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Design of a Single Axle Regenerative Brake System for a 2005 Thru-the-Road Hybrid Electric Chevrolet Equinox

John William Miller
University of Tennessee - Knoxville

To the Graduate Council:

I am submitting herewith a thesis written by John William Miller entitled "Design of a Single Axle Regenerative Brake System for a 2005 Thru-the-Road Hybrid Electric Chevrolet Equinox." I have examined the final electronic copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Mechanical Engineering.

David K. Irick, Major Professor

We have read this thesis and recommend its acceptance:

William R. Hamel, Gary V. Smith

Accepted for the Council:
Carolyn R. Hodges

Vice Provost and Dean of the Graduate School

(Original signatures are on file with official student records.)

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___Anne Mayhew___
Vice Chancellor and
Dean of Graduate Studies

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A Thesis
Presented for the
Master of Science
Degree
The University of Tennessee, Knoxville

John William Miller
May 2006

Dedication

This work is dedicated to my wife Amanda. Her constant support and encouragement during this project always lifted my spirits and helped to keep me focused on the goal. She is my best friend and greatest inspiration.

Acknowledgments

I would like to thank Dr. David Irick for giving me the opportunity to attend graduate school and work with him the last few years. The experience has brought me in contact with many new engineering techniques that have truly added to my knowledge and interest of the field. I want to thank him for his guidance and input to this project. His dedication to his work, the students he teaches, and his work ethic to "never admit defeat" stand as a motto that I will take with me in my career.

I want to thank Dr. William Hamel and Dr. Gary Smith for their willingness to participate on my graduate committee. Their participation and input is highly valued.

David Smith has been a great friend and mentor during my time at the university. I want to thank David for his advice and help with questions on control theory and software/ hardware concerns.

Gary Hatmaker and Danny Graham have always been ready and willing to lend their help whenever I needed an expert electrician, machinist, or friend. Their input and advice has always led me in the right direction.

I would also like to thank the Department of Energy for financial support through the Graduate Automotive Technology Education (GATE) Center. The GATE Center provides opportunities for graduate students to be involved in the field of advanced automotive systems.

Abstract

The University of Tennessee through its mechanical engineering department is participating in an advanced vehicle design competition, Challenge X, sponsored by the Department of Energy and General Motors. Participants are challenged with redesigning a 2005 Chevrolet Equinox that will minimize energy consumption and reduce emissions while maintaining the stock performance, safety, and utility of the vehicle. The overall vehicle architecture and control system for Tennessee's vehicle entry is described while focus is given to the design of the single axle series regenerative braking system. A series regenerative braking system provides a means of capturing the kinetic energy otherwise lost in the form of heat using conventional hydraulic/friction brakes found in the stock vehicle. The purpose of this thesis is to describe the system design and operation of a tunable single axle regenerative brake system for use in the Challenge X vehicle. A dual stage brake pedal will allow the regenerative system to work in conjunction with the conventional system to provide the braking required to safely stop the vehicle. A system controller will interpret driver input through the brake pedal and vehicle status to determine how to proportion the energy during a braking event. The system is designed to operate "transparent to the driver." The driver should experience the same pedal feel and vehicle performance as with the stock system. Therefore, the system will require calibration to achieve optimal energy conversion and drive quality metrics. The control algorithm for the regenerative brake system is tested using the Powertrain System Analysis Toolkit (PSAT) [12]. A comparison between parallel and series brake strategies is made, showing the benefits of using the series regenerative brake system.

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Nomenclature

Ah	Amp-hour
BMCP	Battery mode control process
C	Aerodynamic drag factor
CIDI	Compression ignition direct injection
D_{aero}	Aerodynamic drag
D_x	Deceleration
EMCP	Energy management control process
ESS	Energy storage system
EV	Electric vehicle
F_b	Front and rear brake force combine
F_{mxf}	Maximum brake force on front axle before lock-up
F_{mxr}	Maximum brake force on rear axle before lock-up
F_{xf}	Front axle braking force
F_{xr}	Rear axle braking force
g	Gravitational acceleration
GATE	Graduate automotive technology education
HIL	Hardware in the Loop
HWFET	Highway fuel economy test
IC	Internal combustion
kW	Kilowatt
μp	Peak coefficient of friction
NIMH	Nickel metal hydride
PM	Particulate matter
PPS	Pedal position sensor
PSAT	Powertrain system analysis toolkit
RBCP	Regenerative brake control process
R_{xf}	Force due to rolling resistance on the front axle
R_{xr}	Force due to rolling resistance on the rear axle
SD	Stopping distance
SIL	Software in the Loop
SOC	State of charge
Ts	Stopping time
TTR	Through the road
θ	Uphill grade
UDDS	Urban dynamometer driving schedule

UQM	Unique Mobility
V	Forward vehicle velocity
ω	Rotational velocity of the wheel
VMCP	Vehicle mode control process
VTS	Vehicle technical specification
W	Vehicle weight
W_d	Dynamic load transfer
W_{fs}	Front axle static load
W_{rs}	Rear axle static load
X	Distance travel during the braking event

Chapter 1

Introduction

Background

The U.S. Department of Energy recognizes the University of Tennessee as a leader in advance automotive systems research as it was chosen to participate in establishing the Graduate Automotive Technology Education Center (GATE). The GATE Center is sponsored by the Department of Energy, and it provides graduate students opportunities to participate in advance vehicle systems research. The goal of the GATE center is to produce graduate level engineers with the knowledge and related experience prepared to contribute to the advancement of the transportation industry. The University of Tennessee's Gate Center focuses on hybrid system control development and advanced powertrain system integration. The work of this project is in direct support of the GATE center.

The basis for this thesis comes from the University of Tennessee's participation in an advanced vehicle design project called Challenge X. The Department of Energy through Argonne National Laboratory, along with General Motors, sponsors and manages the competition. Challenge X is a design competition involving seventeen North American universities. Its brings government, industry, and academia together to find new and innovative approaches to advanced vehicle design University teams are to redesign a 2005 Chevrolet Equinox to minimize the vehicle's energy consumption and environmental impact while maintaining performance, safety, and utility of the stock vehicle. The competition takes place over a three-year period. The competition uses

GM's global design process as a model for vehicle development. The GM model for vehicle design lays out a structured plan for vehicle development that starts with defining the overall purpose of a vehicle. Vehicle definition leads to the development of computer models of vehicle components and various vehicle models. Designers use vehicle simulation software to predict a variety of performance metrics for the respective vehicle models. The results from the vehicle simulations allow designers to determine which combination of vehicle components and vehicle architecture meets the design specifications. Computer simulation streamlines the vehicle design process. The effects to vehicle performance due to changes in vehicle components can be analyzed before physical changes are made. The goal of vehicle modeling is to move from the trial and error type of design to a more efficient process that will decrease the development time of the vehicle and ultimately save money. The simulation process continues throughout all stages of vehicle development. With the components and vehicle architecture chosen, hardware tests of components are completed to verify simulation results, assure the functionality, and establish hardware communication. After component testing is completed, construction of the vehicle begins. Once the vehicle is built, on-road testing of the vehicle takes place. Just as in component testing, vehicle models are refined with data from on road testing when noticeable differences are observed. Vehicle testing and refinement continue until performance specifications are satisfied. The final stages of the GM vehicle development process focuses on resolving drive quality issues in the vehicle. This final stage includes vibration analysis of various components in the vehicle directly affecting the consumers experience while driving the vehicle. Drive quality issues from steering wheel vibration to engine vibration to vehicle operating mode transition

harshness contribute to the overall drive quality of the vehicle. These and other drive quality issues must be resolved before the vehicle is ready for production. A simplified version of the GM Global design process will be applied over the entire three-year Challenge X competition. Each year of the competition marks a transitional milestone in the development process.

During the first year of development, teams evaluate potential vehicle configurations and the available component technologies through literature review and vehicle simulation. The Powertrain System Analysis Toolkit (PSAT), a computer based vehicle simulation package developed by the Department of Energy through Argonne National Laboratory, is used to simulate candidate vehicle configurations. PSAT allows the team to rapidly evaluate vehicle performance. Three-dimensional CAD modeling tools are used to model vehicle components, which provide packaging and mounting information. Limited subsystem testing can occur through software in the loop (SIL) testing. The goal of year-one development is to chose a vehicle architecture, have a control strategy in place, and to develop an implementation plan for year-two.

Year-two development challenges teams to execute the plan from year one. Hardware arrives, and testing begins. Mounting systems are fabricated, and the vehicle is built. The vehicle model is continually refined as data is generated from hardware tests. Simulation of the vehicle continues throughout the vehicle development process and is used not only as a performance prediction tool, but also as an aid for design decisions. Many times component availability changes due to factors such as production holds on components or component discontinues. When such problems arise, the time it takes to search for a replacement can have major effects to the implementation schedule.

Simulation tools provide a quick way to quantify how performance is impacted by changing components. The goal for year-two development is to have an operational mule vehicle reliable enough to begin road testing and controller calibration.

Year-three is the final stage vehicle development. Focus is on resolving drive quality issues and vehicle efficiency improvements through control system refinement. On road tests continue on the vehicle to investigate noise, vibration, and harness problems as the vehicle transitions between operation modes. All changes to the vehicle's subsystems should be complete and the various control modules should be mounted as if in a production vehicle.

The University of Tennessee interprets the GM Global design process with the system engineering approach. Figure 1 illustrates the systems engineering approach. The development of the system design is an iterative process between simulation and testing. It provides progressive feedback to each stage of development. The process begins on the left side of the system "V" diagram by defining the goals of the competition. The team leverages all of its resources to develop different vehicle configurations having the potential to meet performance expectations. The configurations are modeled and computer simulation provides performance information about each. The vehicle architecture is chosen by determining which candidate provides the best balance between performance, component technology availability, and implementation difficulty. SIL and HIL testing allows for the rapid development of the vehicle control system as well as establishing the integrity of system components. Once a certain system level and subsystem level designs have been completed, the design process begins to focus on the right hand side of the "V" process. During this stage of vehicle development, the

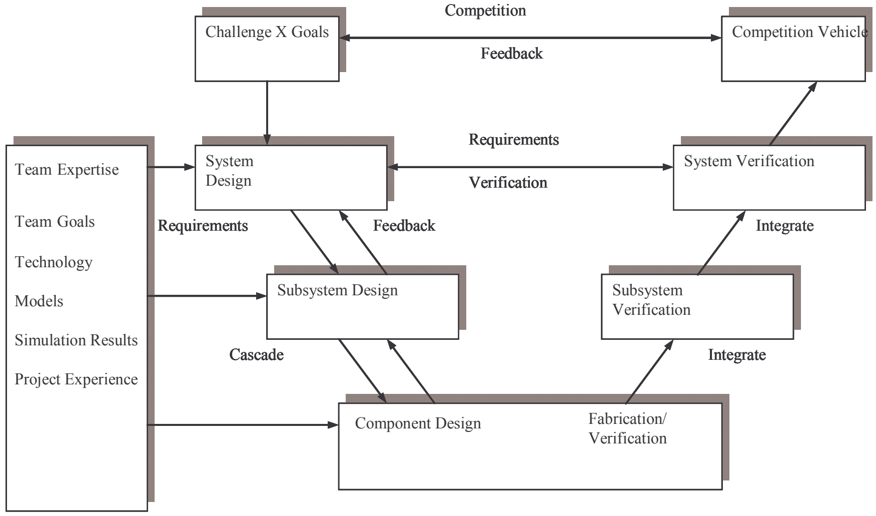


Figure 1 - Systems engineering approach to vehicle design

subsystems are integrated into the vehicle. Road testing of the vehicle begins. The results are used to verify the vehicle simulation results are accurate. The "V" cycle ends with the competition held at the end of each design year. The competition results are used to begin the design of the next year.

Purpose

The University of Tennessee has developed a hybrid electric vehicle platform to meet the performance goals of Challenge X. Chapter 2 will provide a detailed description of the vehicle's architecture and its control system. The vehicle in brief will use a small displacement compression ignition engine to power the front axle and an electric motor to provide traction power to the rear axle of the vehicle. A Nickel Metal Hydride battery pack will provide the energy storage system for the vehicle. The control strategy of the vehicle will focuse on effectively implementing a charge sustaining strategy with an emphasis on maintaining the integrity of the high voltage system and

battery pack. The performance of the vehicle relies largely on the control system's ability to efficiently interpret the input from the vehicle operator, manage energy flow through the vehicle, and to protect the battery pack. The appeal of a hybrid vehicle platform is that it gives the ability to take control of energy flow throughout the vehicle. Consequently, energy management becomes a large part of the overall vehicle control design. Conventional vehicles have a unidirectional energy flow that does not reuse energy consumed to propel the vehicle. The energy from the fuel never returns to the vehicle to be reused. The stored energy in the fuel converts to heat energy as it is burned in the engine. The engine converts the heat energy to rotational mechanical energy to propel the vehicle forward, imparting kinetic energy to the vehicle. A fraction of the energy used to propel the vehicle is consumed in the form of friction by powertrain components to transfer the mechanical energy of the engine. To complete the path, the vehicles brakes are used to convert the vehicle's kinetic energy into work which elevates the temperature of the brake components and ultimately transferred to the environment as heat as it is slowed. Though all of the energy from the fuel is consumed during vehicle operation, much of the energy is wasted. Energy not contributing to propelling the vehicle is considered wasted energy. Wasted energy is normally in the form of heat generated from the burning of fuel and heat generated during braking. Vehicle efficiency improvements focus on reducing the energy consumed during conversion and minimizing the energy wasted.

The hybrid approach to vehicle design attempts to reduce the energy consumption of a vehicle in two ways. The first is to minimize fuel energy consumption by reducing the size of the engine. The reduction of engine size also reduces the energy consumed by

reducing friction loss in engine. Logically, the reduction in engine size leads to a decrease in power availability for the vehicle. This directly affects the vehicle performance metrics such as towing capacity and acceleration. To offset this adverse effect, a secondary tractive-power source is used. In the case of the new hybrid Equinox platform, an electric drive is used to provide the additional power needed to offset the reduced engine power. Figure 2 illustrates the effects of engine downsizing and vehicle hybridization for acceleration and fuel economy [9]. This data was generated from vehicle simulation using PSAT. Both conventional and hybrid powertrains were simulated to show the effects of engine downsizing and hybridization on fuel economy and acceleration of a vehicle. The portion of Figure 2 labeled "Conventional" shows that reducing the engine size increases the fuel economy, but the acceleration of the vehicle suffers considerably.

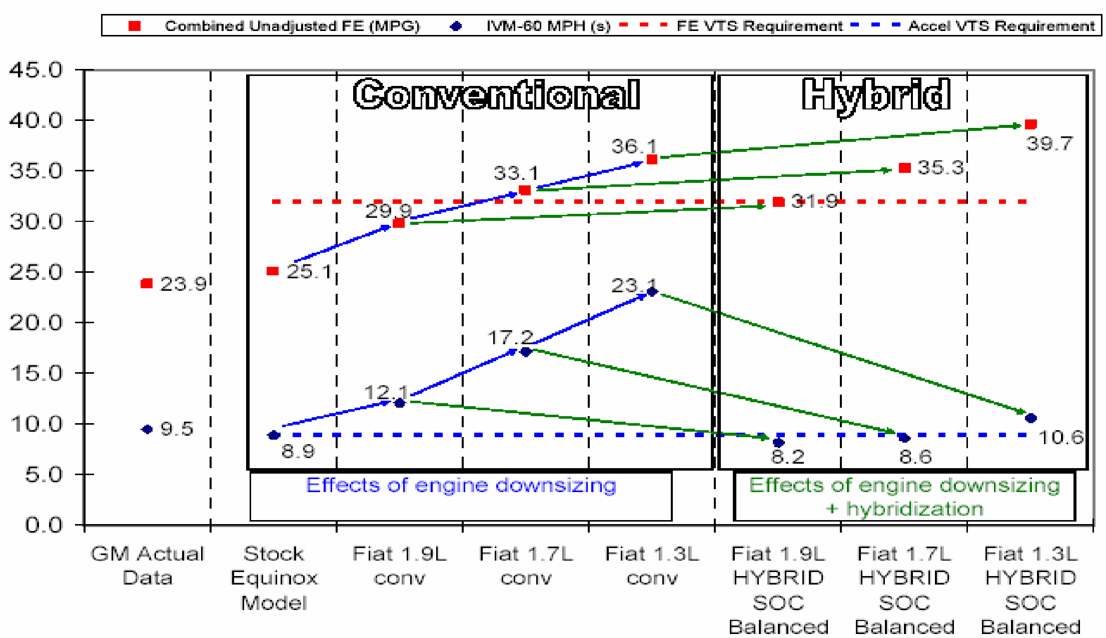


Figure 2 - Effects of vehicle hybridization

The hybridization of the vehicle, the addition of an electric-drive and ESS to the powertrain, brings the vehicle's acceleration to a reasonable level. A weight increase occurs when hybrid components are added to the vehicle's powertrain.

The second way to reduce energy consumption in the vehicle is to implement energy management strategies to take advantage of wasted energy during vehicle operation. Previously, areas of a vehicle's conventional energy cycle were examined to identify these areas of wasted energy. These areas included energy consumed during the transfer of mechanical energy due to inherent friction losses through powertrain components. Reducing the inherent friction losses in the powertrain such as the engine provides relatively small incremental increases in efficiency and energy consumption reduction. While friction reduction is important in reaching for higher efficiencies in the vehicle energy cycle, the energy wasted during braking events contribute to approximately 40 percent of the energy consumed over a typical driving cycle [1]. Energy lost due to braking is most seen in drive cycles simulating urban driving conditions, which include stop and go driving patterns. When a vehicle accelerates from rest and moves to the next stop, all of the kinetic energy produced by the engine and stored in the vehicle, is wasted during the braking event and never recovered. The use of a regenerative brake system (RBS) in a vehicle provides a method for recovering a large portion of this wasted energy. Conventional vehicles, of course, use friction brakes to provide the required braking to slow the vehicle. RBS systems use an energy storage system, electric or mechanical, coupled to the drive axles to apply the required braking torque. Instead of converting the kinetic energy of the vehicle into heat energy in conventional operation, which is lost to the environment; the RBS stores the kinetic

energy in some type of energy storage system (ESS). The energy in the ESS can be used later to propel the vehicle. Typically, for hybrid electric vehicle platforms, an electric generator provides the brake torque to the wheels, and a battery pack or ultra capacitor bank is used for energy storage. Energy can also be store as kinetic energy in flywheels. Using a flywheel as an energy storage device is not usually a feasible choice for energy storage due to system complexity versus the required energy storage capacity needed for long-range vehicles [7].

Hybrid vehicle architectures require the implementation of a regenerative brake system to meet the goal of minimizing energy consumption. The regenerative brake system is an essential part of the energy management strategy. The remainder of this document focuses on the design of a regenerative brake system to be used in UT's hybrid electric Equinox.

Statement of Problem

The hybrid platform for UT's version of the Hybrid Chevrolet Equinox is a Thru-the-Road (TTR) configuration. The IC engine provides power to the front axle, while and electric drive provides power to the rear axle. This configuration allows the high voltage system, including the electric drive, motor controller, and high voltage battery pack, to be isolated from the IC engine and its related components. This is advantageous to the overall system in that if one system experiences failure, the vehicle will continue to operate under limited power. The hybrid system employs many of the features discussed earlier. The engine's displacement is reduced from stock to improve fuel consumption. An electric drive is introduced to provide the supplemental traction power to the wheels,

and regenerative braking will be implemented to recapture the kinetic energy normally lost using conventional brake technology.

Typically, two versions of regenerative brake systems are implemented on hybrid electric vehicles. The first version is called parallel regenerative braking. This design couples the friction brakes and electric generator to provide the brake power to slow the vehicle. The other version is a series regenerative brake system. This type relies solely on the power of the electric generator to slow the vehicle up to a point. During a normal (not panic) braking event, energy is stored in the ESS until the charge limit of the battery pack is reached. At this point, the friction brakes are applied to complete the braking event. Series regenerative braking is the most efficient of the two system designs. The energy management strategy for the hybrid Equinox will employ a series regenerative brake system. Energy conversion and storage during braking will occur through the rear wheels. System isolation in the TTR does not allow the front axle to contribute to the electric braking process. Because single axle braking could possibility present vehicle handling problems during a braking event, the design of the single axle regenerative brake system will require much attention to ensure proper deceleration. Limitations on the operating envelope of the system must be determined to maximize energy conversion while providing provisions for safe and transparent system operation. In addition to the intrinsic system limitations set by the choice of powertrain configuration, competition rules limit the degree in which a brake system can interact with the existing vehicle brake system. The vehicle operator must have direct influence of the brake system during a braking event. This means a system controller cannot interpret the driver's input as a brake pressure to provide braking. Essentially, a dedicated brake-by-wire system cannot

be used in place of the conventional brake system to provide the sole braking power for the vehicle.

Scope

The purpose of the regenerative braking system is to maximize the energy recovery during a braking event. This thesis defines the overall system specifications including definition of the operating envelope and design limitations, development of the control system, and the mechanical design of the pedal assembly. Safety is the major design criteria for the implementation of the regenerative brake system. Due to the vehicle architecture, the regenerative brakes are only active on the rear axle of the vehicle. Safety measures in the system design do not allow the system to interfere with existing safety features in the stock brake system. For example, the regenerative system will be rendered inoperable when the anti-lock brake system is activated. When considering drivability issues such as vehicle stability during cornering, operating parameters such as vehicle speed and steering angle must be determined for which the regenerative brake system could operate safely. Calibration of the system will determine such metrics as the system is tested over a range of operating conditions. System calibration is not in the scope of this project. Rather, the goal of this project is to provide the system, ready for installation into the vehicle. The computer simulation process provides a baseline calibration. The baseline calibration is used as starting point for system optimization, which is defined as maximizing energy recovery during a braking event while maintain vehicle stability and component integrity.

Chapter 2

Hybrid Equinox Vehicle Architecture

Design Process Summary

The purpose for this chapter is to understand the overall vehicle architecture in which the regenerative brake system operates. The design philosophy used to evaluate candidate vehicle architectures was outlined in Chapter 1. This chapter summarizes the development and final design for Tennessee's vehicle entry for Challenge X 2006. The hybrid configuration, vehicle components, and control strategy are discussed. Several hybrid vehicle architectures and combination of components were considered during the design process. A list of selection criterion was chosen to evaluate different powertrains and configurations. Vehicle technical specifications (VTS), team expertise, cost, packaging and component availability were among the selection criterion.

Selection and sizing of vehicle components began with the development of the Vehicle Technical Specification (VTS). The metrics found in the VTS were determined by current consumer expectations and the performance goals selected by the design team. The VTS provides the framework for the architecture selection process. Table 1 outlines these specifications. A matrix of possible configuration and component combinations were developed. These combinations were modeled and vehicle simulations were performed using PSAT. Fuel economy, system efficiencies, emissions impact, control strategies, vehicle weight, and environmental impact were used to aid design decisions [9].

Table 1 - Vehicle technical specification

Description	Test Year	Modeling/Inspection /Testing	VTS(Team Selected)
IVM-60	1,2,3	M,T	$\leq 8.9\,\text{sec}$
50-70	1,2,3	M,T	$\leq 6.8\,\text{sec}$
Vehicle Mass	1,2,3	M,T	$\leq 4200lb$
MPG Combined (EPA)	1,2,3	M,T	$\geq 32.2mpg$
Highway Range	1,2,3	M,T	$\geq 200mi$
Passenger Capacity	1,2,3	M,T	5 passengers
Emissions Cert Level	2,3	T	Tier 2, Bin 5
Trailer Capacity	2,3	M,T	2500 lbs
Cargo Capacity	2,3	I	Approx. stock
Start Time	1,2,3	T	$< 5\,\text{sec}$
Noise Emission	1,2,3	T	$< 75dbA$

The first major design decision was determining which powertrain configuration to integrate into the Equinox. Both the series and power split configurations were eliminated from consideration based on high cost and integration complexity. The pre-transmission parallel configuration was dismissed due to reduced cycle efficiency through reduce regenerative brake capabilities and packaging constraints. Finally, the post-transmission parallel was removed from the list due to packaging constraints and feasibility. This configuration could be executed using wheel motors, but it falls prey to the high component cost.

The Through-the-Road parallel configuration is the only configuration that meets the design constraints set by the team. Though this configuration presents some vehicle integration complexities, it is viewed as an achievable alternative. This configuration provides increased cycle efficiency by providing the ability to use a series regenerative braking system instead a parallel system. The TTR also provides complete segregation of

the powertrain subsystems in the vehicle. Should the vehicle experience a loss of the high voltage system, the vehicle could continue to operate in a limited power mode. Subsystem segregation also provides more design flexibility in the design for packaging and control strategies. More of the free volume in and around the vehicle can be used for mounting powertrain components.

Next, a comprehensive list of possible vehicle components was developed. Both compression ignition and spark ignition technologies were considered for the primary energy converter in the vehicle. In addition, both DC permanent magnet and AC induction motors were considered for electrical energy conversion. The energy storage system would consist of either lead acid or nickel metal hydride battery chemistries. Fuel cells were not considered due to both cost and lack of team experience. This list of components was reduced by choosing configuration/component combinations that provided a power density similar to the stock vehicle, 0.076 kW/kg. Table 2 shows four basic engine/battery combinations that provide a power density comparable to the stock vehicle. The component list was further reduced based on literature research of current technologies and component availability.

