THESIS

CONTROL OF AN 8L45 TRANSMISSION INSIDE THE COLORADO STATE UNIVERSITY ECOCAR 3 2016 CHEVROLET CAMARO

Submitted by

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ABSTRACT

CONTROL OF AN 8L45 TRANSMISSION INSIDE THE COLORADO STATE UNIVERSITY ECOCAR 3 2016 CHEVROLET CAMARO

The hybridization and electrification of vehicles brings new challenges to the engineering and development of automotive control systems. Parallel, single motor pre-transmission hybrid electric vehicles are a preferred design for hybrid vehicles because of the mechanical simplicity, in that the electric motor and engine are on a common axis, connected to the transmission. Mechanically, this configuration enables the electric motor to take advantage of the torque multiplication of the final drive gear and transmission. From a controls perspective, this configuration is complicated because the engine, motor and transmission must work together to achieve the system-level objectives of fuel economy and driveability. These challenges are exemplified in the development of the hybrid 2016 Chevy Camaro developed by the Colorado State University (CSU) EcoCAR 3 team.

The results of this thesis demonstrate model development, model validation, and controls development to control the operation of the electric motor and engine together for driveability and performance during transmission gear changes. A model was developed in MATLAB Simulink to predict the behavior and performance of the 8-speed automatic transmission 8L45 that is stock to the 2016 Chevrolet Camaro. The performance of this model was validated by comparison to on-track vehicle data with <0.3m/s average error in prediction of the vehicle speed trace. A control system was developed to enable control of electric motor torque during shifts which eliminates ignition timing-based torque requests while maintaining driveability-derived shift dynamics.

This work has implications for the design of automatic transmission hybrid electric vehicles with discussion focusing on the potential for integration of learning technologies and minimization of gear lash.

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Completing this thesis would not have been possible without a great deal of help.

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I also want to extend a thanks to Dr. Jason Quinn and Dr. Anthony Marchese for their contributions in the conclusion of this research. My thanks also to anyone who had some small part in reviewing the finished product as it began to take shape or for providing helpful nudges in the right direction.

There is also need to thank Clay Bell, a former graduate student that developed a custom Simscape block for Matlab that was given to me by Dr. Bradley. There was nowhere else to properly give credit for this handy assistance, and without this one block, getting the dynamics of a locking torque converter down correctly would have taken a lot more work given how finicky this kind of simulation can get. Unknown hours were saved by being able to reuse this one bit of previous work.

I must thank my family, my parents, Marsha and Wes Knackstedt have been an unending source of support. I cannot even fully comprehend everything a parent does for their child as they grow, making me who I am, and providing guidance and encouragement through my college education.

My other family have also been a great support as well, each of them has provided some pillar of guidance that I strive to follow, but I will name my siblings, Dillon and Aspen Knackstedt for always making sure I can never think that being the oldest in some way makes me better.

Last, my departed grandparents Richard and Marvel Washnok, hardly a day goes by without me thinking of them despite all the years. I was so blessed to know them as long as I did. To me they were titans of character, strength, compassion, and faith. Living examples of the struggle to be more Christ-like.

Without any one of you, I would not be the person that I am now, and I cannot thank any of you enough for it. I hope that one day I can repay this unending kindness to all of you and others to help make this world a better place to live, grow, and learn.

Livestrong,

Clinton Knackstedt

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$\zeta_{1, 2, 3....}$ Gear ratio between the transmission input and output (excluding torque converter)

ω_{in} Transmission input speed (excluding torque converter)

ω_{out} Transmission input speed

R/S 1,2,3,4 Tooth counts for the ring and sun gears on each of the 4 planetary gears

$\omega_{(R,S,P),(1,2,3,4)}$Angular velocity for an element of one part of the planetary gear sets

T_{Axl} Axle Torque

F_{drag}Aerodynamic drag force

F_{accel}Vehicle acceleration force

F_{rr}Rolling resistance force

r_{wheel} Tire radius

I_{lumped} Lumped powertrain inertia

α_{eng} Powertrain angular acceleration

ρ Mass density of air

C_d Aerodynamic drag coefficient

A_f Effective drag area

v Vehicle velocity

m Vehicle mass

a Vehicle longitudinal acceleration

g Gravitational acceleration

c_{rr} Rolling resistance coefficient

z Count of friction surfaces

μ_{static} Static friction coefficient

r_{eff} Effective radius of the friction surfaces

A_{eff} Effective contact area

$P_{a,b,c,d,e}$Pressure applied to one of the 5 transmission clutches

FD rear differential gear ratio

T_{RL} Torque from vehicle road load

$\mu_{kinetic}$ Kinetic friction coefficient for the clutches

FGR First gear ratio

$I_{1,2,3,4,5,6,7,8}$ Rotational inertia for one of the rotating components in the transmission

I_{ds} Drive shaft inertia

I_{wh} Wheel inertia

I_{cv} Half-shaft inertia

α_{out} Angular acceleration coming out of the rear differential

T_{engine} Engine crankshaft torque

$T_{c,load}$Torque carried by clutch c

$T_{e,load}$Torque carried by clutch e

$T_{engine,inertia}$ Engine inertia torque

$T_{3,inertia}$ Inertia torque from element 3 in the transmission

α_{decel} Engine rotational deceleration during inertia phase

$T_{trans-in,inertia}$ Inertia torque from the input shaft of the transmission

R_{TC} Torque multiplication ratio from the torque converter

T_{int} Predicted inertia phase duration

v_{init} Vehicle speed at the start of the inertia phase of a shift

SGR Second gear ratio

TCR Torque converter speed ratio (input : output)

1.0 INTRODUCTION

1.1 Motivation

The United States has been regulating vehicle fuel economy standards since 1978 for the purpose of improving the nation's energy security. Over time, these regulations have increased the required minimum fleet fuel economy. This corporate average fuel economy (CAFE) is calculated per manufacturer so as the standards increase, each automaker has developed different strategies for meeting or exceeding the regulations. In 2010 a presidential memorandum began the process that created the goal of reaching a corporate average fuel economy of 54.5 mpg for all passenger cars and light-duty trucks, a significant rise compared to the previous history of CAFE standards (Environmental Protection Agency, 2012). It is important to note that the CAFE regulations use the unadjusted Environmental Protection Agency (EPA) fuel economy test results which are significantly higher than performance seen by consumers. The "window sticker rating" seen at the dealership which is much closer to what actual consumers will experience and is based on the EPA adjusted test result.

Based on a report made at the end of 2014, the total fleet performance in model year 2014 was 31.5 mpg, approximately 58% higher since test results were first recorded in 1978 (National Highway Traffic Safety Administration, 2014; University of Michigan Transportation Research Institute, n.d.). The adjusted fuel economy increased to 25.4 mpg during the same time period and has been relatively stagnant since 2014 (UMTRI, n.d.).

Each administration weighs the costs and benefits of how to determine an appropriate requirement, but the overall trend is a push to increase the US fleet fuel economy as a matter of public health and spending. These regulations reduce the amount of exhaust produced and reduces spending on not only fuel, but on transportation expenses in general (Greene & Welch, 2018; Greene et al., 2020a). Moving forward, automakers will have to continue to improve the fuel economy of the vehicles that they sell in order to stay in compliance with federal regulations and public demand. This continuous rate of improvement is made harder since much of the "low hanging fruit" of conventional vehicle efficiency improvement have already been realized. The cost to consumer for integrating these technologies has not been consistent through the life of the CAFE standards, though there has been an upward trend since the Obama administration started the rule-making process to rapidly increase fuel economy though 2025 (Greene et al., 2020b). As new technology penetrates the market, the costs do go down with the scale of production and with refinement of the technology (Lutsey & Nicholas, 2019). But with each year it takes more effort and more resources to continue to improve conventional vehicle powertrains. As we approach the 2025 CAFE requirements this cost will only go up without continuing to find new paths to introduce technology that is not tied to a purely combustion-based system. It has been shown that vehicle hybridization can effectively reduce this cost to automakers while minimizing the incremental cost to consumers (Al-Alawi & Bradley, 2014; Cheah & Heywood, 2011).

The Advanced Vehicle Technology Competition (AVTC) program sponsored by GM and the Department of Energy provides critical education experience to students from many different fields that can help support the research and development of hybrid/electrified vehicles. Each competition guides the student participants through the development process of taking an existing conventional (engine powered only) vehicle and converting it into a hybrid architecture of the student's design. Each competition comes with a new vehicle, pushing students to explore the unique opportunities and challenges when trying to make a design that fits within the constraints of the competition, meets the needs of the customers for that vehicle, while improving the fuel economy. This is particularly the case with the EcoCAR3 competition, which was focused on the Chevrolet Camaro, a vehicle well known for high performance, and less so for fuel efficiency.

1.2 CSU EcoCAR 3 Vehicle Architecture

When the CSU team set out to re-design the Camaro architecture, the most critical goals laid out by the competition were to maintain or improve the vehicle's dynamic performance and safety while improving the fuel economy of the powertrain through hybridization (Knackstedt et al., 2015). Vehicle Technical Specifications (VTS) were provided by the competition to establish a common baseline for stock vehicle performance that could later be used to compare the finished hybrid vehicle. Safety improvements are addressed in other publication about the CSU EcoCAR 3 vehicle and were mainly focused on the integration of advanced driver-assistance systems (ADAS) systems into the vehicle architecture, namely camera-based computer vision (Tunnell et al., 2018). This research focuses on the hybrid operating strategy for the powertrain.

The architecture that was chosen for design and manufacturing was a pre-transmission (P2) parallel hybrid electric vehicle (PHEV). This architecture will use E-85 as the fuel type, meaning the fuel contains up to 85% ethanol in the mixture vs the 15% maximum seen at generic fuel pumps. This fuel was selected since the engine supported the higher ethanol content and it has been shown to reduce net greenhouse gas emissions (Environmental Protection Agency, 2010). The CSU architecture will combine the GM 2.4l LEA EcoTEC with a Remy HVH250 electric motor powered by the A123 Systems 7 module 15s2p battery. Figure 1 below shows the layout of the selected components for the architecture and

Table 1 shows the basic specifications for the main powertrain components. Power ratings that are listed are not based on the physical hardware limits but on the practical limits within the context of the total system. The electric motor for instance can accept an input of 250kW but only if the operating voltage is 700Vdc. Given the selected battery pack the motor can only produce 150kW given the lower pack voltage.

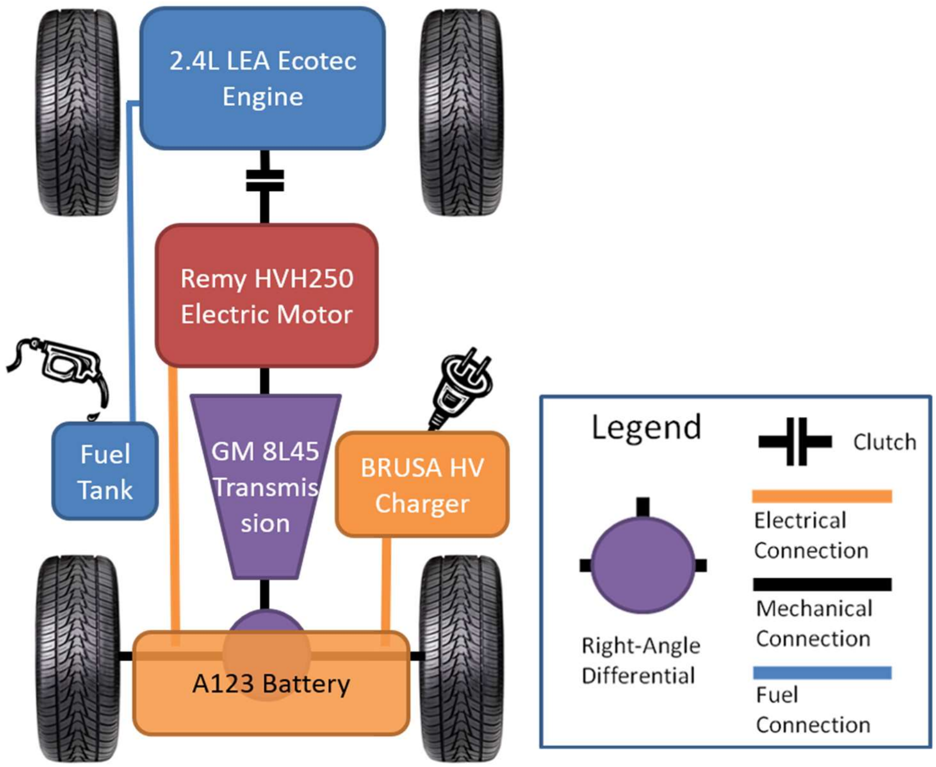

Figure 1: High Level CSU EcoCAR 3 Hybrid Architecture

Table 1: CSU EcoCAR 3 Hybrid Powertrain Component Specs

Component	Name	Power Rating (peak/max use)	Torque/Performance Rating (peak)
Engine	GM 2.4L EcoTEC	135kW	233 Nm
Motor	Remy HVH250-115 DOM	150 kW	408 Nm
Transmission	GM8L45	NA	NA
ESS	A123 7x15s2p	150kW	340Voc
Inverter	Rinehart PM250	150kW	720VDC/600Arms
Clutch	Tilton 5-5" Carbon-Carbon	300 kW	339 Nm

This powertrain was new to the CSU team, and came with many challenges for integration. But the flexibility in the design allowed for the components to each perform to their strengths and mitigate the other's weaknesses. Below is a list of some of the highlights:

- The high-power electric motor and its high torque application rate allows for an improved initial 0-60 mph (mile per hour) take-off time compared to a system purely powered by an engine.
- The positioning of the engine and electric motor on the same axle allows for transmission input torque to change rapidly with the electric motor rather than with the engine. In gas powered vehicles there are multiple methods to control torque production in the engine including fuel injectors, exhaust gas recirculation, and spark plug ignition timing. Many of these methods are hardware dependent, and generally all except spark plug ignition timing are slow to respond in the context of this research. Changing ignition timing has one major downside, fuel burned while the timing is not set at the optimal position will not generate the maximum torque possible. This actuator allows torque to reduce rapidly but comes at the cost of vehicle emissions (Ferguson & Kirkpatrick, 2015).
- Adding a clutch between the engine and electric motor allows the engine to disconnect from the electric motor. With the clutch open the vehicle can operate as an electric vehicle (EV) without the pumping losses incurred by spinning the engine without fuel.
- With the capability of the A123 battery pack, the vehicle is expected to achieve ~30 miles of range before the engine must turn on.

- By directly connecting the motor to both the engine crankshaft and transmission input the main powertrain is relatively simple to design and implement.

With this architecture, the team was setup to have a strong contender in the competition with lots of room to tailor the operation to any driving situation the driver could give it. With the design choices made, the integration was the major focus of the rest of the competition.

1.3 Architecture Challenges

There are multiple challenges that came with the selection of this architecture that had to be resolved through a combination of hardware and software solutions. Some of the key challenges are detailed in the following sections.

1.3.1 Packaging and Design

The 2016 Camaro provided a unique challenge in packaging. The powertrain chosen was simple, but it was also much longer compared to the stock powertrain with the addition of the electric motor and engine disconnect clutch. To accommodate this new powertrain, low profile radiators designed for racing had to be installed that allowed the engine to be pulled as far forward as possible in the engine compartment. At the same time, the transmission mounts had to be redesigned so that the transmission could be mounted further back in the exhaust tunnel. These changes together allowed the powertrain to fit without making larger structural modifications that the competition rules would not have allowed. In addition, the battery pack (which comes as a kit that must be assembled from the 7 modules provided) requires a custom enclosure that not only fits in the tight space of the trunk, but can also accommodate all components of the pack and be structurally sound in the event of a collision.

The limited space in the front of the vehicle required significant modifications to the engine configuration from how it was received. The oil pan for the engine was too deep and would have stuck out below the frame rails so a low-profile oil pan had to be found and other modifications had to be made to the engine itself to accept the low-profile pan. The biggest change was that a 12V starter motor could no longer be mounted to the engine. Without it the only way to start the engine was to use the propulsion electric motor to accelerate the engine up to the idle speed so that stable combustion could begin.

One of the biggest physical integration challenges for this powertrain was the electric motor shaft. The electric motor as delivered, did not come with a shaft for connecting to other components and a custom one had to be designed. This shaft had 2 major purposes. First, it had to allow for motor coolant dispersal, and accept seals to prevent coolant leaks. Second, it had to not only connect to the electric motor, but also to the engine disconnect clutch and the transmission input shaft on either end.

Other publications discuss more about the design of these parts, and how the overall powertrain was designed to fit and operate within the existing vehicle body. While quite heavy, the final assembled powertrain from engine to transmission was a single element that the car body could be lowered onto before it was secured.

1.3.2 Software/Controls

Any strong or mild hybrid will face the challenge on how to utilize the battery charge it has available for a drive cycle as efficiently as feasible. Many vehicles like the Chevy Volt will take a simple default approach of starting out in EV mode, and only starting the engine when the battery charge it too low to continue driving the vehicle. Attentive customers can use the mode controls switch to achieve higher fuel economy, but the vehicle controls do not attempt to do anything more without input from the driver.

Other hybrids, especially ones with smaller battery packs like the Honda IMA (Inzumiura & Ogawa, 2002; Honda, n.d.) and the Toyota Prius (Asher et al., 2016) must develop other methods. Many of these vehicles have electrified systems that cannot produce enough torque to drive the vehicle on their own, but even those that can, must choose between EV mode operation or saving their battery charge for specific events during the drive cycle where they can improve engine efficiency in a targeted fashion (Funston, 2000 ; Holdener, 2004). Methods for this targeted reduction in emissions and fuel consumption are often dependent on the vehicle hardware but can include things such as:

- Using motor torque to offset the driver's request to allow the engine to run with only half the cylinders fueled or less (Yuille et al., 2014).
- Using full EV operation only at low vehicle speeds (Johnston et al., 1998).
- Strategic motor torque use to reduce or eliminate the need for ignition timing-based torque requests for the engine. This thesis focuses on this option, specifically how it can be used during transmission gear shifts

The idea behind any of these strategies is not to simply turn the engine off to drive the vehicle with electricity only. Instead, these strategies leave the engine on for a larger portion of the drive cycle and instead look for opportunities to prevent the engine from operating in inefficient states by supplementing electric motor torque. The key challenge with any of these blended operation strategies is to capture the efficiency improvement opportunities, without significantly impacting the drive quality of the vehicle.

There are also practical controls challenges that had to be resolved for the vehicle to operate as intended. The controls architecture was designed in a web structure as shown in Figure 2. A centralized supervisory controller provided by Woodward interfaced with each component whether another controller, for the primary powertrain components, or direct interfaces for simpler hardware like the engine disconnect clutch.

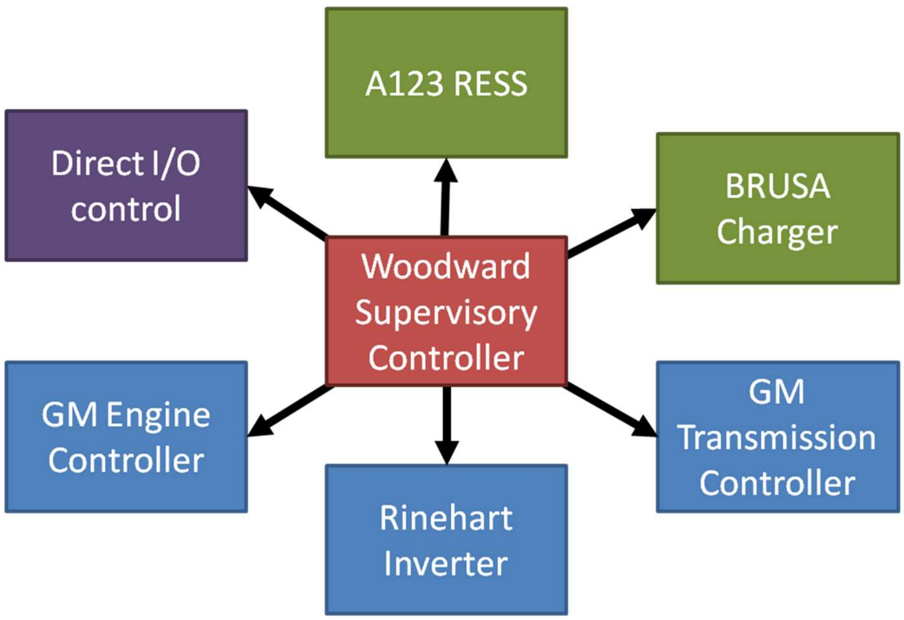

Figure 2: Controller Network Diagram for the CSU EcoCAR3 Vehicle

The biggest challenge with the design was interfacing between the Woodward supervisory controller and other controllers for the engine, transmission, battery, and electric motor. Each requires unique interfaces and expects information from the others. Even components from the same manufacturer did not necessarily have common interfaces that could be directly connected. This required the supervisory controller to act as an intermediary and convert the relevant information between formats.

Once initial communication between controllers was functional and the vehicle could operate, development began on a strategy that would attempt to improve fuel economy while providing the required performance for the vehicle. However, while functional, there were a couple of major drive quality issues that had to be addressed to improve vehicle functionality. First, for the vehicle to be capable of operating in electric vehicle (EV) mode, there needed to be a mechanism for it to exit EV mode when the battery charge is near depletion. If the engine cannot be started before the battery is depleted the driver could experience a loss of propulsion power while on the road. There are various possible solutions with different levels of acceptability, but they are not addressed here as they occur separate from transmission shift event which is the focus of this research.

Second, the vehicle has several issues that impact drive quality when the transmission shifts due to a disconnect between the transmission and the inputs to the software that controls its operation. These issues will be explained more detail in section 1.4.

1.4 Research Questions

The engine and transmission controllers provided by GM, like all the controllers provided by the other sponsors/vendors, come with fixed software which has a finite ability to be adjusted after delivery. GM took on the burden to try to make the engine and transmission more accessible to teams. They provided controllers with slightly modified software that made them easier to work with, like adjusting some of the torque interfaces for the engine. The transmission software was also modified so that the torque converter clutch logic would close much earlier than in the stock vehicle and an interface was added allowing teams to hold it open through a serial data request. Given this special software delivered to the teams, even other means of adjusting engine and transmission software such as aftermarket ECU re-flashing were not available. Still the changes provided were helpful but also left the engine and transmission lacking vital information necessary to achieve high quality shifts. The details on clutch-to-clutch shifting more generally, and how it can go wrong when software does not control the actuators optimally are covered in section 2.

The transmission was delivered with a set value specifying the inertia of rotating components in the powertrain and for each gear shift knows how these components will need to accelerate or decelerate during the shift. This information is used to calculate a torque request during a shift with the purpose of maintaining both output axle torque and shift duration. After the modifications to the powertrain to create the hybrid Camaro there are now 2 different scenarios that both leave the transmission unable to correctly calculate the torque needed.

- In EV mode, the electric motor and half of the Tilton clutch are the only components connected to the input of the transmission leaving the rotational inertia upstream of the transmission input much lower than stock. This could cause input speed surge during the torque phase and cause the transmission to overestimate the torque reduction required in the inertia phase which could lead to noticeable disturbance to the driver in the output torque depending on the severity of the over calculation.
- In Non-EV mode, both the new engine, electric motor, and clutch are connected to the transmission input together. This increases the upstream inertia significantly which has a few possible impacts to the driver. These include causing the engine speed to decrease during the torque phase of the shift and output torque surge during the inertia phase. It could also extend the duration of the inertia phase which would increase wear on the transmission on-coming clutch.

To resolve these issues the following research questions were developed to capture the problems and determine if a solution has been found.

1. Can a controls strategy make the changes to the CSU EcoCAR 3 powertrain invisible to the transmission and the software controlling it, thereby allowing the transmission to operate as if still in the stock configuration?
2. While executing the above strategy, can the electric motor serve as an adequate replacement actuator to respond to torque requests that are normally executed via changes in the timing of cylinder ignition in the engine?

To answer these questions this thesis will show the methods used to develop a virtual model of the vehicle in Simulink that represents the CSU EcoCAR 3 vehicle in the stock configuration. This process includes validation of the virtual model with real-world data collected on the stock vehicle. These methods will also cover modifications made to represent the final powertrain designed by the team, and demonstrating that even in the new architecture, the transmission performance is still consistent. Last, there will be an analysis of how the vehicle can operate in a blended hybrid operation that eliminates the need for ignition timing-based torque requests.

2.0 BACKGROUND

2.1 Powertrain Specifications

2.1.1 Automatic Transmission and Internal Gear Ratios

As shown in Figure 3 the 8L45 transmission consists of 4 planetary gear sets, 3 standard clutches and 2 brake clutches. When the software requests a gear, a corresponding set of 3 of the 5 clutches will close to connect the input and output shafts. Once in that gear, a shift from the current gear to another is accomplished by opening one of the 3 clutches and closing another, based on the new selected gear. The table below shows the clutch pattern and approximate system gear ratio for each of the selectable gears excluding the final drive gear ratio.

