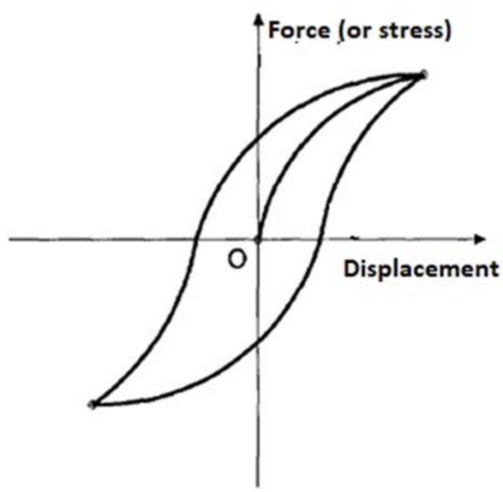


Figure 2.9. Hysteresis phenomenon when the clutch is subjected dynamic loading and unloading forces. Adopted from [16]

2.3.4 Automatic Transmission

The Ford 10R80 automatic transmission has four interconnected planetary gears with six clutches to provide ten forward speeds and one reverse speed. Different gear ratios (i_t) are obtained by fixing the sun, planet carrier and the ring gears of the planetary gear-set and corresponding clutches. The output torque is then transmitted to the propeller shaft.

The model of the transmission consists of the gear-backlash block, torque reduction due to the losses in the transmission, clutch and transmission shaft inertias, gear node inertias, planetary gear sets (gear ratio multipliers) and associated flexibility components. The selection of gear is done by selecting the appropriate gearset with corresponding combination of clutches.

The transmission gear backlash can be modeled with the equations given by Nordin et. al. [17] –

$$T_{ts} = \begin{cases} k_{ts}(\Delta\theta - \theta_{tb}) + c_{ts}(\dot{\theta}_c - \dot{\theta}_t) & \text{(Contact Mode)} \\ 0 & \text{(Backlash Mode)} \end{cases} \tag{10}$$

$$\dot{\theta}_{tb} = \begin{cases} \max\left[0, \Delta\dot{\theta} + \dfrac{k_{ts}}{c_{ts}}(\Delta\theta - \theta_{tb})\right] & \text{if } \theta_{tb} = -\alpha_{tb} \\ \Delta\dot{\theta} + \dfrac{k_{ts}}{c_{ts}}(\Delta\theta - \theta_{tb}) & \text{if } \theta_{tb} < \alpha_{tb} \\ \min\left[0, \Delta\dot{\theta} + \dfrac{k_{ts}}{c_{ts}}(\Delta\theta - \theta_{tb})\right] & \text{if } \theta_{tb} = \alpha_{tb} \end{cases} \tag{11}$$

where, T_{ts} = is the torque transmitted by the transmission output shaft, α_{tb} is the half of the backlash gap between the input and the output transmission shaft, θ_{tb} = transmission

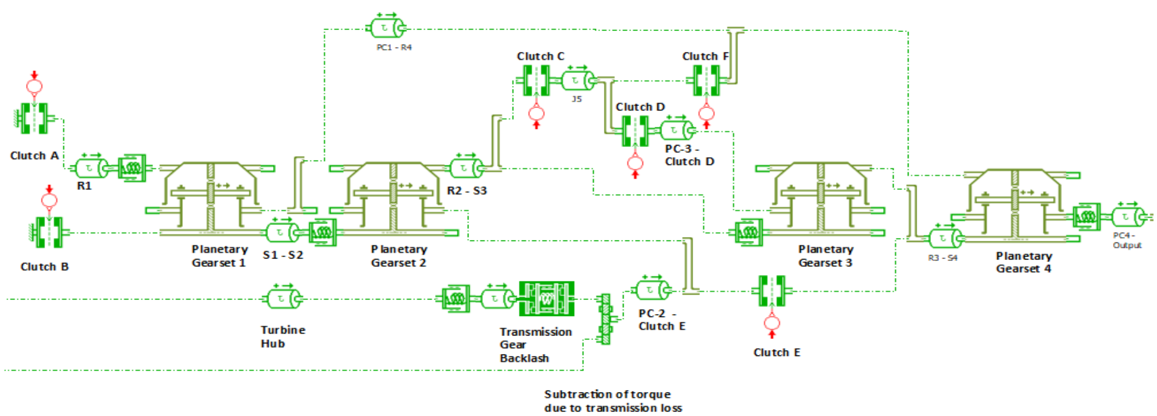


Figure 2.10 The AMESim® model for the automatic transmission

backlash position angle, $\Delta\theta = \theta_c - \theta_t$ is the difference between the transmission input angular displacement θ_c and output shaft angular displacement θ_t.

The torque lost inside the transmission, measured in Nm, can be reduced from the input torque and then multiplied by the corresponding gear ratio. The node inertias and rotational inertias for the shaft and planetary gearset components were included to accurately evaluate the total inertia for the corresponding gear.

2.3.5 Propeller shaft

The propeller shaft is modeled as damped torsional flexibility with rotational inertia J_{ps}, stiffness k_{ps}, and damping coefficient c_{ps} using the equations -

$$T_{ps} = k_{ps}(\theta_t - \theta_{ps}) + c_{ps}(\dot{\theta}_t - \dot{\theta}_{ps}) \tag{12}$$

where, θ_{ps} is the propeller shaft output angular displacement.