Compression ignition engines were the primary focus in the component selection due to higher operating efficiencies and low speed torque characteristics. General Motors offered two low displacement CI engine choices for use in Challenge X, the GM 1.9 liter CIDI and the GM 1.3 liter CIDI.

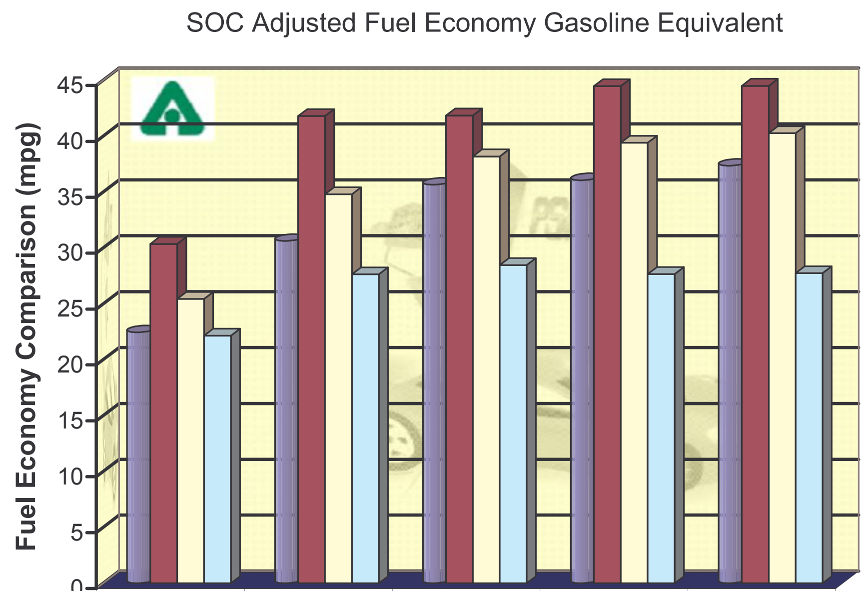
Table 2 - Power density matrix

Powertrain Component Size Combination	Vehicle Mass (kg)	Battery Peak Power (kW)	Engine Peak Power (kW)	Power Density (kW/kg)
Stock	1814	N/A	138	0.076
Small Engine (CIDI)/large battery	1904	60	51	0.058
Medium Engine (CIDI)/large battery	1966	60	75	0.067
Large Engine (CIDI)/small battery	1976	30	103	0.067
Large Engine (SI)/ small battery	1926	30	100	0.067

Both lead acid and nickel metal hydride (NiMH) battery technologies were considered. Lead acid batteries are a low cost and readily available solution to energy storage. The team has used this technology in the past with some success. The drawback to this battery type is the increased weight compared to technologies that provide a higher energy density. Charge and discharge rates also limit motor performance and overall vehicle efficiency. NiMH batteries packs are commercially available and are the choice for several production hybrid vehicles. The advantage of this type of battery is it provides high energy and power densities. Charging and discharging rates are normally adequate for a passenger vehicle, allowing the electric motor to operate in a broader range. The disadvantage of this technology is the high cost to purchase. In addition, the batteries are sensitive to temperature during operation; so more effort to cool the battery pack is needed. Cobasys made several commercial NiMH battery packs available for teams through a proposal process. Tennessee was awarded a battery pack. These packs are turn-key solutions for energy storage. So eliminating cost from the design decision, the team chose to use NiMH battery technology in the vehicle architecture.

The electric motor choice was reduced by choosing electric machines that were readily available to the team and determining which technology best fit the overall architecture. The two motors were available, a Unique Mobility (UQM) permanent magnet motor and a Ballard electric drive system. The team has successful experience with the UQM motor technology. Additional hardware would be needed to adapt the UQM motor to power the rear axle. Packaging constraints also reduce the feasibility of using this motor. On the other hand, the Ballard electric drive system is an electric motor integrated transaxle system and is, therefore, well suited to the TTR configuration as it can directly replace the rear differential to power the rear axle with little modification to the existing rear sub-frame. In addition, both the electric drive and the Cobasys battery pack communicate via CAN allowing a more efficient means of integration into the vehicle controller.

PSAT models of the vehicle component combinations for the TTR configuration were developed, and vehicle simulations were performed. Figure 4 shows the PSAT block diagram for the TTR configuration. The vehicles were exercised using standard driving cycles to determine fuel economy and performance results for each combination. The results from the simulations were used to determine if the VTS targets were satisfied and what design trade offs were required. Figure 3 shows fuel economy performance for a vehicle using a 1.9 liter and 1.3 liter CI engine with two different hybrid strategies for engine operation, 100 percent on and on/off during idle, compared to the stock vehicle.



	base_Equinox	hybrid_1.9L_ENG_ON_100%	hybrid_1.9L	hybrid_1.3L_HV_ENG_ON_100%	hybrid_1.3L
FUDS	22.44826345	30.6194581	35.64836333	36.05474083	37.34597121
FHDS	30.36161635	41.79998396	41.84249037	44.47869811	44.50670241
Combined	25.43	34.81	38.19	39.41	40.26
US06	22.14828602	27.63910364	28.4607153	27.66308918	27.74295156

Figure 3 - Fuel economy impact based on engine size and operating strategy

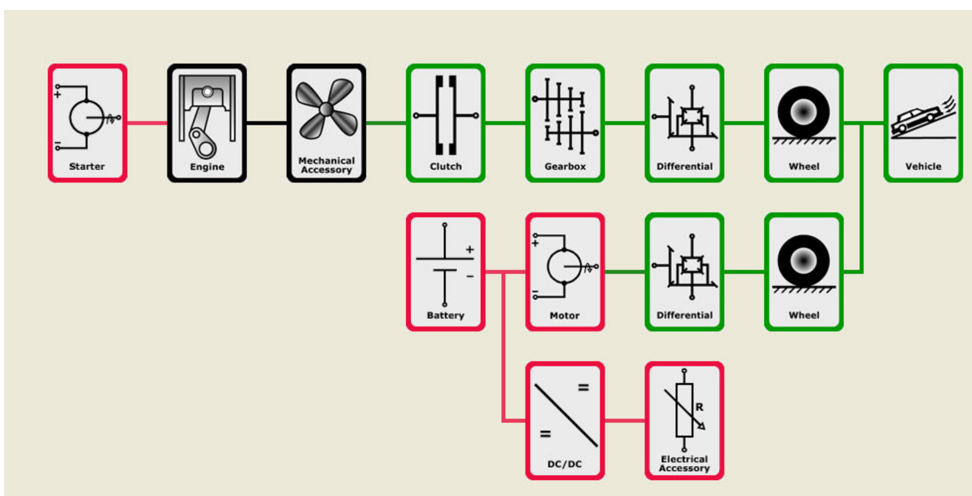


Figure 4 - PSAT thru-the-road block diagram

Final Vehicle Architecture Selection

The Challenge X vehicle platform is a charge sustaining thru-the-road parallel hybrid electric vehicle. A 90kW GM 1.9 liter CIDI engine will provide traction power to the front axle of the vehicle. A soot filter located downstream of the oxidation catalyst controls particulate matter (PM) in the exhaust. A NOx adsorber is planned for NOx reduction. The energy storage system is a Cobasys 336 volt NiMH battery pack with a peak power of 60kW and 9.5 Ah rating. The electric motor is a 65kW peak, 45kW continuous AC induction motor with an integrated transaxle originally used in the EV Ranger project.

Control System Overview

The goals of the control system are to translate the driver intent, maintain the SOC of the battery pack and to protect the high voltage ESS while providing autonomous operation of vehicle components. The vehicle controller must coordinate the interaction of the heat engine, electric motor, battery pack, and stock vehicle controller such as the Equinox Body controller and ABS module. The Vehicle controller is made up of four control processes needed to meet the system goals, Vehicle Mode Control Process (VMCP), Energy Management Control Process (EMCP), Regenerative Braking Control Process (RBCP), and the Battery Mode Control Process (BMCP). Figure 5 shows the block diagram for the vehicle controller.

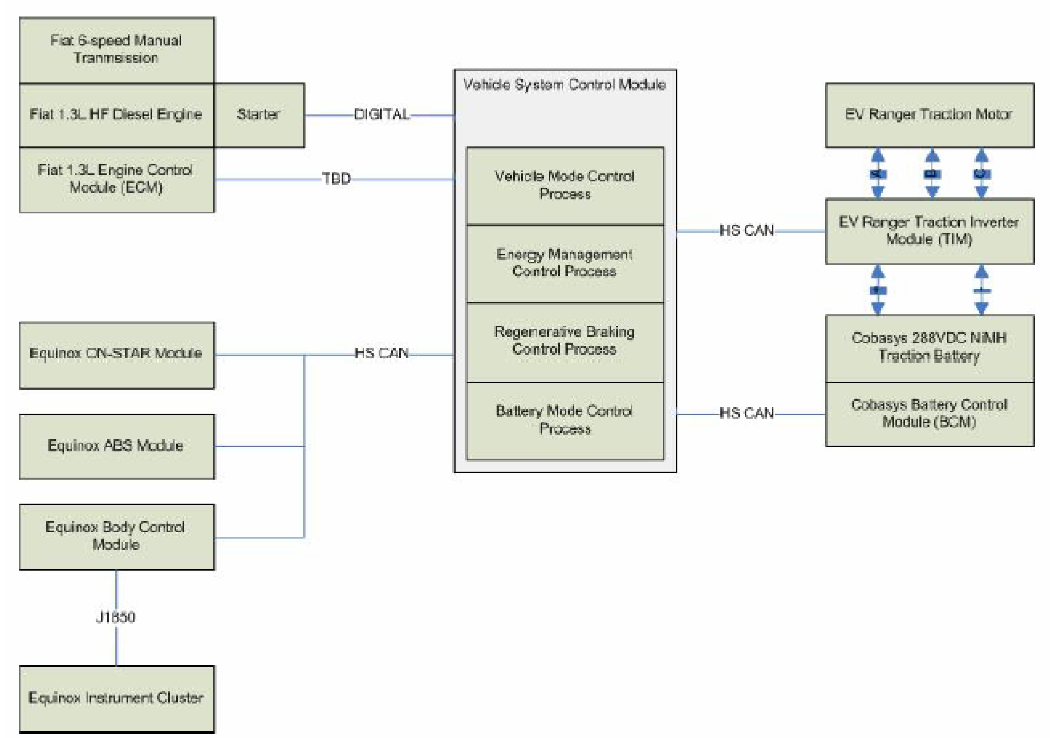


Figure 5 – Vehicle controller block diagram

The VMCP determines in which mode the overall vehicle should operate such as engine on/off operation. The TTR configuration does not allow idle charging of the high voltage system because the electric motor is only connected to the engine through the road. This feature simplifies the vehicle mode control since the engine can only be shutdown during vehicle idle periods and possibility during extended braking events.

The BMCP is tasked with protecting the battery pack by monitoring and providing updated information on state of charge (SOC) and temperature. Upper and lower limits for each parameter are used to limit the power demand the vehicle controller can ask of the battery pack. Battery information is sent to the EMCP for further processing. A calibratable table is used to determine the additional power needed from the engine to maintain SOC during vehicle cruising.

The RBCP is responsible for controlling the electric brake torque during braking events. The controller must monitor parameters such as wheel speed, brake pedal demand, and stock anti-lock brake system operation to safety slow the vehicle. Motor torque demand signals are sent to the EMCP for further processing. The RBCP controller is disabled if the ABS system is needed, to prevent interference with ABS operation.

The EMCP is the most important control process. It controls the interaction between the engine and the electric motor and is tasked with providing the power demanded from the driver, while maintaining the battery SOC. The module blends the energy, based on the output of the other control processes, to determine how to supply the power demanded by the driver.

Chapter 3

Regenerative Brake System Design

History of Regenerative Braking Systems

The concept of regenerative braking has a history of over 120 years. Its progress follows closely with the development of the hybrid electric vehicle and railway systems. The 1880's saw the development of electric locomotives. These machines received their power through a third rail [5]. The ability to control an electric motor between motoring and generating modes were well understood at their "commercial introduction [5]." This lead to the idea of converting the kinetic energy normally wasted through braking the train, into some other useful form. By 1885, Frank J. Spague received patents using regenerative braking in both railroad and automotive applications [5]. As power distribution networks developed in urban areas, more passenger vehicle such as trolleys and subway systems began to use electric power. Regenerative braking systems were implemented in many different ways. The electric energy generated by the train was sent to resistor banks located onboard the train where it was converted to heat. Fans then circulated the heat throughout the train's cars. Later it was discovered that the electric energy could be put back in to the power grid from which the trains received their power. Non-electric regenerative systems that recovered the kinetic energy by storing compressed air through braking locomotive were also developed during this time.

Developments in electric automobiles were progressing along with electric railway systems. Electric automobiles were among the first passenger vehicles

developed. The lack of petroleum distribution networks and under developed roads lead designers to focus their efforts on the electric automobile. By the 1900's, electric vehicle engineering had developed a broad knowledge base in the design and implementation of many different regenerative brake systems. Electric and hybrid vehicles were in the market place and were largely more appealing than internal combustion (IC) vehicles in the early 1900's. Woods dual power automobile introduced in 1917, used both a gasoline engine and an electric motor for propulsion, and it utilized regenerative braking [5]. By the 1920's, the IC powered vehicle had most everything it needed to succeed in the automotive market. The petroleum network was growing rapidly, making it readily available to consumers at inexpensive prices. IC vehicles were also out-performing the electric vehicle in both range and power. Though the electric vehicle had lost its appeal to consumers, electric railroad systems continued to be developed. The railroad industry holds approximately two-thirds of all the patents for regenerative brake systems. This trend, which lasted nearly 50 years, shows the shift away from electric vehicle development.

It was not until the U.S. government recognized motor vehicles were a major contributor to the increasing air pollution problem did electric and hybrid vehicle development begin to emerge again. In the 1960's, engineers began to revisit the advantages of the electric car in the reduction air pollution. The fuel crisis in the 1970's again urged the government to focus even closer at electric and hybrid electric vehicle. The 94th US Congress passed the *Electric and Hybrid Vehicle Research, Development, and Demonstration Act of 1976* [5]. The act called for the assessment of the "overall effectiveness and feasibility of including regenerative braking systems on electric and

other automobiles in order to recover energy." Researchers found that electric regenerative brake systems could be included in electric and hybrid electric vehicle platforms for a nominal amount of money. Regenerative braking in electric and hybrid vehicles is an inherent fit. There are no additional pieces of equipment needed for its integration into the vehicle. Ultimately, it was found that the use of regenerative brake systems were an effective method of recovering kinetic energy by both increasing the range and overall vehicle efficiency. At the time, system limitations such as inadequate battery technology and heavy, inefficient system controllers caused the interest in hybrid vehicle development to decline as the fuel crisis ended [7].

Hybrid vehicle component technologies have greatly progressed through the last 30 years allowing the shortcomings of past hybrid vehicle designs to be realized. The performance of today's hybrids is comparable to their still favored IC competitors. While hybrid vehicles are readily available, sales suffer from consumer fear and lack of trust in the reliability of these vehicles. Hybrids must prove to last as long as conventional vehicles to be considered competitive. As in the 70's during the fuel crisis, the price of petroleum is at an all time high. Recently the price of fuel spiked to over $3.00 per gallon and with a rising national average of $2.30, the public has recognized the need for an alternative to the conventional automobile.

The automotive industry is not the only industry faced with the need to reduce energy consumption. The railroad industry is beginning to see the potential and need for more powerful, more efficient and cleaner operating train engines. Trains transport goods from coastal ports all over the US. Trains transport nearly 42 percent of the freight in the US. Trains out perform freight trucks requiring only a third amount of the amount

of fuel required to move the same amount of cargo. Freight trains are series hybrid vehicles without an energy storage system. A large diesel engine drives a generator to provide electric current to electric motors that give traction power to the rails. As discussed earlier, the railroad industry holds close to two-thirds of the patents on regenerative braking. The applications of regenerative braking on non-electric trains, such as the large freight trains, have been restricted to bleeding off the electric energy as heat to a resistor bank. This is because non-electric trains do not have the ability to store the energy, and they are not connected to a grid. General Electric has developed a new 'hybrid' train that has the ability to capture the kinetic energy through braking the train [11]. The amount of energy produced through braking a large train carrying tons of cargo is enormous. In the past, a storage system was not available that could withstand such a large amount energy flow and be light enough to be a feasible choice for a store system. General Electric's new hybrid will be built from composite materials and will use a lightweight battery technology to implement the regenerative brake system. GE claims the new train will use 200,000 gallons less fuel over its life than conventional trains.

Brake System Fundamentals

Figure 6 shows the forces acting on a vehicle during braking. Newton's Second Law provides the governing equations for brake performance. For brake performance in the x-direction:

$$F_x = m \cdot a_x = -\frac{W}{g} \cdot D_x = -F_{xf} - F_{xr} - f \cdot W - D_{aero} - W \cdot \sin\theta \qquad (3.1)$$

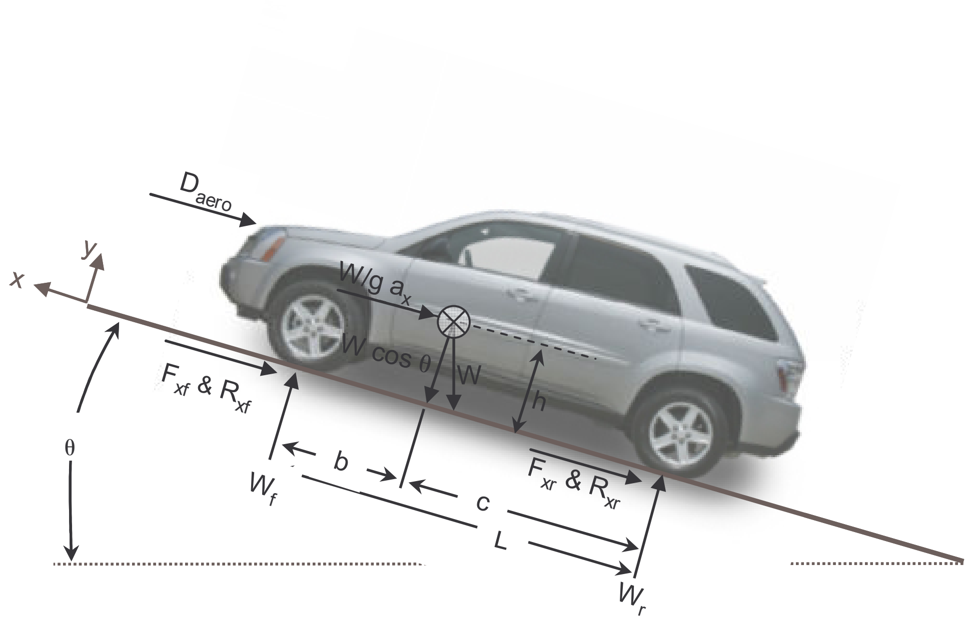


Figure 6 - Forces acting on the vehicle

Where:

W	Vehicle Weight
g	Graviational acceleration
D_x	$-a_x$=Linear Deceleration
F_{xf}	Front axle braking force
F_{xr}	Rear axle braking force
D_{aero}	Aerodynamic Drag
θ	Uphill grade
R_{xf}	Force due to rolling resistance on the front axle
Rxr	Force due to rolling resistance on the rear axle

For simplicity, assume all forces are constant during a braking event and the vehicle speed is slow enough to neglect aerodynamic forces on the vehicle, the following relationship can be derived:

$$D_x = \frac{F_{xt}}{m} = -\frac{dV}{dt} \tag{3.2}$$

Integrating equation 3.2 provides the relationship between stopping time and vehicle speed.

$$\int_{V_0}^{V_f} dV = -\frac{F_{xt}}{m} \cdot \int_0^{t_s} dt \rightarrow V_o - V_f = \frac{F_{xt}}{m} \cdot t_s \tag{3.3}$$

Furthermore, the relationship between velocity and stopping distance can be achieved using the relationship $V = dx/dt$ and substituting for dt in equation 3.3 giving:

$$\frac{V_o^2 - V_f^2}{2} = \frac{F_{xt}}{m} \cdot X \tag{3.4}$$

Where:
X distance travel during the braking event.

Assuming a vehicle comes to a complete stop during a braking event the stopping distance and the time to complete the event is:

$$SD = \frac{V_o^2}{2 \cdot \frac{F_{xt}}{m}} = \frac{V_o^2}{2 \cdot D_x} \tag{3.5}$$

$$t_s = \frac{V_o}{\frac{F_{xt}}{m}} = \frac{V_o}{D_x} \tag{3.6}$$

Assuming constant deceleration, stopping time is related directly to the velocity, but the distance is related to the square of the velocity. This means that if the speed is double the time to stop is doubled, but the distance is quadrupled.

The following equations take the effects of aerodynamic forces into account to determine the stopping distance. The coefficient of aerodynamic drag is determined empirically for each vehicle type.

$$\sum F_x = F_b + C \cdot V^2 \tag{3.7}$$

Where:
F_b Front and Rear brake force combine
C Aerodynamic drag factor

Therefore:

$$\int_0^{SD} dx = m \cdot \int_{V_o}^{0} \frac{VdV}{F_b + C \cdot V^2} \tag{3.8}$$

Integrating the above provides the stopping distance including the effect of aerodynamic drag on the vehicle.

$$SD = \frac{m}{2 \cdot C} \cdot Ln\left[\frac{F_b + C \cdot V_o^{\;2}}{F_b}\right] \tag{3.9}$$

Calculation of brake energy and power provide insight into determining the performance required from a brake system [9]. The kinetic energy absorbed during braking is dependent on the mass of the vehicle and the change in speed during a brake event.

$$Brake\_Energy = \frac{m}{2} \cdot \left(V_o^{\;2} - V_f^{\;2}\right) \tag{3.10}$$

In addition, the average power of a brake event with a final vehicle velocity of zero is given as:

$$Brake\_Power = \frac{m}{2} \cdot \frac{V_o^{\;2}}{t_s} \tag{3.11}$$

Integration of equation 3.11 using the negative power generated during a vehicle drive cycle provides the cumulative brake energy over that cycle.

There are a number of forces acting on the vehicle to resist forward motion other than the brake system. Some of these peripheral forces include rolling resistance, aerodynamic drag, driveline drag, and road grade. These forces are inherent properties in every vehicle and thus depend on a particular vehicle's design parameters such as weight, body style, and driveline type.

Rolling resistance constantly acts in opposition to the forward motion of the vehicle. The force of rolling resistance is related to the distributed weight of the vehicle and the coefficient of rolling resistance as:

$$R_{xf} + R_{xr} = f_r\left(W_f + W_r\right) = f_r \cdot W \qquad \textbf{(3.12)}$$

On average, rolling resistance forces provide approximately 0.01g of deceleration.

Aerodynamic drag forces described earlier are proportional to the square of the vehicle speed times the aerodynamic drag factor. The drag factor includes the dynamic gas relation dependent on air density, the coefficient of drag, which is determined experimentally, and the frontal area of the vehicle. When a vehicle is driven at normal highway speeds, drag forces average to approximately 0.03 g. However, at low vehicle speeds, the effects of aerodynamic forces can be neglected. The regenerative brake system recovers kinetic energy at vehicle speeds less than speeds seen on the highway, thus aerodynamic drag forces may be neglected when calculating the amount of energy absorbed by the regenerative brake system.

Braking forces due to driveline friction such as engine braking and friction forces in mechanical rotating components in the driveline can act to aid the braking process. In

some cases, such as low acceleration events, driveline braking may provide enough power to sufficiently slow the vehicle, but the driveline friction will not provide the adequate amount of power when the vehicle requires high deceleration, at which point, the foundation brake are applied. The regenerative brake system's effectiveness on capturing the kinetic energy may be adversely affected by the presence of such friction. In vehicles equipped with manual transmissions, as used in the Equinox, the brake torque provided by the engine is multiplied by the selected gear ratio. If the clutch is disengaged, the only energy losses due to the driveline would be through the axles and other rotating components connected to the wheels. In this case, the regenerative system would recover much more kinetic energy. On the other hand, vehicles using toque converter transmissions effectively transmit torque from the engine to the transmission, but the power transfer is not effective in the opposite direction. The engine's contribution to the braking of the vehicle is much less in vehicles using automatic transmissions. From the standpoint of the regenerative brake system performance, a vehicle using an automatic transmission is much more desirable to achieve a high-energy conversion.

The road grade braking force can act to be positive or negative on vehicle braking depending on the incline of the road. A negative grade is preferable when regenerative braking is used because the force of gravity rather than the engine is producing the kinetic energy. The braking force from road grade is given by:

$$R_g = W \sin\theta \tag{3.13}$$

Equation 3.14 gives the relationship of the torque acting on the axle to the braking force acting on the ground through the wheel.

$$F_b = \frac{T_b}{r} \tag{3.14}$$

If all of the wheels are rotating, the braking forces are found by using the relationship in 3.14. The braking force can be increased until the interaction of the friction limit of between the tire and the road is reached. At this point, the tire begins to slide along the road. This effect is known as wheel lock. Surface adhesion and bulk hysteresis both contribute to the friction between the tire and the road. Surface adhesion describes the level of contact the rubber of the tire has with respect to road surface. While bulk hysteresis describes the energy loss due to deformation of the tire as it slides over the road surface. Both of these tire characteristics contribute to determining the amount of friction available between the tire and the road. As the tire moves along the road, some amount of slip is always present as the elements move through the tire contact patch. The friction force, tire deformation, and the relative slip increase from the front to the rear of the contact path and is dependent on the amount of braking force applied. Slip is defined as the ratio of the difference of the forward vehicle velocity and rotational speed of the wheel to the forward vehicle velocity.

$$slip = \frac{V - \omega \cdot r}{V} \tag{3.15}$$

Where,
V forward vehicle velocity
ω rotational velocity of the wheel

In general, braking forces increase with slip up to approximately 10% – 20%. If the slip continues to increase beyond this amount, the maximum braking force available decreases until wheel lock occurs. If the wheels lock on an axle of the vehicle, the driver

loses some ability to control the vehicle. It is then desirable to control the brake force on both the front and rear axle such that they reach their respective locking point at the same time. Brake proportioning schemes have been developed to accomplish this.