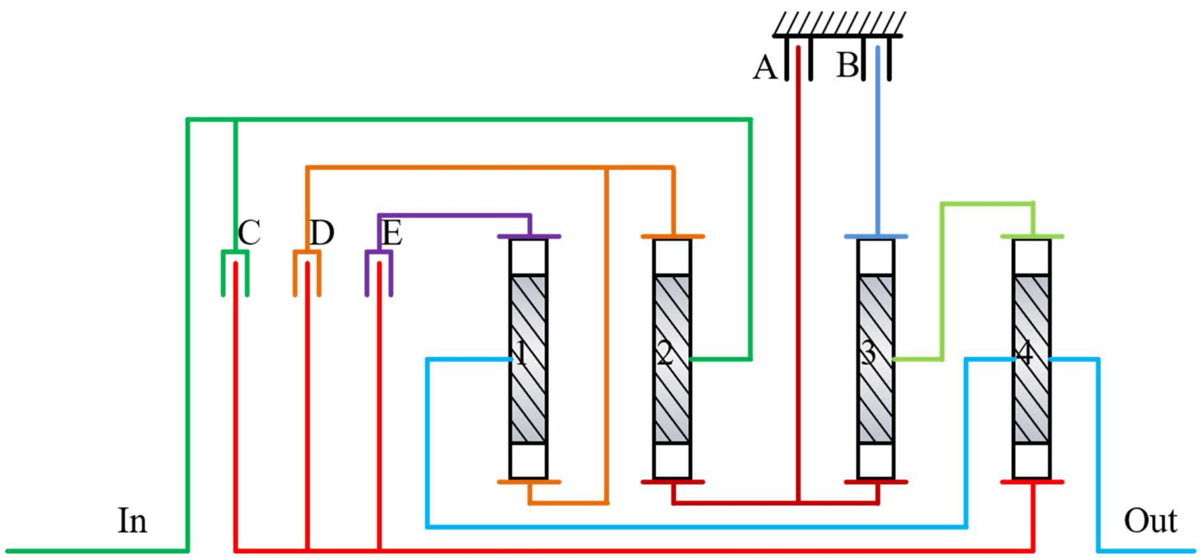

Figure 3: Stick Diagram of the GM 8L45 Transmission Rotating Components (Apakidze, 2014; Read, 2015)

Table 2: 8L45 Clutch Shift Pattern for the GM 8L45 Transmssion (Apakidze, 2014)

		Clutch				
Gear State	Gear Ratio	A	B	C	D	E
Rev	-3.93	X	X			X
N	0	O	O			
1st	4.62	X	X	X		
2nd	3.04	X	X		X	
3rd	2.07		X	X	X	
4th	1.66		X		X	X
5th	1.26		X	X		X
6th	1.00			X	X	X
7th	0.85	X		X		X
8th	0.66	X			X	X

X—carrying torque, O—closed, not carrying torque

For this thesis, GM provided the exact gear ratios needed for each of the planetary gear sets which are needed to model the 8L45 transmission dynamics accurately. In the Appendix, a method for approximating these ratios is shown to allow the same methods to be applied to other vehicle architectures where the exact tooth counts may not be known.

2.1.2 Torque Converter Operation

Critical to the operation of automatic transmissions is the torque converter, depicted in Figure 4. There are some automated manual transmissions that do not require torque converters but they are still present in most automatic transmissions to allow smoother connections between the engine and the transmission gear sets. The torque converter consists of 3 parts that are connected via the motion of transmission fluid inside the case. The engine drives the pump which imparts motion on the fluid through the stator and spins the turbine side which is connected to the transmission input.

This setup allows the torque converter to manage the slip needed between the engine and wheels while the vehicle is not moving. With the engine running, and the vehicle stopped, the torque converter still transmits torque to the wheels, but a small amount that can be easily cancelled by light pressure on the brake pedal by the driver. Transmission setups like this have the benefit of providing continuous torque versus a manual that must close the clutch before getting torque. It also provides additional torque multiplication, offering about a 2x increase when the speed at the turbine is 0 (Maguire et al., 2013).

Torque converters do have one main downside, which is energy transmission efficiency. The ratio of power in to power out changes based on the difference in speed between the turbine and pump sides. The peak efficiency occurs when the 2 sides are close to the same speed and can be up to 90%. To improve efficiency, many torque converters also include a clutch that can bypass the turbine and pump to directly connect the 2 sides to minimize the lost energy.

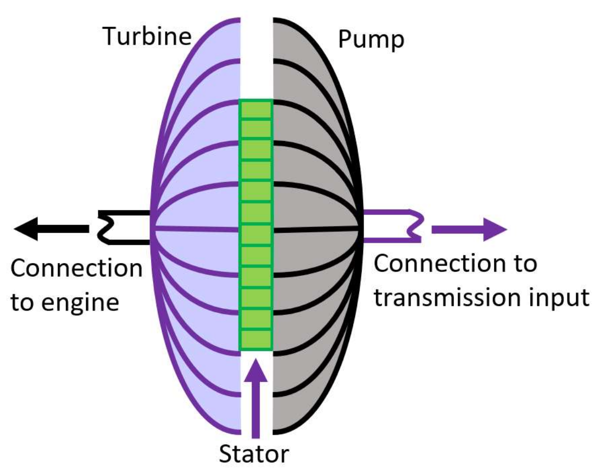

Figure 4: Torque Converter Layout (TCC not shown)

As an example, in Figure 5 a nominal torque converter chart is shown which indicates how the torque multiplication, speed ratio, and K-factor are connected while the clutch is open. When the vehicle motion is stalled, there is typically a torque multiplication of 2 and that torque ratio reduces towards 1 as the input and output synchronize. Similarly, the efficiency starts low at stall and later stagnates at about 90% efficient until the clutch starts to close (Gillespie, 1992).

The K-factor is the last piece needed to fully capture torque converter performance characteristics. Depending on the maximum output of any torque sources connected to the pump side of the turbine there will be a different maximum speed that the system will be able to reach on the pump side of the torque converter when the turbine side is locked. This relationship between stall speed and max input torque is normalized as the constant K-factor and is used for torque converter selection. The K-factor is also used in simulation to calculate how the input and output torques and speeds are related.

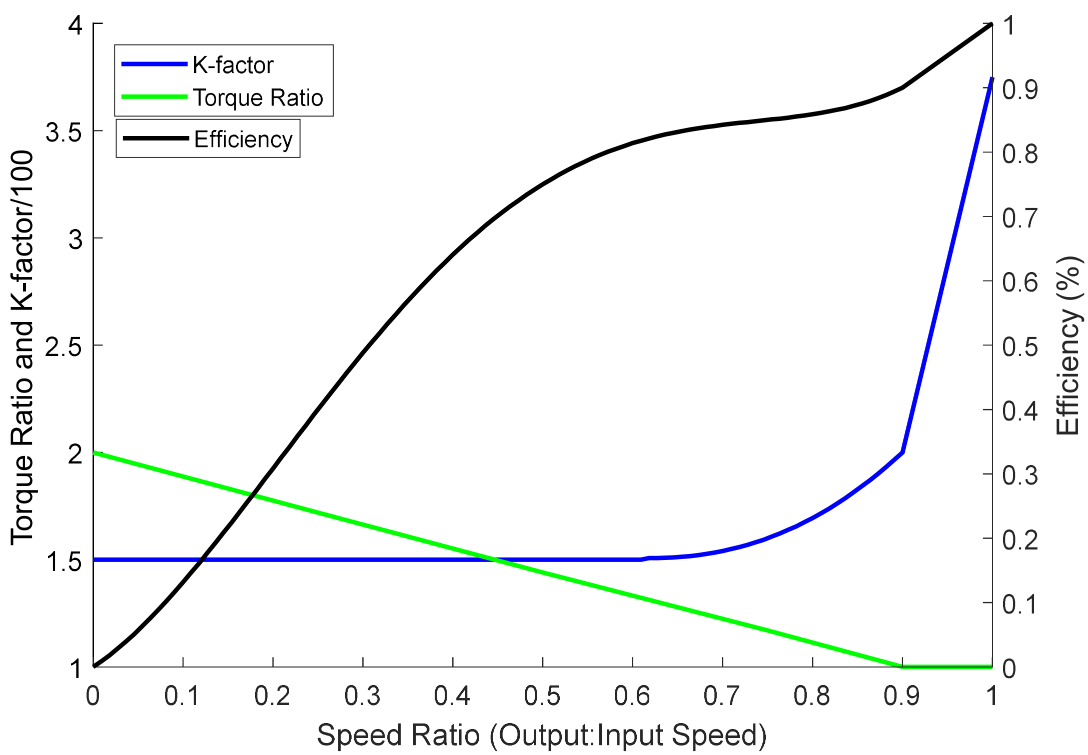

Figure 5: Typical Torque Converter Performance Characteristics Based on Output to Input Speed Ratio

When writing software to manage torque converter operation the key information is the relationship between the input and output (both the speed and torque ratio). When the torque converter characteristics are known it is possible to determine the current operational state with only 2 of the 4 values. In Section 3.2.3 it is shown how the torque converter was integrated into the vehicle model and software to capture the dynamics that it represents in the vehicle architecture.

2.2 How Up-shifts Occur and Common Controls Issues

Given the nature of the challenge of improving shift drive quality, a more detailed model and understanding of the sequence of an automatic transmission shift is necessary. It highlights the considerations needed for each phase of the shift and what the risks are for a non-coordinated response. Throughout this section the impacts of the torque converter are not factored in directly, but instead look more generally on how the clutch control impacts shift performance. Depending on the state of the torque converter clutch (TCC) it may or may not have a torque multiplication effect that would impact the engine torque required to execute the shift smoothly. All impacts from the torque converter are covered in section 3.3.3.

2.2.1 Clutch Operation Sequence

It is important to go over the sequence of a typical Power-on Up-shift as show in Figure 6 and describe in (Haj-Fraj & Pfeiffer, 2001; Maguire et al., 2013). The transmission monitors vehicle conditions to determine when it should start the gear shift process, primarily using the vehicle speed and driver requested torque as inputs into a shift map. There are other conditions that are also monitored but are very hardware dependent such as component/fluid temperatures or the incline/decline of the road. For this research, the transmission controls software could not be modified to alter when shifts would occur. This makes it important to be aware of the conditions that will cause a shift and any software made by the team ultimately had to be able to control the vehicle with gear and shift selection as an input that cannot be altered. Once the transmission determines the shift is to be executed, it will go through 2 phases: 1) the torque phase and 2) the inertia phase.

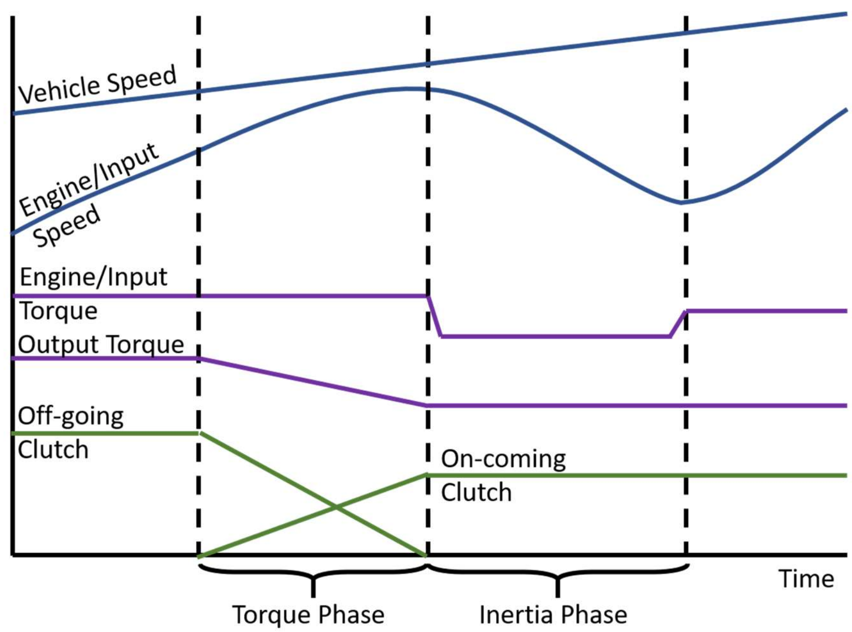

Figure 6: Nominal Power-on Up-shift Dynamics. Exact values for each signal will depend on vehicle conditions.

Before the torque phase can begin, the fluid reservoir for the on-coming clutch must be filled for quick and smooth clutch engagement. To execute this the control solenoid for that clutch will allow transmission fluid to flow towards the on-coming clutch until engagement between the plates starts.

During the torque phase the on-coming clutch is slowly applied while the pressure holding the off-going clutch is reduced. While this transfer of pressure occurs, the transmission stays in the original gear from the perspective of the speed ratio between the input and output of the transmission. Even as the amount of torque carried by the on-coming clutch increases, since the off-going clutch locked, the on-coming must slip continuously.

By the nature of the dynamics of the transmission, as the on-coming clutch pressure increases it will carry more torque between the 2 side of the transmission, and regardless of the pressure on the off-going clutch (as long as it remains locked) the actual torque carried by the locked clutch will reduce. The end of the torque phase is marked by the point when the off-going clutch is no longer carrying torque and is then exhausted, meaning that the 2 sides of the clutch are no longer in physical contact, though they will continue to spin at nearly the same speed. Through this transition, as indicated earlier, the pressure on the off-going clutch must be controlled carefully to prevent it from slipping or otherwise opening early.

By the end of the torque phase, the off-going clutch will be carrying no torque, and it is critical that the fluid pressure on that clutch is near zero so that it can open before the inertia phase starts. If the off-going clutch fluid pressure reduces too quickly, the torque load may exceed the static friction limit and cause the clutch to slip while still carrying torque. Early off-going clutch slippage will cause the engine speed to increase rapidly and the vehicle speed will drop. This is because of the change in kinetic friction coefficients when the clutch starts to slip while that clutch is still transmitting torque.

When the inertia phase starts, the on-coming clutch is carrying all the torque to connect the input and output of the transmission (ignoring the other 2 clutches that remain locked throughout the entire shift), but it is still slipping constantly at the same speed as the gear ratio at the start of the shift. The inertia phase consists of smoothly reducing the transmission input speed and engine speed so that the amount of slip on the on-coming clutch reduces until it can lock.

Once the clutch locks, the shift is complete, and the powertrain is operating in the newly selected gear ratio.

2.2.2 Controlling Input Torque During an Up-Shift

Through the shift process, the transmission must coordinate with the engine to make sure that torque delivered to the wheels has minimal disturbances and stays as close as reasonable to what the driver has requested. In each phase, including before and after the shift, the engine should operate in a specific pattern.

During the torque phase, with the on-coming and off-going clutches both carrying torque to the wheels, they resist each other, meaning that not all the power put into the input of the transmission will be seen at the output. But this can be mitigated by increasing the engine torque. With enough torque the effects of the fighting clutches can be cancelled out allowing the vehicle to continue to operate in line with the driver's expectation.

In the inertia phase, the clutches are no longer fighting, and a different response is needed. To complete the shift, the slip speed of the on-coming clutch needs to go to zero quickly to reduce wear on the clutch. To do this the engine needs to reduce torque since the only alternative, reducing output torque, would not meet the driver's expectations. If the engine does not produce enough torque to drive the clutch, then the transmission input speed will drop as kinetic energy is supplemented to drive the vehicle until the clutch locks.

To accomplish this, the engine needs to drop the torque rapidly and once the clutch locks the torque needs to again increase rapidly. Changing the position of the engine throttle can increase or decrease torque, but it does so slowly in comparison to the length of a shift. To get an appropriate response it is better to change the spark timing. By retarding the spark, fuel ignition in the cylinder occurs later in the stroke than what is optimal, the engine produces less torque rapidly but at the cost of fuel efficiency and emissions. The advantage is that the timing of the ignition point can be changed almost instantly, though it has upper and lower limits. During the inertia phase it is common practice to retard the spark to reduce torque quickly until the clutch locks before resuming normal spark timing.

3.0 METHODS

In order to develop and test a method for improving the vehicle drive quality during a shift, a virtual model was needed for faster software development cycle time than could be achieved with in-vehicle testing. This model was developed in 3 stages. First, the powertrain design and operating dynamics of the stock vehicle were duplicated in a MATLAB simulation. This was followed with making modifications to the plant model that represent the changes that were made to the powertrain, before building the control software that would provide the same vehicle response to the identical driver inputs with the new hardware.

To support this and other development activities a baseline driving evaluation was conducted with the stock vehicle where the vehicle performance was measured through a variety of dynamic tests including block pedal accelerations which is the main source of data for this thesis. This data was used in MATLAB to validate the software-in-the-loop (SIL) model of the various powertrain components.

3.1 Stock Performance Data Collection

To collect the stock vehicle data, the team recorded controller area network (CAN) data from the car while driving on a straight and reasonably flat road. Each test was executed twice going opposite directions to correct for slight variations in elevation on the surface. Beginning from 0 speed, the accelerator was pressed a fixed distance by checking it on the data recorder and the position was held until the vehicle stopped accelerating or until driver safety on the test surface required that we begin braking. Different pedal positions were used to sample across the whole range of travel on the accelerator pedal since each pedal position had a unique acceleration curve.

For the purposes of this study, several key signals were recorded in order to better understand the capture the vehicle operation of the during shifts.

Current Gear Ratio

Transmission Output Speed

Engine Torque

Accelerator Pedal Position

Estimated Axle Torque

Driver Intended Axle Torque (does not include transmission interventions)

Torque Converter Clutch State

Engine Speed

Transmission Turbine Speed (downstream of the torque converter)

Transmission Engine Torque Requests

Commanded Transmission Gear

Actual Transmission Gear

Vehicle Speed

These signals together make it possible to reconstruct the dynamic state of the powertrain components at any point in the drive cycle except for some details such as the transmission solenoid states and reservoir fill levels, which even with direct access to the hardware is difficult to measure.

Since the focus of this research is on the dynamics within a shift, the data was cropped for vehicle simulation so that it consists of a single shift event with a couple seconds surrounding either side of the shift. In Figure 7 and Figure 8 below is a sample of the stock vehicle data that was clipped from a 8% accelerator block pedal maneuver, with the key signals called out. In

Figure 7 the shift in the magenta line indicates the point in the drive cycle where the transmission determined it was time to begin the shift to second gear while the green line represents marks the end of the inertia phase. Within the speed signals in this data sample is an example of early off-going clutch slippage as the engine speed has an unexpected inflection point about a quarter second before the peak which also leads to issues at the end of the inertia phase as the transmission was not expecting it to take as long to synchronize the on-coming clutch. The output speed, (analogous to vehicle speed) simply represents the mostly smooth acceleration under the block pedal maneuver with some disturbance during the shift.

Figure 8 shows the same data sample but focusing on the torque signals from the engine and transmission controllers. The driver intended engine torque represents the ideal engine torque if there was not a gear shift in progress. This means the driver intended torque only accounts for the current gear ratio without considering how the current clutch state would alter the torque needed to drive the vehicle smoothly. It is only visible in the figure in this sample during the inertia phase as otherwise the produced engine torque was identical. The torque phase request represents the minimum torque that must be produce by the engine to maintain sufficient output torque to the wheels while dealing with the competing clutches. Then during the inertia phase and additional calculation is made which is shown as the inertia phase request in Figure 8Figure 7. The inertia phase request is a calculated reduction in engine torque meant to target a specific engine deceleration rate. This deceleration rate is chosen to allow the powertrain to synchronize all rotating components for the new gear by the end of the inertia phase.

Both the torque and inertia phase requests in Figure 8 show spikes that go outside the y-axis range in the graph. When these requests are not active, the signal sent by the transmission controller defaults to the maximum value. All values where the request is active are shown.

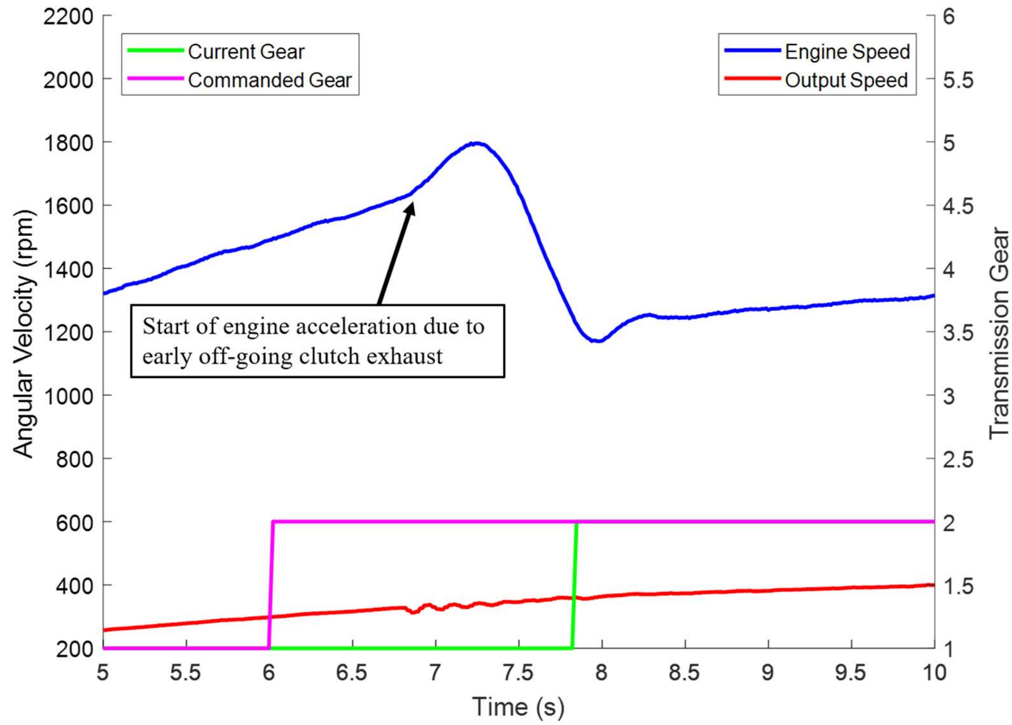

Figure 7: Power-on 1 to 2 Up-shift Speeds 17% Pedal From Recorded Vehicle Data Including Early Off-going Clutch Exhaust

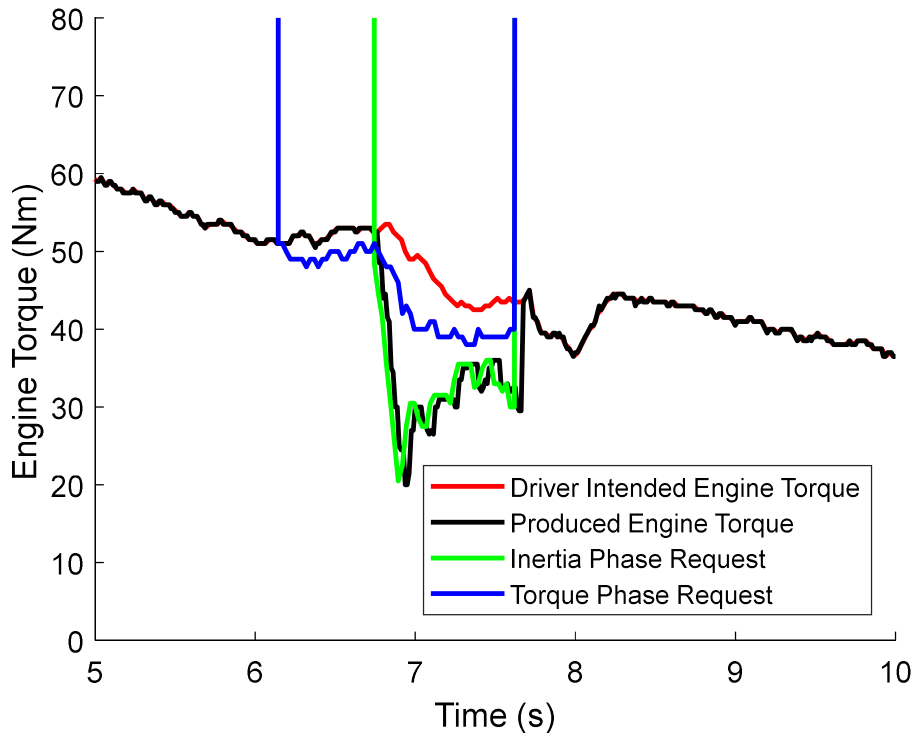

Figure 8: Power-on 1 to 2 Up-shift Torque 17% Pedal From Recorded Vehicle Data Showing Transmission Shift Torque Requests

3.2 Developing the Stock Vehicle Model

With the stock vehicle data collected, it was now possible to start development on a virtual model that could replicate the stock vehicle data including the clutch shift dynamics discussed above. The model was built starting from a basic vehicle body in Simulink adding progressively more complexity until the simulation had enough detail to represent the dynamics of the shift (Mathworks, n.d.). Each phase of development is explained below in roughly the order it was introduced to the model.

3.2.1 Engine, Vehicle Body, and a Basic Clutch

To begin the model development the starting point was to get the basic vehicle parameters represented with Simscape blocks and then validate that the model functions by testing it against the data recorded in the real vehicle. This would make sure that both the model parameters were correct, and that the calculations used to drive the inputs to the model were the right value to match the drive cycle.

For the baseline model, as shown in Figure 9, vehicle body and tire Simscape blocks are used with all the stock vehicle parameters entered in. The vehicle block is a 1 degree-of-freedom (DOF) model of the vehicle body and the road loads it experiences while driving. Lumped inertia represents the rotating powertrain components lumped together, assuming the vehicle is operating without a transmission regardless of speed with a single clutch connecting the engine to the rear wheels. The engine is represented with an ideal torque source with inputs determined by a script that runs before the simulation starts. To manage clutch pressure, like in (Watechagit, 2004), the actual transmission hydraulics were not directly modeled here or later as the model development advanced, instead the clutch pressure was calculated directly based on the amount of pressure needed to maintain the correct transmission output torque given the slip state of that clutch.

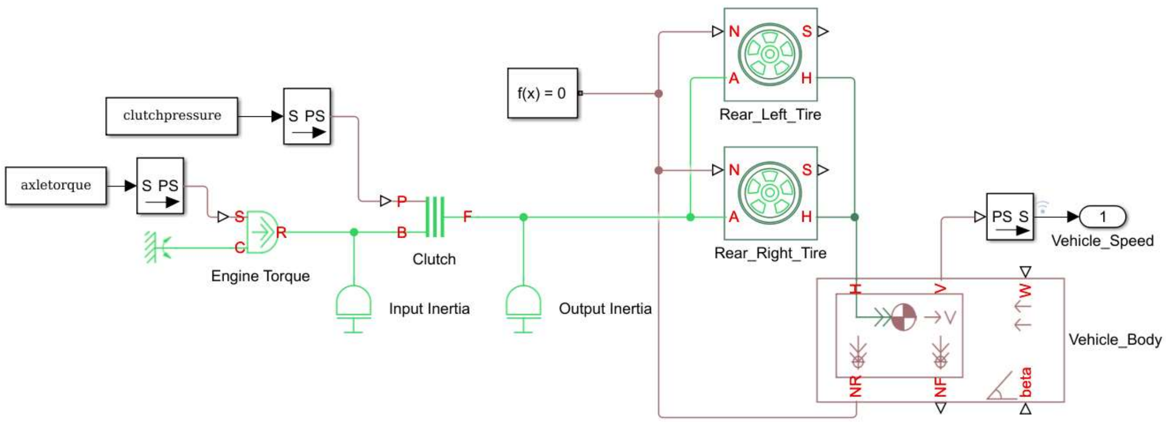

Figure 9: Baseline Stock Vehicle Model Layout Including Engine, Lumped Inertias, Primary Clutch, Driven Tires, and Vehicle Body

With such a simple model using the vehicle speed trace from the sample data it was possible to back-calculate the appropriate clutch pressure and engine torque to keep the clutch locked and accelerate the vehicle body to follow the collected data. But this model was also used to demonstrate if Simscape was up to the task of this study by seeing if using the fundamental physics of vehicle dynamics if you could consistently control the vehicle body model. For this early stage in the model, there is no transmission or other gear ratio included in the model so engine torque is equivalent to axle torque delivered to the wheels. Below are the formulas for calculating axle torque (T_{Axl}) and clutch pressure. Axle torque is based on the aerodynamic drag (F_{drag}), vehicle acceleration (F_{accel}), rolling resistance (F_{rr}), tire radius (r_{wheel}), plus the effecting of the rotating powertrain components described by the lumped rotational inertia (I_{lumped}), and the engine acceleration (α_{eng}). Without anything more complex the clutch pressure is tied directly to what is needed to transmit the full axle torque without slipping.