2.3.6 Final drive:

Final drive can be modeled as a rotating inertia J_{fd}, and a friction torque that could be assumed to be described as a viscous damping coefficient, c_{fd}. For simplification purpose, the inertia and the damping coefficient can be neglected for this evaluation. The complete final drive model also contains a gear speed/torque conversion ratio i_{fd}, a backlash element and a differential element. The differential element provides the driven wheels to be rotated

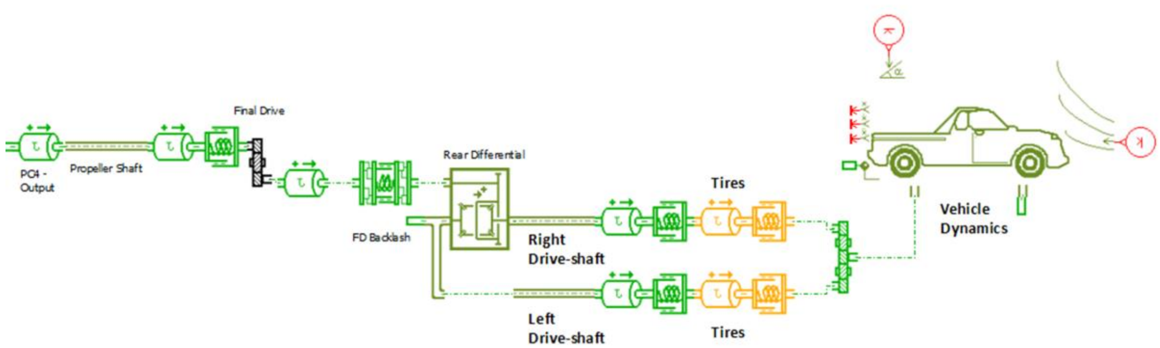


Figure 2.11. The AMESim® model for propeller shaft, final drive (gear multiplication, backlash and differential), drive-shafts, tires and vehicle dynamics

at different speeds. The model of the final drive can be represented using the following equations –

$$i_{fd} = \frac{\theta_{ps}}{\theta_{fd}} = \frac{T_{fd}}{T_{fd}} \tag{13}$$

$$T_{fd} = \begin{cases} k_{fd}(\Delta\theta_{fd} - \theta_{fdb}) + c_{fd}(\dot{\theta}_{ps} - \dot{\theta}_{fd}) & \text{(Contact Mode)} \\ 0 & \text{(Backlash Mode)} \end{cases} \tag{14}$$

$$\dot{\theta}_{fdb} = \begin{cases} \max\left[0, \Delta\dot{\theta}_{fd} + \frac{k_{ps}}{c_{ps}}(\Delta\theta_{fd} - \theta_{fdb})\right] & \text{if } \theta_{fdb} = -\alpha_{fdb} \\ \Delta\dot{\theta}_{fd} + \frac{k_{ps}}{c_{ps}}(\Delta\theta_{fd} - \theta_{fdb}) & \text{if } \theta_{fdb} < \alpha_{fdb} \\ \min\left[0, \Delta\dot{\theta}_{fd} + \frac{k_{ps}}{c_{ps}}(\Delta\theta_{fd} - \theta_{fdb})\right] & \text{if } \theta_{fdb} = \alpha_{fdb} \end{cases} \tag{15}$$

where, T_{fd} is the torque transmitted by the final drive output shaft, α_{fdb} is the half of the backlash gap between the input and the output final drive shaft, θ_{fdb} = final drive backlash position angle, $\Delta\theta_{fd} = \theta_{ps} - \theta_{fd}$ is the difference between the transmission input and output shaft angular displacement θ_{fd}.

2.3.7 Drive-shafts

The drive-shafts can be represented as a component with damped torsional flexibility with inertia J_{ds}, with internal damping c_{ds}, and stiffness k_{ds}.

$$T_{ds} = k_{ds}(\theta_{fd} - \theta_{ds}) + c_{ds}(\dot{\theta}_{fd} - \dot{\theta}_{ds}) \tag{16}$$

where, θ_{ds} is the drive shaft output angular displacement.

2.3.8 Wheels and Vehicle Dynamics

The wheels can be modeled as a rotating inertia with inertia $J_{w,}$ with internal damping c_w, and stiffness k_w.

$$T_w = k_w(\theta_{ds} - \theta_w) + c_w(\dot{\theta}_{ds} - \dot{\theta}_w) \tag{17}$$

where, θ_w is the wheel hub angular displacement.

The vehicle dynamics block consists of modeling the load torque, resulting from the driving resistances of the vehicle. These resistances can be calculated as follows:

- Rolling Resistance,

$$F_{RR} = m_w g(a + bv^2) \tag{18}$$

- Drag Resistance,

$$F_{DR} = \frac{1}{2}\rho_w A_f C_D (V \pm V_{wind})^2 \tag{19}$$

- Gradient Resistance,

$$F_{GR} = m_w g \sin(\theta_r) \tag{20}$$

- Inertia Resistance,

$$F_{IR} = m_w a_w \tag{21}$$

where, m_w = mass of the vehicle, a & b = rolling resistance coefficients, ρ_w = density of air, A_f = frontal area of the vehicle, C_D = coefficient of drag, V = vehicle velocity, V_{wind} = wind velocity, θ_r = gradient of the road, a_w = acceleration of the vehicle.

As rotational dynamics of the vehicle is the focus area, the load torque, which is needed for evaluation, can be obtained as –

$$T_{load} = F_{load} . r_{eff} \tag{22}$$

where, $F_{load} = F_{RR} + F_{DR} + F_{GR} + F_{IR}$ is the total load force, r_{eff} = effective radius of the tire.