Braking introduces a dynamic load transfer acting to increase the normal forces on the front wheels, while decreasing the normal force on the rear wheels, proportional to the deceleration of the vehicle as:

$$W_f = \frac{c}{L} \cdot W + \frac{h}{L} \cdot \frac{W}{g} \cdot D_x = W_{fs} + W_d \tag{3.16}$$

$$W_r = \frac{b}{L} \cdot W - \frac{h}{L} \cdot \frac{W}{g} \cdot D_x = W_{rs} - W_d \tag{3.17}$$

Where,

W_{fs} front axle static load

W_{rs} rear axle static load

W_d dynamic load factor

The maximum brake force for each axle is then given by:

$$F_{xmf} = \mu_p \cdot W_f = \mu_p \left(W_{fs} + \frac{h}{L} \cdot \frac{W}{g} \cdot D_x \right) \tag{3.18}$$

$$F_{xmr} = \mu_p \cdot W_r = \mu_p \left(W_{rs} - \frac{h}{L} \cdot \frac{W}{g} \cdot D_x \right) \tag{3.19}$$

where,

μp peak coefficient of friction

Figure 7 shows the maximum front and rear forces plotted against deceleration in g’s.

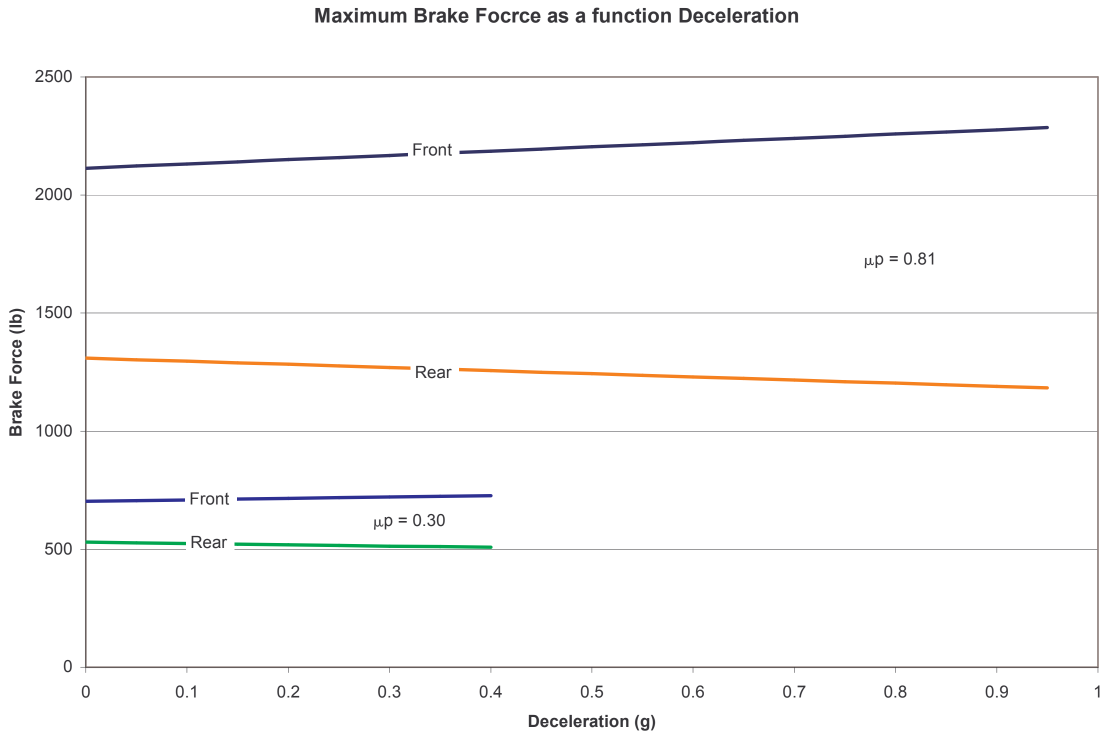


Figure 7 - Maximum brake force plotted as a function of deceleration

Explicit solutions for the maximum front and rear braking forces can be obtained by realizing that the deceleration is a function of the total braking force acting on the vehicle. The following relationships are derived to determine the maximum brake force each axle can experience before lock up occurs as a function of the brake force applied to the opposite axle.

For the front axle (in equation 3.18),

$$D_x = \frac{\left(F_{xmf} + F_{xr}\right)}{M} \tag{3.20}$$

For the rear axle (in equation 3.19),

$$D_x = \frac{\left(F_{xmr} + F_{xf}\right)}{M} \tag{3.21}$$

Substituting these relationships in the equations 3.18 and 3.19 gives explicit solutions for the front and rear maximum brake forces,

$$F_{xmf} = \frac{\mu_p\left(W_{fs} + \frac{h}{L}\cdot F_{xr}\right)}{1-\mu_p\cdot\frac{h}{L}} \tag{3.22}$$

$$F_{xmr} = \frac{\mu_p\left(W_{rs} - \frac{h}{L}\cdot F_{xf}\right)}{1+\mu_p\cdot\frac{h}{L}} \tag{3.23}$$

Brake lockup occurs when the brake force for either or both axles extends beyond the boundary lines form by the relationship in 3.22 and 3.23. Holding the force for either axle to zero provides the origin of the lockup boundary line for the other. As seen in Figure 8, the maximum brake force on the rear axle is less than the front axle. This trend in most cars is attributed to the dynamic load transfer and the weight distribution of the vehicle, which causes the loads on the rear to decrease thus reducing the normal force at the rear.

The maximum brake force that the regenerative brake system can apply is found by holding the front brake force to zero in equation 3.23 and solving for F_{xmr}. Some braking events will require additional torque from the foundation brakes to stop the vehicle. The torque from the foundation brakes at the rear is then added to that of the regenerative braking system. Figure 8 shows that as front braking force is added, the rear lockup limit is reduced. The RBS system must control the torque of the electric motor to not exceed this rear force limit.

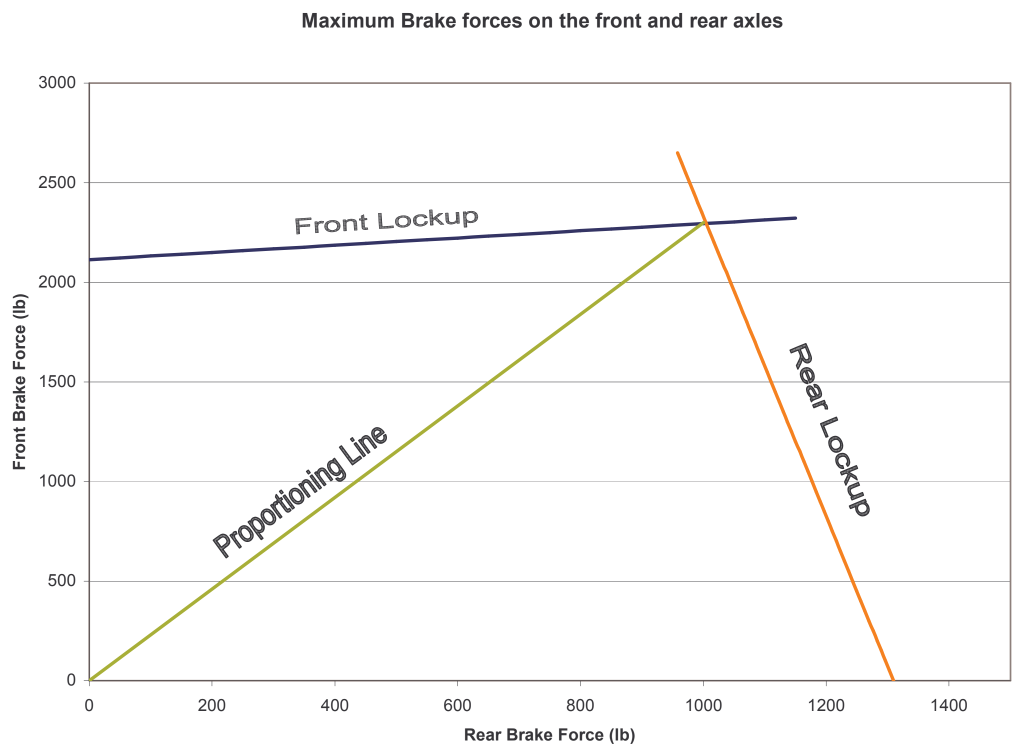


Figure 8 - Maximum brake force for front and rear axles

The foundation brakes on the Equinox utilizes both an anti-lock brake system to prevent wheel lockup during aggressive braking and front to rear proportioning schemes. The regenerative brake system acts only on the rear axle during the light braking events. Wheel lockup adversely affects both brake performance and the handling behavior of the vehicle. During wheel lock the tire does not generate the concerning forces needed to safety keep the vehicle on the road. If lockup is experienced on the front axle, the vehicle will continue in a straight line without the ability to control direction.

Rear wheel lockups tend to cause the vehicle to react to yaw forces putting the vehicle in a rotation [9]. This initial rotation causes the front wheel to turn in the direction of the rotation promoting the continuation of the rotation. If the rotation can not

be controlled the vehicle will not become stable again until the rear of the vehicle has complete one half of a full rotation, leaving the driver facing the opposition direction as when he started. Vehicle rotation due to rear wheel lock-up is regarded as the worse of the two wheel lock events. Therefore, wheel lock is an important parameter for the RBS controller to monitor.

Regenerative Brake Controller Design

The RBS system controller is part of the overall VSCM. The purpose of the RBS controller is to provide transparent interaction between the RBS and the stock brake system while safely converting the kinetic energy into stored electrical energy. The controller uses inputs from the vehicle controller and user-defined operating parameters to determine a torque request to the electric drive. While the goal of the overall vehicle controller is maintaining the integrity of the high voltage system and providing efficient vehicle operation, the RBS controller aims to maintain vehicle stability during braking. The controller is structured to monitor the status of vehicle parameters related to vehicle braking and to react to changes potentially causing harm to components or the driver. These parameters include vehicle speed, brake pedal position, and wheel lock. Each of these parameters and their respective control functions are discussed below.

Table 3 lists the input variables to the RBS controller. The UTK vehicle controller calculates vehicle speed using the front wheel speeds, and the value is used by the controller to limit system operation above a user-defined limit. The Equinox tracks wheel speeds for all four wheels and these values are transmitted on the CAN bus located in the vehicle. These signals are used by the RBS controller to calculate wheel slip on the

Table 3 - RBS controller input variables

Input Variable Name	description
vehicle_speed	Vehicle speed calculated from the front wheel speed sensors. Globally in VSCM (KPH)
pedal_input_voltage	Voltage being supplied to the pedal position sensor. The output of the sensor is scaled based on this signal.
pedal_sig	Pedal position sensor sends three signals to be used as redundant signals.
max_ available_torque	Maximum motor torque available during a braking event. This signal is provided by the vehicle controller
LR_wh_sp	Left Rear wheel speed is used to calculate wheel slip. Signal is provided by GMLAN bus.
RR_wh_sp	Right Rear wheel speed is used to calculate wheel slip . Signal is provided by GMLAN bus
LF_wh_sp	Left front wheel speed is used to calculate wheel slip . Signal is provided by GMLAN bus
RF_wh_sp	Right Front wheel speed is used to calculate wheel slip . Signal is provided by GMLAN bus
spd_thresh	Calibratable input from the user. Sets the max operating speed at which the controller will allow a regen request.
wh_slip_thresh	Target amount of wheel slip during wheel lock.

rear axle by comparing rear wheel rotation to front wheel rotation. The stock vehicle uses these signals for traction control and ABS braking.

Other inputs to the controller are signals used to interpret the output of the RBS brake pedal. The first input is the pedal reference voltage. This is the 5-volt signal supplied to the pedal position sensor (PPS). The PPS outputs three signals, one rising and two falling. The slope of these signals is based on a percentage of the reference voltage at zero sensor rotation and full rotation. The details of the PPS are discussed in the RBS pedal design section. The reference voltage is input to the controller to maintain the proper signal slopes in the event supply voltage fluctuations occur. Should a position signal lose voltage the controller uses the next available signal until there is no signal from the RBS pedal, at which point the RBS system is disabled.

The speed threshold and wheel slip threshold inputs are constant values used to set system operating limits. The speed at which the RBS system is engaged determines the amount of energy recovery the system can provide. On the other hand, the RBS system only acts on the rear axle and is therefore more prone to vehicle instability should wheel lock occur. Limiting the speed at which the RBS system is active limits the amount of energy converted and stored, and increases the chance for a driver to recover from a rear wheel only lock-up. As the RBS system is verified through in-vehicle test, the vehicle speed at which the system is allowed to convert energy will be increased. A twenty percent wheel slip is the average target for most brake systems to maintain during braking. Should the wheel slip rise above this point, the wheel is considered to be locked. Conventional ABS systems modulate the brake torque in 0.5 – 1 sec intervals to maintain a twenty percent slip. The wheel slip threshold input sets the target for the RBS

control to maintain during a locked wheel situation. Testing may show that a twenty percent wheel slip is not appropriate; therefore, the user can easily change this value to the desired amount. Initially, the wheel slip threshold is set to twenty percent.

Finally, the maximum available motor torque is a parameter calculated by the UTK hybrid vehicle controller and is available on the signal bus. The RBS controller attempts to provide the maximum amount of regenerative brake by demanding the maximum motor torque available with a full RBS pedal signal. If the controller detects wheel lock, the torque requested from the motor is reduced to unlock the wheel and provide the maximum amount of brake torque allowable.

Figure 9 shows the top-level Simulink model of the overall UTK vehicle controller RBS controller. At the top left of Figure 10, the input signals are put on a signal bus to simply the block diagram. These inputs were described above. The signals at the top left are constant value controller parameters such as the speed threshold and wheel slip threshold values. Additional parameters are also shown in the top level. These are unused parameter constants that may be added to the controller if needed in the future, and they include SOC and charge current threshold values pertaining to the high voltage system protection. Currently the BMCP has control authority for these parameters, but future RBS controller refinement may include them. Signal tagging is used to further simply signal routing within the controller schematic. Green tags are constant value user defined system parameters. Orange tags are wheel slip/lock signals calculated by the controller. Red tags are output control from the Stateflow logic controller. Finally, blue tags, used in the RBS torque command block represent signals supplied by the UTK vehicle controller. Discussion of each block in the controller will

be in order of propagating signal calculation starting with wheel slip and lock determination moving to the Stateflow logic block and finally the RBS torque command block. The top level of the controller is seen in Figure 9.

Wheel slip calculation and wheel lock determination are the main purposes of the RBS controller. It is not likely that the vehicle will experience wheel lock under normal operating conditions. If emergency braking is needed, the driver will be actuating the foundation brake and the stock anti-lock brake system will be engaged. However, it is possible the wheels will become locked during times at low vehicle speeds, where the electric motor has max torque, or if the friction coefficient is low. The wheel slip detection and the calculation of wheel slip and wheel lock determination diagram is shown in Figure 10. The inputs to the wheel slip block are the angular velocity of all four wheels and the wheel slip threshold. The outputs are percent wheel slip and a 0 or 1 wheel lock signal - 0 for unlocked and 1 for locked. The stock anti-lock brake system monitors all wheel speeds to prevent lock. The RBS system only prevents wheel lock on the rear wheel during electric braking. Wheel slip calculation compares the forward motion of the vehicle to the rotation of a particular wheel as in equation 3.15. The average rotation speed of the front wheels is calculated and used as the forward motion of the vehicle since no brake torque is applied to the front wheels that would cause lock-up. The wheel slip detection block uses the rear wheel angular speeds to calculate the percent difference in the wheels speed relative to each other to determine which wheel is locked as follows:

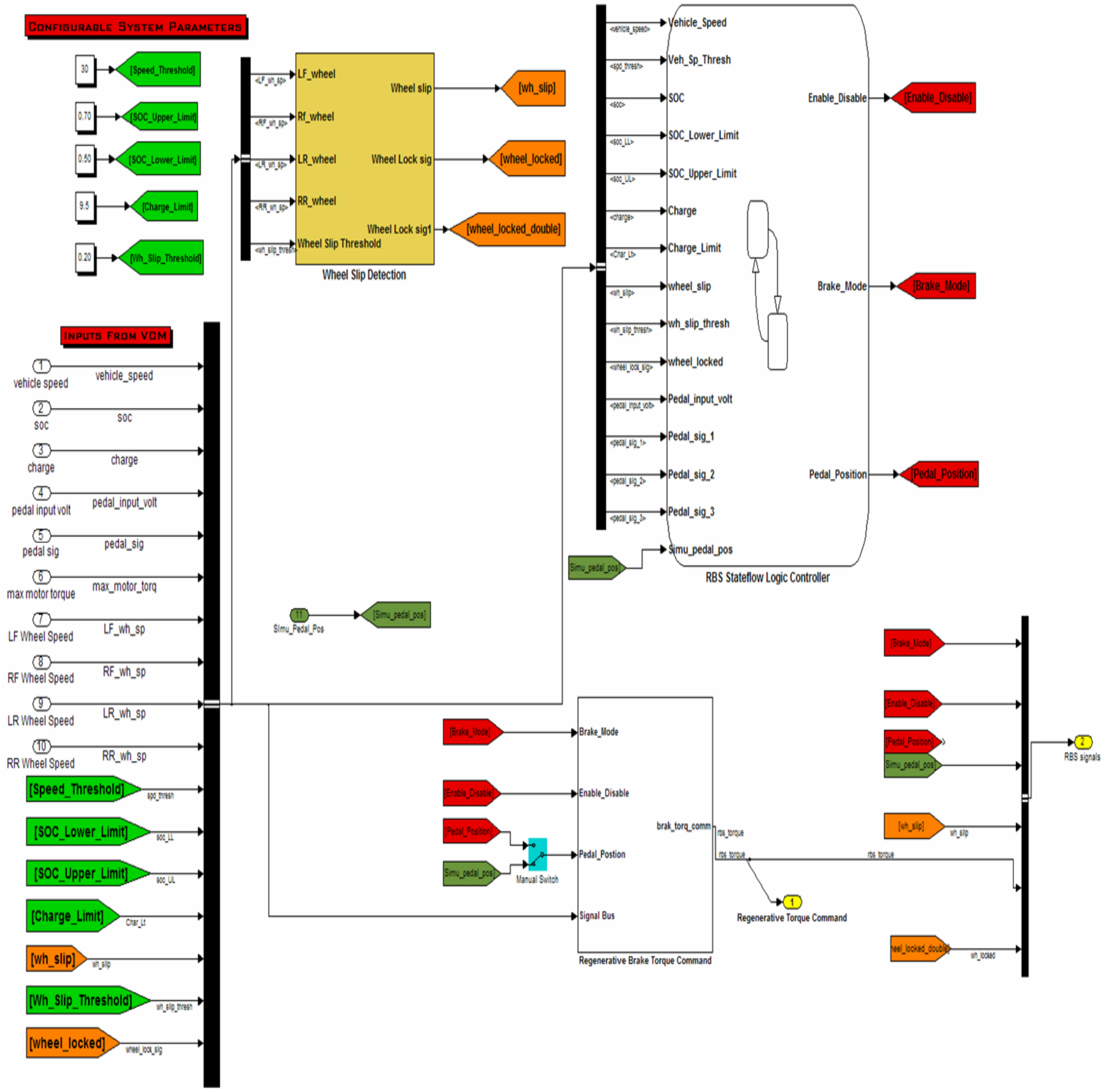


Figure 9 - Top level of the RBS controller

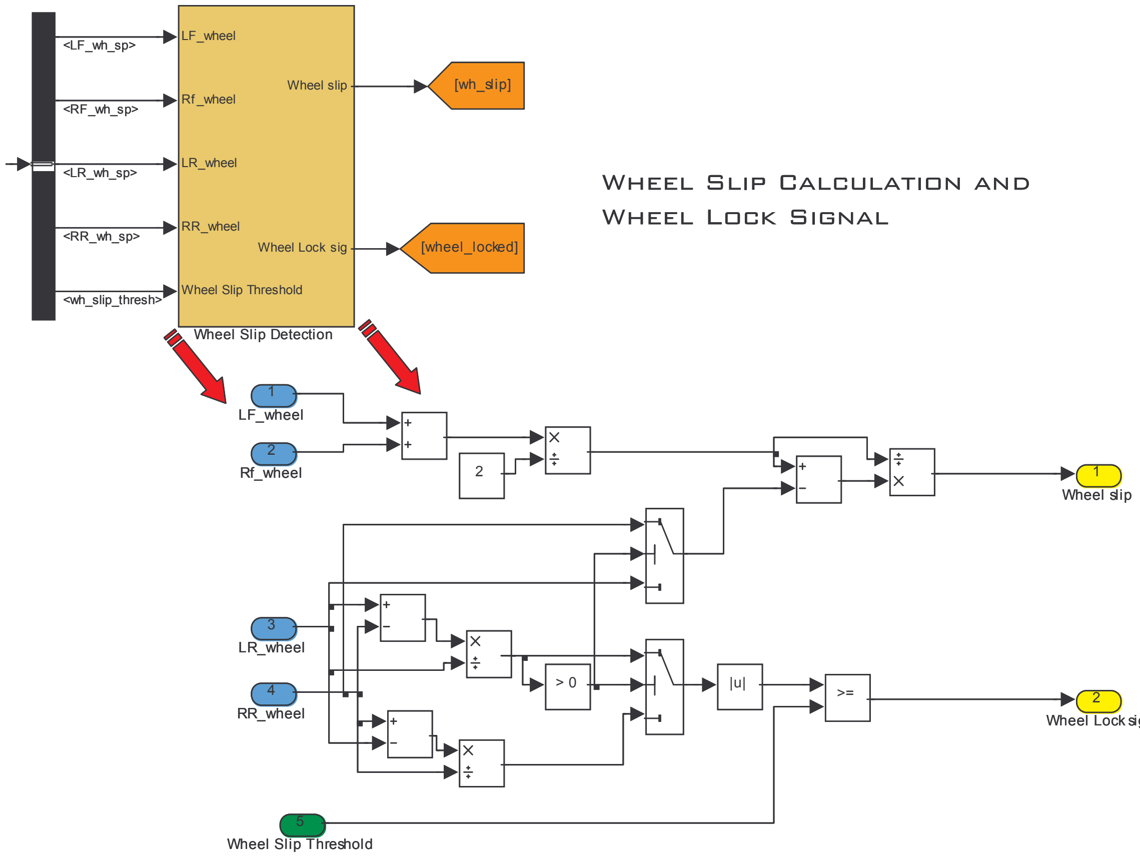


Figure 10 - Wheel slip block diagram

$$\frac{LR\_wheel - RR\_wheel}{LR\_wheel} \rightarrow \begin{bmatrix} positive & ? \\ negative & ? \end{bmatrix}$$

$$and$$

$$\frac{RR\_wheel - LR\_wheel}{RR\_wheel}$$

If the expression is positive, the right rear wheel is rotating slower than the left. The right rear is in a possible lock-up scenario and its angular velocity is used in the wheel slip calculation comparing it with the angular velocity of the front wheels. A negative value says the left rear wheel is rotating slower than the right wheel. In this case, the left rear wheel is in possible lock-up scenario and its angular velocity is compared to the front wheels. A compare to zero logic block and signal routing switch is used to choose which signal to be sent for wheel slip calculation. If wheel slip is greater than the threshold value, a high value is sent to the Stateflow logic controller where the system is then put into a limited operation mode. Wheel slip is used in the torque command block as the feedback signal for the PID control of the brake torque. Differences in wheels speeds during turning are neglected. It is assumed most braking events will be performed with little to no turning. If braking is needed during such conditions and the wheel speed difference is greater than the wheel speed difference limit, the RBS system will reduce the amount of electric brake available to ensure vehicle stability is maintained.

The RBS logic controller determines in which state the RBS system should be functioning. The logic controller was developed in Simulink/Stateflow. Stateflow uses a flow chart programming style allowing users to draw blocks representing different states of a system. Arrow lines represent state transitions. Conditions, similar to “while” and

“for” statements normally used in programming languages are added to each line allowing a transition to the next State when the condition is met. Actions such as calculations can take place in several ways, during a transition, entry into a new state block, throughout the duration of the state block, and upon exiting the block. Input and output variables to and from the Simulink environment, respectively, are set up to be used to provide condition variables to transition between states and to output data to be used by other parts of the controller.

The input variables to the logic controller come from the signal bus and have been described earlier. The three output signals are used in the torque command block. Brake_Mode is used to trigger torque control blocks related to a particular mode of operation of the RBS system. The application of this signal is described in the discussion of the toque command block. Pedal_sig is the scaled (0-1) pedal position signal. Scaling is calculated in the Pedal Signal Select state block in Stateflow. Finally, a system enable/disable signal is used as a system on/off switch. For full and limited modes of operation, the system is enabled and torque can be requested. On the other hand if the vehicle is operating outside system design specifications no torque will be requested.