Equation 1: Basic Axle Torque Formula

$$T_{Axl} = \left(F_{drag} + F_{accel} + F_{rr}\right) * r_{wheel} + I_{lumped} * \alpha_{eng}$$

Equation 2: Axle Torque Component Forces

$$F_{drag} = \frac{\rho * C_d * A_f * v^2}{2}, F_{accel} = m * a, F_{rr} = m * g * c_{rr}$$

Equation 3: Basic Clutch Pressure Formula

$$Clutch\ Pressure = \frac{T_{Axl}}{z * \mu_{static} * r_{eff} * A_{eff}}$$

At this stage, no torque response delay is modeled so the torque changes are instantaneous. Additionally, at this stage the torque needed to account for the acceleration of rotating components individually was not factored in. That was added later in the model development. Even with these simplifications, the results shown in Figure 10 a and b demonstrate that the model works and follows the speed trace accuratel.

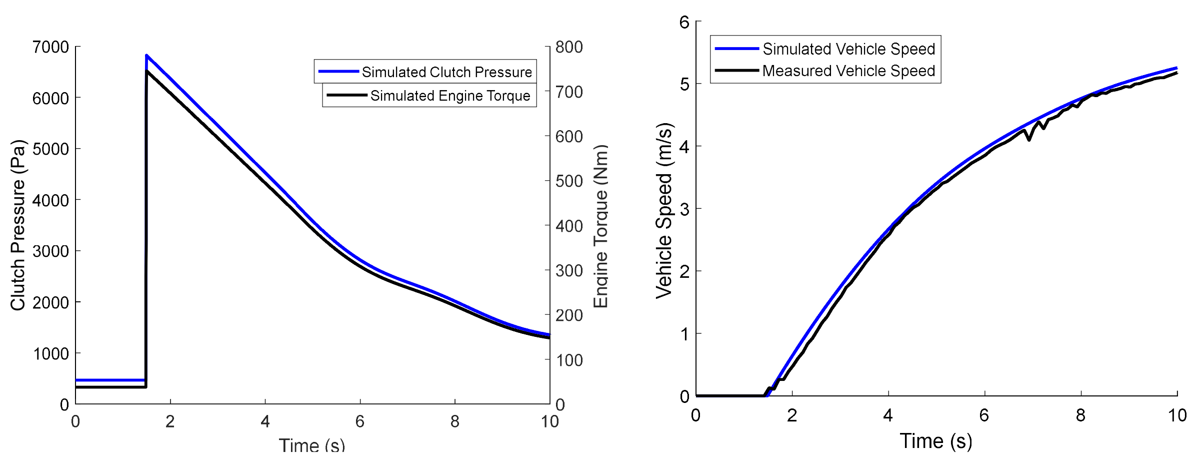

Figure 10 a&b: Basic Simulation Validation Against Recorded Vehicle Speed with Calculated Engine Torque and Clutch Pressure

3.2.2 Transmission Integration and Internal Clutch Controls

For the next stage of the model development, it was critical to include a detailed model of the transmission. Many EcoCAR vehicle models choose to neglect the internal transmission dynamics, instead using a Simscape block that allows you to specify the gear ratio at each timestep, because the transmission controller is provided by GM and the teams have minimal ability to impact its function. But understanding the separation between torque and inertia phases is critical here. The first transmission model was simple, just meant to test out the use of a planetary gear set that was configured for first gear only as shown in Figure 11.

Soon the model expanded to a full transmission model with clutch pressure inputs as shown in Figure 12 based on the strategies outlined in (Mishra & Srinivasan, 2015). It also saw the addition of the rear differential meaning all major physical components were in the model except the torque converter. Inside the transmission plant model is the Simscape equivalent of Figure 3 above with all planetary gears, clutches, and tooth ratios.

Rotational inertias were collected from GM directly for all the rotating part in and out of the transmission though exact values are not provided here. If those values had not been available, it is possible they could have been estimated through detailed coast down testing with the vehicle, or at worst measured directly after disassembly. The full design of the transmission plant is shown in Figure 13. The design has five inputs that are dedicated for controlling individual clutches inside the transmission.

For initial testing, only 1-2 up-shifts were considered. Following the clutch shift pattern in Table 2, the model shows that clutch A & B were locked since they remain closed in both gears and were defaulted to a pressure high enough that they should never have a chance to slip. Clutch E is given no pressure since it is not used to complete either gear. Last, clutch C & D are connected to input blocks so that they can be controlled through the phases of the upshift. Equations for calculating the model inputs are listed below.

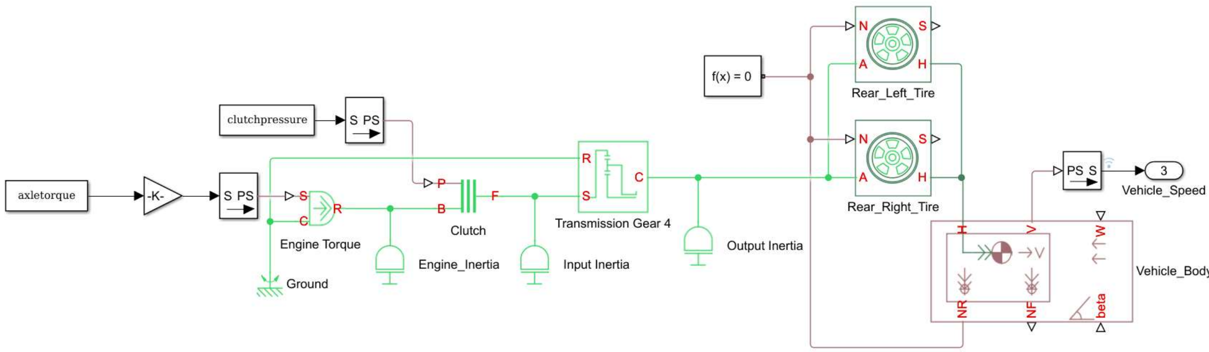

Figure 11: Intermediate Simscape Transmission Model A

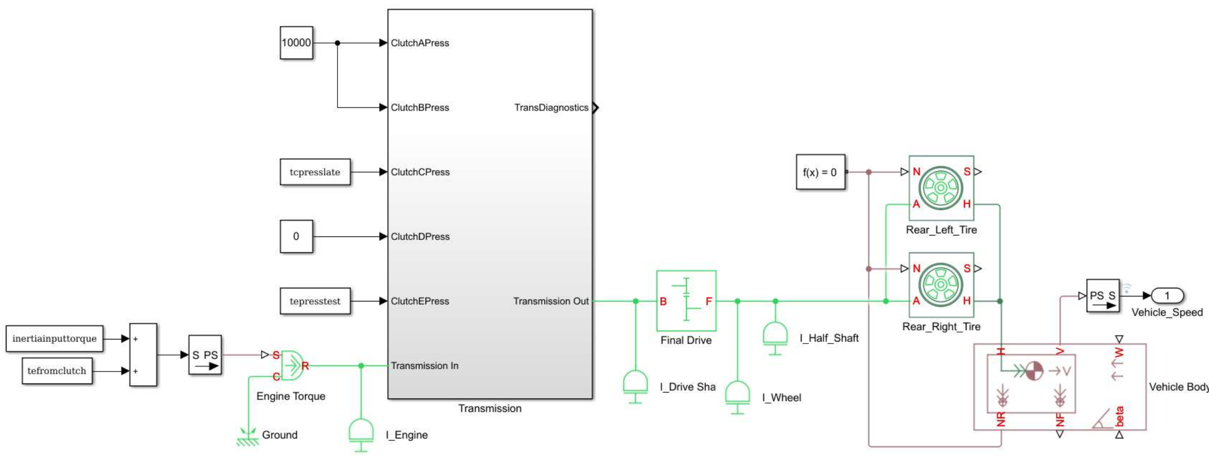

Figure 12:Intermediate Simscape Transmission Model B

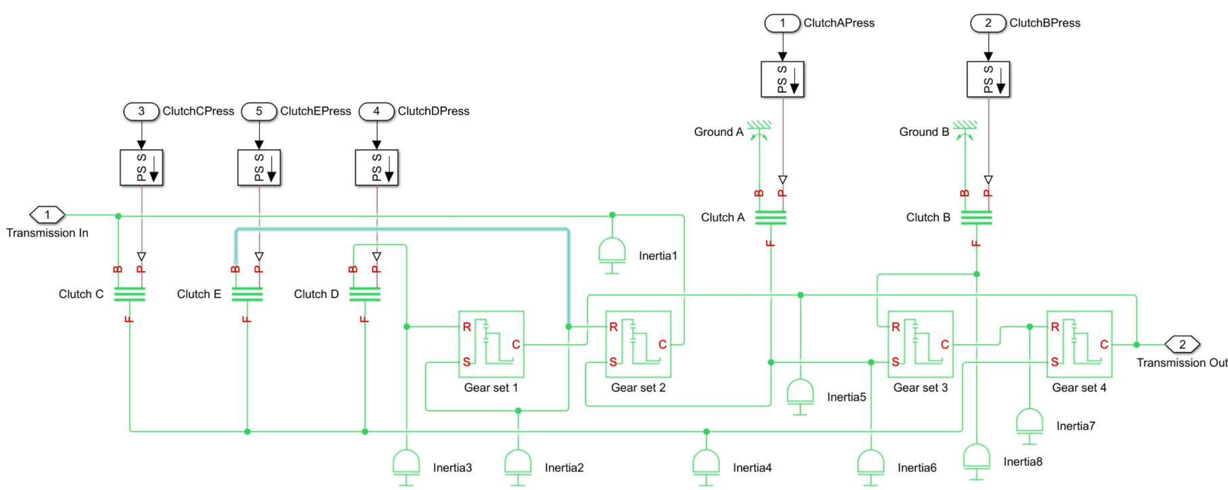

Figure 13: Inside the Transmission Simscape Plant Model

To drive this model, first the clutch pressures were calculated, taking into account the 4 different kinematic states through the shift sequence (locked in first, torque phase, inertia phase, locked in second). For clutch C, while locked in first gear it must carry all the torque required to drive the vehicle with a non-slipping clutch. During the torque phase it must ramp the pressure down steadily to 0 without allowing it to slip and remains at 0 for the rest of the shift event. Clutch E follows the opposite path, starting open, ramping up pressure to provide torque to drive the vehicle while slipping during the torque phase. In the inertia phase the clutch pressure is maintained as the slip speed drops, and once locked the clutch continues to drive the vehicle without slipping. In this model both pressure ramps for the clutches were calculated with 2 points at the beginning and end of the torque phase using linear interpolation between them.

In the equations below, the torque carried by each of the ramping clutches is calculated based on the changing pressure on the on-coming clutch, which shifts the load between the off-going (P_c) and the on-coming (P_e) clutch. In addition to the physical parameters of the clutch plates we have the addition of the overall first gear ratio (FGR), the final drive ratio of the rear differential (FD), and the road load torque (T_{RL}) defined below. The point of reference for determining the start and end time of the torque and inertia phases are based on when the corresponding torque request signals come from the transmission. Section 3.3.2 will go over the method for predicting these transitions in the software.

Equation 4: Clutch C Pressure During a 1-2 Shift

$$P_c = \left\{ \frac{T_{RL}}{FD * z * \mu_{static} * r_{eff} * A_{eff}} * \frac{1}{FGR}, 0 \leq t \leq torque\ phase\ start \right\}$$

$$\{0, inertia\ phase\ start \leq t \leq shift\ event\ end\}$$

Equation 5: Clutch E Pressure During a 1-2 Shift

$$P_e = \{0, 0 \leq t \leq torque\ phase\ start\}$$

$$\left\{\frac{T_{RL}}{FD * z * \mu_{kinetic} * r_{eff} * A_{eff}} * \frac{1}{FGR}, inertia\ phase\ start \leq t \leq inertia\ phase\ end\right\}$$

$$\left\{\frac{T_{RL}}{FD * z * \mu_{kinetic} * r_{eff} * A_{eff}} * \frac{1}{FGR}, inertia\ phase\ end \leq t \leq simulation\ end\right\}$$

In this case, clutch pressure equations are identical except for using the kinetic/static friction coefficients depending on the clutch slip state. This is because the relationship that changes the gear ratio from first to second in the transmission happens between these clutches and the transmission input. With the increased model complexity, the rotational inertia needs to be considered in more detail now that there are multiple rotating components with different angular accelerations based on the internal gears of the transmission. For calculating the road load (the torque needed to drive the vehicle meet the drive trace) both the physics of the vehicle body dynamics and all the inertial loads from the downstream components are used as shown in Equation 6. In this equation α_{out} represents the speed of the output of the transmission. Each rotational inertial (I) inside the transmission is grouped based on whether its acceleration is tied to the input or output transmission speeds as indicated in Equation 6 and Figure 14.

Equation 6: Road Load Calculation with Downstream Inertia in 1st and 2nd Gear

$$T_{RL} = T_{axl} + (I_{ds} + I_5) * \alpha_{out} * FD + \frac{(I_{wh} + I_{cv}) * \alpha_{out}}{FD} + I_4 * FGR * FD * \alpha_{out}$$

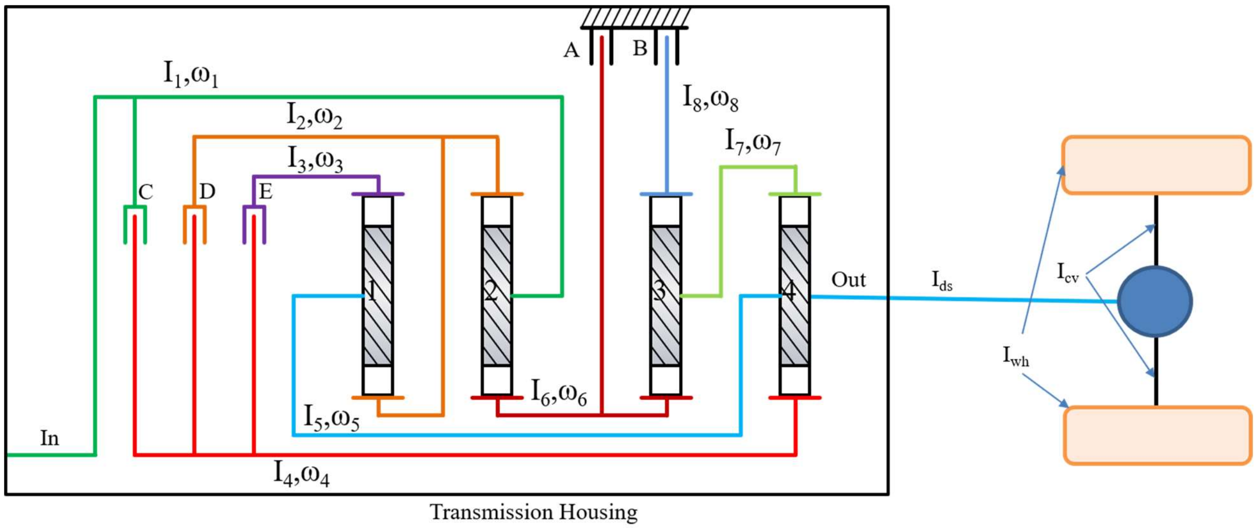

Figure 14: Transmission Stick Diagram with Downstream Rotating Components

The last step for operating this stage of the model is to take the torque load on the clutches and convert that into an engine torque and add in the effects of upstream inertia, meaning rotating elements that have their speed and acceleration fully determined by the engine speed through all phases of the shift as shown in Figure 15. The equations below start from the engine torque and work down towards terms already defined above to produce a complete system of equations that allow for calculating the engine torque needed to follow the drive trace. The end result is the engine crankshaft torque (T_{engine}) request which is based on the torque carried by the on-coming ($T_{e,load}$) and the off-going ($T_{c,load}$) clutches, plus the torque impacts of the upstream inertia acceleration ($T_{engine,inertia}$), and the "floating" inertia that requires input and output acceleration to calculate ($T_{3,inertia}$). Similar to previous equations, a linear torque ramp was assumed during the clutch swap so only the end points are defined, using Matlab's interpolation to fill the intermediate time steps.

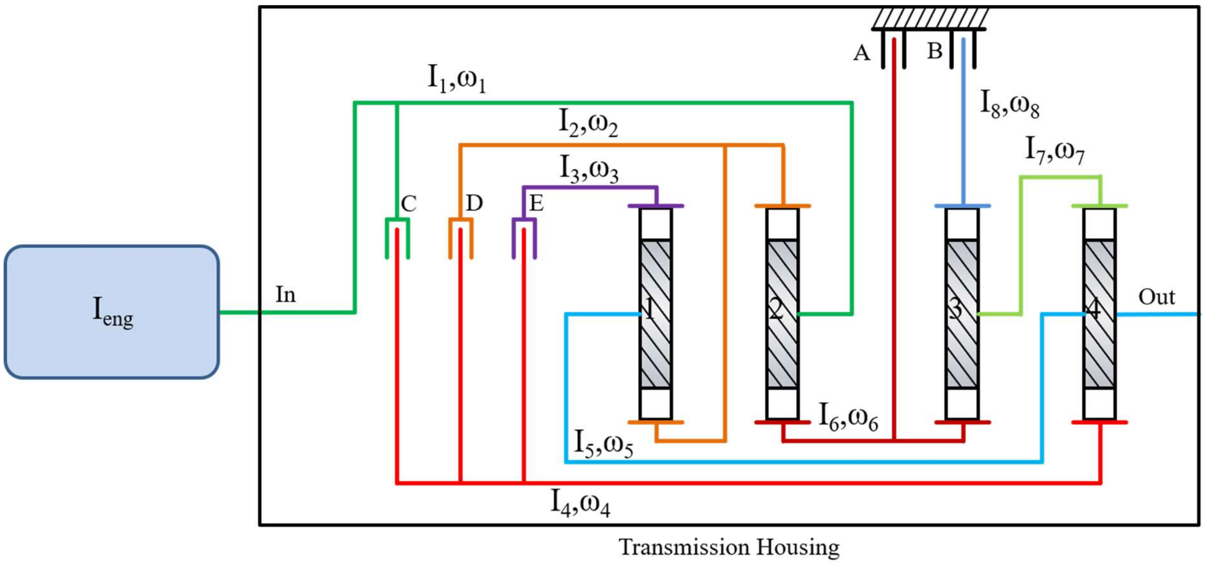

Figure 15: Transmission Stick Diagram with Upstream Rotating Components

Equation 7: Engine Torque From Clutch Load During a 1-2 Shift

$$T_{engine} = T_{c,load} + T_{e,load} * \frac{\mathrm{R2}}{(\mathrm{R2+S2})} + T_{engine,inertia} + T_{3,inertia}$$

Equation 8: Clutch C Torque Load During 1-2 Shift

$$T_{c,load} = \left\{\frac{T_{RL}}{FGR}, 0 \leq t \leq torque\ phase\ start\right\}$$

$$\{0, inertia\ phase\ start \leq t \leq shift\ event\ end\}$$

Equation 9: Clutch E Torque Load During 1-2 Shift

$$T_{e,load} = \{0, 0 \leq t \leq torque\ phase\ start\}$$

$$\left\{\frac{T_{RL}}{FGR}, inertia\ phase\ start \leq t \leq simulation\ end\right\}$$

For the upstream and "floating" inertia torques a new term α_{decel} is defined which is the predicted engine deceleration that occurs during the inertia phase. For this stage in development, this value was pre-calculated before the start of the simulation by simply calculating a linear deceleration rate that matches the engine speed before and after the inertia phase and the duration of the inertia phase. During the 1-2 shift the rotational mass that is not locked to either the input or output is only I_3 making the inertia torque for this single element almost negligible, but during the inertia phase it experiences a high acceleration to the point that the engine torque needed is significant. During the inertia phase there is no constraining formula that can solve for the inertia torque of I_3 from the acceleration of the planet and sun gears directly. As a substitute in $Q(t)$, which here will represent the acceleration ratio between the transmission input, and the output at every time t through the inertia phase.

Equation 10: Upstream Inertia Torque Compensation During a 1-2 Shift

$$T_{engine,inertia} = \left\{ \begin{matrix} \left(I_1 + I_{eng}\right) * \frac{(R4+S4)}{S4} * \alpha_{out} + I_2 * \frac{(R2+S2)^2}{S2} * \frac{(R4+S4)}{S4} * \alpha_{out}, \\ 0 \le t \le inertia\ phase\ start \end{matrix} \right\}$$

$$\left\{ \left(I_1 + I_{eng}\right) * \alpha_{decel} + I_2 * \frac{(R2+S2)^2}{S2} * \alpha_{decel}, inertia\ phase\ start \le t \le inertia\ phase\ end \right\}$$

$$\left\{ \begin{matrix} \left(I_1 + I_{eng}\right) * \frac{R2}{S4} * \alpha_{out} + I_2 * \frac{(R4+S4)}{S4} * \frac{(R2+S2)}{R2} * \alpha_{out}, \\ inertia\ phase\ end \le t \le simulation\ end \end{matrix} \right\}$$

Equation 11: Engine Inertia Torque Load from I3 During a 1-2 Shift

$$T_{3,inertia} = \left\{ \begin{matrix} \left[\left(\frac{R1+S1}{R1} * \frac{S4}{R4+S4}\right) - \left(\frac{S1}{R1} * \frac{R2+S2}{R2}\right)\right] * I_3 * \left[\frac{R1+S1}{R1} - \left(\frac{S1}{R1} * \frac{R2+S2}{R2} * \frac{S4+R4}{S4}\right)\right] * \alpha_{out}, \\ 0 \le t \le inertia\ phase\ start \end{matrix} \right\}$$

$$\left\{ \begin{matrix} \left[\left(\frac{R1+S1}{R1 * Q(t)}\right) - \left(\frac{S1}{R1} * \frac{R2+S2}{R2}\right)\right] * I_3 * \left[\frac{R1+S1}{R1} * \alpha_{out} - \left(\frac{S1}{R1} * \frac{R2+S2}{R2}\right) * \alpha_{in}\right], \\ inertia\ phase\ start \le t \le inertia\ phase\ end \end{matrix} \right\}$$

$$\left\{ \begin{matrix} \left[\left(\frac{R1+S1}{R1}\right) - \left(\frac{S1}{R1} * \frac{R4+S4}{S4}\right)\right] * \left[\frac{R2+S2}{R2}\right] * I_3 * \left[\frac{R1+S1}{R1} - \left(\frac{S1}{R1} * \frac{S4+R4}{S4}\right)\right] * \alpha_{out}, \\ inertia\ phase\ end \le t \le simulation\ end \end{matrix} \right\}$$

With all these enhancements to the model, it is now possible to generate much more realistic engine torque and clutch pressure signals. These new signals, shown in Figure 16 a & b, capture most of the dynamics of the shift event while still following the original drive trace.

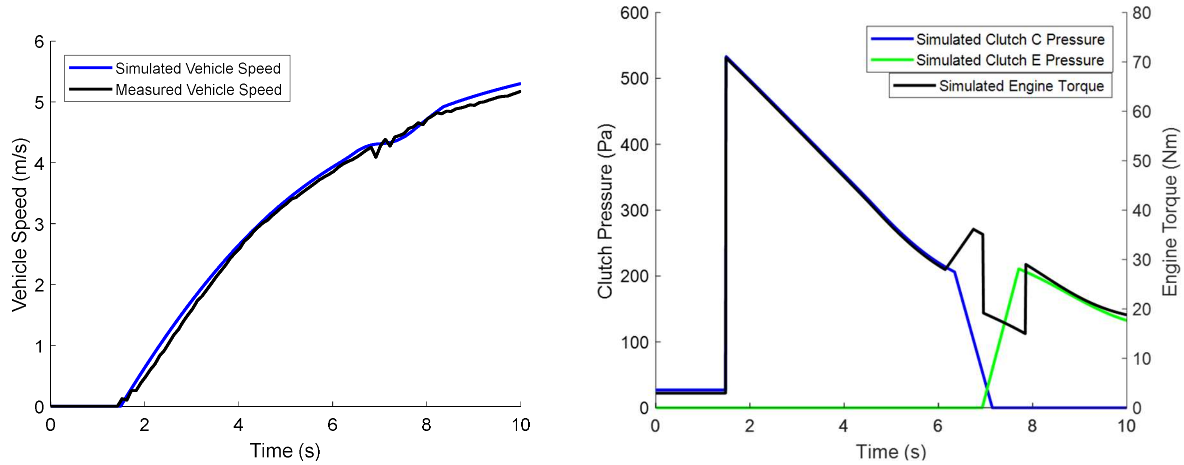

Figure 16 a&b: Intermediate Simulation Validation against Recorded Vehicle Speed with Calculated Clutch Pressures and Engine Torques

3.2.3 Torque Converter Integration and Modeling

To make the conventional vehicle model complete for the purposes of this research, the last powertrain component to add is the torque converter between the transmission input and the engine crankshaft as shown in Figure 17. Parameters for the K-factor and torque ratio as a function of the speed ratio were provided by GM and put into a custom torque converter block that supported a torque converter clutch with a binary on/off control that was tied to the data recorded in the drive trace.