3 Results and Discussion

The driveline model is simulated using the Simulink® solver using the Model-exchange interface. The post-processing of the simulation results is done using the respective software interface, i.e. the Simulink® model results are evaluated using Simulink® while the AMESim® results in AMESim®.

The model consists of two main inputs affecting the system behavior, viz. the requested torque and the load torque. The driver requested torque is generated using the Simulink® signal generator block based on the test requirements while the load torque depends on the rolling resistance, drag resistance and the gradient of the road. While the backlash is in the contact mode, the driver requested torque directly affects the powertrain inertia, eventually affecting the rest of the model. But, when backlash traversal occurs, the model is decoupled for that instant. The characteristics of the corresponding backlash block in this mode depends on the state of the components connected to either side of the block.

The model is analyzed for different tip-in and tip-out scenarios. These open-loop control method, where the requested torque is the controlling element, is used to understand the effect of torque variations on the driveline model when it is subjected to tip-in and tip-out conditions. The results of the qualitative evaluation of the complete driveline model are discussed in the subsequent section.

3.1 Driveline Model Evaluation

To evaluate the functionality of the model, two major scenarios were tested to verify the complete model behavior. Tip-in condition is when the vehicle is rapidly accelerated, while in the tip-out condition, the vehicle is allowed to coast down. The initial tests for tip-in and tip-out scenarios were set up to prevent any backlash traversal and so both the backlash elements, the transmission and the final drive backlash remain in positive contact mode. It was then followed by tests in which the backlash position changes from one side to the other. Most of the tests were evaluated at fifth gear as higher degree of oscillations were observed for that gear. The test results are shown below.

3.1.1 Tip-out and Tip-in conditions – Positive Contact mode

3.1.1.1 Tip out at the rate of 150 Nm/s in positive contact mode

For Tip-out case as shown in Figure 3.1, the model was tested at the following conditions–

- Engine torque request – Ramp down at the rate of 150 Nm/s from 200 Nm to 50 Nm at 9 seconds
- Torque converter clutch (TCC) capacity – Constant at 700 Nm (No-slip condition)

In Figure 3.1(a), as soon as the requested torque (red line) is ramp down at the rate of 150 Nm/s at t = 9s to 50 Nm, the torque delivered by the engine follows the requested torque.

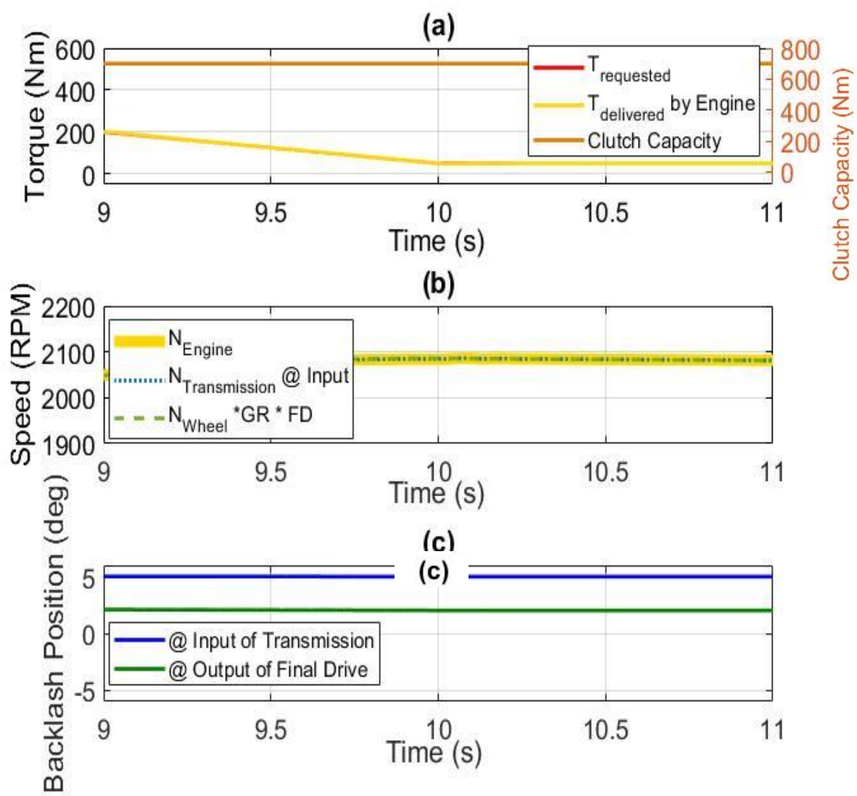


Figure 3.1. Tip-out case for positive contact mode with torque converter clutch locked and requested torque ramp down at the rate of 150 Nm/s at t = 9s

The engine speed gets stabilized to 2090 rpm at 10.5 seconds (Figure 3.1(b)). Also, the transmission speed and the reflected wheel speed at the transmission input shaft are same as the engine speed. Both, the transmission backlash and the final drive backlash stays in the positive contact mode i.e. +5° for transmission backlash and +2° for final drive throughout the evaluation period, as shown in Figure 3.1(c).