The state chart, seen in Figure 11, is organized into five overall system state blocks, system parameter blocks and a system mode selection block. Dashed lines identify these blocks and they are numbered in order of execution. The order of execution determines how the block affects the safety and operation of the vehicle. The blocks at the top of the chart are parameter states. Each state monitors the activity of the respective system parameter. These parameters execute first because they determine if the vehicle is operating safely enough to request electric braking. These states also

determine in which mode the system should operate. The mode select block uses information from the parameter block to make decisions regarding which mode is appropriate for RBS controller operation. Each block will be discussed in order of execution.

The pedal signal selection block (1) is used to calculate a 0-1 scaled pedal position from the voltage output of the PPS. Three voltage signals are fed into the logic controller, one rising and two falling signals. sig_1 is the rising signal and sig_2 and sig_3 are the falling signals. The linear slope of each signal is based on a percentage of the reference voltage powering the sensor at zero sensor shaft rotation and full shaft rotation. Table 4 lists these values.

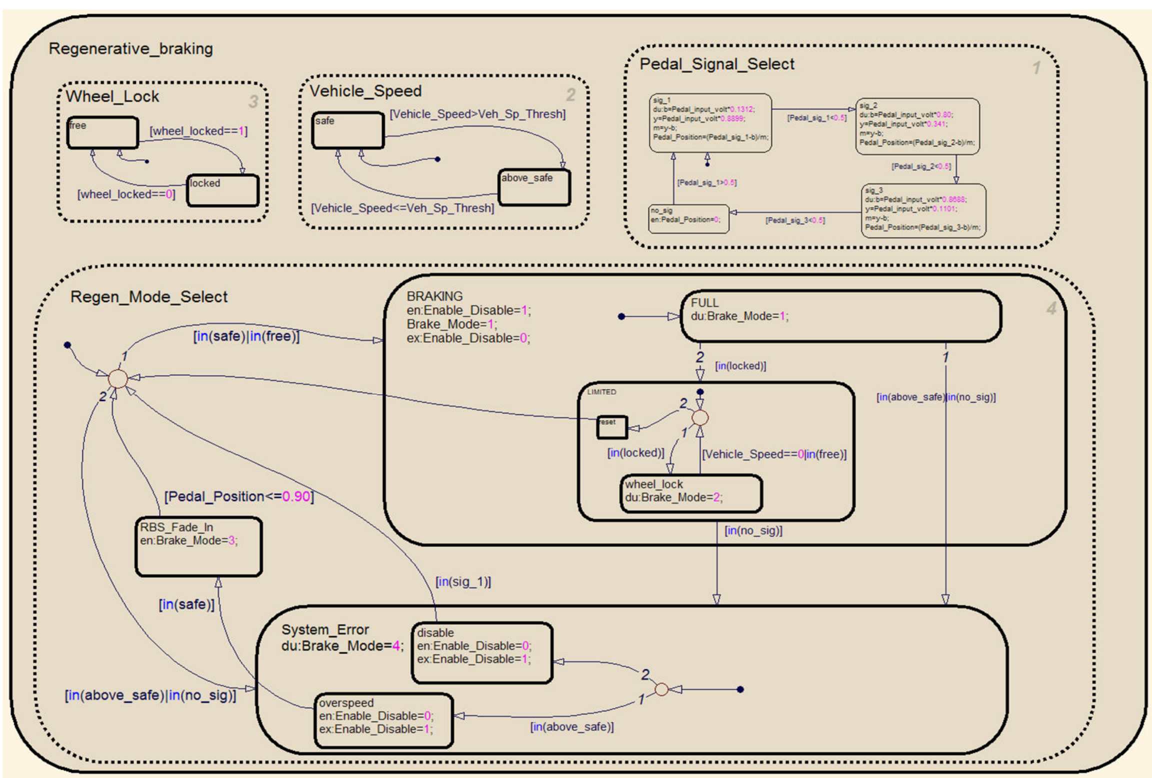


Figure 11 - Simulink/Stateflow RBS logic controller

Table 4 - PPS percent of reference voltage at zero and full shaft rotation

	Zero Degree Rotation % of Reference Voltage	Full Degree Rotation % of Reference Voltage
Sig_1	13.12	88.99
Sig_2	80	31.10
Sig_3	86.88	11.01

Scaling, based on the reference voltage, corrects for any fluctuation in the reference voltage being supplied to the sensor. The following equations are used to determine the 0-1 scaled signal.

$$b = Pedal\_input\_voltage \times Zero\_rotation\_\%\_Pedal\_input\_volt$$
$$y = Pedal\_input\_voltage \times Full\_rotation\_\%\_Pedal\_input\_volt$$
$$m = y - b$$
$$Pedal\_signal(0-1) = (Pedal\_sig\_1\_2\_3 - b)/m$$

The Pedal Signal Select block is structured using four blocks, one for each PPS output signal and a disabled state block. The code first looks at sig_1 to calculated a scaled pedal position signal. If this signal drops out, giving a voltage output below 0.5 volts, the controller transitions to the next block to calculate a scaled pedal signal. This process continues until there is no signal from the sensor. In this case, the RBS system will be disabled and no torque will be requested from the motor. The disabled block outputs a zero pedal position signal upon entry into the block. This triggers the Regen Mode block into the disable state to further ensure no torque request will be commanded. The RBS system is not activated until the signals are reset to their normal operating specifications. Each block calculates the scaled pedal position for each respective PPS output during the time it is active. The scaled pedal position is output from Stateflow to

be used in the Torque command block. The driver provides feedback to the system through the RBS brake pedal thus closing the loop on the control system.

The second parameter block is the vehicle speed block. This block is organized into two states, safe and above safe. Upon activation the block defaults to the safe state. The speed threshold parameter is used as the condition to transition to the above safe state block. If the speed of the vehicle is above the threshold value when braking begins, the system is disabled. Once the vehicle speed drops below the speed threshold the system becomes active.

Finally, the Wheel Lock parameter block monitors the wheel slip and determines if the vehicle is experiencing a wheel lock scenario. Like the vehicle speed block, there are two states the vehicle can be in when considering wheel lock, locked and unlocked. The default state is the unlocked position. Should wheel slip rise above the threshold value, the controller receives a logic high from the wheel slip detect block from Simulink. A transition is made from the unlocked state to the locked state. The Regen Mode Select block then moves into the wheel lock state.

There are many different ways to set up logic flow in Stateflow. The structure for the RBS controller is to use the state blocks within each parameter block as switches to trigger transitions within the Regen Mode Select block. The parameter blocks monitor the system activity while the Regen Mode Select uses the actions of the parameter block to decide how the RBS system should react.

An arrow with a dot on the end represents the default logic flow upon entry to a block. In the Regen Mode Select block, the logic flow starts at a logic node where the controller determines which path to take. Logic flow can only take one path at a time,

and two paths are available at this node. The first determines if the RBS system can be safely operated based on the state of the parameter blocks. If the Vehicle speed parameter is in the safe state and the Wheel lock parameter is in the unlocked state, the RBS system will move into the Braking block. Similarly, if the Vehicle speed parameter block is in the above safe block or the pedal signal block is in the no signal state, then the RBS controller goes directly to the System error state until the system is deemed safe to operate. The Braking Block and the System Error block are the two major states in the mode select block. Four modes of operation are controlled in these states. The Braking state controls modes one and two, and the System Error state controls modes three and four. Brake Mode is an output of a numeric value of 1-4 representing the four modes of braking. It is used in the torque command block to trigger the appropriate control blocks. Table 5 summarizes the brake modes.

Upon entry into the Braking state, Enable_Disable and Brake_Mode output variables are set to one. The FULL block is the default state within Braking. While, the controller is in this state the system is in mode 1, and the RBS system asks for the maximum regenerative brake torque available from the motor, limited only by the

Table 5 - RBS brake modes output description

Mode	Description
Brake_Mode (1)	Full – normal system operation, max regen requested
Brake_Mode (2)	Limited – wheel lock detected, anti-lock routine activated to maintain 20% slip
Brake_Mode (3)	RBS Fade In – activated when braking event begins above speed threshold and crosses below threshold
Brake_Mode (4)	System Disable – no regen requested, possible system failure

demand from the driver. Transitions from the FULL state occur if the Wheel Lock parameter block moves to the locked state or if the pedal signal select block moves to the no signal state. If wheel lock is detected the Braking state moves from FULL to LIMITED. When the LIMITED state is entered, a logic node is used to confirm a wheel lock scenario. If wheel lock is confirmed, the controller enters the wheel lock block where brake mode two is activated. The wheel lock state is executed until the wheel become unlocked or the vehicle comes a stop at which point the system is reset. Brake reset sends the logic flow back to the default node where the system decides again which general state to operate. Should the pedal position signal be lost during the wheel lock state, the system again is disabled. At this point the driver is expected to react and engage the foundation brakes via greater pedal pressure where the stock anti-lock system will take over should the wheels remained locked.

The System Error block is used to disable the system should a failure occur during system operation. Each block in the Regen Mode Select block has a pathway to the System Error block. Brake Mode 4 is assigned to this state. Two possible situations grant entry into the System Error state. The first case is the lost of the pedal position signal at any point during the operation. The system is not reset until the pedal signal is restored. Secondly, if the vehicle speed is above the speed threshold value at the beginning of a braking event, the Vehicle Speed parameter block will move into the above state. At that point, the controller enters the System Error block and is identified as violating the speed threshold by entering the over speed block. It is important to identify this type of violation because many braking events will begin above the speed threshold and have a long enough duration that the threshold is crossed. If the system

remains disabled, the system loses many opportunities to capture energy. Therefore, the controller allows the System Error State to be exited once the vehicle drops below the speed threshold. Upon exiting the controller moves to the RBS Fade In state. This state initiates Brake Mode 3.

The purpose of the RBS Fade In state is to gradually increase the electric brake torque as not to cause large unexpected changes in the deceleration of the vehicle. Remember the RBS Fade In block is only used when the vehicle at the beginning of an event is above the speed threshold and the RBS system is disabled. This means the friction brakes are being used to provide the braking and the RBS pedal signal is at 100 percent. If the controller moves directly to the FULL operating state when the speed threshold is crossed, the RBS controller would ask for the maximum amount of regen motor torque. The sudden torque addition would most likely lock the wheels and startle the driver, compromising vehicle safety. Therefore, the RBS Fade In increases the brake torque slowly until the driver reacts by releasing the pressure on the friction brakes allowing the electric brakes to provide the brake power desired. The controller only exits the RBS Fade In block when the driver has completely released the friction brakes and the RBS pedal is at 90 percent travel. When the controller has exited the fade in block, the system returns to the Regen Mode select default logic node.

The last portion of the controller is the RBS Torque Command Block. This subsystem of the overall RBS controller uses the logic output and system signals on the signal bus to send a brake torque request to the electric motor. This subsystem is organized in a similar fashion as the top level view of the controller. The input signals from Simulink and Stateflow are located on the left side of the diagram, Figure 12.

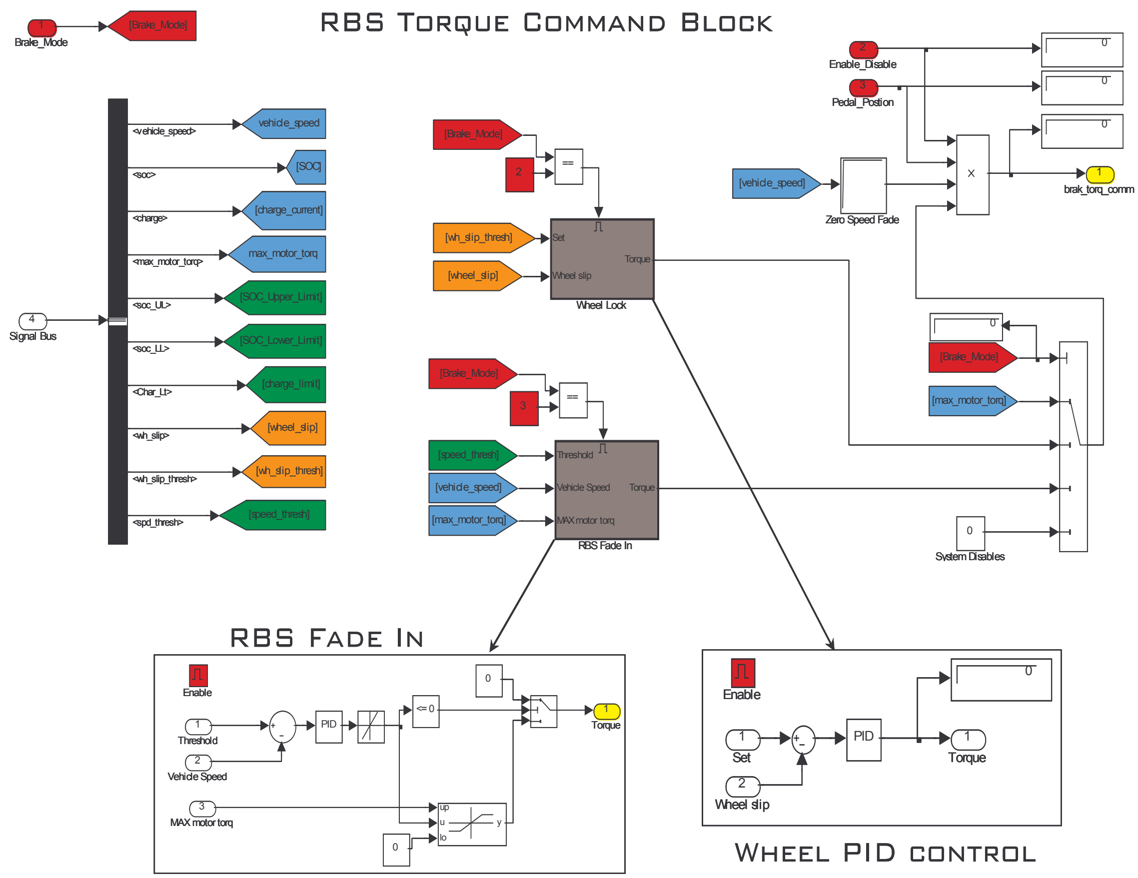


Figure 12 - RBS torque command block details

The control algorithm is executed in the center and the command torque is output on the right. The diagram is uses the Brake_Mode signal from Stateflow as a trigger to activated the appropriate set of torque control. A selector switch, located in the bottom right of Figure 12, is used to route the torque command signals. Input ports on the switch are numbered 1-4 from top to bottom and correspond to the respective brake mode number. Maximum available torque is input to port one. When the Stateflow controller is operating in the FULL regen mode, the brake mode signal is equal to one. The maximum torque available from the motor is commanded.

The PID control of wheel slip is located on port 2. Again, when the brake mode signal is “equal to” 2, this signal is passed through the switch. The wheel slip control subsystem uses the brake mode signal as a trigger to initiate the control calculation. An equal to logic block is used to as the trigger mechanism. When Brake_Mode equals 2, the logic block sends a logic high, and calculation begins. This triggering method saves processor time and bus space on the microprocessor because the code is only executed when needed, instead of calculating unused data and constantly dumping it off the processor bus. The wheel lock subsystem uses a PID control to adjust the torque command to maintain the wheel slip at the threshold value set by the user. The threshold value is the set point for the controller and the calculated wheel slip is the feedback signal.

Port 3 on the signal selection switch is dedicated to the RBS Fade In torque request subsystem. The RBS fade in uses the same trigger mechanism as the wheel lock subsystem. Brake_Mode 3 is associated with this block. The purpose of this subsystem is to gradually increase the electric brake torque command when the braking event is

started at a speed greater than the threshold speed and continues to below the threshold speed. The threshold speed is the set point and vehicle speed is the feedback signal. The PID control tries to correct of the error in the two signals. As the vehicle-speed moves further away from the set point, the commanded torque is increased in proportion to this difference. The PID control is tuned to response slowly to the error so the driver will not experience a sudden addition of brake torque and is reset once the vehicle stops. The torque will continue to build until the driver begins to retard the brake pedal, thus releasing the friction brakes at a point in which the RBS system can provide the brake power desired to slow the vehicle. A rate limiter block is used to further control the gradual addition of motor torque. This rate limiter can be used to quickly calibrate the system to the desired feel.

The last input to the signal selector switch is a constant block outputting a zero value. This port is, of course, activated when the logic controller is in the System Error state outputting a value of four for brake mode.

The torque command signal coming through the selector switch is further controlled by the brake pedal position signal and a zero speed fade curve. The pedal position signal multiplies the torque command giving the driver the ability to demand a percentage of the motor torque available up to the maximum torque available. Should the driver desire more brake torque than the system can provide, the friction brakes can be engaged to compensate. The zero speed fade curve is used to limit the percentage of brake torque as a function of vehicle speed. The electric motor is capable of providing its maximum torque at zero vehicle speed. This curve, which a Simulink look-up table, reduces the amount of torque that can be requested as the vehicle speed approaches zero.

This table can be calibrated to the desired feel through vehicle testing. Note that relaxing the curve to smooth operation serves to reduce the effectiveness of the RBS system.

Brake Pedal Assembly Modification

The RBS system requires two brake inputs to execute a series regenerative brake scheme. The stock brake pedal is used to actuate the friction brakes and a regen pedal controls the electric brakes. Though there are two brake systems in the vehicle, it is required that the driver only actuate one brake pedal, meaning that the driver should not have to push one pedal for the electric brake and another for the friction brake. Brake pedal actuation should not be different from what a driver normally experiences with the stock brakes. Thus, the modified pedal assembly should be design in such a way to provide conventional pedal feel. The driver will feel some difference in pedal actuation as the overall brake system is blending of two separate systems. The RBS controller reacts to system transients to provide a safe amount of brake torque, and there will be times that this torque is varied to control the vehicle. Comfortable pedal actuation depends largely on individual drivers. Therefore, the modified pedal assembly will allow adjustments to be made, such as adjustable pedal height and pressure, after the system is installed into the vehicle.

Actuation Strategy

The brake actuation strategy attempts to use the electric brakes first during a braking event by partitioning brake pedal travel into electric and friction brake actuation. The first few degrees of pedal travel are dedicated to actuating the RBS system. During

this part of pedal travel, a pedal position signal is sent to the RBS controller. If the desired deceleration cannot be met by a 100% pedal position signal from the RBS pedal, the driver can engage the foundation brakes by continuing pedal travel until the friction brakes provide additional brake torque. In the case in which the RBS controller is disabled, the driver will not receive any brake torque in the electric brake region of pedal travel. There will be a dead band of pedal travel in such cases until the friction brakes are engaged. Likewise, the brake torque in the electric region will vary depending on the operating conditions of vehicle components. For the most part, the controller attempts to provide a consistent torque pattern during operation so the driver can learn the feel of the brake system.

Brake Pedal Assembly Layout

Two pedal brackets enable system partitioning, the stock pedal bracket and the RBS pedal bracket. A pedal bracket is defined as the moment arm on which the pedal is attached. The stock pedal bracket is connected to the master cylinder through a push rod connected to a point below the pivot on the bracket. The stock bracket is located to the left side, front view, of the brake assembly bracket. Originally, it rotated on a pivot pin that spanned the width of the assembly bracket. This width provided adequate spacing to add another pedal bracket adjacent to the stock bracket without interfering with the travel of the master cylinder push rod. Figure 13 shows the stock brake pedal assembly. The RBS bracket was located adjacent to the stock bracket and pivots on the same centerline as the stock bracket. Two pivot pins, one for each respective pedal, replaced the original pivot pin. This provided free space between the two pedal brackets to mount the pedal

position sensor (PPS). Mounting the sensor on the outside of the assembly bracket could not be realized due to space constraints under the dashboard. A PPS mounting plate was installed to support the PPS and RBS pivot pin. The RBS bracket drives the PPS through the RBS pivot pin. An extension spring gives the RBS pedal a return force providing the ergonomic pedal resistance the driver expects.

Challenge X rules stipulate that a mechanical connect between the stock brake pedal and the foundation brake be maintained. Therefore, the RBS bracket makes a mechanical connection to the stock pedal bracket when it has reached 100 percent of its designated travel. This pedal travel is an adjustable distance. The brake pedal pad was removed from the stock bracket and placed on the RBS bracket. The stock bracket is engage once the back of the pedal pad meets with it. The original length of the bracket remained the same to maintain the proper leverage and actuation torque of the foundation brakes. The following section discusses the RBS assembly design.

Regenerative Brake Assembly

The RBS brake assembly is shown in Figure 14. Two pedals work together to provide transparent operation of the combined brake system. The brake pedal is located on the RBS bracket as the driver actuates the electric system first during a braking event. At the top of the assembly, the two pedal are separated by the PPS and the master cylinder push rod. The RBS bracket is bent to travel side by side with the Stock pedal. This allows the brake pedal pad to be located in the same position and size as the stock pedal. Space limitation and throttle pedal interference would not allow this configuration to be used.

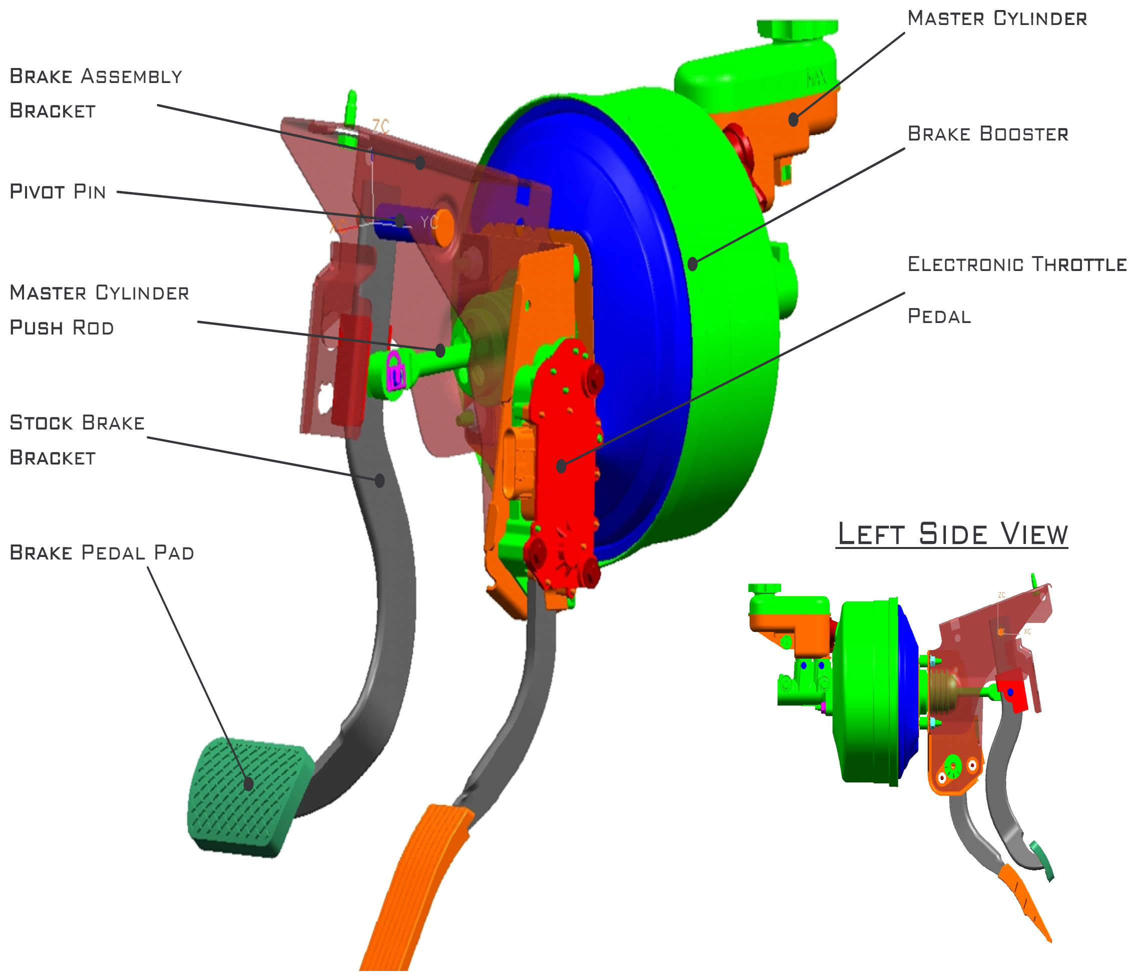


Figure 13 – Stock brake assembly

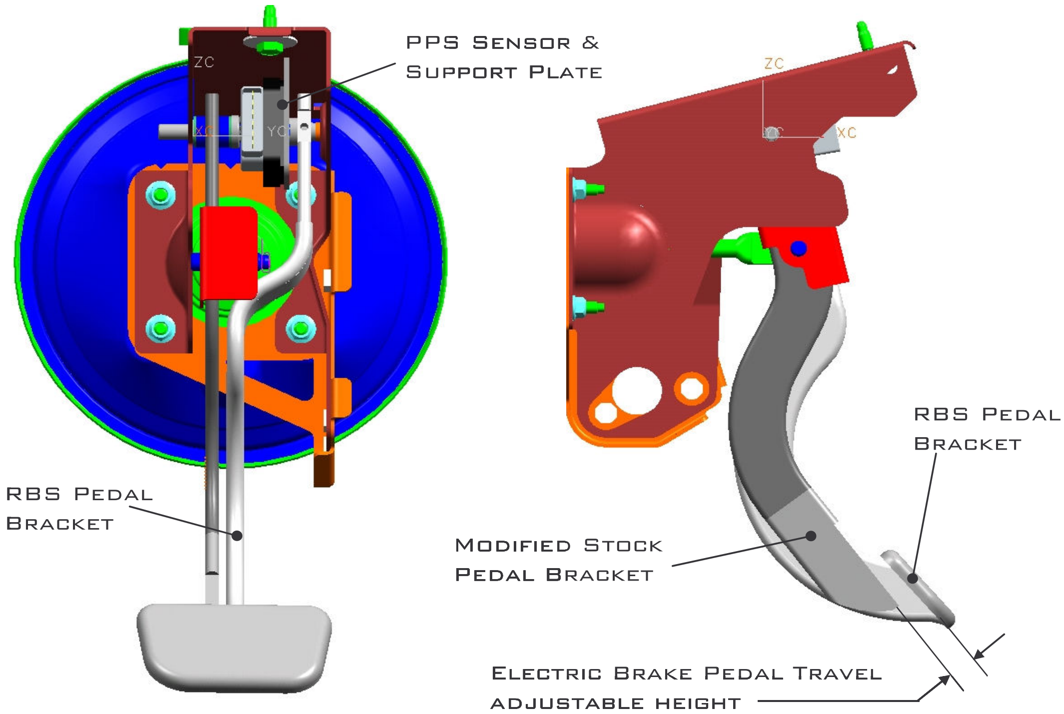


Figure 14 - RBS brake system

The driver is accustom to a pedal resistance proportional to acceleration. The National Highway Traffic Safety Administration (NHTSA) backed research in 1970 to determine optimal pedal force gain properties [3]. Figure 15 shows the results from this study. Emergency brake decelerations can reach up to 0.8g, but the RBS system will not be active in such a case. These situations occur over a short time so energy recovery would not be significant and are uncommon for normal drive cycles. The regenerative brake system has the capacity to absorb all the kinetic for most urban driving cycle where the deceleration is approximately 0.1g [2]. Using the results from the NHTSA with a deceleration of 0.1g, the RBS pedal should provide a pedal force gain between 0.02 and 0.009 g/lb. The RBS pedal provides this return force through an extension spring attached to a moment arm extending from the back of the RBS pedal bracket. It was