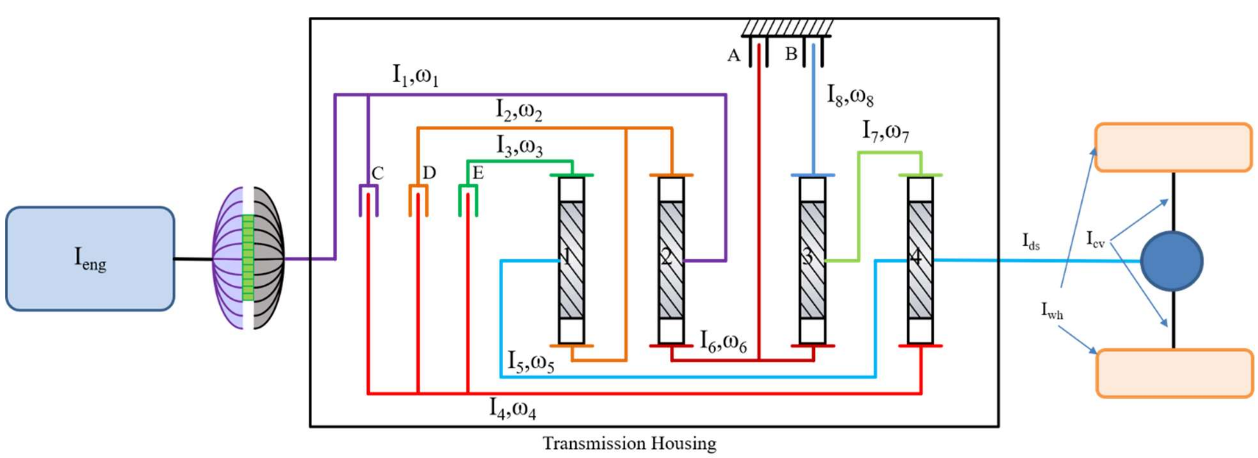

Figure 17: Full Conventional Powertrain Model

Another change to the model that is needed is to add in friction brakes. To simulate the torque converter effects properly, it is critical that the engine be allowed to idle before launch so that all the powertrain components are at the right speed. With the torque converter, this also means that significant torque will be put on the wheels just as it would in an automatic transmission vehicle if you release the brake at a stop. To correct this rolling forward so we can better follow the vehicle trace, the brakes are used to lock the wheels until just before the accelerator pedal is pressed.

As mentioned in the background, when the TCC is open the velocity between the input and output will be different, meaning there will be different accelerations for calculating inertia torque in addition to a torque multiplication that comes from the fluid action. To control the expanded model Equation 7 can be reused with some alterations to account for the torque converter with an open clutch. In these equations $T_{trans-in,inertia}$ represents the inertia torque for the portions of the transmission upstream of the swapping clutches plus the turbine of the torque converter, but not the pump side, or the engine inertia, those are covered by $T_{engine,inertia}$ which uses I_{eng} for the combined inertia of that shaft.

Equation 12: Engine Torque from Clutch Load During a 1-2 Shift for the Full Model

$$T_{engine} = \frac{T_{c,load} + T_{e,load} * \frac{\mathrm{R2}}{(\mathrm{R2+S2})} + T_{trans-in,inertia} + T_{3,inertia}}{R_{TC}} + T_{engine,inertia}$$

Equation 13: Upstream Inertia Torque Compensation During a 1-2 Shift for the Full Model

$$T_{trans-in,inertia} = \begin{Bmatrix} I_1 * \frac{(R4+S4)}{S4} * \alpha_{out} + I_2 * \frac{(R2+S2)}{S2}^2 * \frac{(R4+S4)}{S4} * \alpha_{out}, \\ 0 \leq t \leq inertia\ phase\ start \end{Bmatrix}$$

$$\left\{ I_1 * \alpha_{decel} + I_2 * \frac{(R2+S2)}{S2}^2 * \alpha_{decel}, inertia\ phase\ start \leq t \leq inertia\ phase\ end \right\}$$

$$\begin{Bmatrix} I_1 * \frac{R2}{S4} * \alpha_{out} + I_2 * \frac{(R4+S4)}{S4} * \frac{(R2+S2)}{R2} * \alpha_{out}, \\ inertia\ phase\ end \leq t \leq simulation\ end \end{Bmatrix}$$

Equation 14: Engine Inertia Torque During a 1-2 Shift for the Full Model

$$T_{engine,inertia} = \alpha_{eng} * I_{eng}$$

With these equations within the vehicle model, all the necessary powertrain dynamics have been captured that are needed to answer the research questions. Furthermore, the vehicle model has been shown that it can replicate the real-world test results accurately. With this in place, the modifications can be made to demonstrate a controls solution that can meet the goals for the hybrid architecture.

3.3 Control Strategy Development

With a functional model covering all the stock vehicle powertrain components it is almost ready to be applied to the hybrid powertrain. Key to this strategy will be to have a system that can predict how a shift will play out when the decision to shift gears is made by the transmission, and then controlling torque production in a system with variable rotational inertia. These last sections will cover the modifications to support real-time vehicle control, conversion to a hybrid powertrain, and the outlines of a basic system for trying to improve vehicle efficiency during shifts.

3.3.1 Model Prep for the Control Strategy

With the model development described above there is one major fallback that has been useful for model development but is a barrier for ever implementing a controls strategy into a real vehicle. The model, as developed, runs fully off the data recorded from the stock vehicle, and all command signals are calculated by a script before the model runs. For instance, during the inertia phase when the engine decelerates to the new synchronization speed by using the pre-recorded data it was possible to know exactly how much the vehicle would accelerate, and what exact speed the engine would be at when the on-coming clutch locks. With that knowledge it was possible to calculate an extremely accurate inertia torque requests. Similar conveniences exist elsewhere in the model and need to be replaced with predictions or estimations based on data available in the real time operation, or on the recent past.

In the scope of this project, the only actuators that will be controlled directly by logic are the electric motor and engine torque commands. Other control parameters in the model such as torque converter clutch state, transmission clutch pressure, and gear selection among others will be left to the control of other embedded controllers on the vehicle or are directly controlled by the driver.

To start building the control strategy, the first step was to take

Equation 13 and all the steps leading up to it and shift the work from a Matlab script into Simulink blocks that are tied to the model parameters (vehicle speed, engine speed, etc.). Starting with the road load and working back to the engine crankshaft torque.

3.3.2 Shift Trigger and Predictions

To enable torque estimations during the shift in real-time, the control strategy needs to predict the length of the torque and inertia phase to have an idea of when to expect the phases to change and how fast the engine will need to deceleration while in the inertia phase.

The simplest way to do this was to the collected stock vehicle data to see if there is a way to correlate the length of the shift phases to the vehicle state when the decision to shift is made. Various curve fitting options were tried using accelerator pedal and vehicle speed as inputs, but accelerator pedal by itself seemed to predict the behavior quite well on its own for the 1-2 shifts.

Figure 18 shows the results from the curve fit toolbox in Matlab. Equation 15 shows the curve fit formula to apply the coefficients shown in

Table 3 to predict the shift phase length when using x as the accelerator pedal position calculated as a percentage of total travel.

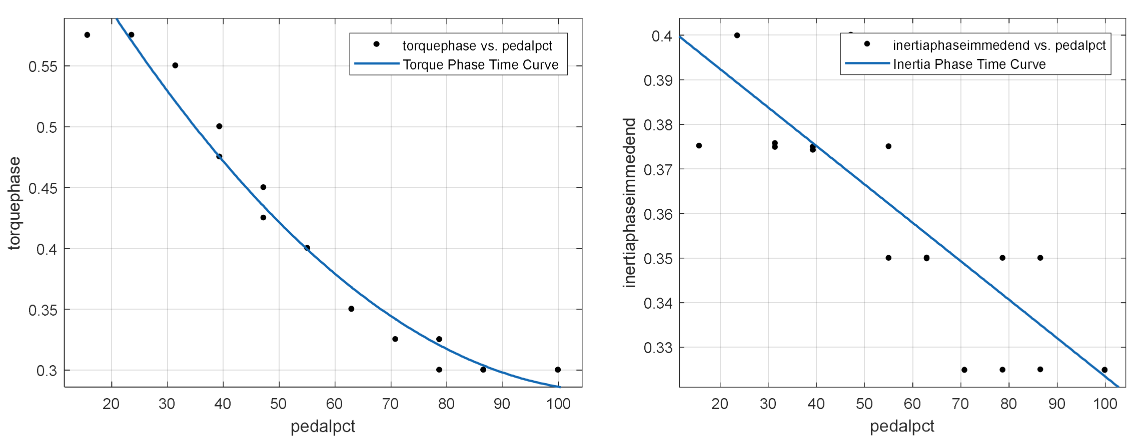

Figure 18 a&b: Torque & Inertia Shift Phase Length Curve Fit Results and Raw Data

Equation 15: Curve Fit Formula for Shift Phase Length Predictions

$$T(x) = P_1 * x^2 + P_2 * x + P_3$$

Table 3: Shift Phase Length Prediction Parameters

Parameter/Result	Torque Phase	Inertia Phase
P1	3.84e-5	0
P2	-0.0085	-0.00086
P3	0.75	0.41
SSE	0.0070	0.0042
R-squared	0.96	0.67

The last thing to determine is the synchronization speed for the end of the inertia phase. Using Equation 16, the current vehicle speed and acceleration at the start of the shift is used to predict what engine speed is needed for the on-coming clutch to bring slip to zero by the end of the predicted inertia phase. Knowing the engine speed at the start of the inertia phase, the predicted target speed at the end of the shift, and the predicted inertia phase length; the engine deceleration rate can be determined, and the inertia phase torque can be determined in real-time.

Equation 16: Inertia phase engine deceleration during a 1-2 shift

$$\alpha_{decel} = \frac{(a * T_{int}) + v_{init}}{2 * \pi * r_{wheel}} * FD * SGR * TCR$$

3.3.3 Final Stock Vehicle Simulation Results

After adding some controls to limit the max rate of change of the engine torque signal the finished real-time model is functional. The final model is able to take in a physics-based torque request, compensate for the inertia state of the powertrain and determine a crankshaft torque that will meet the drive trace as shown in the figures below.

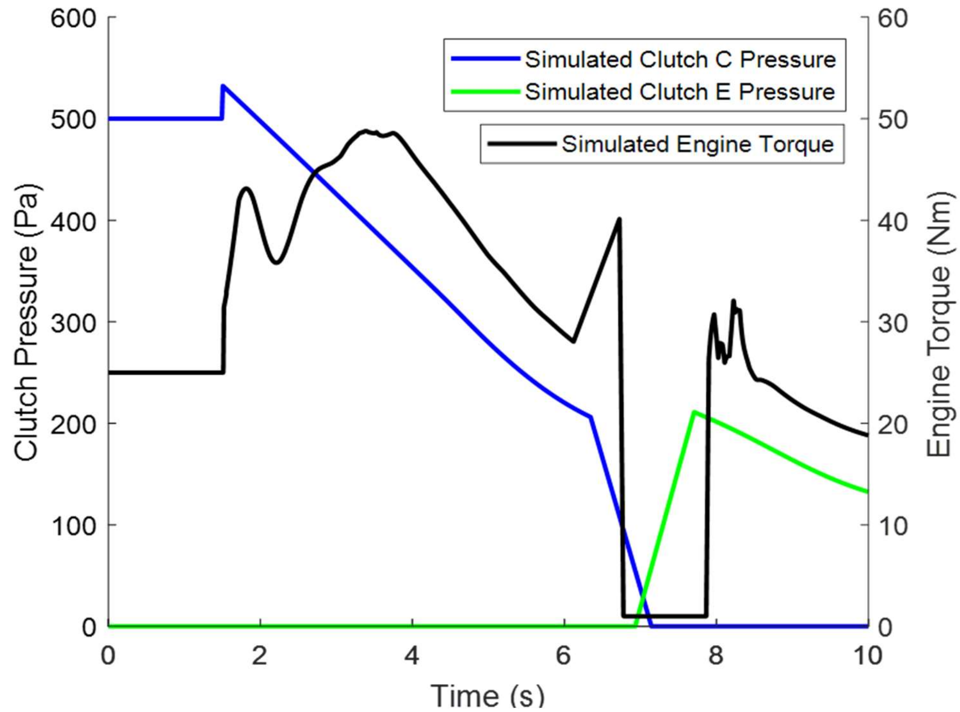

Figure 19: Engine and Clutch Pressure Commands for the Full Conventional Model

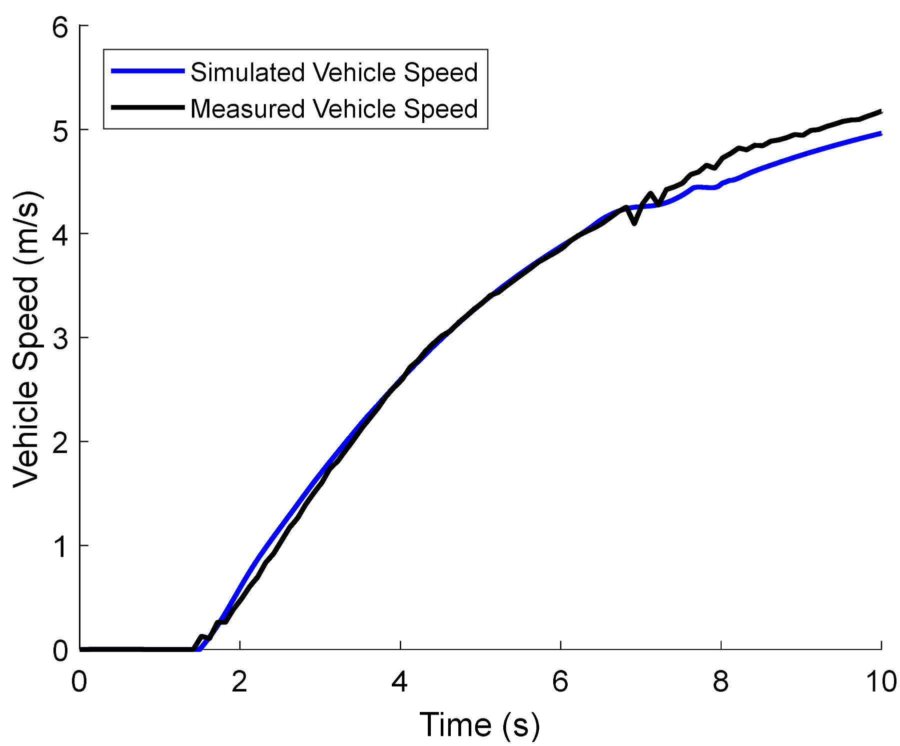

Figure 20: Comparing the Recorded and Simulated Vehicle Speed for the Full Conventional Model

3.3.4 Modifications for the Hybrid Setup

Up to this point the model has been of the Camaro as it was delivered to the university before it was converted to a hybrid. With a model that can both simulate transmission dynamics and run in real-time there is a great foundation to convert the model to the hybrid powertrain and make a strategy that can take advantage of the electric motor torque response.

As shown in Figure 1, the electric motor is placed between the engine and the torque converter of the transmission meaning that the inertia from the electric motor gets added to the engine and torque converter pump inertia. There is also a clutch between the engine and the electric motor that allows the vehicle to operate as a purely electric vehicle meaning there are 2 states to account for in the model depending on the clutch state. For this model, a clutch and an additional torque source was added, the engine to motor clutch was not given any control strategy. Mid drive cycle changes in or out of EV mode were not considered.

3.3.5 Spark Cut Negation

This research does not seek to integrate transmission control in a vehicle with an overall hybrid operating strategy on its own. There are always multiple considerations to be made, when developing an electrified vehicle, on how to use the electric battery charge. This becomes more challenging with a muscle car since there is always going to be a strong bias to see how the electric motor can be used to improve vehicle performance that will do little to improve fuel economy. This thesis does not seek to try to manage this larger task. Instead, the objective is to focus on drive quality improvements that come from correcting inaccuracies in the transmission torque requests and secondarily the efficiency gains that can be found through the specific application of electric motor torque during transmission shifts. During the shift, the control strategy should attempt to maximize the mechanical effort extracted from any fuel that goes to the engine during that time window.

In a conventional powertrain, to execute the inertia phase of the shift the engine must adjust spark ignition timing to reduce engine torque faster than the air throttle can respond. This comes at the cost of vehicle efficiency since the fuel being burned is not generating the full mechanical effort that is possible, it is effectively wasting some of the fuel. With an electric motor on the same shaft as the engine that can adjust torque much faster than the throttle, it is possible with some planning to use the electric motor to provide the torque reduction instead.

The only real challenge here is to try to make sure that whenever possible, the electric motor is making enough torque to be able to meet the transmission's needed torque reduction without exceeding the hardware limits. An additional consideration is to try to reduce or eliminate the need for the electric motor to cross the zero-torque line. If this is not done, the electric motor will cause part of the powertrain to go through backlash, which can cause bumps in torque or audible rattles that can be distracting to the driver when the gear teeth change engagement on the electric motor.

To create a system to control the electric motor during these shifts, existing logic from the vehicle model was repurposed which include the following pieces described above:

- Inertia phase length prediction
- Engine synchronization speed prediction
- Turbine inertia torque calculations

Normally, the first 2 of these functions only executes in the one controller cycle when the desire to shift was indicated by the transmission. The purpose of this single execution is to take a snapshot of the current vehicle conditions and predict based on that how the shift will execute.

For the motor controls, this same logic is run continuously so that in every loop of the software an estimate is calculated as if the transmission had requested a shift. These estimates include the following with respect to a hypothetical up-shift request:

- What speed the engine is likely to be at the end of the torque phase
- What speed the vehicle will be at when the inertia phase ends
- How long it would take to go through the inertia phase from the peak engine speed

With all these factors you can calculate the torque reduction that is needed. Once calculated, this resultant torque was requested of the electric motor directly when possible, but there were a few considerations that were made so that the motor torque and engine torque would mesh well and smoothly.

3.3.5.1 Operating outside the shift (during acceleration):

While accelerating, the vehicle conditions can change rapidly in a way that significantly impacts the magnitude of a torque reduction that would be needed during the inertia phase. This can result in an increase in the torque cut prediction in a short amount time. If the motor were to immediately provide that torque it could cause the engine torque to decrease torque production to continue to meet the driver's request even as the driver's total torque request is increasing. To balance the torque sources, a rate limit was applied on the motor torque commands to prevent rapid changes since it would not be an effective use of the battery's electrical charge. This can result in cases where, once the transmission does shift, the motor torque is not large enough to stay positive while taking on the full torque reduction during the inertia phase and must provide negative torque.

3.3.5.2 During the Torque Phase:

Many of the conditions from before the shift is selected still apply during the torque phase, but a rate limit is not guaranteed to keep the engine torque stable during the torque phase, instead a check is made to only allow engine torque to maintain its current value or increase while the motor makes sure the driver's demand is still met. The only exception to this would be if the motor is not capable of meeting the driver's request under those conditions and the engine is forced to reduce torque. None of the recorded stock vehicle data demonstrates an example of this happening so it was not addressed.

3.3.5.3 During the Inertia Phase:

In the inertia phase, the main goal is to try to keep the engine torque at a similar value, and if it must change, to make the change with the throttle valve and not with spark timing. Any transmission messages to the engine should be interrupted and the engine torque should only change if the motor is not able to meet the driver's expectation and the transmission's needs. During the inertia phase, the transmission input speed needs to drop rapidly while the slipping on-coming clutch gets closer to the synchronization speed. To limit wear on the clutches and maintain transmission operating life, it is important that the electric motor reduce the net torque to the transmission so that the target speed is reached within the predicted inertia phase time window.

While backlash controls are not present in the logic it is something that would have to be assessed during in vehicle testing to see if there is a major disturbance to the driving experience when or if the motor switches from positive to negative torque. There will always be a major reduction in torque needed the moment the inertia phase starts since to synchronize the engine to the new gear; a lot of inertia kinetic energy needs to be removed from the powertrain upstream of the slipping clutch. It may not be practical in many cases to even attempt to cover this torque cut without the electric motor torque crossing the zero-line which will make backlash something important to watch.

3.4 Simulation Error Quantification

With the model and control strategy developed, the last thing needed is a method to compare the results of the simulations that will be run against the data that was recorded in the stock vehicle. For this purpose, the Mean Absolute Error (MAE) method was selected. MAE provides results that can simplify a full simulation into a single value that can easily identify outliers in the data and clearly indicate the magnitude of any error between the simulation and real-world data. Both data sets were sampled at 1 millisecond intervals throughout the duration of the simulation and the MAE was calculated as shown below where n is the total number of samples, p_i is a sampled value from the simulation results, and r_i is the corresponding sampled value from the real-world data. The results of this calculation is the average difference between the real world data and the simulation results in m/s.

Equation 17: Mean Absolute Error Formula

$$MAE = \frac{\sum_{i=1}^{n} |p_i - r_i|}{n}$$

4.0 RESULTS

To show that the proposed strategy will work, this section documents the simulation results that prove the original research questions. There are 3 stages to these results. First, to prove that the MATLAB model is accurately representing the vehicle dynamics. It was configured for the stock conventional vehicle to show that the model reasonably follows the real-world data. Second, show that the control strategy makes the changed powertrain “transparent” to the vehicle performance by running the simulation against the same vehicle data. To be complete, the results are shown with the vehicle configured into the final powertrain design, but setup to run as both a hybrid with the engine-to-motor clutch closed, and in EV mode with the clutch open. Last, execute the torque control strategy to show how in the loop-to-loop operation of the controller, it is possible to achieve the objective of eliminating the need to adjust engine ignition timing during the inertia phase.

This same process was run against all useable real-world data that was available. To aggregate the results from each sample the MAE is shown for the delta between the vehicle speed recorded on the stock vehicle vs. the simulated vehicle speed. This makes it clear if the model is reasonably fitting the stock data and shows the degree of change when switching from the stock vehicle simulation to the final powertrain. To aid with the analysis, all simulations against the same stock vehicle data are shown together to see each step.

While only shifts from first to second gear were simulated, further shifts occur with the same process, but with a different pair of clutches changing engagement and a different configuration of the rotating elements sorted into 3 groups:

- Elements whose acceleration is tied only to the transmission input rotational velocity.
- Elements who acceleration is tied only to the transmission output rotational velocity.
- Elements that are driven both by the input and output.

With this framework the same strategy for shifts from first to second gear could easily be extended to the others.

4.1 Sampling Low, Medium, and High Accelerator Pedal Data

To review some results in more detail, 3 data sets were selected representing low, medium, and high accelerator pedal positions to try to cover the range of expected vehicle response. Below each is reviewed in full including the stock vehicle simulation and the modified powertrain in hybrid and EV mode with a look at how the torque split strategy functioned in the hybrid mode simulation. Each of these demonstrate the model and the torque control strategy working as intended through a shift from first to second gear.

4.1.1 - 17% Accelerator Pedal Sample A

In this first data sample, the brake pedal was not released until 4 seconds into the data so the analysis and plots will be limited between 4 and 9 seconds into the simulation to cover the relevant time window. Looking at the vehicle speed the model exceeds the simulated results, as shown in Figure 21, but the MAE is <0.3 m/s.

The transmission pressures in Figure 22 show the separation of the shift phases clearly as the handoff from clutch C to clutch D occurs. During the handoff, we see engine torque increase rapidly as it compensates for the torque loss that comes from the competing clutches. Once clutch D is carrying all the torque, while still slipping, the inertia phase torque reduction occurs. As the engine speed dropped, as shown in Figure 23, during the inertia phase it eventually reached the new synchronized speed causing clutch D to lock and the pressure on that clutch then drops to a value appropriate to the static friction coefficient.

There is a noticeable drop in engine speed that comes with a small surge in transmission input speed that occurs at about 8.3 seconds that can be seen in Figure 23, this occurs because the speed differential between the 2 sides of the torque converter drops low enough that the calibrations on the transmission controller determined it was time to lock the torque converter clutch to improve power transmission efficiency meaning the engine and transmission input speed had to synchronize.