3.1.1.2 Tip in at the rate of 500 Nm/s in positive contact mode

For Tip-in case as shown in Figure 3.2 from 11 to 11.5 seconds, the model was tested at the following conditions –

- Engine torque request – Ramp up at the rate of 500 Nm/s from 50 Nm to 300 Nm at 11 second
- Torque converter clutch (TCC) capacity – Constant at 700 Nm (No-slip condition)

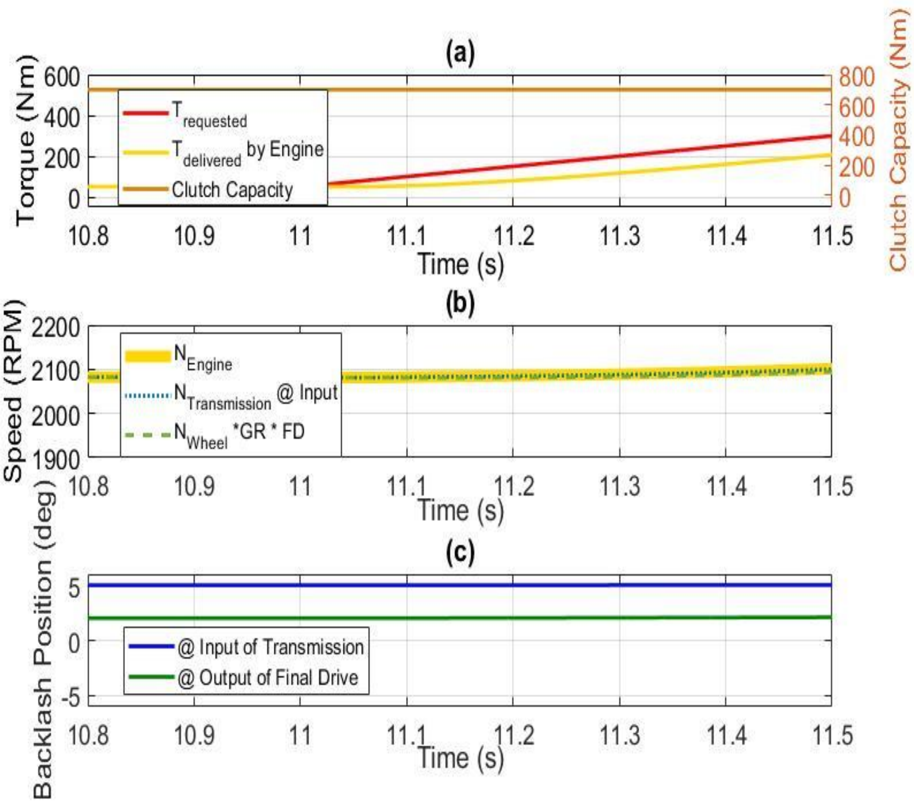


Figure 3.2. Tip-in case for positive contact mode with torque converter clutch locked and requested torque ramp up at the rate of 500 Nm/s at t = 11s

In Figure 3.2(a), as soon as the requested torque (red line) is excited by a ramp increase at t = 11s to 300 Nm in 0.5 seconds, the torque delivered by the engine starts to increase but after a delay of 50 milli-seconds, which is equivalent to the engine time delay. The torque increase conforms to the condition of positive contact mode of the backlash and hence, power transfer to the wheel through the driveshaft. The engine speed increases slightly from 2090 rpm to 2110 rpm from 11 to 11.5 seconds (Figure 3.2(b)). Also, the transmission speed and the reflected wheel speed at the transmission input shaft are nearly same as the engine speed. Both, the transmission backlash located at the input of the transmission shaft, and the final drive backlash placed at the output of the final drive, stays in the positive contact mode i.e. +5° for transmission backlash and +2° for final drive throughout the evaluation period, as shown in Figure 3.2(c).

3.1.2 Tip-out and Tip-in conditions – Backlash Traversal mode

3.1.2.1 Tip out at the rate of 150 Nm/s in backlash traversal mode

The model is evaluated for a tip-out scenario, using the following conditions –

- Engine torque request – Ramp down at the rate of 150 Nm/s from -30 Nm to 200 Nm at 9 seconds
- Torque converter clutch (TCC) capacity – Constant at 700 Nm (No-slip condition)