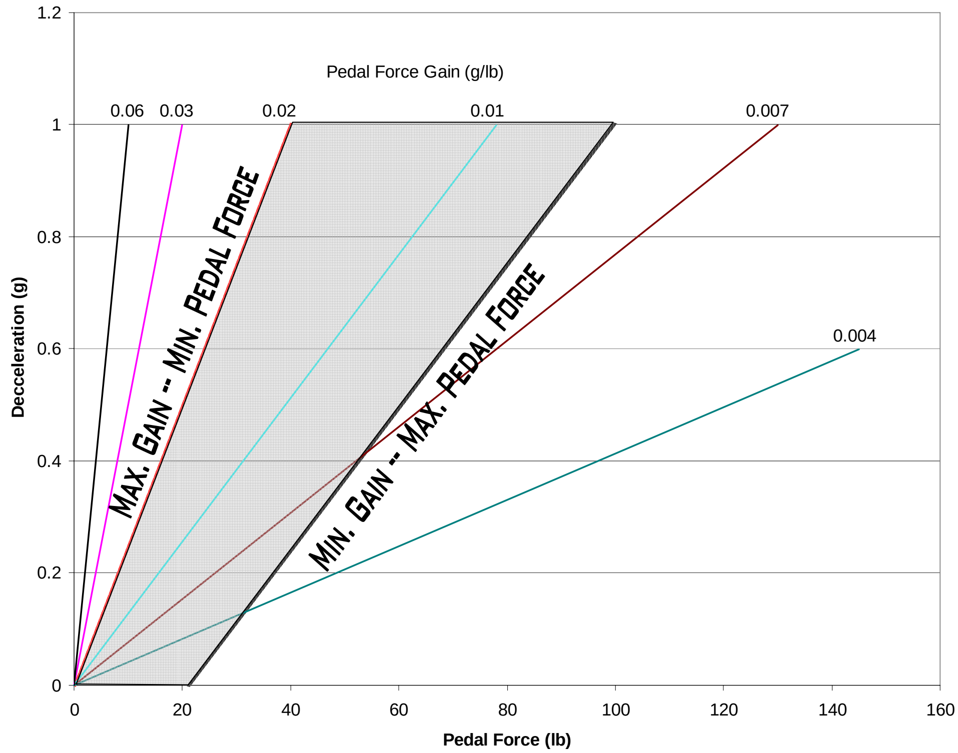


Figure 15 - Optimal pedal force gain

assumed the return force at the maximum pedal travel should be approximately 10lb. The PPS can rotate a total of 27 degrees. This provides the RBS pedal a travel of 5.5 inches from unloaded position until the pedal is nearly depressed to the floor. During normal operation the pedal travel does not go to the floor. The only time the pedal reaches the floor is during brake line bleeding procedures. During this procedure, the PPS actuation pin should be disengaged from the RBS bracket by loosening the lock screw. The pedal force gain of the two pedals is added together once contact is made. Note that the additional pedal force from the RBS pedal should not drastically change that of the stock system and should be adjustable once the brake assembly is installed in the vehicle.

The RBS pedal bracket is shown in Figure 16. Summing moments about the rotation point of the RBS bracket and assuming a 10lb pedal force at the maximum RBS pedal travel provides the appropriate spring constant for the extension spring.

$$\sum M = F_1 \cdot L_1 - F_2 \cdot L_2 = 0$$
$$\max\_moment\_arm\_displacement = y_{\max} = L_1 \cdot \sin 27^o = 1.81in$$
$$\max\_pedal\_travel = x_{\max} = L_2 \cdot \sin 27^o = 5.45in$$
$$F_1 = \frac{F_2 \cdot L_2}{L_1} = k \cdot y_{\max} \rightarrow k = \frac{F_2 \cdot L_2}{y_{\max} \cdot L_1} = \frac{10 \cdot 12}{1.81 \cdot 4} = 16.57 lb/in$$

The extension spring providing the return force must have a spring constant of 16.57 lb/in to provide 10lb of force at the pedal. This force provides the driver some feeling of deceleration in proportion to pedal travel. The extension spring is attached to the pedal at the end of the moment arm and to the assembly bracket from a mounting point on the lower side of the assembly. The travel adjustment pin allows the pedal height of the RBS pedal to be changed. When the pedal height is changed, the PPS output signal must be reset. This is done by adjusting the RBS pivot pin lock screw located on the RBS bracket. To change the height, i.e. electric pedal travel, the lock screw holding the RBS pivot pin should loosened, allowing the pedal bracket to freely rotate about the pivot pin. Once the desired pedal height is met, the lock screws can be tightened thus locking the bracket to the pivot pin and the travel adjustment pin can be set.

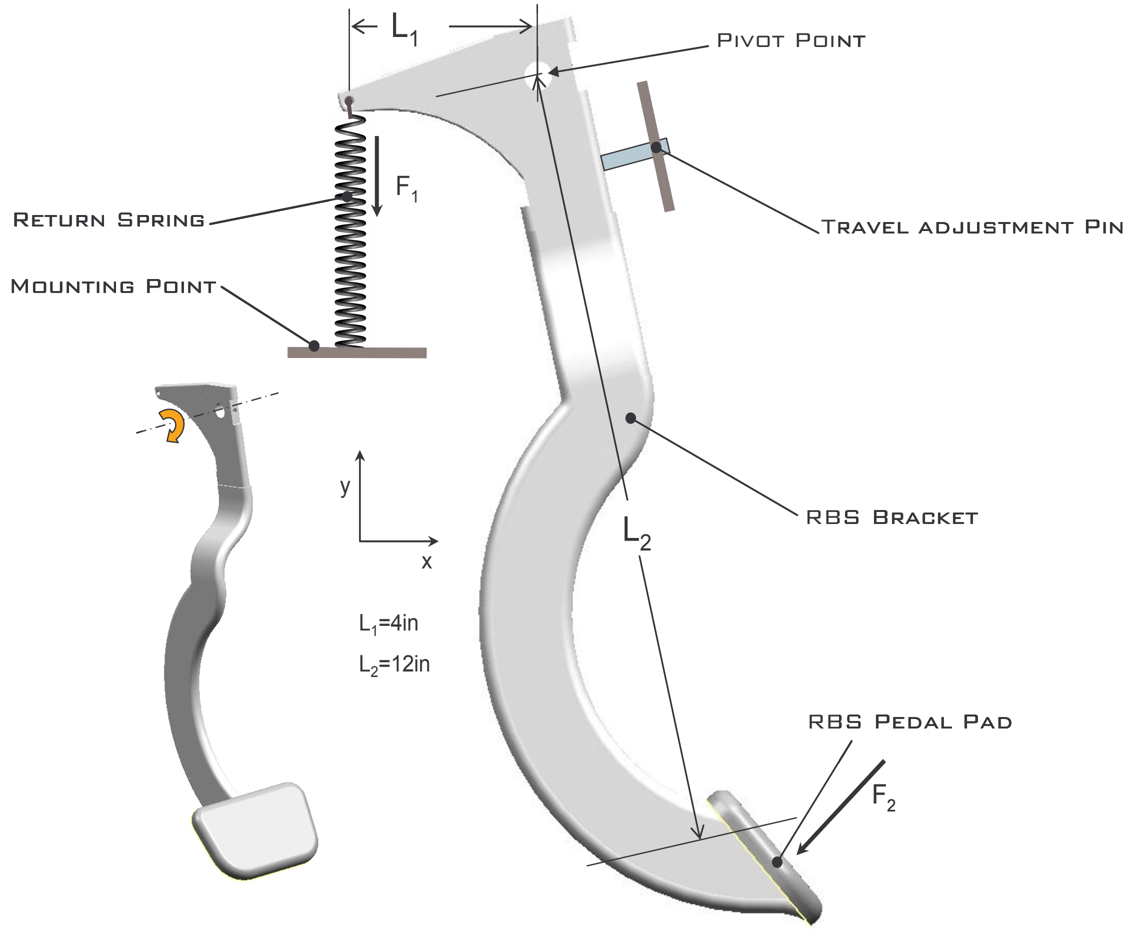


Figure 16 - RBS pedal bracket

Pedal Position Sensor and RBS Assembly

The pedal position sensor is a CTS model 503 series. This is a standard sensor generally found on electronic throttle pedals. CTS donated the sensor for use in the project. The sensor uses three potentiometers providing redundant pedal signals to the RBS controller. There is one rising signal and two falling signals. It operates on a 5-volt source and has a rotational sweep of 27 degrees. The RBS bracket actuates the PPS mounted on a sensor support plate attached to the assembly bracket between the stock bracket and RBS bracket. Figure 17 shows an exploded view of the RBS pedal assembly.

Under-dash space constraints did not allow the senor to be mounted on the outside of the brake assembly bracket. Therefore, the PPS support plate was added between the two pedal brackets to be used to mount the PPS and to support the PPS actuation pin. The actuation pin fits through the brake assembly bracket, RBS pedal bracket, PPS support plate, and centering plate. The end of the pin is machined to fit into the PPS. Setscrews in the RBS bracket allow the pin to rotate with the bracket thus actuating the PPS. Unlocking the set screws allow the RBS bracket to rotate freely on the actuation pin. There are nine pins on the sensor. Each potentiometer within the PPS requires a voltage high, low, and wiper signals. Figure 18 shows both the pedal and pin-outs for the PPS. The pedal position signal is scaled in the RBS controller to output a 0-1 signal based on the length of pedal travel, zero in the null position and one when the RBS pedal engages the stock bracket. The position signal is linear; therefore, the scale of the signal is based on the sensor output at the null position and at the full position.

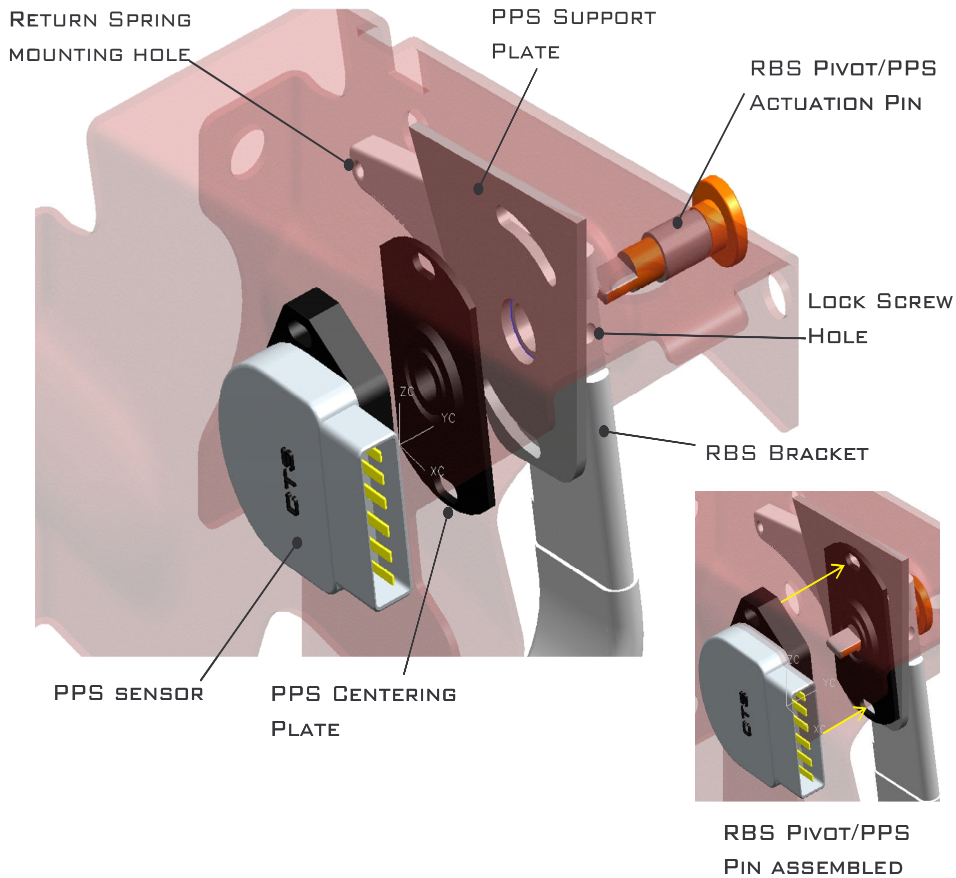


Figure 17 - RBS assembly exploded view

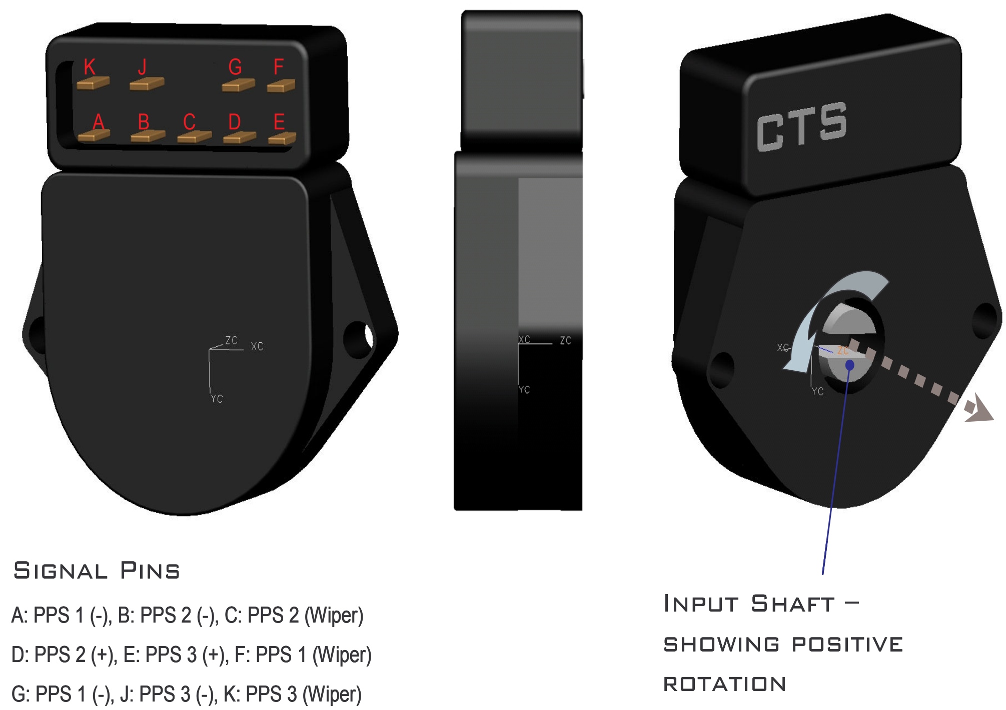


Figure 18 - Pedal position senor pinout

The functional travel of the RBS pedal is actually greater than the distance from the null to full positions because once the RBS pedal engages the stock bracket, the two move together as the friction brakes are actuated. The RBS pedal signal is saturated at the value of one so the torque command does not exceed the maximum torque available from the system.

Pedal Height Adjustment

The pedal height can be adjusted within the limits of the PPS sweep distance. The setscrews must be loosened to allow the RBS bracket to rotate about the PPS actuation pin, and the height adjustment pin, shown in Figure 16 should be set to the desired height. Then the lock screws can then be tightened to lock in the actuation pin so it rotates with

the RBS bracket. Finally, the PPS signal must be calibrated for the new travel distance. This is accomplished by reading the values of the three output signals in the null and full travel positions and changing their values in the Stateflow logic controller.

Chapter 4

Simulation of the RBS system

Introduction

Challenge X emphasizes the different roles that model based vehicle simulations play in the vehicle development process. During the June 2006 competition, a portion of the fuel economy event score is based on the team's ability to accurately predict the vehicle's fuel economy performance through model based simulations. Chapter 2 summarized the design process and final vehicle architecture selection process of the UT hybrid Equinox in which PSAT was used to evaluate candidate vehicle/component combinations. The results from the study provided information need to determine which combination of vehicle component would best meet the VTS requirements. Vehicle modeling and simulations also provide a means to evaluate different vehicle control and energy management strategies to quickly determine which methods allow the vehicle to perform within the VTS specifications. This chapter uses vehicle simulations to confirm the functionality of the RBS control algorithm and to demonstrate of improved energy conversion with the use of a series regenerative brake strategy compared to a parallel brake system. The overall vehicle controller including the RBS controller was integrated into a vehicle model developed using PSAT. Figure 19 shows the overall PSAT vehicle Simulink model for the TTR Equinox.

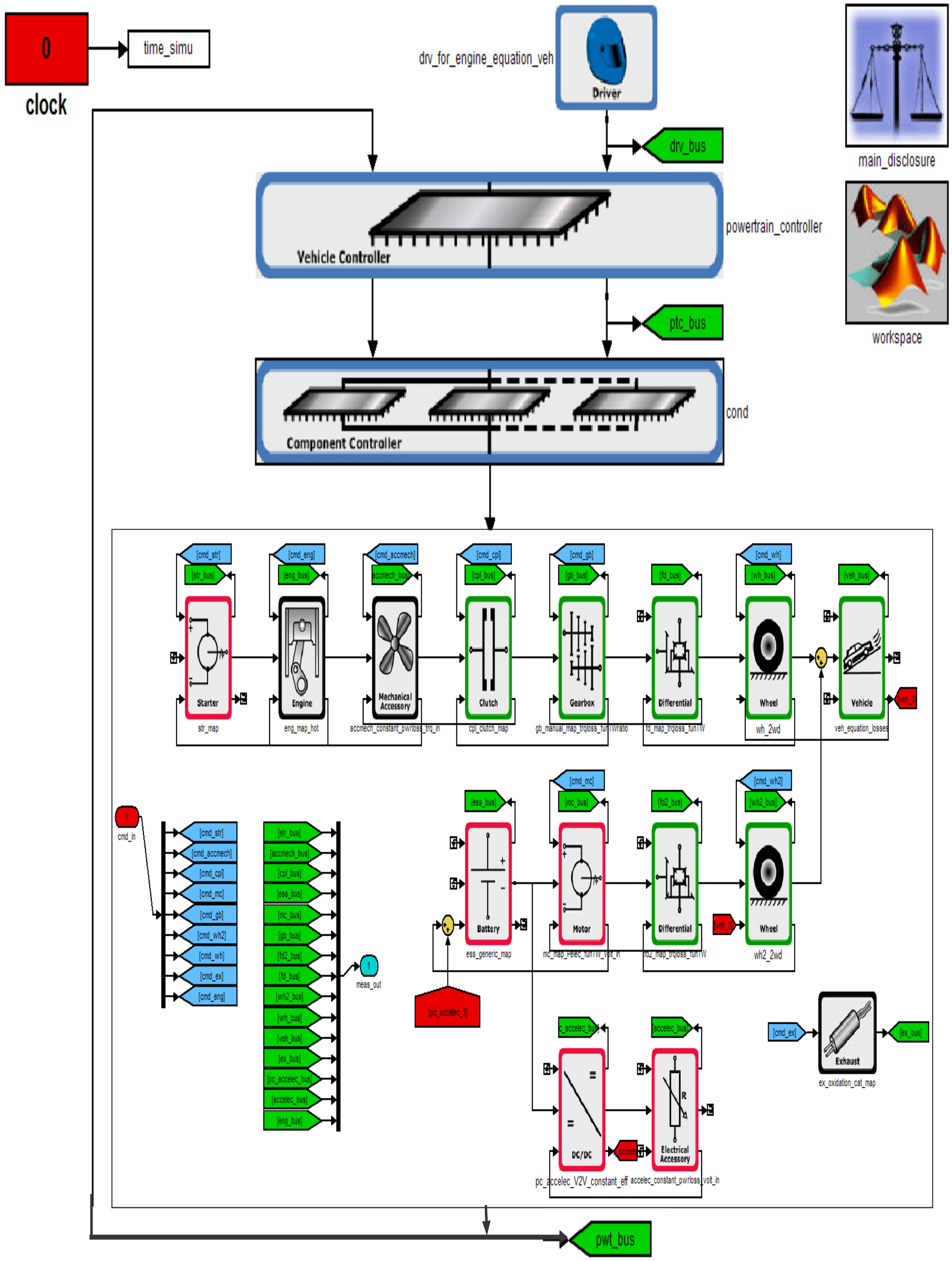


Figure 19 - PSAT vehicle model Simulink block diagram

Simulation Setup

PSAT uses a feed-forward control structure so that the output of each component model is the input to the next component. The components are placed in the order in which they would be found in the vehicle. A vehicle speed trace is input into the vehicle model through the driver demand block. The driver model provides a driver torque request to the vehicle controller that determines how to control the traction power sources based on the information from the powertrain component blocks. Outputs from the traction power sources propagate through the powertrain components (power source to gear boxes to differentials, etc.) until torque commands reach the wheels. The wheel models provide input to a vehicle dynamics model. This model calculates the vehicle speed based on the input from the wheels. Vehicle speed is then used as the feedback to the driver block to correct the driver power demand based on the difference between the desire vehicle speed and the actual vehicle speed.

Figure 20 shows the driver demand block. As described above, the driver block determines how much torque to request from the powertrain so the vehicle can follow the speed trace. PI control of the desired speed from the speed trace and the actual calculated model vehicle speed is used to determine the driver torque. This block represents a driver attempting to follow a speed trace, and the output is used to determine both accelerator and brake pedal positions as inputs to the vehicle controller. This block also provides the "key on" signal to start the vehicle.

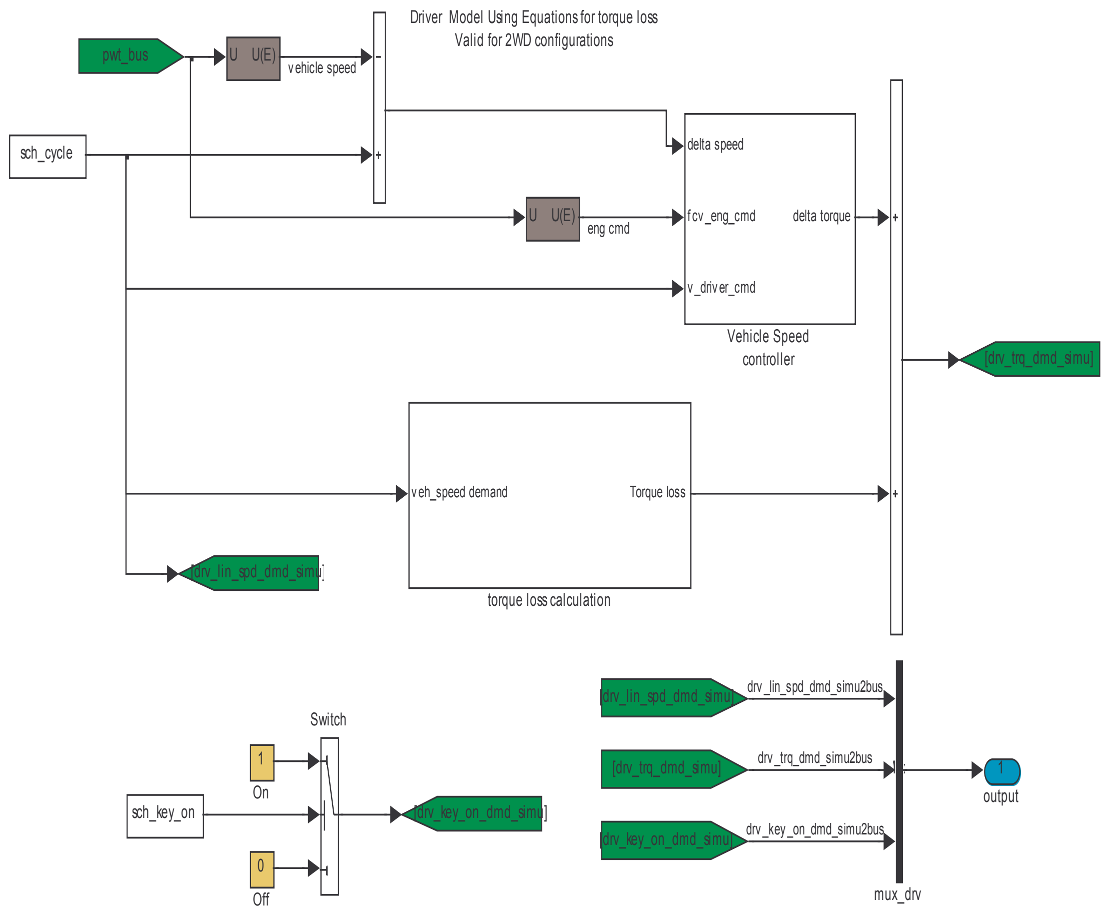


Figure 20 - PSAT driver demand calculation block

The UTK vehicle controller was integrated into PSAT. Three blocks are used to accomplish the integration, illustrated in Figure 21. The input and output Organizer blocks are used to convert PSAT variables to variables used by the UTK controller and then to convert the output of the controller back to PSAT variables, respectively. The accelerator and brake pedal signals are calculated in the input block. The output organizer block is used to actuate the foundation brake based on either a series or parallel electric brake strategy. The UTK controller is located within the center block.