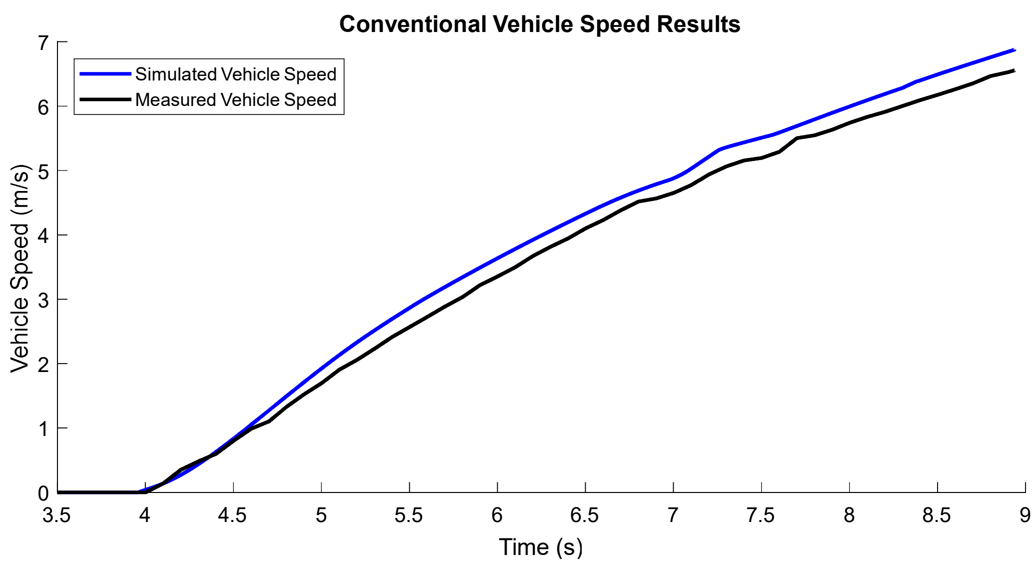

Figure 21: 17% Accelerator Pedal Vehicle Speed Simulation Results Against the Recorded Data for the Stock Vehicle Configuration

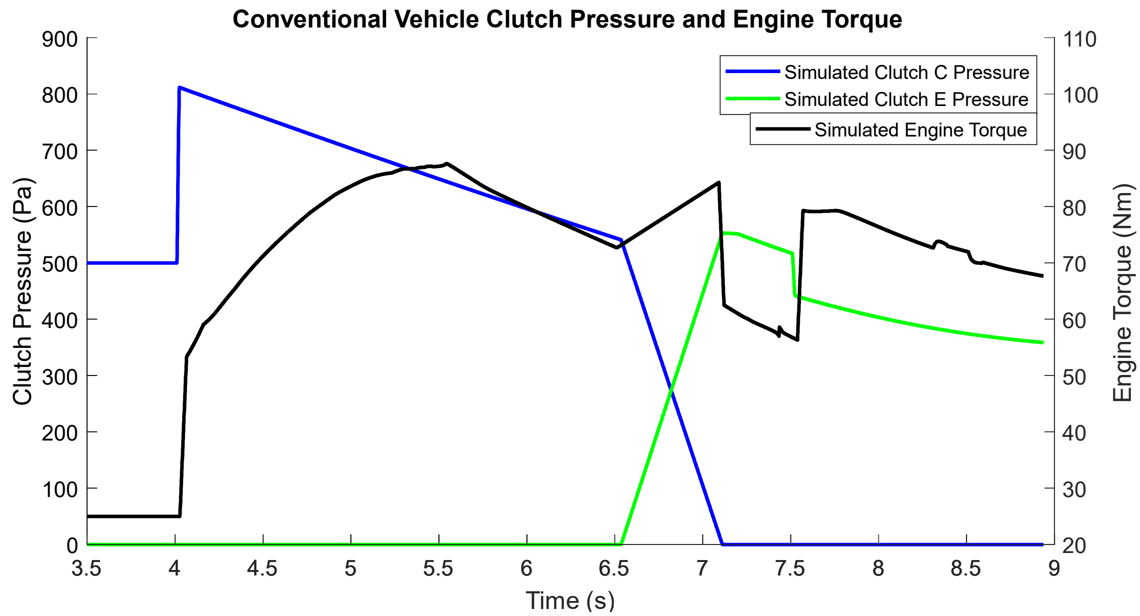

Figure 22: 17% Accelerator Pedal Clutch Pressures and Engine Torque Simulation Results for the Stock Vehicle Configuration

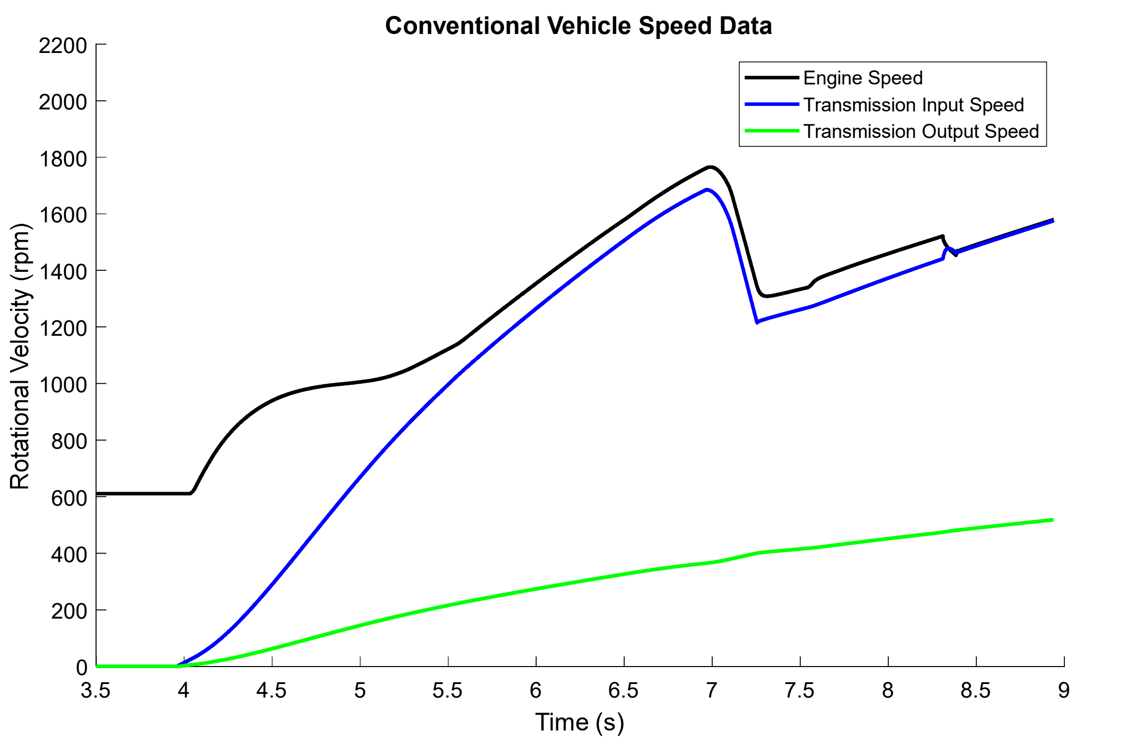

Figure 23: 17% Accelerator Pedal Engine and Transmission Speed Simulation Results for the Stock Vehicle Configuration

Once switching to the hybrid powertrain, the first test is to run the simulation with the clutch between the engine and electric motor closed and try to maintain the shift quality while using the motor to provide the torque change in the inertia phase. In Figure 24, we can see that the model is still following the recorded data with similar accuracy with an MAE of 0.24 m/s. The clutch pressures, axle torque, and speeds also all show similar behavior as shown in Figure 25 and Figure 26. This means that the torque changes have compensated for the change in rotational inertia successfully as a significant error would have cause an unexpected surge or decrease in the speed of the vehicle, or engine and electric motor.

When looking at the torque split in Figure 27 the electric motor hovers around 0 Nm until the vehicle begins to approach the start of the torque phase and the predictive logic starts to calculate a meaningful amount of torque needed for a potential shift. The motor torque starts to increase without reaching a point where the motor dominates the total powertrain torque production. It continues to ramp up the torque until the inertia phase is triggered and it drops. The motor charges the battery briefly as the electric motor and engine slow down to the new synchronized speed. Once synchronization occurs the motor torque returns to hovering at 0 Nm and letting the engine resume as the primary torque provider while waiting for the next up-shift.

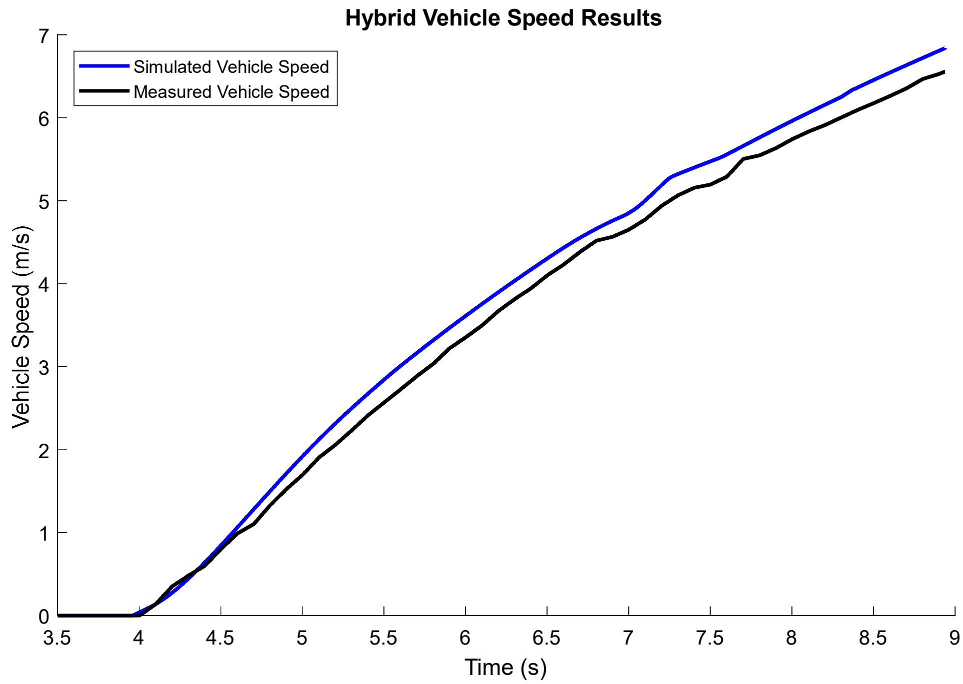

Figure 24: 17% Accelerator Pedal Vehicle Speed Simulation Results for the Hybrid Configuration Against Real World Data

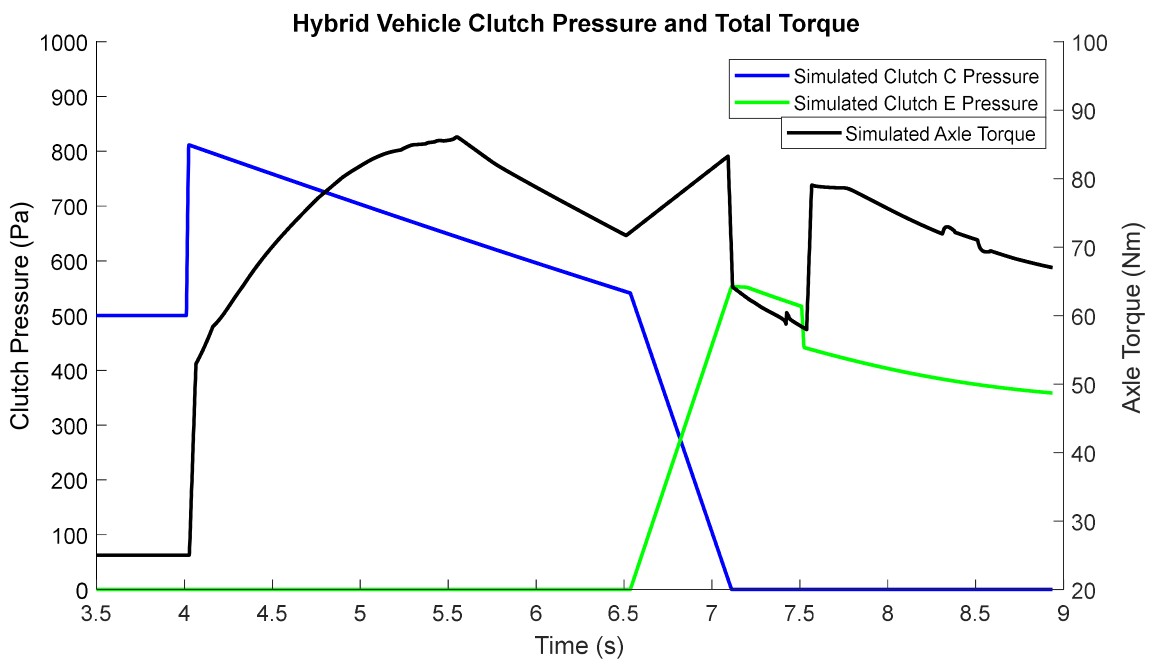

Figure 25: 17% Accelerator Pedal Clutch Pressures and Axle Torque Simulations Results for the Hybrid Configuration

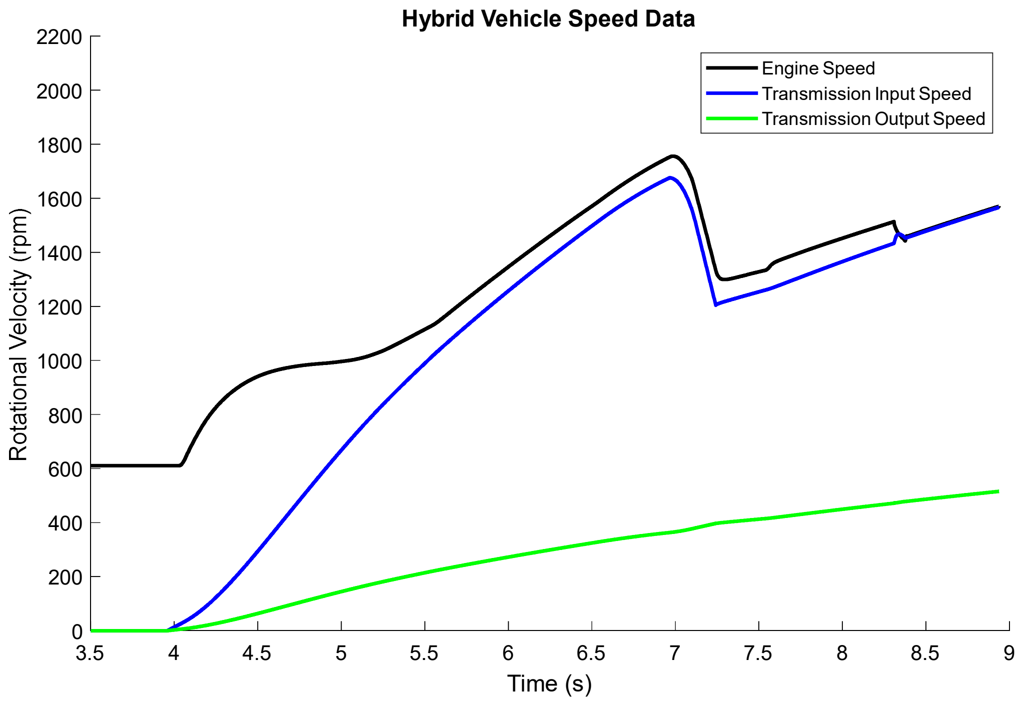

Figure 26: 17% Accelerator Pedal Engine and Transmission Speed Simulation Results for the Hybrid Configuration

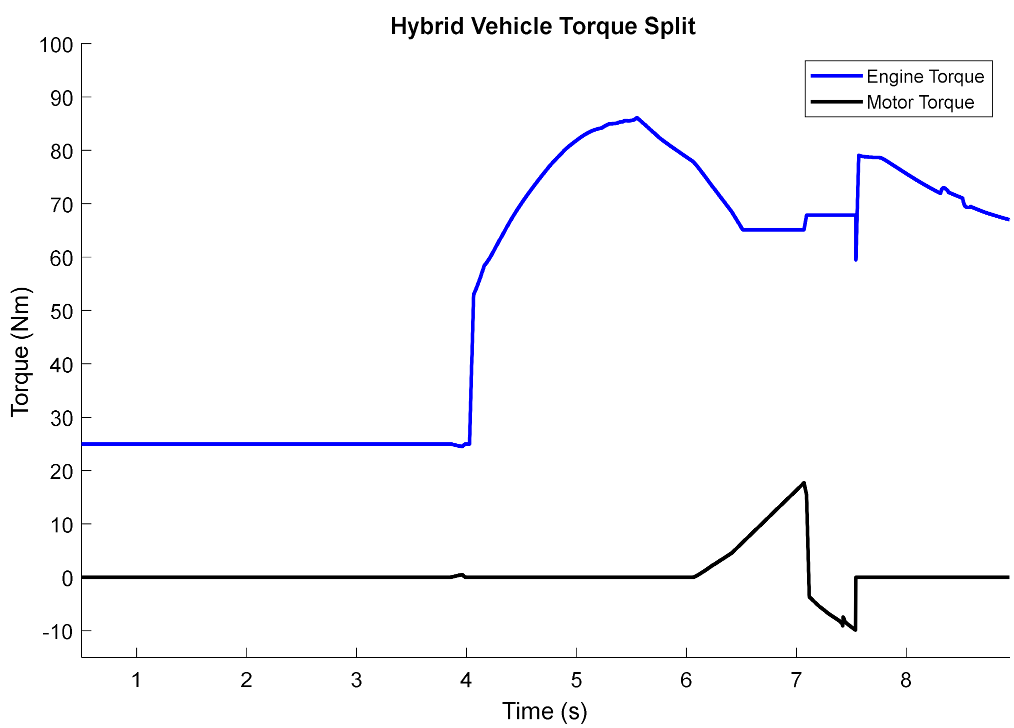

Figure 27: 17% Accelerator Pedal Engine and Electric Motor Torque Simulation Results for the Hybrid Configuration

The last check is to open the clutch between the engine and electric motor and re-run the simulation in EV Mode. With only the electric motor connected to the transmission input the rotational inertia is much lower causing changes in the results both here and in most other data sets. The vehicle speed follows much closer to the trace since the powertrain inertia has a much-reduced impact on vehicle dynamics and the needed torque is more strongly connected to only the vehicle road loads.

This also means that the torque provided by the motor is sensitive to driver pedal position changes and vehicle conditions which can cause it to swing up and down rapidly. Last, with the electric motor connected to the transmission input, often referred to as a P2 hybrid configuration, it still must interact with the transmission during the torque and inertia shift phases and the torque changes can still be seen in Figure 29.

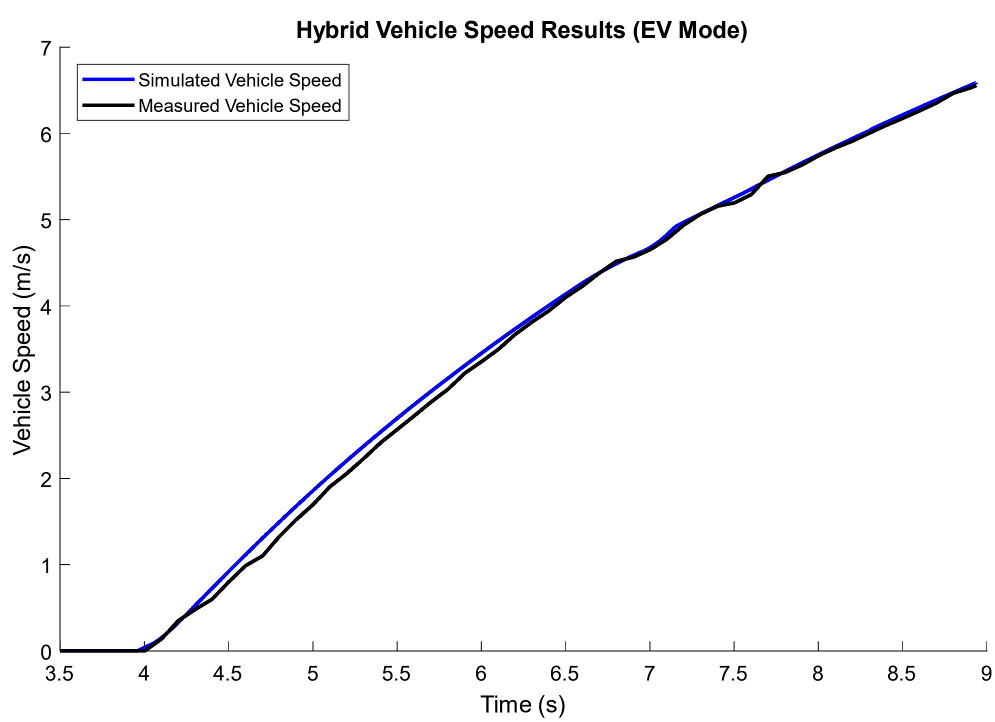

Figure 28: 17% Accelerator Pedal Mode Vehicle Speed Simulations Results in EV Mode Against Recorded Data

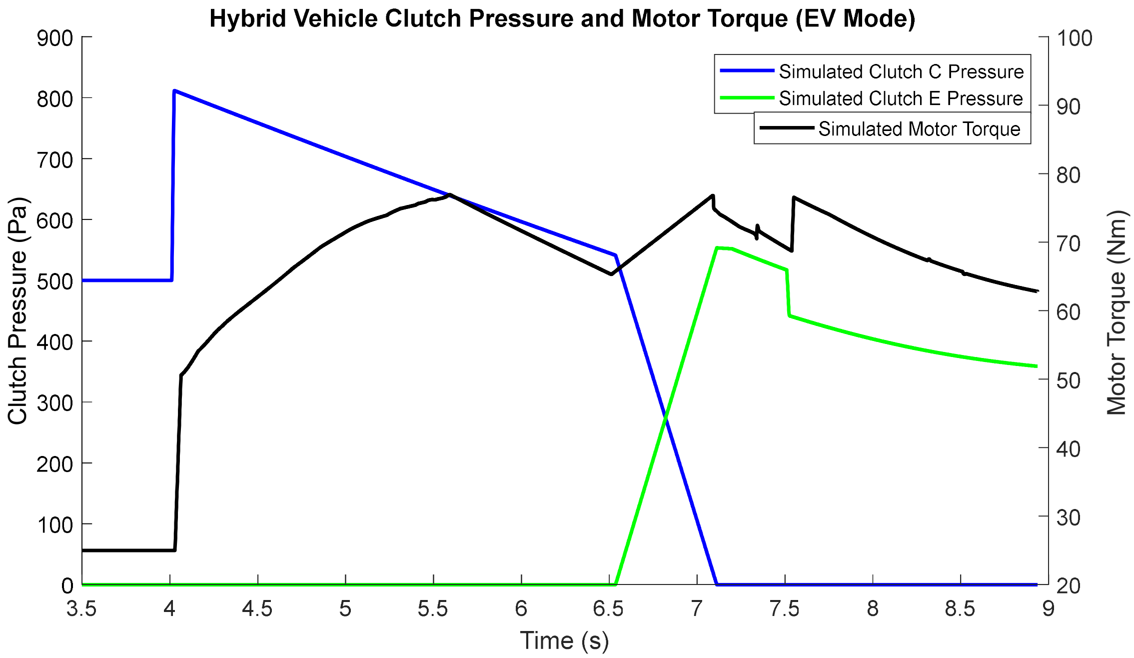

Figure 29: 17% Accelerator Pedal Clutch Pressures and Motor Torque Simulation Results in EV Mode

4.1.2 - 42% Accelerator Pedal Sample A

In this second sample the relevant time window is a bit shorter as the higher vehicle acceleration means the shift to second gear occurs sooner. The error in this sample between the recorded data and simulated vehicle speed (Figure 30, Figure 33, and Figure 37) is also much smaller than in the previous sample. All the figures below show similar outcomes to the 17% pedal results, the biggest difference being that many times during the shift the sequence the transmission opens and closes the torque converter clutch, making the engine and transmission input speeds shift in and out of synchronization (See Figure 32 and Figure 35). Otherwise the clutch pressures and axle torques fall in line with expectation (See Figure 31, Figure 34, and Figure 38), and the hybrid torque split between engine and motor runs as planned as well (Figure 36).

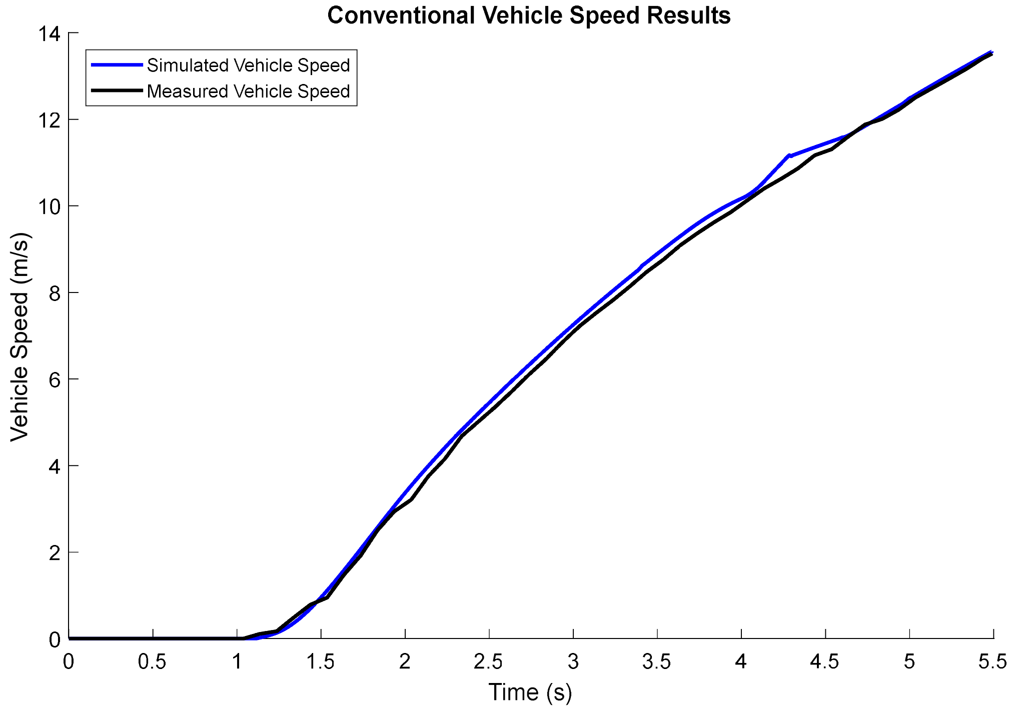

Figure 30: 42% Accelerator Pedal Vehicle Speed Simulation Results Against Recorded Data for the Stock Configuration

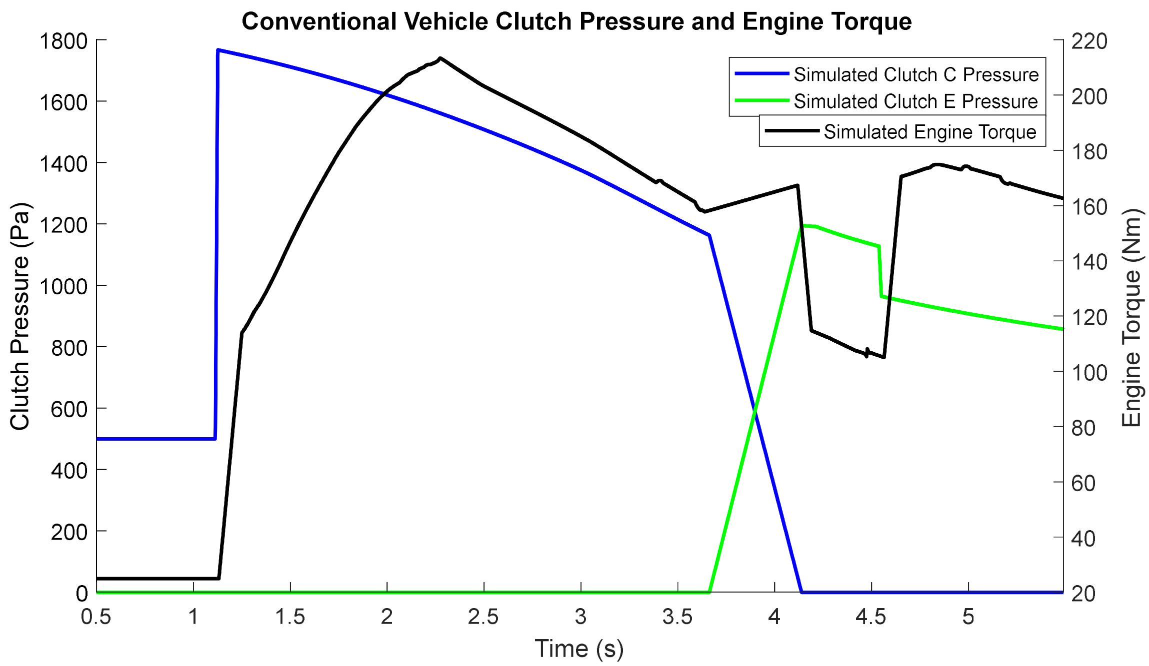

Figure 31: 42% Accelerator Pedal Clutch Pressures and Engine Torque Simulation Results for the Stock Configuration