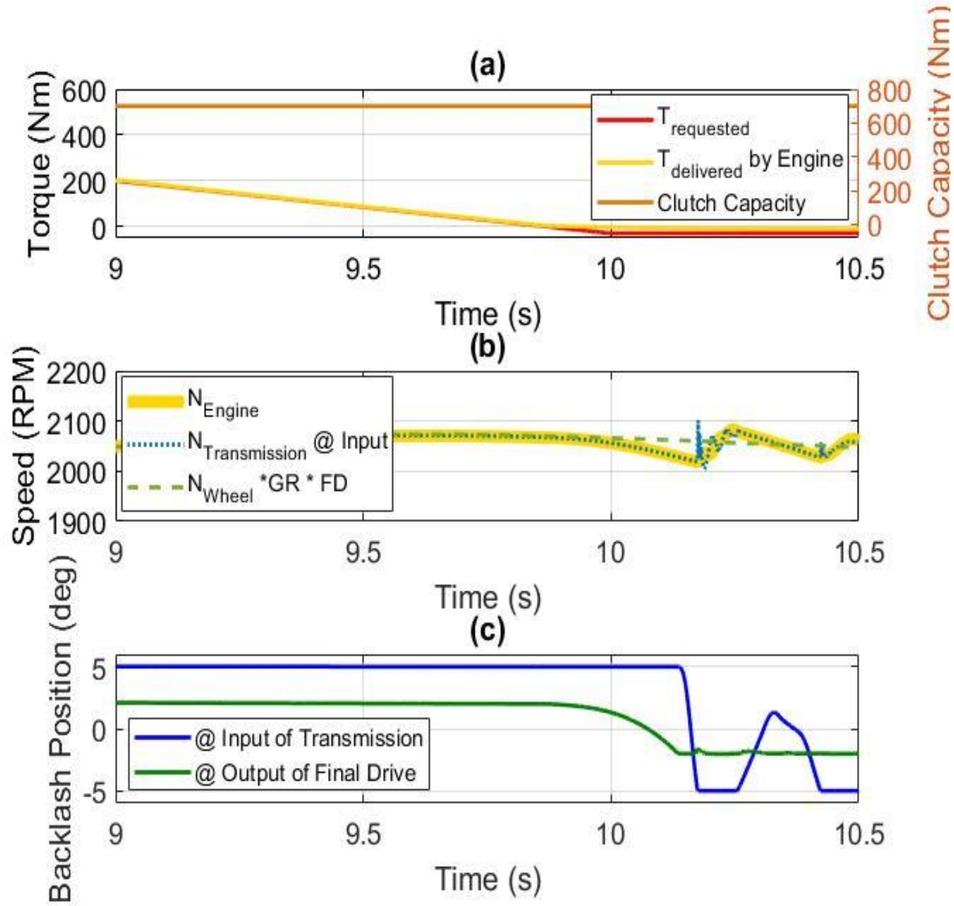


Figure 3.3. Tip-out case for backlash traversal mode with torque converter clutch locked and requested torque ramp down at the rate of 150 Nm/s at t = 9s

Figure 3.3(a) shows the requested torque (red line) is ramped down from a positive value to a negative torque to induce deceleration in the driveline model. This negative value leads to engine motoring at -10 Nm. The reduction in the torque delivered by the engine to 0 Nm (at t = 9.9 sec) triggers the final drive backlash to shift from the positive contact mode to the negative contact mode. Post final drive backlash traversal, the transmission backlash begins to move towards the negative contact mode (at t = 10.14s). The backlash position change leads to propagation of oscillations in the driveline due to the in-built rotary elasticities and damping. These oscillations also cause the backlash gap to open again and come back to negative contact mode, as it can be seen in Figure 3.3(c) at t = 10.33s.

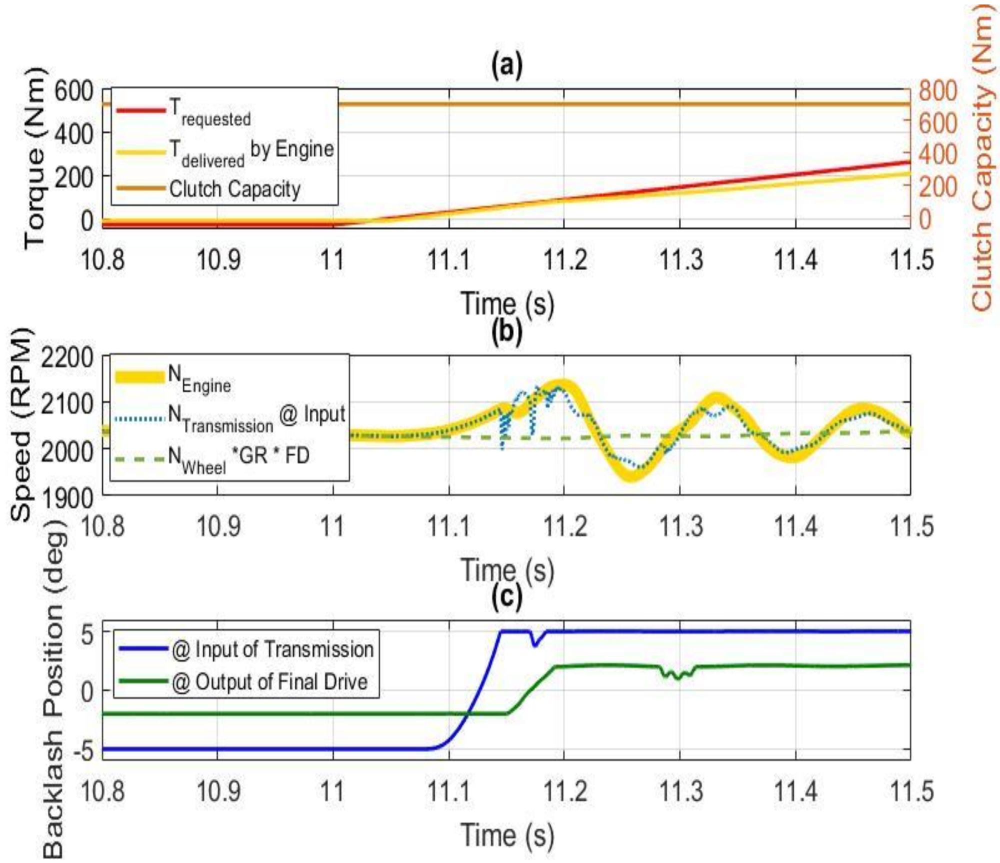


Figure 3.4. Tip-in case for backlash traversal mode with torque converter clutch locked and requested torque ramp up at the rate of 500 Nm/s at t = 11s

3.1.2.2 Tip in at the rate of 500 Nm/s in backlash traversal mode

The tip-in case from a negative requested torque to a positive value triggers backlash traversal across the transmission and final drive element (Figure 3.4). This will lead us to verify the backlash dynamics present in the model. The conditions at which the model was evaluated are as follows –

- Engine torque request – Ramp up at the rate of 500 Nm/s from -30 Nm to 220 Nm at 11 second
- Torque converter clutch (TCC) capacity – Constant at 700 Nm (No-slip condition)