The accelerator and brake pedal demands are 0 to 1 and 0 to $^-1$ signals, respectively, calculated using the driver torque demand. Figure 22 shows the block diagram used to calculated these signals. The accelerator pedal demand uses the positive

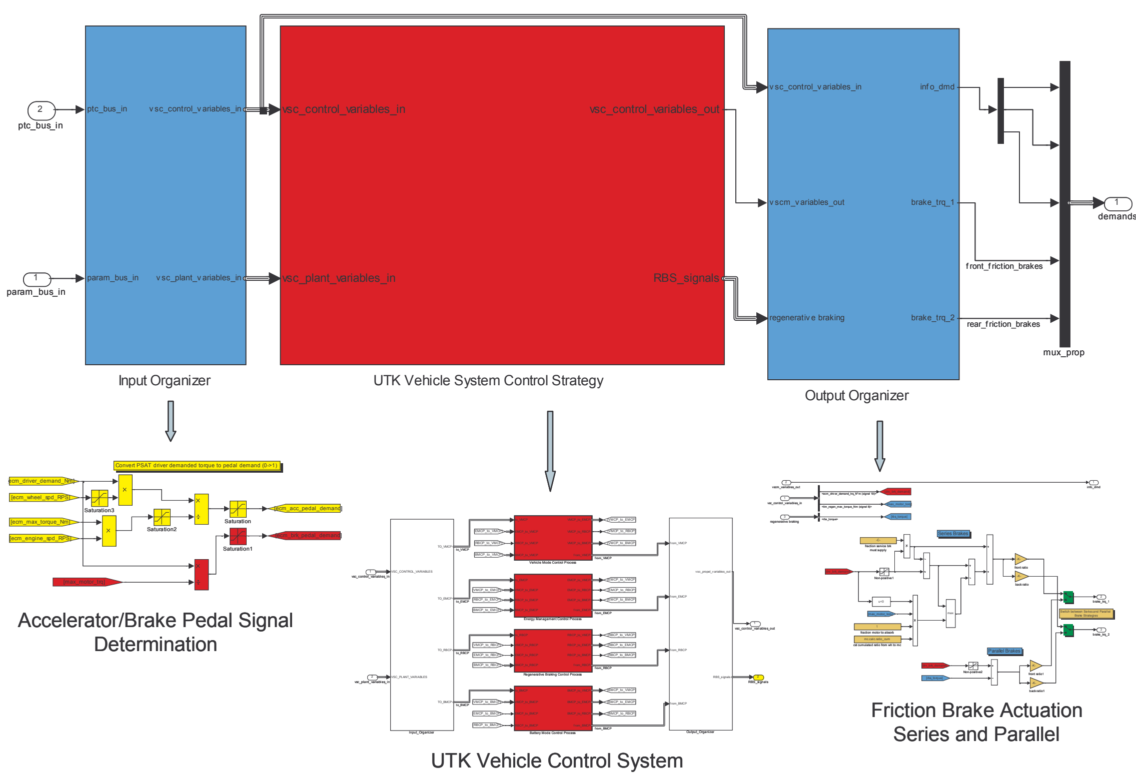


Figure 21 – Integration layout of the UTK controller in PSAT

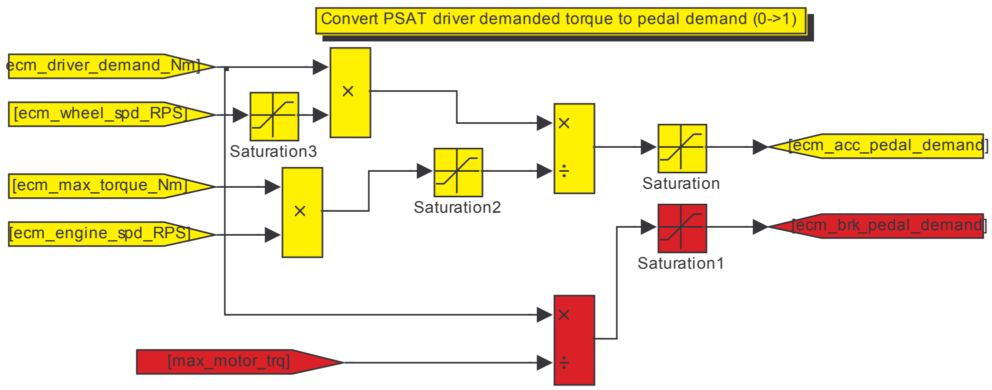


Figure 22– Conversion of PSAT driver demand into pedal demand

driver power demand divided by the maximum available engine power. The brake pedal demand uses the ratio of the driver demanded torque and the maximum available electric motor torque and saturation block to provide the zero to negative one brake signal. Recall that the RBS controller requires three signals to determine pedal position. These signals are only useful once the controller is integrated into the vehicle and are not needed for simulation purposes. The simulation brake pedal position is used in place of these signals.

One of the purposes of the simulation process is to show improved brake energy recovery using a series regenerative strategy compared to a parallel strategy. Figure 23 shows the block diagram used to simulate the two strategies. The braking block is used to determine if, and to what degree, the friction brakes are applied during a braking event. The series braking strategy compares the driver demanded brake torque at the drive wheel to the RBS torque acting on the wheel.

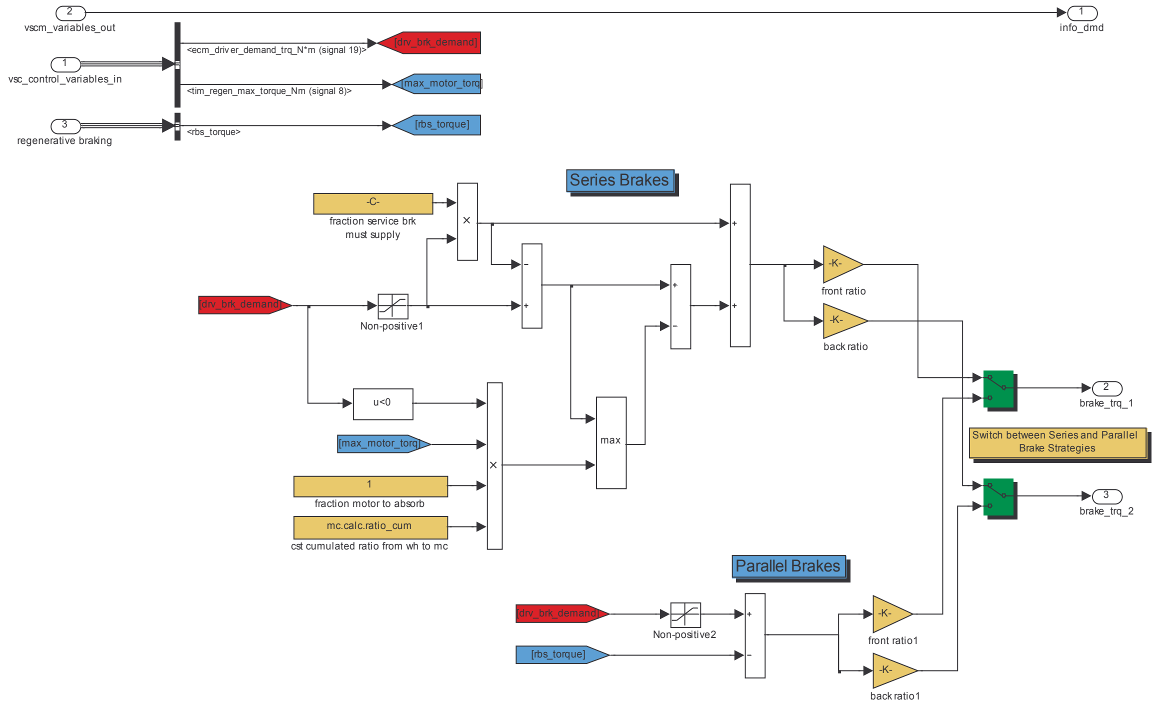


Figure 23 - Simulation friction brake actuation block diagram

The RBS torque acting on the wheel is calculated by multiplying the maximum available motor torque, the fraction of torque the motor should use, 100% for our case, and the cumulative gear ratio from wheel to motor through the final drive. Both the driver brake demand and the RBS torque are compared at the "max" logic block. This block passes the larger of the two signals. The signals are negative so the least negative signal is passed. The signal is then subtracted from the driver brake demand to determine how much friction braking is required to meet the driver demand. The friction brake torque is distributed evenly to the front wheels and rear wheels. This is not realistic for an actual brake system. As described in Chapter 3, there is a dynamic load transfer in which the normal load on the front axle is increased and is decreased on the rear axle

during a braking event. More work can be done on the front axle before lock up occurs than the rear axle. Conventional brake systems are proportioned to apply more force on the front than the rear as a result of the dynamic load transfer. PSAT does not calculate such effects as wheel slip and dynamic forces acting on the vehicle affecting the stability and drive quality, but rather calculates and traces the energy used by the various components throughout the powertrain as it is driven on a drive cycle. So proportioning the brake forces from front to rear does not affect the calculated total brake energy, as it is the sum of energy absorbed at both axles.

If the RBS torque meets the required brake demand, the driver brake demand is subtracted from itself and no friction torque is applied to the wheels. On the other hand, if the driver torque demand is greater than the RBS torque, the RBS torque is subtracted from the driver demanded torque and the value of the torque difference is then applied via the friction brakes to meet the brake demand. In this way, the electric brakes are used first, and the friction brake are only used if additional brake torque is needed. The series brake block simulates the function of the RBS system as designed.

Virtual manual switches, shown in green, are used to switch between the series and parallel brake strategies. The parallel strategy used assumes the friction brakes and the RBS system work together for the complete braking event. The motor braking torque was limited to 50 percent of its maximum output and is subtracted from the driver brake demand. In this way, the friction brake torque is applied in parallel with the RBS system to provide braking. The energy conversion comparisons between the two systems are discussed later in this chapter.

Simulation Results of the RBS Controller

With the vehicle controller integrated into the PSAT environment, vehicle simulations were executed to confirm RBS controller functionality and to compare energy conversion performance between the selected regenerative brake strategies. Two standard driving cycles were selected to evaluate the brake system. The Urban Dynamometer Driving Schedule (UDDS) is a low speed drive cycle that represents a vehicle driving in the city. The stop and go cycle in the UDDS allow the RBS system to activate many times throughout the city cycle and is thus much more effective. The other drive cycle chosen for the simulation process is the Highway Fuel Economy Test (HWFET). This cycle represents the vehicle driving on a typical highway with higher vehicle speed and fewer braking events.

Figures 24 and 25 show the RBS controller output over the total drive cycles. The results shown here illustrate the RBS controller using a series braking strategy. The RBS torque command is shown with the maximum available torque and friction brake torque over each cycle. The functionality of the controller can be seen by viewing regions of high brake activity at a closer view. Two regions within the drive cycle are used to show how the RBS system reacts to driver demand. The RBS torque (shown in red) provides the majority of the brake torque needed to meet the driver's brake demand.

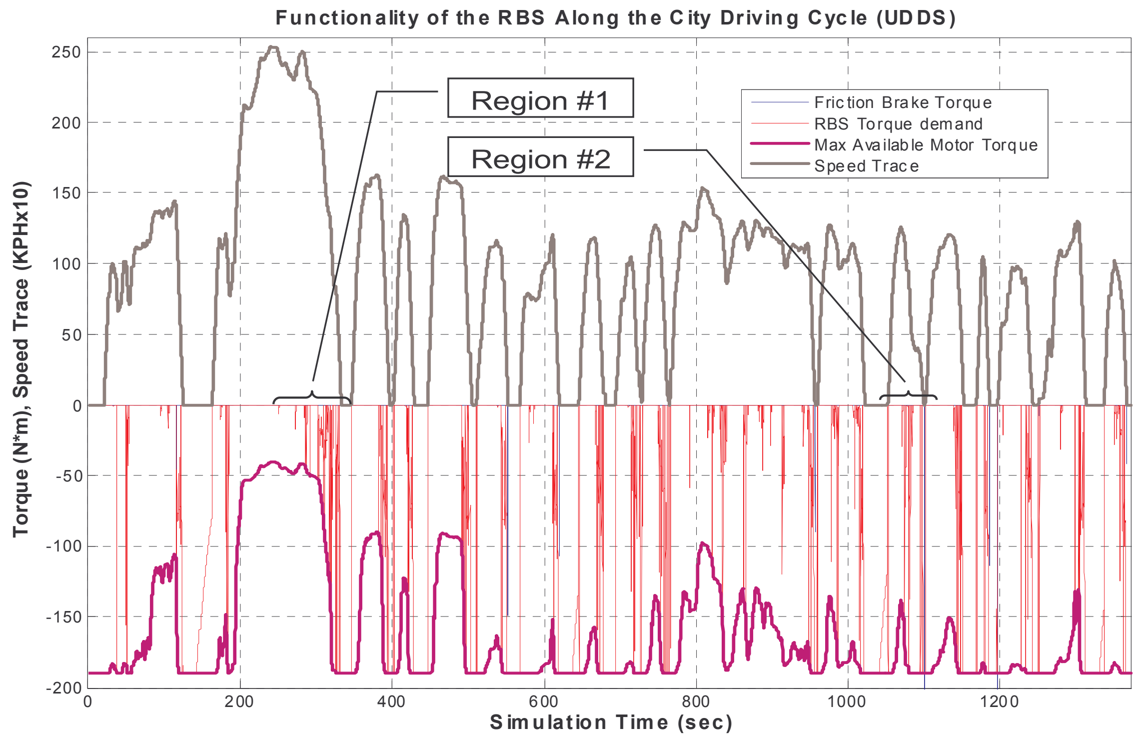


Figure 24 – Simulation output of the RBS controller for the UDDS

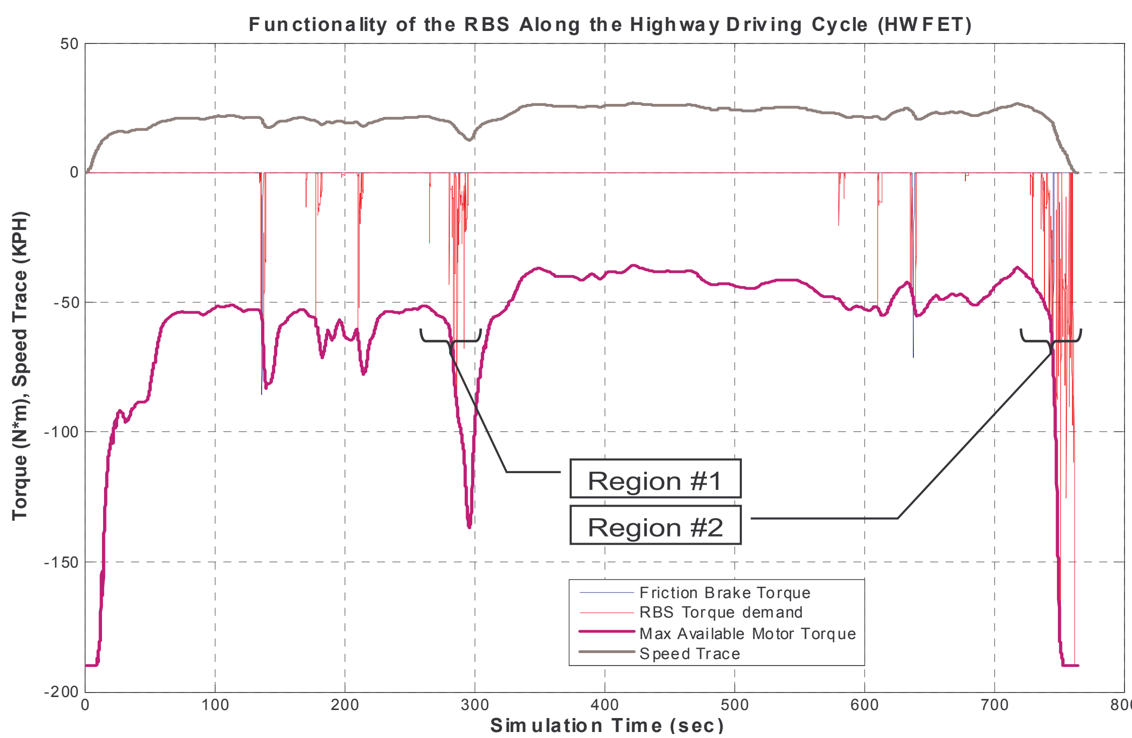


Figure 25 – Simulation output of the RBS controller for the HWFET

Figure 26, 27, 28, and 29 show the RBS torque output, friction brake output, and driver torque demand for regions one and two of the UDDS and HWFET drive cycles. The RBS torque follows closely with the driver demanded brake torque. It is clear the RBS system provides the majority of the brake torque needed to meet the drivers demand for both cycles. There was a peak in the driver demand block in figure 30 in which both brake systems were actuated. During this time, the RBS torque followed the maximum available motor curve as designed.