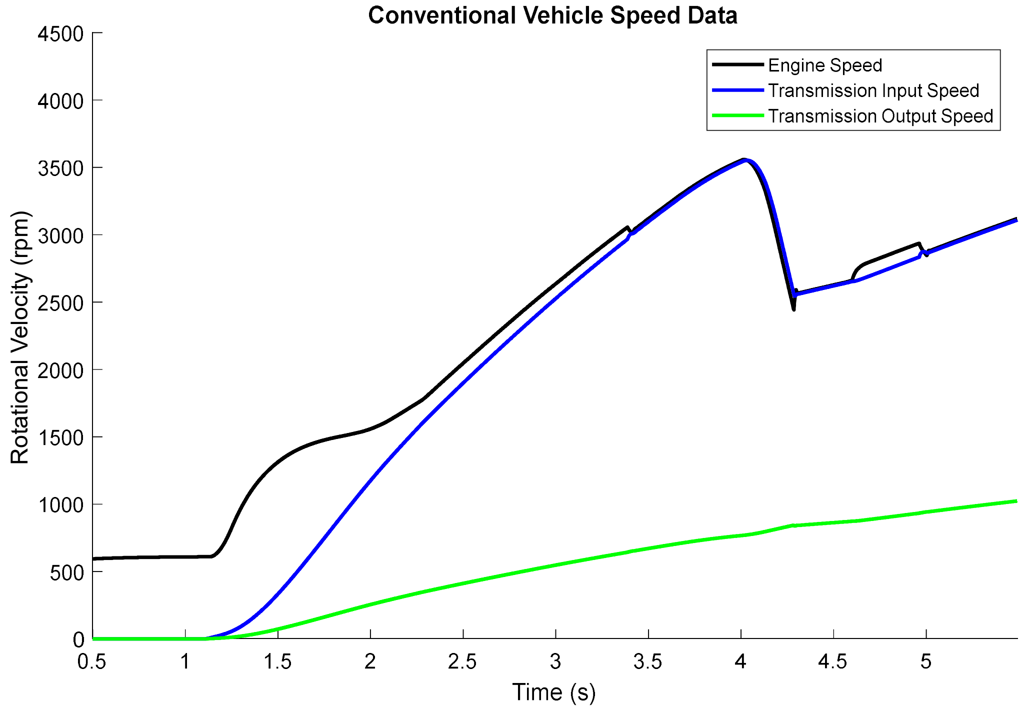

Figure 32: 42% Accelerator Pedal Engine and Transmission Speed Simulation Results for the Stock Configuration

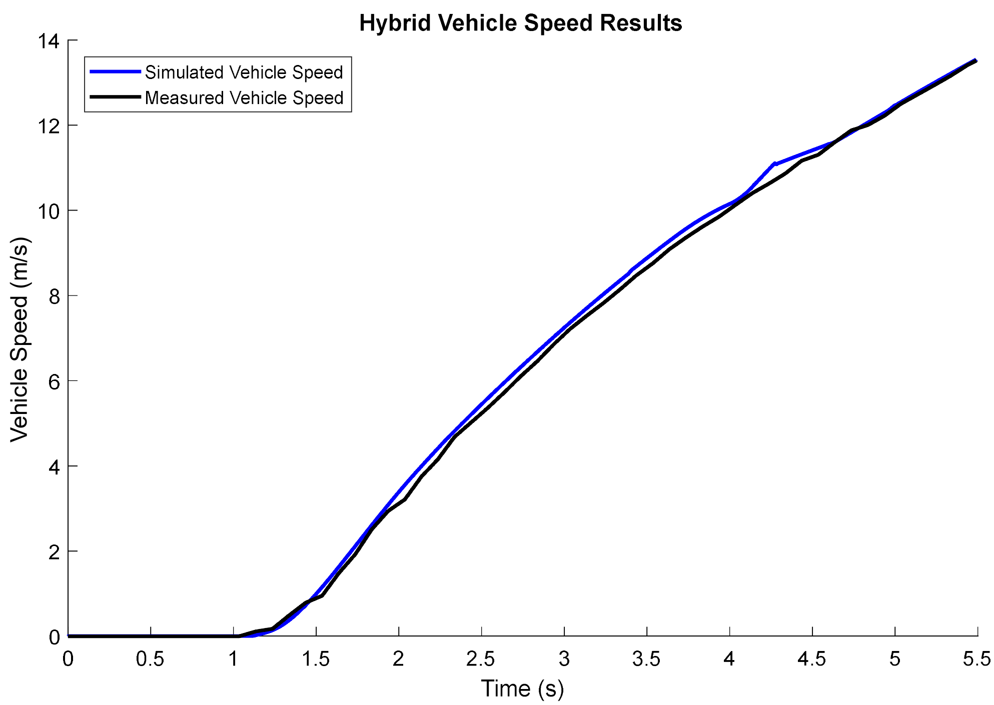

Figure 33: 42% Accelerator Pedal Vehicle Speed Simulation Results in the Hybrid Configuration Against the Recorded Data

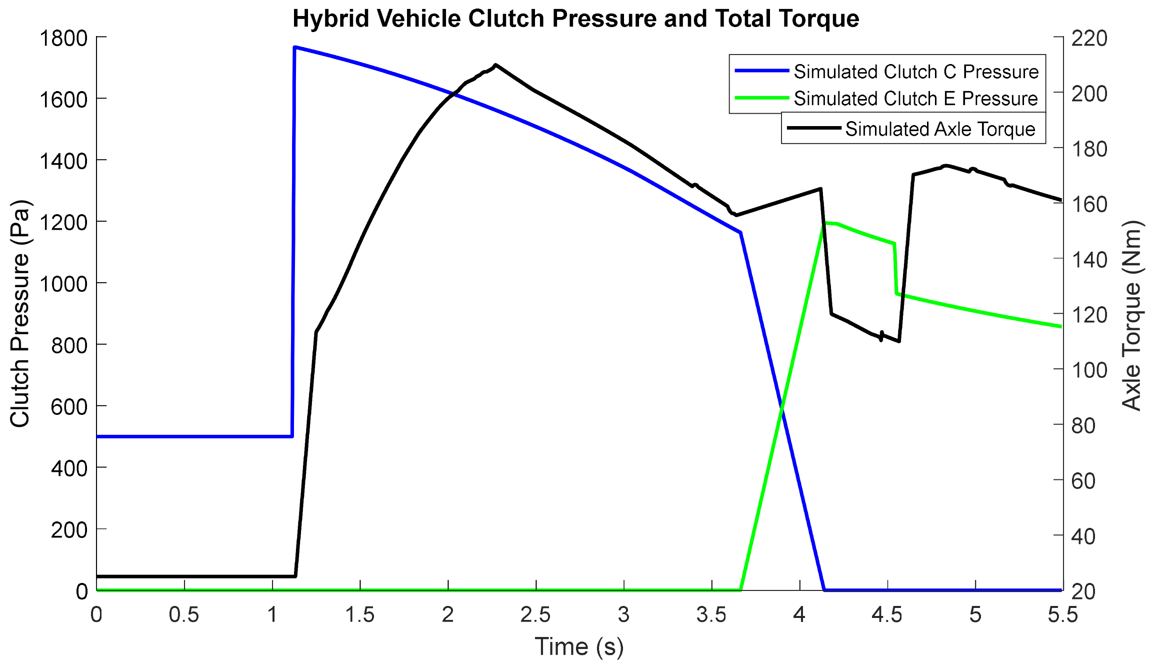

Figure 34: 42% Accelerator Pedal Clutch Pressures and Axle Torque Simulation Results for the Hybrid Configuration

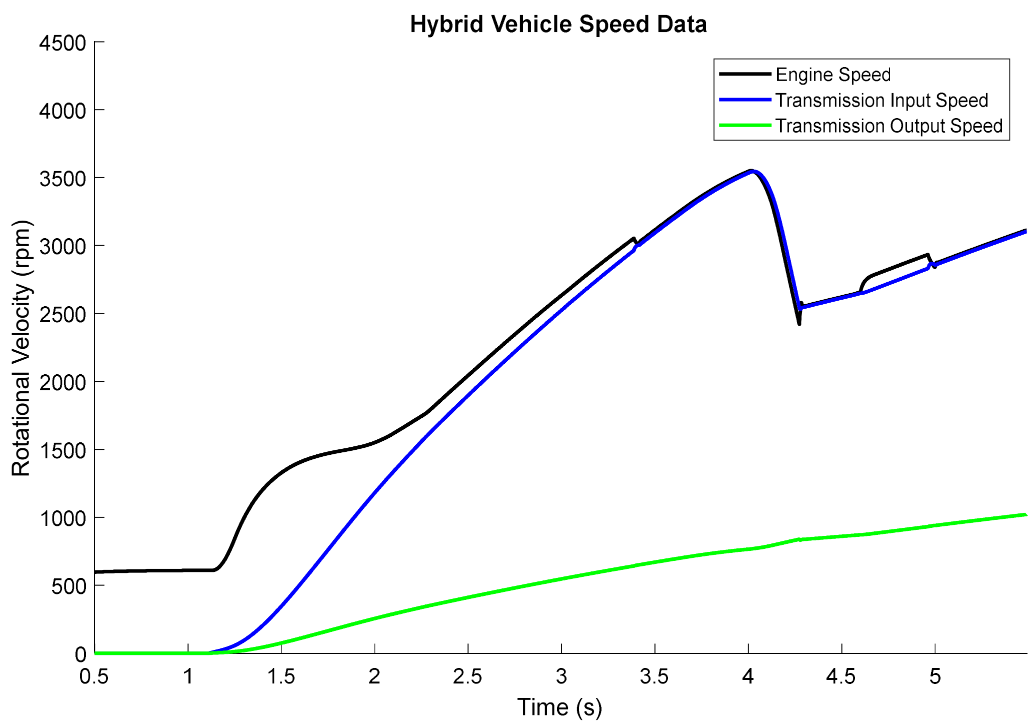

Figure 35: 42% Accelerator Pedal Engine and Transmission Speed Simulation Results for the Hybrid Configuration

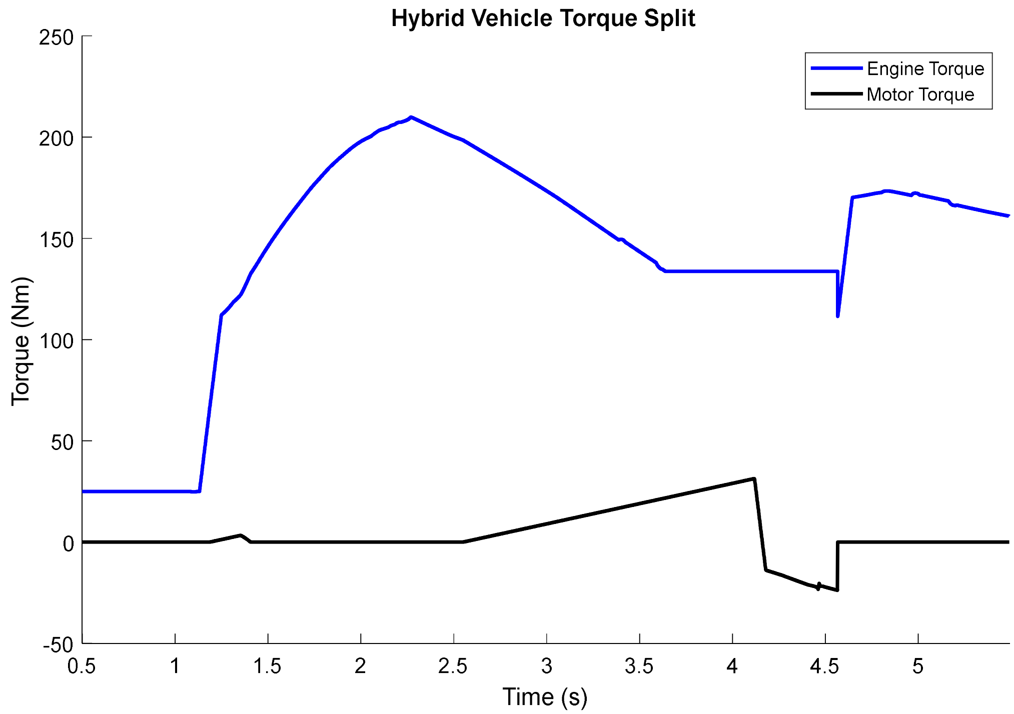

Figure 36: 42% Accelerator Pedal Engine and Electric Motor Torque Simulation Results for the Hybrid Configuration

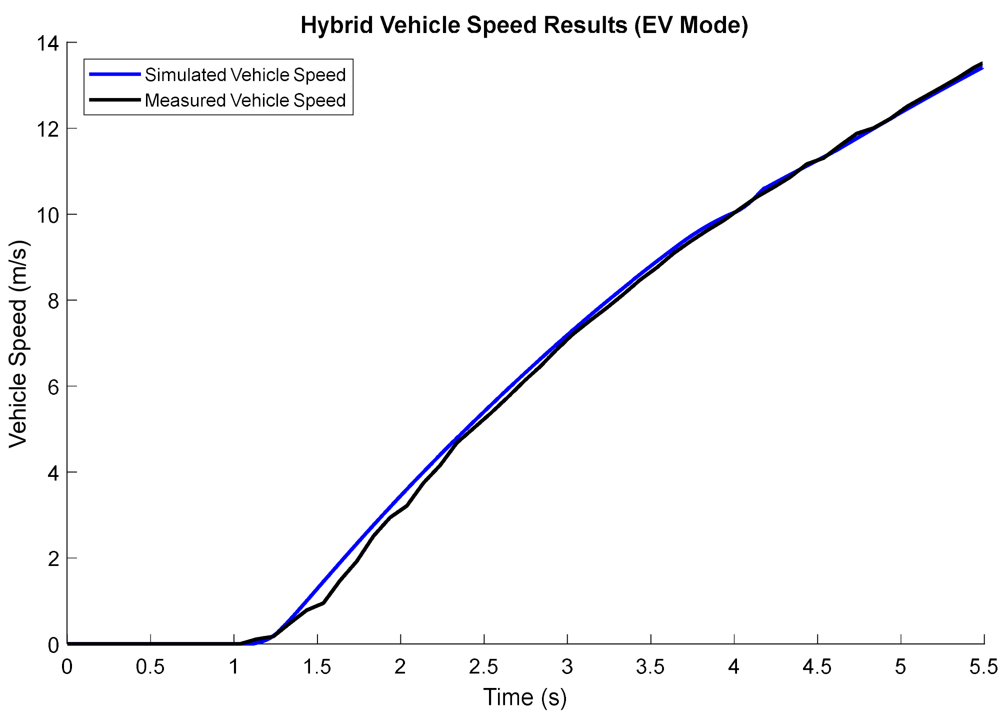

Figure 37: 42% Accelerator Pedal Vehicle Speed Simulation Results in EV Mode Against the Recorded Data

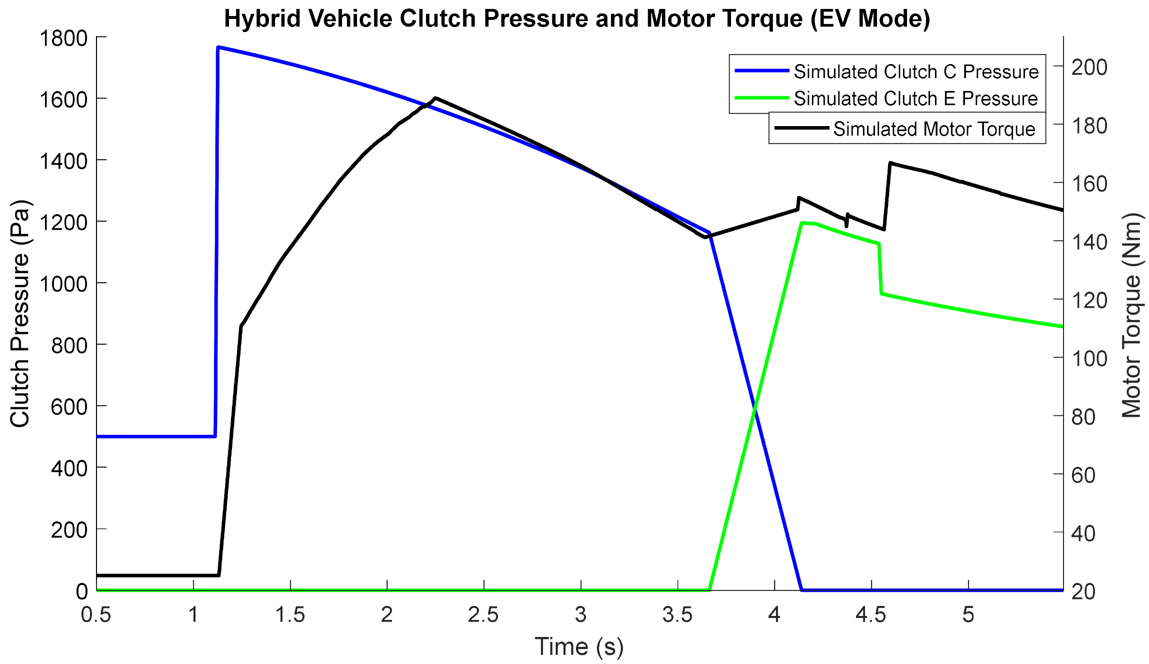

Figure 38: 42% Accelerator Pedal Clutch Pressures and Motor Torque Simulation Results in EV Mode

4.1.3 - 100% Accelerator Pedal Sample B

The last data sample shown in detail covers one of the 100% pedal samples covering the most extreme case of straight-line acceleration available. There are a couple of differences that this edge case sees in the data.

First, the data recording started right before the brake pedal was released. Normally there is a few seconds of delay that allows time for the Simulink model to wind up the rotating powertrain components and get the transmission input to a solid idle speed. While it did not negatively impact the simulation error in a significant way, it is a factor that changes the results slightly.

Second, the high pedal position causes the engine and transmission to wait a long time to shift from first to second gear. The engine and/or motor must go to a much higher speed to operate at the maximum torque multiplication. This is often done in performance vehicles where at near maximum accelerator pedal input, the transmission will expect the engine to operate near the maximum safe speed to improve the performance.

Despite these changes the model and torque control strategy continue to function well under the high acceleration environment with minimal deviation in the vehicle speed trace (See Figure 39, Figure 42, and Figure 46). Notably, compared to the other shifts the torque phase is much shorter relative to the inertia phase (See Figure 40, Figure 43, Figure 45. and Figure 47). The transmission and engine/motor speeds also behave normally aside from the short time between the start of the simulation and the brake release. The last significant change is that the torque converter clutch locks before the shift even occurs and staying locked for the rest to the time region (See Figure 41 and Figure 44).

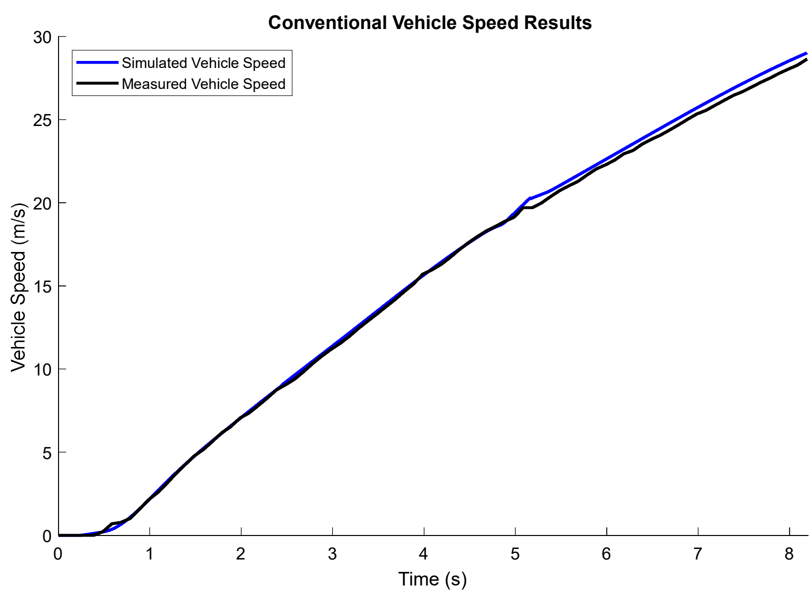

Figure 39: 100% Accelerator Pedal Vehicle Speed Simulation Results Against Recorded Data for the Stock Configuration

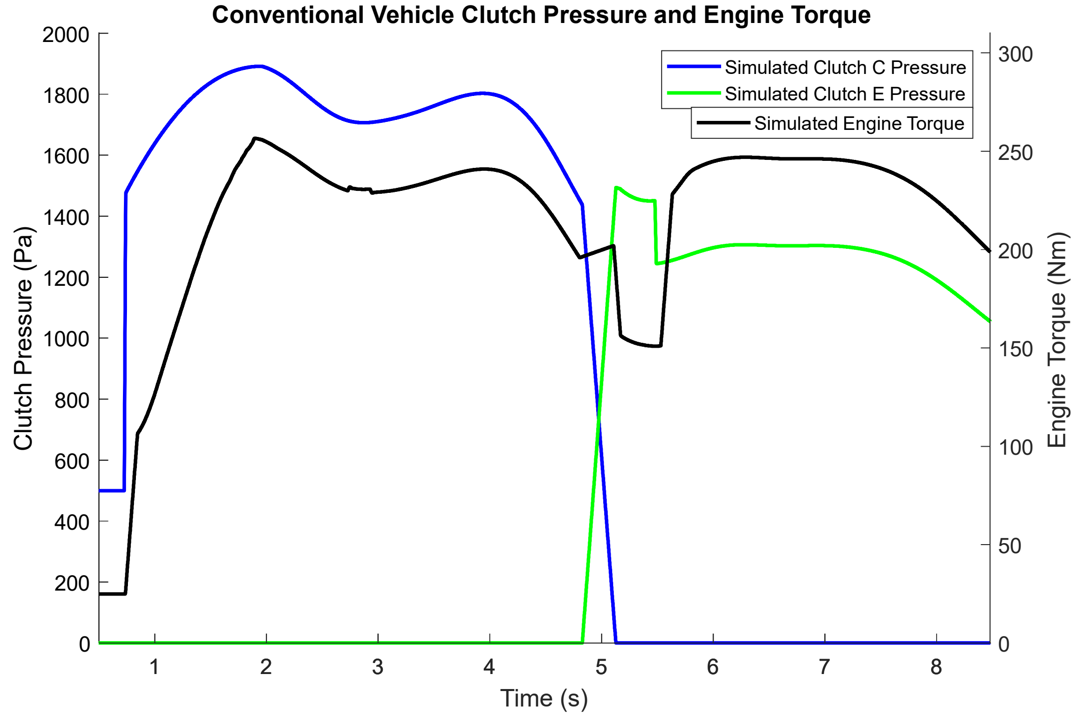

Figure 40: 100% Accelerator Pedal Clutch Pressures and Engine Torque Simulation Results for the Stock Configuration

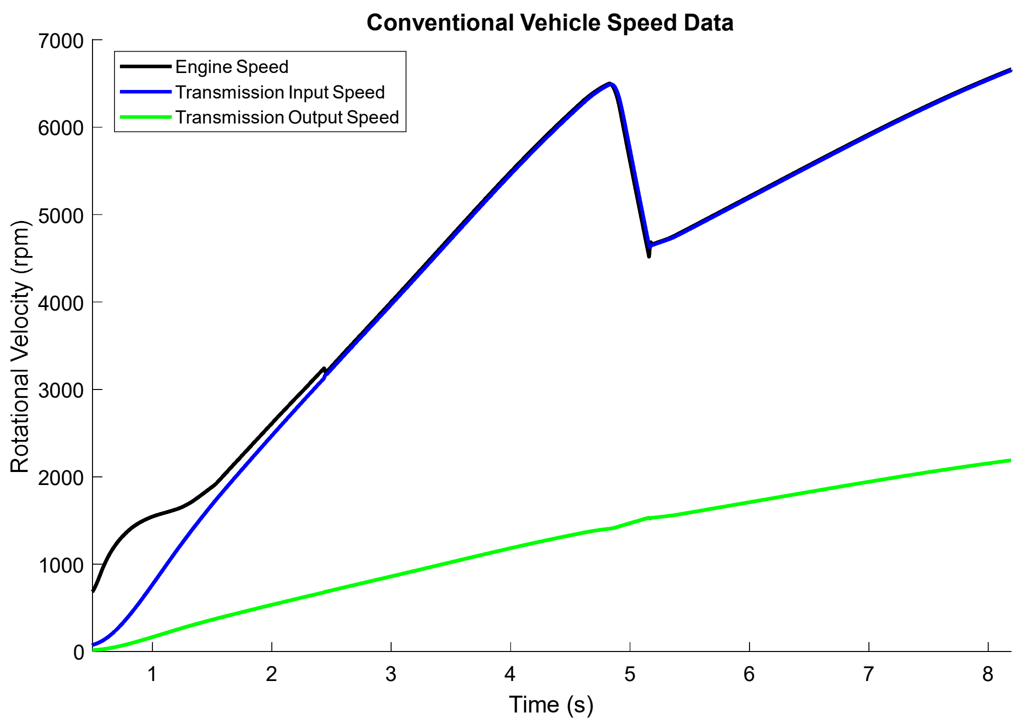

Figure 41: 100% Accelerator Pedal Engine and Transmission Speed Simulation Results for the Stock Configuration