In Figure 3.4(a), the requested torque (red line) is initially provided a negative value (-30 Nm) such that the engine is in constant deceleration mode. The value of the torque delivered by the engine is around -10 Nm and the engine speed is reducing, which shows that the engine is driven by the load torque provided by the wheels and eventually by the transmission. From Figure 3.4(c), it is observed that the backlash position of both the backlash elements is on the negative side, i.e. -5° for transmission backlash and -2° for final drive. The engine is then provided with a ramp increase in requested torque at t = 11s from -30 Nm to 220 Nm in 0.5 seconds. As soon as the torque delivered by the engine becomes positive at t = 11.07s, the engine and the transmission speed increases. Moreover, the transmission backlash starts to traverse from negative contact mode to positive contact mode. The wheel speed reflected at the transmission input shaft remains constant at 2026 rpm. Right after the transmission backlash reaches positive contact mode at t = 11.14s, the final drive gets the driving torque leading the backlash position at the final drive to change to positive contact mode (at t = 11.19s). This change introduces oscillations in the driveline at frequency of 7-8 Hz, as observed in the engine speed (thick yellow line) and the transmission input speed (dotted blue line) in Figure 3.4(b). The transmission speed reduces at the instant where the transmission backlash connects to the positive contact side and then increases again. The reflected wheel speed slowly increases after the final drive backlash has been traversed.

3.1.3 Effect of rate of change of requested torque

3.1.3.1 Tip in at the rate of 1000 Nm/s in backlash traversal mode

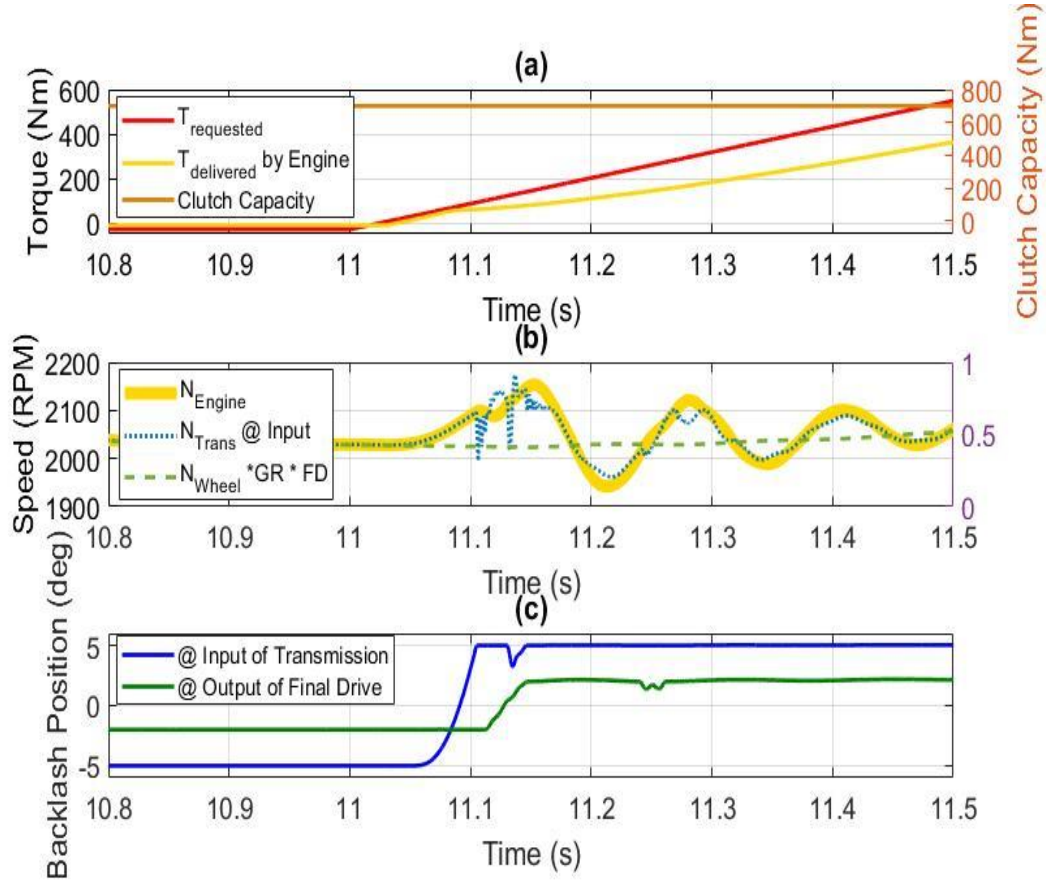


Figure 3.5. Tip-in case for backlash traversal mode with torque converter clutch locked and requested torque ramp up at the rate of 1000 Nm/s at t = 11s

The tip-in case from a negative requested torque of -30 Nm to a positive value of 520 Nm at the rate of 1000 Nm/s (Figure 3.5) can be compared with the tip-in case shown in Figure 3.4 with the rate of change of requested torque is 500 Nm/s. This will lead us to understand the non-linear dynamics of the driveline model. The conditions at which the model was evaluated in Figure 3.5 are as follows –

- Engine torque request – Ramp up at the rate of 1000 Nm/s from -30 Nm to 520 Nm at 11 second

- Torque converter clutch (TCC) capacity – Constant at 700 Nm (No-slip condition)

In Figure 3.5(a), the rate of change of requested torque from 500 Nm/s to 1000 Nm/s seconds makes the oscillation more prominent in the driveline. The frequency of the oscillations remains unchanged (7-8 Hz) but the amplitude of those oscillations increases as observed from transmission input speed from Figure 3.5(b).