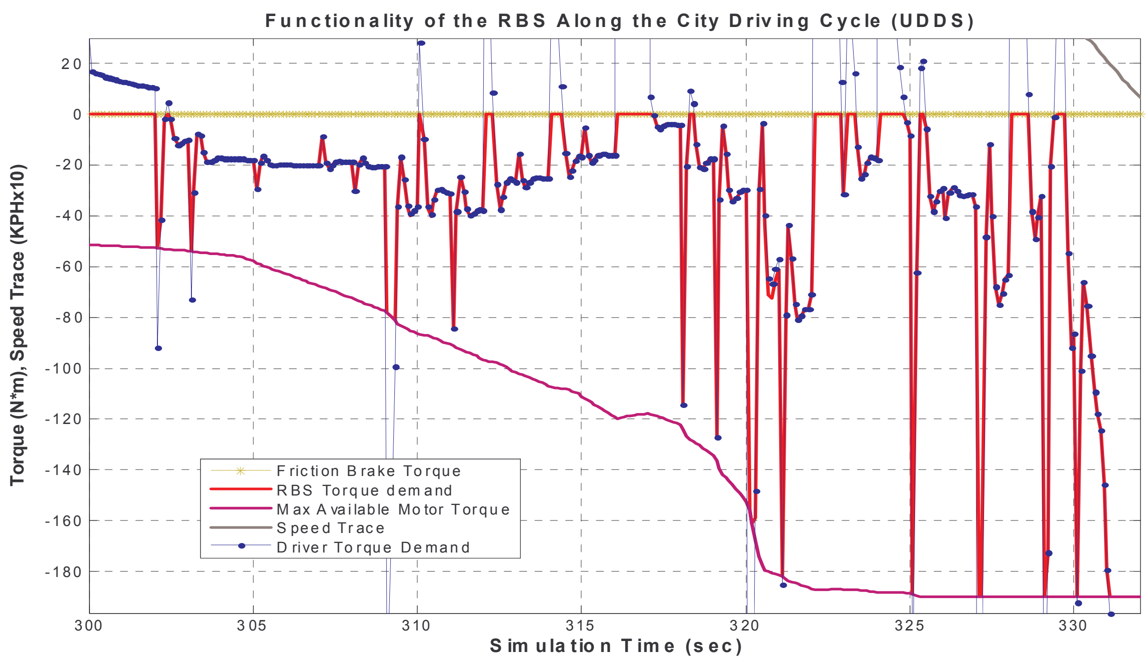


Figure 26 – RBS functionality in region one of the UDDS

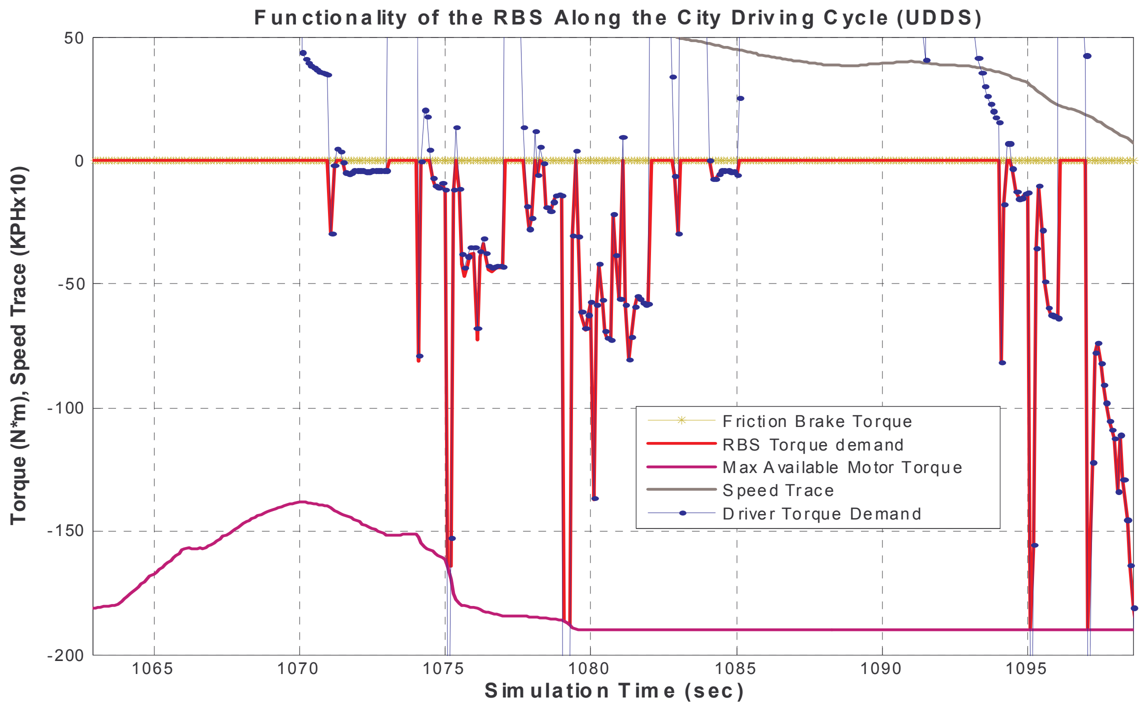


Figure 27 – RBS functionality in region two of the UDDS

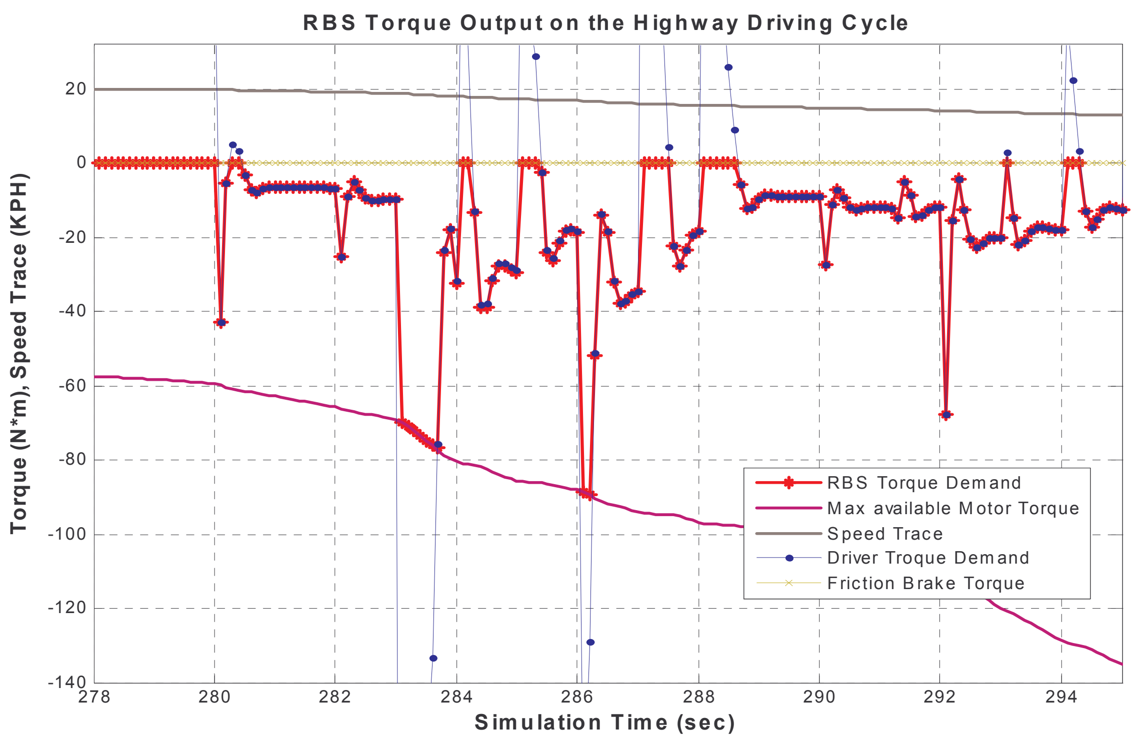


Figure 28 - RBS functionality in region one of the HWFET

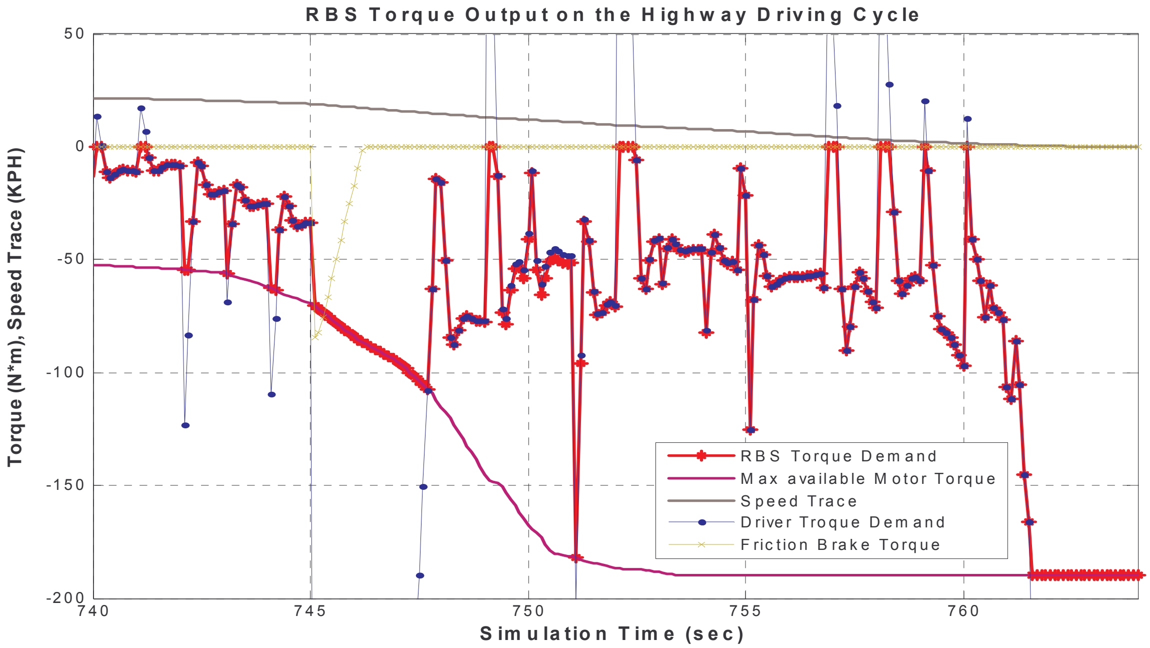


Figure 29 – RBS functionality in region two of the HWFET

The parallel braking strategy was executed over the UDDS. Figure 30 and 31 shows the overall torque sharing between the RBS and friction brakes. The friction brakes share nearly 50% of the brake torque demand from the driver. PSAT calculates various component and vehicle efficiencies, fuel economy, and other vehicle performance metric for each simulation. Full documentation of the simulation results are located in the Appendix. Among these metrics is a value of percent brake energy recuperated. This value is determined as the ratio of energy recuperated at the battery divided by the amount of energy available at the wheels during a braking event. This value is useful for determining how effective the regenerative brake system and respective strategies are in converting the kinetic energy to electric energy during braking.

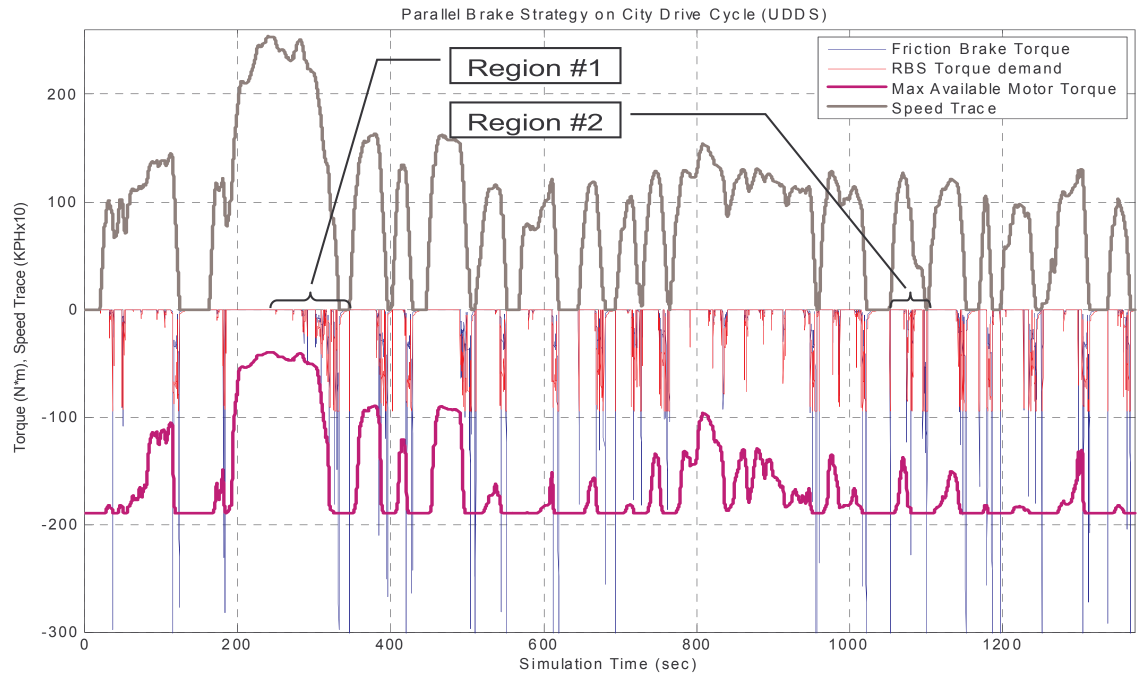


Figure 30 – Parallel brake strategy along the UDDS

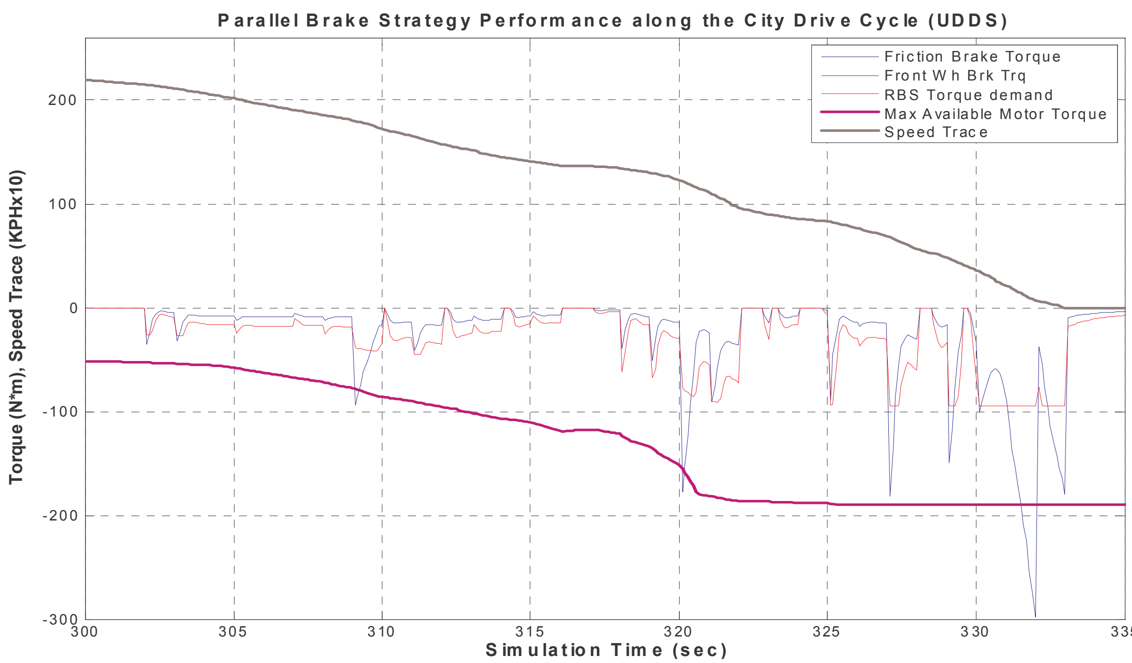


Figure 31 – Region one from overall parallel braking plot

Table 6 shows the comparison of the series and parallel brake strategies based on the percentage brake energy recuperated as well as other metrics, along the UDDS drive cycle. The brake energy recuperated metric also accounts for power losses through the drivetrain and power loss through electric accessories as well. Though the brake system has the ability to provide the entire brake torque needed over the drives cycles shown, part the energy is loss to heat generated during energy storage.

Table 6 - Simulation results for series and parallel brake strategies on the UDDS

	Series Braking	Parallel Braking
Brake Energy Recuperated	80.03%	66.21%
Motor Efficiency @ deccel	93.03%	86.72%
Engine Efficiency	25.50%	21.33 %
Energy Loss Rear wheel	148.46 Wh	196.97 Wh
Energy Loss Front wheel	148.46 Wh	196.97 Wh

Limitations of Vehicle Simulations

Two operations of the RBS controller were not simulated due to model limitations. The wheel lock control and RBS Fade In operation are functions used in the controller to provide transparent operation of the RBS controller to the driver. The wheel lock control is used to unlock the wheels if wheel lock is detected during electric braking. The functionality of the wheel lock system could not be shown using the PSAT vehicle model because the wheel models do not model the tire in contact with the road including wheel slip calculations. Calibration of the wheel slip control should be completed through vehicle testing. The RBS Fade In function in the controller is a driver feel function. When a brake event is initiated above the threshold speed and continues below the threshold thus activating the RBS system, the RBS Fade In allow the RBS torque command to be gradually increased until the driver responds by releasing the friction brakes. The purpose of this function is to prevent a sudden increase in brake torque when the RBS system is activated from above the threshold speed. This function is a driver feel metric and should be calibrated through vehicle testing.

Conclusion

A series regenerative brake strategy provides a hybrid electric vehicle with higher cycle efficiency by capturing more of the vehicles kinetic energy during braking compared to a parallel braking strategy. The thru-the-road hybrid electric configuration only allows the electric brake system to act on one axle unless two electric motor where used. From the analysis in Chapter 3 describing the dynamic load transfer that occurs during braking, more energy can be converted if the system was active on the front axle

than the rear axle. It is recommend that if packaging allows a regenerative brake system should be included on the front axle for a thru-the-road hybrid configuration, if not both.

Future Work

The RBS system has been design to provide a calibratable system with a focus on flexible vehicle integration and operation. Many aspects of the RBS system can be adjusted to meet the specifications of the driver in both feel and performance. Integrating the RBS system into the vehicle architecture will require calibration in several areas of the controller. The Zero Speed Fade curve allows the RBS torque to be reduced as the vehicle comes to a stop providing smooth operation. The Fade In control needs to be calibrated to allow the RBS torque to increase rapidly enough to effectively convert the kinetic energy to electric energy without causing the driver to react to a unstable system. Vehicle testing is needed to increase the operating speed threshold allowing the RBS system to operate at higher vehicle speeds while maintaining vehicle stability.

List of References

List of References

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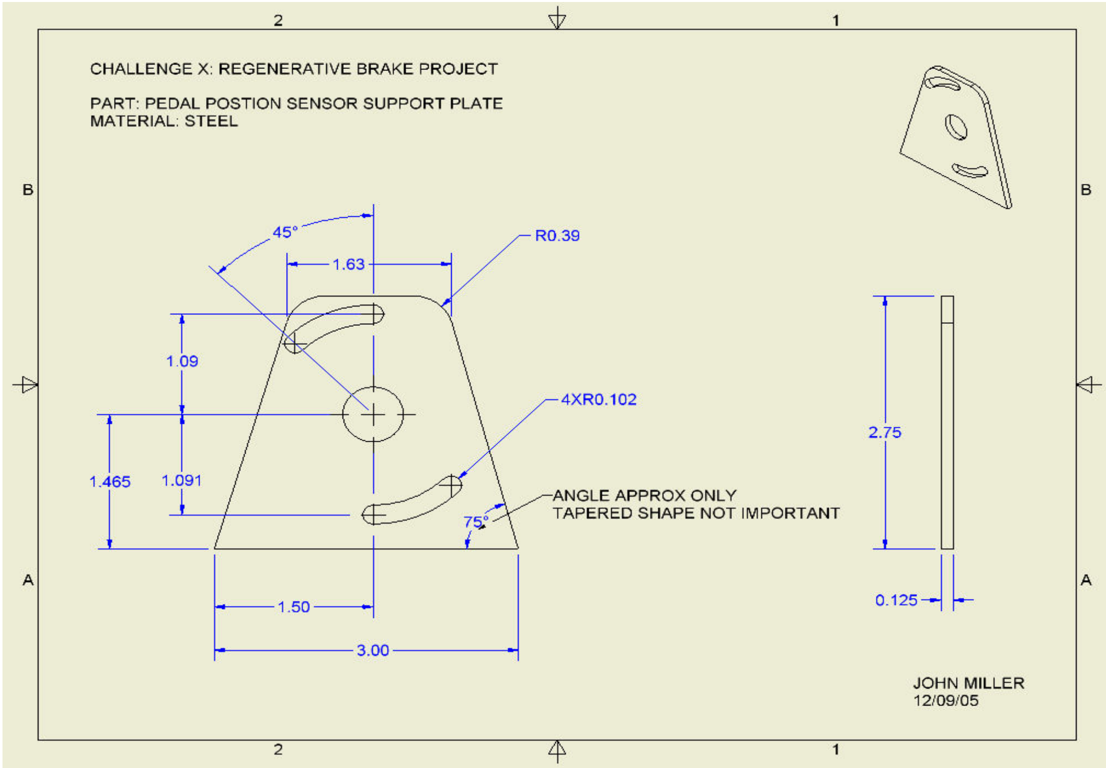
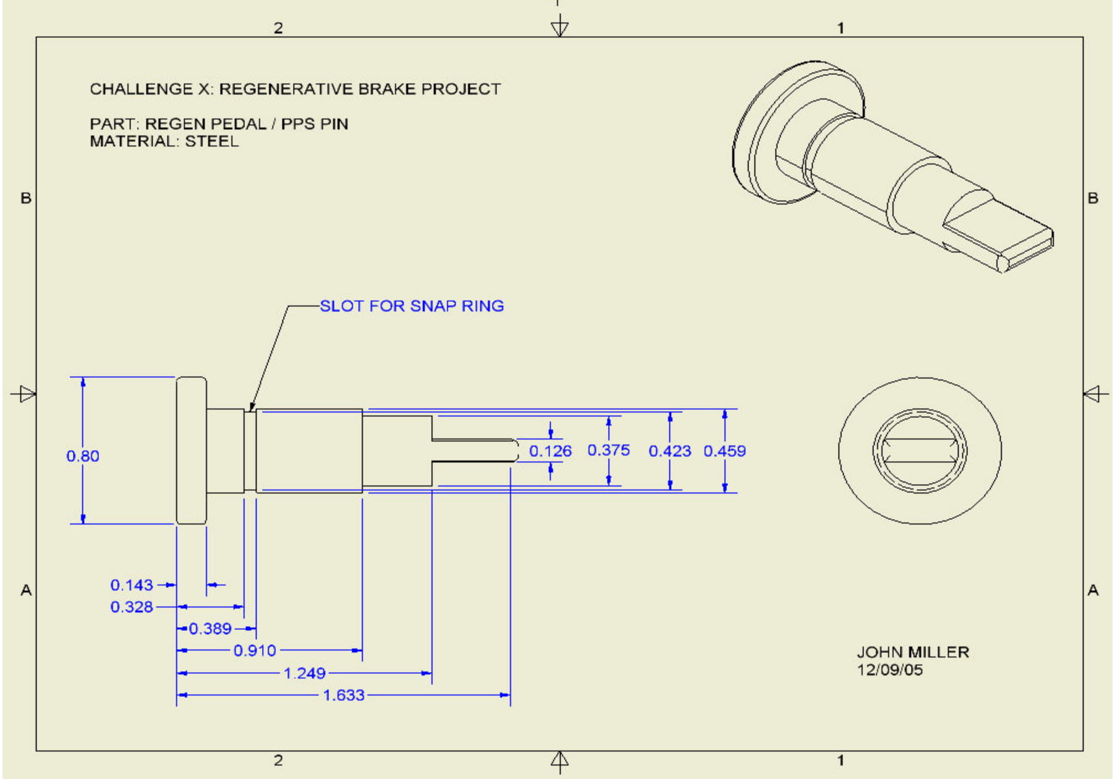
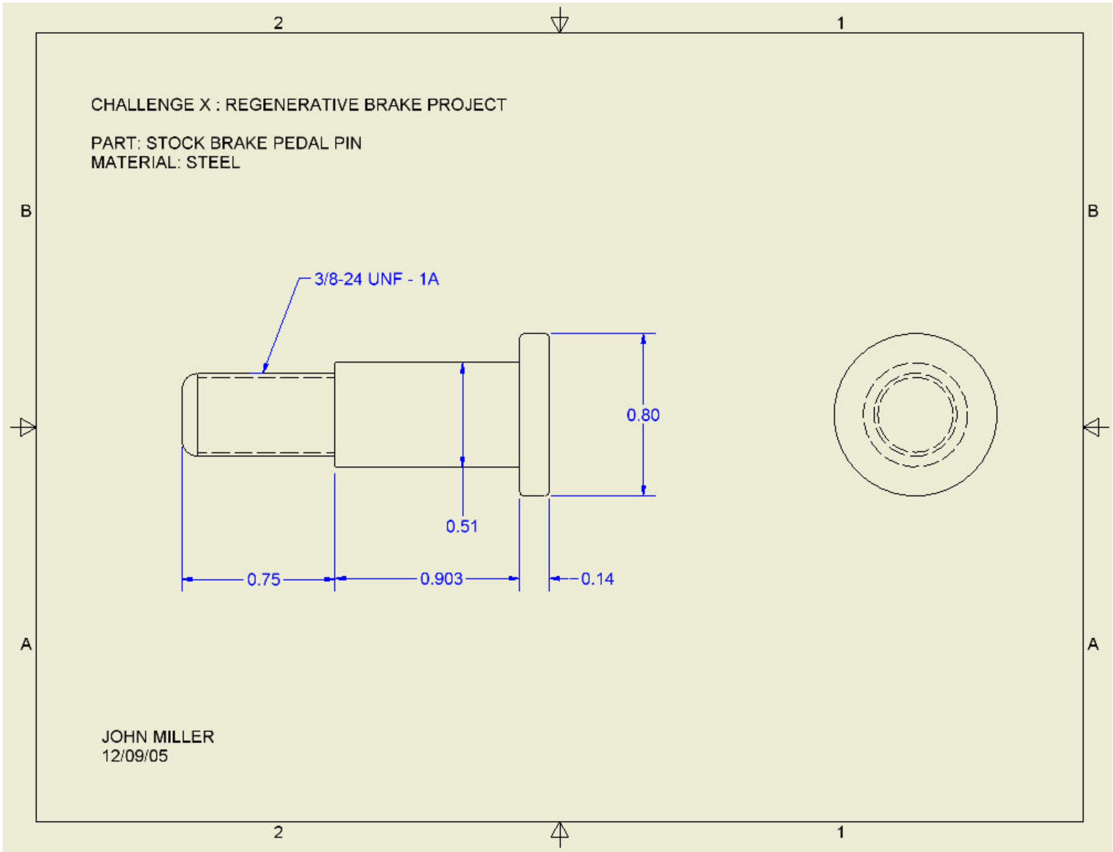
Appendix

Appendix I – CAD Drawings






2
1
B
A
CHALLENGE X : REGENERATIVE BRAKE PROJECT
PART: STOCK BRAKE PEDAL PIN
MATERIAL: STEEL
3/8-24 UNF - 1A
0.80
0.51
0.75
0.903
0.14
JOHN MILLER
12/09/05

CTS®

503 Series

Electronic Pedal Sensors

Typical Application: Pedal Position

Adjustable Option

Non-Adjustable Option

	Type	Adjustable	Non Adjustable
FEATURES / BENEFITS:	Potentiometers	2 Maximum	3 Maximum
	Switches	Multiple Switch combinations	
	Mates with	6 pin Amp Mini timer Connector (P/N C-282236)	Packard 150 Series Metri-Pack Connector (P/N 12065425)
		CW and CCW Rotation Available	
		Sealed	
		Field Proven in Advanced Drive-By-Wire Systems	
		Assured Registration Accuracy	
		Low Cost	
		Customizable Electrical Outputs	
		Laser Marking of P/N and Logo on Cover	
		Gold or Tin Plated Terminals Available	

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Date: 4/5/04
Revised: 01/29/03

TYPICAL SPECIFICATIONS

	Type		Adjustable	Non Adjustable
ELECTRICAL	Electrical Travel:		Maximum 65°	Maximum 30°
	Recommended Supply Voltage (Vref):		5 V	
	Output Voltage:		5% to 95% Vref	
	Max. Current Draw (I):		5.0 mA, each Circuit	
	Overall Resistance (ROA):		1 to 5 KO ± 30% (typical)	
	Series Resistance (RS):		400O to 1.7 KO ± 30% (typical)	
	Recommended Load Resistance (RL):		= 200 KO	
	Index Reference Points:		±1.0° X Nominal Gain (%/degree)	
	Ideal Linearity (point-slope):		= 1.0° X Nominal Gain (%/degree)	
	Increment Linearity (slope deviation):	Initial:	1.0 ± 0.7 X Nominal Gain (%/degree) per 0.1°	
		After Durability Test:	1.0 ± 1.4 X Nominal Gain (%/degree) per 0.1°	
	Hysteresis:		= 0.5° X Nominal Gain (%/degree)	
	Output Correlation:		All Potentiometric Outputs to correlate to each other within ± 0.8° X Nominal Gain (%/degree).	
MECHANICAL	Rotor return torque		14-70 Nmm	
	Max Mount Bolt Torque		2.8 Nm	
	Mechanical Shock		25 g	
DURABILITY	Full Stroke Durability		= 4 X 10^6 at -30° to 80°C and ambient to 80% Humidity, cycling rate <4 Hz	= 4 X 10^6 at -40° to 85°C and 0-95% Humidity
	Dither Stroke Durability		48h at 80°C, 30 Hz	= 25 X 10^6 at 25°C with 1° Stroke
	Vibrational durability		(sine) 8 hrs in each of 3 planes, 6 G 10-2000 Hz	(random) 20 hrs. in each of 3 Planes, 2.5 GRMS 10-2000 Hz
ENVIRONMENTAL	Salt Fog:		96 hrs. with 5% Neutral Salt at 35°C	
	Dust:		9 hrs. per SAE J726 with 0.003 to 0.014 g/cm^3 Concentration	
	Fluid Compatibility:		Fifteen Fluids Tested	
	Thermal Shock:		-40° to 125°C	
	Meets normal underhood and passenger compartment requirements			

Note: Standard sensor outputs are provided in this data sheet as a guideline. CTS can customize potentiometric outputs within the ranges suggested herein to meet customer application requirements.