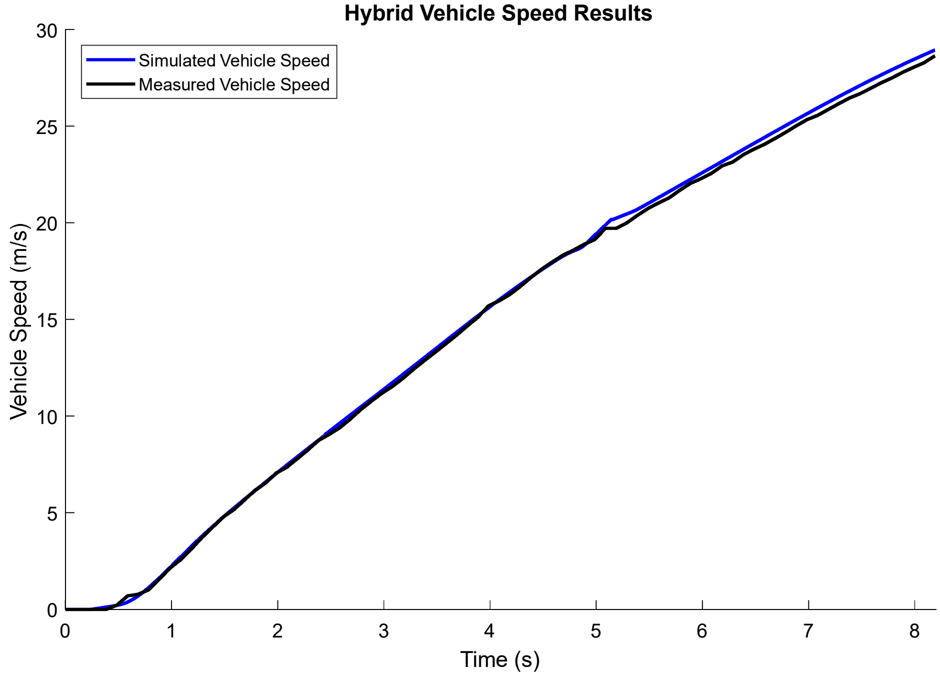

Figure 42: 100% Accelerator Pedal Vehicle Speed Simulation Results for the Hybrid Configuration Against Recorded Data

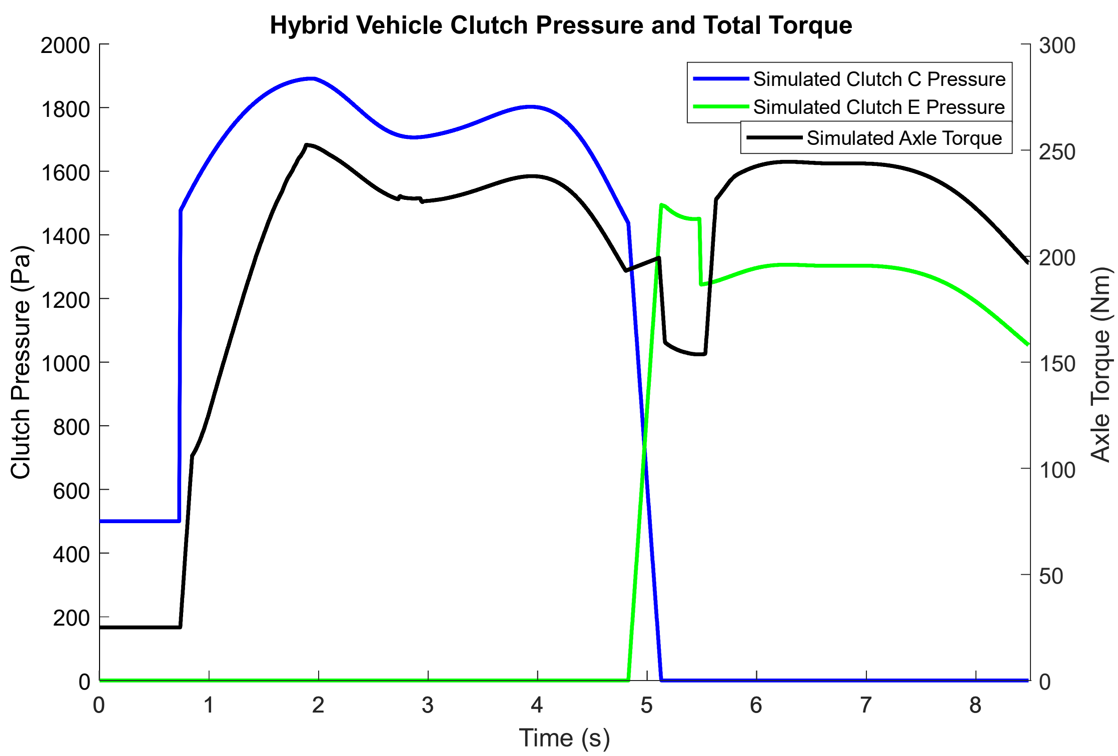

Figure 43: 100% Accelerator Pedal Clutch Pressures and Axle Torque Simulation Results for the Hybrid Configuration

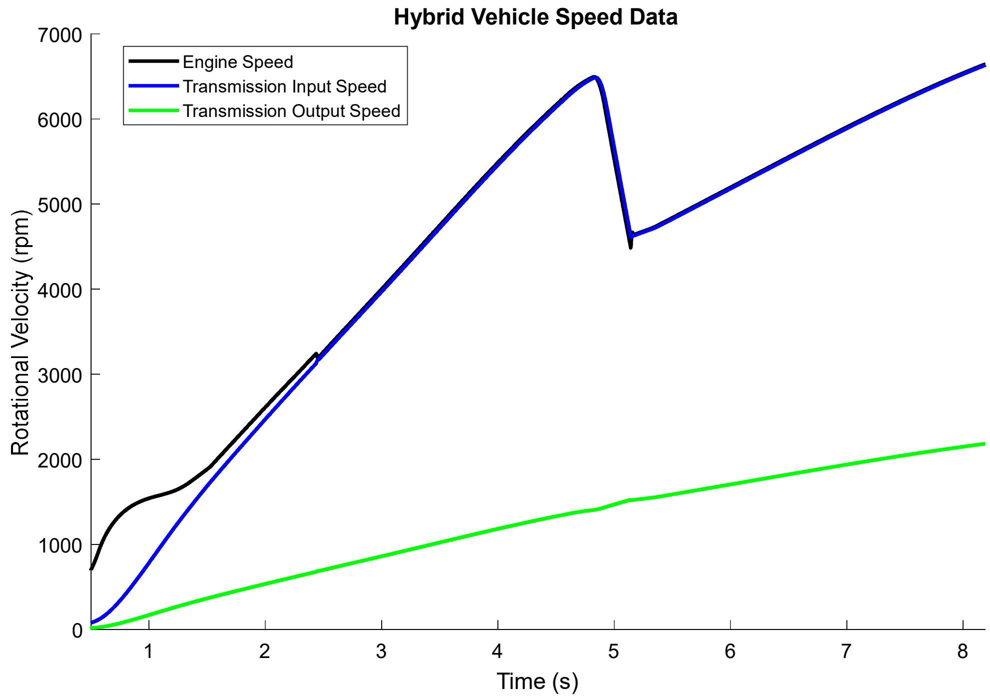

Figure 44: 100% Accelerator Pedal Engine and Transmission Speed Simulation Results for the Hybrid Configuration

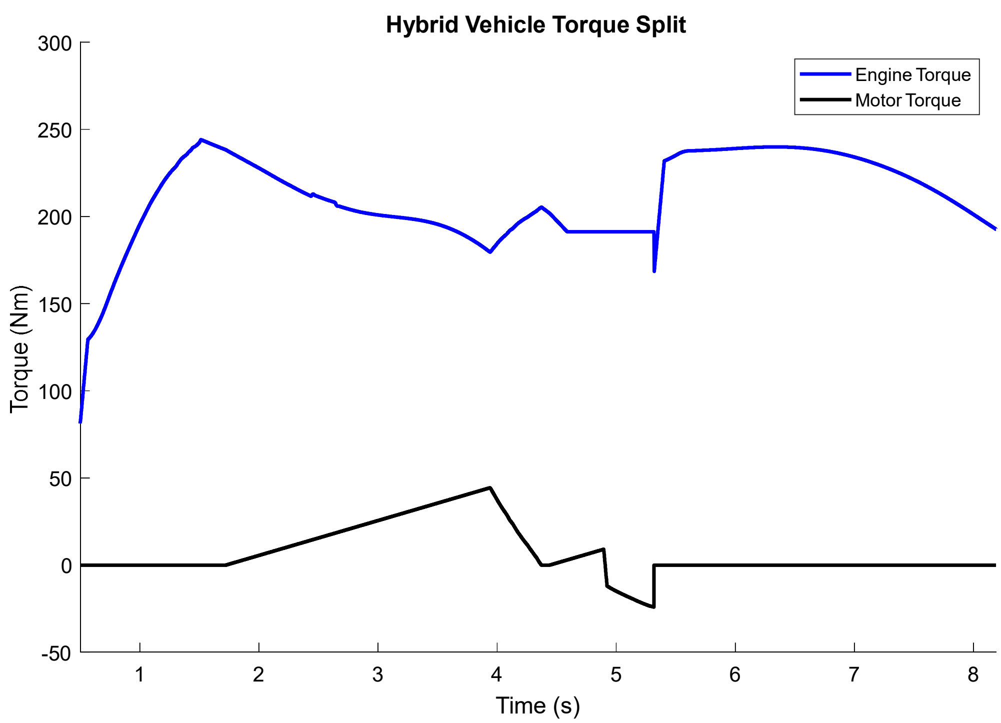

Figure 45: 100% Accelerator Pedal Engine and Electric Motor Torque Simulation Results for the Hybrid Configuration

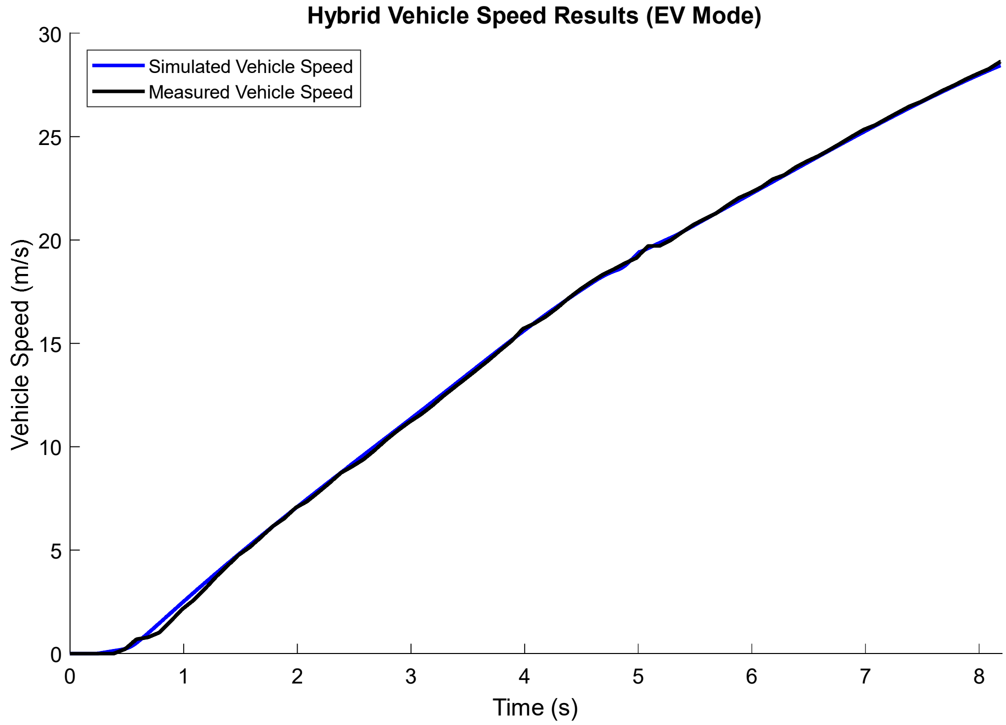

Figure 46: 100% Accelerator Pedal Vehicle Speed Simulation Results in EV Mode Against Recorded Data

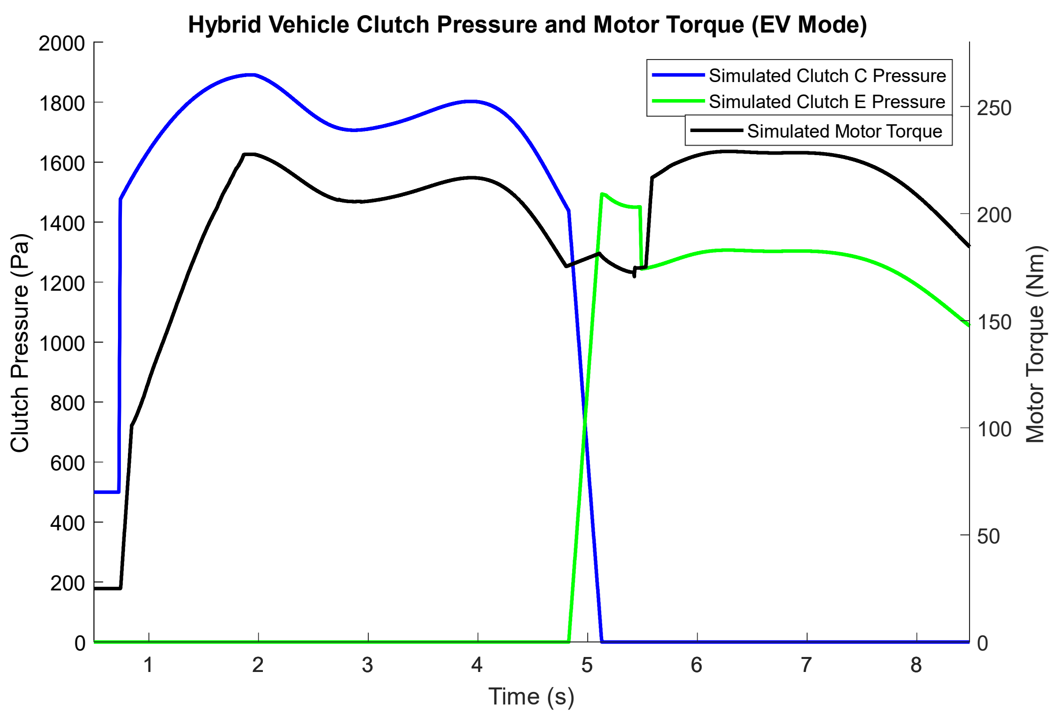

Figure 47: 100% Accelerator Pedal Clutch Pressures and Motor Torque Simulation Results in EV Mode

4.2 Reviewing Simulation Errors

When reviewing the data there were also a few data files that showed more significant error in the vehicle speed trace (MAE >0.4m/s), or there were odd discontinuities in the vehicle speed trace. Both of them have been pulled out here to look at the data and explain why the results from these samples have issues due to the recorded stock vehicle data.

4.2.1 - 25% Accelerator Pedal Sample B

The simulation data for this sample seems to produce the results you would expect for a 25% accelerator pedal position. A couple plots of the stock vehicle simulation are included below (Figure 48, Figure 49, and Figure 50). The issue appears in the recorded vehicle data where something causes the vehicle speed to lag below expectation. Looking at other recorded signals it is not clear what the source is since the accelerator pedal position and axle torque command signals are appropriate for a straight-line acceleration of this magnitude. Other signals that might explain the issue are not available in the record. One possibility is that the rear wheels may have lost traction while the front wheel were used to record the vehicle speed, but that is just speculation. Regardless, the recorded vehicle speed data is anomalous and so the simulation error is not considered in the final analysis.

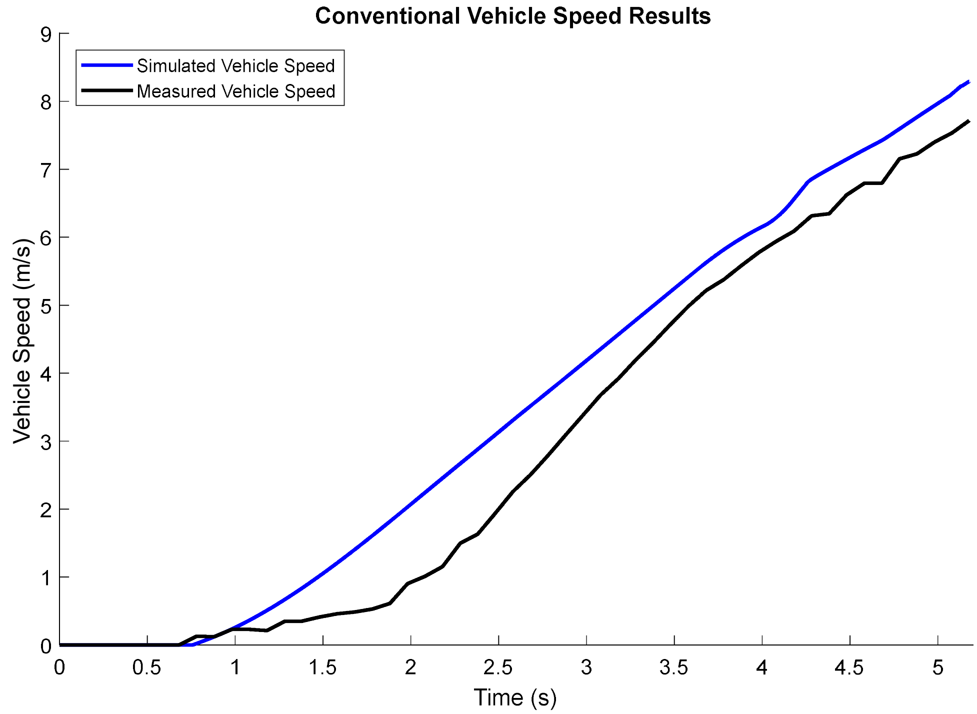

Figure 48: 25% Accelerator Pedal Vehicle Speed Simulation Results Against Unusual Recorded Data

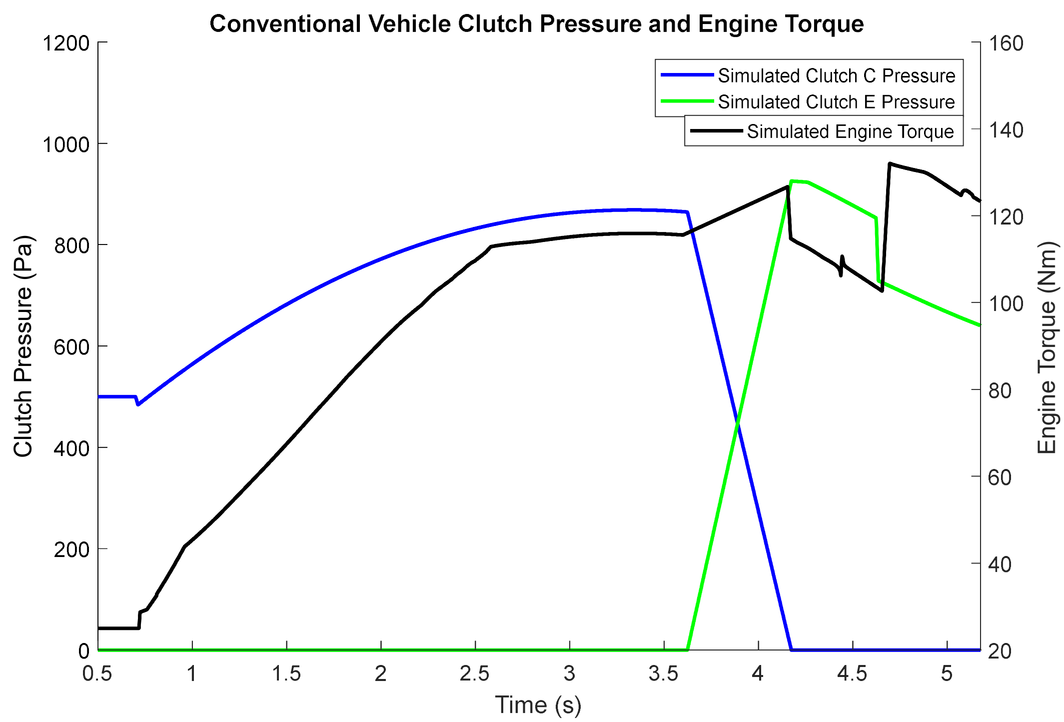

Figure 49: 25% Accelerator Pedal Clutch Pressures and Engine Torque Simulation Results for the Stock Configuration

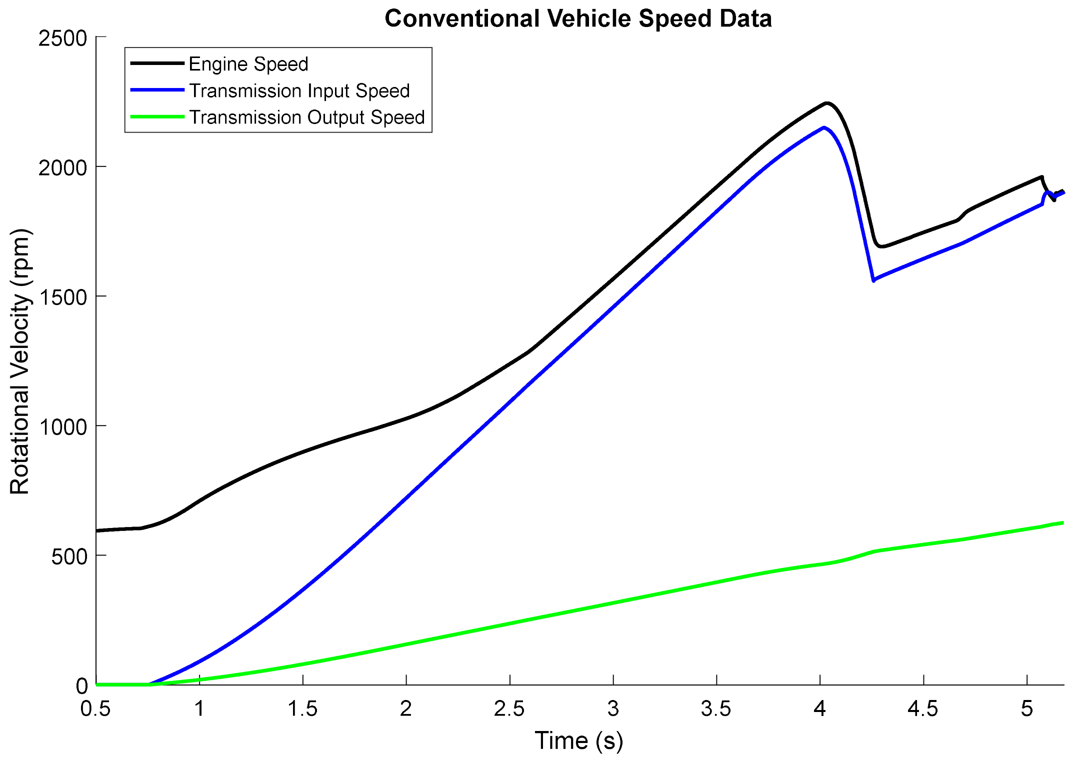

Figure 50: 25% Accelerator Pedal Engine and Transmission Speed Simulation Results for the Stock Configuration

4.2.2 - 42% Accelerator Pedal Sable B

This data sample has 2 issues that make it suspect for use in validating the research questions. The minor issue is that similar to the 100% accelerator pedal sample b mentioned in section 4.1.3, there is minimal time between the start of the data recording and the start of vehicle motion, not allowing for sufficient time for the model to reach idle steady state before the brake is released. The major issue is that similar to 25% accelerator pedal sample b mentioned in section 4.2.1 that while there is no obvious discontinuity in the recorded accelerator pedal and axle torque signals recorded on the stock vehicle, the recorded vehicle speed is inconsistent (Figure 51) during the first 2 seconds making it unreliable to test the quality of the simulation and torque strategy. Though, the simulation again does seem to provide reasonable results as well (Figure 52 and Figure 53) even if there is not a baseline to make a comparison against.