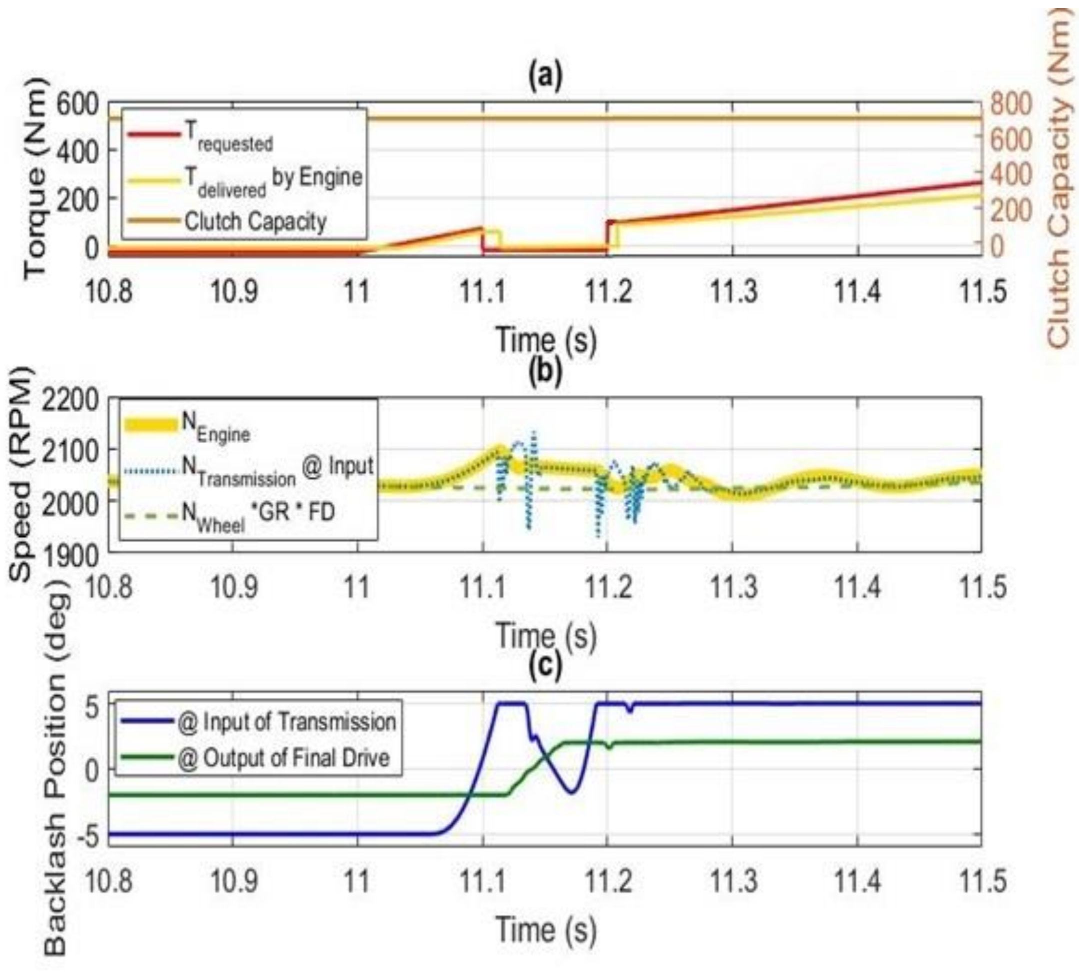


Figure 3.6. Tip-in case for backlash traversal mode with torque converter clutch locked and requested torque ramp up at the rate of 500 Nm/s at t = 11s with additional dip in the requested torque during traversal

3.1.4 Effect of torque shaping during backlash traversal mode

3.1.4.1 Tip in at the rate of 500 Nm/s in backlash traversal mode with reduction in torque during traversal

In the backlash traversal mode, some degree of open loop control can be achieved by modulating the requested torque signal such that the oscillations prevalent in the driveline model are reduced. The conditions at which the torque modulation was evaluated are as follows –

- Engine torque request – Ramp up at the rate of 500 Nm/s from -30 Nm to 520 Nm at 11 second
- Torque converter clutch (TCC) capacity – Constant at 700 Nm (No-slip condition)
- Torque dip at t = 11.1s

Figure 3.6(a) shows reduction in the requested torque to –20 Nm at t = 11.1s when the final drive backlash is open and traversing to the positive contact side. This reduction can also occur at the time of transmission backlash traversal, but this evaluation explains its effect on torque transfer during final drive backlash position change only. As seen in Figure 3.6(b), the engine speed and the transmission speed are observed to fluctuate quite less when compared to the case without torque reduction in Figure 3.4(b). This shows that the driveline oscillations can be controlled using this technique.

3.1.4.2 Tip in at the rate of 1000 Nm/s in backlash traversal mode with reduction in torque during traversal

With an increased rate of change of requested torque (1000 Nm/s), the torque reduction is made at t = 11.07s to observe its effect on both the transmission and the final drive backlash. As seen in Figure 3.7(a), the requested torque is reduced to -20 Nm for 0.1s during which both the backlashes change from negative to positive contact mode one after the other. At t = 11.1s, the transmission backlash rebound is seen to be suppressed when compared to Figure 3.5. This dip in the requested torque assists in slow backlash traversal and hence, reduced impact torque when it reaches the positive contact mode. It provides reduction in driveline oscillations besides reduced wear of the components.