508 Series

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Revised: 01/29/0

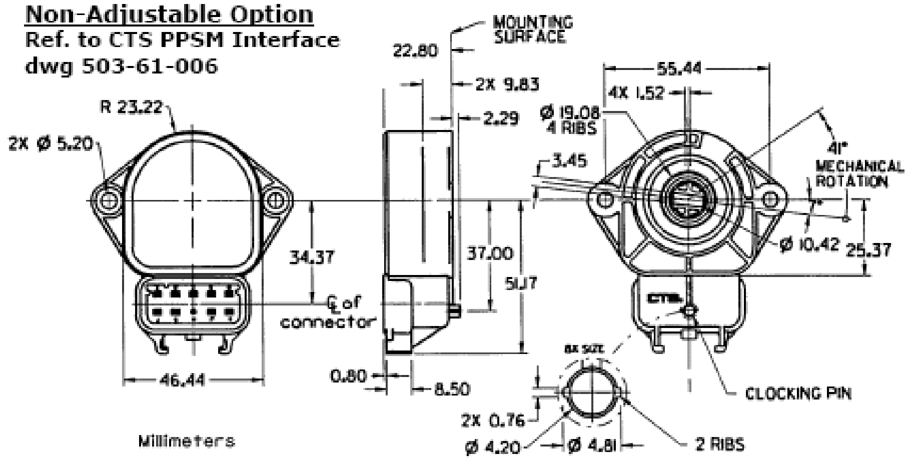
503 Series

Mechanical Interfacing to Pedal

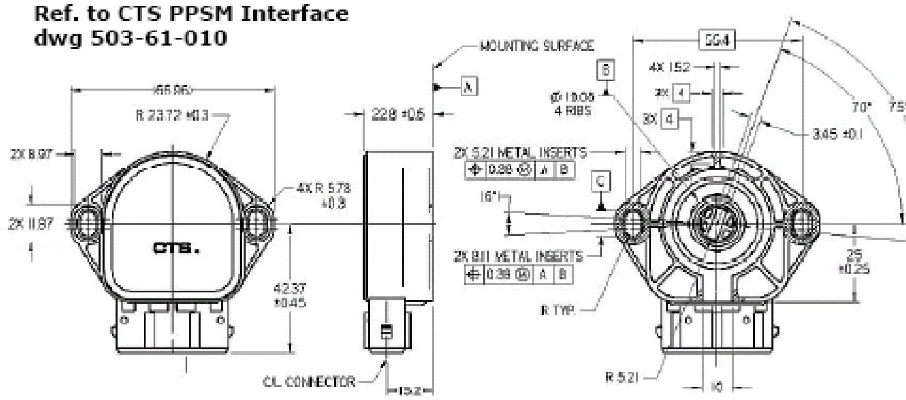
Two key features on the 503 series housing assure accurate fixing to the pedal:

1. Pilot Hub (Ø 19.08, 4 ribs): The mounting boss (surrounding the sensor's rotor) is designed to assure concentricity and has four (4) plastic ribs which deform as the mounting boss is forced into a 18.98 mm +0.0/-0.05 diameter pilot bore in the pedal. The pedal manufacturer must assure the pilot bore is concentric to the drive shaft and that adequate bearing support of the shaft is provided.
2. Clocking Pin (non-adjustable option): The 503 sensor's clocking pin (located on the back of the connector) assures angular registration to the pedal. Note that mounting-bolt hole locations do not perform this function. The clocking pin has two (2) plastic crush ribs which deform when inserted into a 4.76 mm ±0.05 diameter mating hole in the pedal.
   Elongated mounting-bolt holes (adjustable option): The 503 sensor's adjustable mounting-bolt hole location and size assure angular registration to the pedal. They also allow the sensor's index voltage to be adjusted within a finite range by rotating the sensor relative to the pedal drive shaft.

Non-Adjustable Option
Ref. to CTS PPSM Interface dwg 503-61-006



Adjustable Option
Ref. to CTS PPSM Interface dwg 503-61-010



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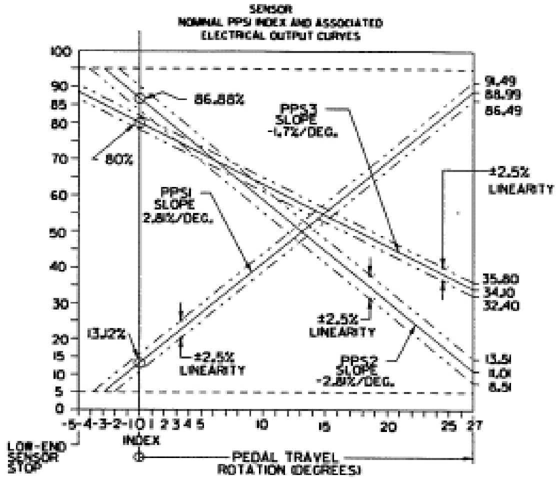
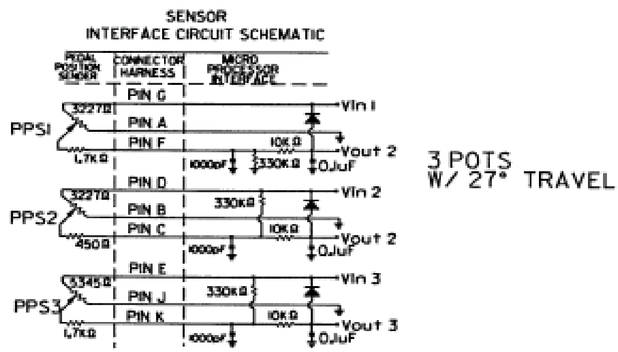
Non-Adjustable Option

Standard Electrical Output





PIN ALLOCATION (REFERENCE)

E PPS3 (+)	D PPS2 (+)	C PPS2 WIPER	B PPS2 (-)	A PPS1 (-)
F PPS1 WIPER	G PPS1 (+)	H (UNUSED)	J PPS3 (-)	K PPS3 WIPER

Other Available Non - Adjustable Outputs

Version	Outputs	Sensor Slope(s)	Switch(es)	Plateau(s)	Travel
Standard	3 sensors	PPS1 2.81%/DEG PPS2 –2.81%/DEG PPS3 –1.70%/DEG	None	None	27°
Customized 1	2 sensors 1 switch	PPS1 3.085%/DEG PPS2 –3.085%/DEG	1 (NO)	High & Low End	30°
Customized 2	3 sensors	PPS1 5%/DEG PPS2 -5%/DEG PPS3 –2.87%/DEG	None	High & Low End	14°
Customized 3	2 sensors	PPS1 3.346%/DEG PPS2 –3.346%/DEG	None	High End	23°
Customized 4	3 sensors	PPS1 3.398%/DEG PPS2 -3.398%/DEG PPS3 -4.724%/DEG	None	High & Low End	30°
Customized 5	1 sensor 2 switches	PPS1 2%/DEG	2 (NO&NC)	None	30°

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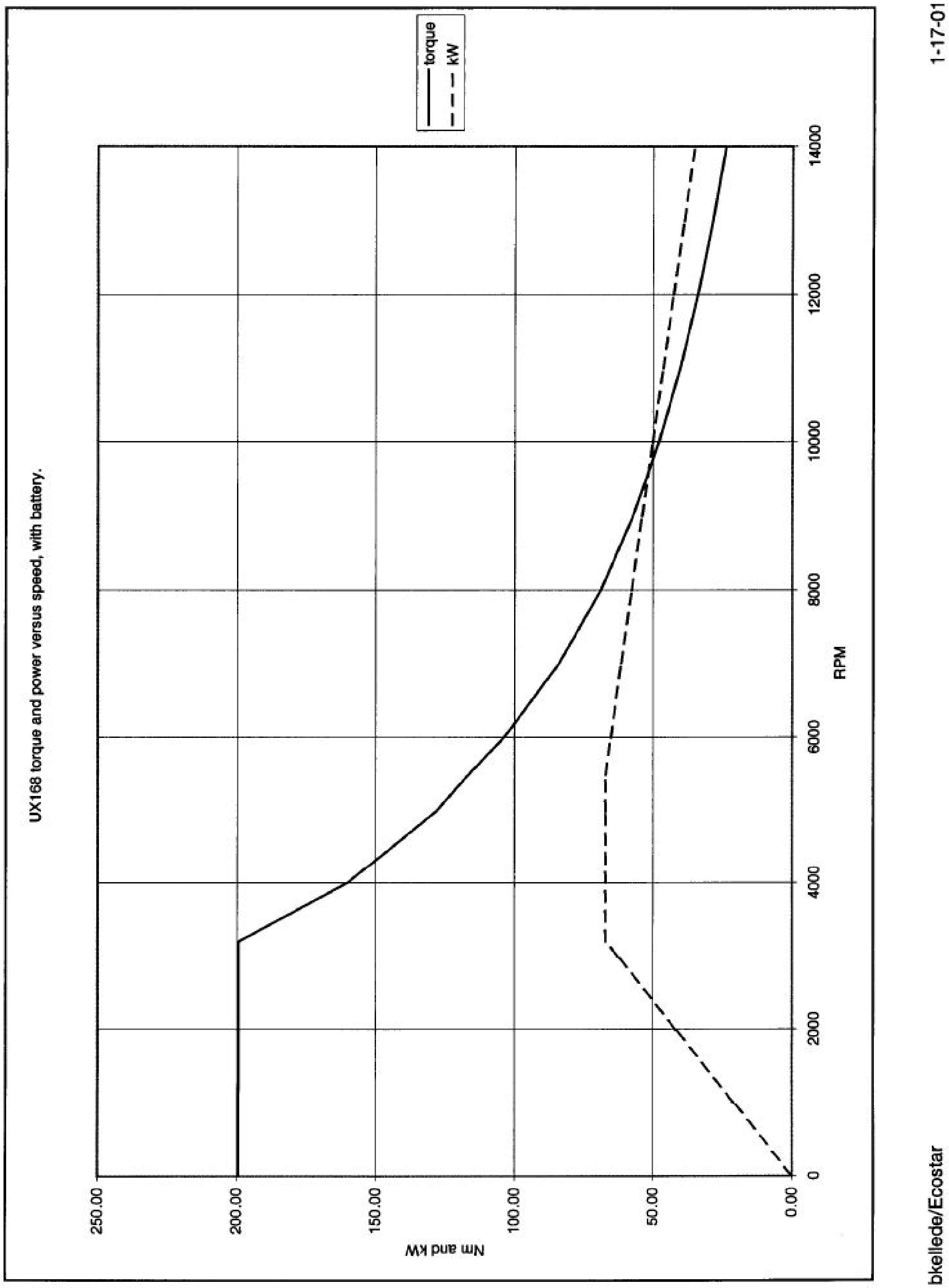

UX168 torque and power versus speed, with battery.
Nm and kW
250.00
200.00
150.00
100.00
50.00
0.00
0
2000
4000
6000
8000
10000
12000
14000
RPM
torque
kW
bkellede/Ecostar
1-17-01

Appendix III – PSAT Vehicle Simulation Results

****
*********************** SIMULATION RESULTS ***************************
****

THRU-THE-ROAD HYBRID ELECTRIC CHEVROLET EQUINOX
PARALLEL BRAKE STRATEGY
Cycle: 1.0 time the UDDS
Cycle distance: 7.29 miles
Cycle distance: 11.73 km
Vehicle mass: 1954.92 kg
*********** THERMAL INFORMATION ***********
** Engine
Fuel economy: 34.98 mile/gallon
Fuel economy gasoline equivalent : 31.45 mile/gallon
Fuel mass: 0.66 kg
HC emissions: 0.0000 g/mile
CO emissions: 0.0000 g/mile
NOx emissions: 0.0000 g/mile
CO2 emissions: 288.24 g/mile
*********** ESS ELECTRICAL INFORMATION ********
ESS Electric-only: -13.83 Wh/mile
Initial ESS SOC : 70.00 %
Final abs ESS SOC: 72.19 %
Final usable ESS SOC: 72.19 %
*********** TOTAL ELECTRICAL INFORMATION ********
Electric-only: -13.83 Wh/mile
*********** COMBINED INFORMATION **********
PSAT onboard lumped ess eff gasoline eq.: 33.88 mile/gallon
PSAT onboard chg dis ess eff gasoline eq.: 33.98 mile/gallon
PSAT onboard thermal/regen partioned lumped
ess eff gasoline eq.: 33.88 mile/gallon
Student competition onboard gasoline eq.: 32.74 mile/gallon
PSAT offboard gasoline eq.: 32.63 mile/gallon
Student competition offboard gasoline eq.: 32.34 mile/gallon
Energy loss accmech : 294.76 Wh
Energy loss cpl : -287.54 Wh
Energy loss ess : 24.49 Wh
Energy loss mc : 163.80 Wh
Energy loss gb : 86.35 Wh
Energy loss fd2 : 31.67 Wh
Energy loss fd : 50.02 Wh
Energy loss wh2 : 196.97 Wh
Energy loss wh : 196.97 Wh
Energy loss pc_accelec : 0.00 Wh
Energy loss accelec : 0.00 Wh
Energy loss eng : 6121.09 Wh
Energy loss aero: 492.58 Wh
Energy loss drag: 148.00 Wh
Energy loss grade: 0.00 Wh

Total Energy loss : 7519.15 Wh
Total Energy used @ vehicle: 1593.19 Wh

Engine Fuel Energy Use: 7780.68 Wh
Fuel Cell Fuel Energy Use: 0.00 Wh
Total Fuel Energy Use: 7780.68 Wh
ESS Net Energy Use (propel + regen): -76.36 Wh
ESS2 Net Energy Use (propel + regen): 0.00 Wh
Total Net Energy Use (propel + regen): -76.36 Wh
Combined Power Source Energy Use: 7704.32 Wh
Powertrain Energy Use In @ wheel (w/regen): 8644.72 Wh
Powertrain Energy Use Out @ wheel (w/regen): 8481.14 Wh
Percentage Braking Energy Recuperated : 66.21 %

***** COMPONENT AVERAGE EFFICIENCIES *******

Engine efficiency : 21.33 %
Motor efficiency : 83.09 %
Motor efficiency during acceleration: 77.90 %
Motor efficiency during deceleration: 86.72 %
Transmission efficiency : 95.80 %

************ OTHER INFORMATION ************

** Engine
Weight Specific Fuel Consumption Gas Eq. (WSFC): 1.63 gallon/100mile/ton
Mass of fuel needed to travel 320 miles: 28.92 kg

*************** STATISTICS ****************

Absolute average difference on vehicle speeds: 0.47 mile/h
Percentage of the time the trace is missed by 2mph: 2.07 %
Total time the trace is missed by 2mph: 28.40 s
Greatest percentage deviation from the trace in mph: 96.32 % at time 1264.20 s
Absolute deviation from the trace in mile/h: 10.09 at time 1264.00 s
Save directory: par_2t2wd_dm_UTK_UDDS_031506_164044

****
************************ SIMULATION RESULTS ****************************
****

THRU-THE-ROAD HYBRID ELECTRIC CHEVROLET EQUINOX
SERIES REGENERATIVE BRAKING STRATEGY

Cycle: 1.0 time the UDDS
Cycle distance: 7.32 miles
Cycle distance: 11.78 km
Vehicle mass: 1954.92 kg

*********** THERMAL INFORMATION ***********

* Engine
Fuel economy: 30.28 mile/gallon
Fuel economy gasoline equivalent : 27.22 mile/gallon
Fuel mass: 0.76 kg

HC emissions: 0.0000 g/mile
CO emissions: 0.0000 g/mile
NOx emissions: 0.0000 g/mile
CO2 emissions: 333.25 g/mile

*********** ESS ELECTRICAL INFORMATION ********

ESS Electric-only: -14.13 Wh/mile

Initial ESS SOC :	70.00	%
Final abs ESS SOC:	73.81	%
Final usable ESS SOC:	73.81	%

*********** TOTAL ELECTRICAL INFORMATION ********

Electric-only:	-14.13	Wh/mile

*********** COMBINED INFORMATION **********

PSAT onboard lumped ess eff gasoline eq.:	46.27	mile/gallon
PSAT onboard chg dis ess eff gasoline eq.:	48.06	mile/gallon
PSAT onboard thermal/regen partioned lumped ess eff gasoline eq.:	46.27	mile/gallon
Student competition onboard gasoline eq.:	42.55	mile/gallon
PSAT offboard gasoline eq.:	36.76	mile/gallon
Student competition offboard gasoline	36.30	mile/gallon
Energy loss accmech :	294.29	Wh
Energy loss cpl :	-263.95	Wh
Energy loss ess :	36.38	Wh
Energy loss mc :	144.33	Wh
Energy loss gb :	99.61	Wh
Energy loss fd2 :	35.53	Wh
Energy loss fd :	64.02	Wh
Energy loss wh2 :	148.46	Wh
Energy loss wh :	148.46	Wh
Energy loss pc_accelec :	0.00	Wh
Energy loss accelec :	0.00	Wh
Energy loss eng :	6727.71	Wh
Energy loss aero:	491.53	Wh
Energy loss drag:	148.01	Wh
Energy loss grade:	0.00	Wh
Total Energy loss :	8074.37	Wh
Total Energy used @ vehicle:	1611.77	Wh
Engine Fuel Energy Use:	9030.31	Wh
Fuel Cell Fuel Energy Use:	0.00	Wh
Total Fuel Energy Use:	9030.31	Wh
ESS Net Energy Use (propel + regen):	-798.85	Wh
ESS2 Net Energy Use (propel + regen):	0.00	Wh
Total Net Energy Use (propel + regen):	-798.85	Wh
Combined Power Source Energy Use:	8231.46	Wh
Powertrain Energy Use In @ wheel (w/regen):	9526.41	Wh
Powertrain Energy Use Out @ wheel (w/regen):	3440.21	Wh
Percentage Braking Energy Recuperated :	80.03	%

***** COMPONENT AVERAGE EFFICIENCIES *******

Engine efficiency :	25.50	%
Motor efficiency :	85.12	%
Motor efficiency during acceleration:	52.75	%
Motor efficiency during deceleration:	93.03	%
Transmission efficiency :	96.23	%

************ OTHER INFORMATION ************

** Engine

Weight Specific Fuel Consumption Gas Eq. (WSFC):	1.88	gallon/100mile/ton
Mass of fuel needed to travel 320 miles:	33.41	kg

*************** STATISTICS ****************

Absolute average difference on vehicle speeds:	0.60	mile/h
Percentage of the time the trace is missed by 2mph:	5.08	%
Total time the trace is missed by 2mph:	69.70	s

Greatest percentage deviation from the trace in mph:	96.49 %	at time 1264.00 s
Absolute deviation from the trace in mile/h:	10.13	at time 1264.00 s
Save directory:	par_2t2wd_dm_UTK_UDDS_031506_171341	

****
************************ SIMULATION RESULTS ****************************
****

THRU-THE-ROAD HYBRID ELECTRIC CHEVROLET EQUINOX
PARALLEL BRAKE STRATEGY

Cycle:	1.0 time the HWFET	
Cycle distance:	10.23	miles
Cycle distance:	16.46	km
Vehicle mass:	1954.92	kg

*********** THERMAL INFORMATION ***********

** Engine

Fuel economy:	45.75	mile/gallon
Fuel economy gasoline equivalent :	41.13	mile/gallon
Fuel mass:	0.71	kg
HC emissions:	0.0000	g/mile
CO emissions:	0.0000	g/mile
NOx emissions:	0.0000	g/mile
CO2 emissions:	220.25	g/mile

*********** ESS ELECTRICAL INFORMATION ********

ESS Electric-only:	-12.06	Wh/mile
Initial ESS SOC :	70.00	%
Final abs ESS SOC:	73.29	%
Final usable ESS SOC:	73.29	%

*********** TOTAL ELECTRICAL INFORMATION ********

Electric-only:	-12.06	Wh/mile

*********** COMBINED INFORMATION **********

PSAT onboard lumped ess eff gasoline eq.:	43.85	mile/gallon
PSAT onboard chg dis ess eff gasoline eq.:	44.01	mile/gallon
PSAT onboard thermal/regen partioned lumped ess eff gasoline eq.:	43.85	mile/gallon
Student competition onboard gasoline eq.:	43.55	mile/gallon
PSAT offboard gasoline eq.:	42.91	mile/gallon
Student competition offboard gasoline eq.:	42.78	mile/gallon
Energy loss accmech :	211.08	Wh
Energy loss cpl :	-28.23	Wh
Energy loss ess :	8.63	Wh
Energy loss mc :	155.66	Wh
Energy loss gb :	31.40	Wh
Energy loss fd2 :	9.09	Wh

Energy loss fd :	62.15	Wh
Energy loss wh2 :	214.81	Wh
Energy loss wh :	214.81	Wh
Energy loss pc_accelec :	0.00	Wh
Energy loss accelec :	0.00	Wh
Energy loss eng :	5690.94	Wh
Energy loss aero:	1626.13	Wh
Energy loss drag:	207.82	Wh
Energy loss grade:	0.00	Wh
Total Energy loss :	8404.30	Wh
Total Energy used @ vehicle:	706.65	Wh
Engine Fuel Energy Use:	8341.40	Wh
Fuel Cell Fuel Energy Use:	0.00	Wh
Total Fuel Energy Use:	8341.40	Wh
ESS Net Energy Use (propel + regen):	-114.72	Wh
ESS2 Net Energy Use (propel + regen):	0.00	Wh
Total Net Energy Use (propel + regen):	-114.72	Wh
Combined Power Source Energy Use:	8226.68	Wh
Powertrain Energy Use In @ wheel (w/regen):	8489.93	Wh
Powertrain Energy Use Out @ wheel (w/regen):	2646.07	Wh
Percentage Braking Energy Recuperated :	80.15	%

***** COMPONENT AVERAGE EFFICIENCIES *******

Engine efficiency :	31.78	%
Motor efficiency :	73.93	%
Motor efficiency during acceleration:	-29.28	%
Motor efficiency during deceleration:	93.51	%
Transmission efficiency :	98.75	%

************ OTHER INFORMATION ************

** Engine

Weight Specific Fuel Consumption Gas Eq. (WSFC):	1.24	gallon/100mile/ton
Mass of fuel needed to travel 320 miles:	22.11	kg

*************** STATISTICS ****************

Absolute average difference on vehicle speeds:	0.14	mile/h	
Percentage of the time the trace is missed by 2mph:	0.24	%	
Total time the trace is missed by 2mph:	1.80	s	
Greatest percentage deviation from the trace in mph:	92.96 %		at time 3.10 s
Absolute deviation from the trace in mile/h:	2.93	at time 3.70 s	

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************************ SIMULATION RESULTS ****************************
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THRU-THE-ROAD HYBRID ELECTRIC CHEVROLET EQUINOX
SERIES BRAKE STRATEGY

Cycle:	1.0 time the HWFET	
Cycle distance:	10.23	miles
Cycle distance:	16.46	km
Vehicle mass:	1954.92	kg

*********** THERMAL INFORMATION ***********

** Engine

Fuel economy:	37.11	mile/gallon
Fuel economy gasoline equivalent :	33.37	mile/gallon
Fuel mass:	0.87	kg

C emissions:	0.0000	g/mile
CO emissions:	0.0000	g/mile
NOx emissions:	0.0000	g/mile
CO2 emissions:	271.57	g/mile

*********** ESS ELECTRICAL INFORMATION ********

ESS Electric-only:	-15.28	Wh/mile
Initial ESS SOC :	70.00	%
Final abs ESS SOC:	74.14	%
Final usable ESS SOC:	74.14	%

*********** TOTAL ELECTRICAL INFORMATION ********

Electric-only:	-15.28	Wh/mile

*********** COMBINED INFORMATION **********

PSAT onboard lumped ess eff gasoline eq.:	47.93	mile/gallon
PSAT onboard chg dis ess eff gasoline eq.:	47.88	mile/gallon
PSAT onboard thermal/regen partioned lumped ess eff gasoline eq.:	47.93	mile/gallon
Student competition onboard gasoline eq.:	52.07	mile/gallon
PSAT offboard gasoline eq.:	44.75	mile/gallon
Student competition offboard gasoline eq.:	44.46	mile/gallon
Energy loss accmech :	211.08	Wh
Energy loss cpl :	-17.80	Wh
Energy loss ess :	28.71	Wh
Energy loss mc :	154.01	Wh
Energy loss gb :	42.87	Wh
Energy loss fd2 :	27.68	Wh
Energy loss fd :	83.00	Wh
Energy loss wh2 :	210.09	Wh
Energy loss wh :	210.09	Wh
Energy loss pc_accelec :	0.00	Wh
Energy loss accelec :	0.00	Wh
Energy loss eng :	6782.75	Wh
Energy loss aero:	1626.08	Wh
Energy loss drag:	207.78	Wh
Energy loss grade:	0.00	Wh
Total Energy loss :	9566.34	Wh
Total Energy used @ vehicle:	714.39	Wh
Engine Fuel Energy Use:	10283.04	Wh
Fuel Cell Fuel Energy Use:	0.00	Wh
Total Fuel Energy Use:	10283.04	Wh
ESS Net Energy Use (propel + regen):	-904.74	Wh
ESS2 Net Energy Use (propel + regen):	0.00	Wh
Total Net Energy Use (propel + regen):	-904.74	Wh
Combined Power Source Energy Use:	9378.30	Wh
powertrain Energy Use In @ wheel (w/regen):	10323.99	Wh
Powertrain Energy Use Out @ wheel (w/regen):	4155.67	Wh
percentage Braking Energy Recuperated :	88.92	%

***** COMPONENT AVERAGE EFFICIENCIES *******

Engine efficiency :	34.04	%
Motor efficiency :	85.83	%
Motor efficiency during acceleration:	83.04	%
Motor efficiency during deceleration:	94.42	%
Transmission efficiency :	98.72	%

************ OTHER INFORMATION ************

** Engine

Weight Specific Fuel Consumption Gas Eq. (WSFC):	1.53	gallon/100mile/ton
Mass of fuel needed to travel 320 miles:	27.25	kg

*************** STATISTICS ****************

Absolute average difference on vehicle speeds:	0.15	mile/h
Percentage of the time the trace is missed by 2mph:	0.52	%
Total time the trace is missed by 2mph:	4.00	s
Greatest percentage deviation from the trace in mph:	92.96 %	at time 3.10 s
Absolute deviation from the trace in mile/h:	6.41	at time 7.00 s

Vita

John was born in Murfreesboro, TN in 1980. He attended Oakland High School graduating with honors. In the fall of 1999, John began studying at the University of Tennessee with a major in mechanical engineering. He was awarded the Boeing scholarship for undergraduate research during his senior year for work on a lean NOx trap for lean burn natural gas engines. He graduated in the fall of 2003 with a B.S. in mechanical engineering, and started graduate school the following spring. He was awarded a GATE Fellowship, which allowed him to focus his studies on the development of hybrid electric vehicles. As a fellow, john worked to manage senior automotive design teams and to update the automotive laboratory.