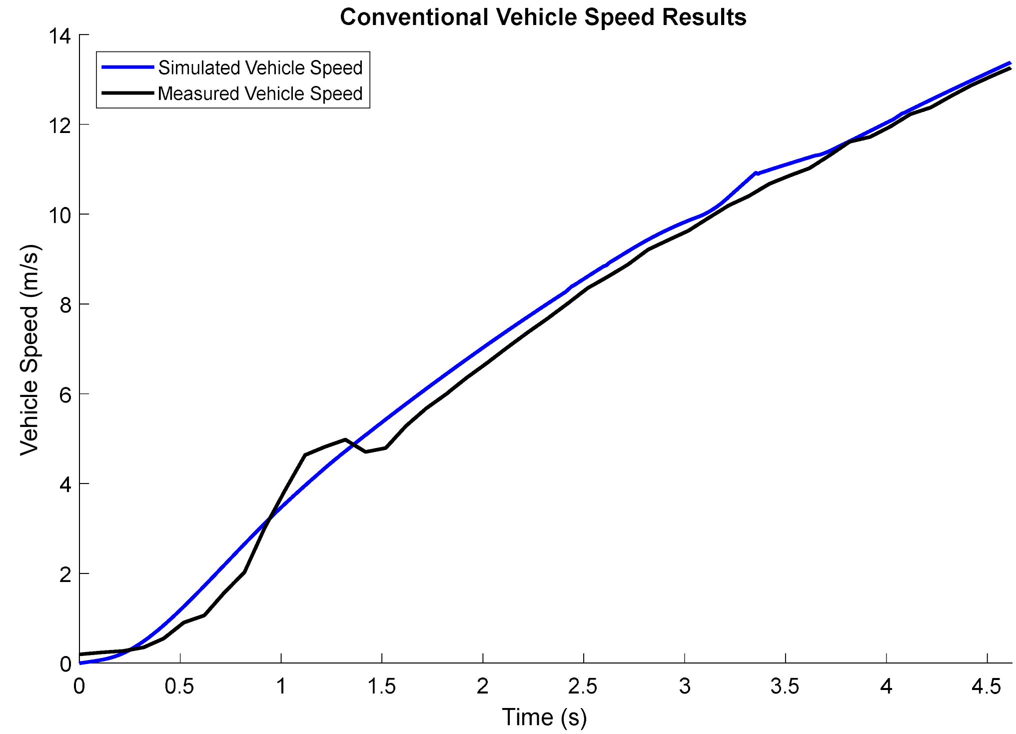

Figure 51: 42% Accelerator Pedal Vehicle Speed Simulation Results Against Unusual Recorded Data for the Stock Configuration

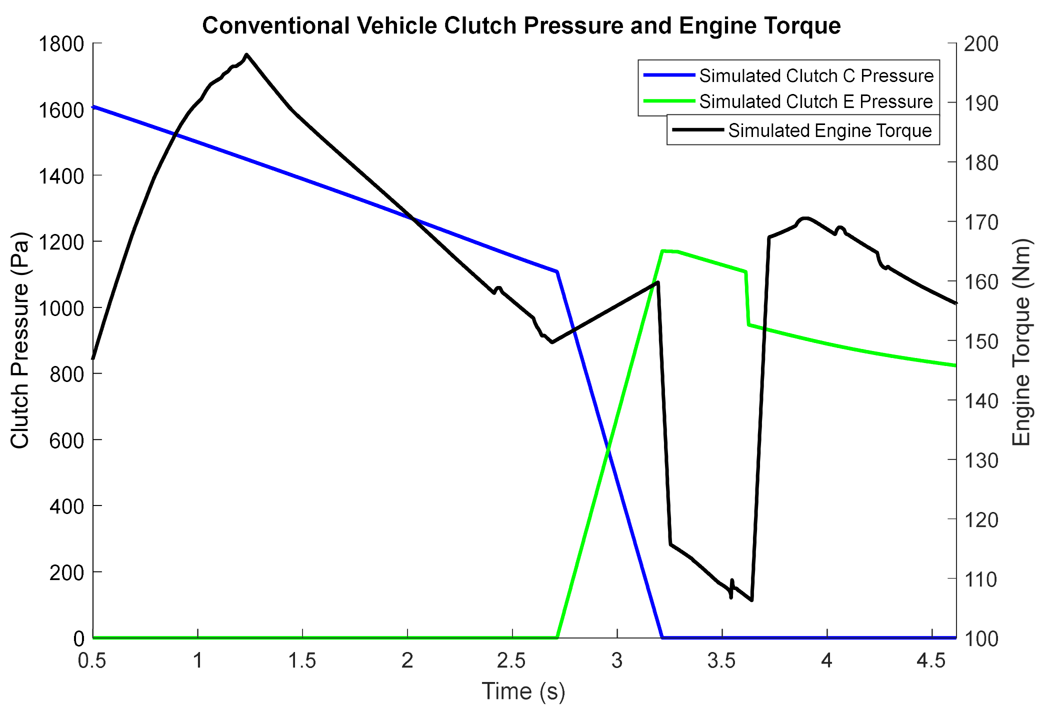

Figure 52: 42% Accelerator Pedal Clutch Pressures and Engine Torque Simulation Results for the Stock Configuration

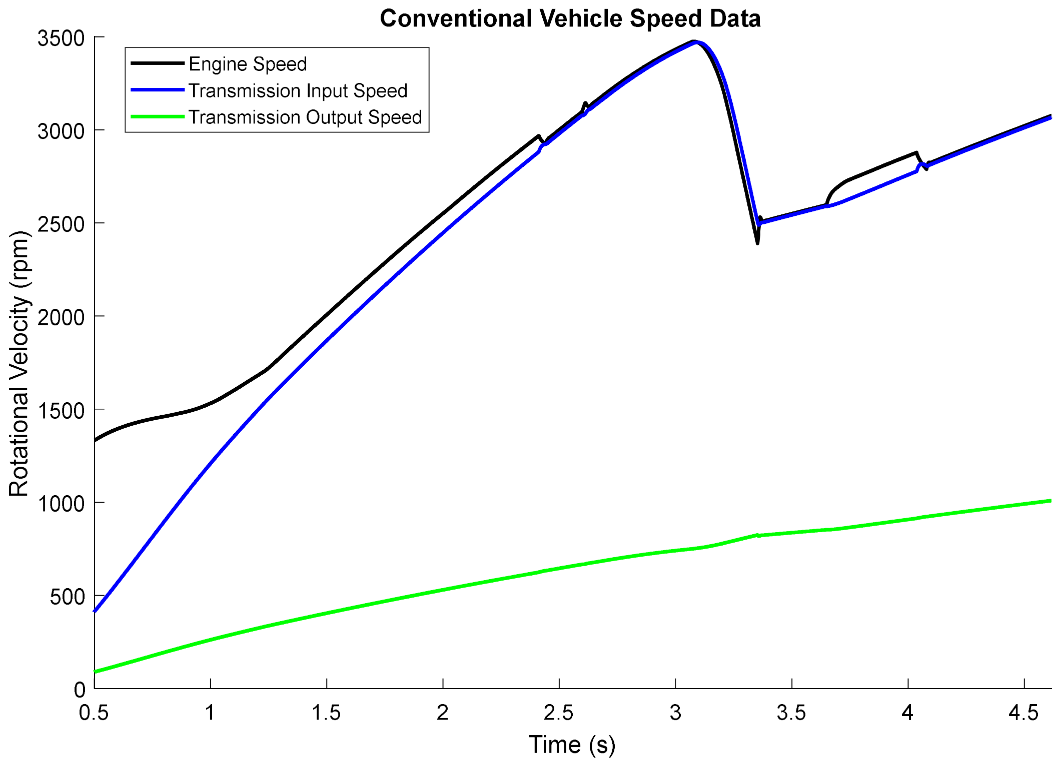

Figure 53: 42% Accelerator Pedal Engine and Transmission Speed for the Stock Configuration

4.3 Summary of Simulation Results

In the table below, all the data for 1-2 shifts is summarized with the MAE which was calculated as described above. The MAE is not the same for all data sets and outside the data sets covered in section 4.2, all the MAE values are staying at reasonable values and are consistent between all configurations.

Table 4: Summarized Mean Absolute Percentage Errors for all 1-2 Shifts

Data Sample	Stock Vehicle m/s Error	Hybrid Mode m/s Error	EV Mode m/s Error
8%a	0.08	0.08	0.07
8%b *	-No data available-		
17%a**	0.27	0.24	0.05
17%b	0.15	0.15	0.10
25%a	0.43	0.41	0.28
25%b***	0.60	0.57	0.39
33%a	0.39	0.38	0.23
33%b	0.37	0.35	0.26
42%a**	0.17	0.16	0.12
42%b***	0.28	0.27	0.24
50%a	0.37	0.36	0.27
50%b	0.20	0.20	0.18
58%a	0.18	0.18	0.20
58%b	0.19	0.19	0.17
67%a	0.26	0.24	0.20
67%b	0.14	0.14	0.19
75%a *	-No data available-		
75%b	0.23	0.21	0.19
83%a	0.10	0.09	0.17
83%b	0.16	0.14	0.15
92%a	0.09	0.09	0.19
92%b	0.18	0.16	0.18
100%a	0.24	0.21	0.09
100%b**	0.23	0.20	0.09

* Simulation could not be completed.

** Data reviewed in section 4.1 as good results.

*** Data reviewed in section 4.2 for the reason behind high MAE or unusual data output.

As a note, the data for 8% pedal b was incomplete, not all the required signals were recorded so it was not practical to try to run the simulation with it, the data for 75% pedal a failed to converge when running the simulation in Matlab and a reason could not be easily determined so the data sample was excluded.

To get a general picture of the trends across all the data samples the average of all MAE results was taken (excluding those called out in section 4.2, or the notes below Table 5). These averaged MAE results show the consistent fit between the stock and hybrid configurations and an interesting correlation.

The stock vehicle and hybrid mode simulations have almost the same MAE, (0.21 m/s stock, 0.20 m/s hybrid), while the average MAE for EV Mode is only 0.17m/s. The reason for this is likely that the inertia of the electric motor in EV Mode is so much lower than in the hybrid or stock model that given the electric motors ability to rapidly change torque, the effects of rotational inertia are much less, and the model can follow the trace much closer.

This indicates that the model can accurately replicate stock vehicle conditions and shows that the torque control strategy can accurately command torque both in EV and Hybrid operation to fulfill the transmissions needs during a shift.

5.0 FUTURE WORK

There are many places that this strategy could be expanded in the future beyond just integrating it into the physical vehicle and tuning some of the calibrations to the real hardware. Increasing the fidelity of the shift control strategy would be balanced against other priorities in vehicle development. The potential improvements outlined below would have variable improvements to drive quality and/or fuel economy which would have to be balanced against other improvements that could be made to the vehicle that may have a larger impact.

Some of the options for improvement include the integration of closed loop controls. Given the short duration of shifts it can often be overlooked, but there are several things that could cause non-standard shifts if closed loop controls are not present. These include things such as the driver changing their control inputs to the vehicle or other environmental factors such as road grade. Any of these can impact the length of the overall shift or individual shift phases. Closed-loop controls could be used to monitor the progress through the shift versus the expected change over time and compensate for deviations. This could be pushed further to include learning algorithms that could adapt to longer term changes like, driver character (how each individual driver interacts with the vehicle controls), towing a trailer or other significant cargo, and other external factors that could cause the vehicle response to change in the short or long term from the nominal case. Some of these considerations are also discussed in the introduction of (Mishra & Srinivasan, 2015).

Other improvements include working more on the torque split strategy to eliminate backlash during the inertia phase torque cut completely. If it is not feasible to eliminate gear backlash entirely, another option would be to design a strategy to control the motor as it moves through the backlash to reduce the driver's ability to sense the change in gear tooth engagement. Controlling this gear lash is often a significant challenge on many hybrid or electrified vehicles as they get close to market release.

One data point that was lacking as an input to the torque control strategy was a method to characterize the response delay of the engine to a command. This would have to be managed during the integration with the physical vehicle since the solution described here calculates the torque that needs to be produced each loop, but to achieve that there would need to be some degree of over-scaling in the engine torque request to make it achievable.

Last, as mentioned before, the focus of this thesis was on the operation during the shift and the torque split that could take advantage of the hybrid powertrain. What was not in scope of the thesis was an over-arching torque strategy for the vehicle. This would include discussions critical to a Chevy Camaro about how much deference to give to improving the drive quality and performance metrics on a muscle car over using the electric motor to improve fuel economy. Some middle ground would have to be found, and then this shift control strategy would have to be integrated into the larger strategy. This could be a torque authority handoff during the shift, or just a suggestion or input into the larger strategy that can be followed or ignored based on the status of the vehicle hardware such as the state of charge in the high voltage battery.

6.0 CONCLUSIONS

Through this thesis the feasibility of taking control of torque commands from the transmission during gear shift was explored. Given the changes to the torque architecture of the CSU EcoCAR 3 vehicle due to design decisions of the team and the inability to modify transmission control software some issues in drive quality were bound to arrive as vehicle fidelity progressed. The systems built in the transmission software to provide a smooth and uninterrupted delivery of torque during gear changes were bound to become unreliable since the software was not built with an ability to manage different, or changing rotational inertias connected to the transmission input directly. While there were closed loop controls to make sure shifts would be successful, performance would be degraded, and transmission lifespan could be shortened.

To overcome this, a model was developed that could represent the dynamics of the vehicle both in the original stock configuration, and as a modified hybrid. This included most of the transmission internals and the 5 friction clutches inside it. As the model increased in complexity it was checked against the stock vehicle data to ensure that it was representing the dynamics of the real vehicle as accurately as the simulation would allow until of all major rotating elements were represented with their proper states before, during, and after a shift event.

Once the model of the stock vehicle was validated to the recorded data it was modified slightly to represent the student's designs. These changes primarily consisted of replacing the stock engine with a smaller E-85 engine, an electric motor, and an engine disconnect clutch. With these changes to torque sources and inertia, a control strategy was built that could predicted the length and dynamics of an upcoming shift when the transmission made the decision to shift gears. But rather than continuing to follow the recommendations from the transmission on the amount of torque to produce, the control system compensated for the changed inertia while following the transmission shift process to make sure the engine speed and torque changed as needed to ensure consistent torque delivery to the wheels. While doing so, the control system also selected specific torque distributions between the electric motor and the engine to eliminate the need to alter engine ignition timing during the inertia phase when a sudden drop in net torque is needed at the transmission input. By eliminating the need for this timing change, the engine can continue to operate at a higher efficiency point through the entire shift and significantly improve the mechanical effort produced by all fuel consumed during that time.

The results of the simulation across a wide variety of shifts from first to second gear show that the strategy can adapt to multiple different situations when shifts occur and the developed control strategy could be integrated into the broader torque control strategy for the vehicle as part of a larger fuel economy improvement plan.

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APPENDIX

A.1 Transmission Equations and Base Relations

A.1.1First Gear:

In first gear the A, B, and C clutches are closed.

$$(R+S)\omega_P=R\omega_R+S\omega_S$$

$$\omega_{S3}=\omega_{R3}=\omega_{P3}=\omega_{S2}=\omega_{R4}=0$$

$$\zeta_1=\frac{\omega_S}{\omega_P}=\frac{S4+R4}{S4}=4.62$$

A.1.2 Second Gear:

In second gear clutches A, B, and D are closed. Therefore, the same gears which did not move in 1st gear remain stationary.

$$\omega_{S3}=\omega_{R3}=\omega_{P3}=\omega_{S2}=\omega_{R4}=0$$

$$(R4+S4)*\omega_{P4}=S4*\omega_{S4}$$

Where ω_{P4} is the output speed of the transmission. Since clutch D is closed $\omega_{S4}=\omega_{R2}$ and ω_{P2} is equivalent to the transmission input

$$(R2+S2)*\omega_{P2}=R2*\omega_{R2}=R8*(\frac{R4+S4}{S4}*\omega_{P4})$$

$$\zeta_2=\frac{\omega_{P2}}{\omega_{P4}}=\frac{R2*(R4+S4)}{S4*(R2+S2)}=3.04$$

A.1.3 Third Gear:

In third gear clutches B and D remain locked while A opens and C closes. In this configuration all components of gear set 2 rotate at the input speed which means that ω_{S3} is the input and ω_{P4} is the output speed.

$$\omega_{R2}=\omega_{P2}=\omega_{S2}=\omega_{S3}=\omega_{S1}=\omega_{S4}=\omega_{In}$$

$$\omega_{P1}=\omega_{P4}=\omega_{Out}$$

$$\omega_{R3}=0$$

Starting at gear set 4,

$$(R4+S4)*\omega_{P4}=R4*\omega_{R4}+S4*\omega_{S4} \qquad (1)$$

$$\omega_{R4}=\omega_{P3} \qquad (2)$$

$$(R3+S3)*\omega_{P4}=S3*\omega_{S3} \qquad (3)$$

Using 2 substitute (3) into (1),

$$(R4+S4)*\omega_{P4}=R4*\left(\frac{S3*\omega_{S4}}{R3+S3}\right)+S4*\omega_{S4}=\left(\frac{R4*S3}{R3+S3}+S4\right)*\omega_{S4}$$

Where ω_{S4} is ω_{In}

$$\zeta_3=\frac{\omega_{S4}}{\omega_{P4}}=\frac{(R4+S4)*(R3+S3)}{R4*S3+S4*R3+S4*S3}=2.07$$

A.1.4 Fourth Gear:

In fourth gear clutches B and D remain closed again and C opens and E closes. In this configuration ω_{P4} is the output and ω_{P2} is the input.

$$\omega_{S1}=\omega_{P1}=\omega_{R1}=\omega_{R2}=\omega_{P3}=\omega_{S4}=\omega_{P4}=\omega_{R4}=\omega_{Out}$$

$$\omega_{P2}=\omega_{In}$$

$$\omega_{R3}=0$$

$$(S3+R3)*\omega_{P3}=S3*\omega_{S3}+R3*\omega_{R3}=S3*\omega_{S3} \qquad (1)$$

$$\omega_{S3}=\omega_{S2} \qquad (2)$$

$$(S2+R2)*\omega_{P2}=S2*\omega_{S2}+R2*\omega_{R2} \qquad (3)$$

Using 2 substitute (3) into (1),

$$(S3+R3)*\omega_{P3}=S3*\left(\frac{(S2+R2)*\omega_{P2}-R2*\omega_{R2}}{S2}\right)$$

Where ω_{P3} and ω_{R2} are both equal to ω_{Out} and ω_{P2} is equal to ω_{In}. After manipulation,

$$\zeta_4=\frac{\omega_{P3}}{\omega_{P2}}=1+\frac{S2*R3}{S3*(R2+S2)}=1.66$$

A.1.5 Fifth Gear:

In fifth gear clutches B and E remain closed while D opens and C closes. The primary input and output gears remain the same as in fourth gear.

$$\omega_{P4}=\omega_{P1}=\omega_{Out}$$

$$\omega_{R1}=\omega_{P2}=\omega_{S4}=\omega_{In}$$

$$\omega_{R3}=0$$

$$(R4+S4)*\omega_{P4}=S4*\omega_{S4}$$

$$\omega_{R4}=\omega_{P3}$$

$$(S3+R3)*\omega_{P3}=S3*\omega_{S3}+R3*\omega_{R3}=S3*\omega_{S3}$$

$$\omega_{S3}=\omega_{S2}$$

$$(S2+R2)*\omega_{P2}=S2*\omega_{S2}+R2*\omega_{R2}$$

$$\omega_{R2}=\omega_{S1}$$

$$(S1+R1)*\omega_{P1}=S1*\omega_{S1}+R1*\omega_{R1}$$

Now, replacing ω_{In} and ω_{Out} and working backwards,

$$(R4+S4)*\omega_{Out}=S4*\omega_{In}+R4*\left(\frac{S3*\omega_{S3}}{R3+S3}\right)$$

$$(R4+S4)*\omega_{Out}=S4*\omega_{In}+\left(\frac{S3*R4}{R3+S3}\right)*\left(\frac{(R2+S2)*\omega_{In}-R2*\omega_{R2}}{S2}\right)$$

$$\omega_{R2}=\frac{R1+S1}{S1}*\omega_{Out}-\frac{R1}{S1}*\omega_{In}$$

$$\zeta_5=\frac{\omega_{In}}{\omega_{Out}}=\frac{R4+S4+\frac{R2*S3*R4*(R1+S1)}{S2*S1*(R3+S3)}}{S4+\frac{R4*S3*(R2+S2)}{(R3+S3)*S2}+\frac{R1*R2*R4*S3}{S1*S2*(R3+S3)}}=1.26$$

A.1.6 Sixth Gear:

In sixth gear clutches C and E remain closed while B opens and D closes.

$$\omega_{S1}=\omega_{P1}=\omega_{R1}=\omega_{S2}=\omega_{P2}=\omega_{R2}=\omega_{S3}=\omega_{P3}=\omega_{R3}=\omega_{S4}=\omega_{P4}=\omega_{R4}=\omega_{In}=\omega_{Out}$$

$$\zeta_6=\frac{\omega_{In}}{\omega_{Out}}=1$$

A.1.7 Seventh Gear:

In seventh gear clutches C and E remain closed while D opens and A closes.

$$\omega_{R1}=\omega_{P2}=\omega_{S4}=\omega_{In}$$

$$\omega_{P1}=\omega_{P4}=\omega_{Out}$$

$$\omega_{S2}=\omega_{S3}=0$$

$$(R2+S2)*\omega_{P2}=(R2+S2)*\omega_{In}=R2*\omega_{R2}+S2*\omega_{S2}=R2*\omega_{R2}$$

$$\omega_{S1}=\omega_{R2}$$

$$(R1+S1)*\omega_{out}=R1*\omega_{In}+S1*\left(\frac{R2+S2}{R2}\right)*\omega_{In}$$

$$\zeta_7=\frac{(R1+S1)*R2}{S1*S2+R2*(R1+S1)}=0.85$$

A.1.8 Eighth Gear:

In eighth gear clutches A and E remain closed while C opens and D closes.

$$\omega_{P2}=\omega_{In}$$

$$\omega_{S1}=\omega_{P1}=\omega_{R1}=\omega_{R2}=\omega_{S4}=\omega_{P4}=\omega_{Out}$$

$$\omega_{S2}=\omega_{S3}=0$$

$$(R2+S2)*\omega_{P2}=(R2+S2)*\omega_{In}=S2*\omega_{S2}+R2*\omega_{R2}=R2*\omega_{R2}$$

$$\zeta_8=\frac{R2}{R2+S2}=0.66$$

A.1.9 Reverse Gear:

In reverse clutches A, B, and E are closed.

$$\omega_{P2}=\omega_{In}$$

$$\omega_{P4}=\omega_{P1}=\omega_{Out}$$

$$\omega_{S2}=\omega_{S3}=\omega_{P3}=\omega_{R3}=\omega_{R4}=0$$

Using,

$$\omega_{S4}=\omega_{R1}$$

$$\omega_{S1}=\omega_{R2}$$

$$(R+S)*\omega_P=R*\omega_R+S*\omega_S$$

$$(S4+R4)*\omega_{Out}=S4*\omega_{S4}=S4*\left(\frac{S1+R1}{R1}*\omega_{Out}-\frac{S1}{R1}*\omega_{S1}\right)$$

$$\left(S4+R4-\frac{(S1+R1)*S4}{R1}\right)*\omega_{Out}=-\left(\frac{S1*S4}{R1}\right)*\omega_{S1}$$

$$\left(S4+R4-\frac{(S1+R1)*S4}{R1}\right)*\omega_{Out}=-\left(\frac{S1*S4}{R1}\right)*\left(\frac{(R2+S2)}{R2}*\omega_{In}\right)$$

$$\zeta_{Rev}=\frac{R2*(S1*S4-R1R4)}{S1*S4*(S2+R2)}=-3.93$$

A.2 Method for Estimating the Planetary Gear Ratios:

Given that gear tooth counts or gear ratios for each of the four planetary gear sets are not publicly available (using the information in Figure 3 and Table 2) the ratios of each gear set can be calculated by following the path from input (ω_{In}) to output (ω_{Out}) of the transmission and relating it to the final ratio for that gear (ζ_1). This relationship also defines how all the internal rotating components in the transmission connect to the input and output shafts. This becomes important later for modeling the vehicle and calculating torque commands. The equations below were derived to relate the input and output speed of the transmission for each of the selectable gears using the tooth counts of the relevant ring and gear tooth counts, Rx and Sx respectively. A detailed description on how these equations were derived for all gears is contained in the appendix but follows a process similar to that outlined in (Nezhadali & Eriksson, 2015).

Equation 18: General Equation for Planetary Speed Ratios

$$(R+S)\omega_P=R\omega_R+S\omega_S$$

Equation 19: Locked Gears in First Gear

$$\omega_{S3}=\omega_{R3}=\omega_{P3}=\omega_{S2}=\omega_{R4}=0$$

Equation 20: Final Gear Ratio Equation

$$\zeta_1=\frac{\omega_S}{\omega_P}=\frac{S4+R4}{S4}=4.62$$

Equation 21: Second Gear Relation

$$\zeta_2=\frac{\omega_{P2}}{\omega_{P4}}=\frac{R2*(R4+S4)}{S4*(R2+S2)}=3.04$$

Equation 22: Third Gear Relation

$$\zeta_3=\frac{\omega_{S4}}{\omega_{P4}}=\frac{(R4+S4)*(R3+S3)}{R4*S3+S4*R3+S4*S3}=2.07$$

Equation 23: Fourth Gear Relation

$$\zeta_4=1+\frac{S2*R3}{S3*(R2+S2)}=1.66$$

Equation 24: Fifth Gear Relation

$$\zeta_5=\frac{R4+S4+\frac{R2*S3*R4*(R1+S1)}{S2*S1*(R3+S3)}}{S4+\frac{R4*S3*(R2+S2)}{(R3+S3)*S2}+\frac{R1*R2*R4*S3}{S1*S2*(R3+S3)}}=1.26$$

Equation 25: Sixth Gear Relation

$$\zeta_6=\frac{\omega_{in}}{\omega_{out}}=1$$

Equation 26: Seventh Gear Relation

$$\zeta_7=\frac{(R1+S1)*R2}{S1*S2+R2*(R1+S1)}=0.85$$

Equation 27: Eighth Gear Relation

$$\zeta_8=\frac{R2}{R2+S2}=0.66$$

Equation 28: Reverse Gear Relation

$$\zeta_{rev}=\frac{R2*(S1*S4-R1R4)}{S1*S4*(S2+R2)}=-3.93$$

With these equations defined, approximate ring to sun gear ratios can be determined for all 4 planetary gear sets. As will be seen later, the actual tooth count for the planetary gear is not needed, only the ratio between the sun and ring gears. A simple method to calculate these gear set ratios is to use Matlab's optimization problem solver toolbox.

The initial guess values are taken from the end of (Apakidze, 2014) which are decent integer approximations that provide a good initial guess. The result from this start point are as follows, after 106 iterations.

Table 5: Planetary Gear Set Approximated Ring to Sun Gear Ratios

Gear Set	Ratio (R_x/S_x)
1	1.9282
2	1.905
3	1.9283
4	3.6283

```
function F = transtoothcalc(x)
F = [ ((x(8)+x(4))/x(8))-4.62;
       ((x(2)*(x(4)+x(8)))/(x(8)*(x(2)+x(6))))-3.04;
       (((x(4)+x(8))*(x(3)+x(7)))/((x(4)*x(7))+(x(8)*x(3))+(x(8)*x(7))))-2.07;
       (1+((x(6)*x(3))/(x(7)*(x(2)+x(6)))))-1.66;
       ((x(4)+x(8)+((x(2)*x(7)*x(4)*(x(1)+x(5)))/(x(5)*x(6)*(x(3)+x(7)))))/(x(
       8)+((x(4)*x(7)*(x(2)+x(6)))/(x(6)*(x(3)+x(7))))+((x(1)*x(2)*x(7)*x(4))/
       (x(5)*x(6)*(x(3)+x(7))))))-1.26;
       ((x(2)*(x(1)+x(5)))/((x(5)*x(6))+(x(2)*(x(1)+x(5)))))-0.85;
       (x(2)/(x(2)+x(6)))-0.66;
       ((x(2)*((x(5)*x(8))-(x(1)*x(4))))/(x(5)*x(8)*(x(6)+x(2))))+3.93;
       ];



% initial guess values
  x0 = [77; 78; 77; 103; 39; 42; 39; 29];
% solver options
  options = optimoptions('fsolve','Display','iter', 'TolX', 1e-   6,'TolFun',
1e-6);
 [x,fval] = fsolve(@transtoothcalcpublic,x0,options)
```