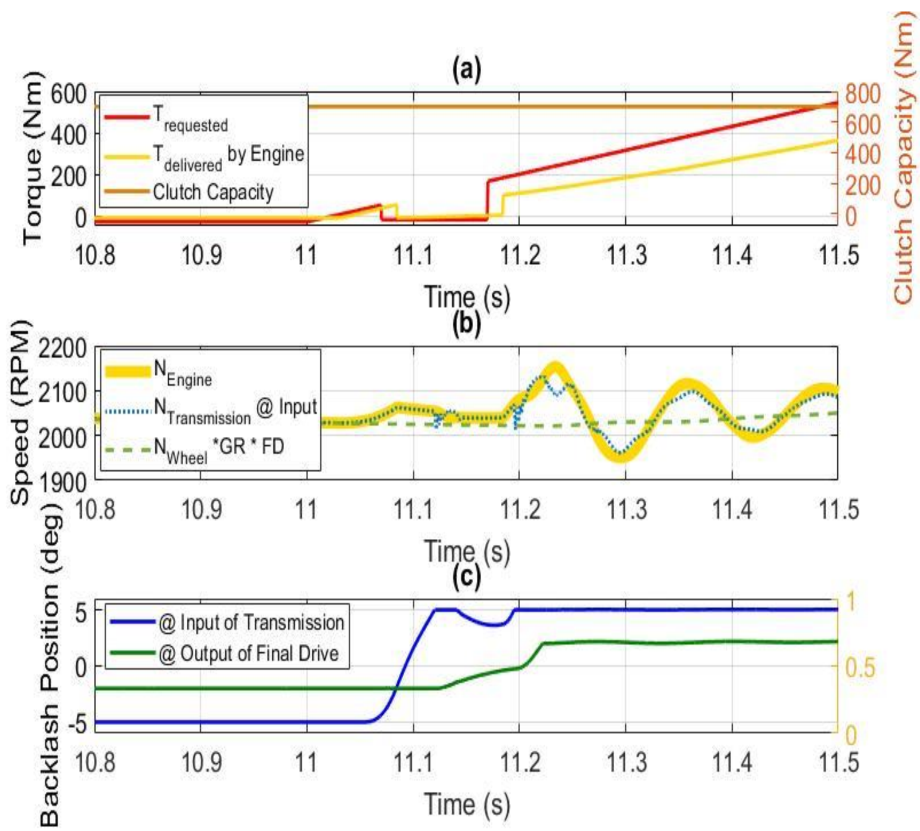


Figure 3.7. Tip-in case for backlash traversal mode with torque converter clutch locked and requested torque ramp up at the rate of 1000 Nm/s at t = 11s with additional dip in the requested torque during traversal

4 Conclusions and Future Work

This report broadens our understanding of the efforts put to develop the Ford F150 vehicle driveline model and captures its dynamics while evaluating it to understand clunk-and-shuffle, and effect of backlash status and position on performance of the driveline. This report also summarizes the qualitative evaluation of the complete driveline model simulated using a Model-exchange interface with Simulink® and AMESim® along with their respective post processing capabilities. Different scenarios for tip-in and tip-out along with commonly encountered combinations were run using the simulations. The key inferences from the simulations are as follows:

The joint simulation of the high-fidelity driveline model in AMESim® and control-oriented model in Simulink® will provide a platform to evaluate the estimators and control system while considering the fast dynamics of the non-linear system in future. The initial evaluation of the model with backlash in positive contact mode for tip-in and tip-out case confirms that the model exhibits the expected dynamics of the components such as the transmission and final drive backlash, the flexibilities of the propeller shafts, drive-shafts and tires. Also, based on the wheel speed reflected at the transmission input shaft for tip-in and tip-out scenarios, it can be inferred that the stiffness and damping coefficient of the tires are essential to capture the right damping characteristics of the vehicle.

Moreover, when the model is evaluated for a tip-in case from a negative torque to a positive torque, the effect of these flexibilities is quite noticeable. The oscillations induced in the driveline post tip-in are at 7-8 Hz frequency, which die out eventually due to damping present in the torque converter damper, transmission input shaft, propeller shafts, two drive-shafts and both the tires. This frequency of oscillations is similar to the results found in the literature [9]. Also, with higher rate of change of torque input (1000 Nm/s), the driveline experiences increased amplitude of the shuffle as observed in the engine speed. Preliminary evaluation using the torque shaping during the backlash traversal shows reduction in the amplitude of the oscillations for different rate of change in requested torque.

4.1 Future Work

As proposed future work, the following areas can be considered for improvement –

- The results obtained in the report can be validated using different means – both analytical and experimental. A thorough analysis of the effect of individual parameters of the components on complete driveline performance can be studied next.
- The driveline component models downstream to the transmission can be developed and validated to capture their natural frequency by performing a jounce test on the vehicle
- A preliminary torque shaping showed improvement in controlling the torque transfer during the backlash traversal mode, reducing the driveline oscillations slightly. Hence, a closed loop torque-based control system needs to be developed to actively control the torque to mitigate the low frequency oscillations and also to compensate for the effects of the backlash, clutch and tire slip, and drive shaft flexibility.
- A reduced order model can be further derived from the currently developed full-order model to develop the estimators and controllers for actively reducing the oscillations.
- Estimators can be developed for calculating the parameters required to identify and quantify backlash state in the driveline.
- Additional components in a vehicle with four-wheel drive such as transfer case, additional propeller shafts and drive-shafts will change the flexibility of the system. This will also affect the control system to be developed for improved drivability. Hence, an additional model will be needed to evaluate the additional degrees of freedom of the system.
- Also, the effect of addition of the vertical dynamics of the unsprung masses, the vertical and longitudinal dynamics of the sprung mass, flexibilities associated with the engine mounting system and more comprehensive tire model could be evaluated for different scenarios to evaluate its effect on the driveline dynamics.

- A clutch slip control method can be developed and utilized to reduce the driveline oscillations.

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